Technology Collaboration Programme on
Heat Pumping Technologies (HPT TCP)

Annex 39

A Common Method for Testing and Rating of Residential HP and AC Annual/Seasonal Performance

Final Report

Operating Agent: Sweden

2016

Report no. HPT-AN39-1

Published by

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Production

IEA Heat Pump Centre, Borås, Sweden

ISBN 978-91-88349-66-8
Report No. HPT-AN39-1

Preface
This project was carried out within the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) which is an Implementing agreement 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 over 40 Implementing Agreements.

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 Heat Pumping Technologies Programme. 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.

The IEA Heat Pump Centre
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 advance and disseminate knowledge about heat pumps, and promote their use wherever appropriate. Activities of the HPC include the production of a quarterly newsletter and the 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.

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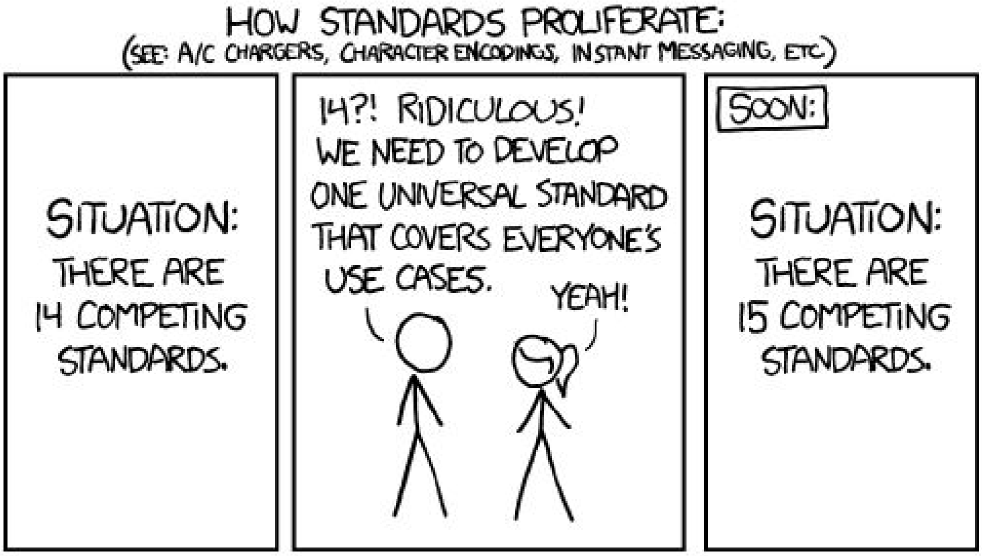
IEA – HPT TCP – Annex 39

A common method for testing and rating residential HP and AC annual/seasonal performance

Final report



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EXECUTIVE SUMMARY

It is important to have reliable information on both the heat pump itself, and how it is influenced by the surrounding system and the climatic conditions under which it operates. This annex focuses on lab methods and related standards, in order to improve them, harmonize and create a better understanding of differences between these. As Figure 1 shows, there could be large deviations between lab tested performance and real performance.

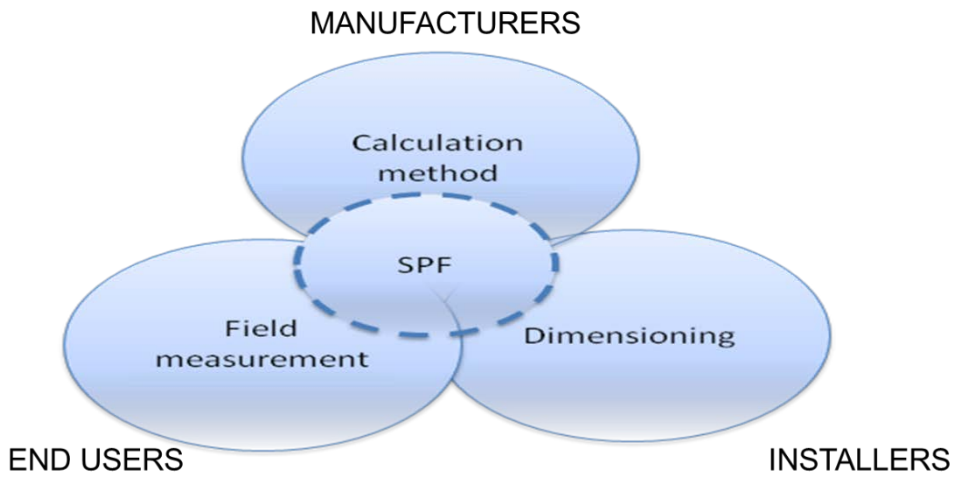

Figure 1. SPF can be obtained in many different ways.

The matrix in Table 3 below is a summary of the most important standards studied in the project. It is divided into different categories trying to sort out the content of the different standards.

Strengths and weaknesses with current methods have been analysed and SWOT analysis of existing standards have been done. It was shown for example that the standard EN 14511 covers not only capacity measurement but also safety in operation and different temperature levels on sink side. EN 14511 is broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) and different funding programmes in Europe. The Standard is not covering capacity controlled heat pumps and the Nominal capacity of capacity controlled HPs is not clearly defined. In EN 14511 circulation pumps are included in the testing procedure only a small amount is integrated in the calculation. After the closing of this annex, updates to the EN14511 standard have occurred.

	HELPFUL To achieving the objectives	**HARMFUL** To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ The standard covers not only capacity measurement but also safety in operation ▪ Different temperature levels on sink side provided	WEAKNESSES ▪ Capacity controlled heat pumps are not covered ▪ Nominal capacity of capacity controlled HPs is not clearly defined ▪ Circulation pumps are included in the testing procedure
EXTERNAL FACTORS	▪ Broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) ▪ Broadly accepted for different funding programs in Europe ▪ Might change to an ISO standard OPPORTUNITIES	▪ Large effort to modify and adapt because referenced within a variety of other standards THREATS

Comparisons on how different calculation methods predict the seasonal performance have been performed in the Annex, showing that the calculation almost always underestimates the real performance of the heat pumps, but that they are very close to real performance.

A comparison was made for one field monitored site, where monitored SPF was used as a benchmark. As can be seen from Figure 18 below, all calculation methods have underestimated the SPF compared to the monitored value (messung).

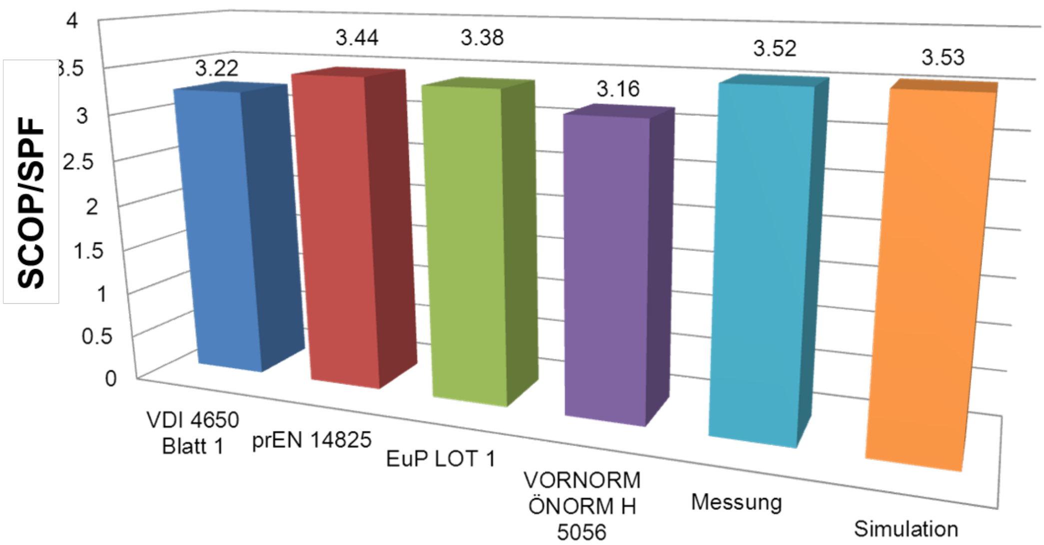

Figure 2. Comparison of different calculation methods with one field monitoring site.

The fact that there are numerous methods for calculating SPF, taking into consideration different national geographic conditions and other special conditions, there was a quite clear view that calculation methods for different climates may need to be local, but considering the test points for lab test standards (Table 8), there is not that many points that differ. It would therefore be of interest to make a thorough evaluation of the consequences of harmonizing the test point parameters for lab testing.

Table 1. Comparison of main standards in three geographic regions of the world.

	APF (Year-round performance factor)	EN14825 SPF	ANSI/AHRI210/240 SEER
Building load	Load=0 : Outdoor tem. at 17 ℃ Load=82% of rated capacity at outdoor temp. of 0 ℃ in heating	Load=0 : Outdoor tem. at 16 ℃ Load=100%: Outdoor tem. at -10℃	Load=0 : Outdoor tem. at 18.3 ℃ Load=100%: Outdoor tem. at -5℃
Estimated load appearance hours	Heating period: Oct 28 – Apr 14 Heating hours: 6:00 – 24:00 Total heating hour: 3042 hrs	Heating hours of standard region is 4910 hours.	750 ~ 2,750 hours depends on regions.
How to calculate device performance	Performance level of products in every outdoor temperature degree can be calculated by measuring rated and intermediate capacities at 7 ℃ outdoor temperature and maximum capacity at 2 ℃.	Can estimate device performance curve by measuring 4 points between 88 % and 15 % on load curve in relation to outdoor temperature	1.Max capacity line; connect 2 points, measured at 1.7 ℃ & -8.3 ℃ 2.Min. capacity line; connect 2 points, measured at 8.3 ℃ & 16.7 ℃ 3. Intermediate capacity line: Measuring at 1.7 ℃ then make a line.
Measuring points	Total: 3 points; 2 from rated and intermediate capacities at 7 ℃, 1 from max. capacity at 2 ℃.	3 -5 points in 3 regions (basically measured at full rated capacity or 75%, 50% and 25 % of rated capacity.	5 points; 2 at max. compressor rotating speed, 1 at intermediate capacity and 2 at min. capacity
Conditions of Measuring points	Rating and intermediate capacities at 7℃, and max. capacity at 2 ℃	-7.0℃:88% 2℃:54% 7℃:35% 12℃:15%	-8.3℃:Max 1.7℃:Mid. 8.3℃:Max.Min. 16.7℃:Min.
	Outdoor temp. is fixed at rated condition of 7℃ & 2℃	4 measuring points. Part loads are measured at different temps.	5 measuring points at 4 different temps. Part loads are measured at different temps.
Remarks about capacity range	Low capacities are not measured due to lacking fairness considering large measuring errors in low capacities.	Measuring error is large at 15% of rated capacity.	Measuring error is large at 25% of rated capacity.
Frost	Considered in every load between max. and intermediate.		Considered only in performance curve of max. load not in other loads.
How to measure a intermittent operation for majority	Degradation Coefficient Cd is not used. As the lower limit of variable width of heating capacity is rarely below the intermediate operation in Japan, they do not calculate by using Degradation Coefficient Cd.	Degradation Coefficient Cd=0.25 is used. Same as the ARI210/240	Degradation Coefficient Cd=0.25
Stand-by power consumption	Not considered	Suggested that stand-by power consumption should be considered	

By looking into the development of products, the complexity of different building traditions and climatic conditions, we have developed a set of requirements that a completely new test/calculation method should be able to handle. Some of the most important are listed below, but all are presented in the report:

It should be possible to decide the energy demand of the house in the model, either by given reference loads, or by choosing a specific energy demand of the house. This should be separated into space heating and domestic hot water. When the model itself calculates the losses of the house it can be misleading and not sufficient for the actual house. This can be one boundary requirement of the project.

To take into account for the climate at the installation, local climate data, for example Meteonorm climatic data could be a part of the model.

The dynamics of the house/building can be a part of the model. The perceived temperature of the house is not fully consistent with the actual outdoor temperature. At colder temperature dips of for example -15°C, the house will not experience the real outdoor temperature, but experiences a temperature of e.g. -12°C instead (due to internal heat gains). Even the irradiance of the sun differs between the seasons (and different spots). The energy demand of the house is affected from those variances over the year, why it might be an idea to calculate the SPF over monthly periods. For simulations, also the use of a fictive outdoor temperature would be an alternative. The climate data can be adjusted (flattened out) depending on a number of inputs, but a temperature dip is still needed in order to make a proper effect dimensioning (this is dimensioning the entire system such as deep wells etc.).

The model should contain a radiator heat curve where required supply temperature is calculated, an example of this can be found in the thesis of Fredrik Karlsson [6]. At a colder outdoor temperature, the supply temperature should peak; this makes the test scheme tables in EN 14511 deficient. Also other heat distribution systems, such as underfloor heating, heating ventilation air and mixed systems should be included in the model.

Part load performance of the heat pump must be properly taken into account.

Back up heaters is sometimes necessary to complete the energy demand of the house. Back up heaters should be included in the calculation model. Supplementary heating should be possible to choose between different sources of supplementary heat, e.g. electricity, solar or biomass heating.

The possibility to include the production of domestic hot water to the SPF calculations would be an advantage. It should also be described how this shall be measured in tests alternatively, how the amount of produced domestic hot water shall be estimated. Today there are two main ways how to do the measurements, including the losses or not (one can measure the amount of energy that is obtained by tappings or the amount of tap water the heat pump is producing).

Accumulators should be possible to include in the model.

A model must contain clear system boundaries for what is to be included in the calculations and how measurements are performed.

An outcome of the results should be to see that a properly sized heat pump is the best alternative to install. An oversized heat pump will result in on/off cycling losses etc.

The model must be transparent so it is possible to follow and understand the calculations. The studied models all contain parts that are more or less transparent. For example how the estimation of the number of equivalent heating hours is performed is not shown in any method.

For the calculation, either BIN methods or hour by hour calculations should be possible to be used. The existing calculation models based on heat pump performance testing according to standards are all using BIN models. Therefore, to keep a clear connection to existing test standards, it is the easiest to base a new model on BIN models.
The drawback with this approach might be that dynamic effects, especially in cases with large or well stratified accumulators are not treated in a way that the full potential of these units are revealed. Likewise, solar irradiation gains might not be treated properly.

To better compare heat pumps' benefits with other heating technologies, but also to better understand performance of heat pump, a number of other measures could be used to understand:

a. The improvement potential of heat pumps and heat pump systems
b. The competitiveness of heat pumps in environmental performance compared to other competing technologies

Figure 3 below shows the different boundaries for characteristic factors for benchmarking the systems according to primary energy, final energy, usable energy, SPF, PEF and PER. The needed parameters for calculating PEF, AE and0020PER are described.

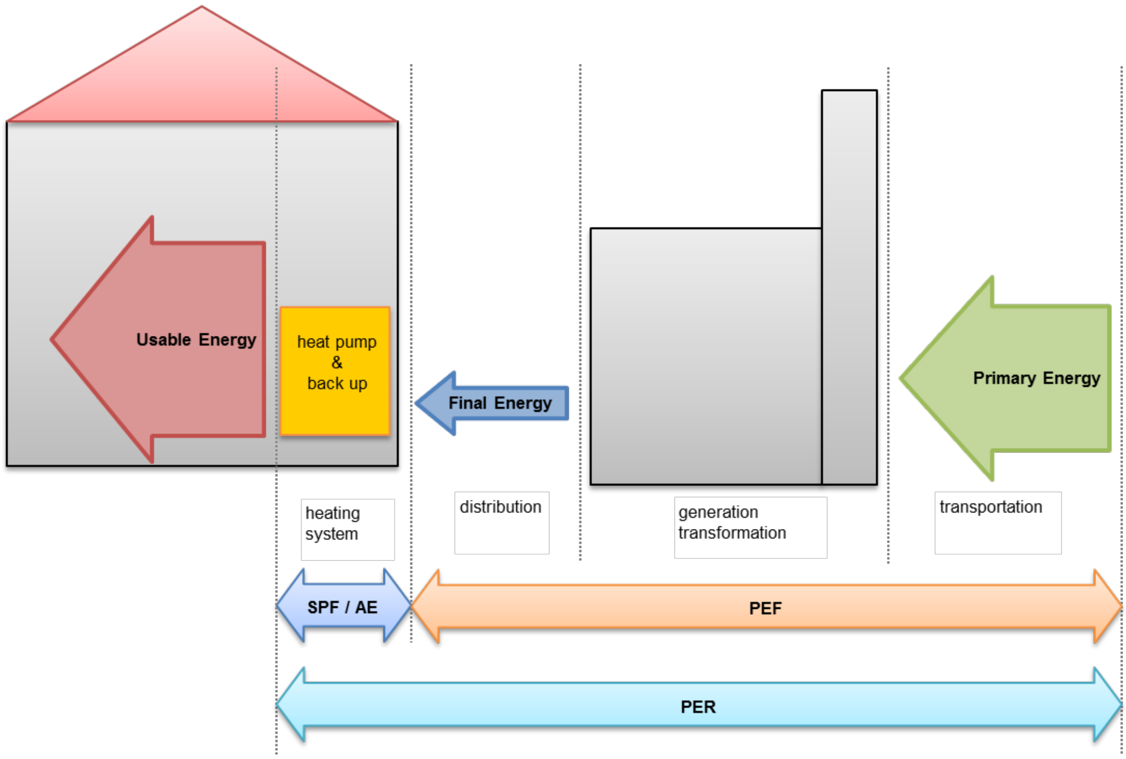

Figure 3. Boundaries for characteristic factors

Conclusions from the Annex work

This annex give proposals for harmonizing test standards, but also extends to give suggestions for building test chambers in an similar way, and propose alternative measures to describe the technical, environmental and financial performance of heat pumps. Much work was carried out in the separate national teams, and the results were presented in workshops. The conclusions from the results are summarized in bullets below, but in order to gain more insight, it is recommended that the conclusions of the national reports are read as well.

- The difference in test points in different regions doesn't differ a lot, why there is the possibility to harmonise many test points. By harmonising the test points, the road is open to come closer with the calculations and certifications that are based on these test points.
- Harmonisation should be made to test points, so that a similar set of test points are tested in the test labs. There must be room for local (national, regional) variations, especially regarding climatic conditions and building demand profiles. Therefore a matrix of test conditions could include the necessary test points, and voluntary test points that should need to be tested for certain markets (e.g. in cold climates, one -15°C point should be included).

- Harmonisation of test standards should happen on ISO level, since this is the global forum for standardisation. Regional/national standards should align with the ISO standards when they are published.
- Timing between revisions of standards is a threshold to harmonisation. Ideally, an agreement should be made between standardisation organisations to make revisions e.g. every five years with a limited revision time, with possibility to harmonise standards at every revision.
- We have reached a conclusion that harmonization of the standards in respective countries is difficult. Even so, we believe that we will be able to create annual performance evaluation standards that seem to be uniform as far as possible.
- Even though this annex have found many possibilities to harmonize standards, we have concluded that a number of new calculation and simulation methods have been developed during this project, which is moving in the opposite direction of the thoughts of this annex.
- As simulation becomes more and more accepted to define building integrated heating performance, there should be very transparent models for both buildings, heating systems and with regards to climatic data. Very clear operating ranges for different relations should be defined etc. There is otherwise the possibility that the final performance numbers are compromised by uncertainties in simulations models.
- To promote heat pump simulation, one IEA HPT annex could be performed, developing a library of annotated and accepted heat pump and building models.
- The IEA HPT could from this annex develop a set of calculation templates for evaluating other performance metrics but SPF, both for installers and for end users. These templates should be Final energy use, Primary energy consumption, CO2 emissions reduction and Cost performance. This makes it much more clear to end consumers to understand the financial and environmental consequences when installing a heat pump

ABSTRACT

In this Annex, the following was aimed to be developed:

1) Common calculation methods for SPF using a generalised and transparent approach, ***based on repeatability and reliable test data from laboratory measurements.***

2) Establish comprehensive test methods based on further development of existing test standards. The test standards should include test conditions needed for the future SPF calculations.

3) A method to evaluate additional heat pump performance, e.g. Carbon Footprint, Primary Energy Saving or Energy Savings.

A matrix of existing standards, test methods and monitoring protocols was assembled, and similarities and differences between these were studied. A swot analysis was performed and proposals for how a harmonisation of test points for lab testing could happen were developed. By this harmonisation, manufacturers could test the performance of heat pumps in any accredited lab and then apply for all certificates that require tested performance, globally.

In order to better understand the accuracy of calculation methods to predict real performance, a comparison of existing calculation methods and results from field measurements were made. All calculation methods have underestimated the SPF compared to the monitored value.

This Annex has also contributed to the development of a whole new standard in the US the IHP (or multifunction HP) test standard, ANSI/ASHRAE 206-2013.

Different methods to calculate and compare heat pumps with other heating technologies were also suggested in this work, mainly based on an LCA perspective, nut also models that calculates the CO2 emissions reduction, or the Primary Energy (PE) use have been proposed.

The Japanese team in the Annex has proposed that test chambers should be built and instrumented in the same manner in order to minimise uncertainties between test sites.

1 INTRODUCTION

1.1 Background

To achieve an excellent working heat pump system, the right type of heat pump must be chosen and installed with a matching heat distribution system. For this reason, it is important to have reliable information on both the heat pump itself, and how it is influenced by the surrounding system and the climatic conditions under which it operates. The most common way to describe the performance of a heat pump is through the so called Seasonal Performance factor, SPF. SPF can be obtained through an number of ways, e.g. by calculations of the thermodynamic cycle, trough lab testing under controlled conditions, or trough field monitoring of installed units, see Figure 4-

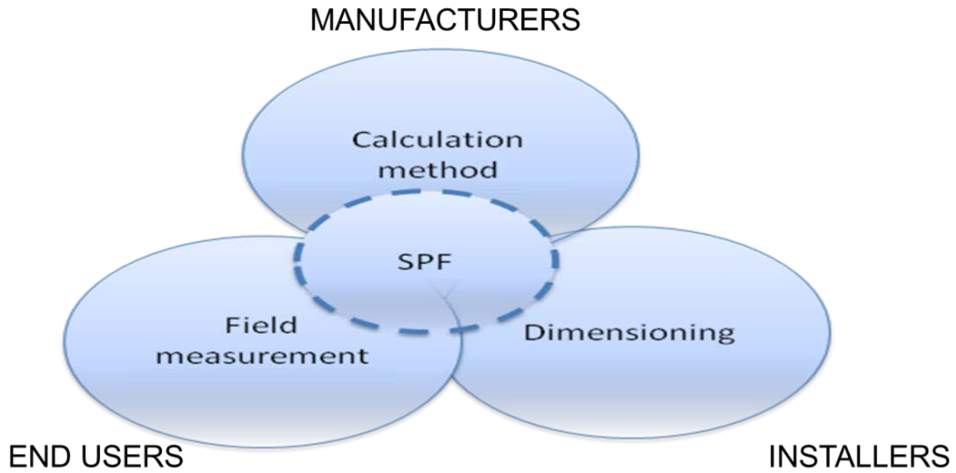

Figure 4. SPF can be obtained in many different ways.

Different groups also need different information, to serve their needs. Examples are:

- *Policy* (EU): **RES – How do heat pumps live up to set goals**

- *Consumer*:
  - **Reliable information** (What you buy is what you get)
    - Cost performance, energy performance
    - Better comparison to other heating products

- *Industry*: **Consumer confidence**
  - Industry wants customers to buy a heat pump also when the old HP is replaced. They also want to benchmark their products to their competitors.

Heat pump quality and performance is increasing. One reason for this is the work with standardisation, including calculation as well as performance testing procedures. The work with standardisation has improved continuously since the late 1970's. Despite this, the performance of heat pumps (Coefficient of Performance, COP) has up to now often been characterised at single operation conditions and at full or rated capacity. These conditions do not always reflect the real performance of the heat pump in practical operation in heating systems. Heat pumps mainly operate intermittently or at reduced capacity, through capacity control or on-off cycling, in climatic conditions that differ from the standard rating conditions. It is therefore important to characterize the Seasonal Performance Factor (SPF) based on a number of operating conditions. The influence of part load or variable capacity on SPF is not fully covered by existing methods for calculation of SPF.

At the time of start for this Annex, the European Seasonal Energy Efficiency Ratio (ESEER) existed and was calculated from a few operating points and is the same for the whole of Europe. Such a method would also be feasible for heating purposes. A method showing the benefits of capacity controlled units was needed to promote more energy efficient heat pump systems/units. Since 2012, SCOP (Seasonal Coefficient of Performance) is defined in a European standard.

A common SPF method would be important for fair comparison between different types of heat pump systems as well as fair comparison with other competing technologies using e.g. fossil fuels. A common SPF (or SCOP) method can later be incorporated in different labelling, rating and certification schemes and is already so in Europe. However, as can be seen in Figure 5, different standards apply globally.

Figure 5. Standards for heat pump performance used in different parts of the world.

There is thus a need for improved transparent and harmonised methods for calculation of heat pump system SPF ***based on repeatability and reliable test data from laboratory measurements***, see Figure 6.

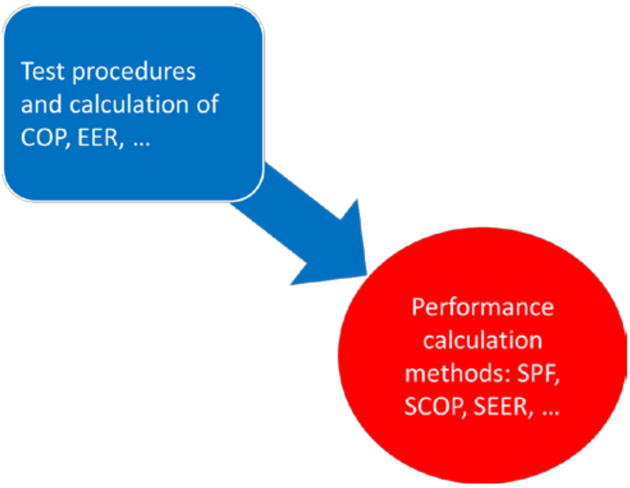

Figure 6. All performance calculation methods rely on lab or field measured performance.

Building types covered by calculation methods analysed in this annex
The energy demand for heating, cooling and use of domestic hot water is influenced by a large number of factors such as the standard of the buildings, the outdoor climate and the behaviour of the end users. In addition to this different heat sources annual variation in enthalpy is strongly influenced by the geographical location. Finally the type of distribution system in the building, e.g. temperature level and variation in supply temperatures needed for a certain outdoor temperature varies and will have a large impact on the system efficiency. Due to the complexity with different building codes, distribution system in buildings, outdoor climate and the variation in behaviour from the end- users. The evaluation and further development of existing test methods and calculation methods should therefore be limited to cover single and multifamily buildings only.

Heat Pump types included in the Annex
The Annex covers the heat pump systems listen in Table 2.

Table 2. Heat pump types included in the Annex.

Type of heat pump	Operating mode				
	Heating only	Heat pump water heating	Heating + Domestic hot water	Heating + Cooling	Heating + Cooling + Domestic hot water
Air/air	x			x	
Air/water	x	x	x	x	x
Brine/Air	x	x	x	x	x
Brine/water	x	x	x	x	x
Water/water	x	x	x	x	x
DX/DX	x	x	x	x	x
DX/water	x	x	x	x	x

Bivalent or hybrid systems, i.e. systems where the refrigerant cycle is combined with an additional heating option in the heat pump unit have been included in the analyses in this annex.

Existing test methods and SPF methods
There is a lack of a common calculation method for SPF covering heating, cooling and domestic hot water, DHW, production. In addition it is a need to have a common calculation method for SPF covering also heat pumps at part load operation/capacity control.
A method showing the benefits of capacity controlled units is needed to promote more energy efficient heat pump systems/units. This standard should be developed according to the procedure illustrated in Figure 7. The development of standards differs between Asia, North America and Europe. The International Standards Organization (ISO) is working on a global standard for SPF calculation (called APF, Annual Performance Factor) in the group TC86, SC6, WG1. Also the TC163 and TC205, also work with SPF (ISO/NP 13612 Heating and cooling systems in buildings -- Method for calculation of the system performance and system design -- Heat pump systems. Within CEN TC 113, a European standard, EN 14825, dealing with how to calculate SEER and SCOP and to test capacity controlled heat pumps has been developed. In Europe, methods for calculating SPF have for a long time mainly based on lab measured values of COP from the standard EN14511. The RES directive (Directive 2009/28/EC) base the amount of renewable energy on an SPF calculation, mentioned in the Annex VII to the directive, where the SPF calculation method was published in march 1, 2013. Another process in Europe is the Energy using Products (EuP) directive (Directive 2005/32/EC) that puts thresholds on efficiency on a number of products, including heat pumps. Heat pumps have been affected by Ecodesign and Energy Labelling Directive and different methods for declaring seasonal performance have been developed within ENER Lots 1, 2 and 10 and Ecodesign and Energy Labelling regulations have now been published for different types of products. In Japan, there exists three standards for calculating Annual Performance Factor (APF), namely JRA4046, JRA4048 and JRA 4050. These cover different capacity ranges and application areas. The latest update was for the JRA4046 (most relevant) in 2004, which is now becoming old.
In North America, standards, as AHRI 210-240-2008, ANSI/AHRI 390-2003, ANSI/AHRI 870-2005, ARI 320-98, ARI 325-98, and ARI 330-98 have been relevant.

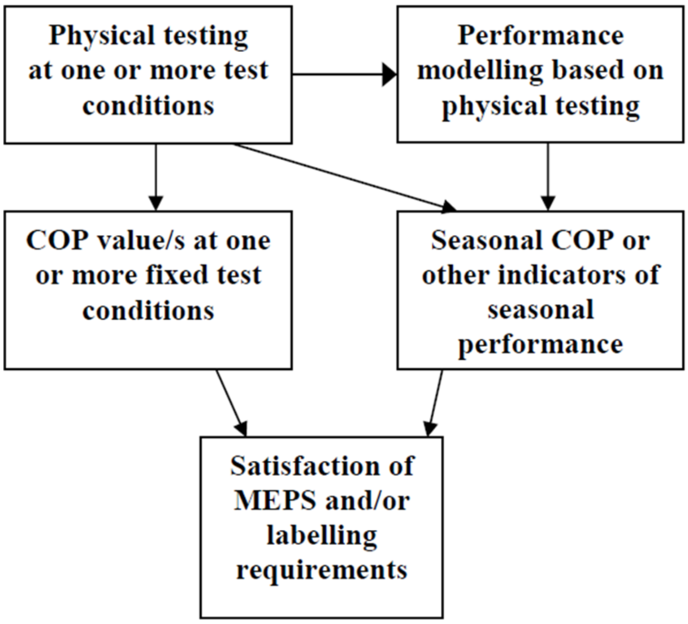

Figure 7. Procedure for testing, calculating and rating of heat pumps.

European RES (Renewable Energy Sources) Directive
Heat Pumps using aerothermal, geothermal or hydrothermal energy as a source are defined as renewable in the European RES Directive if the SPF is above a specific value (SPF > 1.15/η, and η is presently estimated as 0.455, giving that SPF > 2.5 makes the heat pump defined as renewable. The amount of renewable energy is calculated as: $\mathbf{E_{RES} = Q_{usable} * (1 - 1/SPF)}$**.**

Need from Industry
Presently a large number of national standards for both testing and calculation of SPF exist around the world. There is a request from manufacturers to have globally common testing methods and common SPF methods, since this would simplify for them to export heat pumps to different countries. The question has been highlighted in the European countries after the RES Directive was approved. Also in Japan the existing standards need to be updated, and a common methodology is desired.

In addition to this, end users need to have reliable information in the selection procedure both between different heat pump systems, how it is influenced of the system as well as in the process to compare heat pumps with other technologies. Finally it is important in the communication to stakeholders and policy makers to communicate transparent, reliable and comparable information about the energy performance of heat pumps in comparison with other technologies, including fossil-fuel systems.

Challenges
It is not an easy task to define a common calculation method for SPF. The building standard and the type of distribution system in the building vary between regions and countries. The supply temperature influence COP and the building standard influence the energy demand. In addition the outdoor climate and the end users behaviour influence the heating demand, cooling demand and demand for domestic hot water. Finally the temperature/ enthalpy of the heat source is dependent on the outdoor conditions. All these variations that influence the SPF, Seasonal Performance Factor, is a challenge. A real value of the SPF has to be calculated for each specific installation, from field measured data. However, a simplified general approach is necessary for comparison and improved understanding of the heat pump system. The simplest way is to make all calculations for one specific building in one specific climate. Another approach could be to define a limited number of regions with typical climate and buildings.

In conclusion, there is a need for an improved transparent and harmonised method for calculation of heat pump system SPF based on repeatability and reliable test data from laboratory measurements. The need for a harmonised SPF method from a European point of view is driven by the RES Directive and a request for harmonization from the manufacturers. In addition a common SPF method could be used in different labelling, rating and certification schemes. Even if the project is driven from an European perspective, it would very beneficial to reflect the experience from all regions covered by the IEA HPP.

The outcome from the Annex was described as a proposal for a common transparent SPF calculation method for domestic heat pumps including heating, cooling and domestic hot water production.

The idea is to make pre-normative research, which later can be incorporated in the standardisation (ISO and CEN) in the same way as IEA HPP Annex 28 earlier was. The outcome from Annex 28 was successful and the developed calculation method has already been integrated in the EN 15316-4-2 standard from the European standardisation organisation CEN in the framework of the European Energy Performance of Buildings Directive (EPBD).

The main sector targeted for the SPF calculation method is international standardisation organisations (e.g. ASHRAE, CEN, JRAIA, and ISO) heat pump associations and manufacturers. The manufacturers need a common transparent calculation method for SPF for further deployment of the technology.
Reliable data is important for design of the heat pump system for each unique installation. To have reliable tools for design is of importance for the installers of the systems, but also for policy level, such as IEA, DOE, EU and NEDO. The quality of the system design is of importance for the reputation for the technology and the market growth. This project can deliver further knowledge to the manufacturers so they later can improve their existing design tools.

1.2 Objectives and scope of the project

The calculation method for SPF should cover the following heat pump systems and operating modes:

- Single- and multi-family buildings.
- heating, cooling and DHW
- capacity control

The objective was to
1) Establish common calculation methods for SPF using a generalised and transparent approach. The focus is on a fair comparison between different heat pump types, but also for comparison between different competing technologies, such as pellet boilers, gas boilers, etc.

2) Establish comprehensive test methods based on further development of existing test standards will be evaluated. The test standards should include test conditions needed for the future SPF calculations. The annex therefore mainly cover lab methods, Figure 8.

3) A method to evaluate additional heat pump performance, e.g. Carbon Footprint, Primary Energy Saving or Energy Savings.

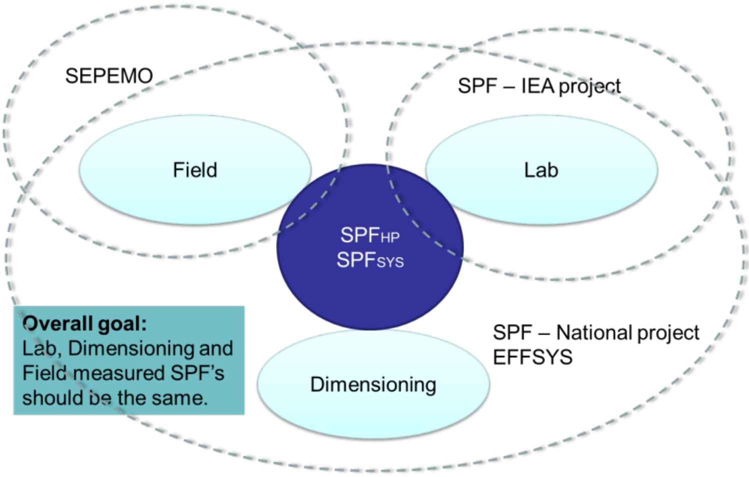

Figure 8. This Annex related originally to methods connected to lab testing.

1.3 Project participants

The following persons participated in at least one meeting with contributions to the Annex:

SP	Roger Nordman	SE
AIT	Ivan Malenkovic	AT
AIT	Christian Köfinger	AT
AIT	Heinrich HUBER	AT
FHNW	Andreas Genkinger	CH
FHNW	Thomas Afjei	CH
Fraunhofer ISE	Marek Miara	DE
Fraunhofer ISE	Thore OLTERSDORF	DE
VTT	Riikka Holopainen	FI
VTT	Ari Laitinen	FI
Aalto University	Lari Eskola	FI
Aalto University	Juha Jokisalo	FI
Aalto University	Kai Siren	FI
SULPU	Jussi Hirvonen	FI
EDF	Christine Arzano-Daurelle	FR
CRIEPI	Choyu WATANABE	JP
CRIEPI	Hashimoto	JP
HPTCJ	Makoto Tono	JP
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HPTCJ	Ms. Terao,	JP
Mie University	Hirota	JP
Waseda University	Kiyoshi Saito	JP
KIER	EJ Lee	KR
AgentshapNL	Onno KLEEFKENS	NL
Isso	Jaap Hoogeling	NL
AHRI	Karim Amrane	US
ASHRAE	Wayne Reedy	US
DOE	Antonio Bouza	US
ORNL	William Craddick	US
ORNL	Jerry Groff	US
ORNL	Van Baxter	US

1.4 Annex execution

The Annex was executed through a number of workshops, were the researchers presented their outcomes. The annex was originally organised in the following tasks:

Task 1 Survey and evaluation of existing testing methods and calculation methods for SPF

Each participating country completed a spreadsheet to provide
1. Strengths (advantages) and
2. Weaknesses of current test and calculation methods for a number of common systems and product types.

They also made an analysis of what's missing in present methods. The second phase was to provide detail on measurements, test conditions, rating methods, etc. for these current methods and to suggest ways to correct deficiencies of these.

Survey and evaluation of existing calculation methods for SPF including:

- Information collecting, analysis and description of current methods
- Evaulation of the need for improvements of existing standards to include capacity control, new refrigerants, etc.
- Arrive in recommendations for use at different locations, or suggestions for a global harmonised standard
- Better understanding of how others do their calculations, by understanding the definition of performance metrics.

The surveys were reported in workshops, and a compilation is made in this report.
The critique was made as a SWOT analysis.

Survey and evaluation of existing test methods for heat pumps

- Information collecting and analysis and description
- Need for improvements of existing standards to include capacity control, new refrigerants,
- Group critique for and against different methods in different aspects.
- Arrive in recommendations for use at different locations, or suggestions for a global harmonised standard
- Questionnaire about capabilities for testing in different countries

The critique could be made as a SWOT analysis.
The outcome is
1) A matrix of technical data from existing standards. Evaluation of recommendations for improvements will be presented.
2) A proposal to a specification for a new calculation method

The surveys on existing methods will answer how well they live up to the requirements specified in Task 1. This can then serve as a decision on which model to develop further, or to develop a completely new model.

Task 2 Matrix definition of needs for testing and calculation methods
The definition of a matrix for evaluation of existing testing and calculation methods should contain a specification of the needs for testing and calculation methods, such as e.g.

- *Measurement requirements* (what has to be measured) in existing standards for measurement of combined operation (heating, cooling, DHW) that can be used as input for an SPF calculation.
- *Measured test conditions needed* from laboratory measurements for calculation of SPF in different regions.
- *Measurement requirements* (what is measured) in exiting standards for measurement of combined operation (heating, cooling, DHW) that can be used as input for an SPF calculation.
- *Measured test conditions* in exiting standards.

Examples are:

- Type of building
- Type of distribution system, temperature levels.
- Heating, cooling and DHW demand
- Outdoor conditions
- Different heat sources
- Type of model (bin…)

This matrix could be seen as the specification of requirements for developing a testing method that fulfils the annex goals, or as a roadmap for harmonisation of national standards.

Task 3 New calculation methods for SPF/ commonly accepted definitions on how SPF is calculated
The target of task 3 was to improve existing and/or develop a common transparent calculation method for SPF on a system level based on technical data from laboratory measurements. Important observations from this part are the need for number of regions, number of building types, number of climate zones etcetera. This may be different for different countries, system types, etc. and that likelihood should be recognized.
Better understanding of how others do their calculations, by understanding the definition of performance metrics was an important component in this task.

During the execution of the Annex, it became apparent that the lack of harmonisations is large, for example in Europe, the number of climatic regions have been set to three for the whole of Europe in the EPBD, while France have defined five climatic regions and Sweden have four alone. Since climate has so large impact on the heat pump performance both regarding heat demand and heat source temperature, the trend is nowadays to perform heat pump system modelling for anticipated climates and buildings, based on lab tested COP values. The researchers have therefore in this task looked into different ways of modelling heat pump systems.

Task 4 Identify improvements to existing test procedures
Existing test standards were identified, and suggestions for development to cover the needs identified in task 2 are proposed. Further development of test procedures for capacity control is expected, especially for combined heating and DHW production. The main objective of this task has been to suggest improvements in current testing methods by showing how they can be implemented in existing standards.

Task 5 Validation of SPF method
A validation of calculation method against already ongoing or completed field measurements, and improvement of the proposed calculation method if needed was foreseen.
Since no single new method was proposed in this Annex, this task was not completed.

Task 6 Development of an alternative method to evaluate heat pump performance
The objective with this task was to find ways to make valid comparisons with other heating systems, using performance indices such as carbon footprint and/or primary energy use.
In order to evaluate the heat pump performance in alternative ways, methods were discussed and developed to enable the calculation of the energy savings potential and the CO_2 reduction potential, or the primary energy consumption, from heat pumps. It was agreed among the project partners that a wide set of methods showing many aspects of the positive aspects of using heat pumps could be used. Which method is chosen will depend on the needs from the end user.

Task 7 Communication to stakeholders
Task 7 dealt with communication to stakeholders. Most important in this respect was to send the report to relevant stakeholders within standardisation committees, and secondly to inform the research community about the activities.

The results could be used to harmonise standards and propose improvements for increased harmonisation in existing standards.

2 SURVEY AND EVALUATION OF EXISTING TESTING METHODS AND CALCULATION METHODS FOR SPF

2.1 Studied methods for calculation of SPF

The matrix in Table 3 below is a summary of the most important standards studied in the project. It is divided into different categories trying to sort out the content of the different standards. All AHRI standards mentioned above refers to ASHRAE standard 37 for the description the test method and requirements for testing. The purpose of the AHRI standards is to provide test and rating requirements, requirements for operating and the like for different kinds of heat pumps. The standards EN 255-3, prEN 255-3, TS14825 and prEN14825 all refers to the standard EN 14511 for requirements to fulfil the test method. For data input to the calculations of the calculation method EuP ENER Lot 10 and to some extent EuP ENER Lot 1 and EN15316-4-2, one is referred to the test results from standard EN 14511. It should be noted that during the course of this task of the Annex, EN14825 was not approved, but existed as a prEN-standard, that is a not finally set standard. All analysis was done considering this prEN-standard. Nor existed EN16147, which later replaced EN255-3, when this task was performed.

The first category “type of standard” shows whether the standard describes a test method for laboratory tests, for field tests and if it includes a calculation model for the calculation of seasonal performance factor.

The second category “type of heat pump” describes what kind of heat pumps that is included in the standard or test method.
The third category “Operation” describes the type of operation that is treated by the standard. The different types of operation can be heating mode, cooling mode or heating of domestic hot water. The column called “combined operating” refers to the simultaneous production of heating and/or cooling and the heating of domestic hot water. The last column within this category “part load conditions” shows if the standard includes the operation of the heat pump in part load.

The intention of the fourth category “requirements” is to show whether the standard has any requirements of testing to reach accurate test results. Typical requirements could be that steady state has to be reached before the measurements are performed, requirements of maximum deviations from the stated measurements and a largest permissible uncertainty of measurements of the tests. The last column within this category shows whether the standard gives any recommendations of how the measurements shall be performed, such as the placement of sensors.

Table 3. Matrix of standards and their application.

	Type of standard			Type of heat pump			Operation					Requirements				Aspects in capacity calculations					Calcula-tions of	
	Laboratory tests	Field tests	Calculation model for SPF	ASHP	GSHP/WSHP*	AIR/AIR	Heating	Cooling	Domestic hot water	Combined operating	Part load conditions	Steady state	Permissible deviations	Uncertainty of measurements	Test set up/ performance of measurement	Pumps and fans included	Defrost period	Standby losses	On/off cycles capacity regulation	Other	COP/EER	SPF or SCOP/SEER
NT VVS 076				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 115				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 116				x	x	x	x	x				Δ	Δ	x	x						x	
SP 1721						x	x	x				x	x	x	x						x	
ASHRAE standard 37	x			x	x	x	x	x				x	x	x	x		x			x		
AHRI 210/240	x					x	x	x			x	x	x	Δ	x				x		x	x
AHRI 870-2005	x				x		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 390-2003	x					x	x	x			x	Δ	Δ	Δ	Δ		x				x	
AHRI 320-1998	x				x*		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 325-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
AHRI 330-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
EN14511	x			x	x	x	x	x				x	x	x	x	x	x					
EN 255-3	x			x	x				x			x	x	x	x	x	x	x			x	
TS14825	x			x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	
EN14825	x		x	x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	x
EN15316-4-2			x	x	x	x	x		x	x	x	α	α	α	α	α	α	α	x	α		x
EuP Lot 1	x		x	x	x		x				x	x	x	x	x	x	?		x		x	x
EuP Lot 10	x		x			x	x	x		x	x	Δ	Δ	Δ	Δ	x	x		x		x	x

The sign “Δ” means that the standard refers to another standard where the requirements are fulfilled.

The sign “α“ means that the method is a calculation method that does not include requirements from a specified test method.

The fifth category “Aspects in capacity calculations” describes aspects that are taken into account in the capacity calculations. It describes whether liquid pumps and fans are included in the effective power absorbed by the unit. The “Defrost period” column describes whether the defrost periods are taken into account when measuring and calculating the capacity of the heat pump. The “standby losses” column means that standby losses are measured and taken into account when calculating the capacity of the heat pump. The NT VVS 076 and NT VVS 115 both mention that it is necessary to take standby losses into account when calculating the SPF, but there is no method of how to measure the losses. Both the standards for measuring the production of domestic hot water EN 255-3 and prEN 255-3 states methods of how to measure the standby losses, but the way of taking the standby losses into account when calculating the COP differs a lot between the standards. “On/off cycles and capacity regulation” shows whether the standard treats what kind of capacity regulation that is used by the heat pump. The last column “other” shows whether there are other important aspects apart from the earlier mentioned ones, which are taken into account in the capacity calculations. It shows that for some of the methods mentioned in the standard ASHRAE 37 adjustments of the line loss capacity and duct losses are made.

The last category “calculations of” describes the calculated outcome of the standard. The NT VVS standards provide simple equations of how to calculate SPF without a calculation model.

In this Annex, methods to measure annual performance factors (APFs) for heat pump equipment in Japan are specified under the following Japanese Industrial Standards (JIS):

1) JIS C 9612 : Room air conditioners
2) JIS B 8616 : Package air conditioners
3) JIS C 9220 : Residential heat pump water heaters

Japanese air conditioner manufacturers currently use APFs to evaluate the fundamental performance of their products.
Japan focused on air conditioners with rated cooling capacities of 10 kW or less under JIS C 9612 room air conditioners, which are categorized as air/air (heat pump class) and heating and cooling

2.1.1 Other methods including calculation models

Besides the models mentioned above there are several other standards and models that can be used in order to find an appropriate model to calculate a seasonal performance factor. The ones studied in this project are shortly summarized in this chapter.

EN 15316-2-3 Heating systems in buildings – Method for calculation of system energy requirement and system efficiencies – Part 2-3: Space heating distribution systems

This method calculates the system thermal losses and the auxiliary energy demand of water based distribution system for heating circuits (primary and secondary), as well as the recoverable system thermal losses and the recoverable auxiliary energy. The calculations are related to a design effect and design heat load of the accounted zone (EN 12831). Correction factors are provided for a number of different conditions, these conditions can for example be corrections for the size of the building, for systems without outdoor temperature compensation, efficiency and part load. The method can be applied for any time step (hour, day, month or year).

EN 13790:2008, Energy performance of buildings – Calculation of energy use for space heating and cooling (ISO 13790:2008)

This standard provides a calculation method for the assessment of the annual energy use of buildings. Factors that are taken into account are for example the heat transfer by transmission and ventilation of the building when heated or cooled to constant internal temperature, contribution of internal and solar heat gains to the building energy balance and the annual energy use for heating and cooling.
There are two different main methods that are used by the standard, one where the heat balance is calculated during a sufficiently long time (one month or a season) and dynamic effects of the building are taken into account by an empirically determined gain and/or loss utilization factor and one method where the heat balance is calculated over small time steps (typically one hour) and the heat stored in, and released from, the mass of the building is taken into account.

EN 12831 Heating systems in buildings – Method for calculation of the design heat load

This standard is used to calculate the design heat losses of a heated space; the result is then used to determine the design heat load at standard design conditions. The temperature distribution (air and design temperature) is assumed to be uniform. The climatic data that is used for the calculations are the external design temperature and the annual mean external temperature.

Factors taken into account are for example size of the building, type of building, activities inside the building, type of room, interior, building envelope and ventilation.

2.1.2 SPF Modelling

A number of standards/methods for the calculation of seasonal performance factor are investigated. Some of the methods only contain a calculation model while some of them also contain instructions of how to test the heat pumps. The calculation models that are studied in this project are prEN14825:2009 draft Nov 09, EN 15316-4-2:2008, EuP ENER LOT 1 and EuP ENER Lot 10.

EN 15316-4-2:2008

Heating systems in buildings – method for calculation of system energy requirements and system efficiencies – Part 4-2: space heating generation systems, heat pump systems

15316-4-2 is a calculation model for the calculation of system energy requirements and system efficiencies. Input product data for the calculations, like heating capacity and COP are determined according to European or national test standards. The method treats calculations for space heating, production of sanitary hot water and combined operation of space heating and sanitary hot water production in either simultaneous or alternating operation. Presently there is no European standard for testing DHW production and space heating simultaneously; therefore a national standard shall be used instead. As an example in this standard calculations based on testing of a DHW cycle performed according to EN 255-3 during heating operation are done, see Annex D in EN 15316-4-2:2008.

System boundaries

The method takes into account different physical factors that can have impact on the SPF and required energy input. For example type of generator, type of heat pump, variation of heat source and sink temperature, effects of compressor working in part load (on-off, stepwise, variable speed units), and system thermal losses, Figure 9.

Losses due to ON/OFF cycling are considered small and negligible unless part load testing data or national values are available. If part load data is not available the stand-by auxiliary energy is considered enough for the degradation of COP in part load operation.

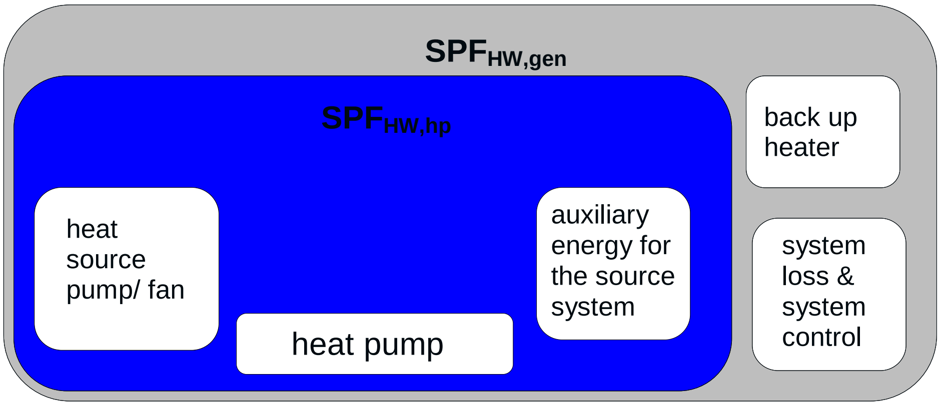


Figure 9. System boundaries in EN 15316-4-2:2008.

Input to the calculations

Two performance calculation methods for the generation subsystem are described corresponding to different applications (simplified or detailed estimation). The differences between the two methods are the required input data; the operating conditions taken into account and the calculation periods.

The simplified method

The considered calculation period is the heating season and the performance data is taken from tabulated values for fixed performance classes of the heat pump. Operating conditions are taken from typology of implementation characteristics, which means that they are not case specific. This method is in particular suitable when limited information of the generation subsystem exists.

The detailed method

This method is a temperature bin method where the specific operating conditions of each individual heat pump can be considered. The bins describe frequency of the outdoor temperature and the calculations are carried out with operating conditions for the heat pump that corresponds to the heat energy requirement of the space at each bin. The operating conditions of the bins are characterized by an operating point in the centre of each bin and in the calculations it is assumed that this point represents the operating conditions of the whole bin. The standard contains one example of climate; it represents the climate of Gelterkinden in Switzerland and span from -11°C-35°C with a resolution of one bin per K. Appendix A in EN 15316-4-2:2008 shows how to calculate bins using meteorological data for the actual spot. There are examples in the standard that uses only four bins, but with lower resolution, see figure 4 in EN 15316-4-2:2008. There are some criteria when choosing the bin resolution. The bins has to be evenly spread out over the operating range, operating points should be chosen at, or close to test points and the number of bins shall reflect the changes in heat source and sink temperatures. COP values and heat capacity can be interpolated from tested values to fit the bins.

The heat energy requirement of the distribution subsystem can be evaluated if the heat load for space heating and domestic hot water is known. The heat load for space heating is calculated based on cumulated heating degree hours which are defined by the difference between the outdoor air temperature and the indoor design temperature at the different bins. Analogously the DHW load depicted as constant daily profile can be cumulated.

Back up heaters can be accounted for, both for space heating and for sanitary hot water production. If no information about electrical back up heaters is given, an efficiency of 95% is used.

Input data for calculation with the bin method according to chapter 5.3.2 requires indoor design temperature, heat energy requirement of the space heating distribution subsystem according to EN 15316-2-3, type and controller setting of the heat emission system heat pump characteristics for heating capacity and COP according to test standards, results for part load operation according to prEN 14825, system configuration like back-up heater calculated according to 15316-4-1 and installed heating buffer storage, power of auxiliary components (pumps etc.). It also requires input data for the DHW-production for example heat energy requirement of the distribution subsystem according to EN 15316-3-2 etc.

Output of the model

Two different seasonal performance factors can be calculated by using this model.

$SPF_{HW,\ gen}$ is the total seasonal performance factor of the generation subsystem. It includes the heat pump in space heating mode and production of sanitary hot water, the backup heater, the space heating distribution system and auxiliary energy.

$SPF_{HW, hp}$ is the seasonal performance factor of the heat pump with regard to the heat produced by the heat pump. It includes the heat pump in space heating mode and production of sanitary hot water, the auxiliary energy input for the source system and the auxiliary energy for the heat pump in standby mode.

EuP ENER LOT 10 (Ecodesign and Energy Labelling Requirements in Europe)

The methods developed and proposed by EuP ENER LOT 10 applies to "residential room conditioning" appliances (air conditioners and ventilation) with cooling power ≤ 12kW. It describes a calculation model for calculating the seasonal energy efficiency for operating in heating or cooling mode. This model was further developed in EN 14825 which later was harmonized with the Ecodesign and Energy Labelling regulations which were the result of the work within EuP ENER Lot 10. Thereby the model described below was replaced by the one of EN14825:2013. However, most of the findings reported below for the prEN-version of the standard are also valid for EN14825:2013.

System limits

The model can be used to calculate the seasonal performance factor for an air/air heat pump. The model does not include any losses from the house. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for. System boundaries defined in the EuP ENER LOT 10 are described schematically in Figure 10.

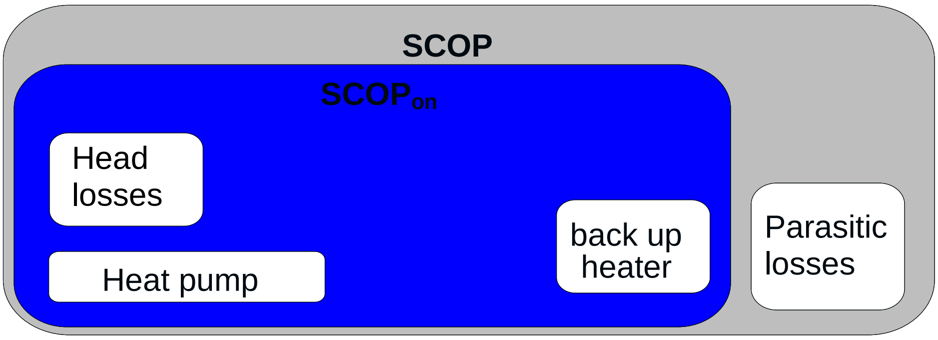


Figure 10. System boundaries in EuP ENER LOT 10.

Input to the calculations

To use the calculation model provided by the excel sheet the load profile of the building, P*designh* has to be selected. There are nine different sizes to choose between from size 3XS to XXL that spans from 1.1 kW to 19.2 kW. The function of the heat pump is set to either "heat only" or "heat and cool" and the type of heat pump is set to "split" or "multi-split".

The model is a bin method with three different climates for the heating season and one for the cooling season. A table declares the number of bin hours occurring at each bin temperature, T_j, for each specific climate. The lowest temperature for each climate respectively is declared the design temperature, T_{design}. The part load ratio (of the building), pl_j, is calculated from Equation 1 below:

$$Pl_j = \frac{(T_j - 16)}{(T_{design} - 16)} \quad \text{(Equation 1)}$$

The reference annual heating demand, Q_{HE} is decided in kWh for each climate as a product of P*designh* and the number of full load heating hours that corresponds to each climate.
Load fractions fracA, fracC and fracW indicate the fraction of the total heating demand (load) occurring in a specific bin at a specific climate. The fractions are given by Equation 2:

$$frac_j = \frac{n_j * pl_j}{\sum_{j=1}^{40} n_j * pl_j} \quad \text{(Equation 2)}$$

Input to the calculations is the COP and capacity of the heat pump at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C(-15°C is only required for the colder climate). The heat pump should be tested at part load to deliver the required heat load of the building at each temperature level. At this point the paper version is not consistent. In one way it says that the capacity of the heat pump at each bin shall complete the energy demand of the building at the part load declared by the product of the annual reference heating demand, Q_{He}, and $frac_j$, but in one way it says that the energy demand is declared by the product of the part load ratio, Pl_j, and P*design.* However the excel sheet uses the first alternative and therefore care should be taken when deciding the operating points (the required effect at each temperature bin) for testing the heat pump. This alternative does not provide any effect balances. Since one house is chosen for the calculations the required effect at each outdoor temperature should be the same among the climates, but this is not the case.

In cases where the heating power supplied by the heat pump is not enough to cover the energy demand of the building in a specific bin, the difference is filled up by a backup heater with a declared capacity of COP=1. Deciding the part load from the product of Q_{He}, and $frac_j$, might result in an underestimated effect demand and therefore underestimate the required backup heating.

Instructions of how the heat pump shall be tested are given in the method for each type of operation respectively; fixed capacity units, staged capacity units and variable speed capacity units.

A degradation factor Cd, which is the efficiency loss per kW of output power when cycling the heat pump, is decided from a specific cycling test.

The energy consumption for the heat pump when operating in thermostat off mode, off mode and crankcase heater mode is decided in tests, but is only required for the calculation of SCOP.

The turndown ratio for heating, which is the lowest steady state over the maximum power and the binlimit, which is the lowest operating temperature of the heat pump, is used as input to both of the SCOP calculations.

<u>Output from the model</u>
This model is used to calculate two different seasonal performance factors:

COP_{ON} is a seasonal performance factor for the heat pump that includes electricity of the backup heater. COP_{ON} is calculated by the total electricity used by the heat pump and the backup heater over the total heat demand of the building. (LhpC_tp*COPC_tp+resC_tp)/LhsysC_tp

SCOP is a seasonal performance factor which unlike COP_{ON}, also includes the electricity consumption of auxiliary energy for the heat pump operating in thermostat off mode, off mode and crankcase heater mode.

The energy of the backup heater is included in all seasonal performance factors that results from the excel-calculation sheet.

The annual electricity consumption split up in supplementary heating, heat pump operation and auxiliary heating is given from the calculations.

The annual carbon emission and label energy class is also result of the calculations.

prEN14825 (2009)

This standard was under development when this task was performed. The prEN-version was later replaced by a version accepted by all European member countries of the CEN TC 113, first in 2012. Thereafter a new version was voted and accepted in 2013, which was later harmonised with the regulations resulting from EuP ENER Lot 10 (Regulation 2012/626 and 2012/206).

The standard aims to cover the laboratory testing and a calculation model for SPF calculations for electric driven heat pumps. The heat pumps are tested at a number of different part load conditions (4-6) designed for heating or cooling the house to a set temperature of 16°C at different outdoor temperatures. Different test conditions are given for each type of heat pump.
This standard serves as an input for the calculation of the system energy efficiency in heating mode of specific heat pump systems in buildings, as stipulated in the standard EN15316-4-2:2008.

System limits
The model can be used to calculate the seasonal performance factor for air/air- ground source- and air source- heat pumps, but also other products. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for at outdoor temperatures below the bivalent point. The system boundary in SPF 4 applies. (Data is treated according to EN14511 where the effect of heat sink and source pumps and ventilation fans is corrected to overcome the pressure differences of the heat pump), Figure 11.

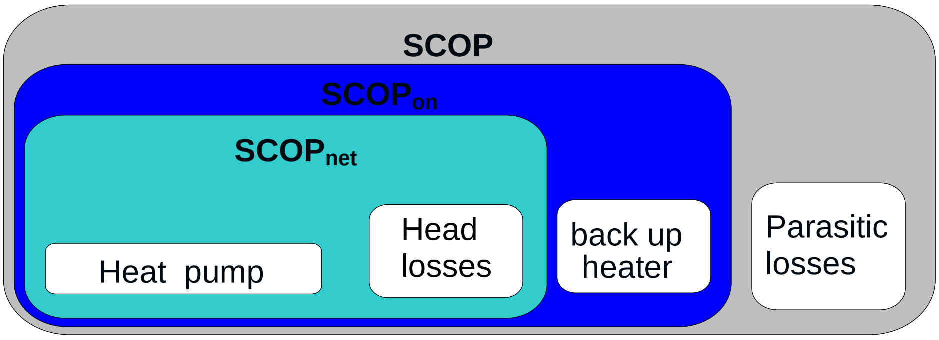


Figure 11. System boundaries according to prEN14825 (2009).

Input to the calculations

The calculation of the seasonal performance (SPF or SEER) is performed using a temperature bin method where each bin represents one degree Celsius and the number of bin hours occurring at the corresponding temperature is given. The cooling season is represented by one climate that span from 17°C-40°C while the heating season is represented by three different climates: one colder, one average and one warmer, that span from -22°C-16°C, see table 29 and 30 in prEN 14825:2009 draft Nov 09. Each climate corresponds to one design temperature and one design heat load of the building.

The heating/cooling demand and the number of bin hours for the different climates are determined as templates, taking different aspects into account; the climate, type of building and building characteristics, set point and set back settings and internal gains. Those aspects also decide the number of hours in which the heat pump works in active mode, thermostat off mode, standby mode, crankcase heater mode or off mode. The electricity consumptions at the different modes are determined from tests. These effects are called the parasitic losses.

Input to the calculations is the COP and capacity of the heat pump tested at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C (-15°C is only required for the colder climate). The heat pump shall be tested in equivalence with standard EN 14511, with the same test methods, test set up, uncertainty of measurements and the way of evaluating data. The heat pump shall be tested at part load to deliver the required heat load of the building at each temperature level. Instructions of how the heat pump shall be tested by means of part load and type of operation; fixed capacity units, staged capacity units and variable speed capacity units, are given in this method.
The required part load for the building at the test points are given by Equation 3:

$$Part\ load\ ratio = \frac{(T_j - 16)}{(T_{design} - 16)} \qquad \text{(Equation 3)}$$

Where T_j is the outdoor (bin) temperature and T_{design} is the lower temperature limit of the selected climate.

If the declared capacities of a unit matching with the required heating/ cooling demand the corresponding COP/EER value is to be used. This may occur with staged capacity or variable speed capacity units. If the declared capacity is higher than the heating/cooling loads, the unit has to cycle on/off. Then a degradation factor (Cd (air/air or Water/air) or Cc (others)) has to be used to calculate the corresponding COP/EER values. Cd and Cc can be determined by testing; else a default value of 0.25 and 0.9 respectively is used.

The bivalent temperature, which is the lowest temperature when the heat pump can deliver 100% of the heat demand of the building, is necessary to use in the excel sheet distributed with the version of the standard. The design heat demand of the building is a consequence of the stated bivalent temperature.
The reference annual heating demand, kWh/a, is given by the product of the full load in heating P_{design} and the equivalent number of heating hours.

The operation limit of the heat pump is set to the lower temperature limit for which the heat pump can operate.

<u>Output of the model</u>
With above input the excel sheet gives two different SCOP: $SCOP_{NET}$ and $SCOP_{ON}$. $SCOP_{NET}$ is the seasonal performance factor for the heat pump, while $SCOP_{ON}$ also includes the electricity and heat delivered to the building from a backup heater.
The paper version of the standard also calculates a seasonal performance factor, SCOP that includes the parasitic losses of the heat pump. The effect from each operational mode is tested according to the standard while the corresponding operational hours for each mode respectively are found in a reference table.

EuP ENER LOT 1 - Boiler testing and calculation method

This model was originally proposed within EuP ENER Lot 1. However, the resulting regulations referred to EN14825:2013 instead of this model in its transitional methods. A version of EN14825 to be harmonized with the resulting regulations (2013/811 and 2013/813) has been developed and will be published shortly.

This model is used to calculate the specific seasonal energy efficiency *etas* of a space heating boiler. The model contains possibilities to include several different types of space heating appliances in the efficiency calculations, such as boilers, heat pumps, electricity or solar systems. The types of heat pumps included in the model is air source and ground source heat pumps tested in either floor heating- or in radiator heating mode. The model only applies for space heating.

<u>System limits</u>
Heat pump data is taken from tests according to EN14511, therefore the head losses from heat source fans or liquid pumps are taken into account in the heat capacity and COP data. This model also includes the heat sink liquid pump, Figure 12.

The model takes into account the net space heating demand, Lh, of the house. The heat demand of the house is a consequence of the choice of the load profile and the so-called system losses Lsys. The size of Lsys depends on the characteristics of the boiler and the installation characteristics. The system losses include fluctuation losses, stratification losses, distribution losses, buffer losses and timer losses, which are set as a percentage that is depending on the heat demand.

The model also includes losses from control, auxiliary equipment and system buffer standing losses.

A back up heater is used to cover up the energy demand that the heat pump cannot deliver.

The electricity use in the model is accounted with the primary energy factor 2.5.

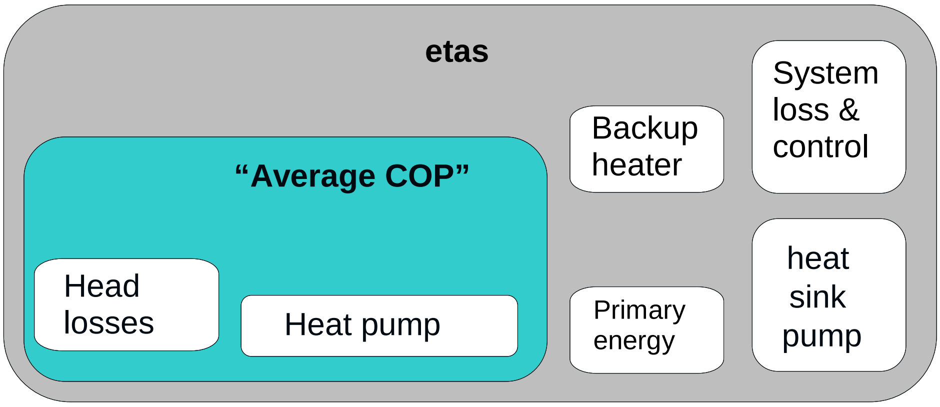


Figure 12. System boundaries according to EuP ENER LOT 1.

Input to the calculations

The test method for testing the heat pump refers to best testing practice e.g. EN 14511 (see document 7) except for some deviations. The test points are similar to the test points in EN14511:2007, but the temperatures of the return/feed temperature differs, see table IV.2 of the standard. In LOT 1 the temperature difference between Treturn and Tfeed gets larger the higher temperature of the Tfeed. Only three test points are necessary to calculate the seasonal energy efficiency by using this model.

The calculation uses a temperature bin method to evaluate the seasonal energy efficiency, etas. There are three different climates to choose among, warmer (+2°C), average (-10°C) and colder (-22°C), see table I.1, LOT 10. Each bin describes the equivalent number of hours corresponding to the bin temperature with a resolution of one bin/K. Input data to the calculations can be either the test points given in this method or test points given in EN 14511.

The maximum heating capacity, Pmax, at the different climates is calculated from the heating capacity data obtained in the test. It is not possible to choose the size of the required heat load for the building, but is given by the model for each bin level based on the capacity of the heat pump. To meet the lower heat load requirements at the different bin levels, the heat pump is assumed to work in part load condition. The heat pump does not have to be tested in part load operation; instead the model uses a degradation factor, Cd, to calculate the COP when working in part load condition. Cd can either be obtained from tests or a default value, Cd=0.15, can be used.

For fixed capacity units the default is COPmin= 0.89*COP at power output Phpmin=0.5*Php.
For staged capacity units the default is COPmin= 0.975*COP at power output Phpmin=0.5*Php.
For variable capacity units the default is COPmin= COP at power output Phpmin=0.4*Php.

It is optional to choose whether the heat pump operates with night set back or not. The bin assumes constant night temperatures during night set back to +1°C, +6°C and 0°C for each climate respectively.

Other inputs to the calculations is type of heat pump, type of operation of the heat pump, type of control of the heat pump, type of heating (floor heating or radiator heating), minimum source operating temperature, the effect of auxiliary equipment and backup electricity heater.

Other possible energy sources can also be chosen, but this chapter only treats the heat pumps.

<u>Output from the calculations</u>
The model calculates the energy use and losses based upon constant fractions. The fraction of the energy use and the different losses is displayed by the model. A diagram shows the energy supply per temperature bin and how it is covered from different energy sources. The seasonal energy efficiency, etas, is calculated.

Etas = Lh/Q_{tot}+cctrl *where* $Q_{tot}=Lh + L_{sys} + Q_{gen} + Q_{el}$

etas is the net space heating demand of the house over the sum of the generated heat of the system. Q_{tot} is the sum of the space heating demand (Lh), the losses from the heating system (L_{sys}), the primary energy losses of the energy input to the system (Q_{gen}) and the energy needed by the auxiliary equipment such as control and heat sink pumps (Q_{el}).

All electricity used by the heat pump and the backup heater is multiplied by a primary energy factor of 2.5. The model is not transparent. It is tricky to follow the outputs of the model since it consists from several excel-sheets and the information turns up all over. It is also difficult to understand all steps of the calculations. To be able to compare the results with field measurements and prEN14825 a value of SPF, the so called “average COP” (see the system boundaries) is calculated without the system losses. Average COP corresponds to $SCOP_{net}$ in prEN14825.

SP-method A3 528

SPA3 528 is a calculation program that is used to calculate the seasonal performance factor and energy saving over the year for houses having a defined heating requirement. It can be used for air/air heat pumps, air source heat pumps and ground source heat pumps. The heat loss from the house is defined in the program and given as the total loss factor, k-value, of the house [W/K]. The method can be used to calculate the energy requirement of a building with a k-value of either 109 W/K or 199W/K. A duration diagram of the outdoor temperature can be calculated from the mean annual temperature and together with the loss factor, the area under the duration curve gives the actual power requirement.

The heat pump is tested in accordance to EN 14511 at outdoor temperatures of -15°C, -7°C, +2°C and +7°C with an indoor temperature of +20°C. The heat pump is also tested in part load conditions according to CEN/TS 14825 at +7°C (75% and 50%) and at +2°C (50%). The lowest ambient temperature is assumed to be -15 °C and no heating is assumed to be required for ambient temperatures above +17 °C. The output data from the tests, thermal heat capacity and electrical input power, is used as input to the calculations.

2.1.3 AFP-model of Japan

Prior to calculating APFs, performance measurement needs to be conducted pursuant to JISC9612. Total load is calculated by multiplying loads in relation to outdoor temperature by load hours, while total power consumption is calculated by multiplying energy consumption in relation to outdoor temperature by energy consumption hours. APF's can be calculated by dividing total load by total power consumption (Figure 13). More details about the calculation method can be found in the Japanese national report in Appendix I.

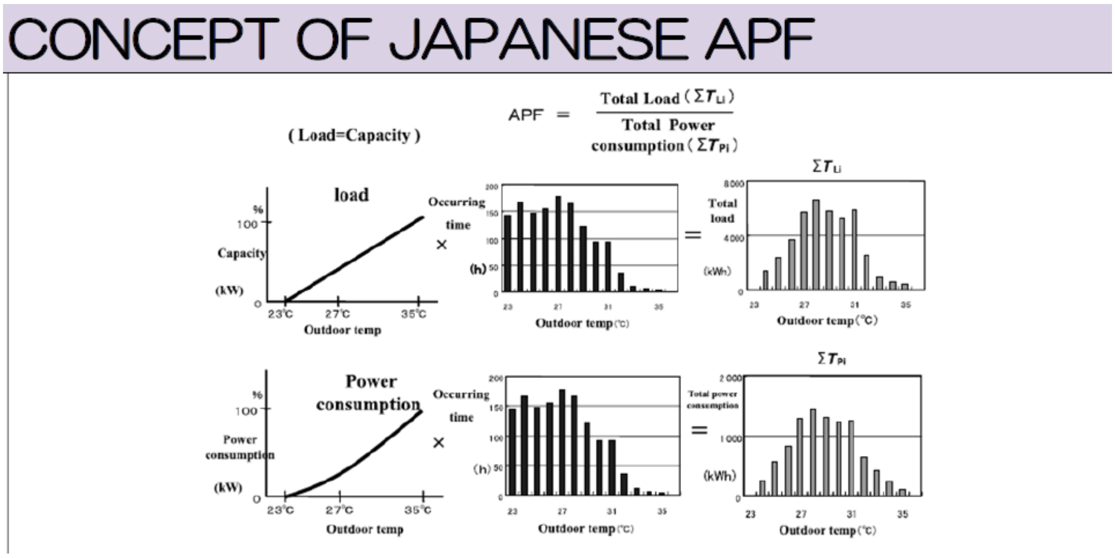


Figure 13. Concept of Japanese APF.

2.1.4 SIA 384/3, Energy consumption of heating systems for buildings

A new standard, SIA 384/3, has been developed in Switzerland. This model includes -amongst others- heat-pumps. It uses a SPF calculation based on a BIN method, where HP performance data is obtained from EN 14511 tests as input, and climate data is available from www.energycodes.ch. For the standard, a calculation model in Excel has been released. The well-known «WPEsti» (also known as «JAZcalc» in Austria) now implements SIA 384/3. A screen shot from the calculation tool can be seen in Figure 14. This work is described in more detail in Appendix IV.

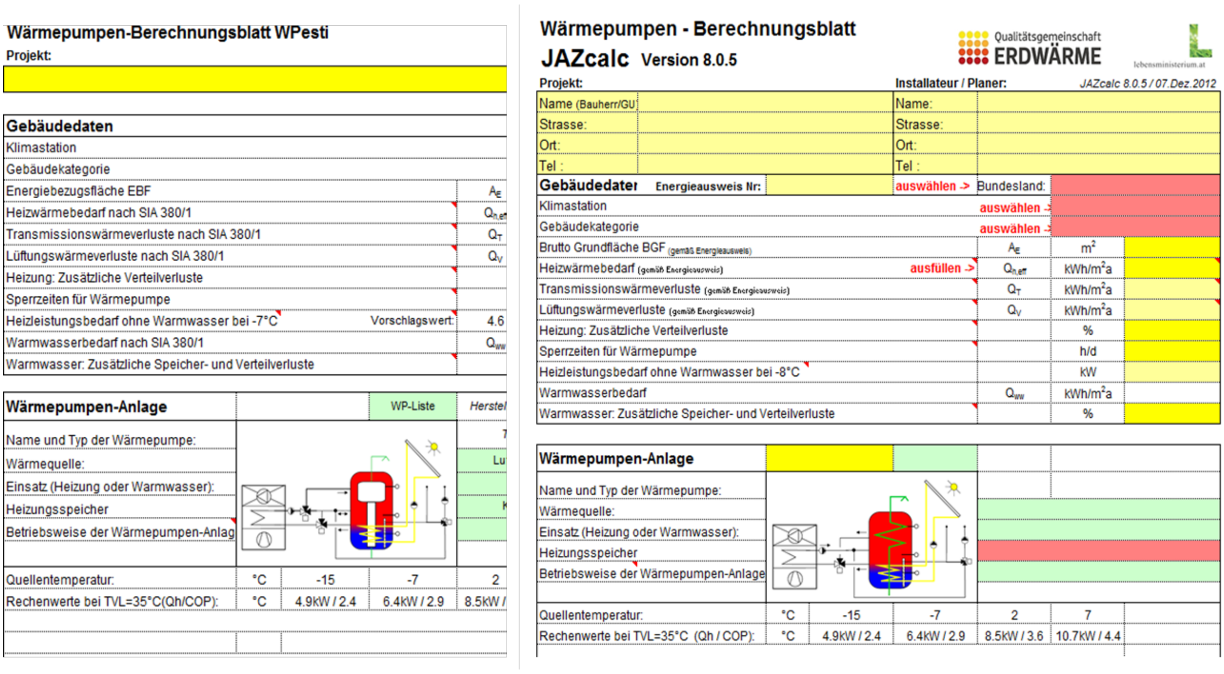


Figure 14. Calculation tool for the SIA 384/3 standard.

2.1.5 NEN 7120, Netherlands

In netherlands, Energy Performance for Buildings is governing much of the development of methods. In the NEN 1720 standard, EPC (Energy Performance Coefficient) is standard. By increased performance to EPC=0.4 in 2015, better heating products are expected in the market.

There is a method for determination of the efficiency of air to water heat pumps for heating purposes (included in Appendix VIII).

- This appendix describes a method for determination of the efficiency of air to water heat pumps. The efficiency and energy fraction of hybrid systems, heat pumps with external supplementary heating, is also described.
- The efficiency and the energy fraction of air to water heat pumps depend on several building specific variables. Therefore this method can be applied in two ways:
    - For specific buildings the efficiency and the energy fraction can be determined
    - For a range of building specific variables the efficiency and the energy fraction can be determined. In this case these variables should be classified according to the rules in appendix

Other national standards or calculation models, and more details on some of the above mentioned methods are reported in national reports from Japan, Finland, Switzerland, US in the respective national report appendices.

2.2 Strengths and weaknesses with current methods

2.2.1 prEN14825

Strenghts

Strength of standard prEN14825 is that it includes all kinds of heat pumps (except exhaust air heat pumps). (However the later versions of the standard do also include exhaust air heat pumps). The model treats heat pumps both in heating and cooling operation. The fact that the heat pump is tested in exactly part load should result in more sufficient results compared to degradation coefficients etc. The model is foreseeable and quite easy to follow.

- A strength of standard EN14825 is:
  - it includes all kinds of heat pumps .
  - The model treats heat pumps both in heating and cooling operation.
  - The fact that the heat pump is tested in exactly part load should result in more sufficient results compared to degradation coefficient etc.
  - The model is foreseeable and quite easy to follow.

Weakness

The model is not completely clear with its definitions of part loads. The part load ratio for which the heat pump is to be tested is the part load energy demand of the building at the corresponding temperature bin. To perform the SPF calculations according to prEN14825 the heat pump is tested at a certain climate (A,W or C) and a certain heat load profile for the building. This means that the test data might not be suitable for another climate or another heat load.

It is also not completely transparent since it describes (ANNEX C) the reference heating/cooling demand and the number of hours in each operational mode (active mode, thermostat off mode, standby and crank case heater mode) is decided from weighted climate, type of building, internal gains, set back setting and so on, but there is no reference that describes the calculations. Therefore it is not possible to recalculate the hours to fit specific needs. The climate hours that describes the temperature bins does not seem to be adjusted in any ways since it is the same hours that is used in the model proposed in EuP ENER LOT 1.

Another weakness is that the model does not include domestic hot water. Figure 15 and Figure 16 shows different tapping patterns; Figure 15 show the actual tapping from one field monitoring project in litres per minute. In Figure 16 on the other hand, the different tapping patterns according to EN 16147 is exemplified by the tapping time versus the temperature of the tap water (section C of the figure). Even if the standard EN16147 gives a test of the performance to produce DHW, the tapping patterns are very unlike the actual found in Figure 15. Since also the DHW use is very user dependent, there is the question if using different tapping patterns in marketing is such a good idea. One heat pump bought by a pair of retired citizens might perform very ill if the house is sold to a young family with many kids. The user experience of the same heat pump, with the same performance might then be very different due to the shift in the share of DHW production of the total energy use.

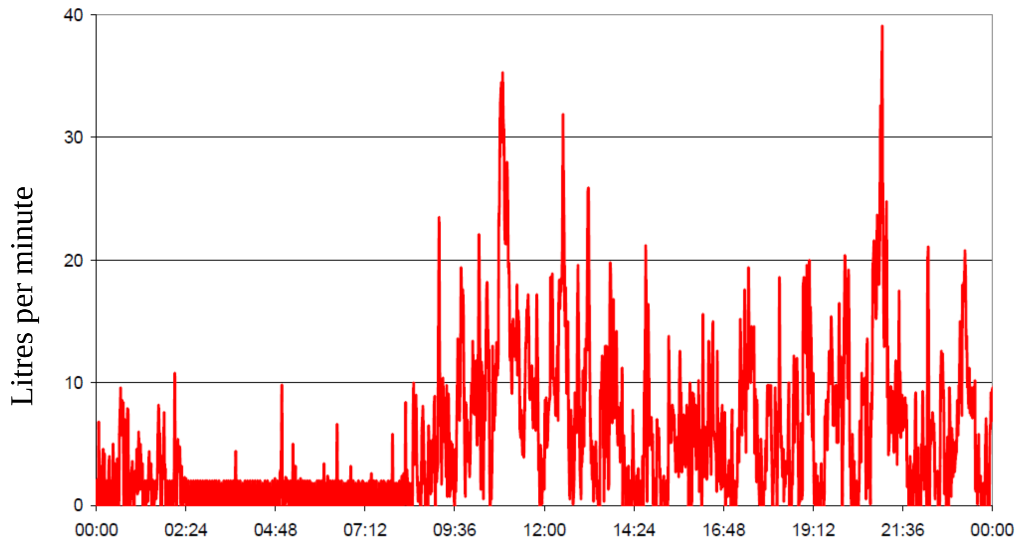


Figure 15. Tapping pattern according to one field monitoring.

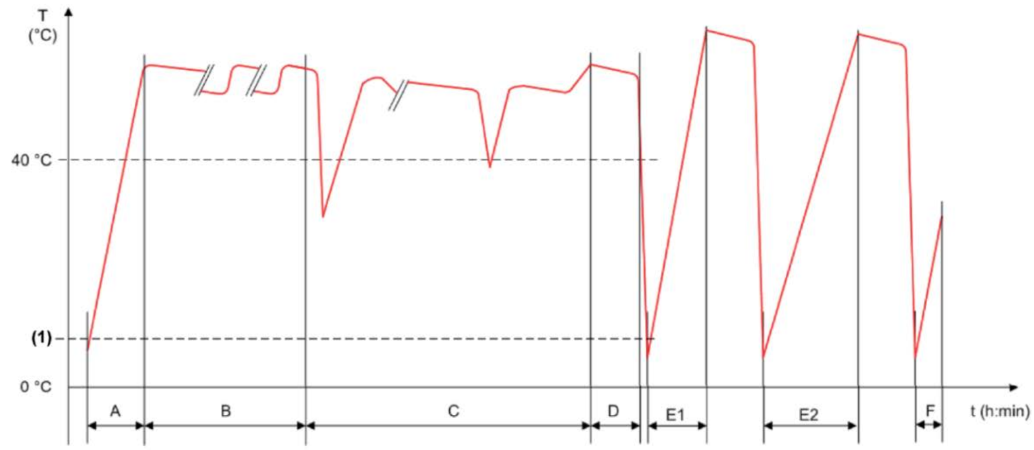


Figure 16. Tapping pattern according to EN16147:2011.

Possibility

The model could be developed so that it would be possible to decide the energy demand of the house. It could also be a possibility to fit the model to your own climate using an individual climate file. Maybe the ground water temperature and thereby the borehole temperature could be climate depending as well.

It should be obvious how interpolations or extrapolations of capacity and/or COP should be performed to avoid differences between different users.

Risk

The performance of water/water heat pumps can be overestimated, especially at the cold climate, since they are only tested at +10°C at the cold side (in reality the ground water temperature can be lower than this). This can also be the case for other ground source heat pumps. However, in later versions brine-to-water heat pumps shall be tested at +0°C and then this risk for overestimation is reduced.

The degradation coefficient Cc might be a disadvantage for a ground source heat pump when default values are used. Cc =0.9 is a larger degradation of GSHP:s than what is shown in reality. (In later versions of the standard, there is an alternative way of calculating a more realistic Cc value based on tests a compressor off state operation). There is a risk that the requirement of having heat pumps tested in part load might lead to extensive laboratory tests, which can be costly. It is also difficult to get sufficient data from existing laboratory tests, since few heat pumps have been tested in part loads.

2.2.2 **EN 15316-4-2**

<u>Strength</u>
This model is very wide and thorough in its content. It treats both room heating and tap water production. The model is adaptable to different climates and the resolution of the temperature bins can be chosen.

The model specifies the requirements and losses of the certain house and defines recoverable respectively unrecoverable energy.

It is not necessary to test the heat pump at the part loads, since there are default values that can be used.

The model can be used to calculate the SPF for the entire system with the building included or only for the heat pump.

<u>Weakness</u>
The strengths of this model could also turn out to be its weaknesses. The wideness of the model makes it complicated and twisty. There are too many aspects that are taken into account in the calculations. The standard refers to several other standards for calculations of losses and needs. The model requires large knowledge of the house.

The fact that default values can be used to calculate the operation in part load for the heat pumps can result in lower accurateness of the model.

The model does not treat operation in cooling mode.

<u>Possibility</u>
The model can be studied and give input to a new easier model.

<u>Risk</u>
There is a present danger of doing mistakes when using the model. The large amount of data that is taken into account will probably result in much estimation that will differ from case to case and will therefore result in incomparable outcome of the model. Also the same heat pump installation can probably give different results depending on the way it is calculated, (choosing method, input, accuracy and test points).

2.2.3 EuP ENER LOT 1

Strength

Test data from EN 14511 can be used in the calculations. The model provides default values to recalculate the test points to fit the part load of the heat pump for the different kinds of heat pumps (fixed capacity, staged capacity och variable capacity). The capacity and corresponding COP values are then interpolated between the temperature bins. However, the accurateness of the recalculation is unknown.

The model itself has suggested test points with a radiator curve (supply temperature) that is adjusted to the outdoor temperature. At colder outdoor temperatures the supply temperature is higher and at warmer outdoor temperatures the supply temperatures are lower.

The model can be used to calculate how to cover the energy need of the house by using different techniques, for example solar cells, heat pumps and fossil fuel. This is a good thought, but might not be interesting in this project (?).

Weakness

Unfortunately the model still contains bugs and technical mistakes in the equations and the way of thinking. It seems to be adjusted to boilers and bio boilers instead of heat pumps.

The model does not include a power balance, but is doing a temperature balance instead. This makes the distribution of the energy need and the required amounts of backup heat differ from the theoretical needed.

The model includes a decided fraction of heat loss that cannot be escaped from. For example if the heat pump does not use night set back a default penalty loss of 12% from the total delivered energy is drawn off. The losses from the apparatus and system operation are also decided in percentages.

At part load operation there is no change in the system flows. This does not seem right with controlled radiators. (Should the radiators be controlled or is it enough with a displacement/adjustment of the radiator curve?)

The night set back function uses the same night temperature all year around, which is not the case in reality.

It is not possible to choose the energy requirement of the house; instead the energy demand is an outcome of the capacity of the heat pump. If the heat pump is not comprehensive also the fraction of backup heat is needed to decide the energy demand of the house.

GSHPs are disadvantaged when recalculating the operation data to part load operation. The ground source heat pumps are degraded by a factor 0.89 at 50 % of the delivered capacity. (The Cd factor, i.e. the on/off control, is overestimated for water borne systems)

Even though the program is transparent it is hard to understand and follow the calculations. The interface of the program is not very friendly and can easily confuse the user.
The model does not include domestic hot water.

Possibility
Making the ground water and borehole temperature climate dependent might lead to results more close to its actual installation spot.

Risk
The model is not adjusted to fit heat pumps and is disadvantaging heat pumps. Despite this the COP and capacity of water to water heat pumps can be overestimated since they are tested at +10°C at the cold side (this can also happen to ground source heat pumps, but probably not to the same extent).

2.2.4 ErP ENER LOT 10

Strength
This model can be used both in heating and cooling mode and it has three different climates both for the cooling season and the heating season. The model has reference heating/cooling demands to choose between.

Weakness
The model takes only air to air heat pumps into account. In accordance to prEN14825 the test points for the heat pump has to be chosen specifically to fit the chosen climate and heat profile of the house.
In accordance to LOT 1 the model does not include an effect balance at each temperature bin. This results in that the heat demand of a house at a specific temperature bin is different at different climates and that the heat requirement of a backup heater is misleading.
The model does not seem to be entirely consistent, partly it is contradicting itself.

Possibility
To make the model usable at other spots it would be better to make it possible to use other climates. Now the model only provides a number of specified heat loads of the house. It would be useful to be able to freely choose the heat demand of the house. There is a risk though, that since the heat pump has to be tested in part load, it has to be tested at each specific heat requirement.

Other types of heat pumps could be included in the model. The model only provides the SPF (SCOP) with the backup heater included. For comparable reasons, it would be useful to include a SPF with backup heater excluded.

Risk
It is not obvious whether the excel model is compatible with the standard. There are also some calculations in the standard that seems to be incorrect.

2.3 SWOT analysis of existing standards

2.3.1 EN14511:2011

The standard EN 14511 covers not only capacity measurement but also safety in operation and different temperature levels on sink side, Table 4. EN 14511 is broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) and different funding programms in Europe. The Standard is not covering capacity controlled heat pumps and the Nominal capacity of capacity controlled HPs is not clearly defined. In EN 14511 circulation pumps are included in the testing procedure only a small amount is integrated in the calculation. As it is a standard which is referenced very often, changes have a large influence to other standards.

Table 4. SWOT analysis of EN14511:2011.

	HELPFUL To achieving the objectives	HARMFUL To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ The standard covers not only capacity measurement but also safety in operation ▪ Different temperature levels on sink side provided	WEAKNESSES ▪ Capacity controlled heat pumps are not covered ▪ Nominal capacity of capacity controlled HPs is not clearly defined ▪ Circulation pumps are included in the testing procedure
EXTERNAL FACTORS	▪ Broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) ▪ Broadly accepted for different funding programs in Europe ▪ Might change to an ISO standard OPPORTUNITIES	▪ Large effort to modify and adapt because referenced within a variety of other standards THREATS

2.3.2 EN 15879-1

The EN15879-1 is in principle identical to EN 14511 with all strengths, weaknesses, opportunities and threats but covering direct exchange to water heat pumps, Table 5. The main problem is that it is not part of EN 14511 and therefore revisions have to be done separately.

Table 5. SWOT analysis of EN 15879-1.

	HELPFUL To achieving the objectives	HARMFUL To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ The standard covers not only capacity measurement but also safety in operation ▪ Different temperature levels on sink side provided	WEAKNESSES ▪ Capacity controlled heat pumps are not covered ▪ Nominal capacity of capacity controlled HPs is not defined ▪ Only on/off heat pumps are covered ▪ Circulation pump is included in the testing procedure
EXTERNAL FACTORS	▪ Accepted and used for quality assurance ▪ Broadly accepted for different funding programs in Europe OPPORTUNITIES	▪ Closely linked but not part of EN 14511 ▪ Market of DX-HPs is decreasing THREATS

2.3.3 EN 16147

The standard EN16147 covers Domestic hot water heat pumps. It provides test procedures which account storage losses and uses the concept of tapping cycles. The main weakness is that no reference tapping cycle is defined and it is hardly possible to compare different products, Table 6.

Table 6. SWOT analysis of EN 16147.

	HELPFUL To achieving the objectives	HARMFUL To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ Storage losses are accounted for ▪ Concept of tapping cycles	WEAKNESSES ▪ Definition of efficiency figure (COP instead of PF) ▪ Five different tapping cycles without one common cycle ▪ Maximum sample time (10 s) to large for accurate measurement ▪ Unrealistic high maximum tap water temperatures
EXTERNAL FACTORS	• The revision is currently ongoing • Only international standard for DHW HPs • First standard to apply EU mandate M324 for DHW tapping cycles OPPORTUNITIES	▪ Generally not accepted due to unrealistically low COPs THREATS

2.3.4 **VDI 4650**

In 2014 a revision of the VDI 4650 standard was performed. Reasons for this included: unclear source for the correction factors used, Lack of calculation method for inverter heat pumps, inflated results for calculation of high temperature heat pumps and inflated results for calculation of domestic hot water

The revision of the VDI 4650 included a simplified method for the calculation of the seasonal performance factor of heat pumps SCOP. As input for the calculation, COP values and correction factors are used. These correction factors include both space heating and DHW production, as illustrated in Figure 17 below.

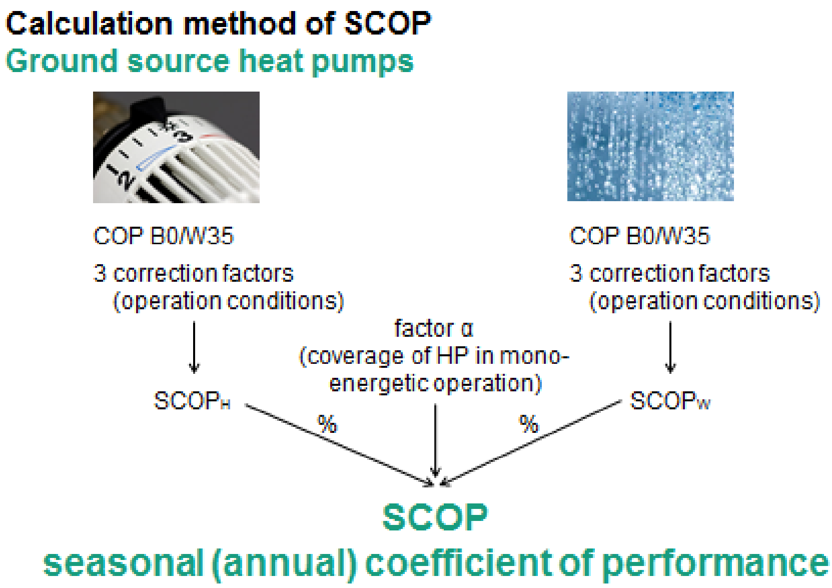


Figure 17. Method for calculating SPF from COP and correction factors in VDI 4650.

New factors for DHW were introduced, and in a comparison between VDI 4650 and EN 16147:

Heat pump, thermal power 6 kW by B0 / W35 and COP_N of 4,9 with a buffer storage (DHW Storage), volume of 175 litres and power demand of heat pump in standby mode of 35 W, using a tapping profile XL.

By calculating the SPF's using the EN standard, and the old and revised versions of VDI 4650, the following results were obtained:

- 2,85 efficiency according to EN 16147
- 3,26 efficiency according to EN 16147 corrected with standby – loses
- 3,60 efficiency according to "new" VDI factors
- 4,39 efficiency according to VDI 4650 Part 1: 2009

As can be seen, a much better accordance with EN 16147 has now been reached.

Comparison
VDI 4650 vs. „WP Effizienz" Project (Monitoring)

The revised method was also benchmarked to some field monitoring results performed in the WP Effizienz project. The field sites with well-known heat pumps were calculated for the actual climatic conditions, and the results are shown in Table 7.

Table 7. Comparison of monitored and calculated SPF. VDI 4650 vs. „WP Effizienz"

	measurement	VDI 4650	difference
	air to water heat pumps		
R1	3.23	3.22	-0.3%
R2	2.79	3.49	25.1%
R3	3.23	3.24	0.3%
R4	2.56	3.73	45.7%
	brine to water heat pumps		
R5	4.03	4.35	7.9%
R6	3.61	4.14	14.7%
R7	3.30	3.76	13.9%
R8	3.55	3.91	10.1%
	water to water heat pumps		
R9	3.66	4.20	14.8%
R10	2.90	3.71	27.9%

Comparison between different methods VDI 4650 vs. other methods

A comparison was also made for one field monitored site, where monitored SPF was used as a benchmark. As can be seen from Figure 18 below, all calculation methods have underestimated the SPF compared to the monitored value (messung). Figure 19 show that with a 90% or 95% confidence interval on the simulated values, all results are within the error margin.

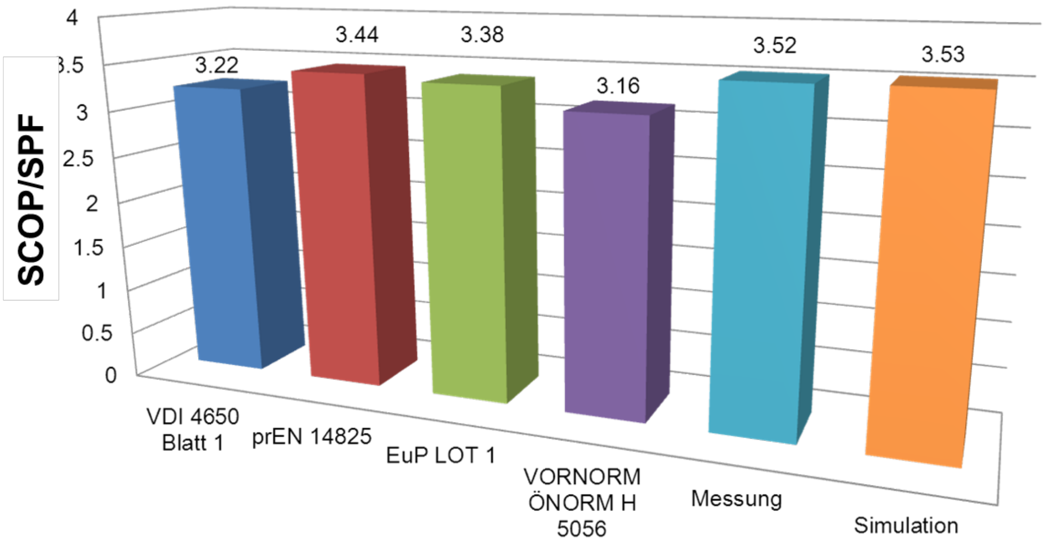


Figure 18. Comparison of different calculation methods with one field monitoring site.

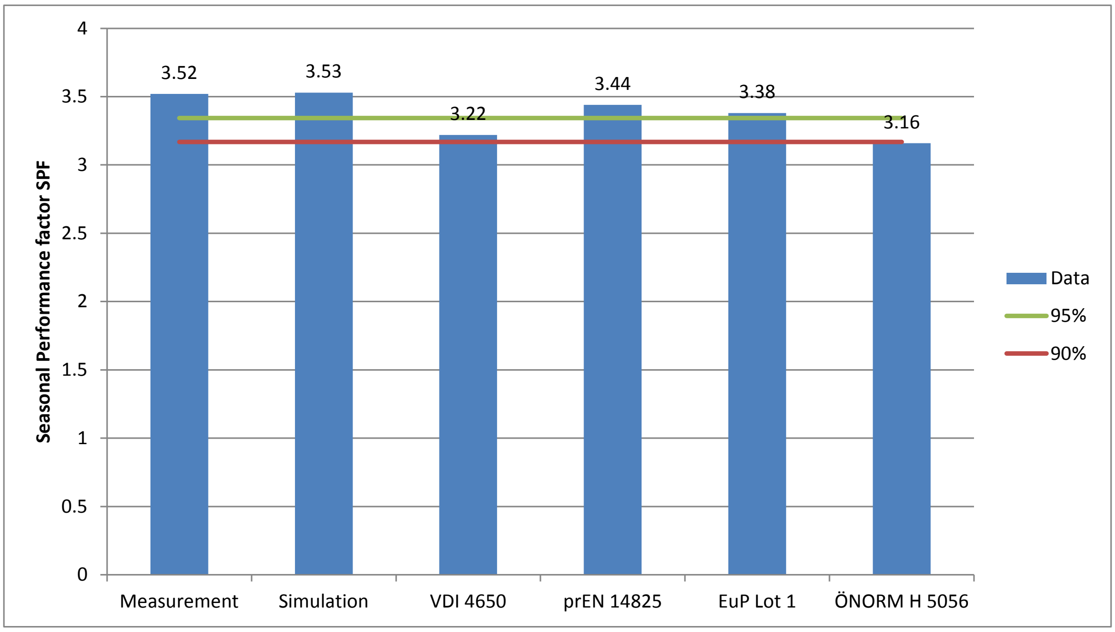


Figure 19. Data with 90% and 95% confidence intervals drawn.

2.4 Local standards, with understanding of differences between them

Based on the analysis above, and the fact that there are numerous methods for calculating SPF, taking into consideration different national geographic conditions and other special conditions, there was a quite clear view that calculation methods for different climates maybe need to be local, but considering the test points for lab test standards (Table 8), there is not that many points of difference. It would therefore be of interest to make a thorough evaluation of the consequences of harmonizing the parameters for lab testing. As an example, what would the difference be if the zero load temperature in all standards was set to 17 °C? This would imply lowering the temperature for US conditions by 1,3 K, and increasing the temperature for European conditions by 1 K. By looking into Table 8 below and making adjustments to standards in a way that all testing requirements are alike, a global standard for testing could be developed.

Table 8. Comparison of main standards in three geographic regions of the world.

	APF (Year-round performance factor)	EN14825 SPF	ANSI/AHRI210/240 SEER
Building load	Load=0 : Outdoor tem. at 17 ℃ Load=82% of rated capacity at outdoor temp. of 0 ℃ in heating	Load=0 : Outdoor tem. at 16 ℃ Load=100%: Outdoor tem. at -10℃	Load=0 : Outdoor tem. at 18.3 ℃ Load=100%: Outdoor tem. at -5℃
Estimated load appearance hours	Heating period: Oct 28 – Apr 14 Heating hours: 6:00 – 24:00 Total heating hour: 3042 hrs	Heating hours of standard region is 4910 hours.	750 ~ 2,750 hours depends on regions.
How to calculate device performance	Performance level of products in every outdoor temperature degree can be calculated by measuring rated and intermediate capacities at 7 ℃ outdoor temperature and maximum capacity at 2 ℃.	Can estimate device performance curve by measuring 4 points between 88 % and 15 % on load curve in relation to outdoor temperature	1.Max capacity line; connect 2 points, measured at 1.7 ℃ & -8.3 ℃ 2.Min. capacity line; connect 2 points, measured at 8.3 ℃ & 16.7 ℃ 3. Intermediate capacity line: Measuring at 1.7 ℃ then make a line.
Measuring points	Total: 3 points; 2 from rated and intermediate capacities at 7 ℃, 1 from max. capacity at 2 ℃.	3 -5 points in 3 regions (basically measured at full rated capacity or 75%, 50% and 25 % of rated capacity.	5 points; 2 at max. compressor rotating speed, 1 at intermediate capacity and 2 at min. capacity
Conditions of Measuring points	Rating and intermediate capacities at 7℃, and max. capacity at 2 ℃	-7.0℃:88% 2℃:54% 7℃:35% 12℃:15%	-8.3℃:Max 1.7℃:Mid. 8.3℃:Max.Min. 16.7℃:Min.
	Outdoor temp. is fixed at rated condition of 7℃ & 2℃	4 measuring points. Part loads are measured at different temps.	5 measuring points at 4 different temps. Part loads are measured at different temps.
Remarks about capacity range	Low capacities are not measured due to lacking fairness considering large measuring errors in low capacities.	Measuring error is large at 15% of rated capacity.	Measuring error is large at 25% of rated capacity.
Frost	Considered in every load between max. and intermediate.		Considered only in performance curve of max. load not in other loads.
How to measure a intermittent operation for majority	Degradation Coefficient Cd is not used. As the lower limit of variable width of heating capacity is rarely below the intermediate operation in Japan, they do not calculate by using Degradation Coefficient Cd.	Degradation Coefficient Cd=0.25 is used. Same as the ARI210/240	Degradation Coefficient Cd=0.25
Stand-by power consumption	Not considered	Suggested that stand-by power consumption should be considered	

3 COMPARISON OF EXISTING CALCULATION METHODS AND RESULTS FROM FIELD MEASUREMENTS

The study presented here is based on results from the Fraunhofer ISE field measurements in the project WP Effizientz. Calculations of predicted performance were made by SP for a number of different calculation methods.

3.1 Heat (and cooling-) demand of the house

This study is focused on heat pumps for indoor heating. The study is made in houses with different heat demand. The ground source heat pumps in this study are considered monovalent, but it is difficult to determine the actual energy demand of the house. When using the calculation models the required heat load of the house is decided by the capacity of the heat pump.

The studied air to air heat pump is not monovalent. The energy demand of the house with the heat pump installation was estimated in the field study. When using the calculation models the energy demand of the house were tried to be the same as in the field study.

3.2 Indoor climate

The indoor climate is expected to reach 20°C for all models. In the calculation models the heat pump is used to reach a temperature of 16°C. Internal gains are expected to contribute to the last temperature increase.

The actual indoor temperature has not been measured in the Fraunhofer ISE field measurements. Thereby it is not possible to compare the real indoor temperatures with the temperatures estimated in the calculation models.

3.3 Outdoor climate

The outdoor climate follows the climate of the year. The calculation models use the same temperature climate when calculating SPF for the ground source heat pumps. The climate corresponds to a European average climate, Strasbourg, with the coldest temperature of -10°C.

The field measurements of the ground source heat pumps are carried out in Germany. The heat pumps installations used for the SPF calculations are spread over the country, from the Hamburg area in the north to Stuttgart in the south. In the calculation models the average climate is chosen as the climate mostly corresponding to the German.

The air to air heat pump installation is made in a climate that is similar to the “colder” climate. Therefore the colder climate is used in the calculation models when calculating SPF for the air to air heat pump.

3.4 Definition of SPF field measurement system boundaries

In the EU project “SEPEMO-Build” four SPF’s with different system boundaries were defined. The definitions from the SEPEMO project have been used for calculating the SPF for the field measurements. The four defined SPF’s are:

SPF_1 includes only the heat pump unit itself. Thereby SPF_1 is identical to the average COP for the measured period.

$$SPF_1 = \frac{Q_{HP}}{W_{HP}} \qquad \text{(Equation 4)}$$

SPF_2 consist of the heat pump unit and the equipment needed to make the heat source available the heat pump.

$$SPF_2 = \frac{Q_{HP}}{W_{HP}+W_{heat\_source\_pump}} \qquad \text{(Equation 5)}$$

SPF_3 represents the heat pump system SPF. SPF2 includes the heat pump and the heat source pump as in SPF_2, but also the back up heater.

$$SPF_3 = \frac{Q_{HP}+Q_{back\_up\_heater}}{W_{HP}+W_{heat\_source\_pump}+W_{back\_up\_heater}} \qquad \text{(Equation 6)}$$

SPF_4 includes all parts relates to SPF_3, additionally SPF_4 also includes the distribution of the heat.

$$SPF_4 = \frac{Q_{HP}+Q_{back\_up\_heater}}{W_{HP}+W_{heat\_source\_pump}+W_{back\_up\_heater}+W_{heat\_sink\_pump}} \qquad \text{(Equation 7)}$$

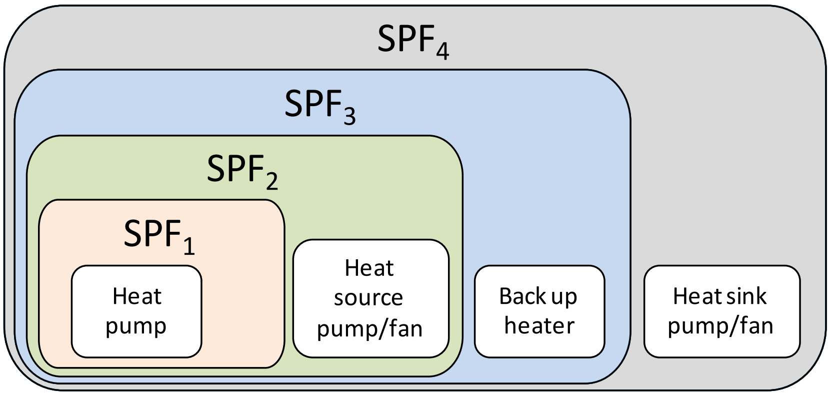


Figure 20 System boundaries for calculations of SPF

3.5 Calculation of SPF

In this study SPF is calculated for three of the four different system boundaries and categories by using data from the field measurements. The system boundaries used are SPF_2, SPF_3 and SPF_4. The categories are “heating only”, “heating and domestic hot water production” and “domestic hot water production”.

For facilities where the installed heat pump also is tested in a laboratory, the laboratory test results are used to calculate SPF by using the calculation models. This is the case for seven different ground source heat pumps and one air to air heat pump.

This section will explain how the calculations were performed and what assumptions were made for the different models.

<u>Field measurements</u>

Ground source heat pumps
All analysed heat pump systems were installed in German single family houses with floor heating. The heat pump is more or less covering full capacity, only a very small amount of backup heat has been used during the year of measurements. The heat pumps in the study have been installed during the years 2000 and 2005. The data used for the SPF calculations are based on field measurements carried out during one year, with one exception the SPF for site no. one (1) is based on data measured from January to August.

The calculations of SPF's are based on the field measurements data from the Fraunhofer study. In the data we have received from the Fraunhofer study the total energy consumption for the heat pump system and its components is presented as well as the energy consumption divided into energy used for space heating and energy used for production of domestic hot water.

In this project we have not been able to evaluate exactly how these allocations have been made. For some of the studied installation sites a part (up to 20 %) of the total electricity consumption has been allocated neither to space heating nor to the domestic hot water production. This is mainly the case for the electricity consumption. For the heat produced no energy gap is seen between the total energy production and the energy divided into space heating and domestic hot water.

The calculated SPFs in the study are based on the energy allocated to the space heating only, this in order to make the results comparable to the results from the calculation models in prEN14825 and Lot 1, which not include the production of domestic hot water.

Air to air heat pumps
The field measurement of the air to air heat pumps is carried out in single family houses located in the Borås area of Sweden. All houses in the study have electricity driven radiators for back up heating. The field measurements are based on SP method 1721. From the field measurements SPF_2 and SPF_3 has been calculated as described below.

The electricity consumed by the heat pump, W_{HP}, is measured continually while the produced space heating is measured at five "performance tests" done at different outdoor temperatures. During the performance tests the heating capacity of the heat pump is measured during stable conditions and is thereby not including any defrost period. Therefore the calculated COP for each test point is based on data from only a part of the operating cycle.

The total amount of heat produced during the total measuring period needs to be calculated based on the five performance tests. The calculations are made as follows:

$$Q_{HP\_year} = \sum W_{HP\_month} * COP_{average\_month} \qquad \text{(Equation 8)}$$

COP is calculated from the performance tests at made at different outdoor temperatures. During the performance test both the electricity consumed (W_{HP}) and the produced heat (Q_{HP}) is measured and COP can be calculated as Equation 9 shows:

$$COP = \frac{Q_{HP\_test}}{W_{HP\_test}} \qquad \text{(Equation 9)}$$

From the five performance tests the COP for the heat pump can be expressed as a function of the outdoor temperature. From this function an average COP for each month is calculated based on the average temperature for the month. Knowing the electricity produced each month by the heat pump the SPF_2 can be calculated:

$$SPF_2 = \frac{Q_{HP\_year}}{W_{HP\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year}} \qquad \text{(Equation 10)}$$

The electricity consumed by the backup heaters is also measured, thereby SPF_3 is calculated as:

$$SPF_3 = \frac{Q_{HP\_year}}{W_{HP\_year} + W_{backup\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year} + W_{backup\_year}} \qquad \text{(Equation 11)}$$

Due to lack of data from laboratory testing of the heat pump models included in the field measurement, this study has only been able to compare the SPF values from the field measurements with the SPF calculated with the calculation models prEN14825 and Lot 10 for one of the tested heat pumps.

prEN14825

Ground source heat pumps

When using prEN14825, data according to Table 9 has to be filled in. The chosen climate, “average” gives that Tdesign is -10°C. Tbivalent is the outdoor temperature where the capacity of heat pump covers the heat demand of the house. It is set to minus 10°C, to make the heat pump comprehensive, like in the field study. T OL, the operation limit temperature, is set to -25°C. This temperature declares where the heat pump no longer can operate. The model calculates Pdesign as a result of Tbivalent and is the heat demand of the house at Tdesign.

Table 9. Data for EN 14825 calculations.

T design	-10 °C
T bivalent	-10 °C
T OL	-25 °C
P design	8,81 kW

The test conditions for the heat pumps were taken from Table 20 in the standard, brine to water heat pump, average climate and low temperature application. The unit is assumed to be a fixed capacity unit with fixed outlet temperature. The heat pumps in the study where all tested in full load according to EN 14511. For the part load conditions the COP was calculated by using equation 12 in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The capacity and COP at Tbivalent and TOL is set to the maximum, while the COP for the delivered capacity at the different outdoor temperatures is calculated by using equations from the standard prEN14825. The default degradation factor where Cc=0.9 is used.

Air to air heat pumps
The colder climate is chosen for the calculations, since this climate is similar to the climate where the field installation is. The bivalent operation point of the heat pump is calculated by using SPA3528, which is another model for the calculation of SPF. The bivalent point is 0°C. The operation limit point was set to -20°C.

At -7°C the heat pump operates in full load to deliver heat to the house. At +2°C and at +7°C the heat pump operates in part load. COP for part load operation is interpolated by using linear interpolation between existing test points. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 50% and 57% of the heat pump capacity. At +12°C the required heat load is so small that the heat pump is assumed to cycle on/off. The capacity of this point is calculated by using equation 11 in the standard. The COP for the bivalent point is interpolated from test points in full load operation at +2°C and -7°C.

<u>*Lot 1*</u>
In Lot 1 there are some general inputs that has to be filled in into the excel sheet. The following inputs are used:

- Reduced setback: Yes
- Radiator (with setback): No
- Floorheat (24h): Yes
- Control: 4 – Weather ctrl BT
- Pump: 3 fixed speed
- Pump timer: 24h
- Buffer: No
- Tmino: -25°C

The only heat generator in use is heat pump. No back up heater is included in the calculations.

The default degradation factor, Cd= 0.15, is used. Default is also used for hpaux (=30W) and hpsb (=10W). The test conditions are taken from the reference test conditions in table V.3. in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The model recalculates the test data to fit with the test conditions of Lot 1 (table V.2.).

Data for part load operation is calculated from equations of "option B" at page 27 in the standard, where COPmin= 0.89*COP at power output Phpmin=0.5*Php for a fixed capacity unit.

From Lot 1 two different results are obtained, “etas” and “average COP”. Etas are calculated by involving the primary energy factor of 2.5 which makes it difficult to compare with other calculated SPF. However, “average COP” corresponds to SPF 1.

Lot 10

Air to air heat pumps
The design load of the house is chosen to 8,5kW, which is the design load that best corresponds to the size of the house in the field measurement. The house in the field is installed in a climate, similar to “colder” climate, therefore “colder” is chosen. The test points for the calculation are given in a table at page 24 in LOT 10 Annex II. The heat pump is tested according to EN 14511 and CEN/TS 14825 for part load conditions.

The heat pump is a variable capacity heat pump, but since the heat pump is not tested at exactly the required heat effect (within ±3%), the calculations of COP has to be performed in accordance with a staged capacity unit.

At -15°C and -7°C the delivered capacity from the heat pump is lower than the house requires; capacity and COP data are taken from operation in full load at these outdoor temperatures. An exception from the standard is made, since the standard proposes a recalculation of the COP at those points. The recalculation does not seem to make sense and is therefore ignored.

At +2°C and +7°C the heat delivered from the heat pump exceeds the required heat from the house and is therefore operated in part load. COP for part load operation is interpolated by using the equation for staged capacity units at page 26 in the standard. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 57% and 44% of the heat pump capacity.

The heat pump is not tested at +12°C. Full load operation at +12°C is extrapolated from test data at +7°C and +2°C. 50% part load is extrapolated from 50% part load operation at +7°C and +2°C. COP for the required effect is extrapolated by using this data. Each extrapolated COP value is corrected with a degradation factor of 0.975.

Default values are used for the degradation factor (Cd=0.1), turndown ratio heating (=25%), thermostat off mode (50W), crankcase heater mode (=10W) and off mode (=10W). The bin limit is set to -20°C.

3.6 Analysis of the results

The results from the SPF calculations of the different heat pump installations in field is compared with the results obtained from the laboratory data used in calculation models.

Ground source heat pumps
Most of the heat pumps installed in field operates both in floor heating mode and produces domestic hot water. The measurements include both kind of operations and the results are presented in the Table 10 and Figure 21 below. SPF for domestic hot water production is always lower compared to operation in heating mode. The energy

balances is not 100% complete for the field measurement, which is quite common in field measurements, due to monitoring uncertainties.

Table 10. The table shows two different SPF from the field measurements in two different levels. SPF for heating and DHW (domestic hot water) is lower than SPF for heating only. This is because COP for domestic hot water production is lower than COP for heating

Results field measurements				
	Heating and DHW	Heating and DHW	Heating only	Heating only
	SPF 1	**SPF 3**	**SPF 1**	**SPF 3**
site 3	3,70	3,46	4,66	
site 6			3,86	3,43
site 8	4,13	3,02	4,71	
site 9	3,97	3,64	4,53	
site 11	3,62	3,32	4,71	4,56
site 13	2,71	2,55	3,99	3,83
site 14	4,14	3,55	5,43	5,16

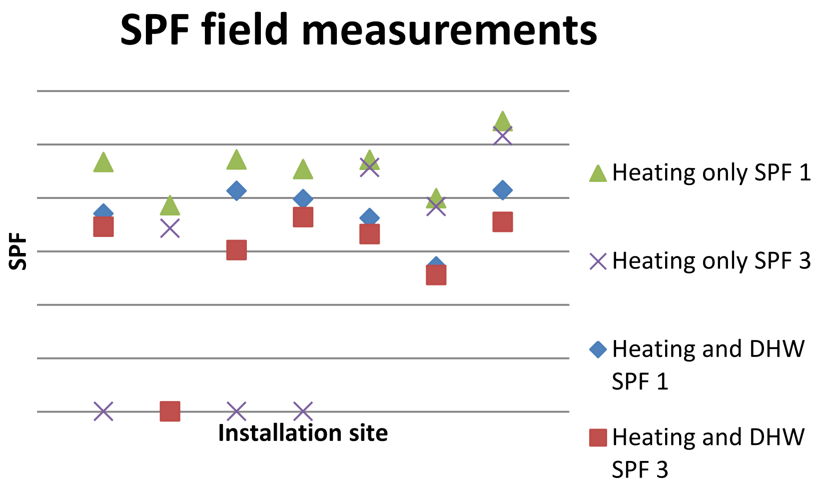


Figure 21. The figure show SPF results from two different SPF, "heat only" and "heat and DHW" (domestic hot water heating) at two different levels, "SPF 1" and "SPF3", from field testing.

The conditions for measurements in a laboratory and in field differ with respect to various factors e.g. the boundary conditions. SPF1 in field measurements includes the electrical energy from the heat source brine pump, while "average COP" and "$SCOP_{net}$" only includes the head losses. This could make the electrical energy use a little larger for the field measurements, but on the other hand "average COP" and "$SCOP_{net}$" also contain head losses for the heat sink side which SPF1 does not. The electrical energy from the heat sink pump for SPF1 is included in SPF3. Table 11 and Table 12 show the results from using the calculation models Lot 1 and prEN14825.

Table 11. The table shows the results from using Lot 1. Average COP is comparable with SPF 1 from the field measurements. Pdesign shows the maximum effect needed for the house.

Results Lot 1			
	avg COP	etas	Pdesign
site 3	3,57	1,05	7,7
site 6	3,49	1,03	7,6
site 8	3,49	1,02	5,9
site 9	3,49	1,03	7,6
site 11	3,83	1,12	7,2
site 13	3,88	1,12	5,8
site 14	3,88	1,12	5,8

Table 12. The table shows results from using prEN14825. SCOPnet is comparable with SPF 1 from the field measurements. Pdesign shows the maximum effect needed for the house.

Results prEN14825			
	SCOPon	SCOPnet	Pdesign
site 3	3,66	3,66	8,81
site 6	3,58	3,58	8,7
site 8	3,6	3,6	7,17
site 9	3,58	3,58	8,7
site 11	3,96	3,96	9,64
site 13	4,02	4,02	8,01
site 14	4,02	4,02	8,01

Since the ground source heat pumps in this study is considered comprehensive, the comparison of the results are mainly done for SPF 1 from the field measurements and SPF that corresponds to SPF 1 from the calculation models, “average COP” from Lot 1 and “SCOPnet” from prEN14825. The results are presented in the figure below.

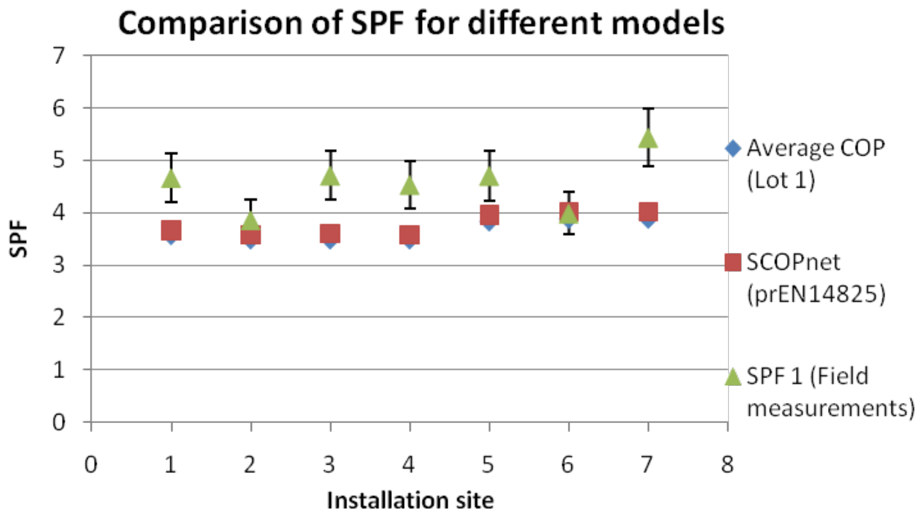


Figure 22. The figure shows a trend that SPF1 is higher compared to "average COP" and "SCOPnet". Field measurements imply a higher uncertainty compared to measurements in a laboratory. The bars of error show an error of ±10% to cover the margins of error.

There are two main differences between "average COP" and "$SCOP_{net}$":

- There are differences in degradation for part load operation

- Lot 1 does not make an effect balance of the heating demand of the house at each outdoor temperature.

The last factor results in that the design effect, P_{design}, turns out to be larger for the house when using $SCOP_{net}$ compared to "average COP". The result show that P_{design} for "average COP" is approximately 13-28% lower compared to "average COP" and SPF is approximately 3-4% lower. The degradation of COP is a little bit tougher when using Lot 1 compared to using prEN14825. The comparison is illustrated in Figure 23 below.

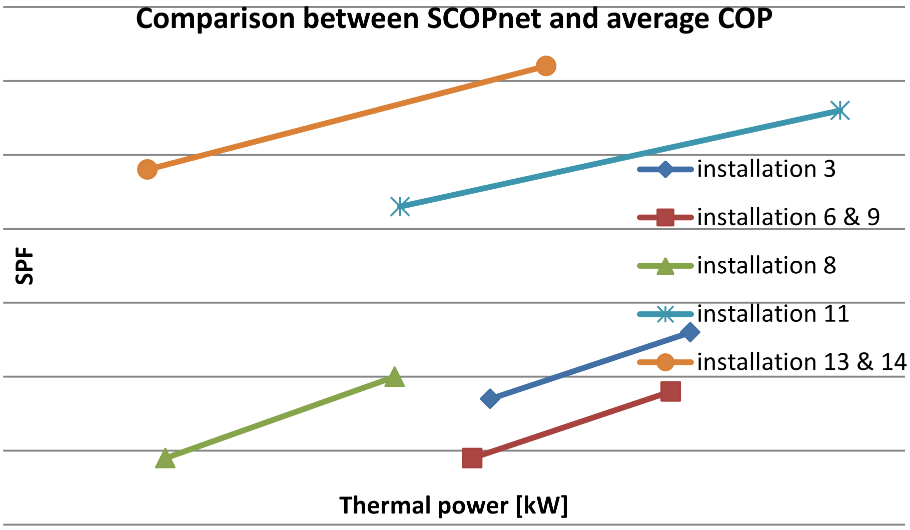


Figure 23. The figure illustrates the differences in the result when using Lot 1 and prEN14825. The lower value corresponds to Lot 1 and the higher value corresponds to prEN14825.

Air to air heat pump
Laboratory test data was available for one of the air to air heat pumps that were studied in field. SPF from the field study and SPF from the calculation models are presented in Table 13below. SCOPnet is the SPF for the heat pump that corresponds to SPF 1. SCOPon is SPF for the heat pump with the backup heater included and corresponds to SPF_{H2}. SCOP is SPF for the heat pump with both backup heater and parasitic losses included. There are some problems by comparing the laboratory test data and the data from field testing, since the field tests do not include the defrosting periods. Therefore SPF from field testing might turn out a little higher than in reality.

Table 13. The table shows a comparison of result for the air to air heat pump.

Results field measurements			
	SCOPnet	**SCOPon**	**SCOP**
prEN 14825	2,52	1,96	
Field measurement	2,4	2,1	
Lot 10		2,15	2,12

3.7 Conclusions from comparisons

Some of the field installations show different SPF_{H1} despite that the same heat pump model is installed. This can be an indication of how important the sizing of the heat pump is. An oversized heat pump results in for example more part load operation and causes standby losses.

The calculation models show that there can be benefits when installing a heat pump where the bivalent temperature is higher than the lowest operation temperature of the year, even though backup heating is necessary.

4 NEEDS FOR IMPROVED TESTING AND CALCULATION METHODS

4.1 Simulation of whole building performance including heating system

Based on test data and detailed modelling of buildings, climate etc., many countries move into the direction of simulating the heating system in a typical building. For this to work, climatic data, performance data, building material data and numerous thermophysical properties and relations are needed to perform the calculation. For a European building, the related standards that could be used for this purpose are illustrated in Figure 24.

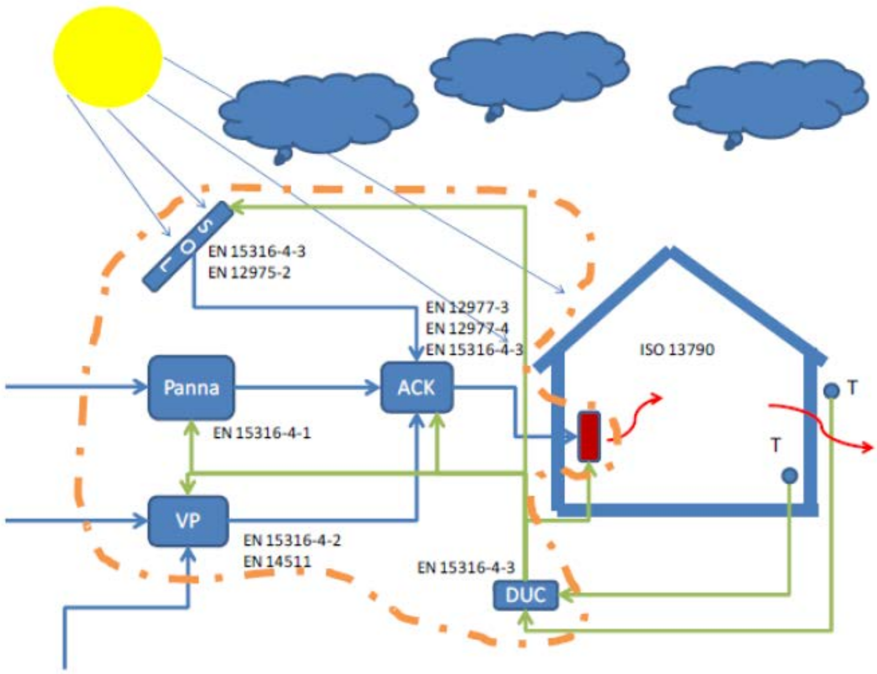


Figure 24. Modelling and simulating buildings and their integrated heating systems requires inputs from many standards.

4.2 The importance of climatic conditions

As can be seen in Figure 25 - Figure 27, many different attempts are made to establish zones of similar climatic conditions, where it is believed that the same heating requirements should appear. In Figure 26, the three climatic regions that were established in the EU are shown. Sweden belongs in this classification to cold climates (Blue colour). However, when the Swedish building council sets up requirements on the building energy performance, four different regions are used in sweden only (Figure 25). From this, it should be concluded that it is better from a requirement perspective to develop harmonised lab testing test points, and then leave it to the local policy makers to set up requirement specifications. The problem that can arise with this way of treating the requirementsetting process is that even if the products have been tested for the same test points in the lab, local requirements can trigger a larger variety of models to comply with the requirements.

Figure 25. Climatic zones in Sweden building regulations.

Figure 26. Climatic regions in EU regarding Ecodesign and Energy label requirements.

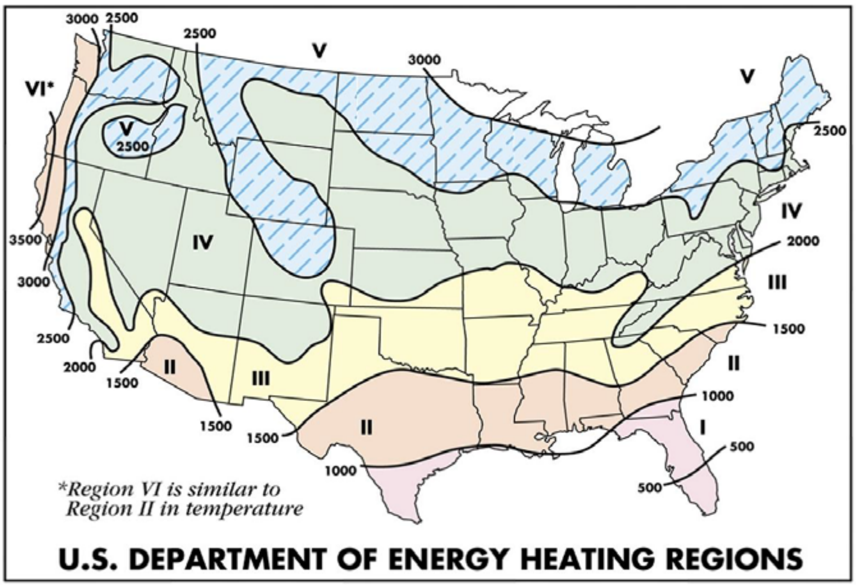


Figure 27. Climatic regions in the US used for setting local requirements on heating appliances.

4.3 Simulation coupled to the determination of SPF

In Finland, a simplified model for the calculation of SPF was developed. The simplified calculation method for heat pumps can be used for the heat pump energy calculations but not for calculating the dimensioning power of a heat pump. The electricity use of a heat pump used for heating, the space heating and warm service water heating energy production and the additional heating demand for warm service water heating and space heating can be calculated using the simplified calculation method.

The electricity use of a heat pump consists of the electricity use for heating energy production and the electricity use of the auxiliary devices. The total electricity use is calculated by means of the produced heating energy for space heating and warm service water heating, and the seasonal performance factor (SPF factor). The assumptions for the additional heating energy calculation method and calculation of the SPF exemplary values are that the lowest operation temperature of the outdoor air heat pump is -20 °C. It is also assumed, that the outdoor air heat pumps and ground source heat pumps heating both the spaces and warm service water, heat in turns either the warm service water or spaces so that the warm service water heating is the primary function.

Details of the method can be found in Appendix VI.

4.3.1 Japan Methods development

In Japan, a more detailed method was developed in this project for analyzing air conditioners in both heating and cooling operation. This method looks at the proportion of the operation at part load conditions, Figure 28. It was found in the project that operation at load from 15 to 25 % is a very large share of the total operation time. At these loads the COP is quite good.

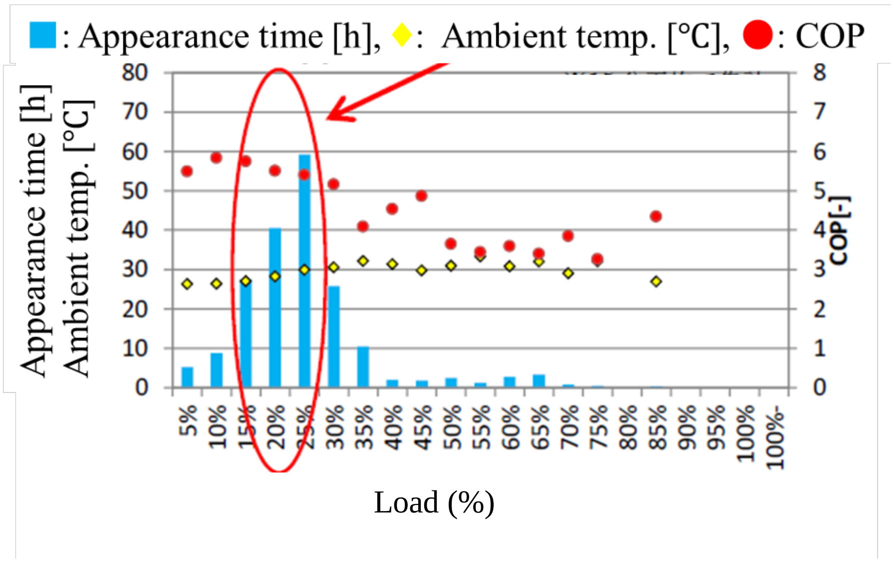


Figure 28. Appearance time of air-conditioner's load in cooling mode.

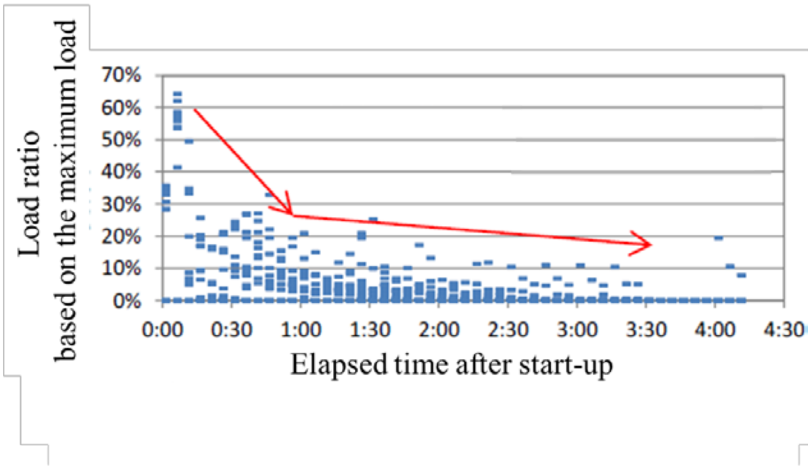


Figure 29. Air-conditioning load is very large just after start-up.

It was also found that the mean air-conditioning load is not so large during normal operation, but very large initially, Figure 29. This comes from the need of the user to quickly cool down the room after entering it. COP decrease in small load of cooling mode and heating mode respectively, Figure 30. This method is described in detail in Appendix I.

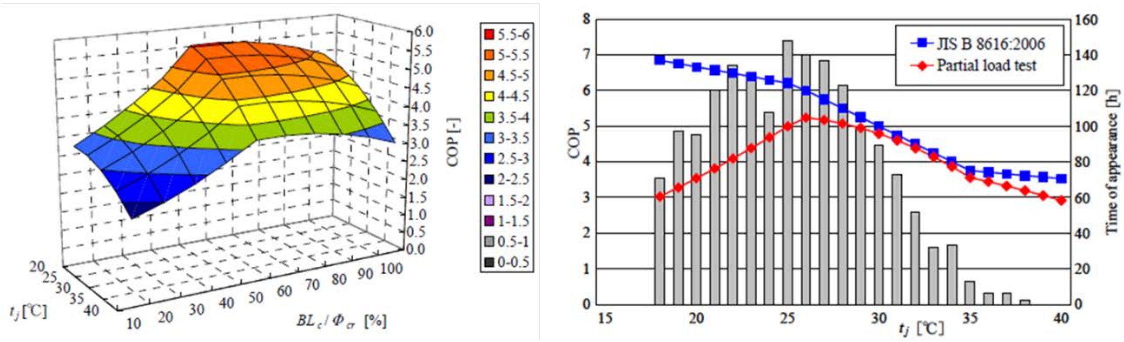


Figure 30. Cooling COP surface of tested HP and comparison of COPs in cooling season.

Reason why COP decrease in small partial load is that operation mode goes into intermittent driving. Normally, it is believed that in the air-conditioner that has inverter system, this kind of intermittent operation doesn't occur.

A newly suggested SPF calculation method for cooling mode has been developed in this project. In this method, 8 points data have to be acquired. The new model gives much better fit to data than previous JIS method, Figure 31.

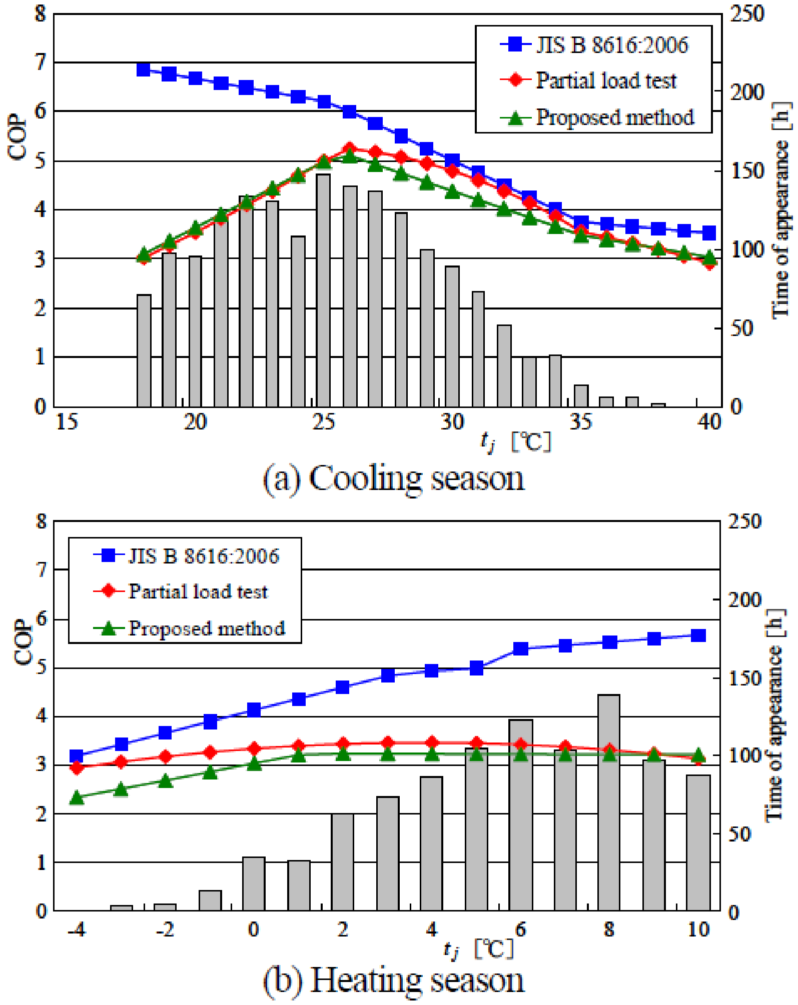


Figure 31. Comparison of COP's for office (JIS model).

4.4 Test Procedures that Accurately Reflect Energy Use and Savings for Advanced HVAC/WH Equipment

During the course of this annex, the US team had their initial focus on integrated heat pump (IHP) for space heating, water heating, space cooling, dedicated dehumidification, etc. It was concluded that account for sources of energy savings were not included in current tests, e.g., energy saved when water heating is combined with space cooling & heating. To develop a new method, ASHRAE standard project committee (SPC) 206 was formed – “Method of Test for Rating Multi-Purpose Residential Heat Pumps for Space Conditioning, Water Heating and Dehumidification”. The scope included air-to-air and water/brine-to-air heat pumps. It applies to air-source equipment rated below 19,000 W (65,000 Btu/h), and water-source, ground water-source, ground-source closed loop, and direct geoexchange equipment rated below 40,000 W (135,000 Btu/h). The standard 206 is an extension of and consistent with:

- ANSI/ASHRAE Standard 37

- ANSI/ASHRAE Standard 116 AHRI Standard 210/240
- ANSI/AHRI/ASHRAE ISO Standard 13256-1
- AHRI Standard 870
- ANSI/ASHRAE Standard 118.2
- Standard 206 should replace:
- ANSI/ASHRAE Standard 137
- and various waivers from DOEs Central Air Conditioner and Central Air Conditioning Heat Pump Test Procedure

The method was developed so that it provides for up to 7 modes of equipment operation

  - Space conditioning only
  - Dedicated water heating
  - Dehumidification+space conditioning
  - Dehumidification
  - Space conditioning+water heating
  - Dehumidification+space conditioning+water heating
  - Dehumidification+water heating

The standard method of test supports metrics for integrated appliances that allow easy comparison to metrics for individual appliances:

- For Air Source Equipment:
- SEERca; HSPFca; EFca
- For Ground or Water Source Equipment:
- EERca; COPca; EFca
- "ca" = combined appliance, referring to values for the integrated appliance, rather than the comparable metric for a standard appliance

5 NEW CALCULATION METHOD FOR SPF/ COMMONLY ACCEPTED DEFINITIONS ON HOW SPF IS CALCULATED

5.1 Requirements specification for a new or improved method

What should be included/ not included in the model?

It should be possible to decide the energy demand of the house in the model, either by given reference loads, or by choosing a specific energy demand of the house. This should be separated into space heating and domestic hot water. When the model itself calculates the losses of the house it can be misleading and not sufficient for the actual house. This can be one boundary requirement of the project.

To take into account for the climate at the installation, spot climate data, for example Meteonorm, could be a part of the model.

The dynamics of the house/building can be a part of the model. The perceived temperature of the house is not fully consistent with the actual outdoor temperature. At colder temperature dips of for example -15°C, the house will not experience the real outdoor temperature, but experiences a temperature of e.g. -12°C instead (due to internal heat gains). Even the irradiance of the sun differs between the seasons (and different spots). The energy demand of the house is affected from those variances over the year, why it might be an idea to calculate the SPF over monthly periods. For simulations, also the use of a fictive outdoor temperature would be an alternative. The climate data can be adjusted (flattened out) depending on a number of inputs, but a temperature dip is still needed in order to make a proper effect dimensioning (this is dimensioning the entire system such as deep wells etc.).

When it gets to ground source heat pumps the temperature of the ground is varying during the year. The model could include a correction for this, or have the undisturbed ground temperature as a function of time. It could also be expressed as a function where the ground source temperature is a function of the outdoor temperature over the year.

The model should contain a radiator heat curve where required supply temperature is calculated, an example of this can be found in the thesis of Fredrik Karlsson [6]. At a colder outdoor temperature, the supply temperature should peak; this makes the test scheme tables in EN 14511 deficient. Also other heat distribution systems, such as underfloor heating, heating ventilation air and mixed systems should be included in the model.

Part load performance of the heat pump must be properly taken into account, and be based on relevant testing standards.

Night set back is a choice in some calculation models; this is not relevant for heat pumps and should not be a part in the new calculation model.

Back up heaters is sometimes necessary to complete the energy demand of the house. Back up heaters should be included in the calculation model. Supplementary heating should be possible to choose between different sources of supplementary heat, e.g. electricity, solar or biomass heating.

The possibility to include the production of domestic hot water to the SPF calculations would be an advantage. It should also be described how this shall be measured in tests alternatively, how the amount of produced domestic hot water shall be estimated. Today there are two main ways how to do the measurements, including the losses or not (one can measure the amount of energy that is obtained by tappings or the amount of tap water the heat pump is producing).

Accumulators should be possible to include in the model.

A model must contain clear system boundaries for what is to be included in the calculations and how measurements are performed.

An outcome of the results should be to see that a properly sized heat pump is the best alternative to install. An oversized heat pump will result in on/off cycling losses etc.

The model must be transparent so it is possible to follow and understand the calculations. The studied models all contain parts that are more or less transparent. For example how the estimation of the number of equivalent heating hours is performed is not shown in any method.

For the calculation, either BIN methods or hour by hour calculations could be used. The existing calculation models based on heat pump performance testing according to standards are all using BIN models. Therefore, to keep a clear connection to existing test standards, it is the easiest to base a new model on BIN models.
The drawback with this approach might be that dynamic effects, especially in cases with large or well stratified accumulators are not treated in a way that the full potential of these units are revealed. Likewise, solar irradiation gains might not be treated properly.

6 ALTERNATIVE METHODS TO EVALUATE HEAT PUMP PERFORMANCE

To better compare heat pumps' benefits with other heating technologies, but also to better understand performance of heat pump, a number of other measures could be used to understand:

c. The improvement potential of heat pumps and heat pump systems
d. The competitiveness of heat pumps in environmental performance compared to other competing technologies

6.1 The improvement potential of heat pumps and heat pump systems

Seasonal performance factor, SPF is not the only measure to check that a heat pump system is working properly. In this section, some examples are chosen to illustrate this fact. Therefore, there is a need to have additional benchmarking key numbers to facilitate system performance analysis.

The following operating situations illustrate why SPF alone cannot be judged alone in order to understand heat pump performance:

- Part load operation in an oversized system
- (Full load operation in an undersized system)

Large share of backup in an undersized system
Large share of DHW to space heating (typical for low energy buildings)

By relating SPF to the Carnot efficiency, evaluation of heat pump unit performance could be made, and improvement areas could be identified where the performance deviates from Carnot "standard" performance. The Carnot efficiency is the relation between the efficiency under real conditions and the theoretically maximum reachable efficiency. The theoretical performance of heat pumps is often referred to the COP_{Carnot} (Equation 12). Together with the measured COP_{real}, ε_{Carnot} is calculated according Equation 13. This parameter enables benchmarking the system and analysing if the heat pump unit is operating properly.

$$COP_{Carnot} = \frac{T_1}{T_1 - T_2} \qquad \text{(Equation 12)}$$

$$\varepsilon_{Carnot} = \frac{COP_{real}}{COP_{Carnot}} \qquad \text{(Equation 13)}$$

Figure 32 presents the COP for an ideal heat pump as a function of temperature lift, where the temperature of the heat source is 0°C. In addition the range of actual COPs for various types and sizes of real heat pumps at different temperature lifts is shown. The Carnot-efficiency varies from 0.3 to 0.5 for small electric heat pumps and 0.5 to 0.7 for large, very efficient electric heat pump systems.

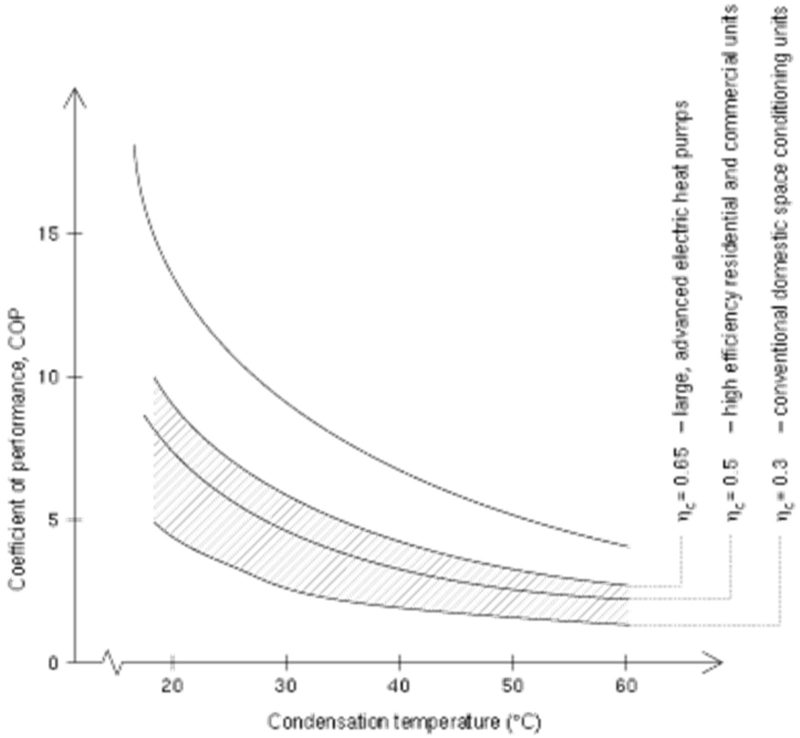


Figure 32: Carnot Efficiency [24]

One problem with relating the performance to Carnot efficiency is that the sink and source temperatures of the heat pump changes over the year, and taking an average temperature would not give good values for an annual evaluation. One way to cope with this is to use the SEI – Seasonal Energy Index, which has been developed by IOR, VDMA, and then further developed in [25]. The SEI is based on field monitoring, which makes it more complex, but it is an efficient means of increasing performance deficits. SEI answers the question how efficient the process is in a measured point. The measured value can be compared with values for other conditions. In this way SEI is a general indicator. The difference tells about the performance in the measured point according to ideal performance, other measured points or dimensioning data. It shows the potential for optimization and the quality of COP. A scale for identification of good performance in the system has been proposed, based on the measurement results.

Carbon footprint of heat pumping technology

Energy saving potential

6.2 The competitiveness of heat pumps in environmental performance compared to other competing technologies

To compare heat pumps with other heating technologies from an environmental perspective, the CO2-emissions per kWh of heating/Cooling /DHW should be calculated and compared.
This can be made by looking at the heat pump system only (building level comparison), or by including upstream emissions (global level).
Both these levels should be compared in an LCA perspective. This includes that the annual CO_2e-emissions and energy demands of the compared heating systems (heat pump, gas, oil, coal, …) and also the impact on global warming during the whole lifetime of the systems are analysed. In the EU project SEPEMO [26, 27], a model was developed, which we also refer to in this annex. The LCA is normally restricted to the use phase of the heat pump and the recovery of the refrigerant. The functional unit is chosen to be **one kWh of useful energy delivered to the building**.

Other studies have shown that the emissions related to the production of the heating equipment as well as the end of life phase is small compared to the emissions during the user phase, including the fuel production for the impact categories. Therefore the focus is on the user phase concerning the CO_2e-emissions and the primary energy (PE) demand. Additionally the TEWI method can be used for calculations of the emissions of CO_2-eqvivalents during the whole operating time of the system, including the recycling of the refrigerant at the end of life phase.

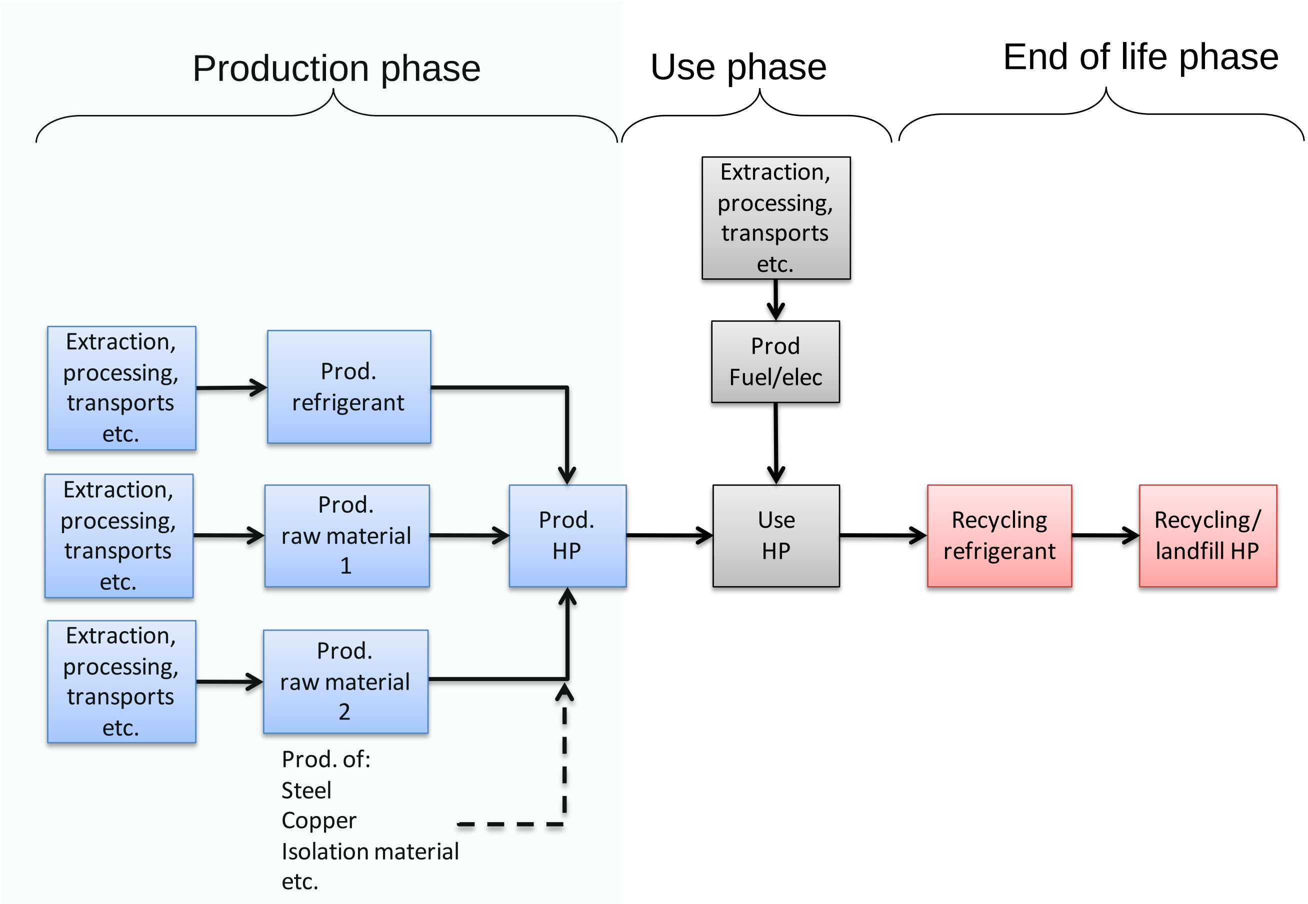


Figure 33: heat pump life cycle

Additionally to the comparison of the SPF the systems should be analysed according there primary energy efficiency, the so called primary energy ratio PER. The PER points out how much usable energy can be generated per primary energy input (Equation 14).

$$PER = \frac{\text{usable energy}\,[\text{kWh}]}{\text{primary energy}\,[\text{kWh}]} \qquad \text{(Equation 14)}$$

Figure 3 shows the different boundaries for characteristic factors for benchmarking the systems according to primary energy, final energy, usable energy, SPF, PEF and PER. In the next chapters the needed parameters for calculating the PER are described.

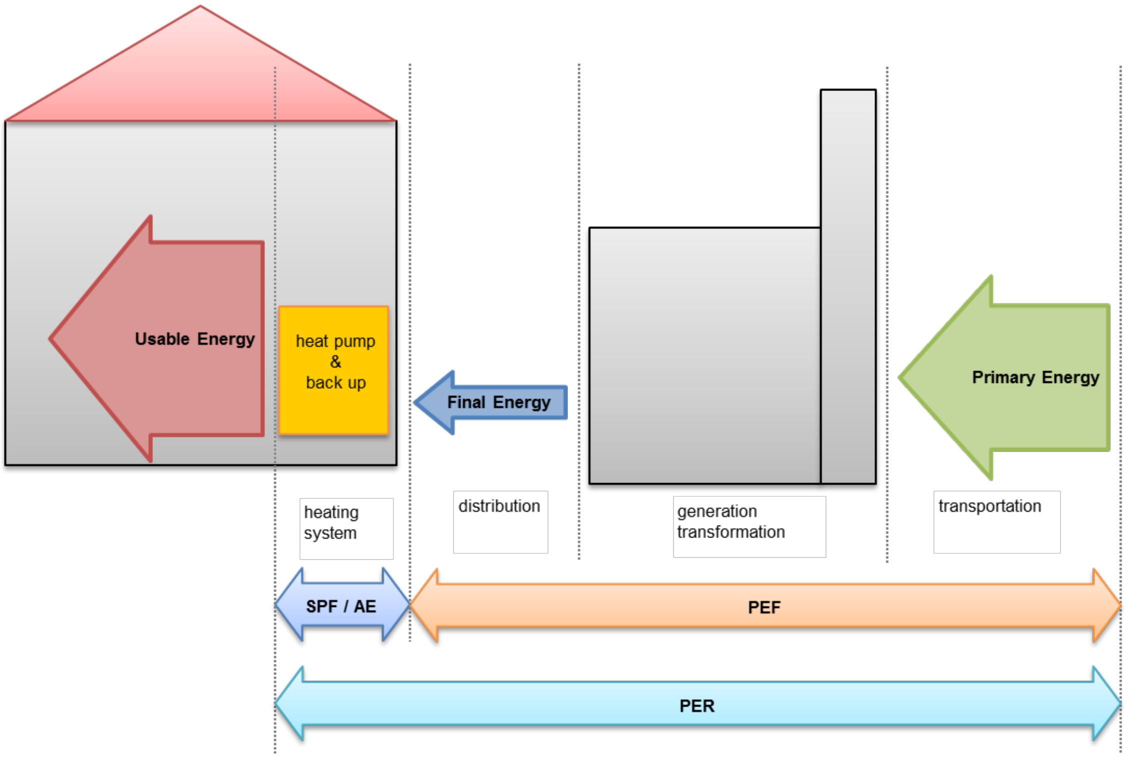


Figure 34. Boundaries for characteristic factors

6.2.1 Primary Energy demand

The annual PE demand can be calculated on the basis of the final energy and PE factor f_p of the different energy carriers of the different systems (Equation 2).

$$\text{primary energy}\,[\text{kWh}] = \text{final energy}\,[\text{kWh}] * f_p \,\frac{[\text{kWh}]}{[\text{kWh}]} \qquad \text{(Equation 15)}$$

6.2.2 Final Energy

The final energy consumption of the different heating systems will be calculated based on the measured usable energy and the different efficiencies of the systems. For the backup heating systems the annual efficiency (AE) will be taken for the calculation (Equation 16).

$$\text{final energy}_{\text{back up systems}}\,[\text{kWh}] = \frac{\text{usable energy}\,[\text{kWh}]}{\text{AE}\,[\text{-}]} \qquad \text{(Equation 16)}$$

The final energy for heat pumps will be calculated with the seasonal performance factor SPF (Equation 17). As the SPF is >1 the final energy demand will be smaller than the usable energy demand.

$$\text{final energy}_{\text{heat pumps}}\,[\text{kWh}] = \frac{\text{usable energy}\,[\text{kWh}]}{\text{SPF}\,[\text{-}]} \qquad \text{(Equation 17)}$$

6.2.3 Primary energy factor, f_p

The PE factor f_p is defined as PE demand per final energy (Equation 18).

$$f_p = \frac{\text{primary energy [kWh]}}{\text{final energy [kWh]}} \qquad \text{(Equation 18).}$$

According to EN 15603:2008 [28] the PE is described as energy that has not been subjected to any conversion or transformation process. The PE includes non-renewable energy and renewable energy. If both are taken into account it can be called total PE [28]. For the calculation according [26] the PEF_T will be used. Table 14 shows the PEF_T and PEF_R of the selected energy carriers for calculating the PE demand of the different heating systems. Since the EN 15603 data are relatively old, we have used the data from Eurostat, also shown in Table 14.

Table 14. PE factors [28, 29].

	Primary energy factor		
	Non-renewable (Resource)	**Total, EN15603**	**EUROSTAT**
Fuel oil	1,35	1,35	1
Natural gas	1,36	1,36	1
Anthracite	1,19	1,19	1
Electricity Mix UCPTE	3,14	3,31	-
Electricity Mix (EU)	3,31	2,65	2,2

One reason that emission data from different sources differ are the use of different system boundaries. It is a problem that the process many times is treated as a black box which makes it difficult for the user to know what sub processes, efficiency values, allocations etc. the data set is based on.

6.2.4 Heat pump energy supply ratio

For bivalent heat pump systems the heat pump energy supply ratio HPESR is calculated according to Equation 19**Fel! Hittar inte referenskälla.**, to show how much of the usable energy is supplied by heat pump and the backup heater.

$$\text{HPESR}\,[\%] = \frac{\text{usable energy}_{\text{hp}}\ \text{[kWh]}}{\text{usable energy}_{\text{hp}} + \text{usable energy}_{\text{bu}}\ \text{[kWh]}} \qquad \text{(Equation 19).}$$

7 IMPROVING THE CUSTOMER UNDERSTANDING OF FACTORS AFFECTING SPF

An important part of marketing is to show good values of COP or SPF. For the end consumer, the other factors we have described in Chapter 6 could be just as important, to show that the product in fact is a means to reduce CO2 emissions and primary energy use. It is also important to inform the user about how space heating and DHW production affects the heat pump performance and annual cost of heating. Fraunhofer made a simple but yet very illustrative Excel model, where different buildings can be compared, and where slide bars makes it easy for the end user to vary different parameters and see how annual cost is affected. This model is a very good example on how to inform and simultaneously educate the end user.

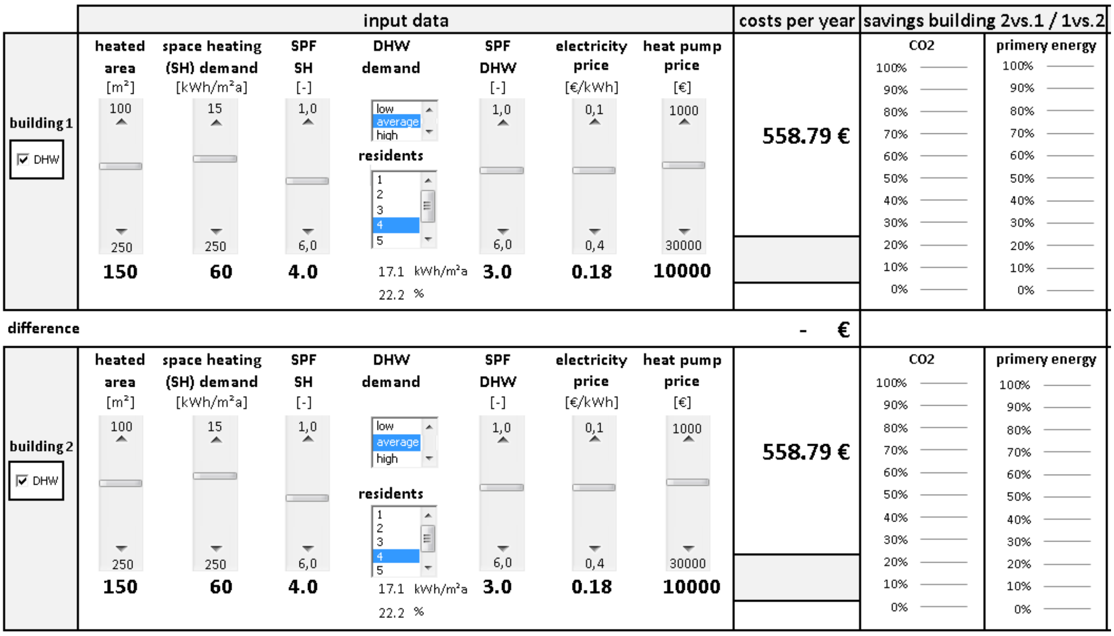


Figure 35. Excel model developed by Fraunhofer ISE to compare the factors affecting heat pump system cost and environmental performance.

8 CONCLUSIONS AND RECOMMENDATIONS

This annex give proposals for harmonizing test standards, but also extends to give suggestions for building test chambers in an similar way, and propose alternative measures to describe the technical, environmental and financial performance of heat pumps. Much work was carried out in the separate national teams, and the results were presented in workshops. The conclusions from the results are summarized in bullets below, but in order to gain more insight, it is recommended that the conclusions of the national reports are read as well.

- The difference in test points in different regions doesn't differ a lot, why there is the possibility to harmonise many test points. By harmonising the test points, the road is open to come closer with the calculations and certifications that are based on these test points.
- Harmonisation should be made to test points, so that a similar set of test points are tested in the test labs. There must be room for local (national, regional) variations, especially regarding climatic conditions and building demand profiles. Therefore a matrix of test conditions could include the necessary test points, and voluntary test points that should need to be tested for certain markets (e.g. in cold climates, one -15°C point should be included).
- Harmonisation of test standards should happen on ISO level, since this is the global forum for standardisation. Regional/national standards should align with the ISO standards when they are published.
- Timing between revisions of standards is a threshold to harmonisation. Ideally, an agreement should be made between standardisation organisations to make revisions e.g. every five years with a limited revision time, with possibility to harmonise standards at every revision.
- We have reached a conclusion that harmonization of the standards in respective countries is difficult. Even so, we believe that we will be able to create annual performance evaluation standards that seem to be uniform as far as possible.
- Even though this annex have found many possibilities to harmonize standards, we have concluded that a number of new calculation and simulation methods have been developed during this project, which is moving in the opposite direction of the thoughts of this annex.
- As simulation becomes more and more accepted to define building integrated heating performance, there should be very transparent models for both buildings, heating systems and with regards to climatic data. Very clear

operating ranges for different relations should be defined etc. There is otherwise the possibility that the final performance numbers are compromised by uncertainties in simulations models.

- To promote heat pump simulation, one IEA HPT annex could be performed, developing a library of annotated and accepted heat pump and building models.
- The IEA HPT could from this annex develop a set of calculation templates for evaluating other performance metrics but SPF, both for installers and for end users. These templates should be Final energy use, Primary energy consumption, CO2 emissions reduction and Cost performance . This makes it much more clear to end consumers to understand the financial and environmental consequences when installing a heat pump

9 REFERENCES

1. Seasonal performance rating of heat pump water heaters, G.L. Morrison et al. / Solar Energy 76 (2004) 147–152

2. Schibuola L, 2000. Heat pump seasonal performance evaluation: a proposal for a European Standard. Applied Thermal Energy 20, 387–398.

3. PARAMETER ESTIMATION BASED MODELS OF WATER SOURCE HEAT PUMPS, HUI JIN, Ph.D. Thesis, Oklahoma State University, December 2002

4. Sherwood G. Talbert, William G. Atterbury, Thomas A. Klausing, and Frank E. Jakob, Relevance of Existing Heat Pump Testing and Rating Methods to Specifying a Method of Test for Engine-Driven Heat Pumps, ASHRAE Project 780-RP, Technical Committee 7.4: Unitary Combustion-Engine Driven Heat Pumps, June 1996

5. Ekonomisk och driftserfarenhetsmässig utvärdering av bergvärmepumpar, Utarbetad av Per Levin, Projektengagemang Energi och klimatanalys Danderyd, februari, 2008

6. Capacity Control of Residential Heat Pump Heating Systems, Karlsson, F., Ph.D. Thesis, Chalmers university of Technology, Sweden, 2007.

7. TWO LARGE FIELD-TESTS OF NEW HEAT PUMPS IN GERMANY, Miara, M. 9th IEA heat pump conference, Zurich, 2008

8. Årsmätning på fem bergvärmeanläggningar i Sjuhärad, Energimyndigheten, 2005

9. CEN 2000. EN12309-2:2000 Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW – Part 2: Rational use of energy, CEN, Brussels.

10. DIN, 1988, DIN 33830-4:1988 Wärmepumpen; Anschlußfertige Heiz-Absorptionswärmepumpen; Leistungs- und Funktionsprüfung, DIN, Berlin

11. JSA 2011, JIS B 8622:2009 Absorption refrigerating machines. JSA, Tokyo

12. CEN, 2011, EN 14511:2011 Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling, CEN, Brussels.

13. ANSI, 2009, ANSI/ASHRAE Standard 37-2009 Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment. ANSI, New York

14. AHRI, 1998, AHRI Standard 320-98 Water-Source Heat Pumps. AHRI, Arlington

15. AHRI, 1998, AHRI Standard 325-98 Ground Water-Source Heat Pumps. AHRI, Arlington

16. AHRI, 1998, AHRI Standard 330-98 Ground Source Closed-Loop Heat Pumps. AHRI, Arlington

17. CEN, 2011, EN14825:2011 Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling – Testing and rating at part load conditions and calculation of seasonal performance, CEN, Brussels.

18. CEN, 2011, EN 16147:2011 Heat pumps with electrically driven compressors - Testing and requirements for marking for domestic hot water units. CEN, Brussels

19. AHRI, 2000, ANSI/ARI 560: Standard for Absorption Water Chilling and Water Heating Packages. AHRI, Arlington

20. ANSI, 2008, ANSI/ASHRAE 182-2008: Method of Testing Absorption Water-Chilling and Water-Heating Packages. ANSI, New York

21. VDI, 2003, VDI 4650-1: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water. VDI, Düsseldorf

22. VDI, 2010, VDI 4650-2: Simplified method for the calculation of the annual coefficient of performance and the annual utilisation ratio of sorption heat pumps - Gas heat pumps for space heating and domestic hot water. VDI, Düsseldorf

23. CEN, 2008, EN15316-4-2:2008 Heating systems in buildings – Method for calculation of sys-tem energy requirements and system efficiencies – Part 4-2: Space heating generation systems, heat pump systems, CEN, Brussels

24. http://www.heatpumpcentre.org/en/aboutheatpumps/heatpumpperformance/Sidor/default.aspx

25. Method and guidelines to establish Sys-tem Efficiency Index during field measurements on air conditioning and heat pump systems, Lane, A-L., Benson, J., Eriksson, L., Fahlén, P., Nordman, R., Haglund Stignor, C., Berglöf, K., Hundy, G., Effsys plus project EP18, Energimyndigheten, 2014. http://effsysplus.se/wp-content/uploads/2012/02/EP18-Slutrapport-20140630rev0704.pdf

26. Evaluation method for comparison of heat pump systems with conventional heating systems - Concept for evaluation of CO2-reduction potential, Andreas Zottl, Markus Lindahl, Roger Nordman, Philippe Rivière, Marek Miara, Deliverable 4.3 of the Sepemo Build project, 2011

27. Benchmarking method of seasonal performance under consideration of boundary conditions., Zottl, A.,Nordman, R., Miara, M., Deliverable 4.4 of the Sepemo Build project, April 2012

28. EN 15603, 2008: Energy performance of buildings — Overall energy use and definition of energy ratings, 2008
29. GEMIS (Global Emission Model for Integrated Systems) v 4.7, Oeko-Institut

10 APPENDICES

10.1 Appendix I – National report Japan

Annex 39 Japan final report

Part1

1. SCOPE OF JAPAN NATIONAL TEAM CONTRIBUTION

2. INTRODUCTION OF SYSTEM SIMULATION TOOL "LCEM"

3. METHODOLOGY OF HP CYCLE PERFORMANCE ANALYSIS

4. COMPARISON OF THE YEAR-ROUND PERFORAMANCE FACTORS IN THE WORLD

5. CURRENT STATUS OF MEASUREMENT TECHNOLOGY

6. TECHNICAL PAPER OF “NEW CALCULATION METHOD OF ANNUAL PERFORMANCE FACTORS FOR VRF SYSTEMS”

Part2

7. FIELD EXPERIMENTAL EVALUATION OF AIR-CONDITIONING SYSTEM PERFORMANCE RESULTS REPORT (SUMMARY)

8. OUTLINE OF THE BUILDING ENERGY STANDARD IN JAPAN AND EVALUATION OF HEAT PUMPS IN THE STANDARD

Preface

Part I

Part I summarizes the views about the possibility of global harmonization of annual performance evaluation standards for air conditioners. As the annual performance evaluation standards for air conditioners are used for evaluation of many air conditioning products, it is not practical to require manufacturers to acquire a lot of data without any reason. Accordingly, we have no other choice but to rely on the data acquired through the minimum combination of outdoor and indoor temperatures for annual performance prediction. Also, in making measurement, the performance of air conditioning equipment must be evaluated in an extremely accurate manner.

In spite of the above, in addition to the differences in the calculation methods for performance evaluation among the standards of Japan, Europe, the United States and other regions, there exist differences in accuracy of measuring equipment and the required conditions for combination of measurement temperatures. Therefore, as Japan national team, we have reached a conclusion that harmonization of the standards in respective countries is difficult. Even so, we believe that we will be able to create annual performance evaluation standards that seem to be uniform as far as possible. Since Japan's Annual Performance Factors (APFs) have a rich track record of air conditioning equipment evaluation with assured high accuracy, we explain that it is the most appropriate method to be employed by the respective countries.

In this report, Chapters 4 and 5 of Part I analyze and compare the evaluation methods of the annual performance evaluation standards for air conditioners in Japan, Europe and the United States. Also, the chapters give descriptions on quantitative assessment methods of uncertainties in measurement of air conditioning equipment, that were put under review in Japan, and the highly accurate performance measurement equipment at present and their future. By this, they indicate Japan's APFs have a rich track record of air conditioning equipment evaluation with assured the highest accuracy.

Moreover, in annual performance evaluation of multiple packaged air conditioning systems for use in buildings, Chapter 6 discusses the newly suggested idea of evaluation method that is capable of greatly increasing accuracy by adding measurement points.

Also, about the simulation techniques for heat pumps, Chapter 2 introduces Life Cycle Energy Management (LCEM) tool that combines heating, ventilation and air conditioning (HVAC) performance curve with building envelope characteristics, room heat load and outside temperature, while Chapter 3 outlines a highly accurate HVAC cycle performance simulation that coincides well with the actual measurement results.

Part II

Part I discusses the methods to evaluate annual performance of air conditioning equipment based on the annual performance evaluation standards. As a matter of course, all of the installed air conditioning equipment operate throughout the year in a way different from that assumed in the annual performance evaluation standards and the thermal load patterns are respectively different. In evaluation of such systems, the manufacturers' highly accurate, standards-based product evaluation methods differ greatly from the user-side on-site evaluation methods of actually installed air conditioning equipment systems. Therefore Japan national team has reached a conclusion that harmonization of the standards is almost impossible. These user-side methods which are employed primarily by the end-users or contractors should be easy to use and be able to take installation conditions into consideration. On the other hand these methods have problem of errors often caused by on-site poor measurement or rough estimation of air conditioning equipment performance. While looking at the current moves toward adoption of smart energy systems and wider use of many types of renewable energy heat sources, associated with introduction of various types of air conditioning equipment, we anticipate a problem that an appropriate evaluation of actually installed air conditioning equipment becomes difficult in case that each type of air conditioning equipment employs a different performance evaluation method.

Consequently, Japan national team considers that it is necessary to evaluate to some extent the user-side methods for actually installed air conditioning equipment systems. Therefore, in Part II, Chapter 7 describes the actual loads, operational conditions, etc. based on the field tests of air conditioning equipment in Japan. Chapter 8 explains the performance evaluation methods proposed in our country, which are established primarily by the users for performance evaluation of actually installed air conditioning equipment systems.

1 SCOPE OF JAPAN NATIONAL TEAM CONTRIBUTION

Methods to measure annual performance factors (APFs) for heat pump equipment in Japan are specified under the following Japanese Industrial Standards (JIS):

1) JIS C 9612 : Room air conditioners
2) JIS B 8616 : Package air conditioners
3) JIS C 9220 : Residential heat pump water heaters

Japanese air conditioner manufacturers currently use APFs to evaluate the fundamental performance of their products.

As the scope of Annex 39 covers single- and multiple-family buildings, Japan focuses on air conditioners with rated cooling capacities of 10 kW or less under JIS C 9612 room air conditioners, which are categorized as air/air (heat pump class) and heating and cooling (operating mode) in Table 1.1. This report deals with capacity-controlled air conditioners. Although water heater APFs are specified under JIS C 9220, they are excluded from the Japan's scope as they are calculated by using loads that do not comply with international standards.
APF indexes in this report are widely used in Japan for labeling and the Top Runner Program implemented by the Japanese government to promote energy conservation efforts.

Table 1.1 Japan's Scope

Type of heat pump	Operating mode				
	Heating	Domestic hot water	Heating & domestic hot water	**Heating & cooling**	Heating, cooling & domestic hot water
Air/air	x			○	x
Air/water	x	x	x	x	x
Brine/water	x	x	x	x	x
Water/water	x	x	x	x	x
DX/DX	x	x	x	x	x
DX/water	x	x	x	x	x

Japan national team focuses on Air / air system, however, there are a variety of heat pump systems in Japanese market. The below table shows typical examples of multi-purpose heat pump and hybrid energy source heat pump which are available in Japanese market. It seems considerably difficult to harmonize the standards for these heat pumps.

Table 1.2 Varieties of heat pump systems in Japanese market

Type	Manufacturer
Domestic hot water supply combined with Floor Heating	Hitachi Appliances
Domestic hot water supply combined with Floor Heating	Panasonic Corporation
Air/Water Space Heating	Toshiba Carrier Corporation
Air/Water Space Heating	Mitsubishi Electric
Floor Heating	Daikin
Heat pumps combined with Solar heating system	Yazaki Corporation
Heat pumps combined with gas furnace system	Rinnai Corporation

2 INTRODUCTION OF SYSTEM SIMULATION TOOL "LCEM"

This section introduces a system simulation tool "LCEM" and examples of simulation using the tool. The LCEM tool was developed by Government Building Services of the Ministry of Land, Infrastructure, Transport and Tourism (MLIT) as a lifecycle energy management support tool, and is available to the public for free.
This spreadsheet program-based tool is user-friendly and can easily be handled by practitioners. The tool can be used to model a total HVAC system including the primary and secondary sides, and allows simulation of annual energy consumption with system efficiency under partial load conditions taken into account. The tool uses an object cell system that allows users to set the system boundaries as they like.

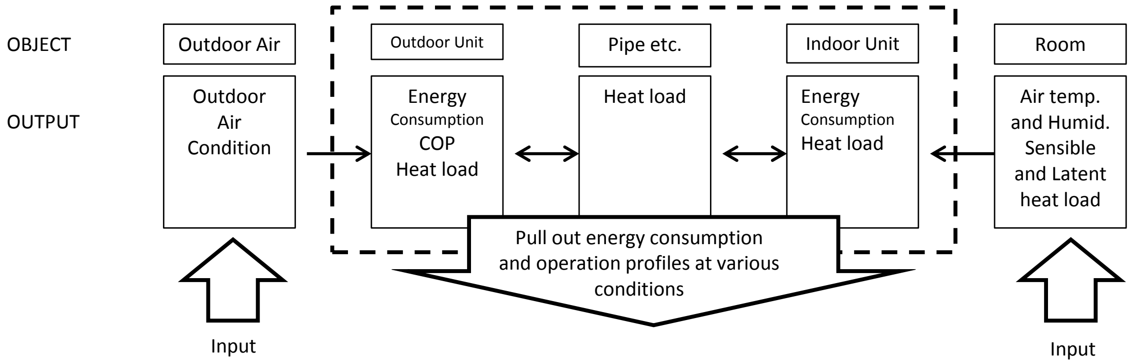


Figure 2.1 LCEM tool concept (example of multi-split type air-conditiong system) [1]

Figure 2.1 shows the concept of the LCEM tool for a multi-split type air-conditiong system as an example. As shown in the figure, the user can place parts called "objects" in a line to model the system. Each object stores the information related to a component of the system and associated arithmetic expressions to calculate the energy consumption. From the boundary conditions received, the objects will calculate the energy consumption.

Taking the multi-split type air-conditiong system in the figure as an example, the following outlines the flow of object simulation. The indoor unit object calculates the thermal load and the required fan power from the indoor thermal load and fresh air load. The outdoor unit object calculates the power consumption from the outdoor air condition, indoor unit (coil) inlet air condition and partial load characteristics based on the indoor unit thermal load. Furthermore, the outdoor unit capacity is modified with the capacity

[1] The indoor unit objects in the figure represent four objects: indoor unit (coil), indoor unit (fan), room and outside air load objects.

correction value calculated by the refrigerant piping object. The capacity correction value is determined by taking the piping length and the difference in equipment height into account.

The following shows an example of simulation of a multi-split type air-conditiong system using the LCEM tool. As for the typical floor of an office building in Tokyo, the indoor load was assumed as shown in Figure 2.2. The HVAC system components are listed in Table 2.1. The system consists of one outdoor unit with a cooling capacity of 28 kW and seven indoor units making up three lines. The outdoor unit has performance characteristics as shown in Figure 2.3.

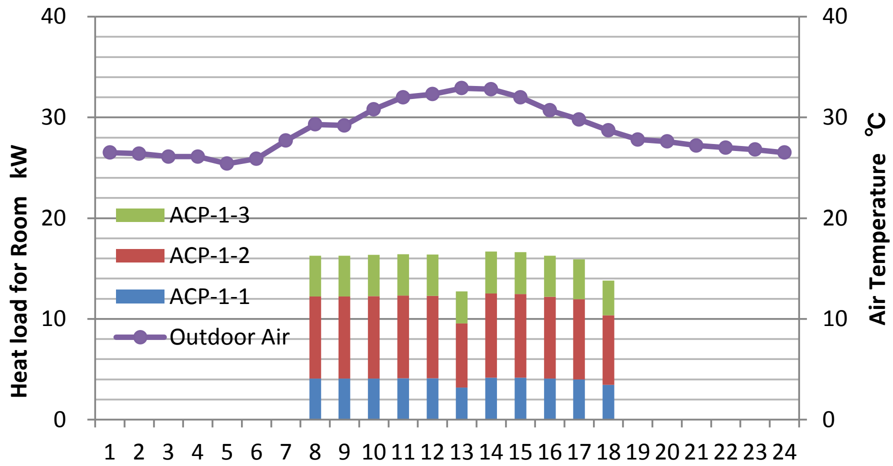


Figure 2.2 Indoor load on the typical floor of a hypothetical building (during a typical day of August)

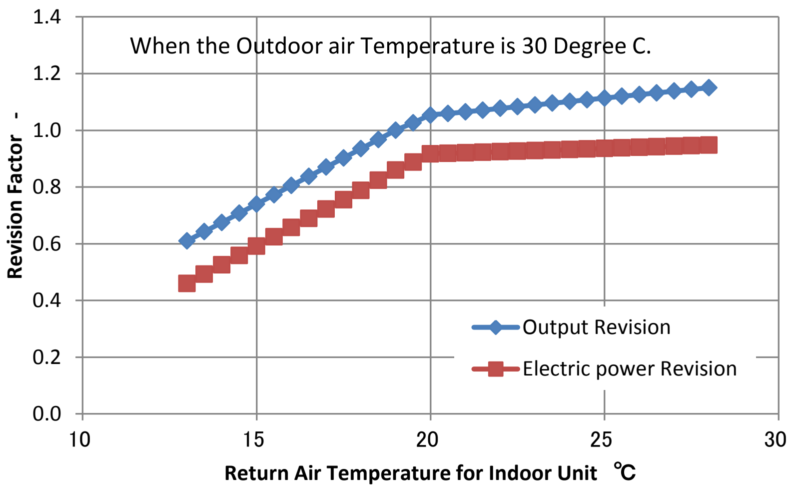


Figure 2.3 Performance characteristics of outdoor unit

Table 2.1 List of target HVAC system components

Symbol	Component name	Specifications	Phase-voltage	Power consumption [kW]	Number of units
ACP-1	Outdoor unit	Cooling/heating capacity: 28 kW/31.5 kW	3-200	7.64/8.45	1
ACP-1-1	Indoor unit	Ceiling concealed duct type Air volume: 1,650 m^3/h Cooling/heating capacity: 11.2 kW/12.5 kW	1-200	0.34	1
ACP-1-2	Indoor unit	Ceiling concealed duct type Air volume: 480 m^3/h Cooling/heating capacity: 2.8 kW/3.2 kW	1-200	0.03	4
ACP-1-3	Indoor unit	Ceiling concealed duct type Air volume: 480 m^3/h Cooling/heating capacity: 2.8 kW/3.2 kW	1-200	0.03	2

Figure 2.4 shows the thermal load and supply air temperature of the indoor unit on the ACP-1-1 line. The figure also shows the thermostat on time for the indoor unit. The air was supplied at a fixed temperature of 11℃. As a result, a higher thermal load than the indoor load was processed. This is because of the extra cooling of latent heat, which was required to keep supply-air temperature as low as 11℃. When the ACP-1-1 indoor unit with a rated air-conditioning capacity of 11.2 kW was applied with an indoor air thermal load of around 4 kW (partial load), the daily mean thermostat on time was 22 minutes per hour.

Figure 2.5 shows the changes in output, capacity and load factor of the outdoor unit with time. According to the figure, the maximum capacity of the outdoor unit after correction for the outside air condition and indoor unit suction temperature is different from the rated cooling capacity of 28 kW. The daily mean load factor (ratio of output to maximum capacity) is 0.86.

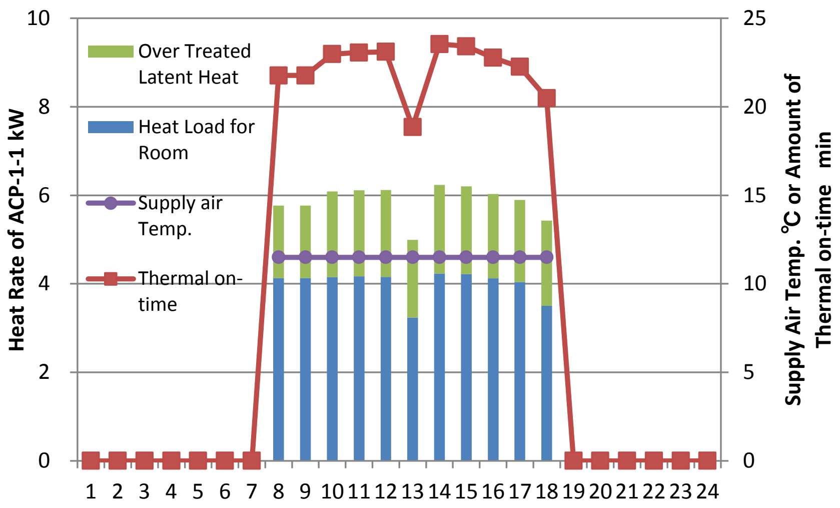


Figure 2.4

Changes in thermal load and supply air temperature of ACP-1-1 indoor unit with time

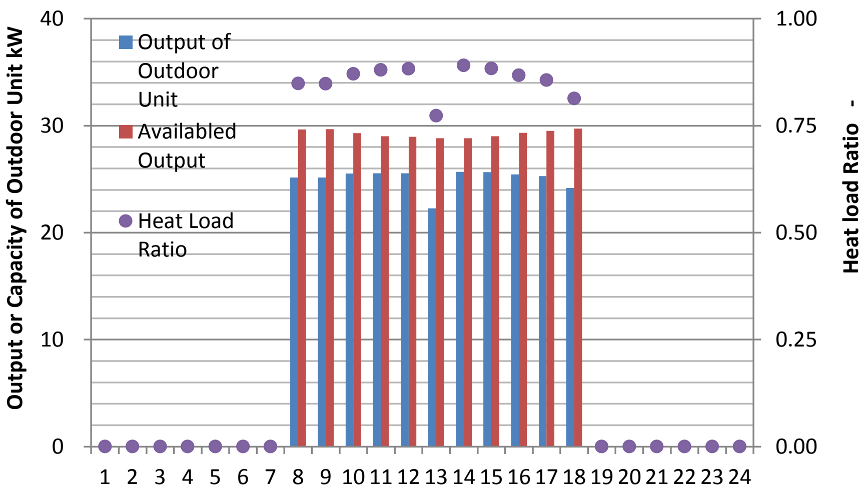


Figure 2.5 Changes in output, capacity and load factor of outdoor unit

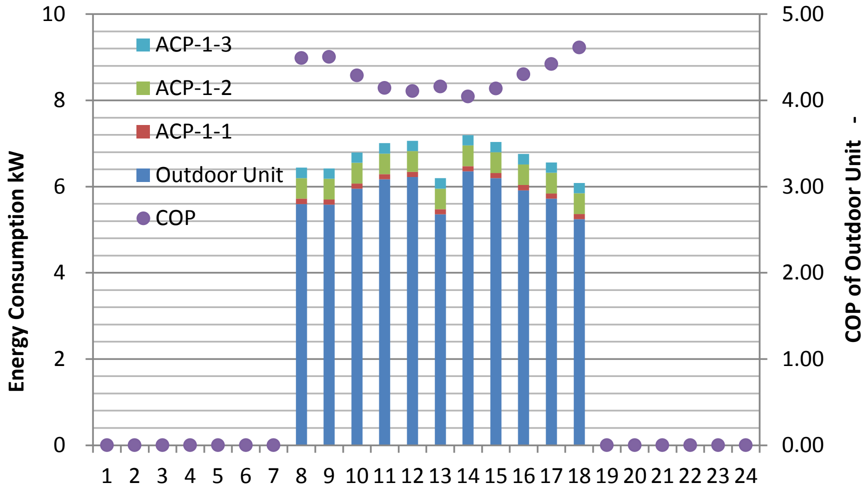


Figure 2.6 Power consumption of air-conditioning system and COP of outdoor unit

Figure 2.6 shows how power consumption of the air-conditioning system and COP of outdoor unit change with time. The power consumption of indoor unit indicates the power consumed to run the indoor unit fans. These fans are controlled to provide a constant air volume and kept running to deliver air even when the thermostat is off. The COP of outdoor unit is 4.29 on a daily average basis while the rating is 3.66.

This completes the description of the LCEM tool and an example of simulation of the multi-split type air-conditiong system with the tool. The example has revealed that the LCEM tool can be used to simulate annual operation of air-conditioning systems with efficiency under partial load conditions taken into account.

We will continue using the LCEM tool for ongoing studies.

3 METHODOLOGY OF HP CYCLE PERFORMANCE ANALYSIS

3.1 Introduction

In this chapter, we introduce a simulator which is capable of evaluating the performance of compression-type heat pumps in detail. 'Energy Flow + M' is a simulator that is based on modular analysis theory. With this simulator, users can calculate the steady-state and unsteady-state performance of compression-type heat pumps without using complex calculation methods or mathematical models.

This simulator can analyze the performance of compression-type heat pumps as well as many energy systems such as adsorption heat pumps, dehumidifiers, fuel cells and solar thermal collectors. It can also analyze the performance of detailed elements such as heat exchangers as well as large energy systems that combine several energy systems.

3.2 Mathematical model for compression-type heat pump

As an example, let us introduce a mathematical model for the compression-type heat pump we will use for calculation. This system consists of a compressor, heat exchanger, expansion valve and accumulator. Figure 3.1 shows a schematic model of each element.

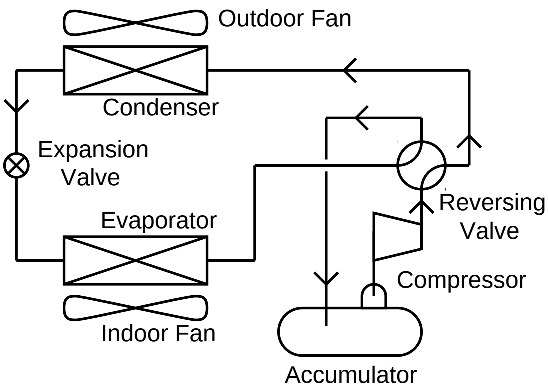


Figure 3.1 Cycle of compression-type heat pump

3.2.1 Compressor model

We consider the compressor model, a steady-state model, because the time delay is very small. The model for the compressor is shown in Figure 3.2.

The continuity equation is as follows:

$$\rho_{R,O} v_{R,O} S_{R,O} - \rho_{R,I} v_{R,I} S_{R,I} = 0 \tag{1}$$

The energy equation is as follows:

$$\rho_{R,O} v_{R,O} h_{R,O} S_{R,O} - \rho_{R,I} v_{R,I} h_{R,I} S_{R,I} = W \tag{2}$$

The isentropic efficiency is calculated as follows:

$$\eta_{\mathrm{ad}} = \frac{h_{\mathrm{R,O,ad}} - h_{\mathrm{R,I}}}{h_{\mathrm{R,O}} - h_{\mathrm{R,I}}} \tag{3}$$

$$s_{\mathrm{R,O,ad}} = s_{\mathrm{R,I}} \,. \tag{4}$$

The volumetric efficiency is calculated as follows:

$$\rho_{\mathrm{R,I}} v_{\mathrm{R,I}} S_{\mathrm{R,I}} = \frac{\omega}{2\pi} \rho_{\mathrm{R,I}} \eta_{\mathrm{V}} V \tag{5}$$

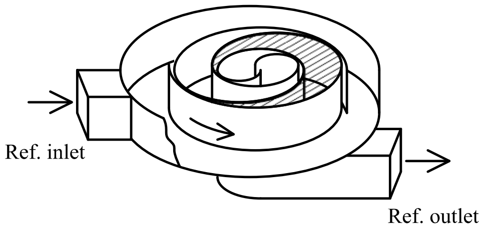


Figure 3.2 Compressor model

3.2.2 Heat exchanger model

A model for the heat exchanger is shown in Figure 3.3.

(1) Refrigerant side

The continuity equation is as follows:

$$\frac{\partial \rho_{\mathrm{R}}}{\partial t} + \frac{\partial \left(\rho_{\mathrm{R}} v_{\mathrm{R}} \right)}{\partial z} = 0 \tag{6}$$

The pressure drop is calculated as follows:

$$\frac{\partial P_{\mathrm{R}}}{\partial z} = -f_{\mathrm{R}} \frac{2 \rho_{\mathrm{R}} v_{\mathrm{R}}^2}{D_{\mathrm{In}}} \tag{7}$$

The energy equation is as follows:

$$\frac{\partial \left(\rho_{\mathrm{R}} u_{\mathrm{R}} \right)}{\partial t} + \frac{\partial \left(\rho_{\mathrm{R}} v_{\mathrm{R}} h_{\mathrm{R}} \right)}{\partial z} = -\frac{L c_{\mathrm{In}}}{S_{\mathrm{In}}} q_{\mathrm{In}} \tag{8}$$

The energy equation of the pipe is as follows:

$$\rho_{\mathrm{T}} C_{\mathrm{T}} \frac{\partial T_{\mathrm{T}}}{\partial t} = \frac{L c_{\mathrm{In}}}{S_{\mathrm{T}}} q_{\mathrm{In}} - \frac{A_{\mathrm{FC}} + \eta_{\mathrm{FIN}} A_{\mathrm{FIN}}}{S_{\mathrm{T}} L} \left(q_{\mathrm{Out}} + j_{\mathrm{Out}} h_{\mathrm{VAP}} \right) \tag{9}$$

(2) Air side

The continuity equation of the whole air is as follows:

$$\rho_{\mathrm{A,O}} v_{\mathrm{A,O}} L_{\mathrm{A}} - \rho_{\mathrm{A,I}} v_{\mathrm{A,I}} L_{\mathrm{A}} = \frac{A_{\mathrm{FC}} + \eta_{\mathrm{FIN}} A_{\mathrm{FIN}}}{L} j_{\mathrm{Out}} \tag{10}$$

The continuity equation of water is as follows:

$$\rho_{\mathrm{A,O}} v_{\mathrm{A,O}} X_{\mathrm{A,O}} L_{\mathrm{A}} - \rho_{\mathrm{A,I}} v_{\mathrm{A,I}} X_{\mathrm{A,I}} L_{\mathrm{A}} = \frac{A_{\mathrm{FC}} + \eta_{\mathrm{FIN}} A_{\mathrm{FIN}}}{L} j_{\mathrm{Out}} \tag{11}$$

The pressure drop is ignored:

$$P_{\mathrm{A,O}} - P_{\mathrm{A,I}} = 0 \tag{12}$$

The energy equation is as follows:

$$\rho_{\mathrm{A,O}} v_{\mathrm{A,O}} h_{\mathrm{A,O}} L_{\mathrm{A}} - \rho_{\mathrm{A,I}} v_{\mathrm{A,I}} h_{\mathrm{A,I}} L_{\mathrm{A}} = \frac{A_{\mathrm{FC}} + \eta_{\mathrm{FIN}} A_{\mathrm{FIN}}}{L} \left(q_{\mathrm{Out}} + j_{\mathrm{Out}} h_{\mathrm{VAP}} \right) \tag{13}$$

$$j_{\mathrm{Out}} = \begin{cases} 0 & T_{\mathrm{DP,A,O}} < T_{\mathrm{A,O}} \\ f\left(P_{\mathrm{A,O}}, h_{\mathrm{A,O}}, T_{\mathrm{DP,A,O}}\right) & T_{\mathrm{DP,A,O}} = T_{\mathrm{A,O}} \end{cases} \tag{14}$$

(3) Heat transfer performance and friction factor

The internal heat transfer coefficient is calculated as follows:

$$q_{\mathrm{In}} = \alpha_{\mathrm{In}} \left(T_{\mathrm{R}} - T_{\mathrm{T}} \right) \tag{15}$$

The external heat transfer coefficient is calculated as follows:

$$q_{\mathrm{Out}} = \alpha_{\mathrm{Out}} \frac{\left(T_{\mathrm{T}} - T_{\mathrm{A,I}}\right) - \left(T_{\mathrm{T}} - T_{\mathrm{A,O}}\right)}{\ln\left\{\left(T_{\mathrm{T}} - T_{\mathrm{A,I}}\right) / \left(T_{\mathrm{T}} - T_{\mathrm{A,O}}\right)\right\}} \tag{16}$$

The internal heat transfer coefficient is calculated by the following correlations.

In the case of single-phase flow, we use the Dittus-Boelter correlation:

$$\alpha_{\mathrm{In}} = \mathrm{Nu}_{\mathrm{R}} \frac{\lambda_{\mathrm{R}}}{D_{\mathrm{In}}} \tag{17}$$

$$\mathrm{Nu}_{\mathrm{R,SP}} = 0.023 \mathrm{Re}_{\mathrm{R}}^{0.8} \mathrm{Pr}_{\mathrm{R}}^{n} \tag{18}$$

In the case of two-phase boiling flow, we use the Yoshida et al. correlation:

$$\frac{\alpha_{\mathrm{In,TP}}}{\alpha_{\mathrm{In,SP,LIQ}}} = 3.7 \left\{ Bo \times 10^{4} + 0.23 \left(Bo \times 10^{4} \right)^{0.69} \left(\frac{1}{X_{\mathrm{tt}}} \right)^{2} \right\}^{0.44} \tag{19a}$$

In the case of two-phase condensation flow, we use the Nozu et al. correlation:

$$\mathrm{Nu}_{\mathrm{R,f}} = 0.018 \left(\mathrm{Re}_{\mathrm{R,L}} \sqrt{\frac{\rho_{\mathrm{R,LIQ}}}{\rho_{\mathrm{R,VAP}}}} \right)^{0.9} \left(\frac{x_{\mathrm{R}}}{1 - x_{\mathrm{R}}} \right)^{0.1 x_{\mathrm{R}} + 0.8} \left(\mathrm{Pr}_{\mathrm{R,L}} + \frac{8 \times 10^{3}}{\mathrm{Re}_{\mathrm{R,LIQ}}^{1.5}} \right)^{1/3} \left(1 + \frac{C_1 H}{\mathrm{Pr}_{\mathrm{R,LIQ}}} - 0.2 \frac{H_{\mathrm{v}}}{\mathrm{Pr}_{\mathrm{R,VAP}}} \right)$$

(19b)

$$\mathrm{Nu}_{\mathrm{R,b}} = 0.725 \left(\frac{Ga\, \mathrm{Pr}_{\mathrm{R,LIQ}}}{H} \right)^{0.25} \frac{\left\{ 1 + 0.003 \sqrt{\mathrm{Pr}_{\mathrm{R,LIQ}}}\, C_3^{\left(3.1 - 0.5/\mathrm{Pr}_{\mathrm{R,LIQ}}\right)} \right\}^{0.3}}{\left(1 + C_2 C_4 \right)^{0.25}} \tag{20}$$

The internal friction factor is calculated by the following correlations.

In the case of single-phase flow, we use the Blasius correlation:

$$f_{\mathrm{R,SP}} = 0.079\,\mathrm{Re}_{\mathrm{R}}^{-0.25} \tag{21}$$

In the case of two-phase flow, we use the Chisholm correlation:

$$f_{\mathrm{R,TP}} = 0.079\,\mathrm{Re}_{\mathrm{R,LIQ}}^{-0.25}\,\phi_{\mathrm{L}}^{2} \tag{22}$$

$$\phi_{\mathrm{L}}^{2} = 1 + \frac{20}{X_{\mathrm{tt}}} + X_{\mathrm{tt}}^{2} \tag{23}$$

Here, the external fin heat transfer coefficient and friction factor are calculated by the Seshimo-Fujii correlations:

$$\alpha_{\mathrm{Out}} = Nu_{\mathrm{A}}\,\frac{\lambda_{\mathrm{A}}}{D_{\mathrm{ec}}} \tag{24}$$

$$Nu_{\mathrm{A}} = 2.1\left(\frac{Re_{\mathrm{A}}\,Pr_{\mathrm{A}}\,D_{\mathrm{ec}}}{L_{\mathrm{x_A}}}\right)^{0.38} \tag{25}$$

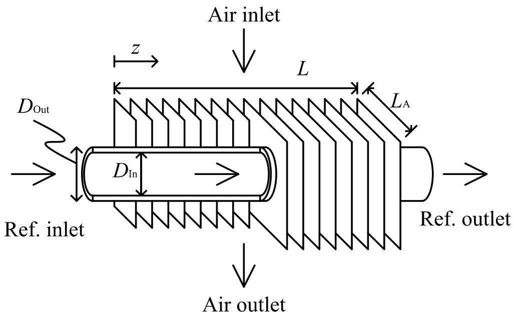


Figure 3.3 Heat exchanger model

3.2.3 Expansion valve model

The expansion valve model equations are shown below.

The continuity equation is as follows:

$$\rho_{\mathrm{R,O}} v_{\mathrm{R,O}} S_{\mathrm{R,O}} - \rho_{\mathrm{R,I}} v_{\mathrm{R,I}} S_{\mathrm{R,I}} = 0 \tag{26}$$

The energy equation is as follows:

$$\rho_{\mathrm{R,O}} v_{\mathrm{R,O}} h_{\mathrm{R,O}} S_{\mathrm{R,O}} - \rho_{\mathrm{R,I}} v_{\mathrm{R,I}} h_{\mathrm{R,I}} S_{\mathrm{R,I}} = 0 \tag{27}$$

The flow coefficient is used to calculate the mass flow:

$$\rho_{\mathrm{R,I}} v_{\mathrm{R,I}} S_{\mathrm{R,I}} = C_{\mathrm{V}} S \sqrt{2\rho_{\mathrm{R,I}}\left(P_{\mathrm{R,I}} - P_{\mathrm{R,O}}\right)}$$

(29)

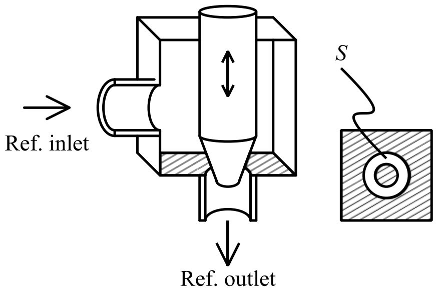


Figure 3.4 Expansion valve model

3.2.4 Accumulator model

An accumulator model is shown in Figure 3.5.

The continuity equation is as follows:

$$V\frac{d\rho_{\mathrm{R}}}{dt}+\rho_{\mathrm{R,O}}v_{\mathrm{R,O}}S_{\mathrm{R,O}}-\rho_{\mathrm{R,I}}v_{\mathrm{R,I}}S_{\mathrm{R,I}}=0 \tag{28}$$

The pressure drop is ignored:

$$P_{\mathrm{R,O}}=P_{\mathrm{R}} \tag{29}$$

$$P_{\mathrm{R,I}}=P_{\mathrm{R}} \tag{30}$$

The energy equation is as follows:

$$V\frac{d\left(\rho_{\mathrm{R}}u_{\mathrm{R}}\right)}{dt}+\rho_{\mathrm{R,O}}v_{\mathrm{R,O}}h_{\mathrm{R,O}}S_{\mathrm{O}}-\rho_{\mathrm{R,I}}v_{\mathrm{R,I}}h_{\mathrm{R,I}}S_{\mathrm{R,I}}=0 \tag{31}$$

$$h_{\mathrm{R,O}}=\begin{cases}h_{\mathrm{R}} & \left(x_{\mathrm{R}}=1.0\right)\\ h_{\mathrm{R,Sat,VAP}} & \left(x_{\mathrm{R}}<1.0\right)\end{cases} \tag{32}$$

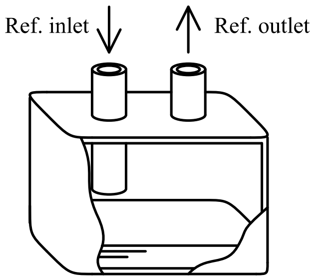


Figure 3.5 Accumulator

3.2.5 Validity of the model

The validity of the model is described in the following chapter.

3.3 How to use ‘Energy Flow + M’

The method of using ‘Energy Flow + M’ will now be introduced. In this simulator, the above mathematical model is invoked.

1st step: start-up

A Microsoft® Excel® spreadsheet is used to start this simulation, as shown in Figure 3.6.

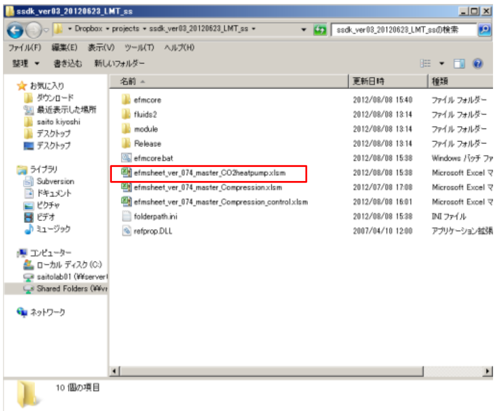


Figure 3.6 Run ‘Energy Flow + M’

The main sheet and workspace are shown in Figures 3.7 and 3.8. In the main sheet, we can select steady or unsteady calculations, type of refrigerant, time steps, etc.

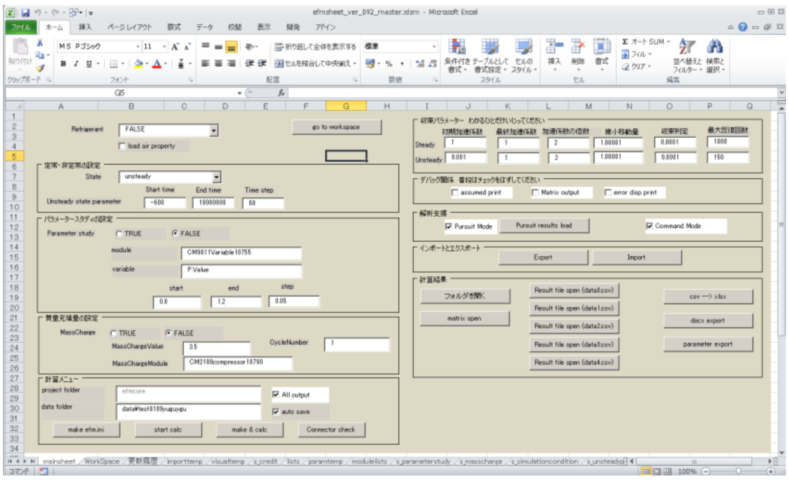


Figure 3.7 Main sheet

In the workspace, each necessary module for the system is selected and put on it.

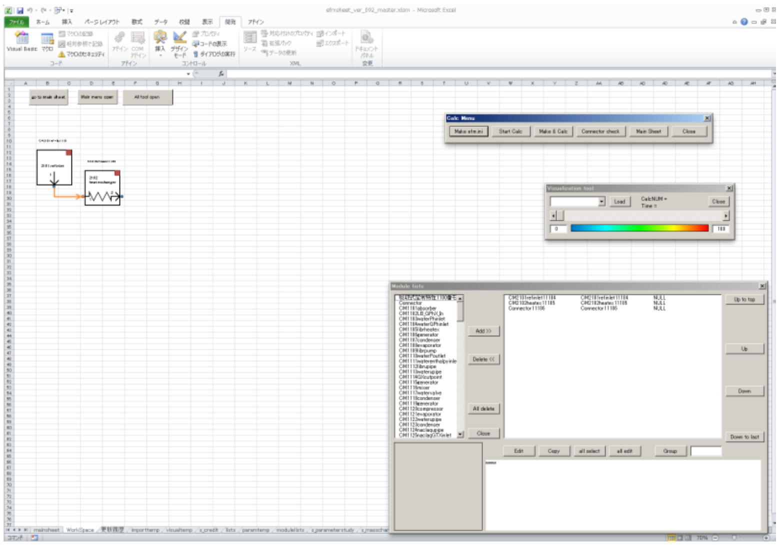


Figure 3.8 Workspace

2nd step: choose modules and put them in the workspace

In module lists of the workspace, we select the modules for calculation. By clicking the 'Add' button, the selected module appears in the workspace.

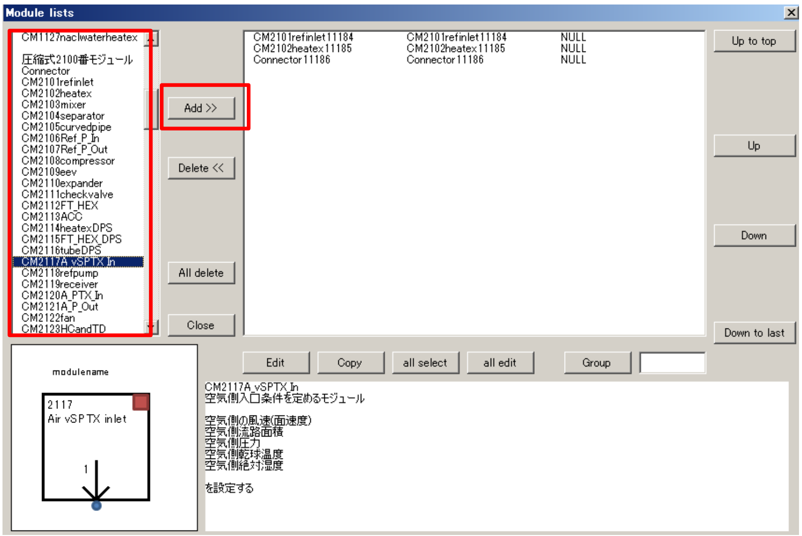


Figure 3.9 Module lists

3rd step: connect each module

The input and output ports of each module are connected as shown in Figure 3.10.

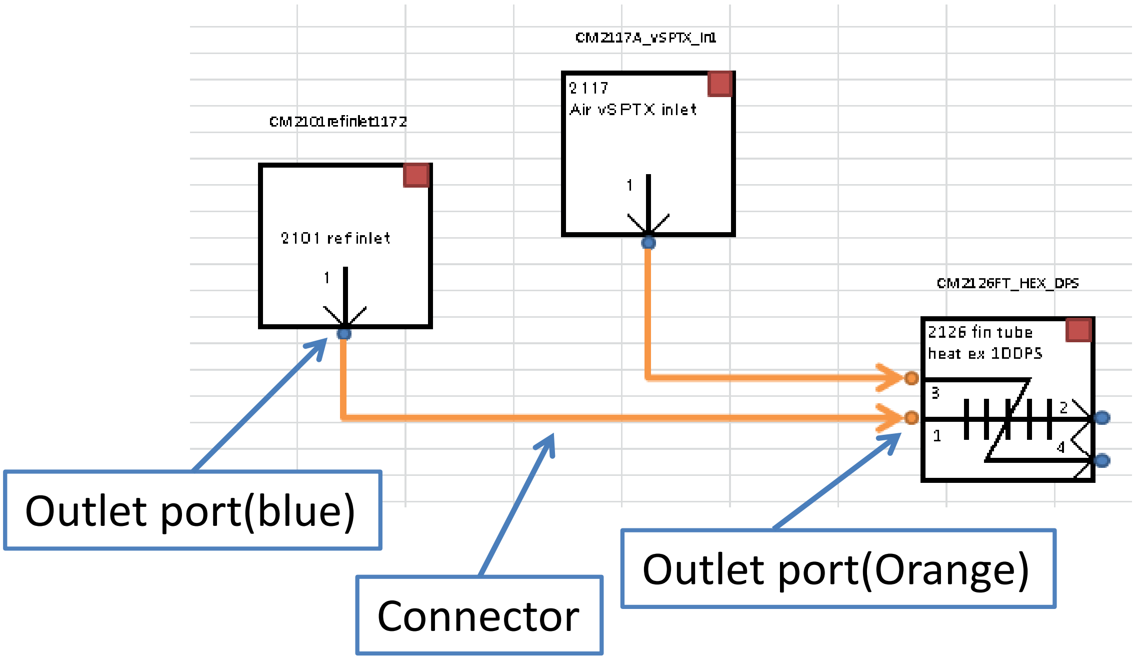


Figure 3.10 Module connection

4th step: set parameters

The parameter editor of each module is opened, and parameters are input such as length, angle, etc., as shown in Figure 3.11.

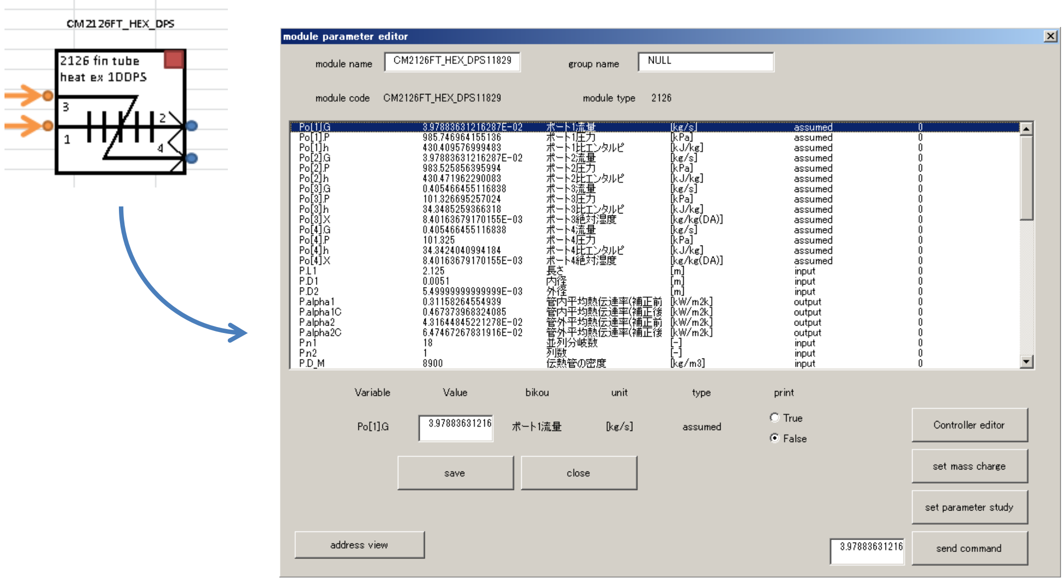


Figure 3.11 Parameter editor

Figure 3.12 represents an example of a heat pump system. All the necessary modules are connected sequentially with each other.

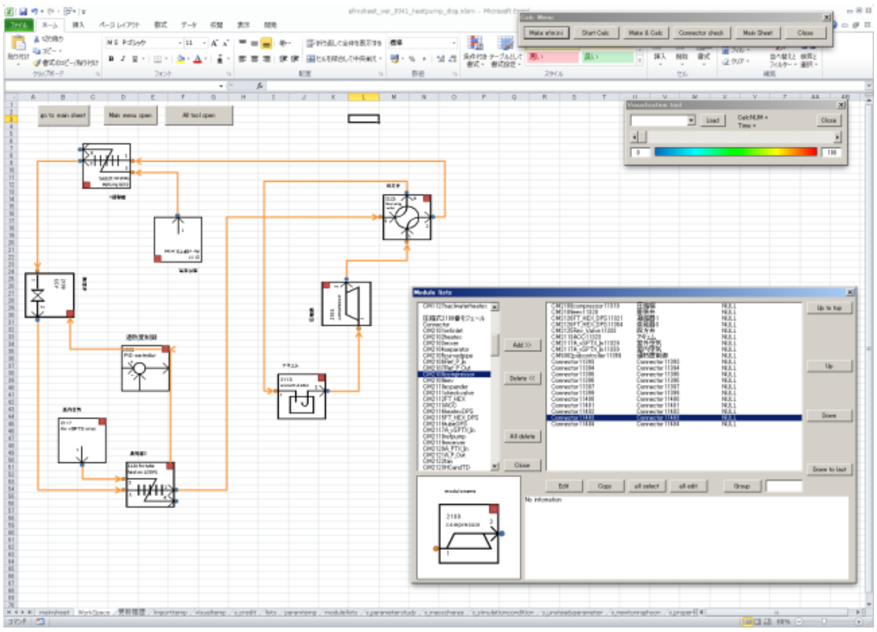


Figure 3.12 Heat pump system

5th step: start simulation

System calculation starts by clicking on the 'make & calc button', as shown in Figure 3.13.

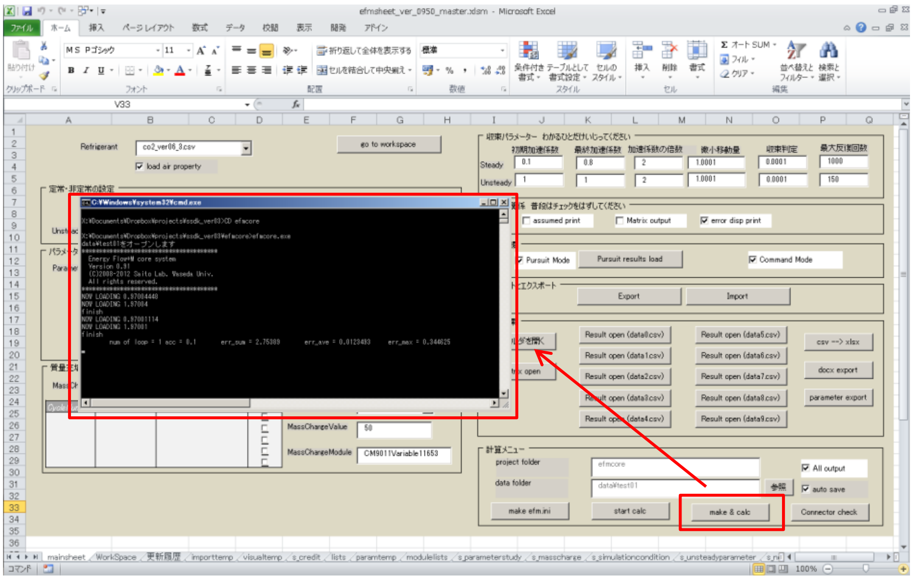


Figure 3.13 Start of simulation

6th step: simulation results

The calculation results are shown in Figure 3.14. The file type is ".csv," therefore, it is easy to use this data in Excel®, for example, to make figures.

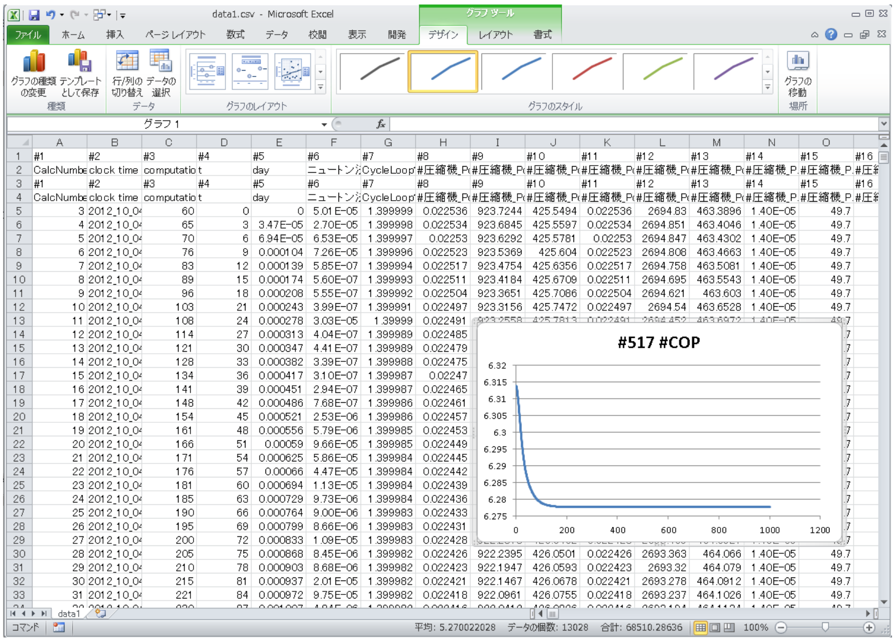


Figure 3.14 Simulation results

3.4 Simulation

Examples of calculations are shown for 'Energy Flow + M'.

3.4.1 Heat exchanger simulation (detailed version)

The calculation for a fin-and-tube-type heat exchanger is shown in Figure 3.15. This heat exchanger is typically used in air-conditioning systems.

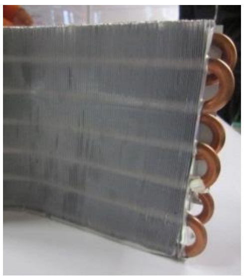


Figure 3.15 Heat exchanger

For example, the calculation model for a simple heat exchanger whose appearance is indicated in Figure 3.16 is shown in Figure 3.17.

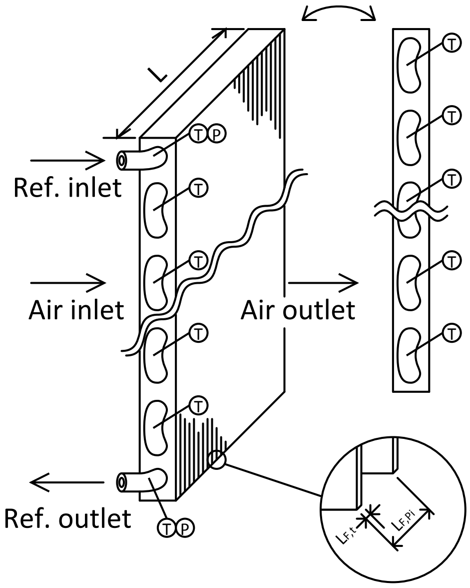


Figure 3.16 Heat exchanger model

The 'Energy Flow + M' model for this heat exchanger is shown in Figure 3.17. Straight and bent tubes are connected. In this calculation, the pressure drop and detailed heat transfer coefficient are considered.

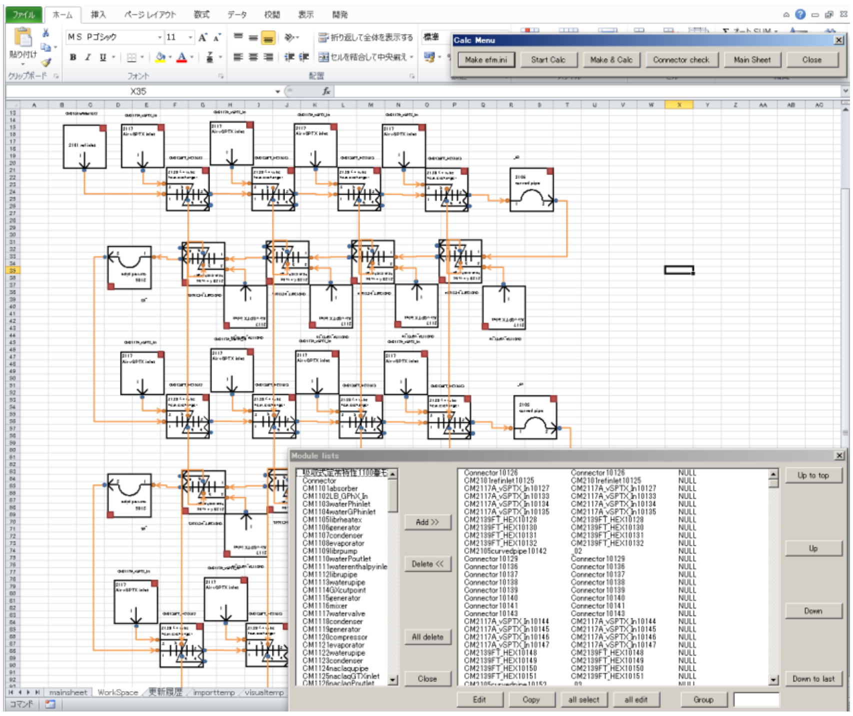


Figure 3.17 Heat exchanger in 'Energy Flow + M'

The simulation results for the condenser are shown in Figure 3.18. The simulation results agree well with the experimental results.

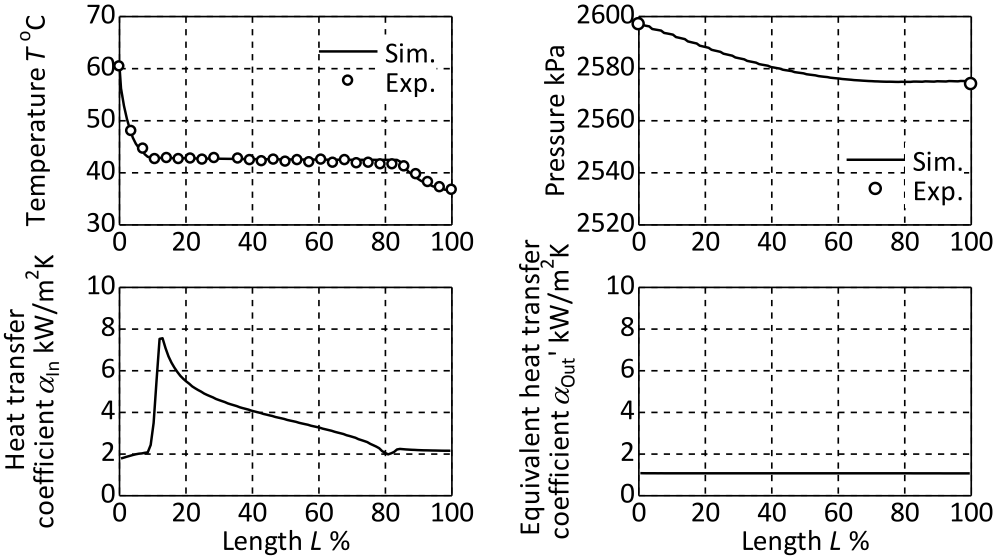


Figure 3.18 Simulation and experimental results

3.4.2 Heat pump simulation (steady state)

A steady-state simulation of a room air conditioner is shown. A photograph and schematic of a room air conditioner are shown in Figures 3.19 and 3.20 respectively.

Figure 3.19 Heat pump system (room air conditioner)

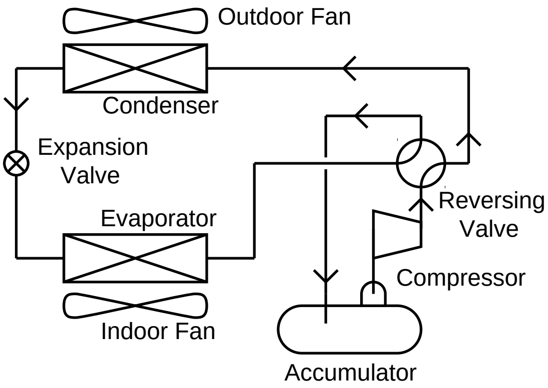


Figure 3.20 Schematic of simulation flow

A model for this system in 'Energy Flow + M' is shown in Figure 3.21. Each module such as compressor, heat exchanger and valve is connected.

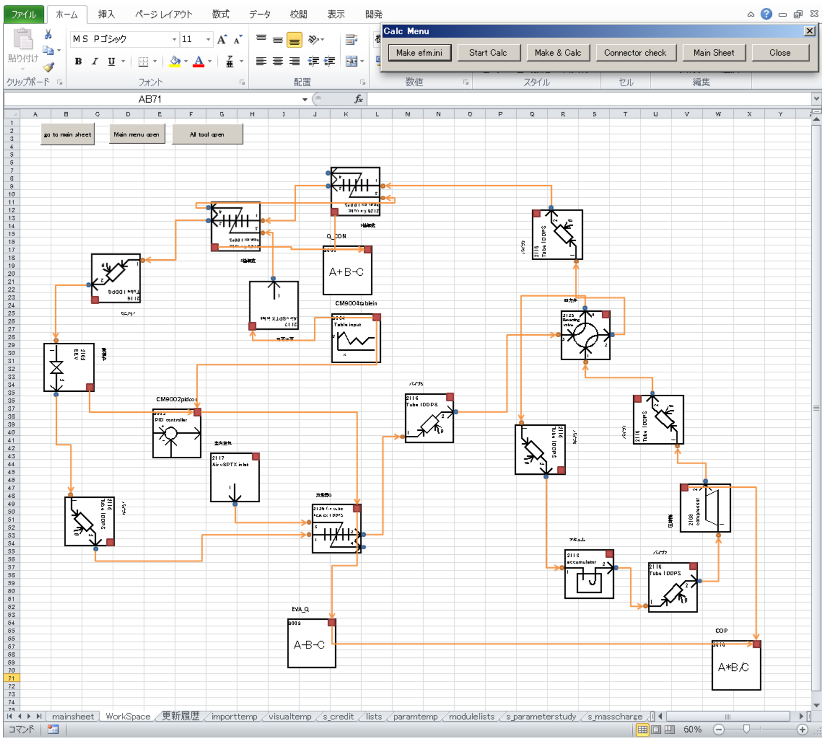


Figure 3.21 Heat pump in 'Energy Flow + M'

A steady-state simulation result is shown in Figure 3.22. Here, the results of the rated and middle operating points for cooling mode are shown. Simulation results depending on outdoor air temperature are shown in Figure 3.23.

The simulation results agree well with the experimental results.

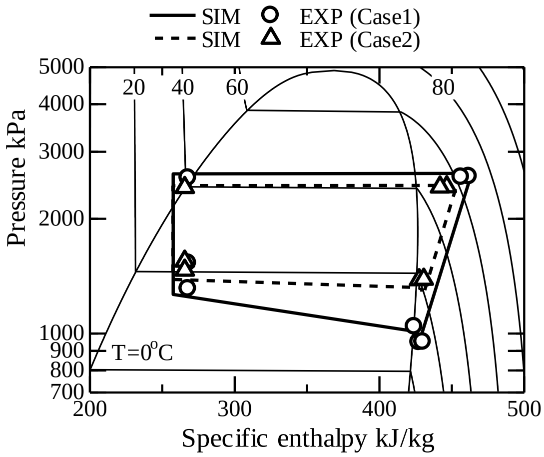


Figure 3.22 Simulation and experimental results on a pressure-enthalpy (ph) diagram

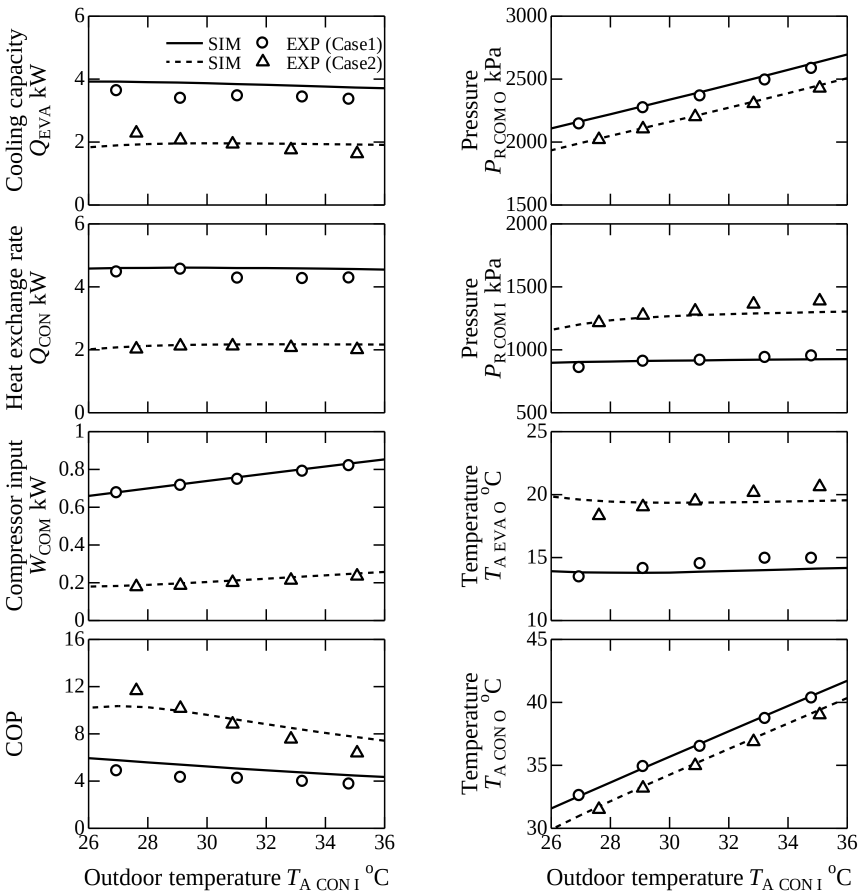


Figure 3.23 Simulation and experimental results

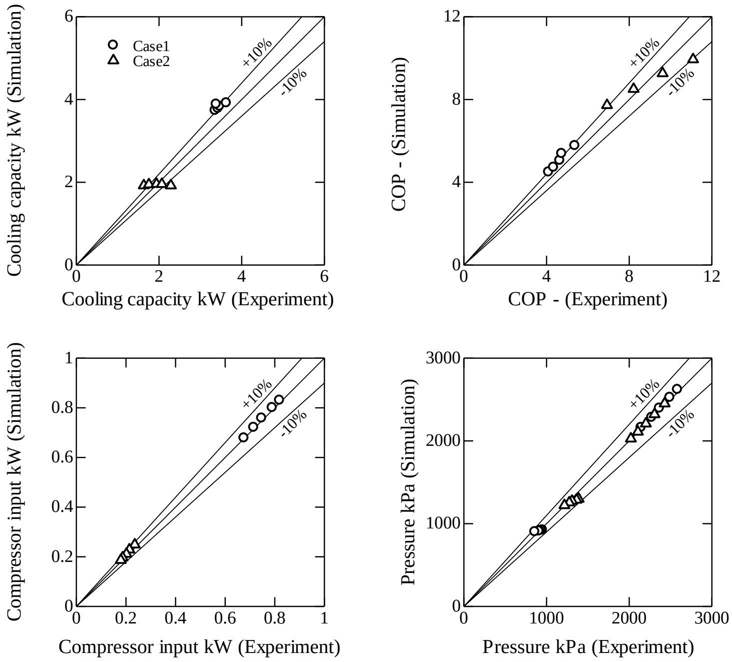


Figure 3.24 Simulation and experimental results

3.4.3 Heat pump simulation (unsteady state)

An unsteady state simulation of the heat pump can be carried out with 'Energy Flow + M'. We apply a package-type air-conditioning system, as shown in Figures 3.25 and 3.26.

Figure 3.25 Heat pump system (package air conditioner)

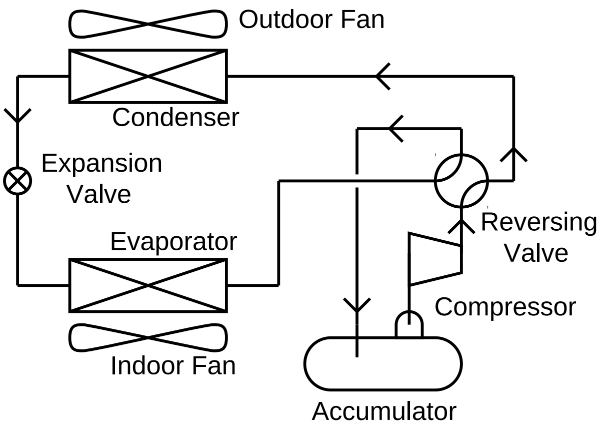


Figure 3.26 Schematic of simulation flow

A simulation model in 'Energy Flow + M' is shown in Figure 3.27. Each module such as compressor, heat exchanger and valve is connected. The compressor operation frequency time series data are directly input.

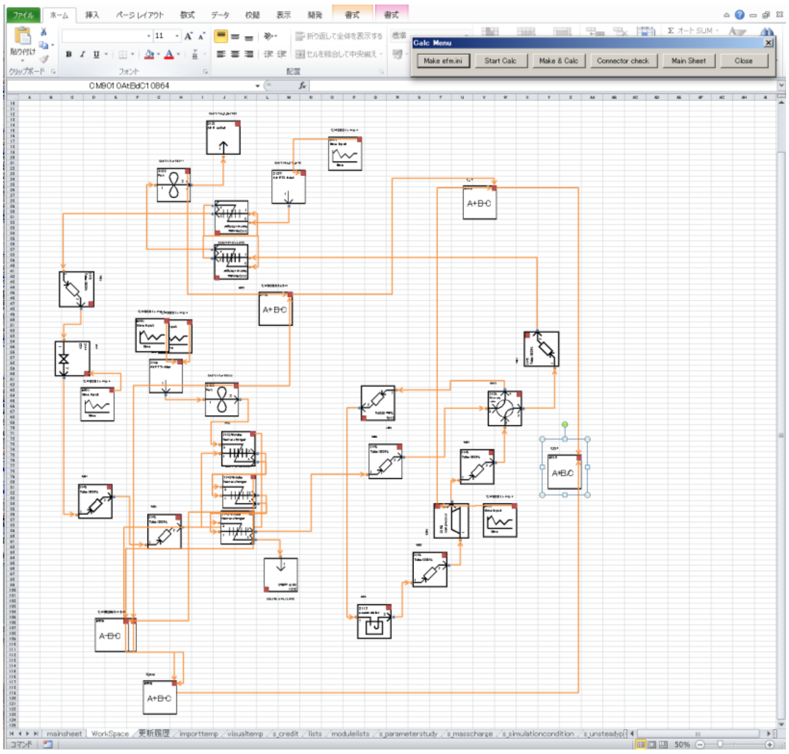


Figure 3.27 Heat pump in 'Energy Flow + M'

Figure 3.28 shows the simulation and experimental results for unsteady intermittent operation. The results agree well with each other.

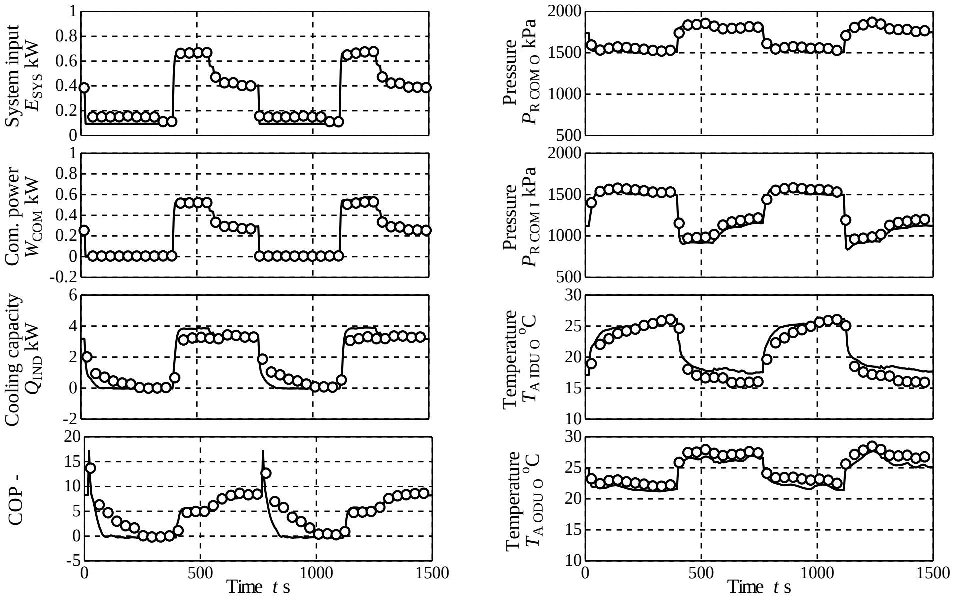


Figure 3.28 Simulation and experimental results

3.4.4 Urban energy system simulation (annual performance simulation)

With ‘Energy Flow + M’, we can carry out large-scale system simulations. For example, we show a simulation in which a heat pump uses energy from a buried pipe as heat source. In this calculation, the heat pump is treated as a module, as shown in Figure 3.29.

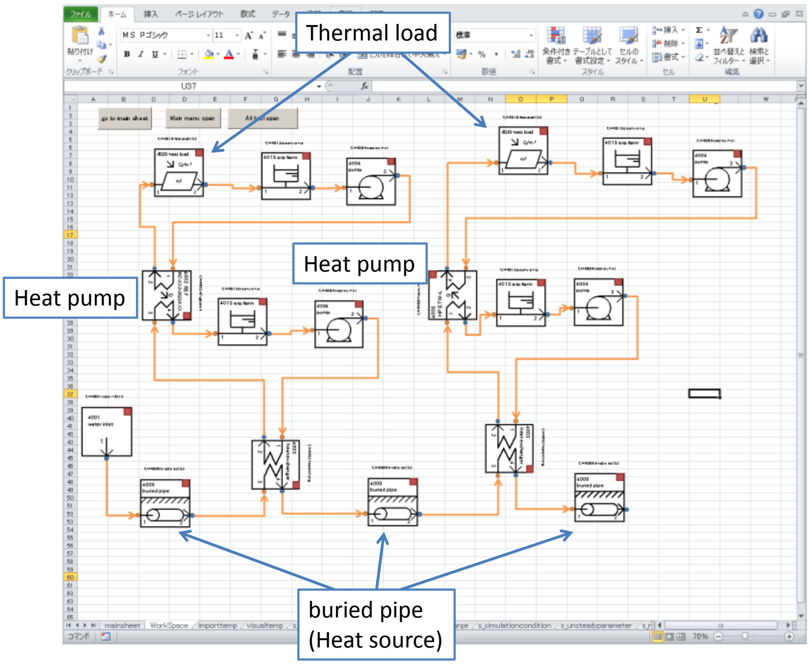


Figure 3.29 Urban energy system

4 COMPARISON OF THE YEAR-ROUND PERFORAMANCE FACTORS IN THE WORLD

4.1 Japanese standard APFs

4.1.1 Outline of APF measurement

APFs are calculated to evaluate energy efficiency of air conditioners in accordance with the actual conditions by taking into consideration the load conditions of periods in relation to outdoor temperature and changes of efficiency along with capacity changes.

Prior to calculating APFs, performance measurement needs to be conducted pursuant to JISC9612. Total load is calculated by multiplying loads in relation to outdoor temperature by load hours, while total power consumption is calculated by multiplying energy consumption in relation to outdoor temperature by energy consumption hours. APFs can be calculated by dividing total load by total power consumption (Figure 4.1).

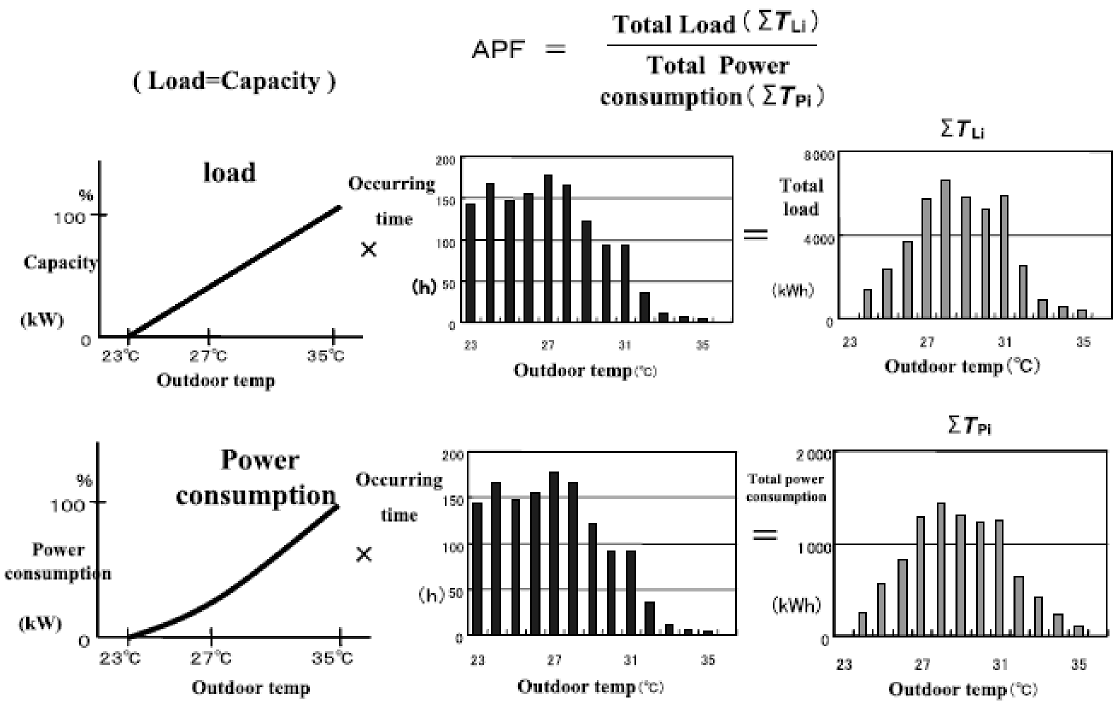


Figure 4.1 APF calculation concept

Temperature and humidity conditions

Temperature and humidity conditions for calculating APFs are summarized in Table 4.1. The conditions are categorized based on indoor and outdoor temperatures as well as rated, intermediate and maximum capacities.

Table 4.1 Temperature and humidity conditions for calculating APFs

		Indoor		Outdoor	
		Dry Bulb (℃)	Wet Bulb (℃)	Dry Bulb (℃)	Wet Bulb (℃)
Cooling Condition	Rated capacity	27	19	35	24
	Intermediate Capacity	27	19	35	24
Heating Condition	Rated capacity	20	15	7	6
	Intermediate Capacity	20	15	7	6
	Maximum Capacity	20	15	2	1

4.1.2 Load

Load conditions for calculating APFs are set forth as follows:

Cooling load is zero at an outdoor temperature of 23℃, and the rated cooling capacity at an outdoor temperature of 35℃ must coincide with the building load at an outdoor temperature of 33℃.

Heating load is zero at an outdoor temperature of 17℃, and the building load at an outdoor temperature of 0℃ must be 82% of the rated heating capacity which is taking cooling/heating ratio of 1.25 into account.

Cooling/heating load hours at each outdoor temperature are determined based on the following assumptions using meteorological data:

Location: Tokyo

Periods: June 2 - September 21 (cooling), October 28 - April 14 (heating)

Operation hours: 6:00 a.m. - 12:00 p.m.

Note: Same types of meteorological data for 18 other locations in Japan, which can be used for performance evaluation, are specified in JISC9612.

Total cooling/heating loads can be the total loads at specific outdoor temperature, which are calculated by multiplying building load by the load hours in relation to outdoor temperature respectively.

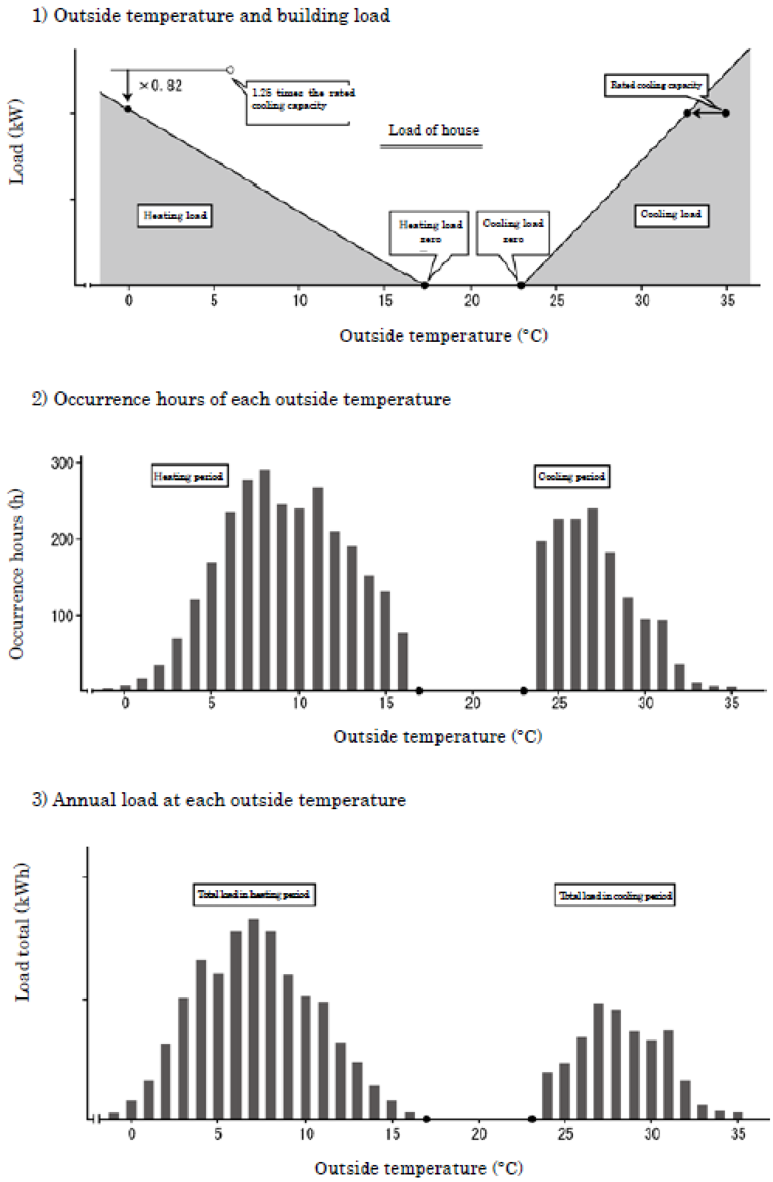


Figure 4.2 Relations between outdoor temperature & building loads, outdoor temperature & appearance hours, and outdoor temperature & loads

4.1.3 Cooling calculation

Figure 4.3 shows capacity, load, energy consumption and efficiency in relation to outdoor temperature in cooling mode respectively.

As shown by the dot plots in Figure 4.3, JISC9612 requires acquisition of actually measured data on rated capacity operation and intermediate capacity operation at an outdoor air temperature of 35℃. The values of rated and intermediate capacities and their energy consumptions at an outdoor temperature of 29℃ are calculated by the below-mentioned collinear approximation expressions that have been derived from the results of many tests conducted to date.

$\Phi(29) = 1.077*\Phi(35)$ $P(29) = 0.914*P(35)$

By collinear approximation of the data acquired at 29 ℃ and 35℃, rated/intermediate capacities and energy consumption along with outdoor temperature are acquired.

Calculation of efficiency

Efficiency is calculated by using the values of the rated capacity/rated energy consumption and the intermediate capacity/intermediate energy consumption at the above-mentioned temperature. Efficiency during cooling operation is classified into three ranges in relation to outdoor air temperature.

Range (a): In the case where building loads are higher than the rated capacity, the efficiency can be found by the above-described collinear approximation equations of rated condition.

Range (b): In the case where building loads are equal to or higher than the intermediate capacity and equal to or lower than the rated capacity, the efficiency in this case is defined as a straight line connecting the efficiency at the temperature where the intermediate capacity and rated capacity match with the building loads respectively.

Range (c): In the case where building loads are lower than intermediate capacity, the efficiency at 23℃ can be found by the above collinear approximation from the values of the intermediate capacity/energy consumption at an outdoor air temperature of 23℃. Then, the efficiency in this case is defined as a straight line connecting the efficiency at the temperature of 23℃ and the temperature where the building load and the intermediate capacity match. During cooling operation, only in the case where the lower limit of variable range of cooling capacity does not decrease to a level equal to or lower than intermediate operation capacity, such lower limit is regarded as the intermediate operation capacity and intermittent operation performance is calculated by using degradation coefficient Cd.

As described above, the efficiency of air conditioners at each outdoor air temperature is calculated in breakdown by classification of operating states, based on the relations between the capacities and the loads.

Calculation of cooling seasonal energy consumption (CSEC expressed in kWh)

Energy consumptions at respective outdoor air temperatures are calculated from the loads and efficiency at the respective temperatures, and the amount of power consumed (kWh) to provide cooling in buildings at respective outdoor air temperatures is calculated from the energy consumption and the number of appearance hours for the respective temperatures. And, CSEC is calculated from the sum total of power consumed in cooling operation at respective outdoor air temperatures.

Calculation of cooling seasonal performance factor (CSPF)

Similarly, total building loads (kWh) can be calculated at respective outdoor air temperatures from loads at respective temperatures and the number of appearance hours for the respective temperatures. Then, cooling seasonal total load (CSTL expressed in kWh) can also be calculated from the sum total of loads at respective outdoor air temperatures, and consequently CSPF is gained by dividing CSTL by CSEC.

$$\mathrm{CSPF} = \frac{\text{Cooling seasonal total load (CSTL)}}{\text{Cooling seasonal eneagy consumption (CSEC)}}$$

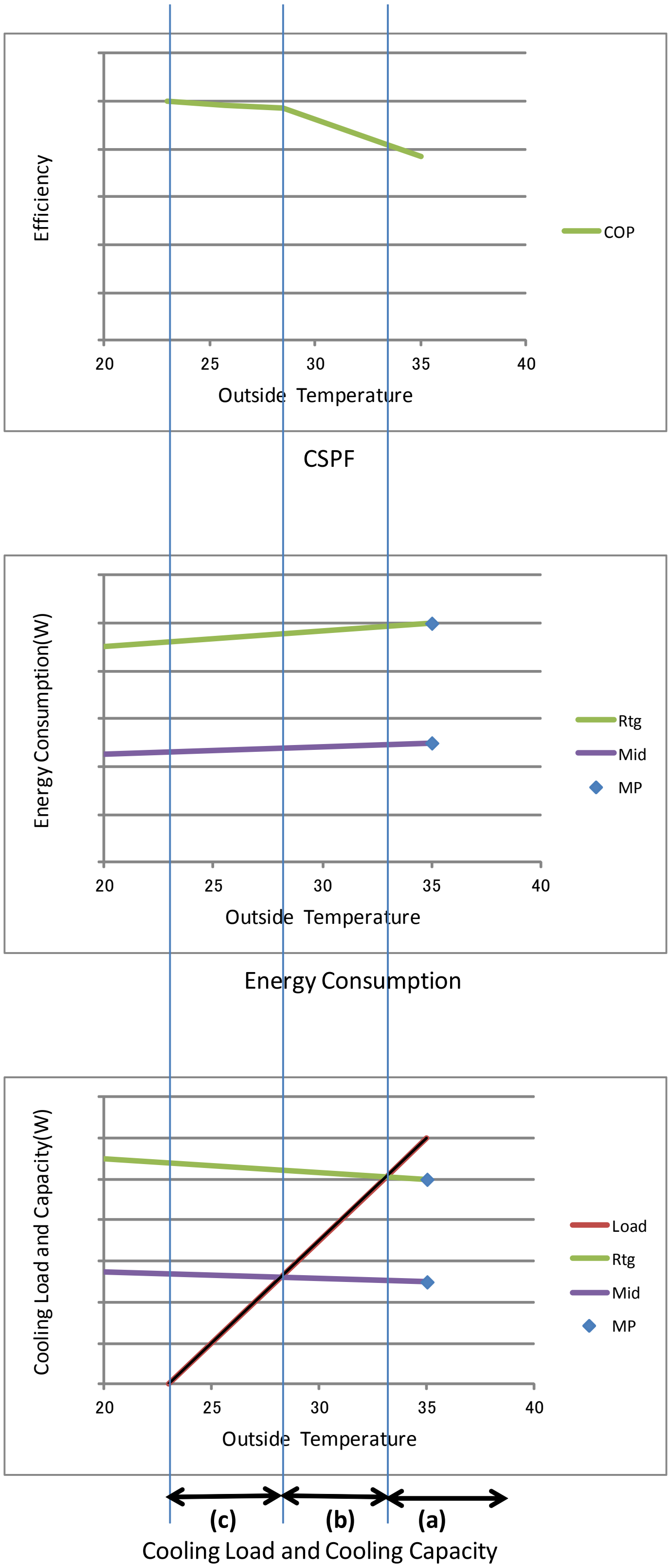


Figure 4.3 Cooling (JISC9612)

4.1.4 Heating calculation

Figure 4.4 shows capacity, load, energy consumption and efficiency in relation to outdoor temperature in heating mode respectively.

As shown by the dot plots in Figure 4.4, JISC9612 requires acquisition of actually measured data on rated capacity operation and intermediate capacity operation at an outdoor air temperature of 7℃ as well as the data on a maximum capacity operation at an outdoor temperature of 2℃.

The values of rated and intermediate capacities and their energy consumptions at an outdoor temperature of -7℃ are calculated by the below-mentioned collinear approximation expressions that have been derived from the results of many tests conducted to date.

$\Phi(-7) = 0.64*\Phi(7)$ $P(-7) = 0.82*P(7)$ (Rated and intermediate capacities)

$\Phi(-7) = 0.734*\Phi(2)$ $P(-7) = 0.877*P(2)$ (Maximum capacity)

It is assumed that frosting is caused at outdoor temperatures of -7℃ to 5.5℃. The decreased values of capacity and energy consumption at an outdoor air temperature of 2℃ are calculated by the following expressions that have been derived from the results of many tests conducted to date.

$\Phi(2) = 1.12*\Phi_{def}(2)$ $P(2) = 1.06*P_{def}(2)$ (Φ_{def}、P_{def} ; defrost condition)

By connecting the data acquired at 2℃ and -7℃, the capacity and energy consumption under frosting conditions are acquired.

Calculation of efficiency

Efficiency is calculated by using the values of the rated capacity/rated energy consumption and the intermediate capacity/intermediate energy consumption at the above-mentioned temperatures and relational equations under frosting conditions. Efficiency during heating operation is classified into five ranges in relation to outdoor air temperatures.

Range (a): In the case where building loads are lower than intermediate capacity, as is the case with cooling operation, the efficiency at 17℃ can be found by the above collinear approximation from the values of the intermediate capacity/energy consumption at an outdoor air temperature of 17℃. Then, the efficiency in this case is defined as a straight line connecting the efficiency at the temperature of 17℃ and the temperature where the building load and the intermediate capacity match. During heating operation, only in the case where the lower limit of variable range of heating capacity does not decrease to a level equal to or lower than intermediate operation capacity, such lower limit is regarded as the intermediate operation capacity and intermittent operation performance is calculated by using degradation coefficient Cd.

Range (b): In the case where building loads are equal to or higher than the intermediate capacity and equal to or lower than the rated capacity, the efficiency in this case is defined as a straight line connecting the efficiency that the intermediate capacity and rated capacity match with the building loads respectively, provided that the data of non-frosting conditions are used in case that the temperature is 5.5℃ or higher and the data of frosting conditions are used in case that the temperature is 5.5℃ or lower.

Range (c): In the case where building loads are equal to or higher than the rated capacity and equal to and lower than the maximum capacity, the efficiency in this case can be calculated from the rated capacity/energy consumption and maximum capacity/energy consumption, both under frosting conditions, and the efficiency in relation to outdoor air temperature can be calculated by connecting the efficiency at the outdoor temperature where the rated and maximum capacity match with the building loads.

Range (d): In the case where building loads are equal to or higher than a maximum capacity at outdoor air temperatures of -8.5℃ or higher, it is assumed that the air conditioners are operated at a maximum energy consumption mode under frosting conditions, and the loads that cannot be covered by the air conditioners are made up for by heaters and the energy consumption of the heaters is also included in calculation.

Range (e): In the case where building loads are equal to or higher than a maximum capacity at outdoor air temperatures of -8.5℃ or lower, it is assumed that equipment is operated at a maximum energy consumption mode under non-frosting conditions, and the amount of power consumed by heaters for additional use is also included in calculation.

As described above, the efficiency of air conditioners at each outdoor air temperature is calculated in breakdown by classification of operating states, based on the relations between the capacities and the loads.

Calculation of heating seasonal energy consumption (HSEC expressed in kWh)

Energy consumption at respective outdoor air temperatures is calculated from the loads and efficiency at the respective temperatures, and the amount of power consumed (kWh) to provide heating in buildings at respective outdoor air temperature is calculated from the energy consumption and the number of appearance hours for the respective temperatures. And, HSEC is calculated from the sum total of power consumed in heating operation at respective outdoor air temperatures.

Calculation of heating seasonal performance factor (HSPF)

Similarly, total building loads (kWh) can be calculated at respective outdoor air temperatures from loads at respective temperatures and the number of appearance hours for the respective temperatures. Then, heating seasonal total load (HSTL expressed in kWh) can also be calculated from the sum total of loads at respective outdoor air temperatures, and consequently HSPF is gained by comparing HSTLs.

$$\mathrm{HSPF} = \frac{\text{Total heating load}}{\text{Total heating energy consumption}}$$

Annual performance factor (APF)

APF is calculated based on the total loads in cooling and heating operations and annual energy consumption.

$$\mathrm{APF} = \frac{\text{Total coolin load} + \text{Total heatin load}}{\text{Total cooling energy consumption} + \text{Total heating energy consumption}}$$

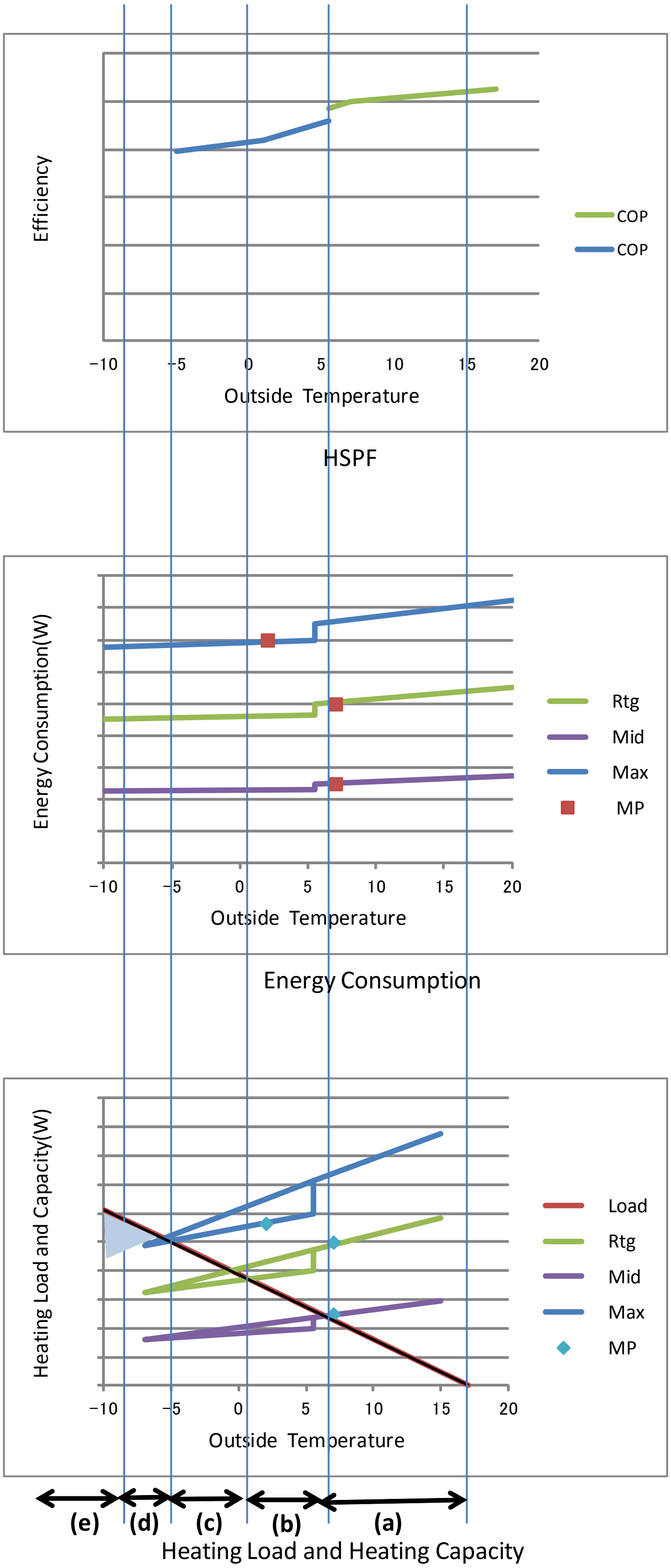


Figure 4.4 Heating (JISC9612)

4.2 European standard EN 14825

European standard is briefly introduced to compare the distinction of Japanese APF. Average heating and cooling seasons are chosen to explain the calculation method. This European standard covers air conditioners, heat pumps and liquid chilling packages and gives the calculation methods for determination of reference seasonal energy efficiency SEER in cooling season and reference seasonal coefficient of performance SCOP in heating season.

<u>Calculation method in case of cooling</u>

<u>Partial load condition in cooling</u>

The partial load condition to calculate SEER of air/air heat pump type is shown in Table 4.2. Cooling load matches with the rated capacity at 35℃, while cooling load reaches to zero at 16℃. Cooling load straight line connects both points. Efficiency and input power are measured under four conditions from A to D along with the load line.

Table 4.2 Partial load condition to calculate SEER of air/air heat pump type

	Partial load calculation	Partial load ratio (%)	Outdoor temperature (℃)	Indoor temperature DB(WB) (℃)
A	(35-16)/(TDesignc -16)	100%	35	27 (19)
B	(30-16)/(TDesignc -16)	74%	30	27 (19)
C	(25-16)/(TDesignc -16)	47%	25	27 (19)
D	(20-16)/(TDesignc -16)	21%	20	27 (19)

TDesignc: Design condition temperature at cooling 35℃

The cooling demand Pc(Tj) can be determined by multiplying the full load value (Pdesignc) by the partial load ratio (%) for each corresponding bin. This partial load ratio (%) is calculated as follows:

Partial load ratio (%) = (Tj-16)/(35-16)

<u>Calculation of efficiency</u>

Efficiency is calculated by the values of the capacity and energy consumptions for each point from A to D. Efficiency along with outdoor temperature is acquired by connecting each efficiency value from A to D.

Figure 4.5 shows these capacities, loads, energy consumptions and efficiency respectively.

<u>Calculation method for seasonal energy efficiency ratio (SEER) in case of cooling</u>

Total loads of cooling period are equal to summation of the product of building loads and appearance hours. Total electricity consumptions in cooling period are equal to summation of the product of electricity consumption and appearance hours as well. And, seasonal energy efficiency ratio in cooling period of heat pump operation (SEERon) can be led by a calculation where total loads in cooling period are divided into total electricity consumption in cooling period.

The reference SEERon is determined as follows:

$$SEER_{on} = \frac{\sum_{j=1}^{n} hj \times Pc(Tj)}{\sum_{j=1}^{n} hj \times \left(\frac{Pc(Tj)}{EER(Tj)} \right)}$$

Where,
Tj = bin temperature;
j = bin number;
n = amount of bins;
Pc(Tj) = cooling demand of the building for the corresponding temperature Tj;
hj = number of bin hours occurring at the corresponding temperature Tj;
EER(Tj) = EER values of the unit for the corresponding temperature Tj.

The EER values at each bin are determined via interpolation of the EER values under partial load conditions A, B, C and D.

The values to be used for j, Tj and hj are determined in Table 4.3 below:

Table 4.3 Bin Number j, outdoor temperature Tj in °C and number of hours per bin hj corresponding to reference cooling season

j	#	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Tj	°C	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
hj	h	205	227	225	225	216	125	218	197	178	158	137	109	88	63	39	31	24	17	13	9	4	3	1	0

SEER is gained from annual total cooling demand as is the case with SEERon divided by the annual electricity consumption. The annual electricity consumption represents the energy consumption during active mode of SEERon with that of thermostat in on mode and standby mode, off mode and that of the crankcase heater.

$$\mathrm{SEER} = \frac{\text{Total cooling demand}}{\text{Annual electricity consumption}}$$

Calculation method in case of heating

Partial load condition

The partial load condition to calculate SCOP of air/air heat pump type is shown in Table 4.4. Heating load matches with the rated capacity at -10℃, while heating load reaches to zero at 16℃. Heating load straight line connects both points. Efficiency and input power are measured under four conditions from A to D along with the load line.

Table 4.4 Partial load condition to calculate SCOP of air/air heat pump type

	Partial load calculation	Partial load ratio (%)	Outdoor temperature (℃)	Indoor temperature DB(WB) (℃)
A	(-7-16)/(TDesignh -16)	88	-7	20
B	(2-16)/(TDesignh -16)	54	2	20
C	(7-16)/(TDesignh-16)	35	7	20
D	(12-16)/(TDesignh -16)	15	12	20

TDesignh: Design condition temperature at heating -10 ℃

The heating demand Ph(Tj) can be determined by multiplying the full load value (Pdesignh) by the partial load ratio (%) for each corresponding bin. This partial load ratio (%) is calculated as follows:

Partial load ratio (%) = (Tj-16)/(-10-16)

Calculation of efficiency

Efficiency is calculated by the values of the capacity and energy consumptions for each point from A to D. Efficiency along with outdoor temperature is acquired by connecting each efficiency value from A to D.

Figure 4.6 shows these capacities, loads, energy consumptions and efficiency respectively.

Calculation method for seasonal coefficient of performance (SCOP) in case of heating

Total loads of heating period are equal to summation of the product of building loads and appearance hours. Total electricity consumptions in heating period are equal to summation of the product of electricity consumption and appearance hours as well. And, seasonal coefficient of performance in heating period of heat pump operation (SCOPon) can be led by a calculation where total loads in heating period are divided into total electricity consumption in heating period.

The reference SCOPon is determined as follows:

$$SCOP_{on} = \frac{\sum_{j=1}^{n} hj \cdot Ph(Tj)}{\sum_{j=1}^{n} hj \cdot \left(\frac{Ph(Tj) - elbu(Tj)}{COP_{PL}(Tj)} + elbu(Tj) \right)}$$

Where,

$Ph(Tj)$ = heating demand of the building for the corresponding temperature Tj, expressed in kW;

hj = number of bin hours occurring at the corresponding temperature Tj;

$COP_{PL}(Tj)$ = COP values of the unit for the corresponding temperature Tj;

elbu(Tj) = required capacity of an electric backup heater for the corresponding temperature Tj, expressed in kW.

The COP_{PL} values and capacity values at each bin are determined via interpolation of the COP_{PL} and capacity values under partial load conditions A, B, C and D.

The values to be used for j, Tj and hj are determined in Table 4.5 below:

Table 4.5 Bin number j, outdoor temperature Tj in °C and number of hours per bin hj corresponding to reference heating seasons “average”

j #	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37
Tj ℃	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1	0	1	2	3	4	5	6
hj h	1	25	23	24	27	68	91	89	165	173	240	280	320	357	356	303	330

j #	38	39	40	41	42	43	44	45	46
Tj ℃	7	8	9	10	11	12	13	14	15
hj h	326	348	335	315	215	169	151	105	74

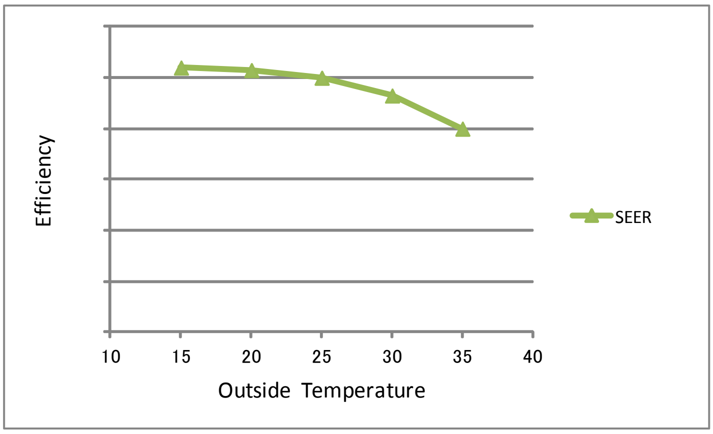
SCOP is gained from annual total heating demand as is the case with SCOPon divided by the annual electricity consumption. The annual electricity consumption represents the energy consumption during active mode of SCOPon with that of thermostat in on mode, standby mode and off mode and that of the crankcase heater.

$$\mathrm{SCOP} = \frac{\text{Total heating demand}}{\text{Annual electricity consumption}}$$

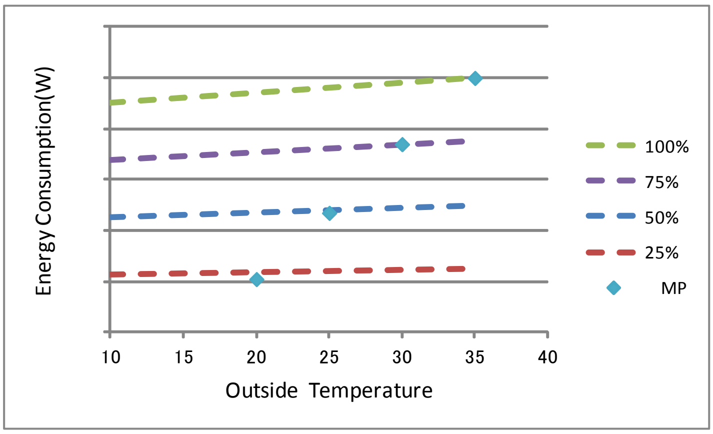
Seasonal performance factor: SPF

SPF is calculated based on the total loads in cooling and heating operations and seasonal energy consumption.

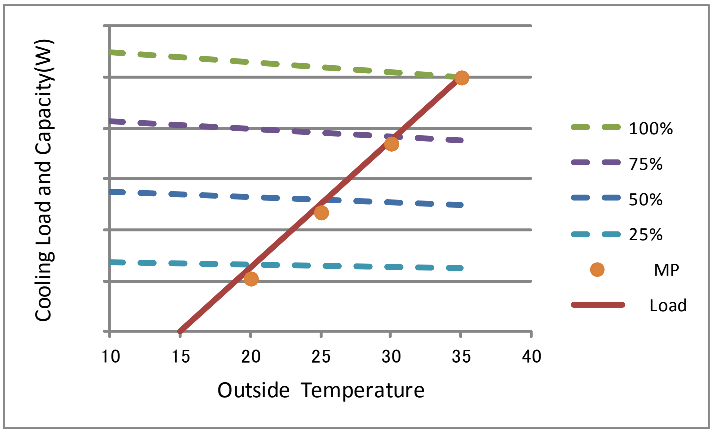
$$\mathrm{SPF} = \frac{\text{Total coolin load} + \text{Total heatin load}}{\text{Total cooling energy consumption} + \text{Total heating energy consumption}}$$



SEER

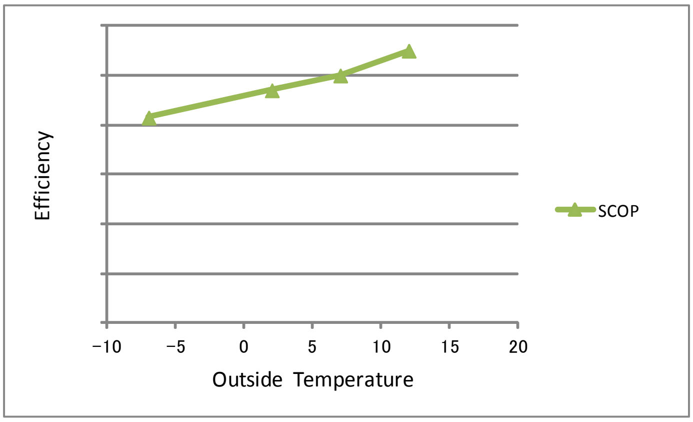


Energy consumption

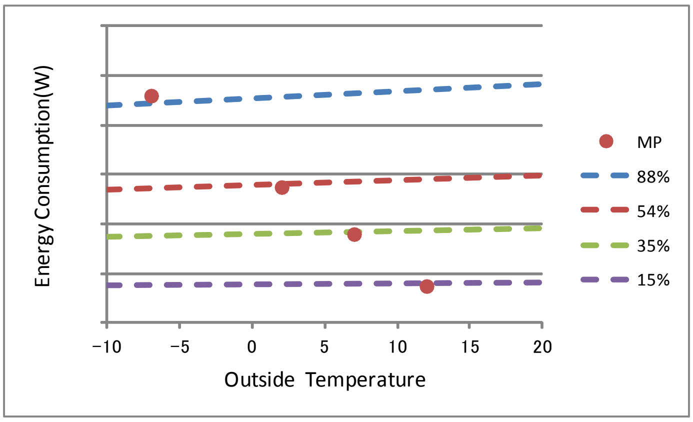


Cooling Load and Cooling Capacity

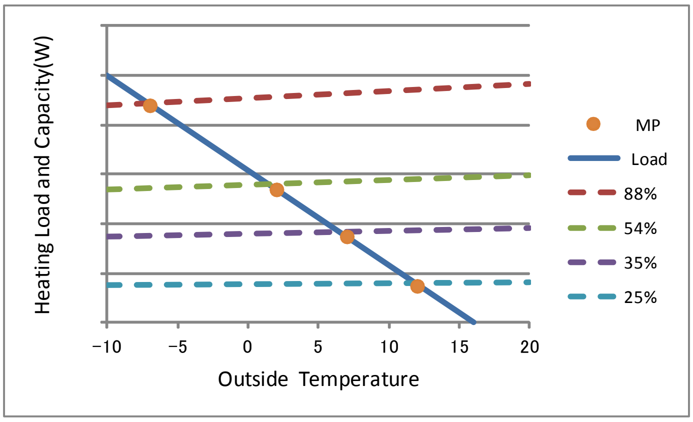
Figure 4.5 Cooling (EN 14825)



SCOP



Energy Consumption



Heating Load and Heating Capacity

Figure 4.6 Heating (EN 14825)

4.3 American standard ANSI AHRI 210/240

American standard is briefly introduced to compare the distinction of Japanese APF. The purpose of this standard is to establish seasonal calculation methods for unitary air-conditioners and air-source unitary heat pumps.

Fractional bin hours for cooling season: the ratio of the number of hours during cooling season when the outdoor temperature fell within the range represented by bin temperature Tj to the total number of hours in cooling season, is shown in Table 4.6.

Similarly, fractional bin hours for the heating season are shown in Table 4.7 for six climate regions.

Table 4.6 Distribution of fractional hours within cooling season temperature bins

Bin Number, j	Bin Temperature Range °F	Representative Temperature for bin °F	Fraction of Total Temperature Bin Hours, N_j/N
1	65-69	67	0.214
2	70-74	72	0.231
3	75-79	77	0.216
4	80-84	82	0.161
5	85-89	87	0.104
6	90-94	92	0.052
7	95-99	97	0.018
8	100-104	102	0.004

Table 4.7 Generalized climatic region information

j	Tj (°F)	I	II	III	IV	V	VI
Region Number		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	*2750
Outdoor Design Temperature, T_{OD}		37	27	17	5	-10	30
j	Tj (°F)	Fractional Bin Hours n_j/N					
1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.042	.087	.094	.008
9	22	.001	.008	021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

* Pacific Coast Region.

Cooling

Cooling load matches with the rated capacity at 37℃, while cooling load reaches to zero at 17℃. Cooling load line connects both points.

Calculation of efficiency

As shown by the dot plots in Figure 4.7, this standard requires acquisition of actually measured data on maximum capacity operation at outdoor air temperatures of 35℃ and 27.8℃, minimum capacity operation at outdoor air temperatures of 27.8℃ and

19.4℃, and intermediate capacity operation at an outdoor air temperature of 30.6℃. Efficiency is calculated by using the values of the maximum capacity/maximum energy consumption and the minimum capacity/minimum energy consumption at the above-mentioned temperatures. Efficiency during cooling operation is classified into three ranges in relation to outdoor air temperatures.
Figure 4.7 shows these capacities, loads, energy consumptions and efficiency respectively.

Range (a): In the case where building loads are driven by the maximum capacity or higher, the straight line of efficiency at the maximum capacity in relation to outdoor air temperatures is acquired from the performance data at the two points of the maximum capacity at 35℃ and 27.8℃.

Range (b): In the case where building loads are equal to or higher than a minimum capacity and equal to or lower than a maximum capacity, efficiency values at the points where building loads match with the maximum capacity and minimum capacity are calculated. The quadratic curve approximation of efficiency applying data of maximum capacity, minimum capacity and intermediate capacity represents efficiency between maximum capacity and minimum capacity.

Range (c): In the case where building loads are driven by the minimum capacity or lower, the straight line of efficiency at the minimum capacity in relation to outdoor air temperatures is acquired from the performance data at the two points of the minimum capacity at 19.4℃ and 27.8℃.
Unlike Japan's APFs, intermittent operation is always considered. The Cd values during intermittent operation may be derived from the calculation using a value of 0.25 or from actual tests.

<u>Calculation of cooling seasonal energy consumption (CSEC expressed in kWh)</u>
Energy consumptions at respective outdoor air temperatures are calculated from the loads and efficiency at the respective temperatures, and the amount of power consumed (kWh) to provide cooling in buildings at respective outdoor air temperatures is calculated from the energy consumption and the number of appearance hours for the respective temperatures. And, CSEC is calculated from the sum total of power consumed in cooling operation at respective outdoor air temperatures.

Calculation of cooling seasonal performance factor (SEER)

Similarly, building loads (kWh) can be calculated at respective outdoor air temperatures from loads at respective temperatures and the number of appearance hours for the respective temperatures. Then, cooling seasonal total load (CSTL expressed in kWh) can also be calculated from the sum total of loads at respective outdoor air temperatures, and SEER is gained by dividing CSTL by CSEC.

$$\mathrm{SEER} = \frac{\mathrm{CSTL}}{\mathrm{CSEC}}$$

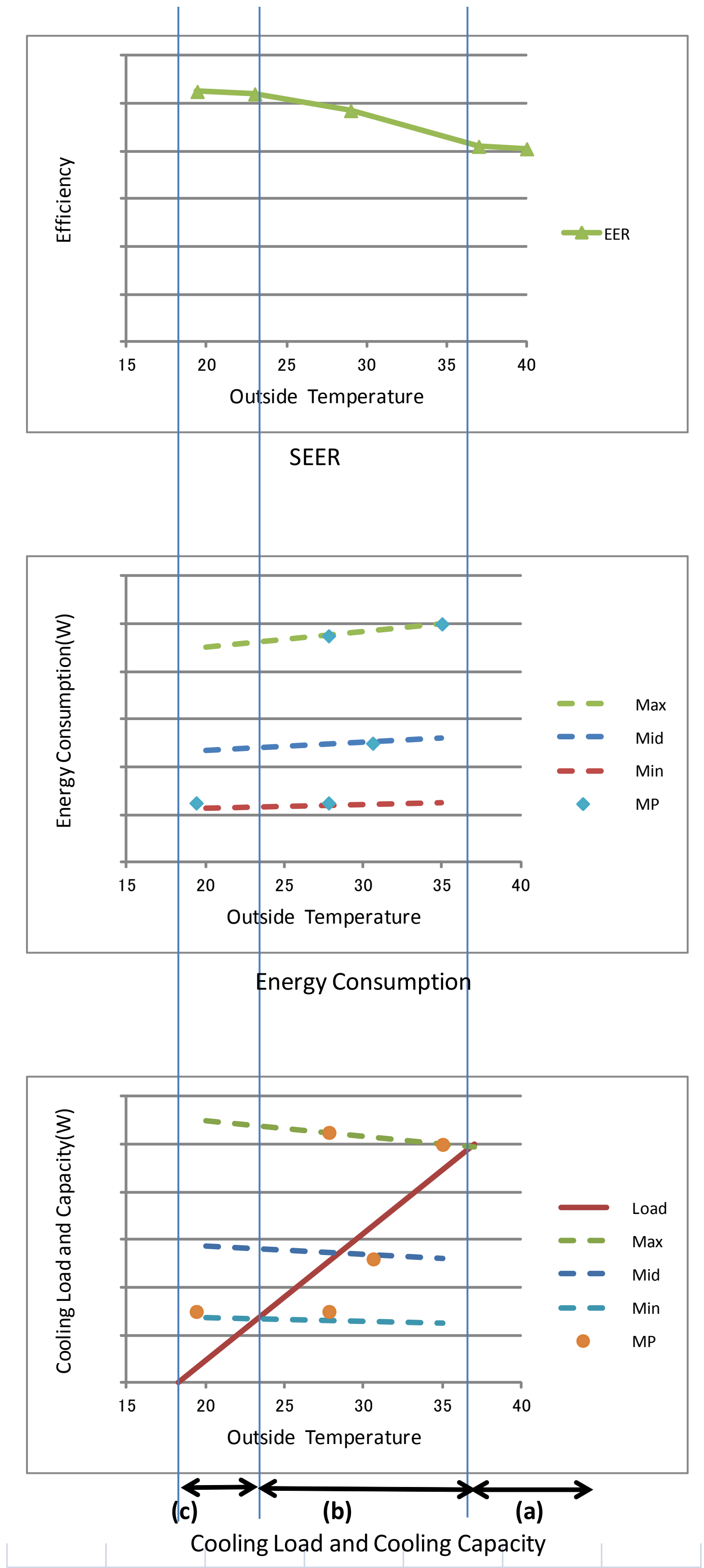


Figure 4.7 Cooling (AHRI)

Heating

Heating load matches with the rated capacity at -5℃, while heating load reaches to zero at 18.3℃. Heating load line connects both points.

Calculation of efficiency

As shown by the dot plots in Figure 4.8, this standard requires acquisition of actually measured data on maximum capacity operation at outdoor air temperatures of 1.7℃ and -8.3℃, minimum capacity operation at outdoor air temperatures of 8.3℃ and 16.7℃, and intermediate capacity operation at an outdoor air temperature of 1.7℃.
Efficiency is calculated by using the values of the maximum capacity/maximum energy consumption and the minimum capacity/minimum energy consumption at the above-mentioned temperatures. Efficiency during heating operation is classified into four ranges in relation to outdoor air temperatures.
Figure 4.8 shows these capacities, loads, energy consumptions and efficiency respectively.

Range (a): In the case where building loads are driven by the minimum capacity or lower, similarly to cooling condition, the straight line of efficiency at the minimum capacity along with outdoor temperatures is acquired from the performance data at the two points of the minimum capacity at 8.3℃ and 16.7℃.

Range (b): In the case where building loads are equal to or higher than a minimum capacity and equal to or lower than a maximum capacity, efficiency at the points where building loads match with the maximum capacity at frosting mode and minimum capacity is calculated. The quadratic curve approximation of efficiency applying data of maximum capacity, minimum capacity and intermediate capacity represents efficiency between maximum capacity and minimum capacity.

Range (c): In the case where building loads are driven by the maximum capacity or higher and outdoor temperature is higher than -8.5℃, it is assumed that the air conditioners are operated under frosting conditions.

The straight line of efficiency at the maximum capacity in relation to outdoor air temperature is acquired from the performance data at the two points of the maximum capacity at -8.3℃ and 1.7℃. The loads that cannot be covered by the air conditioners are made up for by heaters and the energy consumption of the heaters is also included in calculation.

Range (d): In the case where building loads are driven by the maximum capacity or higher and outdoor temperature is lower than -8.5℃, it is assumed that the air conditioners are operated under non-frosting conditions. The straight line of efficiency at the maximum capacity in relation to outdoor air temperatures is acquired from the performance data at the two points of the maximum capacity at -8.3℃ and 8.3℃.
The loads that cannot be covered by the air conditioners are made up for by heaters and the energy consumption of the heaters is also included in calculation.

Calculation of heating seasonal energy consumption (HSEC expressed in kWh)
Energy consumption at respective outdoor air temperatures is calculated from the loads and efficiency at the respective temperatures, and the amount of power consumed (kWh) to provide heating in buildings at respective outdoor air temperatures is calculated from the energy consumption and the number of appearance hours for the respective temperatures. And, HSEC is calculated from the sum total of power consumed in heating operation at respective outdoor air temperatures.

Calculation of heating seasonal performance factor (HSPF)
Similarly, building loads (kWh) can be calculated at respective outdoor air temperatures from loads at respective temperatures and the number of appearance hours for the respective temperatures. Then, heating seasonal total load (HSTL expressed in kWh) can also be calculated from the sum total of loads at respective outdoor air temperatures, and HSPF is gained by dividing HSTL by HSEC.

$$\mathrm{HSPF} = \frac{\mathrm{HSTL}}{\mathrm{HSEC}}$$

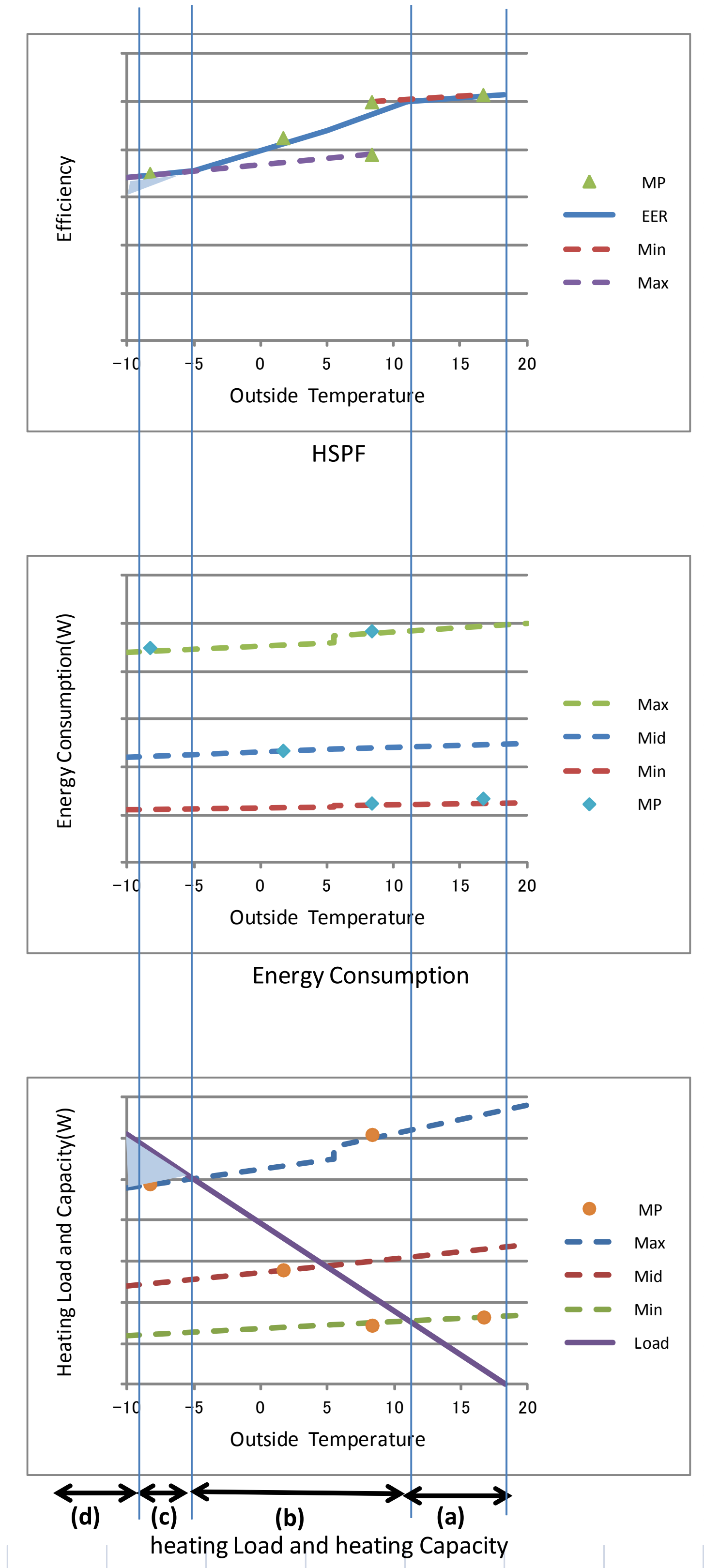


Figure 4.8 Heating (AHRI)

Annual performance factors (APFs)

APFs are calculated based on the total loads in cooling and heating operations and annual energy consumption.

$$\mathrm{APF} = \frac{\mathrm{CLH} \times \mathrm{CSTL} + \mathrm{HLH} \times \mathrm{HSTL} \times \mathrm{C}}{\frac{\mathrm{CLH} \times \mathrm{CSTL}}{\mathrm{SEER}} + \frac{\mathrm{HLH} \times \mathrm{HSTL} \times \mathrm{C}}{\mathrm{HSPF}}}$$

C = 0.77 (correlation factor),

CLH and HLH are estimated based on Table 4.8 or the maps of Figure 4.9 which indicate the cooling and heating load hours respectively.

Table 4.8

Representative cooling and heating load hours for each generalized climatic region

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

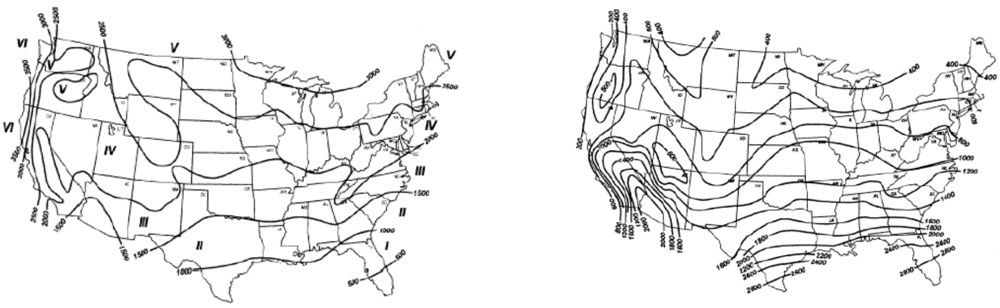


Figure 4.9 Left: heating load hours (HLH_A)
and right: cooling load hours (CLH_A) in the United States

4.4 Comparison of the three standards

APF, EN 14825 and ANSI/AHRI 210/240 are summarized in the list below to identify their features.

Table 4.9 Comparative table of the three standards (cooling)

	APF (year-round performance factor)	EN 14825 SPF	ANSI/AHRI210/240 SEER
Building load	Load = 0: outdoor temp. at 23℃ Load = 100%: outdoor temp. at 33℃	Load = 0: outdoor temp. at 16℃ Load = 100%: outdoor temp. at 35℃	Load = 0: outdoor temp. at 17℃ Load = 100%: outdoor temp. at 37℃
Rationality	Tokyo only	Unique meteorological condition of Strasbourg	Appearance rate of hours a day is unique in cooling period for 6 regions.
Estimated load appearance hours	Cooling period: Jun 2 - Sep 21 Cooling hours: 6:00 - 24:00 Total cooling hour: 2,016 hours	2,286 hours in total	Divided into 6 regions. 200 - 2,400 hours depending on a region.
Measuring points	Total: two points (rated and *int. capacities) * Int. capacity = 1/2 rated capacity	Four points (measured at rated capacity and 74%, 47% and 21% of rated capacity)	Five points: two at max. compressor rotating speed, one at int. capacity and two at min. capacity
Conditions of measuring points	100% and 50% loads at 35℃	35.0℃: 100% 30℃: 74% 25℃: 47% 20℃: 21%	35.0℃: max. 30.6℃: int. 27.8℃: max. min. 19.4℃: min.
	Outdoor temp. is fixed at rated condition of 35℃.	Four measuring points. Partial loads are measured at different temp.	Five measuring points at four different temps. Partial loads are measured at different temp.
Load ranges	Three ranges: over max. load, max. to int. loads and less than int. load depend on relevance to each load and outdoor temp.	No range classification	Three ranges: over max. load, max. - min. loads and less than min. load depend on relevance to each load and outdoor temp. Efficiency curves are taken by connecting their dots in each range.
Performance calculation	Efficiency in each range is calculated by using measured, rated and int. data at an outdoor temp. of 35℃ and empirical approximation equation.	Efficiency curve is estimated by measuring four points between 100% and 21% on load curve in relation to outdoor temp.	Efficiency in each range is calculated by using max. capacity line, min. capacity line and measured int. data. 1. Max. capacity line: connect two points, measured at 35℃ & 27.8℃. 2. Min. capacity line: connect two points, measured at 27.8℃ & 19.4℃.
Capacity range of measurement	Down to int. capacity (1/2 of rated capacity)	Down to 21% of rated capacity	Down to approx. 25% of rated capacity
Remarks about capacity range	Low capacities are not measured due to the possibility of large measuring errors.	Measuring error can be large at 21% of rated capacity.	Measuring error can be large at 25% of rated capacity.

	Instead, empirical equation derived from lots of experiment data is adopted.		
Intermittent operation	Degradation coefficient Cd is not used. As the lower limit of variable range of cooling capacity is rarely below the int. operation in Japan, they do not calculate by using degradation coefficient Cd.	Degradation coefficient Cd = 0.25 is used. Same as AHRI210/240	Degradation coefficient Cd = 0.25 is used.
Stand-by power consumption	Not considered	Stand-by power consumption is considered.	
Bin respective temperature range	1℃	1℃	5°F (2.8℃)

Table 4.10 Comparative table of the three standards (heating)

	APF (year-round performance factor)	EN 14825 SPF	ANSI/AHRI210/240 SEER
Building load	Load = 0: outdoor temp. at 17℃ Load = 82% of rated capacity at an outdoor temp. of 0℃ in heating	Load = 0: outdoor temp. at 16℃ Load = 100%: outdoor temp. at -10℃	Load = 0: outdoor temp. at 18.3℃ Load = 100%: outdoor temp. at -5℃
Rationality	Tokyo only	Divided into standard, cold and warm regions. Each region has its own appearance ratio in accordance with outdoor temp. in heating period.	Divided into six regions. Each region has its own appearance ratio in accordance with outdoor temp. in heating period.
Estimated load appearance hours	Heating period: Oct 28 - Apr 14 Heating hours: 6:00 - 24:00 Total heating hour: 3,042 hours	Heating hours of standard region are 4,910 hours.	750 - 2,750 hours depending on regions
Measuring points	Total of three points: two from rated and int. capacities at 7℃, one from max. capacity at 2℃.	Three - five points in three regions (basically measured at full rated capacity or 75%, 50% and 25% of rated capacity.	Five points: two at max. compressor rotating speed, one at int. capacity and two at min. capacity
Conditions of Measuring points	Rating and int. capacities at 7℃, and max. capacity at 2℃	-7.0℃: 88% 2℃: 54% 7℃: 35% 12℃: 15%	-8.3℃: max. 1.7℃: int. 8.3℃: max. min. 16.7℃: min.
	Outdoor temp. is fixed at rated condition of 7℃ & 2℃.	Four measuring points. Partial loads are measured at different temp.	Five measuring points at four different temps. Partial loads are measured at different temp.

Load ranges	Three ranges: over max. load, max. - int. loads and less than int. load depend on relevance to each load and outdoor temp.	No range classification	Three ranges: over max. load, max. - min. loads and less than min. load depend on relevance to each load and outdoor temp. Efficiency curves are taken by connecting their dots in each range.
Performance calculation	Efficiency in each range is calculated by using measured rated and int. data of an outdoor temp. of 7℃, 2℃ and empirical approximation equation.	Efficiency curve is estimated by measuring four points between 88% and 15% on load curve in relation to outdoor temp.	Efficiency in each range is calculated by using max. capacity line, min. capacity line and measured int. data. 1. Max. capacity line: connect two points, measured at 1.7℃ & -8.3℃. 2. Min. capacity line: connect two points, measured at 8.3℃ & 16.7℃.
Capacity range of measurement	Down to int. capacity (1/2 of rated capacity)	Down to 15% of rated capacity	Down to approx. 25% of rated capacity
Remarks about capacity range	Low capacities are not measured due to the possibility of large measuring errors. Instead, empirical equation derived from lots of experiment data is adopted.	Measuring error can be large at 15% of rated capacity.	Measuring error can be large at 25% of rated capacity.
Frost	Considered in every load between max. and int.		Considered only in performance curve of max. load not in other loads.
Intermittent operation	Degradation coefficient Cd is not used. As the lower limit of variable range of cooling capacity is rarely below the int. operation in Japan, they do not calculate by using Degradation coefficient Cd.	Degradation coefficient Cd = 0.25 is used. Same as AHRI210/240	Degradation coefficient Cd = 0.25
Stand-by power consumption	Not considered	Suggested that stand-by power consumption should be considered.	
Bin temperature range	1℃	1℃	5℉ (2.8℃)

<u>Summary</u>

The major difference between Japan's APF and European/American standards is the estimation method in the low range of capacity. Their features are summarized below.

Japanese APF calculation features

Japan applies practical and empirical estimation methods which allow to calculate

APF with the five-point measurements above intermediate capacity. Performance level of HP products at every outdoor temperature degree can be calculated by using only measurement data of the rated and intermediate capacities.
The estimation at low capacity of this method is reliable, since the correlation equations are derived from lots of experiment data of manufacturers.
As a result, Japanese calculation method is free from difficulty of precise data acquisition at low capacity.
Furthermore, the workload of measurement is relatively low for manufacturers to measure HP products.

The weakness is, for instance, that all HP products are assumed to have unique correlation equation with respect to outdoor temperatures. In addition, the rated condition and intermediate condition are measured at the same outdoor temperature and temperature range for frosting is also the same. These facts cause the intermediate capacity point of cooling and the rated capacity point of heating to be away from the load line plot, which may decrease accuracy. Another weakness is an effect from intermittent operation, which is not considered at the low capacity, and electricity consumption at standby mode is not considered as well.

Features EN 14825 SPF
SPF calculation in European standard is based on the data of 100% load to around 20% load measurement along with load line. Since the data cover widely from high load to low load, this calculation method is idealistic.
On the contrary, from the measurement point of view, the data measured under the conditions of a smaller capacity such as 25% are easily affected by measuring errors. Precise measurement is necessary in low capacity measurement.

Other characteristics regarding measurement are that declared performance is acceptable and measurement activities are not necessary (reasonable grounds are necessary).
In addition, the tolerance of the error is set at 8%.

Features of ANSI/AHRI 210/240 SEER
Calculation under the American standard is based on the data of maximum capacity operation to minimum capacity operation. Since the data cover widely from high load to low load, this calculation method is also idealistic.

On the contrary, from the measurement point of view, the data measured under the conditions of a lower capacity are easily affected by measuring errors. Precise measurement is necessary in low capacity measurement.

5 CURRENT STATUS OF MEASUREMENT TECHNOLOGY

Methods to measure performance of domestic room air conditioners with rated cooling capacities of 10 kW or less are specified in JISB8615.

5.1 Test Facility

The balanced ambient room-type calorimeter and the Psycrometric calorimeter (air enthalpy test method) are created in accordance with JISB 8615. Their schematic diagrams are shown in Figures 5.1 and 5.2 respectively. The tunnel air enthalpy methods are difficult to maintain accuracy at low air volumes, for which the balanced ambient room-type calorimeters are adopted. However, they are easy-to-use and usually adopted for high capacity air conditioners. The balanced ambient room-type calorimeters are usually used for low-capacity air conditioners, but there still exist technical challenges with balanced ambient room-type calorimeters, such as temperature change adjustment and room temperature stabilization.

<u>Balanced ambient room-type calorimeter</u>

A reconditioning apparatus is installed in each of the indoor-side and outdoor-side compartments. The capacity of the reconditioning apparatus is controlled so as to balance cooling or heating capacity of the equipment under test, and as a result, air temperature and humidity of indoor-side and outdoor-side compartments are stabilized under the JIS conditions. In addition, air temperature on both indoor-side and outdoor-side controlled-temp air space in Figure 5.1 is controlled to reach the same temperature as that in indoor-side and outdoor-side compartments, respectively. The amount of heat passing through the two compartments is a known value, since it was measured beforehand. Cooling or heating capacity of the equipment under test can be measured with high accuracy by determining cooling or heating capacity of the reconditioning apparatus under a stable condition.

<u>Low capacity measurement problems in balanced ambient room-type calorimeter</u>

Low capacity measurement problems are mainly caused by a cooling capacity of the reconditioning apparatus which makes use of water (brine). The uncertainty of the liquid flow rate measurement and temperature measurement increases at the measurement of low-capacity air conditioners by using the balanced ambient room-type calorimeters. On top of that, relative uncertainty also increases as the measuring capacity decreases.

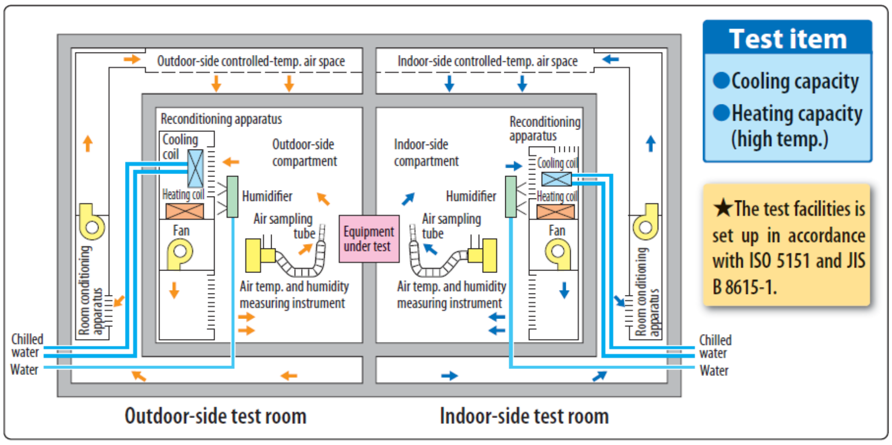


Figure 5.1 Balanced ambient room-type calorimeter

Psycrometric calorimeter (air enthalpy test method)

Air temperature and humidity of indoor-side and outdoor-side test rooms are controlled in accordance with the JIS conditions. Then, the test equipment is installed and operated. The test room is equipped with air temperature and humidity measuring instruments and an airflow measuring apparatus capable of highly accurate measurement. Cooling or heating capacity of the equipment under test can be measured with high accuracy by determining enthalpy from the measurements of temperature and humidity of air entering and leaving the equipment under test.

A variety of tunnel air enthalpy test methods are available. This calorimeter only measures the inside of test rooms using air flow measurement devices in many cases. Because this facility requires less time to measure as compared with balanced ambient room-type, tunnel air enthalpy test methods are suitable to measure the performance of room air conditioners, package air conditioners and car air conditioners, etc. at development stage. Especially, under heating condition in a low temperature, air enthalpy test method is suitable for a frost-defrost testing where the load fluctuation is large, it is difficult for the balanced ambient room-type calorimeters to follow promptly the temperature and humidity fluctuation.

Low capacity measurement problems in air enthalpy test method

It is important to select suitable nozzle size in order to obtain an appropriate level of differential pressure. In case that inadequate nozzle size is selected especially under a low air flow rate condition, relative uncertainty would be large. Moreover, air conditioners which are operated at the lowest capacity and maintain a constant air flow rate bring only a slight temperature difference between the inlet and outlet of indoor air-conditioner. It would reduce the accuracy of measurement as well. In addition, in case of a measurement facility which covers comparatively large capacity, the size of the air flow measuring system is large as well. And, its thermal capacity and the system itself are relatively too large according to the measuring capacity and it requires relatively long time to obtain stable condition. Additionally, relative uncertainty increases as measuring capacity decreases.

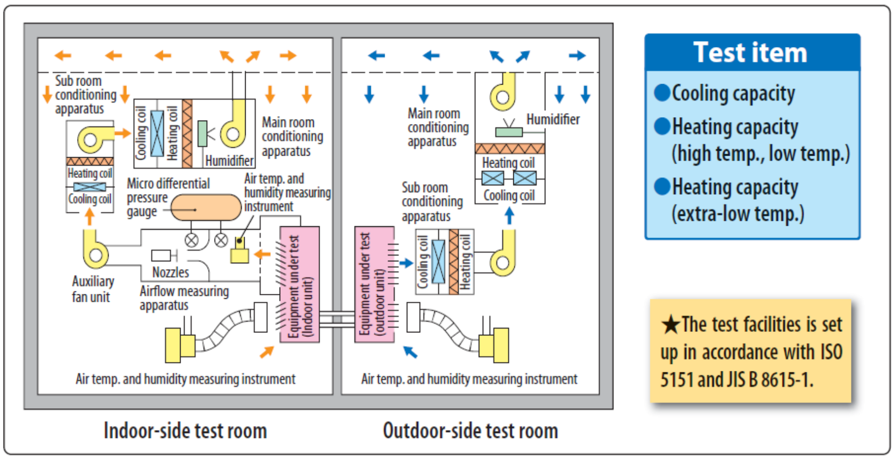


Figure 5.2 Air enthalpy test method

<u>Other verification facility</u>

Only the balanced ambient room-type calorimeter or the air enthalpy test method (tunnel air enthalpy measurement equipment) in test facility is used in Japan, while avoiding to use calibrated room type calorimeters due to their heat leakage problem. This contributes to maintain high accuracy in measurement.

In some cases, calibrated room type calorimeters which are developed in accordance with ISO 5151 are adopted as official measurement equipment in the world rather than balanced ambient room-type calorimeters since the cost is less expensive.

However, they have some problems: both indoor and outdoor test rooms for these calorimeters are single-walled; thermal insulation performance is not good enough for accurate measurement; the inside temperature fluctuates according to the outdoor air temperature; and without controlling the outside temperature around the test room, errors in heat leakage estimation tend to be large. In a certain case, for example, compensation for an air conditioner with a rated capacity of 2,600 W was 400 W.

5.2 Japan's efforts to conduct measurement as precise as possible

Measurement of air conditioner performance requires accurate temperature measurements at many measuring points, flow rate and so on, in order to maintain its accuracy. Japan has seriously paid attention to securing its accuracy and to developing measurement methods for avoiding fluctuation and heat leakage.

5.2.1 Measures for accuracy

Fixed frequency compressor operation

The inverter driven air-conditioner which can promptly follow load fluctuation is popular in Japan. However, for its performance test, Japan has adopted a highly accurate measurement of fixed frequency compressor operation. Each manufacturer's compressor is obliged to be equipped with fixed frequency mode so that they can conduct these kinds of tests. This prevents the test results from receiving effects of rotation speed fluctuation.

5.2.2 Verification programs

Among the verification programs, the performance tests in Japan are conducted by the private sector organization, Japan Air Conditioning and Refrigeration Testing Laboratory (JATL).

JATL is the first organization in Japan to obtain ISO/IEC 17025* certification as an air conditioner testing institute. Japanese manufacturers ensure accuracy of their certified measurement equipment by carrying out periodic mutual evaluation tests every year.

* ISO/IEC 17025 is an international standard that specifies general requirements for the competence of the test and calibration laboratories or organizations.

As a calibration workflow shows below, an applicant's test facility is calibrated against JATL's one (standard facility) followed by an inspection of a test facility on site and its quality maintenance system by JATL. If the applicant's test results agree with those of JATL within ±3%, it is recognized as having adequate capabilities and a "certificate of verification for test facilities" is issued. See below for further workflow.

Calibration workflow:

1) Make a contract on the verification program between Japan Refrigeration and Air

Conditioning Industry Association (JRAIA) and an applicant

2) A test facility is registered to JRAIA by an applicant.

3) The registered test facility is required to be recognized as having adequate capabilities and the "certificate of verification for test facilities" should be issued by JATL.

4) A test facility on site and its quality maintenance system are inspected by JATL.

5) All products of an applicant are registered to JRAIA.

6) An applicant tests all the registered models and puts a “verified” label on the model which passes the test.

7) JATL gets the registered products from the market to conduct a random performance test based on the JIS procedure.

8) JRAIA announces the names of qualified air conditioner models on their website.

JISB8615 requests that air conditioners must have an actual capacity of 97% or more, an actual input of 103% or less, and a higher APF than the values indicated in product catalogues. In order to verify product quality, JATL conducts annual sampling inspections at least one product of a manufacturer from the market. On the other hand, JRAIA has established their own stricter test standard than that of JIS for further reliable quality.

5.2.3 Uncertainty analysis

Uncertainty exists in measuring electricity, temperature, humidity, wind, etc. and uncertainty is required to be quantified.

JATL is carrying out reasonable uncertainty calculations based on Japan Accreditation Board (JAB) guidelines (JAB NOTE 6) and ISO/TS16491 which regulates uncertainty calculation in air conditioning performance test since 2012, since the uncertainty calculations are required to obtain ISO/IEC 17025. This calculation makes it possible to define a limit of measurement capability.

Table 5.1 shows the result of uncertainty analysis. The uncertainty of the wattage is acquired through the uncertainty model.

This uncertainty model method calculates the deviation through the partial differentiation model with a given standard uncertainty of each sensor device.

This expresses 3.5% uncertainty, when the uncertainty is 76 W for the cooling capacity of 2,200 W in this case.

Japan's manufacturers have adopted these guidelines as well and calculated measurement uncertainty for quasi-standard facilities on the same basis of JATL.
The case of above 3% uncertainty at rated capacity indicates relatively large uncertainty in the low capacity test. It must be necessary to improve uncertainty calculation methods further for reduction of uncertainty.

Table 5.1 Uncertainty budget

A primary factor	Standard uncertainty on wattage (W)
Indirect Contribution (Uncertainty except values derived from capacity equation) (This is equivalent to 1.5% of measured capacity)	33.00
Uncertainty caused by measurement of Indoor wall temperature	6.30
Uncertainty caused by measurement of Outdoor wall temperature	6.30
Uncertainty caused by measurement of Indoor partition wall temperature	1.25
Uncertainty caused by measurement of supply water temperature for humidification	1.25
Uncertainty caused by measurement of drain water temperature from test air conditioner	0.01
Uncertainty caused by measurement of humidification	0.16
Uncertainty caused by measurement of drain water temperature from the Re-air conditioner	0.01
Uncertainty caused by measurement of quantity of drain water	0.01
Uncertainty caused by measurement of inlet temperature at cooling coil	9.86
Uncertainty caused by measurement of outlet temperature at cooling coil	9.86
Uncertainty caused by measurement of flow rate of cooling coil	1.65
Uncertainty caused by measurement of total input wattage	9.52
Combined standard uncertainty (W)	38.20
Expanded uncertainty (W)	76.40
Relative expanded uncertainty (%)	**3.47%**

5.2.4 Collaboration with test institutes in other countries

Japan is making efforts to improve measurement performance through round robin tests with overseas laboratories mainly in Asia.
Japan has concluded a technology agreement with the Korea Refrigeration and Air-conditioning Assessment Center (KRAAC) and periodically carries out round robin

test activities with KRAAC. Japan also conducts round robin tests with several laboratories in China and has built a technical relationship with them.

Furthermore, Japan has had technical exchanges of mutual evaluation test, mutual calibration collaboration, provision of training courses under a METI (Ministry of Economy, Trade and Industry) project in ASEAN countries such as Thailand, Vietnam, Indonesia, Malaysia, the Philippines and also in Taiwan.

In addition, we have an altitude issue from the global point of view.
Several laboratories confront issues related to high altitudes above sea level. CEIS in Spain is located 650 m above sea level, for example, there was an 8% disparity between the capacities measured by the manufacturers and CEIS. These laboratories are located at high altitudes where the air pressure is low. As a result, the air mass flow rate of the outdoor/indoor units which apply DC fan motor is diminished and the cooling cycle does not function properly. Consequently, the performance of air conditioners degrades. Since there are some other testing laboratories located at high altitudes, this problem has to be taken into consideration carefully in the future.

Figure 5.3 Test institutes around Japan

SUMMARY

The followings are summary about features in APF measurement.

1. The balanced ambient room-type calorimeter or the air enthalpy test method (tunnel air enthalpy measurement equipment) is mainly adopted in test facilities in Japan and makes it possible to keep high accuracy. On the other hand, the calibrated room type calorimeter is not adopted for possible heat leakage.
2. A concrete calculation method to estimate uncertainty is established based on the JAB guidelines and ISO/TS 16491, and is practiced for ensuring accuracy as is indicated in ISO17025. Uncertainty level of manufacturer's certificated test equipment is considered as the same level of the JATL's.
3. Manufacturers are allowed to test all their models by their certified test equipment. Random performance test on the products from the market is conducted to maintain product quality by JATL.

This table also summarizes the APF measurement features.

Table 5.2 Comparative Table about Performance Measurement

	APF (year-round performance factor), Japan	EN14825 SPF	ANSI/AHRI210/240 APF
Requirement of measurement	Required		
Tolerance of the measurement value	Power consumption: below 103% Capacity: over 97% APF: over 100%		
Inspection institute	Independent inspecting agency : Japan Air Conditioning and Refrigeration Test Laboratory (JATL)		
Calibration	Done at a manufacturer's facility and JATL's one (original facility) followed by an on-site quality check by JATL. (ISO 17025)		
Product inspection	Products from the market are inspected by JATL		
No. of inspected products	One or more products a year per manufacturer		
Round robin test with foreign test institute	Having done round robin tests with test institutes in several foreign countries.		
Uncertainty evaluation	JATL estimates uncertainty properly based on JAB guidelines (JAB NOTE 6) which are to be standardized in ISO.		

Thanks to precisely established calculation methods and test methods, it has been recognized that the APF indexes are reliable and fair, and they are widely used in Japan today. As a labeling index, the APF indexes contribute to society with the top runner program which is implemented by the Japanese government to promote the efforts in energy conservation.

6 TECHNICAL PAPER OF "NEW CALCULATION METHOD OF ANNUAL PERFORMANCE FACTORS FOR VRF SYSTEMS"

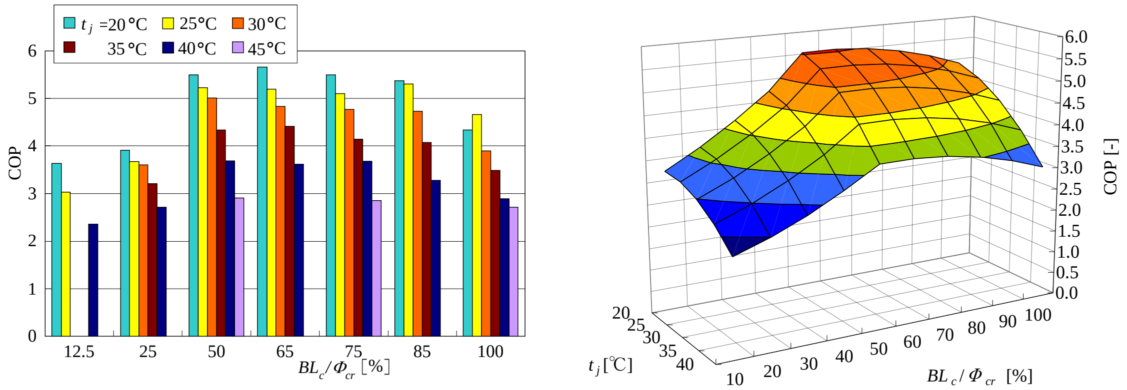
6.1 Introduction

In Japan, the calculation methods of the annual performance factors (APFs) for air conditioners that take into account their partial load performances are specified in JIS C 9612: 2013 for room air conditioners with the rated cooling capacities of less than 10 kW and in JIS B 8616: 2006 for electrically driven packaged air conditioners with the rated cooling capacities of less than 28 kW. As for electrically driven multi-split type air conditioners for building, i.e., VRF systems, with larger capacities, however, the calculation method of APFs is not established yet. For an accurate calculation of APFs for VRF systems, we need to know their detailed partial load performance. It is, however, quite difficult to make clear the whole aspect of their partial load performance because it changes in a complex manner depending on indoor thermal load and outdoor air temperature. Accordingly, in order to develop a calculation method of APFs for VRF systems, we have to first make clear their detailed COP characteristics in partial load operation by the partial load performance test in the test facility. Here, we introduce the new calculation method of APFs for VRF systems proposed based on results of the performance test.

6.2 Results of partial load performance tests

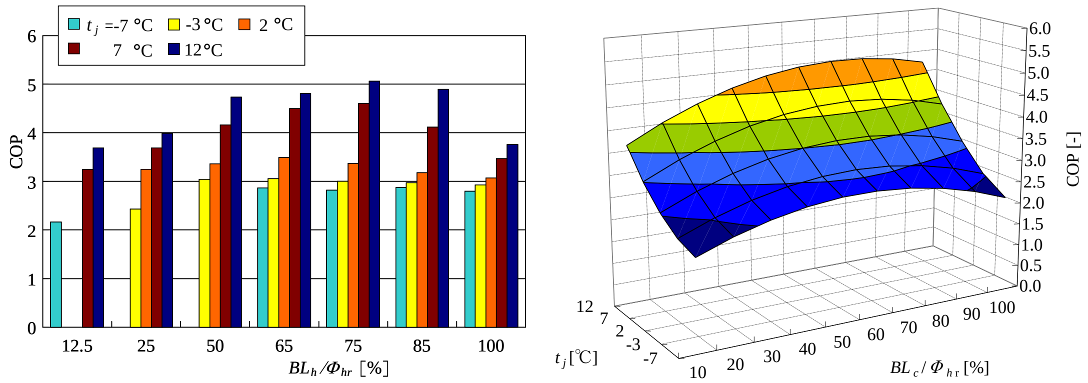
As an example of the results of the partial load performance tests, we present COPs measured with the electrically driven VRF system, manufactured in 2011, with the rated cooling and heating capacities of 45 kW and 50 kW, respectively. In the cooling performance test, dry-bulb air temperature in the outdoor test chamber t_j was raised from 20°C to 45°C, and the indoor cooling load BL_c was raised from 12.5% to 100% of the rated cooling capacity of the air conditioner Φ_{cr}. Air temperature in the indoor test chamber was controlled at 27°C by the tested air conditioner itself to simulate its actual working condition in a building. With this test method, the on-off operations of the compressors that occur under a low load condition can be reproduced in the test facility. Cooling COPs were measured at 36 test points. For the heating performance test, COPs were measured at 31 test points, while maintaining the indoor test chamber air temperature at 20°C. Detailed conditions and procedure of the partial load performance test are described in the reference.

Figure 6.1 (a) shows cooling COPs measured in the partial load performance test. The abscissa shows the cooling load ratio BL_c / Φ_{cr} and the parameter is the outdoor air temperature t_j. For all values of BL_c/Φ_{cr}, COP decreases as t_j rises. At constant t_j, COP shows high values in BL_c/Φ_{cr} = 50 - 75% and decreases as BL_c/Φ_{cr} lowers further. This deterioration of COP in the low thermal load operation was caused by intermittent on-off operations of the compressors. Based on these COP data, we can plot the cooling COP surface of this VRF system as shown in Figure 6.1 (b) by expressing COPs as a function of t_j and BL_c/Φ_{cr}.



(a) Results of cooling COPs (b) Cooling COP surface

Figure 6.1 COPs in cooling mode in partial load performance test



(a) Results of heating COPs (b) Heating COP surface

Figure 6.2 COPs in heating mode in partial load performance test

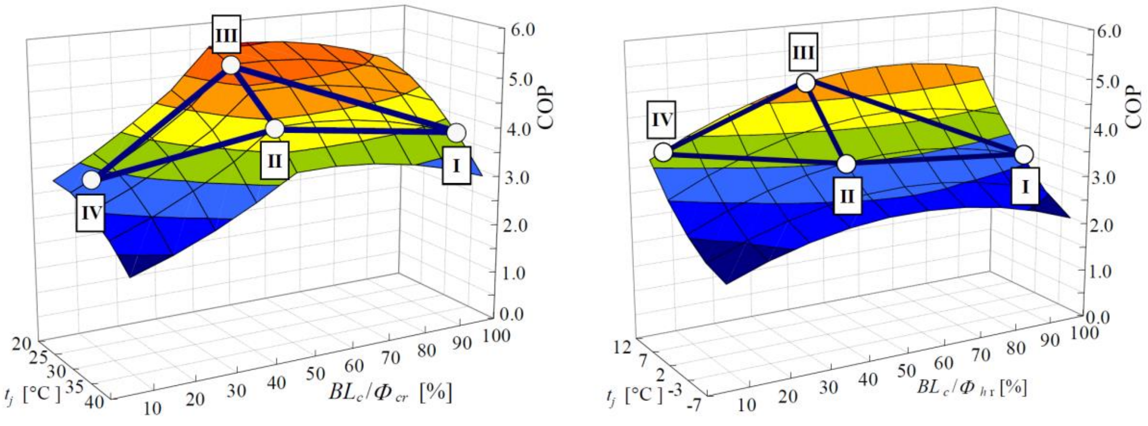
Figure 6.2 shows the results of heating COPs. In the same way as for the cooling COP, the heating COP shows high values in the intermediate heating load range and decreases at lower values of BL_h/Φ_{hr}.

By combining these COP surfaces, the thermal load of a building and load appearance hours in a year, we can determine APF of this VRF system. In order to calculate APF accurately without spending much time and efforts, it is necessary to reproduce such dependence of COP on t_j and BL_c/Φ_{cr} or BL_h/Φ_{hr} as shown above with the smaller number of test points. We have proposed a simple method to reconstruct the COP surfaces of the VRF system based on COPs measured at eight test points in the partial load performance test.

6.3 Concept of new calculation method of APFs

Figure 6.3 (a) shows the concept of the proposed method to reconstruct the cooling COP surface with four test points. The numbers [I] - [IV] on the COP surface designate COPs measured under corresponding t_j and BL_c/Φ_{cr} in the partial load cooling performance test. Considering that cooling COP of the VRF system becomes the maximum in the intermediate thermal load ratio, we divide the COP surface into two regions setting their boundary in the mid thermal load region (the line connecting [II] and [III] on the COP surface). The COP surface in the higher BL_c/Φ_{cr} region is approximated by the plane that contains [I, II, III], and that in lower BL_c/Φ_{cr} is approximated by the plane containing [II, III, IV]. By reconstructing the COP surface with these two planes, the deterioration of COP in the low thermal load operation and the dependence of COP on t_j can be reproduced approximately. The concept for the heating COP is shown in Figure 6.3 (b). Similar to the cooling COP, the heating COP surface is approximated by two planes formed by [I, II, III] and [II, III, IV]. Therefore, totally eight test points of the partial load performance test are needed in the proposed method to express the variation of COP of the air conditioner in a year.

The test points were selected considering that they coordinated with thermal loads of various buildings. Table 6.1 shows the selected test points, and the relationship between these test points and the indoor thermal load models for buildings is shown in Figure 6.4. The indoor thermal loads are expressed as functions of outdoor air temperature t_j. Appearance hours t_j in a year in Nagoya city located in central Japan are also shown by histograms. In addition to the thermal load models for offices and detached shops prescribed in JIS B 8616: 2006, those of home appliance mass merchandisers, food supermarkets and offices obtained by field measurement are shown in Figure 6.4. It is assumed that the maximum thermal load appears in cooling season for a home appliance mass merchandiser and an office, and BL_c/Φ_{cr} = 100% at t_j = 40°C. In a food supermarket, the maximum thermal load is assumed to appear in heating season and BL_h/Φ_{hr} = 100% at t_j = -5°C. Red symbols in the Figure 6.4 correspond to the test points shown in Table 1. The regions enclosed by the selected test points can cover the major parts of the thermal load models for these buildings.

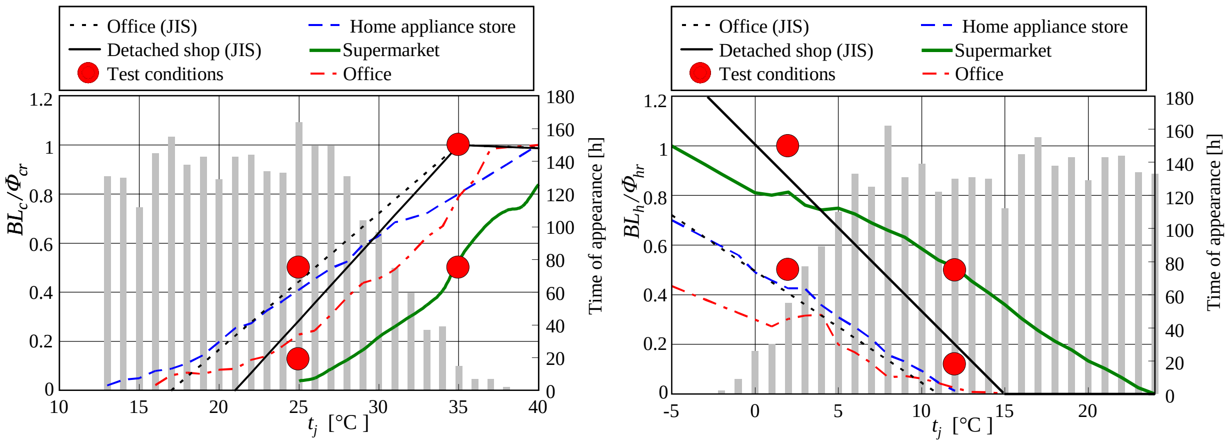


(a) Cooling COP (b) Heating COP

Figure 6.3 Concept of proposed reconstruction method of COP surfaces

Table 6.1 Test points selected in the proposed calculation method of APF

	Cooling tests		Heating tests	
No.	Outdoor air temp.	Thermal load ratio	Outdoor air temp.	Thermal load ratio
I	35°C	100%	2°C	100%
II	35°C	50%	2°C	50%
III	25°C	50%	12°C	50%
IV	25°C	12.5%	12°C	12.5%



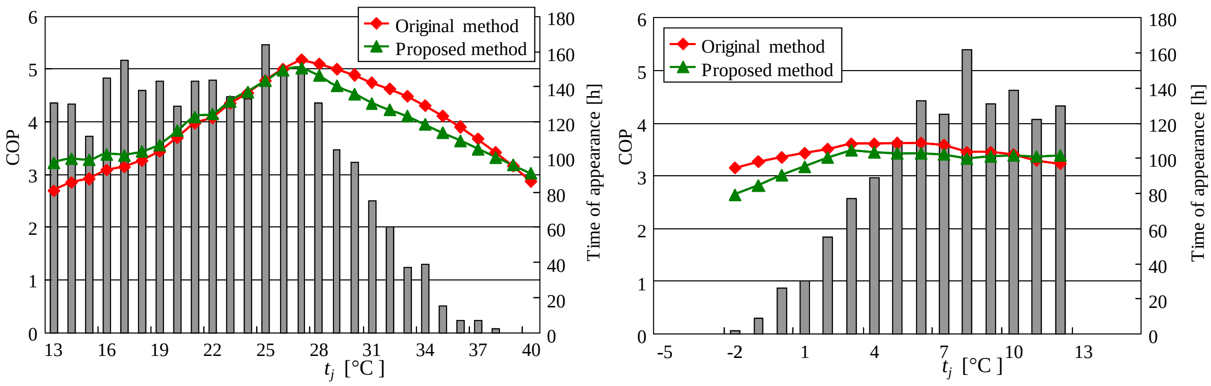
(a) Cooling load (b) Heating load

Figure 6.4 Relationship between indoor thermal load models and selected test points

6.4 Results of COP/APF calculations

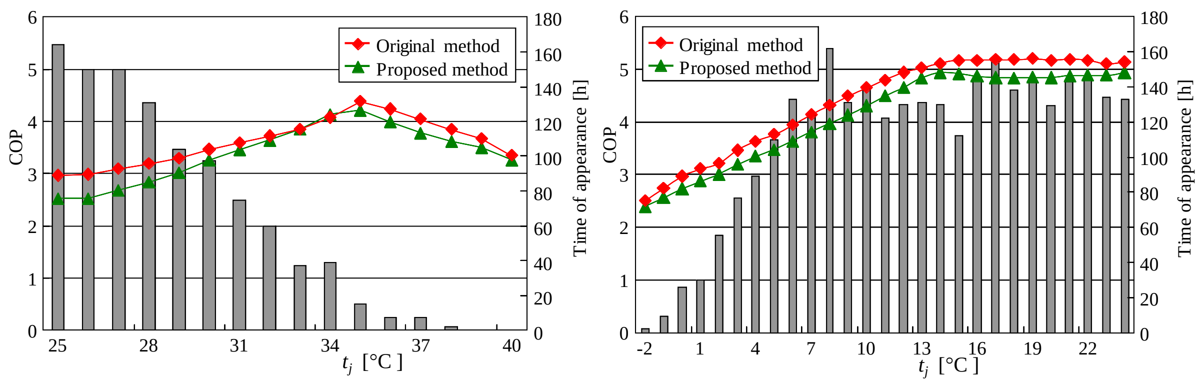
The validity of the new calculation method of APFs proposed here was evaluated by comparing COPs and APFs calculated by this method with those obtained from the original COP surfaces shown in Figures 6.1 (b) and 6.2 (b). Figures 6.5 and 6.6 show variations of COP with t_j calculated based on the original COP surfaces (red lines) and on the approximate ones reconstructed by the proposed method (green lines). The building is used by a home appliance mass merchandiser (Figure 6.5) and a food supermarket (Figure 6.6). Appearance hours of t_j in a year are also shown in these figures. It should be noted here that, since BL_c/Φ_{cr} and BL_h/Φ_{hr} are given as functions of t_j as shown in Figure 6.4, COPs shown in these figures reflect the influences of both t_j and corresponding BL_c/Φ_{cr} or BL_h/Φ_{hr}. Although the proposed method slightly underestimates COPs, the COPs calculated by the proposed method agree well with those calculated from the original COP surfaces.

In Figure 6.7, the seasonal performance factors (CSPFs and HSPFs) and APFs calculated by the proposed method are compared with those obtained from the original COP surfaces.



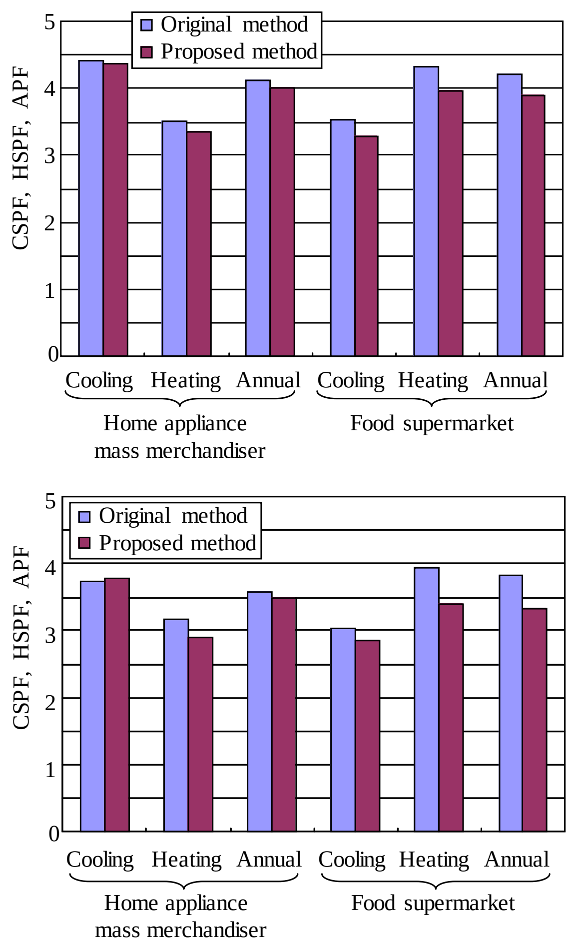
(a) Cooling COPs (b) Heating COPs

Figure 6.5 Comparison of calculated COPs (home appliance mass merchandiser)



(a) Cooling COPs (b) Heating COPs

Figure 6.6 Comparison of calculated COPs (food supermarket)



(a) Machine A, (b) Machine B

Figure 6.7 Comparison of seasonal and annual performance factors of VRF systems

The results for the two different VRF systems, Machines A and B, are presented in this figure. APFs calculated by the proposed method and those determined from the original COP surfaces agree with each other with differences less than 8%. These results prove that the new calculation method of APFs proposed here can predict APFs of the VRF systems accurately with the relatively small number of test points of the partial load performance test.

6.5 Concluding remarks

We have introduced a new calculation method of the annual performance factors proposed for VRF systems with relatively large capacities. The COP surfaces, which show detailed variations of COP with the outdoor air temperature and the indoor thermal load, are reconstructed based on COPs measured at eight test points in the partial load performance test. It has been proved that the proposed method can calculate APFs for VRF systems with errors less than 8%.

6.6 Reference

Miyaoka, Y., Iwata, Y., Nakayama, H., Hirota, M. and Onishi, M. 2014 “Evaluation of Annual Energy Consumption of Multi-split Type EHP Air-conditioner for Buildings,” Proc. 11th IEA Heat Pump Conference, Montreal, paper O.2.4.2, on CD-ROM.

7 FIELD EXPERIMENTAL EVALUATION OF AIR-CONDITIONING SYSTEM PERFORMANCE RESULTS REPORT (SUMMARY)

7.1 Purpose and methodology

7.1.1 Purpose

To assess the adequacy of energy consumption evaluation methods for room air conditioning systems applied by residential energy consumption standards, we gathered measurement data about the actual operational characteristics of air conditioning systems which were close to actual use conditions. These measurements were carried out during the period from July 2010 to February 2011.

7.1.2 Methodology

Six air conditioners (Figures 7.2 and 7.3) with attached sensors were installed in a house (Figure 7.1 and Table 7.1). The air conditioners were operated by turning on and off under the assumption of a five-person family, in accordance with a schedule intended to model that of such family. The heat load of the house was simulated by installing electrical appliances, lighting devices in rooms, and systems to simulate heat generation by human bodies. Additionally, the air conditioner capacity for each room was selected based on the size of the room in which it was placed (measured in terms of *jousuu*, the number of tatami mats in a room). The heat loss coefficient of this house is 3.92W/m2K, which is normal value in the warm-temperature region in Japan. The rated capacities of these six room air conditioners were selected based on JEM 1447 calculating methods of cooling and heating area for room air conditioners which is normally used for selecting the capaity of room air conditioners when installed.

7.1.3 Results of this report

This report describes the results of System 1, a large air conditioning system and System 4, a small one as representative of the rest. The specifications of these systems are collected in Table 7.2.

Figure 7.1 Outside view of the house

Table 7.1 Specifications of the house

Structure	Two-story wooden house
Total floor space	Total: 134.98 m^2 First floor: 80.33 m^2 Second floor: 54.65 m^2
Insulation	Floor: styrofoam 40 mm Ceiling: blowing GW 210 mm Outer walls: glass wool 10 K 50 mm
Specific heat loss coefficient (Q value)	3.92 W/ (m^2・K)

Table 7.2 Specifications of the representative air conditioners

	Cooler		Heater		AFF
	Rated capacity [kW] (Range)	COP [-]	Rated capacity [kW] (Range)	COP [-]	
System 1	6.3 (0.6 - 6.5)	2.90	7.1 (0.6 - 10.4)	4.18	5.00
System 4	2.2 (0.7 - 3.3)	5.57	2.5 (0.7 - 5.4)	6.41	6.10

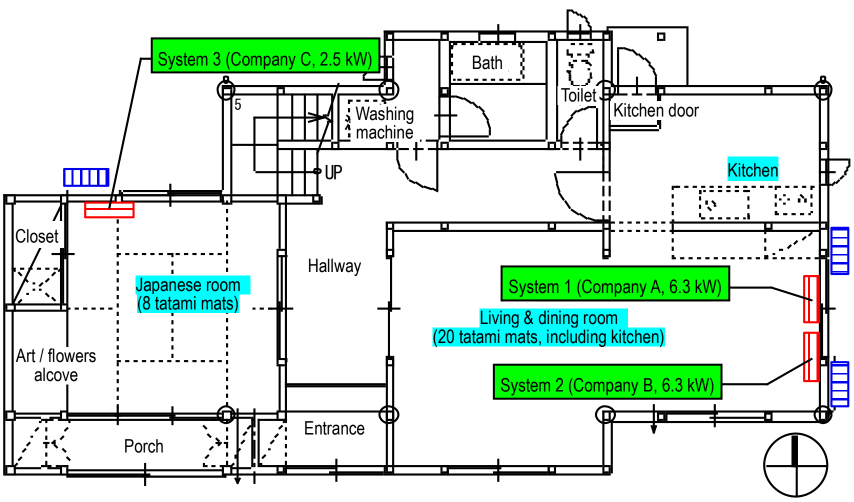


Figure 7.2 First floor ground plan and air conditioner locations

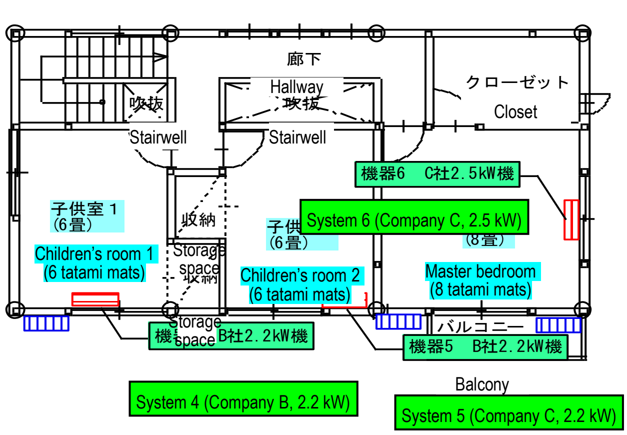


Figure 7.3 Second floor ground plan and air conditioner locations

7.1.4 Method of calculating the amount of heat generated

The amount of heat generated by an air conditioner was calculated utilizing the fan's air flow rate and a sensor placed between the intake and outflow vent. Outflow temperature and humidity were taken as an average of measurements on the left side, center, and right side of the outflow vent.

Air flow was estimated with a formula which relates the number of fan revolutions to air flow. The relationship was obtained beforehand by using highly accurate measuring tools.

The humidity sensor partially placed on the outflow vent (Figure 7.4) blocked air output and dampened the air flow. The specific reduction in air output is discussed in a later section.


Figure 7.4 Sensor on the outflow vent

7.2 Results (cooler)

7.2.1 Large air conditioner

Regarding the large air conditioner (System 1: 6.3 kW) installed in the living room, load factor and the number of hours are indicated in Figure 7.5. And, the relationship between load factor and COP is shown in Figure 7.6. As Figure 7.5 shows, most of operating time is spent at load factors of 15 - 25%. COP (efficiency) is high in these time periods, indicating that the system operates efficiently.

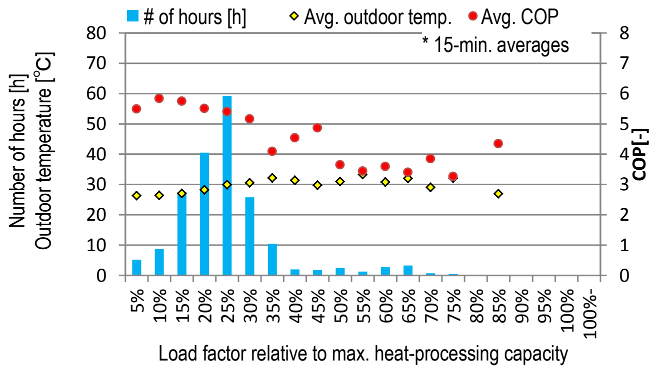


Figure 7.5 Load factor and number of hours (System 1: cooler)

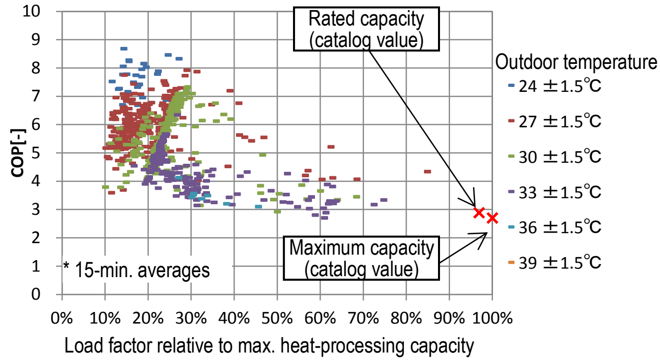


Figure 7.6 Relationship between load factor and COP (System 1: cooler)

7.2.2 Small air conditioners

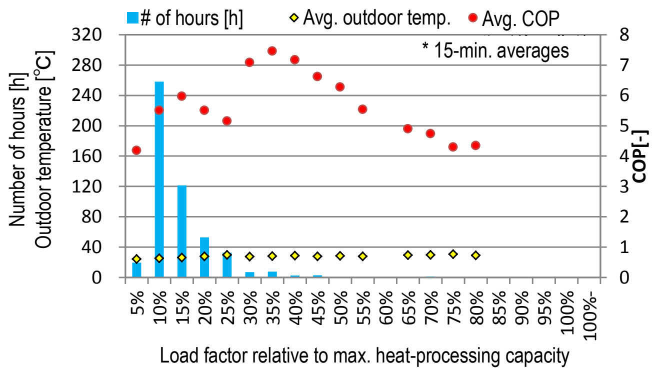


Figure 7.7 Load factor and number of hours (System 4: cooler)

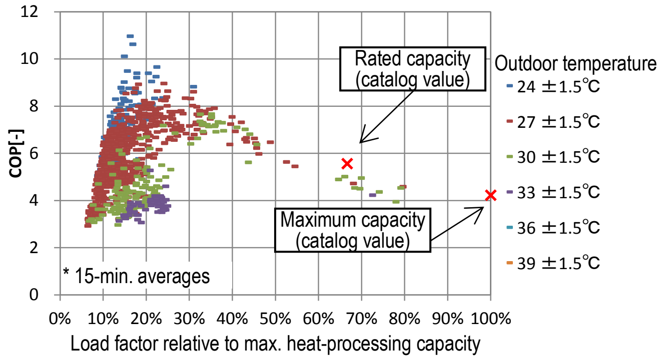


Figure 7.8 Relationship between load factor and COP (System 4: cooler)

7.3 Load on the air conditioner system

7.3.1 Relationship between outdoor temperature and load factor

Generally, load factor correlates well with outdoor temperature. However, the data of experiment in 15-minute intervals appear widely scattered in Figure 7.9 which shows the relationship between outdoor temperature and load factor. The reason is that room conditions affect system load in different ways, even at the same outdoor temperature. For example, the compressor stops working when room temperature reaches the set value, and under these conditions, low load factor is achieved at any outdoor temperature.

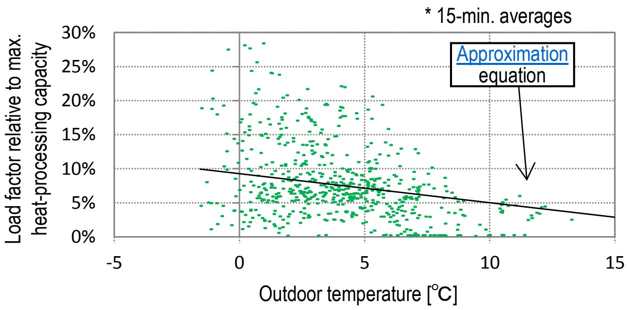


Figure 7.9 Relationship between outdoor temperature and load factor (System 4: heater)

■ Load factor after starting operation

As shown in Figure 7.10, load factor becomes high immediately after starting operation of the air conditioner system, but it shows a downward trend as the system is continuously used. Immediately after starting operation, there is a large difference between room temperature and the set value, leading to an operation in a high load factor to quickly reach the set value. After the attainment, the system turns to a low load factor.

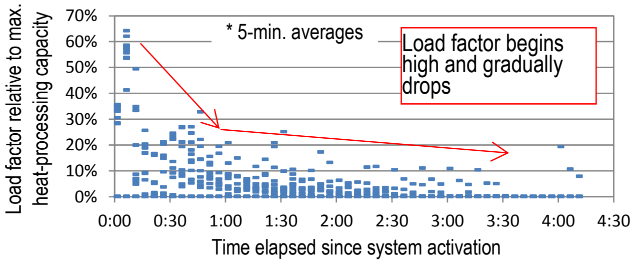


Figure 7.10 Load factor over time after system activation (System 4: heater)

7.4 Results (heater)

7.4.1 Large air conditioner

Regarding the large air conditioner (System 1: 6.3 kW) installed in the living room, load factor and the number of hours are shown in Figure 7.11. And, the relationship between load factor and COP is shown in Figure 7.12. In Figure 7.11, most of operating time is spent at load factors of 15 - 25%. COP (efficiency) is high in these time periods, and hence the system operates efficiently.

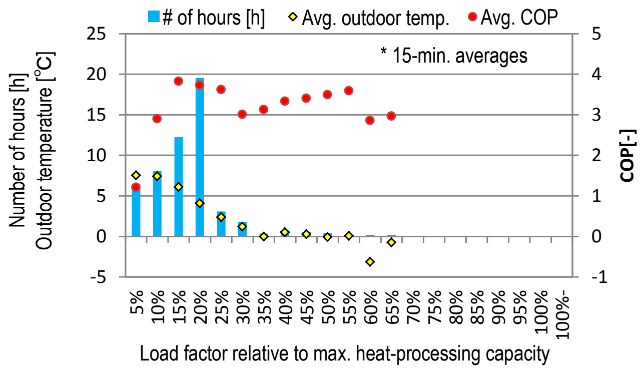


Figure 7.11 Load factor and number of hours (System 1: heater)

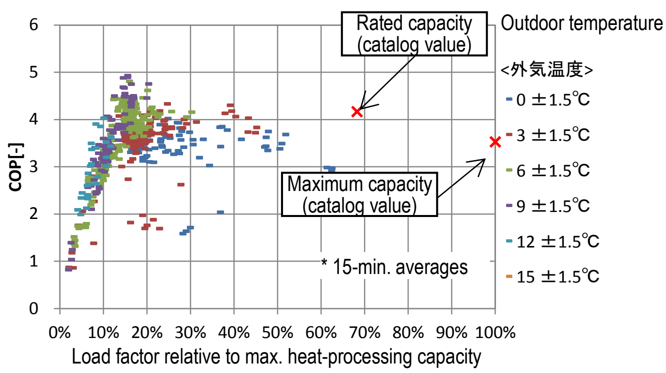


Figure 7.12 Relationship between load factor and COP (System 1: heater)

7.4.2 Small air conditioner

Regarding the small air conditioner (System 4: 2.2 kW) installed in the children's room, load factor and the number of hours are shown in Figure 13. And, the relationship between load factor and COP is shown in Figure 7.14. As Figure 7.13 indicates, most of operating time is spent at load factors of 5 - 10%. COP (efficiency) is generally low, particularly at a load factor of 5% where COP is 1. COP (efficiency) is relatively lower in heater mode than in cooler mode. In particular, heater mode involves defrosting (a system process of removing ice from outdoor units), resulting in a very low efficiency.

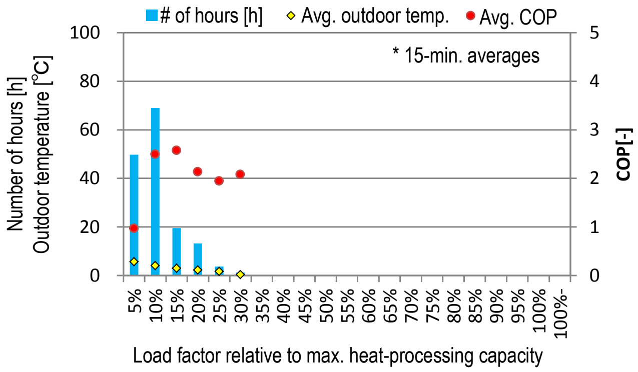


Figure 7.13 Load factor and number of hours (System 4: heater)

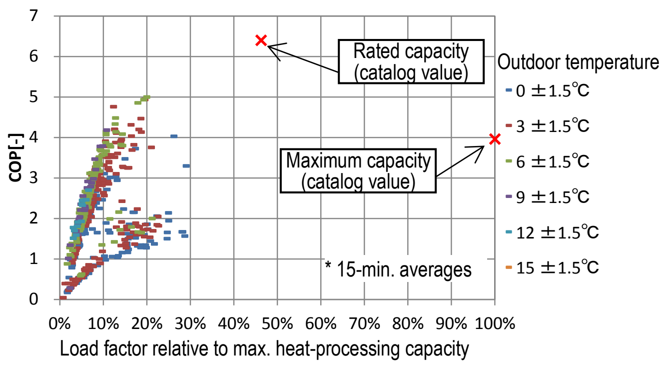


Figure 7.14 Relationship between load factor and COP (System 4: heater)

7.5 Actual room temperature of System 4 in heater mode

The actual performance of System 4 on January 7, when load factors in heater mode were extremely low, is shown in Figure 7.15. The room temperature reached 20℃ 15 minutes after system operation. Afterward, the system works intermittently to maintain room temperature at around 20°C.

Amt. of heat processed [W]

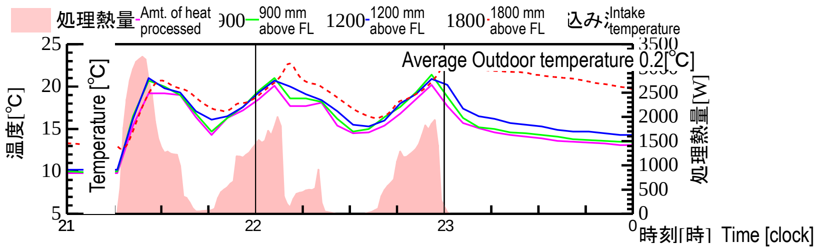


Figure 7.15 Operation data of System 4 (heater)

7.6 Reduced air flow due to the outflow vent sensor

Air flow from the outflow vent was reduced by the sensor installed on the outflow vent (Figures 7.16 and 7.17). For this reason, it is necessary to account for reductions in air flow (heating or defrosting) and electricity consumption. In general, lower air flow leads to a drop in efficiency.

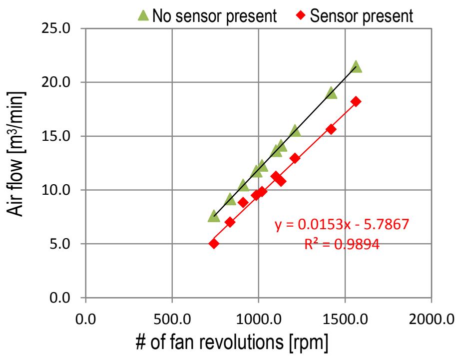


Figure 7.16 Differences in air flow by sensor presence (System 1)

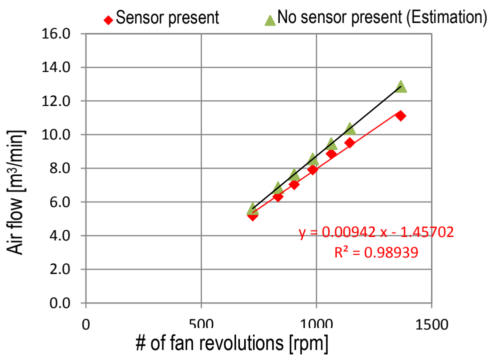
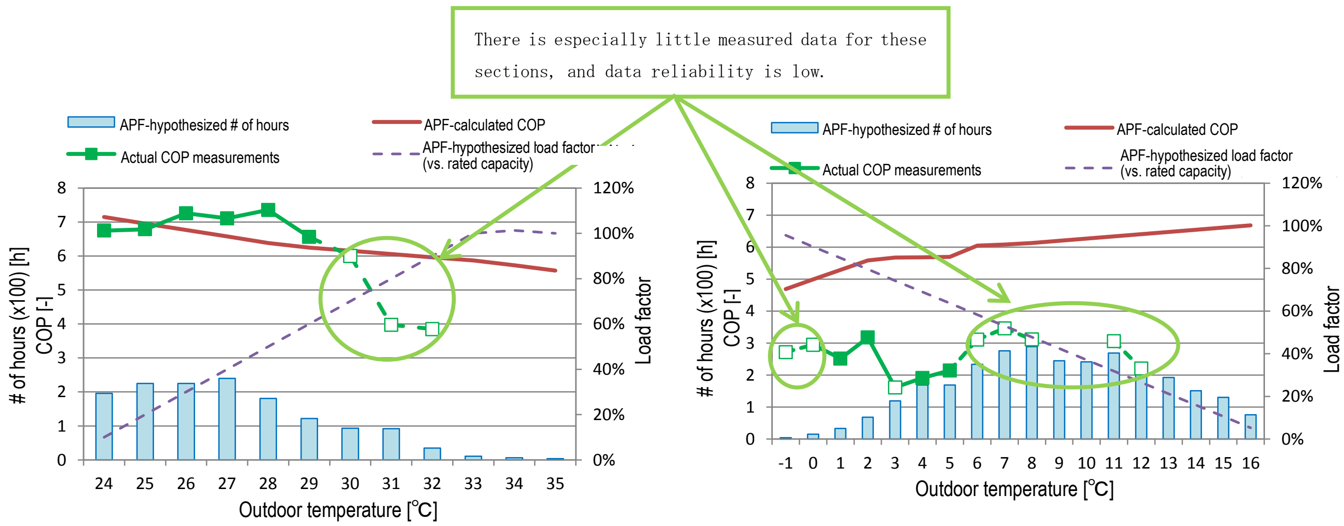


Figure 7.17 Differences in air flow by sensor presence (Figure 4: top of flap)

7.7 Comparison of APF calculations under JIS and actual measurements

Annual performance factor (APF) is currently used as an indicator for air conditioner performance. However, several issues have been raised regarding APF, such as

differences between actual use conditions and the condition for measuring air conditioning system efficiency which is the basis for calculating APF or differences between actual measured values and calculated values of air conditioner efficiency, which are calculated under several load conditions. Figure 7.18 compares air conditioner efficiency calculated by using APF (red) with efficiency based on actual measured values (green). It shows that the efficiency of an air conditioner system in heater mode calculated by APF is higher than the actual measured values (green).
Regarding cooler mode, “APF-calculated COP (red)” and “measured COP (green)” are at approximately equal levels.
Regarding the heater mode, there is a large difference between “APF-calculated COP (red)” and “measured COP (green)”.



(a) Cooler mode (b) Heater mode

Figure 7.18 Comparison of COP calculated by using APF and actual measured COP (System 4)

Notes concerning data interpretation: 1) Measurement data contains various measurement errors, and efficiency measurements may be lower (or higher) than actual efficiency. 2) The actual COP measurements (green) shown here are data aggregated every 15 minutes and include the immediate start time period and time periods in which the air conditioner is in transition. APF calculations assume that load is applied steadily throughout a given hour, and this differs from actual conditions. 3) Actual COP measurement data-points marked with white squares (□) indicate that little data was available, whose reliability is low. 4) The set value of room temperature in the actual measurement was 26°C in cooler mode, which differs from JIS standards (27°C). For heater mode, the set value of room temperature was 20°C, which is equal to JIS standards.

7.8 Summary

The following conclusions were obtained through analysis of experimental results. The effect of sensors in front of the outflow vent on output air flow is compensated.

・This air conditioning system model shows a better performance than catalog (rated) COP in cooler mode, excluding intermittent operation periods.

・This system shows better performance in cooler mode than in heater mode. Additionally, the small air conditioners have a higher catalog COP (COP = 6.41) than that of the large air conditioner (COP = 4.18), while the small air conditioners show a lower COP.

・Although the outdoor temperature has essentially an effect on air conditioner load, the effect of the relationship between room temperature and the set value of temperature is much stronger. As the room temperature approaches to the set value of room temperature, the systems often entered low-load intermittent operation.

・APF is an indicator of air conditioner performance and is useful for comparison between system models. However, there are some aspects in APF that should be reconsidered in evaluating actual performance in real houses, such as assumptions of intermittent operation and load.

・Performance test results of mid-and low-capacity system models agree with the results derived from APF calculation which is used to calculate residential "Top Runner" (best-in-class performance) standards. However, the test results for high-capacity system models show that they consume less energy than predicted.

This test was conducted as a part of work financed by New Energy and Industrial Technology Development Organization (NEDO), entrusted to the Building Research Institute by the University of Tokyo.

8 OUTLINE OF THE BUILDING ENERGY STANDARD IN JAPAN AND EVALUATION OF HEAT PUMPS IN THE STANDARD

8.1 Summary/abstract

Primary energy consumption in the commercial and residential building sectors in Japan represents approximately 34% of the country's total energy consumption and has been increasing steadily. With the aim of decreasing energy consumption, a new building energy standard was implemented in 2013 and the government has announced that it will become mandatory by 2020. This article presents an outline of the standard and how energy performance of heat pumps is evaluated in it.

8.2 Outline of building energy standard

In Japan, the Energy Conservation Act was implemented in 1979 and the first Building Energy Standard was enforced in 1980 under the act. The standard consisted of two parts: one for commercial buildings and the other for residential buildings. The commercial buildings standard evaluates the energy performance of the building envelope and five types of energy demands for air conditioning, lighting, ventilation, hot water and elevators, separately. The residential standard evaluates only the performance of the building envelope. These standards were substantially revised in 2013 and introduced a new common index "designed primary energy consumption" (GJ/year) (Figure 8.1) to represent the energy performances of both commercial buildings and residential buildings. When constructing or renovating a large building with a floor area of 300 m^2 or more, it is mandatory to report the energy performance based on the standard. On the other hand, such reporting is voluntary for small buildings. However, buildings which apply the standard can receive various benefits such as tax benefits and subsidies. Although these energy standards are applied on a voluntary basis for some buildings at present, compliance with them will be mandatory by 2020.

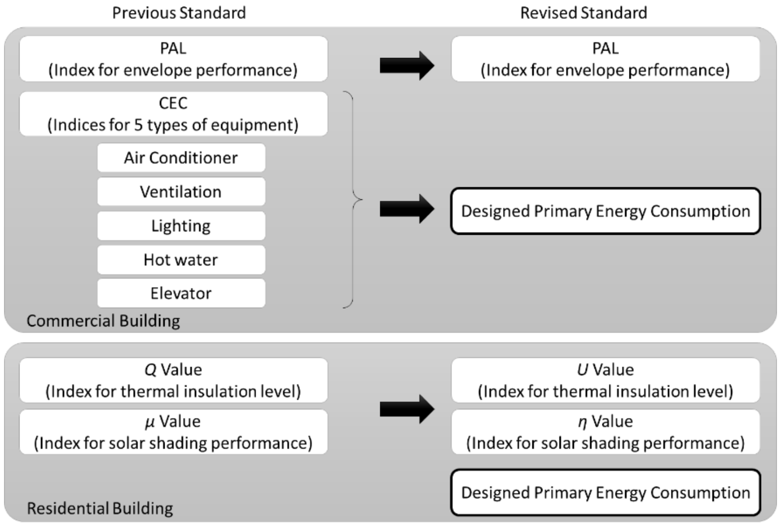
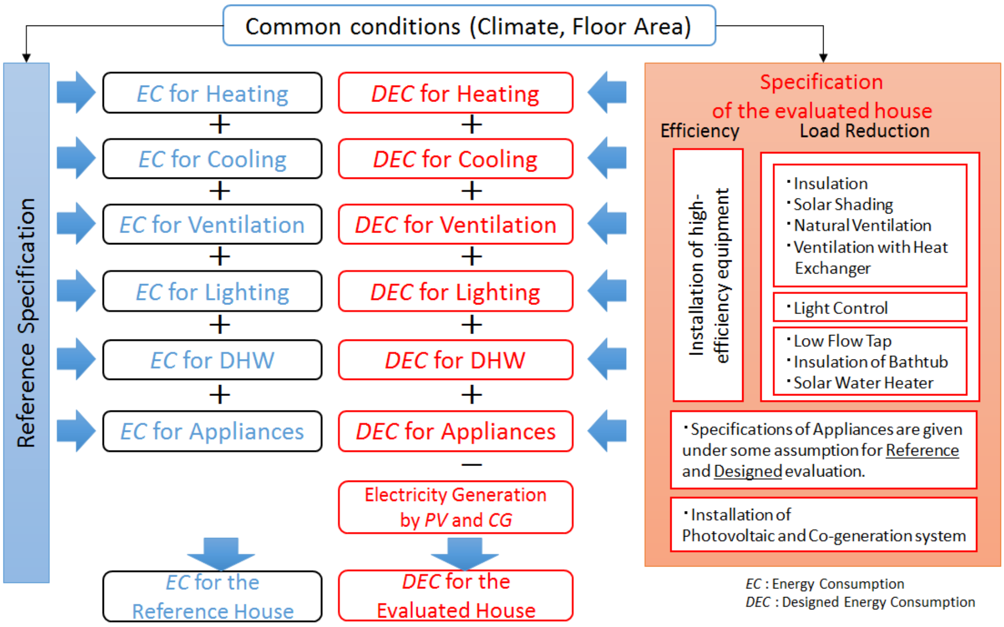


Figure 8.1 Indices for energy performance in the

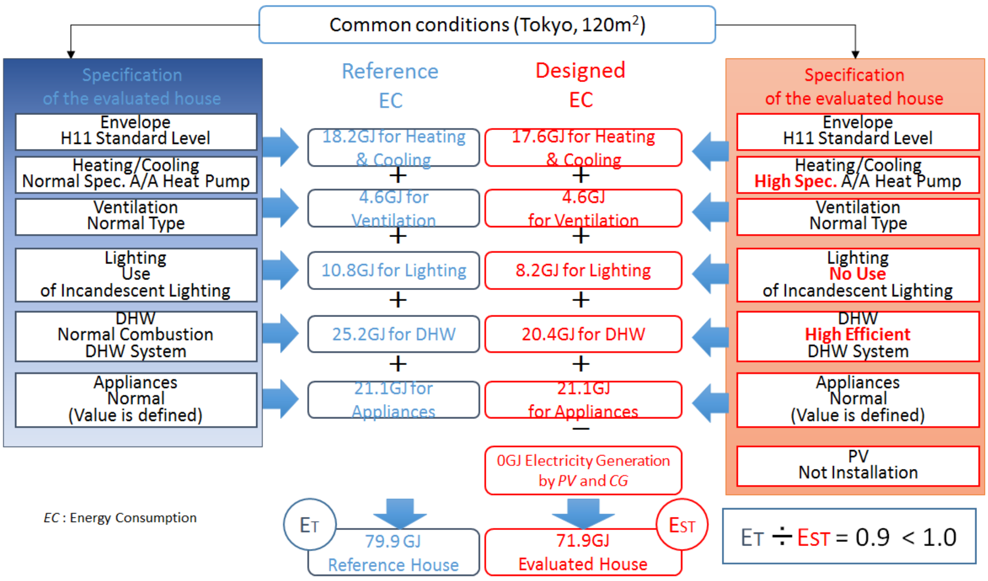
8.3 Design primary energy consumption

In order to comply with the standard, the design primary energy consumption calculated for an applicable building must not exceed the reference primary energy consumption. Both the design and the reference primary energy consumption are based on the same calculation method. The design primary energy consumption must be calculated based on the building envelope and equipment specifications of the building, while the reference primary energy consumption must be calculated based on the building envelope and equipment specifications defined in the standard, which are of equivalent performance to the corresponding features of the building as normally used in Japan (Figure 8.2). Occupants' behavior such as occupation period in a day and duration and frequency of use of the building services is defined in advance for each building/space usage in the standard, and it can, therefore, be said that the design primary energy consumption is a kind of benchmark test value for energy conservation at design stage.

Methods for calculating the design primary energy consumption are mentioned in textbooks and on the web page of the Building Research Institute (BRI) (Figure 8.3). Since these methods are very complex, web-based calculation programmes have also been developed and are available on the BRI home page (Figure 8.4). The programmes are hosted on web servers and local computers simply send the input data to the servers and receive the results from them, which means that designs can be calculated by any computer including tablet-type devices. Applicants normally make calculations and prepare application forms by utilizing these programmes.



a) Structure



b) Example

Figure 8.2 Reference and design primary energy consumptions*

* For residential buildings. For commercial buildings, the energy consumption for elevators is added.

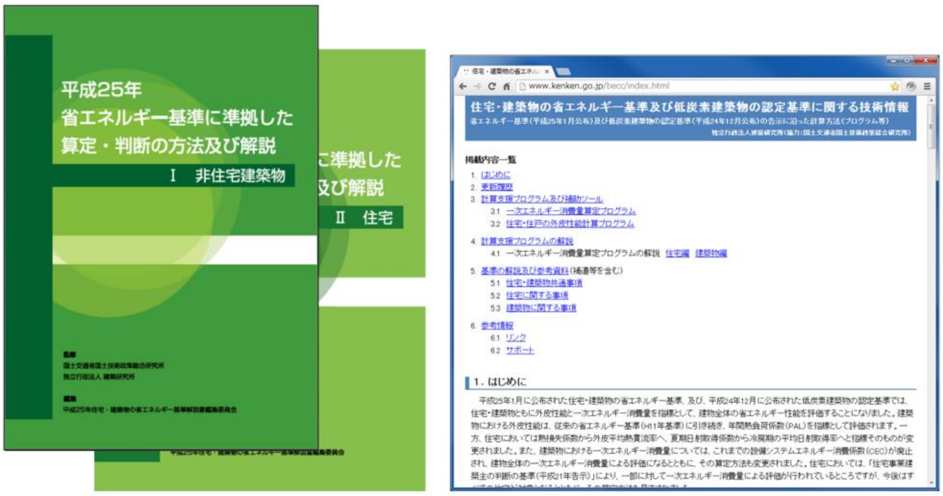


Figure8.3 Textbooks and BRI web page* for the standard

* http://www.kenken.go.jp/becc/

Figure 8.4 Calculation program, application form and labeling

8.4 Performance of heat pumps

Energy consumption for space heating and cooling is calculated on an hourly basis, with design primary energy consumption being the sum of these hourly energy consumptions over a period of one year (Figure 8.5). Since the efficiency of heat pumps depends strongly on the outdoor temperature and the heating or cooling load, efficiency curves for various systems are defined in the standard. These efficiency curves were developed based on various experiments and simulations (Figure 8.6) in a research project by BRI and the National Institute for Land and Infrastructure Management (NILIM). Outdoor temperatures are also defined in the standard. Building areas in Japan are divided into eight climate zones based on the number of heating degree-days and the tables of standardised hourly outdoor temperatures are updated for each revision. The hourly heating and cooling loads are calculated based on the data given in the standard for the building envelope performance and the intended use of the building or space.

Based on this calculation, the differences in efficiency of heat pumps between the different climate zones can be considered. The reduction in average efficiency resulting from mismatching of the heat pump capacity with the heat load including overcapacity of the equipment can also be evaluated. Additionally, operating conditions such as climate and building usage are identical for various types of the equipment (e.g. gas boilers, heat pumps, electric heaters), thus enabling users and designers to compare the energy performance of any types of equipment by using this calculation program.

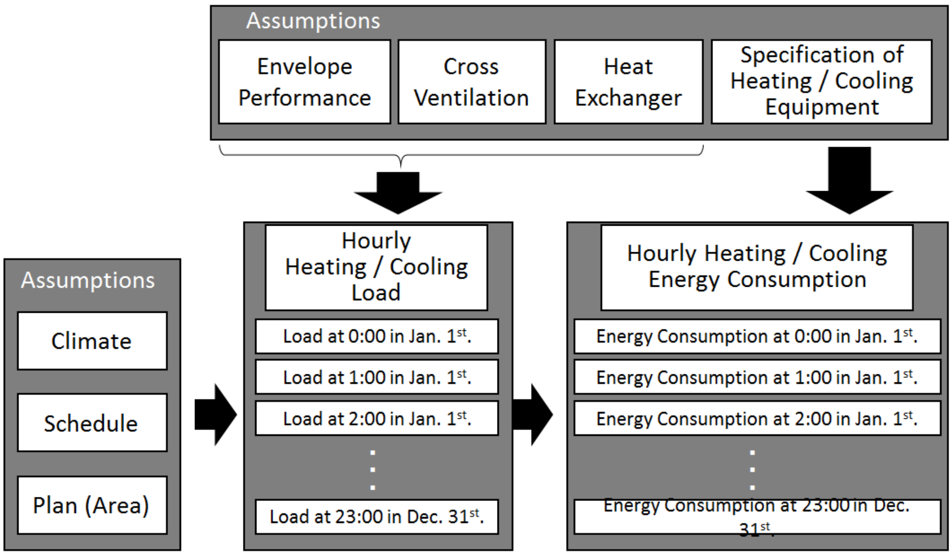
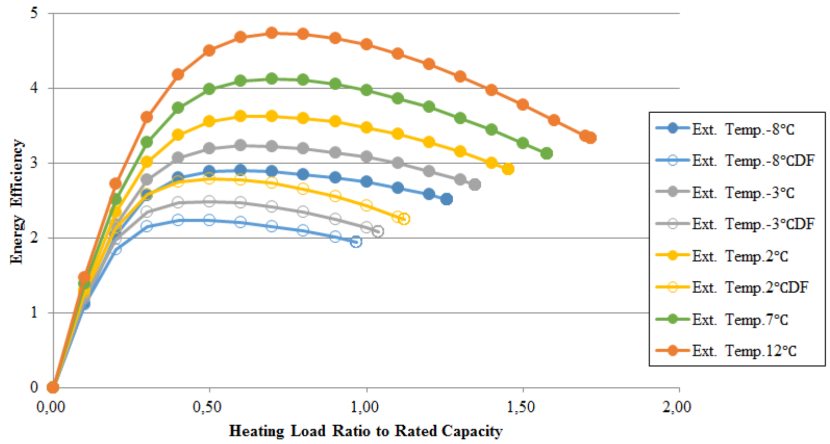


Figure 8.5 Scheme for calculation of space heating/cooling energy consumption



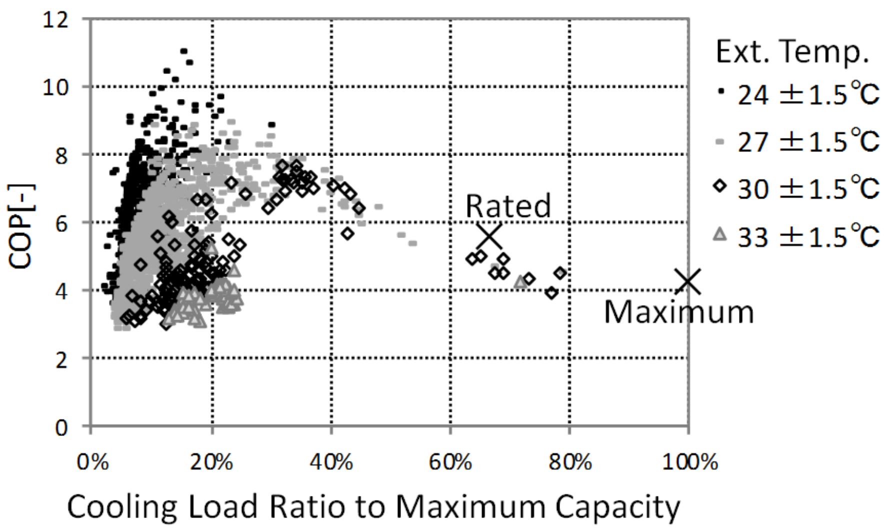


Figure 8.6 Example of heat pump efficiency curves

(Top: curves for calculation, bottom: experimental results)

8.5 Conclusions

This report describes the outline of the recently revised building energy standard and its energy calculation tools as well as how to estimate the energy performance of heat pumps.

There are many indices to represent the energy efficiency of heat pumps such as APF, COP and EER. These indices can be measured values under the rated test conditions or the calculated values assuming the given operating conditions. Alternatively, by applying this energy calculation tools, designers can investigate the effects of changing various building characteristics such as envelope performance, climate and utilization of cross ventilation and heat exchangers, and can also compare the performance of various types of heating systems such as combustion boilers or electric heaters.

We can say that the indices based on a specific and defined set of conditions, such as APF, COP and EER, can be used for comparing the performance of heat pumps of the same type and thus promoting competition in terms of technological development. “Design primary energy consumption” can also be used by designers and users for designing and selecting the equipment for a given building.

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10.2 Appendix II – National report USA

ORNL/TM-2016/654

IEA/HPT Annex 39: US Summary Report – New Standard Method of Test for Multi-function, Integrated Heat Pump Systems, Development and Current Status

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October 31, 2016

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ORNL/TM-2016/654

Energy and Transportation Science Division

IEA/HPT Annex 39: US Summary Report – New Standard Method of Test for Multi-function, Integrated Heat Pump Systems, Development and Current Status

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Date Published: October 31, 2016

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under contract DE-AC05-00OR22725

US Summary Report – New Standard Method of Test for Multi-function, Integrated Heat Pump Systems, Development and Current Status

EXECUTIVE SUMMARY

ANSI/ASHRAE Standard 206-2013 was developed to provide a uniform method of testing for rating the seasonal performance of multi-purpose heat pumps (combined appliances) which perform space conditioning and water heating in residential applications. The heat pumps may also provide additional functions, such as ventilation and/or dehumidification.

The standard provides a single comprehensive procedure for all existing and anticipated multi-purpose heat pumps, covering six different, basic systems: single capacity air-source, dual capacity air-source, variable capacity air-source, single capacity liquid-source, dual capacity liquid-source, and variable capacity liquid-source systems, where liquid-source systems include water-loop, ground-water, ground-loop, and direct geoexchange equipment.

The combined appliances may operate in up to 7 modes: A) space conditioning only; B) space conditioning + water heating; C) dedicated water heating; D) dehumidification + space conditioning; E) dehumidification; F) dehumidification + space conditioning + water heating; and G) dehumidification + water heating.

The standard specifies the test equipment for performing such tests, the physical arrangement of the interconnecting refrigerant and water lines, the data required and the calculations to be used.

The procedures in this standard may be used as a basis for establishing seasonal efficiency ratings for such equipment and for estimating annual energy consumption.

The standard was published in June 2013, and is currently in laboratory use by at least 2 manufacturers.

1 INTRODUCTION

The beginning of the multifunction heat pump was the addition of a simple compressor discharge line heat exchanger, or desuperheater, to transfer heat to the potable water system. Desuperheaters became popular in the 1980s and are still sold today as aftermarket add-ons and as original equipment.

By the mid-1980s there was enough interest in the use of heat pumps for combined space conditioning and water heating that ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) developed Standard 137-1995, “Methods of Testing for Efficiency of Space Conditioning/Water-Heating Appliances that Include a Desuperheater Water Heater”, to address standard testing methods for combination space conditioning and potable water heating appliances. Integrated products being offered at that time were single-speed or single-stage air conditioners and heat pumps with desuperheaters, and therefore Standard 137-1995 dealt only with single stage equipment.

Later, in 1996, ARI (Air-Conditioning and Refrigeration Institute, now AHRI, Air-Conditioning, Heating, and Refrigeration Institute) introduced Standard 290-1996, “Air-Conditioning and Heat

Pump Equipment Incorporating Refrigerant to Potable Water Heating Devices” that dealt with the performance rating of single stage equipment providing desuperheating and full condensing capability.

These standards were then followed by requests for test and rating waivers to DOE (United States Department of Energy) by Carrier for their HydroTech integrated heat pump, and by Nordyne for their Powermiser integrated heat pump. The term “integrated heat pump” referred to heat pump systems that provided both space conditioning and water heating, and in both of these cases could provide water heating as needed, independent of the need for space conditioning.

Integrated heat pumps are continuing to evolve, with DOE, thru ORNL, working with several manufacturers to develop quite advanced systems employing variable-capacity capability (Rice et al., 2013; Rice et al., 2014a, 2014b). As a result of these development activities, ORNL approached ASHRAE and asked that a new method of test be developed for these systems.

ASHRAE established Standard Project Committee (SPC) 206 to develop Standard 206, “Method of Test for Rating of Multi-Purpose Heat Pumps for Residential Space Conditioning and Water Heating”. SPC 206 included representation from all industry segments, made up of 15 voting members, representing manufacturers, testing laboratories, electric utilities, and a university; AHRI was a non-voting member. The first version of this standard was completed and published in June 2013 as ANSI/ASHRAE Standard 206-2013.

2 EQUIPMENT COVERED

ANSI/ASHRAE Standard 206-2013 covers six different basic systems: single capacity air-source, dual capacity air-source, variable capacity air-source, single capacity liquid-source, dual capacity liquid-source, and variable capacity liquid-source systems, where liquid-source systems include water-loop, ground-water, ground-loop, and direct geoexchange equipment. For each of the six basic systems, the combined appliance may operate in up to 7 modes: A) space conditioning only; B) space conditioning + water heating; C) dedicated water heating; D) dehumidification + space conditioning; E) dehumidification; F) dehumidification + space conditioning + water heating; and G) dehumidification + water heating.

While Standard 206-2013 allows for up to 7 modes, the actual number of modes used in a given system covered by the standard may be any number from 2 thru 7 depending on the specific design of the equipment.

For space conditioning only (mode A), the system operates either in space cooling or space heating with no water heating. If the system is performing space cooling, it will be operating at its normal sensible heat ratio. Space conditioning only occurs when there is a call for space conditioning and the hot water storage tank is fully recovered.

The space conditioning + water heating operation (mode B) is the same as the space conditioning only mode except that there is a coincident need for water heating (e.g., the hot water storage tank is not fully recovered so there is a call for water heating). In addition to either space cooling or space heating, heat will be provided to the hot water storage tank per the equipment’s internal controls.

In dedicated water heating (mode C) the system is responding to a call for water heating with no call for space conditioning. In the dedicated water heating mode there may or may not be attendant space conditioning depending on the season and design of the system. If the source of heat for water heating is from outdoor air (or water/ground), there is no indoor space conditioning. If the source of heat for water heating is from indoor air, there will be space cooling irrespective of the season. Depending on the system design, the source of heat for water heating may be either indoor air or outdoor air, ground, or water for either season.

When operating in dehumidification + space conditioning (mode D) there will be space cooling at a lower sensible heat ratio than for the space cooling only mode. How the system achieves the lower sensible heat ratio, and its associated indoor airflow, is solely dependent on the equipment's design and internal controls.

The dehumidification mode (mode E) is the same as for dehumidification+space conditioning mode, except that the system operates at an even lower sensible heat ratio than for the dehumidification+space conditioning mode. There will likely still be some sensible space cooling, but the sensible heat ratio will be the lowest possible for the combined appliance.

The dehumidification + space conditioning + water heating mode (mode F) is the same as the dehumidification + space conditioning mode except that there is a coincident need for water heating as well. In addition to space cooling at a lower sensible heat ratio, heat will be provided to the hot water storage tank per the equipment internal controls.

The dehumidification + water heating mode (mode G) is the same as the dehumidification only mode except that there is a coincident water heating need as well. In addition to space cooling at the equipment's lowest possible sensible heat ratio, heat will be provided to the hot water storage tank per the equipment internal controls.

3 SENSIBLE LOAD

Because the standard accounts for equipment that can operate at various sensible heat ratios it was necessary to develop separate sensible and latent loads for use with the combined appliance performance calculation.

ASHRAE Standard 116-2010, which is the test standard for determining the seasonal efficiency of conventional (e.g., non-integrated) unitary air-source (air-to-air) heat pumps, establishes the total cooling load as 0 W (0 Btu/h) at 18.3°C (65°F) outdoor temperature and increases linearly to equal the unit test total cooling capacity at 35°C/26.7°C/19.4°C (95°F/80°F/67°F) (outdoor dry bulb/indoor dry bulb/indoor wet bulb temperature respectively), divided by 1.1 at 95°F outdoor temperature. Standard 206-2013 multiplies this total cooling load by 0.76 to establish the space design sensible cooling load for the combined appliance.

ASHRAE Standard 116-2010 establishes the sensible heating load as 0 W (0 Btu/h) at 18.3°C (65°F) outdoor temperature and increases linearly to equal 0.77 times the unit test heating capacity at 8.33°C/6.11°C/21.1°C (47°F/35°F/70°F) (outdoor dry bulb/outdoor wet bulb/indoor dry bulb temperature, respectively) at -15°C (5°F). Standard 206-2013 also uses this as the space sensible heating load for the combined appliance.

4 LATENT LOAD

The latent cooling load for the combined appliance in Standard 206-2013 is made up of an internal latent gain and a ventilation latent gain.

The internal latent gain consists of 52.7 W (180 Btu/h) of direct moisture per person and 105.5 W (360 Btu/h) of indirect (showers, cooking, etc.) moisture per person. The number of people is one per ton of total unit cooling capacity at 35°C/26.7°C/19.4°C (95°F/80°F/67°F) with a lower limit of 1 person and an upper limit of 5 persons.

The ventilation latent heat gain due to natural and forced ventilation is based on a total ventilation air flow of 18.9 L/s, (40 cfm) per 3.52 kW (1 ton) of total unit cooling capacity at 35°C/26.7°C/19.4°C (95°F/80°F/67°F). The ventilation air flow is then multiplied by the difference in enthalpy between the moist air for the outdoor average relative humidity in combination with each outdoor temperature bin and the indoor return air. For the cooling season the enthalpy of the indoor cooling return air is based on 26.7°C/19.4°C (80°F /67°F) and for the heating season the enthalpy of the indoor heating return air is based on 21.1°C /14.6°C (70°F /58°F).

5 WATER HEATING LOAD

For purposes of the seasonal bin analysis used in Standard 206-2013, the water heating load is assumed evenly distributed over the hours of the year. The hot water usage is taken as 243.4 liters/day (64.3 gallons/day) from ASHRAE 118.2-2006, which was the current test standard for standalone residential water heaters. Using inlet and outlet water temperatures from ASHRAE 118.2-2006 of 14.2°C (58°F) and 57.2°C (135°F) respectively and a water heater energy factor of 0.90 yields an hourly water heating load of 496.9 W (1696 Btu/h).

6 VENTILATION FUNCTION

Other than the latent load from the ventilation air, no other impact due to ventilation air is included in the standard.

Early in the development of the standard it was intended to have an integrated ventilation function impact the system return air dry and wet bulb temperatures, but this was dropped after it was concluded that the function would have little influence and would add much calculation complexity.

7 TESTS REQUIRED

While certain additional tests are required for a combined appliance, compared to a space conditioning only unit, the standard builds on the test conditions of ANSI/AHRI 210/240-2008, ANSI/ARI/ASHRAE ISO Standard 13256-1:1998, and ANSI/AHRI Standard 870-2005, which were the current rating standards for air-source, liquid-source, and direct geoexchange systems respectively, as the basis of the tests.

In order to minimize the laboratory test burden, all or part of some of the additional tests are conditional or optional. The optional test is at the manufacturer’s discretion, but if the test is not conducted, it is expected that the default value will result in a slight performance penalty.

The number of tests required to describe a given mode of operation increases as that mode of operation is expected to operate over a greater portion of the season. The number of tests also depends on the type and complexity of the system design, with air-source systems requiring more tests than liquid-source, and variable capacity systems requiring more tests than single capacity.

As a specific high detail example, the space conditioning + water heating mode is expected to operate over the entire season, so for single-capacity air-source systems two tests in both the cooling and heating seasons are required to define the basic linear interpolation of performance as a function of outdoor temperature. This base performance is then applied with the cyclic degradation factor of the space conditioning only mode for both cooling and heating, and the defrost degradation factor of the space conditioning only mode for heating.

A low detail example would be the dehumidification mode, which is only expected to operate during the mild temperatures of the cooling and heating seasons. Therefore, the performance of the dehumidification mode is based on only one test, and a change in performance as a function of outdoor temperature is assumed equal to that of the space conditioning + water heating mode. Even that test is optional, and the performance of the dehumidification only mode may be assumed equal to that of the space conditioning performance portion of the dehumidification + water heating mode if the manufacturer so chooses.

8 CALCULATION PROCEDURE

In order to keep the performance calculation for the standard as simple as possible, the standard uses a bin procedure based on the number of hours in 2.78°C (5°F) outdoor temperature bins, similar to that of ASHRAE Standard 116-2010. Because the standard takes into account the performance of combined appliances having multiple modes of operation, the standard requires a number of bin calculations in a specific sequence. The most complex example is a variable capacity system having all seven modes of operation. This requires twelve sequential bin calculations of varying complexity.

Because there is no way to easily predict the coincidence of space cooling (or heating), space dehumidification, and water heating loads, the standard assumes they all exist at the initiation of each hour’s calculation. The calculation procedure begins with the lowest SHR (sensible heat ratio) + water heating mode and progresses to successively higher SHR modes. The system operates in a given mode until it is determined that the next higher SHR mode can meet the remaining latent load in the allotted time. The dehumidification mode (with or without water heating) is defined as having the lowest SHR, with the space conditioning + dehumidification mode defined as having the next lowest (or intermediate) SHR, and the space conditioning mode having the highest (or normal) SHR.

Thus, the first calculation is for the dehumidification + water heating mode (G), and operates to meet either the initial latent load, the initial space sensible cooling load, or the initial water heating load.

The second calculation is then for the space conditioning + dehumidification + water heating mode (F), and operates to meet either the residual latent cooling load, the residual space sensible cooling load, or the residual water heating load.

The third through fifth calculations are for the space conditioning + water heating mode (B) in sequence of low, variable, and high capacity, and operates to meet either the residual space sensible cooling load, or the residual water heating load.

The sixth calculation is for the dehumidification mode (E), and operates to meet either the residual latent cooling load, or the residual space sensible cooling load.

The seventh calculation is for the space conditioning + dehumidification mode (D), and operates to meet either the residual latent cooling load, or the residual space sensible cooling load.

The eighth calculation is for the dedicated water heating mode (C), and operates to meet the residual water heating load.

The ninth calculation is then for auxiliary dehumidification, and operates to meet any residual space latent cooling load.

The tenth thru twelfth calculations are for the space conditioning only mode (A) in sequence of low, variable, and high capacity, and operates to meet any residual space sensible load.

It should be noted that the dedicated dehumidification mode, either with or without water heating, only operates if the space latent cooling load is greater than can be met by the dehumidification + space conditioning + water heating, or dehumidification + space conditioning modes. Similarly, the dehumidification + space conditioning mode, either with or without water heating, only operates if the space latent load is greater than can be met by the space conditioning + water heating, or space conditioning only modes.

A last requirement is that the combined appliance can only operate for one hour of each bin hour. Auxiliary dehumidification, auxiliary space heating, and/or auxiliary water heating is assumed to operate as required within any given bin hour to exactly meet any residual space sensible heating load, residual space latent cooling load, and/or residual water heating load. An unmet residual sensible cooling load is possible at high outdoor temperatures where the space cooling load exceeds the cooling capacity of the unit, just as in ASHRAE Standard 116-2010.

Because the space conditioning + water heating mode, during space heating, can result in increased use of auxiliary space heat due to heat pump system heat being diverted to water heating, the standard allows the manufacturer to specify a priority to space conditioning. With a priority on space conditioning, the run times in heating modes that include water heating are limited such that the ending space conditioning only mode can exactly meet the space heating load.

9 PERFORMANCE DESCRIPTORS

Three performance descriptors are used to completely describe the performance of the combined appliance: a combined appliance space conditioning performance for the cooling season; a combined appliance space conditioning performance for the heating season; and a combined appliance water heating performance on an annual basis.

In all cases, the descriptor ends in a subscript “ca” to designate “combined appliance”.

For air-source systems the cooling season space conditioning descriptor is termed $SEER_{ca}$, and the heating season space conditioning descriptor is termed $HSPF_{ca}$, where SEER and HSPF stand for seasonal energy efficiency ratio and heating seasonal performance factor, respectively, just as it does for the space conditioning only equipment covered by ASHRAE Standard 116-2010.

For liquid-source and direct geoexchange systems the cooling season space conditioning descriptor is termed EER_{ca}, and the heating season space conditioning descriptor is termed COP_{ca}, where EER and COP stand for energy efficiency ratio and coefficient of performance, respectively, just as it does for the space conditioning only equipment covered by ANSI/ARI/ASHRAE ISO Standard 13256-1:1998, and ANSI/AHRI Standard 870-2005. While the liquid-source and direct geoexchange system performance descriptors do not contain an “S” for seasonal, and the unit capacity does not change as a direct function of outdoor temperature, the bin calculation does account for the change in load as a function of outdoor temperature.

EF_{wca} is the annual water heating energy factor, and is defined identically to the energy factor (EF) in ASHRAE118.2-2006. In addition to the “ca” being added, there is also a “w” added to differentiate between the energy factor for water heating and the energy factor for the auxiliary dehumidifier designated as EF_{d} in the standard.

$SEER_{ca}$ and EER_{ca} are defined as the total space cooling energy delivered to the space divided by the total energy used for space cooling, with units of Btu/Wh.

$HSPF_{ca}$ and COP_{ca} are defined as the total space heating energy delivered to the space divided by the total energy used for space heating, with units of Btu/Wh, and Wh/Wh respectively.

EF_{wca} is defined as the total hot water energy delivered divided by the total energy used to heat the water, and is dimensionless.

In all cases the energy consumed by the combined appliance is debited proportionally by function to the useful energy provided. As an example, if the combined appliance provides 10 Wh (34.13 Btu) for space conditioning, either cooling or heating, and simultaneously provides 2 Wh (6.83 Btu) for water heating while consuming 3 Wh (3 Wh) of energy, excluding the potable water pump energy, 10/12 or 2.5 Wh (2.5 Wh) of the energy consumed is debited to space conditioning, and 2/12 or 0.5 Wh (0.5 Wh), plus all of the potable water pump energy, is debited to water heating.

Because the combined appliance and conventional (non-integrated) space conditioning only equipment do not meet identical space conditioning loads (the combined appliances must address sensible+latent loads, while the single function equipment is controlled only to meet sensible loads), an energy adjustment is deducted from the combined appliance energy use to bring $SEER_{ca}$, EER_{ca}, $HSPF_{ca}$, and COP_{ca} into alignment with SEER, EER, HSPF, and COP respectively. The calculation of the energy adjustment requires two additional bin calculations.

First, the performance for the space conditioning only mode is calculated using the data from the space conditioning only tests (mode A) against the total load with the bin procedure presented in ASHRAE Standard 116-2010. The results will be the total capacity supplied and total energy used.

Second, the performance of the space conditioning only mode is again calculated using the data from the space conditioning only tests (mode A), but against the separate sensible and latent loads as determined for Standard 206-2013 load and with the bin procedure presented in ASHRAE Standard 206-2013 with an auxiliary dehumidifier included. The results will be the net total capacity supplied and total energy used.

The difference between the net total capacity of the space conditioning only plus auxiliary dehumidifier system and the total capacity of the space conditioning only system multiplied by the energy consumed by the space conditioning only system provides the energy adjustment to be subtracted from the total combined appliance system energy consumed to make SEERca,

EERca, HSPFca, and COPca directly comparable to the SEER, EER, HSPF, and COP values respectively that U.S. consumers are already quite familiar with.

10 APPLICATION

Several manufacturers are known to have tested product to Standard 206-2013, with one having made some results publicly available.

The unit tested was a variable speed air-source design and operated with modes A, B and C. A series of 24 tests were conducted and system performance parameters were evaluated according to the procedures outlined in the standard.

During testing it was found that the unit controls did not operate the unit in precise agreement with the requirements of the standard, but based on the test results of the prototype system and seasonal/annual efficiencies estimated using Standard 206-2013 procedures, it was found that integrating the space conditioning and water heating functions in a heat pump system can yield annual water heating energy savings of over 60% (Miller, et al, 2016).

11 CONCLUSION

ASHRAE Standard 206-2013 represents a major step forward in development of standard testing methods and seasonal/annual performance metrics of combined appliances. Standard 206-2013 covers a very wide range of equipment because it is intended to address not only those configurations currently being developed, but also other possible configurations that might arise in the foreseeable future. It has introduced the concept of specific latent loads and implemented a means of accounting for them in a seasonal performance descriptor.

Development of a uniform method-of-test along with performance descriptors understandable by the consumer, followed by establishment of a rating standard, will facilitate market penetration of energy saving combined appliances, by allowing consumers to make informed comparisons of the energy performance of combined appliances to that of a suite of separate-function appliances.

12 REFERENCES

ANSI/ASHRAE Standard 206-2013, *Methods of Testing for Rating of Multipurpose Heat Pumps for Residential Space Conditioning and Water Heating*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 1791 Tullie Circle NE, Atlanta GA 30329. www.ashrae.org.

ANSI/ASHRAE Standard 137-1995, *Methods of Testing for Efficiency of Space Conditioning/Water-Heating Appliances that Include a Desuperheater Water Heater*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 1791 Tullie Circle NE, Atlanta GA 30329. www.ashrae.org.

ANSI/AHRI Standard 290-1996, *Air-Conditioning and Heat Pump Equipment Incorporating Refrigerant to Potable Water Heating Devices*, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

ANSI/ASHRAE Standard 116-2010, *Methods of Testing for Rating Seasonal Efficiency of*

Unitary Air Conditioners and Heat Pumps, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 1791 Tullie Circle NE, Atlanta GA 30329. www.ashrae.org.

ANSI/ASHRAE Standard 118.2-2006, *Method of Testing for Rating Residential Water Heaters*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 1791 Tullie Circle NE, Atlanta GA 30329. www.ashrae.org.

ANSI/AHRI Standard 210/240-2008, *Standard for Unitary Air-Conditioning and Air-Source Heat Pump Equipment*, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

ANSI/ARI/ASHRAE ISO Standard 13256-1:1998, *Water-source heat pumps - testing and rating for performance - Part 1: Water-to-air and brine-to-air heat pumps*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 1791 Tullie Circle NE, Atlanta GA 30329. www.ashrae.org.

ANSI/AHRI Standard 870-2005 *Standard for Performance Rating of Direct GeoExchange Heat Pumps*, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

Rice, C. K., Baxter, V. D., Hern, S. A., McDowell, T. P., Munk, J. D., and Shen, B., 2013. "Development of a Residential Ground-Source Integrated Heat Pump", *2013 ASHRAE Winter Conference*, January.

Rice, C. K., Shen, B., Munk, J. D., Ally, M. R., Baxter, V. D., 2014a. "Development of a Variable-Speed Residential Air-Source Integrated Heat Pump", *Conference Proceedings of 11th International Energy Agency Heat Pump Conference*, Montreal, Quebec, Canada, May 12-16

Rice, C. K., Shen, B., Baxter, V. D., Shrestha, S. S., and Uselton, R. B., 2014b. "Development of an Air-Source Heat Pump Integrated with a Water Heating/Dehumidification Module", *Conference Proceedings of 11th International Energy Agency Heat Pump Conference*, Montreal, Quebec, Canada, May 12-16

Miller, W. A., R. G. Berry, Durfee, N., and V. D. Baxter, 2016. "Experimental Evaluation of High Performance Integrated Heat Pump," *Proceedings of the 16th International Refrigeration and Air Conditioning Conference*, Purdue University, July 11-14.

10.3 Appendix III – National report Austria

IEA Heat pump program

IEA HPP Annex 39

Technical Report

Review and SWOT Analysis of Relevant Standards, Guidelines and Software Tools for Testing and Performance Evaluation of Heat Pumps

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März 2015 TR A.1.1

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1 INTRODUCTION

The objective of Annex 39 is to develop a common calculation method for SPF using a generalised and transparent approach, based on laboratory measurement data, to develop appropriate test methods and to extend the performance evaluation to include primary energy depleting and carbon dioxide emission issues.

In Task 1 of the Annex, the goal was to provide a critical review of relevant testing and performance evaluation standards for heat pumps and heat pump systems in each of the participating countries. In Austria, the survey included the following sources: ÖNORM and EN standards, VDI guidelines and EHPA test regulations accepted by some of the national funding bodies.

The reviewed documents can be subdivided into two groups, according to the scope of the document/tool, Table 1:

Table 1: Environmental conditions for units according to the installation site

Test standards and regulations	***Performance calculation standards and guidelines***
EN 14511	EN 14825
EN 15879-1	EN 16147
EN 16147	VDI 4650-1
EHPA Test Regulation	EN 15315 4-2

2 SHORT DESCRIPTION OF RELEVANT STANDARDS AND GUIDELINES

2.1 Standards for heat pump testing and rating at nominal conditions (COP/EER)

2.1.1 EN14511: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling

EN 14511 was drafted by the CEN/TC 113 “Heat Pumps and Air-Conditioning Units”. The current valid version is EN 14511:2011.

The scope of the standard includes electrically driven heat pumps and chillers for space heating and/or cooling. Heat transfer media on heat source and heat sink side can be air, water or brine.

The standard includes four parts:

- Part 1: Terms and definitions;
- Part 2: Test conditions;
- Part 3: Test methods;
- Part 4: Requirements.

EN 14511 defines test procedures for the rating of heat pumps and chillers under steady state conditions (except certain air-source units in heating mode) and at full capacity. The heating and/or cooling capacity of units for hydronic distribution systems is determined by measurement of the volume flow of the heat transfer medium and the inlet and outlet temperatures at the water or brine heat exchanger, taking into consideration the specific heat capacity and density of the heat transfer medium. For air-source units, either the air enthalpy method or the calorimeter room methods can be used. The electrical power input is measured directly.

The unit can be tested under one or more operating conditions depending on the manufacturer’s needs and the type of the unit. One standard rating condition comprises of one temperature and mass flow rate condition of the heat transfer fluid in the evaporator and one in the condenser. Additionally, the unit can be tested under defined application rating conditions, but these tests are not compulsory. Depending on the designation of the unit regarding the type of installation (indoors or outdoors) made by the manufacturers, different environmental conditions according to Rating conditions are given for a number of unit types:

- water-to-water, brine-to-water, water-to-brine and brine-to-brine units in heating and cooling mode;
- air-to-water and air-to-brine units in heating and cooling mode;
- liquid chilling packages for heat recovery condenser and with a remote condenser;
- basic, multiple circuit and modular air-cooled multisplit systems in the heating and cooling mode;
- modular heat recovery air-cooled multisplit systems;

- basic, multiple circuit and modular water-cooled multisplit systems in the heating and cooling mode.

Table 2 must be applied.

Rating conditions are given for a number of unit types:

- water-to-water, brine-to-water, water-to-brine and brine-to-brine units in heating and cooling mode;
- air-to-water and air-to-brine units in heating and cooling mode;
- liquid chilling packages for heat recovery condenser and with a remote condenser;
- basic, multiple circuit and modular air-cooled multisplit systems in the heating and cooling mode;
- modular heat recovery air-cooled multisplit systems;
- basic, multiple circuit and modular water-cooled multisplit systems in the heating and cooling mode.

Table 2: Environmental conditions for units according to the installation site

Unit type	indoor installation in °C dry bulb (wet bulb)	outdoor installation in °C dry bulb (wet bulb)
W/W, B/W	15 to 30	-
A/W with duct connection on the air inlet and outlet side	15 to 30	as inlet air temperatures
A/W without duct connection on the air inlet side	as inlet air temperatures	as inlet air temperatures
W/A, B/A with duct connection on the air inlet and air outlet side	15 to 30	-
W/A, B/A without duct connection on the air inlet and air outlet side	as inlet liquid temperatures	as inlet air temperatures
A/A with duct connection on the outdoor air inlet and outlet side	15 to 30	-
A/A without duct connection on the outdoor air inlet and outlet side	as inlet air temperatures	-
A/A with duct connection on the indoor air inlet and outlet side	-	as inlet air temperatures
W/W, B/W operating in cooling mode	-	25 to 35
W/W, B/W operating in heating mode	-	0 to 7

Only the first two unit types will be discussed here since the others have low relevance for SHP systems.

The temperatures and the mass flow rates of liquid heat transfer media both for the heat source and for the heat sink are fixed for all rating conditions. The mass flow rate remains constant throughout the test. For air as heat transfer media, only the inlet dry bulb temperatures (and for some cases wet bulb temperatures) are fixed. In

Table 3 through Table 6 an overview of all temperature levels is given:

Table 3: Outdoor heat exchanger temperatures in the heating mode

		Outdoor heat exchanger	
		Inlet temperature	Outlet temperature
Heating, standard rating conditions	water	10	7
	brine	0	-3
	outside air - dry (wet) bulb	7	6
	exhaust air - dry (wet) bulb	20	12
Heating, application rating conditions	water	15	*
	brine	5	*
	outside air - dry (wet) bulb	2 (1)	**
		-7 (-8)	**
		-15 (-)	**

Table 4: Indoor heat exchanger temperatures in the heating mode

		Indoor heat exchanger	
		Inlet temperature	Outlet temperature
Heating, standard rating conditions	low temperatures	35	30
	medium temperatures	45	40
	high temperatures	55	47
	very high temperatures	65	50
Heating, application rating conditions	low temperatures	35	*
	medium temperatures	45	*
	high temperatures	55	*
	very high temperatures	65	*

Table 5: Outdoor heat exchanger temperatures in the cooling mode

		Outdoor heat exchnager	
		Inlet temperature	Outlet temperature
Cooling, standard rating conditions	brine and water - cooling tower	30	35
	brine and water - ground coupled	10	15
	air (for water and brine) - dry bulb	35	**
Heating, application rating conditions	air (for water - medium temperatures) -dry bulb	46	**
	air (for water - medium and low temperatures, brin)e - dry bulb	27	**

Table 6: Indoor heat exchanger temperatures in the cooling mode

		Indoor heat exchanger	
		Inlet temperature	Outlet temperature
Cooling, standard rating conditions	water - medium temperatures	23	18
	water - low temperatures	12	7
	brine	0	-5

* The test is performed at the flow rate obtained during the test at the standard rating conditions; **

Not defined

The heating and the cooling capacities for steady state operation are calculated according to eq. 1 and eq. 2:

$$P_H = q \cdot \rho \cdot c_p \cdot \Delta T \qquad \textit{eq. 1}$$

$$P_C = q \cdot \rho \cdot c_p \cdot \Delta T \qquad \text{eq. 2}$$

where

P_H	is the heating capacity;
P_C	is the cooling capacity;
q	is the volume flow rate of the heat transfer medium;
ρ	is the density of the heat transfer medium;
c_p	is the specific heat of the heat transfer fluid at constant pressure;
ΔT	is the difference between inlet and outlet temperatures of the heat transfer medium.

The measured power input includes the overall consumption of the unit including all fans and pumps which make an integral part of the unit. However, in case of integrated liquid pumps and ducted air units, only a part of the respective power input is taken into account which corresponds to the power needed to overcome the pressure drop over the internal heat exchanger (e.g. evaporator or condenser) or the air duct. On the other hand, if a liquid pump or a fan of a ducted unit is not integrated into the unit but needed for the operation, a certain power will be added to the overall power input which corresponds to the pressure drop over an integrated heat exchanger or the duct. Both can be calculated from equations 5 and 8, respectively. For units without duct connection, the entire fan power is included in the total power consumption.

Liquid pumps and fans dissipate one part of their electrical power input to the heat transfer media itself. This is also taken into account in the heat balance for the calculation of the Coefficient of Performance (COP, for heating mode) and Energy Efficiency Ratio (EER, for cooling mode) by adding or subtracting the same amount of power as in the case of the pressure drop consideration from the heating or cooling capacity.

The COP or the EER are thus calculated as follows:

a. if a liquid pump is integrated into the unit:

$$COP = \frac{P_H}{P_E - P_{LP,F}} \qquad \text{eq. 3}$$

$$EER = \frac{P_C}{P_E - P_{LP,F}} \qquad \text{eq. 4}$$

$$P_{LP,F} = \frac{q \times \Delta p_e}{\eta} \qquad \text{eq. 5}$$

b. if a liquid pump is not integrated into the unit:

$$COP = \frac{P_H}{P_E + P_{LP,F}}$$ eq. 6

$$EER = \frac{P_C}{P_E + P_{LP,F}}$$ eq. 7

$$P_{LP,F} = \frac{q \times (-\Delta p_i)}{\eta}$$ eq. 8

where:

$P_{LP,F}$	is the power of the liquid pump or the fan;
P_E	is the total measured electricity input;
η	is the assumed efficiency of the liquid pump or the fan;
q	is the nominal heat transfer medium flow rate;
Δp_e	is the measured external static pressure difference in the heat transfer medium;
Δp_i	is the measured internal static pressure difference in the heat transfer medium.

The efficiency η of the fan is considered to be 0,3 by convention for all units. For liquid pumps, η is calculated according to the following formulae:

$\eta = 0{,}0721 \cdot P_{hyd}^{0{,}3183}$, if $P_{hyd} < 500$ W eq. 9

$\eta = 0{,}092 \cdot \ln\left(P_{hyd}\right) - 0{,}0403$, if $P_{hyd} > 500$ W eq. 10

For the measurement of heating and cooling capacity of water-to-water or brine-to-water units, as well as the cooling capacity of air-to-water units, the measurements are carried out in the steady state condition. This condition is defined as "*(…) when all the measured quantities remain constant without having to alter the set values, for a minimum duration of 1 h, with respect to the tolerances (…) Periodic fluctuations of measured quantities caused by the operation of regulation and control devices are permissible, on condition the mean value of such fluctuations does not exceed the permissible deviations listed (…)*". The tolerances and deviations refer to Table 7 in this document. The respective capacity is calculated as the average value from the recorded temperatures and volume flows.

The measurement procedure of cooling capacity for air-to-water and air-to-air units consists of three periods: preconditioning period, equilibrium period and data collection period.

In the preconditioning period the aimed test conditions should be reached and maintained for at least 10 minutes. This period should preferably end with a defrost cycle. If so, the temperatures and the water flow rates on the indoor heat exchanger should be set not before 20 minutes after the end of the defrost cycle.

In the equilibrium period, steady state conditions according to Table 7 should be maintained for at least one hour. If a defrost cycle occurs during this period, then a transient test procedure will be applied.

Table 7: Permissible deviations from the set values

Measured quantity	Permissible deviations from the set values	
	Arithmetic mean values	Individual measured values
Liquid		
inlet temperature	± 0,2 K	± 0,5 K
outlet temperature	± 0,3 K	± 0,6 K
volume flow	± 1 %	± 2,5 %
static pressure difference	-	± 10 %
Air		
inlet temperature	± 0,3 K	± 10 K
volume flow	± 5 %	± 10 %
static pressure difference	-	± 10 %
Refrigerant		
liquid temperature	± 1 K	± 2 K
saturated vapour temperature	± 0,5 K	± 1 K
Voltage	± 4 %	± 4 %

For the data collection period, two options are possible: steady state or transient test procedures. For the steady state tests a data collection period 70 minutes with a data sampling rate of at least 30 seconds is foreseen. This procedure applies if the value of the quantity *%ΔT* from equation 11 does not exceed 2,5 % during the first 35 minutes of the measurement. If it does exceed 2,5 %, the test must be carried out for transient conditions:

$$\%\Delta T = \left[\frac{\Delta T_i\left(\tau=0\right) - \Delta T_i\left(\tau\right)}{\Delta T_i\left(\tau=0\right)} \right] \cdot 100\,\% \qquad \text{eq. 11}$$

where:

$\Delta T_i(\tau=0)$ is the average temperature difference after first 5 minutes of the measurement;

$\Delta T_i(\tau)$ is the average temperature difference for the entire measurement period;

Table 8: Variations allowed in heating capacity tests when using the transient test procedure

Readings	Variation of values from specified test conditions			
	Arithmetical mean values		Individual readings	
	Interval H	Interval D	Interval H	Interval D
Air temperature entering indoor side				
dry bulb	± 0,6 K	± 1,5 K	± 1,0 K	2,5 K
Air temperature entering outdoor side				
dry bulb (for HX A>5 sqm, x2)	± 0,6 K	± 1,5 K	± 1,0 K	± 5,0 K
wet bulb	± 0,3 K	± 1,0 K	± 0,6 K	-
Inlet water temperature	± 0,2 K	-	± 0,5 K	-
Outlet water temperature	± 0,5 K	-	-	max ± 2 K

If *%ΔT* dose exceed 2,5 % or a defrost cycle occurs during the equilibrium period or during the first 70 minutes of the data collection period, the transient test procedure and tolerances according to

Table 8 will apply. Values marked with H in

Table 8 are to be used for heating intervals, values marked with D for defrosting intervals.

The data collection period is extended to either 3 hours or 3 complete defrosting cycles, whatever occurs first.

The heating capacity in this case is calculated as the average value from the recorded temperatures and volume flows for steady state tests.

Besides the rating tests, EN14511 also considers requirements for failure tests, operating range tests, freeze-up test, marking, technical report, technical data sheet and instructions.

Direct expansion heat pumps are covered by EN 15879-1 [22]. Nominal operating conditions for the indoor heat exchanger are comparable to those of EN 14511. Nominal heat source temperature (temperature of the brine bath in which the evaporator loops are submerged) is 4°C.

2.1.2 EHPA Test Regulation for Brine, Water and Air Source Heat Pumps

The EHPA Test Regulations are based on the standards EN 14511 and EN 15879-1. Additional to the requirements of the standards an extended test of the operating range shall be done. The calculation and the deviation are referenced to the standards and use the values given in the chapter 2.1.1.

For Air sourced heat pumps with brine intermediate circuit the Annex A was developed. This annex specifies additional requirements that apply to air-to-water units consisting of the assembly of an air-to-brine heat exchanger and a brine-to-water heat pump and provided as a factory-made split system by the manufacturer.

To be granted the label these units shall fulfil the requirements described in the testing regulation and in this annex A.

TECHNICAL DOCUMENTATION

The manufacturer shall specify in the technical documentation of the product:

- the minimum outdoor air temperature fro which the brine loop unit is prevented from freezing.
- the corresponding brine type and concentration
- the maximum allowed length of the brine circuit

TEST INSTALLATION

Prior to conduct any test, the test centre shall install the unit as follows:

- .the brine circuit lines shall be installed in accordance with the manufacturer's instructions. The length of the lines shall be 5 m except if the constraints of the test installation make 5 m not possible, in which case a greater length may be used, with a maximum of 7,5 m.
- The lines shall be installed so that the difference in elevation does not exceed 2,5 m.
- the thermal insulation of the lines shall be applied in accordance with the manufacturer's instructions.
- unless constrained by the design, at least half of the brine lines shall be exposed to the outside conditions, with the rest of the lines exposed to the inside conditions.
- the brine circuit shall be filled in with the brine type and concentration as specified by the manufacturer

PERFORMANCE TESTS

For the purpose of the determination of COP, the definition of the effective power input as given in EN 14511-1 is modified as follows:

effective power input

P_E

average electrical power input of the unit within the defined interval of time, expressed in Watt, obtained from:

- power input for operation of the compressor and any power input for defrosting;
- power input of the brine pump of the intermediate brine circuit
- power input for all control and safety devices of the unit;
- proportional power input of the conveying devices (e.g. fans, pumps) for ensuring the transport of the heat transfer media (outdoor air and water) inside the unit.

SAFETY TESTS

In addition to the safety tests as described in the testing regulation under 6.4, the additional tests shall be performed:

Shutting off the heat transfer medium flows

The test is completed by a shutting off the heat transfer medium in any intermediate circuit

Shutting off the intermediate brine flow rate during defrost operating conditions

For units with defrosting system, an additional test shall be conducted at the test conditions specified in Table 4 of EN 14511-4 by shutting off the brine flow rate of the intermediate circuit at the beginning of the defrosting phase.

EN 14511-4 – Table 4 – Shutting off heat transfer medium flows

Type of unit	Outdoor heat exchanger		Indoor heat exchanger			
	Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet water temperature °C	Outlet water temperature °C
Air-to-air units	2	1	20	15 max.		
Air-to-water units	2	1			a	45
Air-to-water units (for floor heating or similar application)	2	1			a	35

[a] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

2.2 Standards for the calculation of SCOP/SEER

2.2.1 EN14825: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – Testing and rating at part load conditions and calculation of seasonal performance

The current version EN 14825:2012 was drafted by the WG7 of the CEN TC113. It was published in April 2012. The revision of the document will, however, start immediately.

The aim of the standard is to give a basis for the comparison of heat pumps, chilling packages and air conditioners on the basis of the Seasonal Energy Efficiency Ratio (SEER) for cooling and Seasonal Coefficient of Performance (SCOP) for heating applications. It provides a description of the calculation method and the part load conditions for three different climates: an average climate, one cold and one warm climate. Corresponding climate data is provided in an Annex to the document.

The standard covers air-to-air, water (brine)-to-air, air-to-water and water (brine)-to-water units. Only the latter two will be discussed.

For the detailed rating conditions and test methods EN 14511 Parts 2 and 3 are used.

The calculation is based on the temperature bin method, which is well described in e.g. [1].

SEER and SCOP are calculated according to:

$$SEER = \frac{Q_{CE}}{\frac{Q_{CE}}{SEER_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \quad \textit{eq. 12}$$

$$SCOP = \frac{Q_{HE}}{\frac{Q_{HE}}{SEER_{on}} + H_{TO} \cdot P_{TO} + H_{SB} \cdot P_{SB} + H_{CK} \cdot P_{CK} + H_{OFF} \cdot P_{OFF}} \quad \textit{eq. 13}$$

where

$SEER_{on}$ is the seasonal efficiency of the unit in active cooling mode;

$SCOP_{on}$ is the seasonal efficiency of the unit in active heating mode;

Q_{CE} is the reference annual cooling demand;

Q_{HE} is the reference annual heating demand;

H_i is the number of hours the unit is considered to work in the modes indicated by the indices;

H_i is the electricity consumption during the modes indicated by the indices.

Indices:

TO thermostat off mode;

SB standby mode;

CK crankcase heater mode;

OFF off mode.

The reference annual heating and cooling demands are obtained from the respective design load multiplied with the equivalent heating or cooling periods in hours. These periods are given in the informative Annex C of the standard, but only for air-to-air and water-to-air units.

The $SEER_{on}$ and $SCOP_{on}$ are determined as follows:

$$SEER_{on} = \frac{\sum_{j=1}^{n} h_j \cdot P_C(T_j)}{\sum_{j=1}^{n} h_j \cdot \left(\frac{P_C(T_j)}{EER(T_j)} \right)} \quad \text{eq. 14}$$

$$SCOP_{on} = \frac{\sum_{j=1}^{n} h_j \cdot P_H(T_j)}{\sum_{j=1}^{n} h_j \cdot \left(\frac{P_H(T_j) - elbu(T_j)}{COP(T_j)} + elbu(T_j) \right)} \qquad \text{eq. 15}$$

where

T_j is the bin temperature;

j is the bin number;

n is the amount of bins;

P_C is the cooling demand of the building for the corresponding temperature T_j;

P_H is the heating demand of the building for the corresponding temperature T_j;

h_j is the number of bin hours in which a certain temperature T_j occurs;

EER (T_j) is the EER value of the unit for T_j;

COP (T_j) is the COP value of the unit for T_j;

elbu (T_j) is the capacity of an electrical heating back up unit for T_j.

If the unit cannot provide sufficient heating capacity for operation points below the bivalent point (part load condition F), the COP values are corrected by introducing an electrical back-up heater with the COP of 1.

For the calculation of the $SEER_{on}$ and $SCOP_{on}$, a unit has to be tested for a certain number of part load conditions according to Table 9 for cooling and Table 10 for heating. The part load ratios (PLR) given in the tables are calculated according to the following equation:

$$PLR = \frac{T_i - 16}{T_{design} - 16} \qquad \text{eq. 16}$$

The power consumption is measured by setting the thermostat to a value which triggers shutting down of the compressor. The auxiliary power consumption is then measured. For the measurement of the power consumption in the standby mode, the unit is stopped by the control device and the power measured. For the crankcase heater power consumption it is only stated, that the measurement has to last for 8 hours after the B temperature condition test. Finally, the off mode test has to take place after the standby test by switching the unit into the off mode while remaining plugged.

The EER and COP values for the part load conditions A, B, C and D (also E and F) have to be provided. If measurements for these conditions are available, the EER and COP values for other temperature bins are generally interpolated between or extrapolated from these values. The cooling and heating demand are calculated from the respective full load value multiplied by the respective part load ratio of the corresponding bin.

Table 9: Part load conditions for the cooling mode

	Part load ratio [%]	Outdoor heat exchanger			Indoor heat exchanger		
		air-to-water	water (brine)-to-water		Fan coil		Radiant cooling
		dry bulb T [°C]	Cooling tower	Groound coupled	Fixed outlet	Variable outlet	
A	100	35	30 / 35	10 / 15	12 / 7	12 / 7	23 / 18
B	74	30	26 / *	10 / *	* / 7	* / 8,5	* / 18
C	47	25	22 / *	10 / *	* / 7	* / 10	* / 18
D	21	20	18 / *	10 / *	* / 7	* / 11,5	* / 18
* water flow rate as determined in the full load test (A)							

For load conditions above the condition A and below the condition D, the same values of EER as for A or D are used, respectively.

Table 10: Part load conditions for the heating mode (average climate)

	Part load ratio [%]	Outdoor heat exchanger			Indoor heat exchanger		
		air-to-water	water (brine)-to-water				
		dry (wet) bulb T [°C]	Ground water	Brine	Application	Fixed outlet	Variable outlet
A	88	-7 (-8)	10 / *	0 / *	low T	* / 35	* / 34
					medium T	* / 45	* / 43
					high T	* / 55	* / 52
B	54	2 (1)	10 / *	0 / *	low T	* / 35	* / 29
					medium T	* / 45	* / 36
					high T	* / 55	* / 43
C	35	7 (6)	10 / *	0 / *	low T	* / 35	* / 27
					medium T	* / 45	* / 33
					high T	* / 55	* / 38
D	15	12 (11)	10 / *	0 / *	low T	* / 35	* / 25
					medium T	* / 45	* / 29
					high T	* / 55	* / 33
E	f (TOL)	TOL			low T	* / 35	* / 34 - (-7-TOL) / (-7-2) x (34-29)
					medium T	* / 45	* / 43 - (-7-TOL) / (-7-2) x (43-36)
					high T	* / 55	* / 52 - (-7-TOL) / (-7-2) x (52-43)
F	f ($T_{bivalent}$)	$T_{bivalent}$			low T	* / 35	* / 34 - (-7-$T_{bivalent}$) / (-7-2) x (34-29)
					medium T	* / 45	* / 43 - (-7-$T_{bivalent}$) / (-7-2) x (43-36)
					high T	* / 55	* / 52 - (-7-$T_{bivalent}$) / (-7-2) x (52-43)
* water flow rate as determined in the full load test (A)							

The bivalent temperature $T_{bivalent}$ is to be set at 2°C or less for the average climate. The operational limit temperature TOL is the lowest outdoor temperature at which the heat pump can still deliver heating capacity and is stated by the manufacturer.

In case that a certain part load condition cannot be reached as stated in Table 9 and Table 10, for example for units with on/off control, the following procedures are to be used (for air-to-water, water-to-water and brine-to-water units):

- Fixed capacity units: The EER or the COP are calculated from the following equations:

$$EER_{B,C,D} = EER_{DC} \cdot \left(\frac{CR}{C_c \cdot CR + (1 - C_c)} \right) \qquad \text{eq. 17}$$

$$COP_{A,B,C,D} = COP_{DC} \cdot \left(\frac{CR}{C_c \cdot CR + (1 - C_c)} \right) \qquad \text{eq. 18}$$

where

EER_{DC} is the EER corresponding to the declared capacity (DC) of the unit at the given temperature conditions for B, C or D;

COP_{DC} is the COP corresponding to the declared capacity (DC) of the unit at the given temperature conditions for B, C or D;

C_c is the degradation coefficient;

CR is the capacity ratio;

The degradation coefficient takes into account the electricity consumption of the unit while the compressor is switched off. It is calculated as

$$C_c = 1 - \frac{\text{measured power in compressor off state}}{\text{full capacity at part load condition}} \qquad \text{eq. 19}$$

If it cannot be determined, the default value of 0,9 should be used.

CR is equal to the heating or cooling demand over the declared capacity of the unit at the same temperature conditions.

The outlet temperatures of the indoor heat exchanger, as indicated in Table 9 for cooling and Table 10 for heating should correspond to the time averaged outlet temperature according to eq. 20:

$$t_{outlet,average} = t_{inlet,full\ load\ test} + (t_{outlet,full\ load\ test} - t_{inlet,full\ load\ test}) \cdot CR \qquad \text{eq. 20}$$

- **Staged capacity units**: The EER or the COP are calculated by the interpolation from the values on either side of the control step of the unit, if the given value (A, B, C or D) cannot be reached within ±3%. If the smallest control step is higher than the required cooling or heating demand, the EER or the COP are calculated as for fixed capacity units.
- **Variable capacity control units**: The capacity for the given part load condition should be reached within ±5% from the stated value. If this is not the case, then the same procedure as for the staged capacity units should be applied.

For fixed capacity units, an alternative test method for part load conditions is given. Thereafter, the test can be performed by obtaining the relevant temperature as a time averaged value over the testing period. It is however unclear which tolerances, data acquisition times etc. should be applied as such a test is not defined in EN 14511.

The standard also defines measurement uncertainties related to the respective heat or cooling load, which are independent of the measurement uncertainties defined in EN 14511.

Finally, informative Annexes A and B give calculation examples for SEER, $SEER_{on}$, $SCOP_{on}$ and $SCOP_{net}$.

2.2.2 EN 16147: Heat pumps with electrically driven compressors - Testing andrequirements for marking for domestic hot water units

The current version of the standard was issued in April 2011. It specifies methods for testing and rating of heat pumps connected to or including a domestic hot water storage tank. It superseded the standard EN 255-3.

Test conditions are similar to those of EN 14511, regarding the heat source and ambient temperatures for the heat pump, Table 11.

Table 11: Test conditions within EN 16147:2011

Type of heat source	Temperature in °C (wet bulb)	Ambient temperature for heat pump in °C	Ambient temperature for storage tank in °C
Outside air, indoor installation	7 (6)	15 - 30	20
Outside air, outdoor installation	7 (6)	heat source temp.	20
Indoor air	15 (12)	heat source temp.	15
Exhaust air	20 (12)	15 - 30	20
Water	10 / 7	15 - 30	20
Brine	0 / -3	15 - 30	20
Direct evaporation	4	15 - 30	20

The uncertainties of measurement and the permissible deviations from the set values are also similar to EN 14511 and will not be given here.

The testing procedure includes the following tests:

- Heating up period – determination of the necessary time to heat up the storage from an initial state until the first turn-off of the compressor by the controls. The heating up time and the electricity consumption are measured;
- Determination of standby power input – power consumption in the standby mode is measured;
- Energy consumption and COP for reference tapping cycles – five tapping cycles according to the energy content of the hot water and type hot water usage are defined. The tapping cycles consist of a series of different types of delivery, which are provided as energy quantities, minimum temperature levels above the cold water temperature and hot water flow rates to be maintained. The consumed electrical energy is measured and corrected by the energy consumptions of fans or liquid pumps, similarly to the procedure described in EN 14511. The coefficient of performance is the ratio of the total useful heat delivered during the whole tapping cycle and the total (corrected) energy consumption during the tapping cycle.
- Determination of a reference hot water temperature and the maximum quantity of usable hot water in a single tapping – the reference hot water temperature is determined by measuring the outlet water temperature from the tank θ_{WH} after the compressor has switched off at the end of the last measurement period for the tapping cycles. The measurement lasts until the outlet temperature falls below 40°C, time t_{40}. The reference hot water temperature θ'_{WH} is calculated from eq. 21:

$$\theta'_{WH} = \frac{1}{t_{40}} \cdot \int_0^{t_{40}} \theta_{WH}(t) \cdot dt \qquad \text{eq. 21}$$

The maximum amount of usable hot water is also determined for the reference temperature difference of 30 K.

- Temperature operating range test – the tests are performed with the minimal and the maximal heat source temperatures, indicated by the manufacturer.
- Safety tests – include shutting off the heat transfer medium flows, cut-off of the power supply and condensate draining.

The efficiency figure defined by the standard, COP_{DHW}, is determined for different, non-stationary operating conditions and thus does not correspond to the definitions of the COP given in other standards. Furthermore, the system boundary includes the hot water storage, thus the storage losses are also included in the energy balance.

2.3 Standards for the assessment of the SPF

2.3.1 VDI 4650-1: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water

VDI 4650-1 "describes an easy, yet sufficiently exact, method for the calculation of the energy efficiency, which takes into account all influence quantities of technical relevance". The currently applicable version (March 2009) expresses the efficiency of the heat pump in terms of the seasonal performance factor, not as annual effort figure as previous versions. The guideline applies to electrically driven heat pumps for heating and/or domestic hot water (DHW) production up to 100 kW heating capacity. Heat sources covered by the guideline are ground water, ground (both boreholes and horizontal ground heat exchangers) and air. Only water-based central heating system is considered on the heat sink side.

The performance of the heat pump is calculated for heating and DHW applications separately and weighted according to the respective contribution to the annual energy demand. Due to differences in the practical annual temperature profile between the ground and ground water system on one hand and ambient air on the other, the guideline treats these two cases separately.

Heating application: The seasonal performance factor for space heating is calculated from eq. 22:

$$\beta_h = \frac{\varepsilon_N \cdot F_\vartheta \cdot F_{\Delta\vartheta}}{F_P} \qquad \text{eq. 22}$$

where

ε_N is the COP of the heat pump according to EN 14511 or EN 255-3 for nominal conditions;

$F_{\Delta\vartheta}$ is the correction factor accounting for deviations in the temperature difference at the condenser between the measurement and operation;

F_ϑ is the correction factor accounting for different operating conditions;

F_P is the correction factor for the energy consumption of the heat source pump.

F_ϑ is given in table form as a function of the ground/water temperature and maximum supply temperature.

$F_{\Delta\vartheta}$ is given in table form as a function of the temperature difference at the condenser during laboratory measurement and at the design point of the heating system.

For F_P, recommendations for different source systems and capacity ranges are given. Separate recommendations are given for water source systems using intermediate heat exchangers in case the water quality is poor.

For air-source systems, the SPF is calculated from eq. 23 (according to DIN V 4701-10):

$$\beta_h = (\varepsilon_{N1} \cdot F_{\vartheta 1} + \varepsilon_{N2} \cdot F_{\vartheta 2} + \varepsilon_{N3} \cdot F_{\vartheta 3}) \cdot F_{\Delta\vartheta} \qquad \text{eq. 23}$$

where

ε_{Ni} are the COPs of the heat pump according to EN 14511 or EN 255-3 for nominal conditions with different source temperatures: -7, 2 and 10°C;

$F_{\Delta\vartheta}$ is the correction factor accounting for deviations in the temperature difference at the condenser between the measurement and operation;

$F_{\vartheta i}$ are the correction factors accounting for different operating conditions at three different air temperatures, as stated above.

For $F_{\Delta\vartheta}$, the same tabular values apply as for ground coupled and water source sytems.

Values for $F_{\vartheta i}$ are provided as tables for three different heating limit temperatures: 15, 12 and 10°C. The parameter is a function of the standard outdoor temperature and the maximum supply temperature.

DHW application: The seasonal performance factor for DHW application is denominated with β_W.

For ground and ground water source heat pumps, equation analogue to eq. 26 is used. However, different ground or ground water dependent values for the correction factor F_ϑ are given in separate tables.
For ambient air-source heat pumps, an analogue procedure to the one described for the heating application is used. For heat pump systems using cellar air as heat source, the seasonal performance factor is calculated from eq. 24:

$$\beta_h = \varepsilon_N \cdot F_1 \cdot 0{,}9 \qquad \text{eq. 24}$$

where

ε_N is the COPs of the heat pump according EN 255-3 for an air temperature of 15°C and water being heated form 15 to 50°C;

F_1 is the correction factor accounting for different hot water temperatures during laboratory measurement.

F_1 is given for water temperatures of 45, 50, 55, 60 and 65°C, linear interpolation for other values is allowed.

For bivalent operation, a table containing the demand coverage α by the heat pump is provided. It is a function of the bivalent point ϑ_{Biv} and the operation mode of the system (alternate, parallel, partly parallel).

The overall seasonal performance factor of the heat pump is calculated by weighting the energy demand for heating and domestic hot water:

$$\beta_{WP} = \frac{1}{x \cdot a/\beta_h + y \cdot a/\beta_w + 1 - \alpha} \qquad \text{eq. 25}$$

If heating or DHW is not provided by the heat pump, the respective term is not considered.

In the final chapter of the document, three calculation examples for three different heat sources are provided.

2.3.2 EN 15316-4-2: Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4.2: Space heating generations systems, heat pump systems

The standard was elaborated by the CEN/TC 228 „Heating systems in buildings“. The current version was published in June 2008.

The scope of the standard covers both heating and DHW heat pumps, in alternate or simultaneous operation. The heat pumps can be driven electrically, with a combustion engine or thermally (absorption only). An overview of the considered heat sources and heat distribution systems is given in Table 12.

Table 12: Part load conditions for the heating mode (average climate)

heat source	heat distribution
air (outdoor and exhaust)	air
ground coupled (direct and indirect)	water
water (surface and ground)	direct condensation

The output data of the described calculations are:

- Driving energy of the system;
- Total thermal losses of the system;
- Total recoverable thermal losses of the system;
- Total auxiliary energy consumption.

EN 15316-4-2 describes two different methods for the calculation of the SPF, which differ regarding the needed input data, the considered operating conditions and the calculation periods:

- Simplified method based on the system typology, which delivers the SPF for the heating season. The input parameters are taken from the tables and do not take into consideration the specific configuration of the system. To use this method, a national annex is needed.
- Calculation based on the temperature bin method, which is explained in the standard itself or e.g. in [1]

The standard gives only the calculation methods, in most cases it does not prescribe which input data to use; in some cases, however, recommendations are provided.

The calculation of the SPF with the temperature bin method is performed following the defined ten steps:

1. Determination of energy requirement of every single bin;
2. Correction of steady state heating capacity / COP for bin source and sink temperature operating conditions;
3. Correction of COP for part load operation, if required;
4. Calculation of generation subsystem thermal losses;
5. Determination of back-up energy requirements of the single bins;
6. Calculation of the running time of the heat pump in different operation modes;
7. Calculation of auxiliary energy input;
8. Calculation of generation subsystem thermal loss recoverable for space heating;
9. Calculation of the total driving energy input to cover the requirements;
10. Summary of resulting and optional output values.

The cumulative heating degree hours should be given in a national annex or available from national standards.

The heating energy demand of the heating distribution system should be calculated according to EN 15316-2-3 [2]. The energy demand for each bin is calculated using a weighting factor calculation based on the heating degree hours for every bin. The domestic hot water demand is also calculated using weighting factors, similar to the heating energy demand.

The heating capacity and the COP <u>for the nominal capacity</u> should be determined according to a European standard. If possible, all relevant operation conditions should be considered or at least the operation conditions given in the standard. If the mass flows on the heat source or heat sink side differ from the design operating conditions, a correction by interpolation or extrapolation is possible.

Also, in order to cover the whole range of heat source and heat sink temperatures, the COP values should be interpolated or extrapolated from the measured values. If the COP for only one operating condition is available, a correction for both heat source and heat sink based on the constant exergetic efficiency can be performed and is described in an informative Annex.

Regarding the heat source, the following temperatures are to be used:

- For air-source heat pumps, the outside air temperature of the bin is to be used;
- For an exhaust air heat pump without heat recovery, the indoor temperature is the source temperature. If a heat recovery is included, combined test results for the heat pump and for the heat recovery unit can be used. Alternatively, an evaluation of the supply temperature according to the temperature variation coefficient of the heat recovery, e.g. according to EN 308;
- For ground coupled or water heat pumps, values from national annexes or standards should be used. If none available, an example is given in an informative annex.

For the DHW, results from the measurements according to EN 255-3 [3] are to be used. Because of oscillating source temperatures, a correction has to be performed on the bases of constant exergy efficiency, same as for the heating operation mode. If no data from the tests are available, an average DHW charge temperature can be calculated.

Finally, the overall COP is interpolated from the test data for the heating and DHW operation mods.

For engine driven heat pumps and absorption heat pumps no reference to applicable test methods is given. It is however stated, that the same corrections regarding operating conditions apply. Examples for input data from measurement for heating are given in the informative annexes.

Regarding part load operation the standard states, that the losses due to the on-off operation are negligible. They are not considered in the calculation, except if considered in the tests which yielded the input data. For the off mode, only the auxiliary energy consumption is regarded. If part load data are available from other standards, e.g. EN 14825, the COP for each operating condition (every bin) should be interpolated and a load factor is to be calculated.

For DHW operation, the start-up losses are already considered in the EN 255-3. For engine driven and absorption heat pumps, the start-up losses have to be considered in the test standards.

Total thermal losses include the losses within the energy generation subsystem, thermal losses from all storages within the system as well as losses in the primary circulation pumps. These losses are accounted for both for the operation and the stand-by times. Some of the losses are recoverable, such as the losses to the heated ambient or the thermal losses of auxiliary components to heat transfer media. These recoverable losses are calculated and added to the energy output of the overall system.

If no storage is integrated in the heat pump (electrically driven) casing, the generator heat losses for the heating operation are not considered if no national standards are available. For engine driven heat pumps, the thermal losses of the engine have to be calculated, but no specific method is given – only references to other standards and possible calculation methods.

If an internal or an external storage is part of the system, the losses to the ambient have to be calculated for every temperature bin. The stand-by heat losses are either given from the storage tests or standard values from an informative Annex are used. The mean storage temperature is obtained from the system control settings. If the temperature in the storage varies according to the heating load, the mean temperature is calculated. For the DHW storage, the same method is applied, only different temperatures according to the regulations have to be taken into account.

For the thermal losses of the primary circulation piping EN 15316-2-3 and EN 15316-3-3 [4] are to be used.

In polyvalent systems, the back-up heating is considered for two reasons:

- If temperatures in the distribution system are needed which are higher than the temperature operation limit of the heat pump;
- The heat pump was not dimensioned to cover the full heating and/or DHW load. In this case two calculation methods are described.

The simplified method is based on the evaluation of the cumulative temperature frequency and the bivalent or low temperature shut-off point. Energy fractions for the heat pump and for the back-up system are obtained. The input data for the calculation are the bivalent or the shut-off point and the operation mode – alternative, parallel or semi-parallel. In all cases, the fraction of the energy delivered by the back-up unit is calculated from the ratio of the area under the cumulative temperature frequency curve representing the energy which is not delivered by the heat pump to the total heating energy needs.

The detailed calculation is based on the evaluation of the running time for 1K bins. The detailed calculation takes also into account also the specific controller settings.

The operation time of the heat pump per bin is calculated from the produced heating energy and the respective heat pump capacity for the operating condition within a single bin.

While the estimation of the running time is quite straight forward in cases where the heat pump produces heating energy and DHW in clearly separated operation cycles, it can be quite difficult to differentiate between these two running times for the simultaneous operation mode, in which both are produced at the same time. The maximum running time in the simultaneous mode can be calculated from the minimum running time needed in both modes. This time can be corrected for different controller settings. The respective energies produced in this operation mode are calculated. From an energy balance, the fractions of the heating and DHW energies can be calculated. Finally, from these energies, the respective running time can be estimated.

For the calculation of the auxiliary energy consumption, the energy consumption of all system components should be considered. Energy already included in the testing standards has to be taken into consideration.

The energy input to the heat pump is calculated as the sum of the energy inputs for every bin, derived from the delivered heat and the heat pump efficiency under the bin operating conditions. Similar calculation is performed for the energy input to the back-up unit.

Finally, two seasonal performance factors (SPF) can be calculated:

- SPF of the generation subsystem during operation, including the heat pump itself and the back-up heater
- SPF of the overall system, including all auxiliary energy consumptions (e.g. for the heat source system and for stand-by operation)

Comprehensive information on various calculation procedures, as well as default values for different parameters used for the calculations is provided in the Annexes. In most cases, the information is available also for thermally driven heat pumps.

3 SWOT ANALYSIS OF ANALYSED DOCUMENTS

The SWOT Analysis was done for the test and rating standards EN 14511, EN 15879-1, EN 16147, and the EHPA – Test regulations. In a further step also the Analysis of the standard EN 14825 and the guideline VDI 4650 have been done.

The standard EN 14511 covers not only capacity measurement but also safety in operation and different temperature levels on sink side. EN 14511 is broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) and different funding programms in Europe. The Standard is not covering capacity controlled heat pumps and the Nominal capacity of capacity controlled HPs is not clearly defined. In EN 14511 circulation pumps are included in the testing procedure only a small amount is integrated in the calculation. As it is a standard which is referenced very often changes have a large influence to other standards.

	HELPFUL To achieving the objectives	HARMFUL To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ The standard covers not only capacity measurement but also safety in operation ▪ Different temperature levels on sink side provided	WEAKNESSES ▪ Capacity controlled heat pumps are not covered ▪ Nominal capacity of capacity controlled HPs is not clearly defined ▪ Circulation pumps are included in the testing procedure
EXTERNAL FACTORS	▪ Broadly accepted and used also as a basis for quality assurance schemes (e.g. EHPA, ErP) ▪ Broadly accepted for different funding programs in Europe ▪ Might change to an ISO standard OPPORTUNITIES	▪ Large effort to modify and adapt because referenced within a variety of other standards THREATS

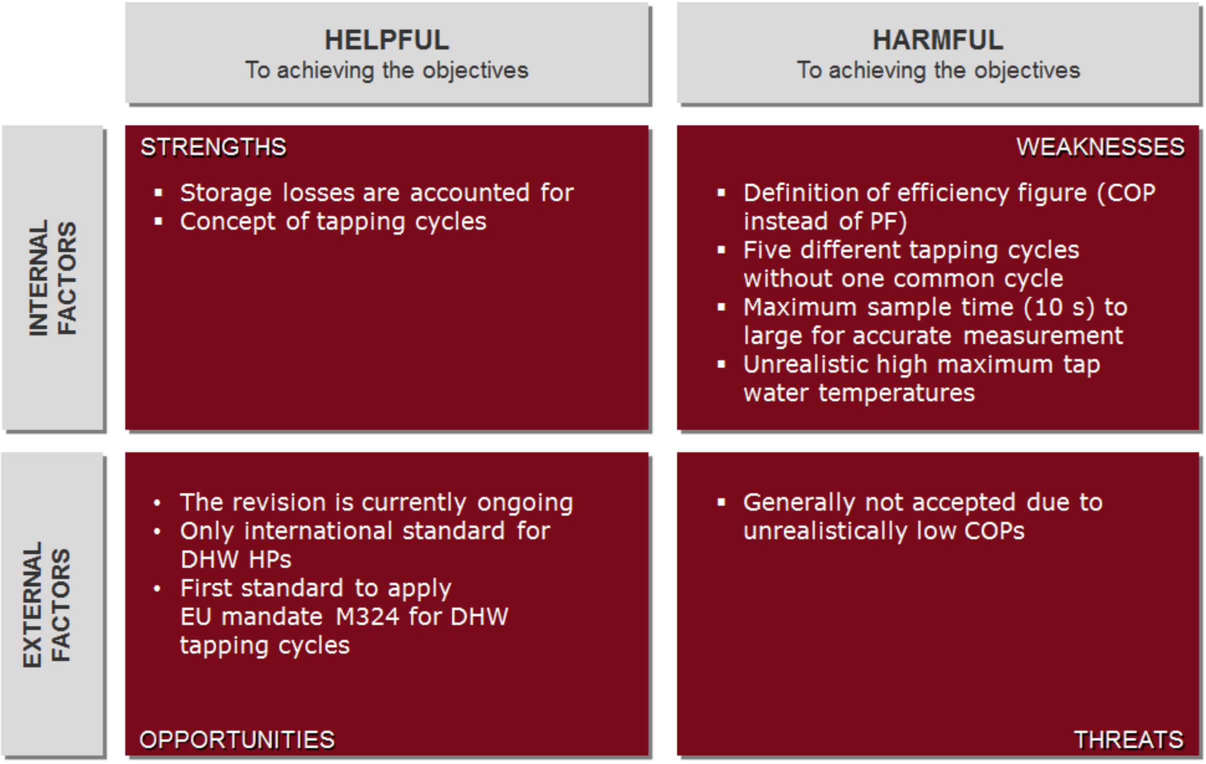
Picture 1 SWOT Analysis of EN 14511

The EN15879-1 is in principle identical to EN 14511 with all strengths, weaknesses, opportunities and threats but covering direct exchange to water heat pumps. The main problem is that it is not part of EN 14511 and therefor revisions have to be done separately.

	HELPFUL To achieving the objectives	HARMFUL To achieving the objectives
INTERNAL FACTORS	STRENGTHS ▪ The standard covers not only capacity measurement but also safety in operation ▪ Different temperature levels on sink side provided	WEAKNESSES ▪ Capacity controlled heat pumps are not covered ▪ Nominal capacity of capacity controlled HPs is not defined ▪ Only on/off heat pumps are covered ▪ Circulation pump is included in the testing procedure
EXTERNAL FACTORS	▪ Accepted and used for quality assurance ▪ Broadly accepted for different funding programs in Europe OPPORTUNITIES	▪ Closely linked but not part of EN 14511 ▪ Market of DX-HPs is decreasing THREATS

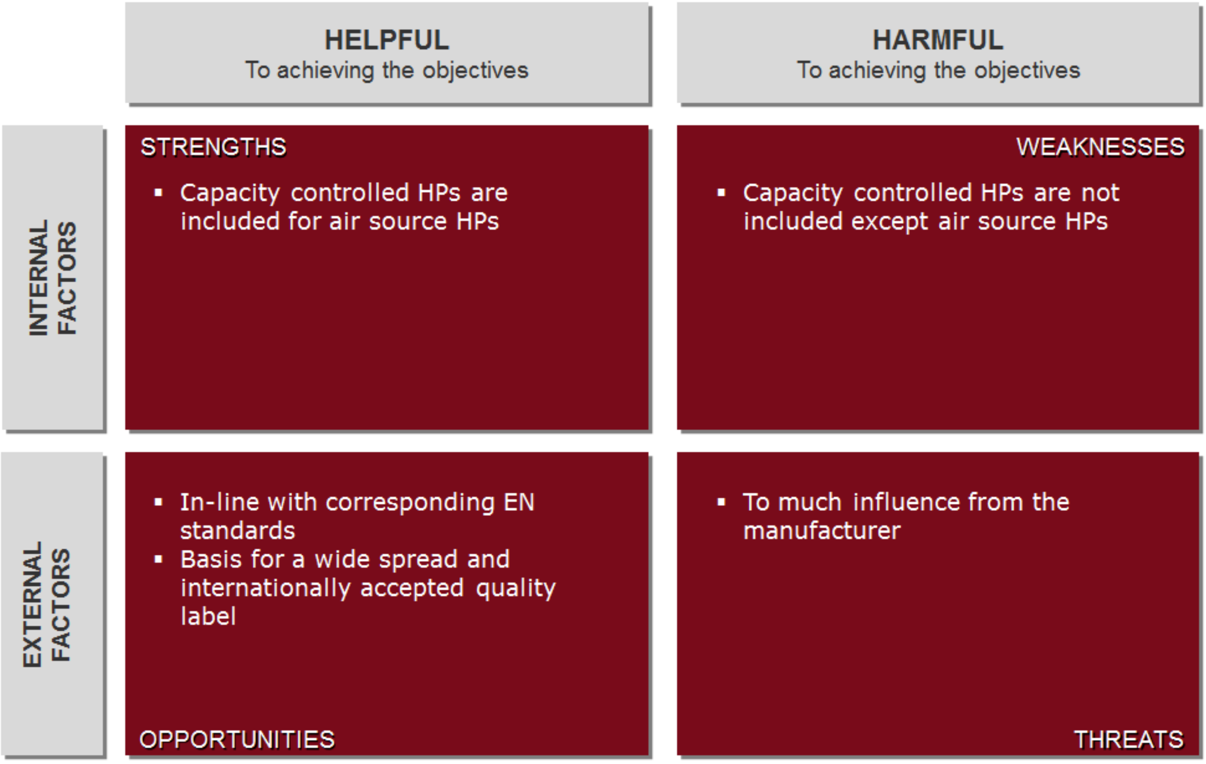
Picture 2 SWOT Analysis of EN 15879-1

The standard EN16147 covers Domestic hot water heat pumps. It provides test procedures which account storage losses and uses the concept of tapping cycles. The main weakness is that no reference tapping ccyle is defined and it is hardly possible to compare different products.



Picture 3 SWOT Analysis of EN 16147

The EHPA-Test Regulation uses the European standards with all strengths, weaknesses, opportunities and threats their as a basis. The different procedures are extended by some additional requirements which are not covered by the standards. The only additional threat of EHPA Test Regulation is that there is a large influence of manufactures.



Picture 4 SWOT Analysis of EHPA Test Regulation

LITERATURE

[1] Wemhöner, C., Afjei, T., 2003, Seasonal performance calculation for residential heat pumps with combined space heating and hot water production (FHBB method), Swiss Federal Office of Energy, Bern

[2] CEN, 2007, EN 15316-2-3: 2007 Heating systems in buildings. Method for calculation of system energy requirements and system efficiencies. Heat generation systems, thermal solar systems. CEN, Brussels

[3] CEN, 1997, EN 255-3:1997 Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors. Heating mode. Testing and requirements for marking for sanitary hot water units. CEN, Brussels

[4] CEN, 2007, EN 15316-3-3:2007 Heating systems in buildings. Method for calculation of system energy requirements and system efficiencies. Domestic hot water systems, generation. CEN, Brussels

[5] European Parliament, Council, 2010, Directive 2010/30/EU of the European Parliament and of the Council of 19 May 2010 on the indication by labelling and standard product information of the consumption of energy and other resources by energy-related products, European Parliament, Concil, Brussels

[6] Melograno, P. N., Fedrizzi, R., Sparber, W., Franchini, G, 2010, Test Procedures for Sorption Chillers Based on the Working Mode, EuroSun 2010, Proc. of the 2nd Int. Conference on Solar Heating, Cooling and Buildings, Graz

<u>Reviewed documents</u>

CEN 2000. EN12309-2:2000 Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW – Part 2: Rational use of energy, CEN, Brussels.

DIN, 1988, DIN 33830-4:1988 Wärmepumpen; Anschlußfertige Heiz-Absorptionswärmepumpen; Leistungs- und Funktionsprüfung, DIN, Berlin

JSA 2011, JIS B 8622:2009 Absorption refrigerating machines. JSA, Tokyo

CEN, 2011, EN 14511:2011 Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling, CEN, Brussels.

ANSI, 2009, ANSI/ASHRAE Standard 37-2009 Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment. ANSI, New York

AHRI, 1998, AHRI Standard 320-98 Water-Source Heat Pumps. AHRI, Arlington

AHRI, 1998, AHRI Standard 325-98 Ground Water-Source Heat Pumps. AHRI, Arlington

AHRI, 1998, AHRI Standard 330-98 Ground Source Closed-Loop Heat Pumps. AHRI, Arlington

CEN, 2011, EN14825:2011 Air conditioners, liquid chilling packages and heatpumps, with electrically driven compressors, for space heating and cooling – Testing and rating at part load conditions and calculation of seasonal performance, CEN, Brussels.

CEN, 2011, EN 16147:2011 Heat pumps with electrically driven compressors - Testing and requirements for marking for domestic hot water units. CEN, Brussels

AHRI, 2000, ANSI/ARI 560: Standard for Absorption Water Chilling and Water Heating Packages. AHRI, Arlington

ANSI, 2008, ANSI/ASHRAE 182-2008: Method of Testing Absorption Water-Chilling and Water-Heating Packages. ANSI, New York

VDI, 2003, VDI 4650-1: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water. VDI, Düsseldorf

VDI, 2010, VDI 4650-2: Simplified method for the calculation of the annual coefficient of performance and the annual utilisation ratio of sorption heat pumps - Gas heat pumps for space heating and domestic hot water. VDI, Düsseldorf

CEN, 2008, EN15316-4-2:2008 Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4-2: Space heating generation systems, heat pump systems, CEN, Brussels

10.4 Appendix IV – National report Switzerland

Schweizerische Eidgenossenschaft
Confédération suisse
Confederazione Svizzera
Confederaziun svizra

Eidgenössisches Departement für Umwelt, Verkehr, Energie und Kommunikation UVEK

Bundesamt für Energie BFE

Schlussbericht vom 23. April 2015

EFKOS – Effizienz kombinierter Systeme mit Wärmepumpe

Swiss country report of IEA HPP Annex 39 «A common method for testing and rating of residential HP and AC annual/seasonal performance»

Fachhochschule
Nordwestschweiz

Datum: 23. April 2015

Ort: Bern

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Zusammenfassung

Moderne Heizsysteme für Wohngebäude basieren heute oft auf Wärmepumpen. Durch technische Fortschritte kommen dabei vermehrt leistungsgeregelte Geräte zum Einsatz, welche eine nochmals verbesserte Effizienz versprechen. Eine Möglichkeit die Nutzung erneuerbarer Energien weiter zu steigern besteht in kombinierten Systemen mit solarthermischer Unterstützung. Hauptziel dieser Arbeit war es, eine Methode zu entwickeln, mit der die Jahreseffizienz solcher Systeme durch Normrechnungen leicht abgeschätzt werden kann. Dazu wurde zuerst ein Simulationsmodell einer konventionellen, nicht leistungsgeregelten Wärmepumpe entwickelt und validiert. Das Modell erlaubt die Bestimmung beliebiger COP Werte aus wenigen, gut verfügbaren Prüfwerten. Das Modell für Wärmepumpen mit fester Kompressordrehzahl wurde darauf aufbauend weiter modifiziert, sodass auch Kompressoren mit variabler Drehzahl abgebildet werden können. Die Simulationsergebnisse zeigen eine gute Übereinstimmung mit Datenblattwerten. Dieses neu entwickelte, halb-empirische Simulationsmodell für Wärmepumpen wurde verwendet um drei verschiedene Heizsysteme mit Luft/Wasser Wärmepumpe und optionaler thermischer Solaranlage zu modellieren. Als Wärmeverbraucher wurde ein typisches Einfamilienhaus im Schweizer Mittelland angenommen. Die Ergebnisse von Jahressimulationen dieser Systeme wurden mit der Temperaturklassenmethode etablierter Normrechenverfahren zur Abschätzung der Jahreseffizienz verglichen. Da bestehende Normen nicht explizit auf leistungsgeregelte Wärmepumpen ausgelegt sind, wurde ein geeignetes Vorgehen anhand der Schweizerischen Norm SIA 384/3 entwickelt. Mit dieser Anpassung erreicht die Temperaturklassenmethode gute Übereinstimmung mit den Simulationsergebnissen, und zwar gleichermassen für herkömmlich on/off geregelte wie auch für leistungsgeregelte Wärmepumpen mit variabler Kompressordrehzahl. Es wird daraus geschlossen, dass die Temperaturklassenmethode so angewendet werden kann, dass damit eine hinreichend genaue Bestimmung der Jahreseffizienz von kombinierten Wärmepumpen/Solarthermie-Systemen für Raumheizung und Warmwasserbereitstellung möglich ist. Mit der Europäischen Ecodesign-Verordnung für Wärmepumpen steht künftig eine genügend grosse Datenbasis zu Verfügung.

Abstract

Modern building equipment for heat generation in dwellings often relies on heat pumps. Recent progresses lead to variable compressor speed heat pump types, which promise an enhanced efficiency. An option to further improve usage of renewable energies is the combination with solar-thermal collectors. A method on how to estimate yearly performance factors in standard calculations of such systems was the main goal of this work. Therefore, an advanced simulation model of a conventional on/off controlled heat pump was developed and validated with measured data. The model allows to set up arbitrary COP values from few, well established rating points. The simulation model for fixed compressor speed heat pump types was then further enhanced by a simple modification which implements variable compressor speed heat pumps behaviour as well. This extension proves to fit well with rating data. Using this newly developed, semi empirical heat pump simulation model, three different heating systems with an air-to-water heat pump as the main heat generator and optional support by solar-thermal collectors were set up for simulations. A single family house with a configuration which is typical for Switzerland was used as a heat load. The outcomes of these simulations were compared to well established temperature class method of yearly efficiency estimation, which is frequently used in standard calculations. As existing standards are not explicitly made for variable capacity heat pumps, an implementation into Swiss standard SIA 384/3 has been elaborated. Applying these modifications for variable capacity heat pumps in each temperature class, a good correlation with simulations can be established. As a conclusion, the temperature class method can be set up to give an accurate estimation of yearly performance of combined heat pump / solar-thermal systems for space heating and sanitary water supply, be it for conventional on/off controlled, be it for modern variable capacity heat pumps. From European Ecodesign and Energy Labelling directives, a sufficiently broad data basis will be available.

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1 Introduction / Goals

The European Union as well as Switzerland, both aim for higher energy efficiency and increased use of renewable energies to reach their respective goals in energy efficiency and reduction of CO_2 emissions. The EUs energy related products (ErP) directive is the most important initiative by the EU to improve energy efficiency by 20% by the year 2020 [1]. The goals have recently been projected to 2030 and aim at a target of at least 27 % renewables share and an improvement of 27 % in energy efficiency [2]. In Switzerland, a new energy strategy has been settled in the aftermath of the Fukushima accident, relying mainly on an increased energy efficiency and higher share of renewables [3]. In the building sector, this regards amongst others heat pump systems for space heating, space cooling and domestic hot water (DHW) production. To prove the efficiency of these devices, internationally harmonized calculation methods for energy assessments are required. Yet there are such tools for conventional on/off controlled heat pumps, but as new technologies such as inverter driven (also known as variable speed) compressors for electrical heat pumps emerge, a knowledge gap opens. These units continuously adjust their heating power output to the actual demand, while conventional heat pumps usually rely on an on/off mode, cycling between full capacity and off-state when demand decreases. The new technology also promises a better efficiency.

While capacity control indeed may lead to a higher performance of the heat pump unit alone, the efficiency of an entire heating system is strongly depending on its implementation at the demand side, basically the sink temperature levels. This means that the whole system, including building quality (insulation level, solar gains through windows, etc.), heat emission system and on-site climate must be taken into account for a meaningful assessment. It's a common practice to calculate seasonal performance factors by a so called bin model, assigning different operating conditions of the heat pump to different outdoor-temperature classes. But the limits of application of such a method are unclear, especially if applied to variable compressor speed heat pumps and combined systems -systems consisting of multiple heat generators- in ultra-low energy buildings, which will become more and more common in the future.

It's one of the main goals of this project to elaborate and verify a calculation process for the assessment of continuous capacity control heat pump systems in such combined systems, consisting of the heat pump and an additional solar thermal subsystem which is designed for domestic hot water preparation and optional space heating. The method to develop should be widely accepted by manufacturers, and preferably be based on already available heat pump rating data such as EN 14511 rating points [4]. The reliability of the bin method shall be proved and –if found to be necessary– other calculation models will be suggested.

2 Method

In a first step, existing standards with relevance for the project have been studied (chapter 3). In this review, the focus was laid on space heating and sanitary hot water preparation. Besides methods used for heat pump assessment, the availability of performance rating data has been identified too in this step. Based on the outcome, a simulation model of an on/off controlled (fixed compressor speed) heat pump has been developed and implemented in Matlab/Simulink [5]. It relies on input data available through European standards EN 14511 and EN 14825 [6], the latter of which has become a major role in Europe's heat pump assessment for Ecodesign and energy labelling purposes. This will secure a wide availability of such rating data, which is required as an input to the model. The model has been validated with measurements from an air to water heat pump (chapter 4). While originally made for on/off controlled devices, it showed to be suitable for variable compressor speed heat pump types too, when some modifications are made. This has been proven by comparing simulation outcomes to datasheet rating values of an air-source device. To do so, a virtual test rig for the modelled heat pump was set up to generate rating data from simulations. This simulated test rig was also used to create data required for the subsequent examination of heat pump assessment methods found in the reviewed standards. These standard calculations have been carried out and were compared against simulation outcomes of identically set up heat generating systems. For the comparison, three typical heat generator systems based on an air to water heat pump and solar thermal subsystem have been chosen. They were applied to a single family house located in Switzerland as a representative load. The building is a simplified version of the reference building developed within joint Annex 38 / Task 44 of the IEA HPP and SHC programmes [7]. The comparisons showed a good match for on/off controlled heat pumps, while some modifications to standard calculation processes were required to reach good correlation for variable compressor speed types too. Three alternatives of such modifications have been examined, they are based on the knowledge gathered during heat pump modelling and standard review. Therefore, again, the finally suggested version relies on readily available data from EN 14511 and EN 14825 rating. The calculation then follows procedures found in SIA 384/3.

3 Existing standards

Many standards concerning heat pump testing and calculation of annual efficiency of heat pump systems for the building sector do already exist. Many of them have been revised to a greater extent in the recent past. The following sections will give a short overview of the actual situation of such standards with relevance for Switzerland. Main point of interest is which operating modes they do cover (space heating, domestic hot water, space cooling) and how variable speed compressors are treated.

3.1 EN 15316-4-2

Named SIA 384.342 in Switzerland, the European standard EN 15316-4-2 [8] is undergoing some major rework at the time of this study. The following section describes the currently applicable EN 15316-4.2 edition, dated 1.12.2008. EN 15316-4.2 is a well introduced standard for calculating efficiencies of heat pump systems for space heating and/or sanitary water preparation. The efficiency evaluation according to EN 15316-4.2 comprises two approaches: One simplified approach, mostly based on system-typology, and a much more precise calculation process taking into account object-specific parameters (such as design temperatures, local climate data and energy demand at the user side). The detailed assessment process is based on a bin model, the standard is applicable for standalone or combined heating/sanitary hot water systems, but does not fully cover part-load conditions as long as no rated data at part load is available. If COP values at full capacity ($COP_{100\%}$) and a corresponding part load (e.g. $COP_{50\%}$) condition are know from measurements, EN 15316-4-2 proposes an interpolation between these values to obtain the part load COP at actual conditions. If no specific data from heat pump rating is available, thermodynamically caused efficiency losses from on/off switching are considered negligible, assuming properly designed systems. Alone auxiliary energy consumption in compressor off states contribute to the reduction of COP at part load when on/off cycling. Regarding part-load conditions of heat pumps, EN15316-4-2 also describes how to establish the distribution of energy demand over the outdoor-temperature class bins from yearly values, as such data is rarely available. As EN 15316-4-2 is specifically made for heating systems, cooling modes are not within the scope of this standard.

3.2 SIA 384/3

The first edition of the Swiss standard SIA 384/3 [9] has been published in spring 2013. The standard fills the gap between the calculation of building energy demands Q_H and Q_{DHW} (heating and domestic hot water) and resulting final energy consumption for various heat generators, see figure 1. Regarding heat pumps, SIA 384/3 describes a very simple typology method based on a few assumptions of boundary conditions, basically a lookup-table for typical system configurations. The standard also allows a more detailed calculation which is based on EN 15316-4-2. It does not contain further information on how to handle capacity controlled units, but describes a method on how to deal with combined systems (e.g. heat pump/solar-thermal): For solar thermal heat generators in combined systems, as long as there is any (minimal) yield, it is handled by priority to cover bin demand.

SIA 384/3 also includes domestic hot water preparation in its scope, which makes this standard very tempting for regulatory energy assessments of heating systems for buildings. This reflects its original purpose. According to the underlaid EN 15316-4-2 nature, the detailed calculation method for yearly energy consumption is based on a bin method. The calculation of heat pump performance under the conditions of each outdoor temperature class is carried out by the approach of assuming a locally constant exergetic efficiency, established from the nearest neighbour rating point. Further details on this subject are given in chapter 7.2.

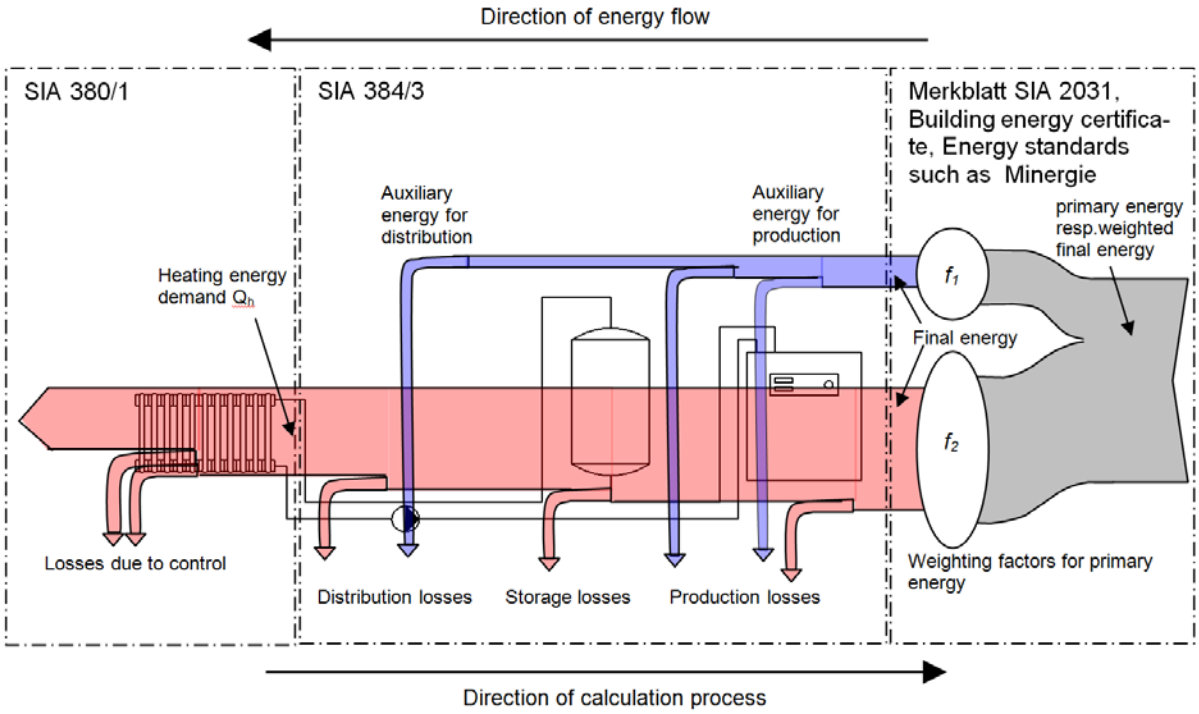


Fig. 1: ***Scope of Swiss standard SIA 384/3***

An alternative method to the setup of COP values via a locally constant exergetic efficiency is also provided. It is using a two-dimensional interpolating polynomial, found by fitting rated performance data. As far as required, default values of the temperature drop across the heat-exchangers (evaporator/condenser) are given and return temperatures from the heating system are assumed to be controlled according to an object-specific heating-curve. It's noteworthy that flow temperature and associated heating power in each bin are both found by an iterative process, as these values are depending on each other. Also, any performance values are calculated at the design mass flow of the heat distribution system, rather than the one derived from heat pump rating. From EN 14511, the latter is defined by inlet and outlet temperatures of the heating water, that is the temperature lift at the condenser at A7 condition (air-to-water heat pumps), where heat pump capacity is markedly higher than at design condition of the heating system. Therefore, the flow derived from heat pump rating is typically much higher than the flow found in an application. Again, details on the effect of this difference will be shown later in section 7.2. SIA 384/3 gives no specific information on the handling of variable capacity heat pumps, whereas the reduction of COP in on/off cycling mode is explicitly treated by adding electrical standby losses to the operational electricity consumption for each bin, as is proposed in EN 15316-4.2.

3.3 EN 14825

The newly revised European standard EN 14825 [6] about "testing and rating at part load conditions and calculation of seasonal performance" of electrically driven compressors for space heating and cooling is based on well-established EN 14511 standard about testing of (electrically) driven heat pumps. The standard is made for the assessment and comparison of a heat pump device on its own rather than for whole heating/cooling systems where object specific data would be required (e.g. climate, space heating demand, etc.). A completely overworked edition has first been published in 2012 and a newly revised version with minor changes was published in 2014. In brief, EN 14825 can be regarded as an extension of EN 14511 by

- Defining part load conditions
- Defining seasonal performance calculation procedures

The part load conditions are defined by a variety of heating curves which represent different applications such as underfloor heating systems or radiators (Tab. 1). The outlet temperatures given are understood as mean temperatures required by the distribution system. They do therefore correspond to the ideal outlet temperatures of a variable compressor speed heat pump. All of these applications are defined for each of three given climates, named "average" (climate of Strasbourg), "warmer" (Athens) and "colder" (Helsinki). They roughly represent the bandwidth of climate conditions in Europe. Seasonal cooling efficiency is designated as "SEER" (for seasonal energy efficiency ratio) while seasonal heating efficiency is called "SCOP" (seasonal coefficient of performance). For both, the calculation is based on a bin method. Heat pump operating hours in each outdoor-air temperature bin are given for each climate. SCOP includes an electric backup heater (if applicable) as well as energy consumption in all compressor off modes ("Off mode", "Thermostat off mode", "Standby", "Crankcase heater"). Yearly operating times for these auxiliary modes are also given for each climate. While SEER and SCOP calculations are based on given load values (operating times and temperatures in each bin), it's still possible to adapt these default values to user defined data.

Condition	A	B	C	D
Inlet temperature (outdoor air)	-7 ºC	2 ºC	7 ºC	12 ºC
Outlet temperature (sink side)	34 ºC	30 ºC	27 ºC	24 ºC
Part load ratio	88 %	54 %	35 %	15 %
Operating hours	24	320	326	169

Tab. 1: ***Example of part load definition from EN 14825:2013 (excerpt). Low temperature application (35 ºC outlet at $T_{designh}$ of -10 ºC), average climate, air-to-water heat pump. Overall operating hours in the average climate are 4'910.***

As mentioned above, SCOP calculation is based on a bin method with given runtimes for each outdoor temperature class (bin). Degradation of COP for on/off controlled heat pumps under part load condition is treated by a degradation factor, named here as f. For air-to-water heat pumps it is

$$f = \frac{CRu}{CC \cdot CRu + (1 - CC)} \tag{1}$$

CC	Degradation coefficient
CRu	Capacity ratio

The capacity ratio CRu is the ratio of the heat demand at the user side (load) and the maximum capacity of the heat pump at the same temperature conditions:

$$CRu = \frac{\dot{Q}_{demand}}{\dot{Q}_{max}} \tag{2}$$

The degradation coefficient CC accounts for additional electricity consumption during compressor off states while on/off cycling at part load. Any other contributions are neglected. From EN 14825:

$$CC = 1 - \frac{\text{measured power of compressor off state}}{\text{total power input (full capacity at the part load conditions)}} \tag{3}$$

If no measured data for the degradation coefficient is available, a worst-case default value of CC = 0.9 shall be used, such that formula (1) simplifies to

$$f = \frac{CRu}{0.9 \cdot CRu + 0.1} \tag{4}$$

From (3), the default value CC = 0.9 would correspond to a standby power consumption of 10 % of operational power input (compressor "on"), which lies in a typical range of some kilowatts. Usually, compressor off power states consume far less electricity. It is therefore strongly advisable to use measured data instead.

The effective part load COP for each bin at outdoor-air temperature Tj can then be established from maximum capacity COP as

$$COPbin(Tj) = COPd \cdot f \tag{5}$$

COPd	COP for maximum capacity at same temperature conditions

This path of estimating part load COP is referenced in ISO 13612-2, where its application to variable compressor speed type heat pumps is described in more depth. Further details on this will be discussed in chapter 3.4.

While both, space heating and cooling are covered by EN 14825 for on/off controlled as well as for variable capacity heat pumps, sanitary water preparation is not in the scope of this standard. Also, no additional heat generators (e.g. solar thermal) can be included in the assessment, besides direct electric backup heaters which are an inherent part of the SCOP determination. The electric backup is treated as a parallel heater which kicks in at high load situations when heat pump capacity is no longer sufficient to cover the demand ($T < T_{bivalent}$). By definition, this add-on heater is included in the SCOP system boundaries of EN 14825.

Though EN 14825 -taken alone- is unsuitable for energy efficiency calculations of combined systems, there is an option to rate combinations of multiple heat generators within the framework of the Energy Labelling directive of the European Union [10]. As both, the Ecodesign and Energy Labelling directives [10][11] are based on EN 14825 calculations, the so called "package label", the energy label for combined systems can be understood as the logical supplement of EN 14825 assessments for heat pump only systems. However, the package label is intended to be easily issued by following a few simple rules only. It is made to give a rough indication of the overall performance to the end user, but cannot accurately describe a complex system in detail. As another shortcoming, the package label can only be calculated for a few given combinations of heat generators. It's expected that the current selection will be complemented in the future.

3.4 ISO 13612-2

The first edition of the international standard ISO 13612-2 [12] about "Heating and cooling systems in buildings - Method for calculation of the system performance and system design for heat pump systems" has been published in May 2014 and "presents methods for the calculation of additional energy requirement of a subsystem in order to satisfy distribution system demand" for space heating, space cooling and combined operation including DHW preparation. The presented method is based on a time-step procedure with hourly calculations as default time step, but well established bin methods may also be applied. As input values, overall energy to be delivered to the distribution system is required for each time step. In contrast to the Swiss SIA 384/3 standard, processing of annual load data to provide demand values for each time step/bin is not treated in ISO 13612-2. Although, methods to establish the coefficient of performance data for each operating condition (time step or bin) from few rating points are thoroughly discussed in the informative Annexes A through C. The calculation process for the COP estimation of on/off controlled and inverter driven heat pumps as described in annex C relies on the procedures found in EN 14825. The relation between actual capacity ratio -called X instead of CRu from EN 14825- and the part load factor -now with subscript X- f_x which normalizes part load COP to full load condition is shown in figure 2. The curves are plotted for both, on/off controlled (top) and variable compressor speed heat pumps (bottom). If applicable, default values given in ISO 13612-2 were used. In any case, at the boundary X=1, the COP is identical for full and part load capacitie, as both states are identical by definition. Thus $f_x(X=1) = 1$. At the opposite side, X=0, the compressor remains off all the time at the part load condition, as the heating demand is zero. Part load COP is therefore zero as well, as no heat is generated. Therefore $f_x(X=0) = 0$.

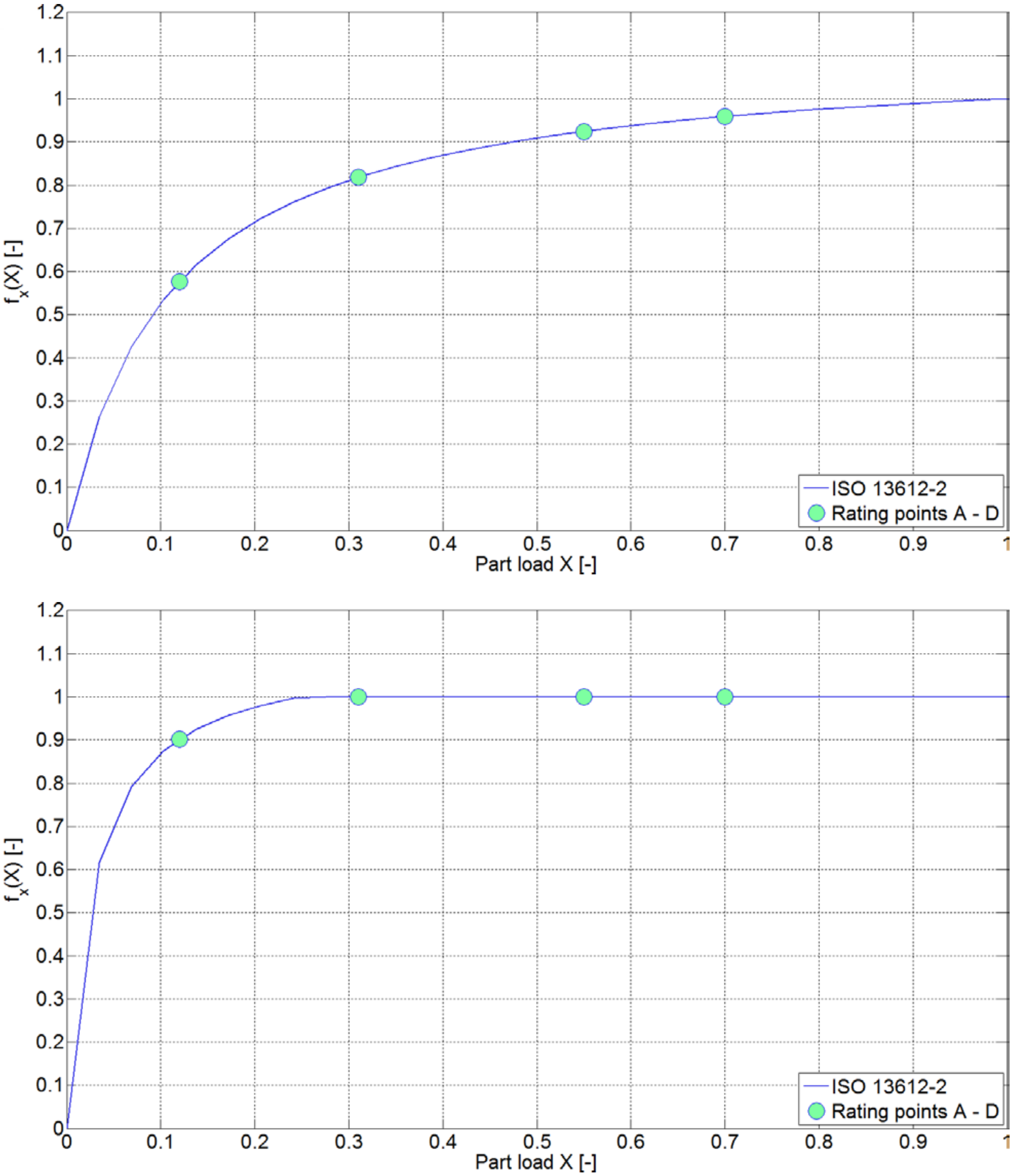


Fig. 2: ***Part load factors f_x for on/off controlled (top) and variable speed (bottom) compressor heat pumps as defined from ISO 13612-2 (default values). Part load rating points are named according to EN 14825 and are arbitrarily chosen for illustration.***

For inverter driven compressors when no measured data is available f_x is assumed to be

$$f_X = 1 \qquad (6)$$

for part load ratios X≥25 % where compressor speed can be regulated. For low part load ratios (X<25 %), it's assumed that the unit falls back to an on/off operation mode, thus the formula (4) presented in EN 14825 is used for these conditions again, now applied to 25 % of full capacity:

$$f_X = \frac{X \cdot \frac{100}{25}}{0.9 \cdot X \cdot \frac{100}{25} + 0.1} \qquad (7)$$

$$X < 0.25$$

Similarly, for on/off controlled heat pumps f_X is

$$f_X = \frac{X}{0.9 \cdot X + 0.1} \qquad (8)$$

$$0 \leq X \leq 1$$

Though establishing the f_X function from measurements accepts rated performance data at any part-load condition X, it’s recommended to use those defined by EN 14825 (named A, B, C, D, E or F) for consistency reasons amongst standards. It is however a drawback that f_X has to be evaluated from part- and full-load at the same temperature levels, which are -at least at the user side- usually not identical with EN 14511 full capacity rating conditions.

3.5 Summary

From the chapters above, the following table gives a summary of which topics of interest are covered by actual standards.

Standard	COP setup (variable speed)	Calculation Method	Bin demand from yearly value	Combined systems[1]	Sanitary hot water mode	Space cooling mode
EN 15316-4-2	no	bin	yes	no	yes	no
SIA 384/3	no	bin	yes	yes	yes	no
EN 14825	no	bin	no	no	no	yes
ISO 13612-2	yes	Time step	no	no	yes	yes

[1] Except direct electric back-up

Tab. 2: ***Overview of scope of the reviewed standards. Additionally to the table above, all standards do also include space heating mode.***

As a conclusion of the previous sections, further elaborations are mainly based on EN 14825 as a source of rated performance data, SIA 384/3 for handling the combination of heat pump and solar-thermal heat generators and finally ISO 13612-2, which describes most extensively procedures how to handle part load efficiencies of variable compressor speed heat pumps.

4 Heat pump modelling

It's one of the main goals of this work to prove and -if necessary- elaborate new calculation methods for yearly efficiency ratings of combined heat pump system. Therefore, simulation models of heat generators using heat pumps have been setup in the Matlab/Simulink simulation environment, using the CARNOT blockset [13] for system modelling. The results of yearly simulations are then regarded as a reference for comparisons with calculation procedures taken from the standards described in section 3. At first, a simulation model for heat pumps has been developed. The detailed modelling is described in the following sections. It has been divided into two subtasks:

- Setup for an on/off controlled heat pump
- Extension for capacity controlled heat pumps

The implementation into Matlab/Simulink is based on [14], a model that has already been incorporated in the CARNOT blockset. The model described here uses a refined calculation procedure, which follows the approach of a constant quality grade. Therefore, additional parameters which allow a simple modelling of the behaviour of the evaporator and condenser are required.

4.1 Introduction

The heat pump model presented here has first been developed in collaboration with the German heat pump association (bwp) for a calculation tool which allows the generation of the European energy label for heat pumps, as required from September 26th 2015 on throughout the European Union (Fig. 3). This web based calculator is available to bwp members and allows to compute EN 14825 part load performance data for fixed speed heat pumps from a few rating points, determined according to well established EN 14511 (2011 or 2007 edition).

Fig. 3: ***Functionality of the bwp energy label tool. If no EN 14825 data is available, at least six EN 14511 rating points are required as input (fixed compressor speed air-to-water heat pumps).***

EN 14825 seasonal performance calculations for air-to-water heat pumps in the average climate of Strasbourg -which is also typical for the Swiss plateau- require performance data from A-7, A2, A7, A12 conditions as well as for the operating limit temperature ($T_{operating\ limit}$) and bivalence temperature ($T_{bivalent}$), see figure 4. As bivalence temperature, any value below 2 ºC (average climate) may be chosen by the manufacturer. $T_{bivalent}$ is not a design value of the heat pump itself, rather than a suggested setting for its application. For any application and climate, a different EN 14825 dataset is required to evaluate corresponding SCOP. All this data can be generated from EN 14511 ratings by the tool (fix compressor speed heat pumps). It requires the input of a minimum of six rating points for air-to-water heat pumps, namely A-7, A2 and A7, each set at two different outlet temperatures (e.g. W35 and W55). Four of these rating points (at A2 and A7) also have to be evaluated without defrosting of the evaporator, as data from EN 14511:2007 and before revisions do not include A12 measurements, where no defrosting should occur. An extrapolation from A2 and A7 -both evaluated without defrosting operation- is therefore suggested in these cases. While performance ratings may be conducted at any pair of fixed outlet temperatures, W35 and W55 conditions are preferred, as depicted in figure 4. These sink-temperature levels cover the typical range of outlet temperatures defined by EN 14511 / EN 14825. For modern low energy buildings W55 is typical for sanitary hot water preparation while W35 is needed for space heating.

Figure 4 gives a graphical overview of which EN 14511 rating points are required (blue) and which can be extrapolated or interpolated if no measured values are available (green). For the sake of convenience, the operating limit temperature was set to design temperature (-10 ºC) in this example. Heat pump performance at the operating temperatures along the given heating curve can then be established from these EN 14511 rated points by an iterative extrapolation (or interpolation) process from W35 and W55 conditions. This process also implements overtemperature at the heat pump outlet due to on/off cycling, as described in EN 14825 Annex D.

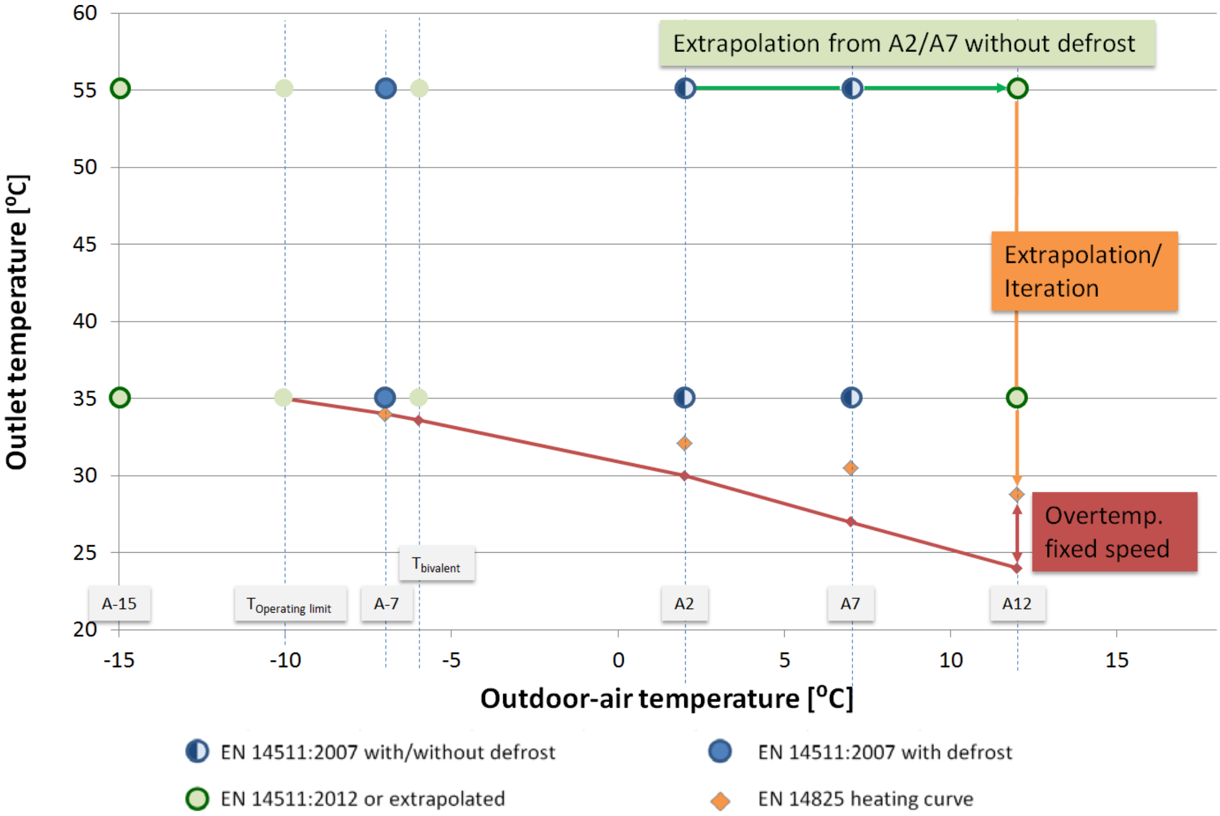


Fig. 4: ***On a basis of six measured test points all required performance data for an EN 14825 assessment process can be calculated (Heating curve according to EN 14825 average climate, low temperature application). Example of a fixed compressor speed heat pump.***

Details of the heat pump modelling -which is based on this web calculator- are given in the following section 4.1.1. It is based on the assumption, that performance data as required by Ecodesign and energy labelling directive of the European Union will be widely available and can therefore be used as input parameters. Besides data from active heating modes (COP, capacity, electrical power) this includes also power consumption in non-active modes ("Standby" and "Off" modes).

4.1.1 On/Off heat pumps

Heat pump modelling was realised with regard to practical usefulness, that is availability of required input data and -as far as possible- accordance with standards. The following explanations are therefore strongly built upon rating data from an exemplary air-to-water heat pump, which was also used for validation of the model. Looking at EN 14511 rating data in table 3, one can observe that there is no strong dependency between heating capacity and flow temperature, as long as conditions on the source side are kept identical. The table shows rating data of the aforementioned air-to-water heat pump. As can be seen, heating capacity reduction of the unit is well below 5 % if flow temperatures increases from 35 ºC to 55 ºC, which is almost the full range of typical outlet temperatures required by modern high efficiency buildings with low flow temperatures.

Quantity	Condition	A-7	A2	A7	A12
Capacity	W35	7.32	9.04	10.21	11.18
Capacity	W55	7.05	8.78	9.98	10.98

Tab. 3: Heating capacity vs. flow temperature of an air-to-water heat pump. Measurements at different source temperatures. Data courtesy of Vaillant Deutschland GmbH & Co. KG (The unit does not necessarily correspond to a series production model).

This leads to the assumption, that for any given source temperature Axx, a simple linear interpolation between outlet temperatures is accurate for the determination of heating capacity at any arbitrary outlet. Unfortunately, while rated data is based on given outlet temperatures, in a real-world application usually return temperatures from the heating systems are known. This is important as today's heat pump technology will lead to an overtemperature of the outlet when demand is low, that is at higher outdoor air-temperatures when heating capacity increases. To correctly model that effect, outlet temperature ($T_{sink,out}$) and heating capacity are calculated in an iterative process based on the given inlet (return) temperature $T_{sink,in}$ and mass flow rate:

$$T_{sink,out}(n+1) = T_{sink,in} + \frac{\dot{Q}(T_{sink,out}(n))}{\dot{m}_{sink} \cdot c} \qquad (9)$$

n	Iteration step
$\dot{m}_{sink}$	Mass flow rate of heating water
$\dot{Q}(T_{sink,out}(n))$	Heat pump capacity at given temperature

Once sink and –given– source temperatures are known, the corresponding COP and electric power consumption may simply be found in a look-up table of rating points, using linear interpolation for intermediate temperature values.

While this will already show fairly good results, it can be improved by taking more care of the electric power consumption, which is much more influenced by the outlet temperature (again at equal source conditions) than heating capacity. Table 4 shows measurement data for the same heat pump that was evaluated in table 3.

Quantity	Condition	A-7	A2	A7	A12
El. power	W35	2.39	2.42	2.30	2.30
El. power	W55	3.20	3.28	3.23	3.25

Tab. 4: ***Electric power consumption vs. flow temperature of an air-to-water heat pump. Measurements at different source temperatures. Data courtesy of Vaillant Deutschland GmbH & Co. KG (The unit does not necessarily correspond to a series production model).***

Comparisons between calculated an measured data show, that the improvement in accuracy achieved by a more sophisticated calculation becomes evident especially at higher flow temperatures and low loads. Instead of simply looking up electrical power consumption, this value can be evaluated via a (linear) interpolation/extrapolation of the exergetic efficiency η_{ex} from the nearest known values (Again, for best results it's important to use rating data close to the operating limits):

$$P_{el} = \frac{\dot{Q}}{COP_{Carnot}(T_{source}, T_{sink}) \cdot \eta_{ex}} \tag{10}$$

η_{ex} interpolated from nearest neighbour rating points

The evaluation of (10) requires the knowledge of the ideal (maximum) COP = COP_{Carnot} of a heat pump which is described by Carnot's law and is depending only on temperature levels of the cold and hot temperature reservoirs. The Carnot process corresponds best to the refrigerant cycle of a real heat pump, so temperatures at the refrigerant level are needed. This introduces more parameters to the model: Mean temperatures at the evaporator and condenser (that is both, inlet and outlet temperatures) and temperature drop across each of these heat exchangers (Fig. 5). Using heating capacity, outlet temperature at the condenser, inlet temperature at the evaporator and corresponding flow rates as input values –given or calculated according to the section above– the temperatures at the refrigerant cycle can then be calculated. Looking at the sink side:

$$T_{sink,ref} = \frac{T_{sink,in} + T_{sink,out}}{2} + \Delta T_{cond} \tag{11}$$

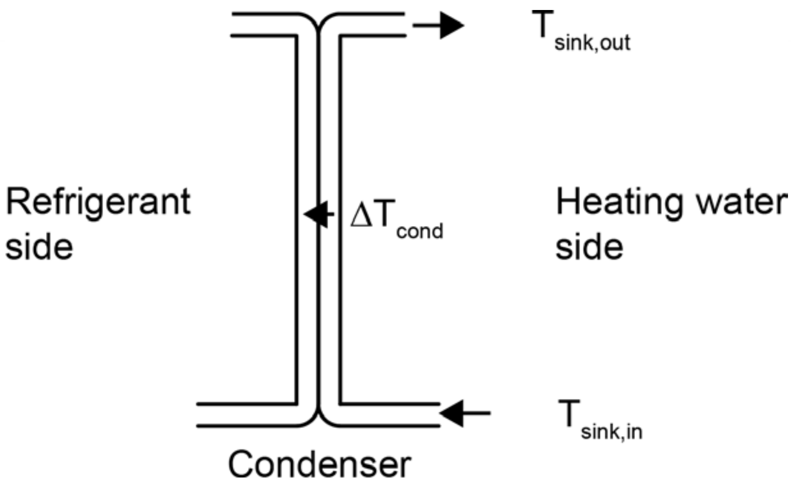


Fig. 5: Evaluation of refrigerant temperature at the heat pump condenser.

While the first summand of (11) can easily be determined from

$$T_{sink,out} = T_{sink,in} + \frac{\dot{Q}_{actual}}{\dot{m} \cdot c} \tag{12}$$

temperature drops ΔT_{cond} and ΔT_{evap} which are depending on the actual heat-exchanger design are usually supposed to be constant for the entire operating range. The model described here estimates actual temperature drop by assuming that this value is proportional to the actual power transmission. Still expressed at the sink side:

$$\Delta T_{cond} = \Delta T_{cond,rated} \cdot \frac{\dot{Q}_{actual}}{\dot{Q}_{rated}} \tag{13}$$

Formula (13) implies, that the NTU is assumed to remain constant over the whole operating range. While shown here for the hot (sink) side of the heat pump, the formulas above can be applied to the source side by just using corresponding subscripts and adjusting operation signs (-/+) where necessary. Finally the Carnot COP at the actual operating point can be evaluated from mean temperatures of the refrigerant:

$$COP_{Carnot} = \frac{T_{sink,ref}}{T_{sink,ref} - T_{source,ref}} \tag{14}$$

It can be seen that another iteration process –at the cold side– is required to find outlet temperature and cooling capacity, which is implicitly required in the Carnot COP calculation. Once all these calculations are carried out, (10) can finally be evaluated too.

4.1.1.1 Implementation

The outlined modelling of a heat pump has been implemented in Matlab/Simulink and follows straightly the path from section 4.1.1. At each simulation time-step a few iterations are required.

The following, essential parameters are needed for the full description of a heat pump in the so called EFKOS-model:

- Heating capacity ($\dot{Q}$) and electrical power consumption (P_{el}) according to EN 14511 at each rated point, evaluated for two different outlet temperatures (preferentially near the operation limits, e.g. at W35 and W 55).
- Temperature drop/rise across the evaporator ($\Delta T_{evap,rated}$) and condenser ($\Delta T_{cond,rated}$) at a specified source/sink temperature pair
- Flow rates at sink ($\dot{m}_{sink,rated}$) and source ($\dot{m}_{source,rated}$) side according to EN 14511 rating conditions (A7)

A thermal inertia of the heat exchangers (and surrounding circuitry) has also been implemented to account for dynamic effects. Further, losses to the ambient are considered by a heat loss coefficient UA. Both values are required, but can be set to zero for reasons of simplicity:

- Heat capacity of condenser (C_{cond}) and evaporator (C_{evap})
- Heat loss coefficient to ambient (UA)

Input values that are required during runtime of the simulation are:

- Actual mass flow at sink side ($\dot{m}_{sink}$)
- Actual inlet temperature at sink side ($T_{sink,in}$)
- Actual mass flow at source side ($\dot{m}_{source}$)
- Actual inlet temperature at source side ($T_{source,in}$)
- Ambient temperature at the site where the unit is installed ($T_{ambient}$)

Figure 6 below shows some simulation outputs of the air-to-water heat pump that has been modelled. The parameter $\Delta T_{cond,rated}$ was chosen to be 5 ºC (water sink) at A7/W35, while for the evaporator $\Delta T_{evap,rated}$ was assumed to be 10 ºC (air source) at A7/W35. While the simulation has been executed for different mass flow rates on the sink side, all other inputs except inlet temperature at the sink side were kept constant. The picture shows the step response for a change of the inlet temperature (return from the heating system) fed to the heat pump at t = 7200 s. As can be seen, COP and outlet temperature are both depending on actual mass flow, as expected for "real" heat pumps. The depicted step in outlet temperature also shows the retarding effect of the thermal inertia simulated by a thermal capacity of some tens of kJ/K for both, evaporating and condensing circuitry.

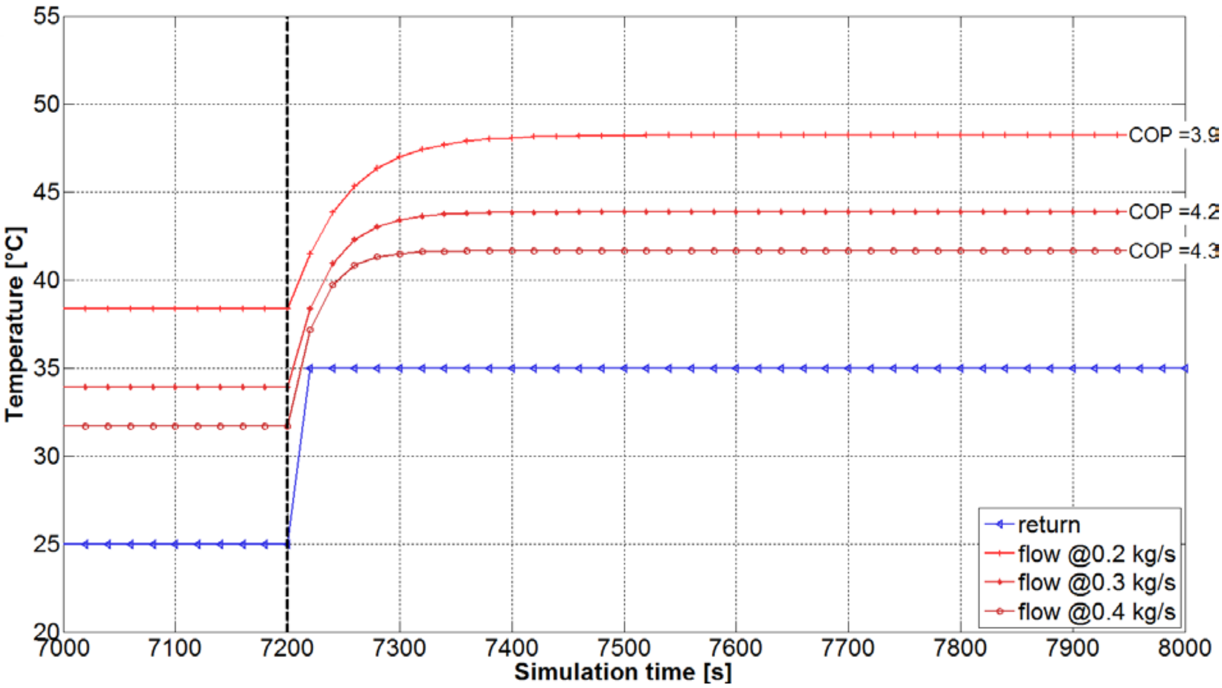


Fig. 6: ***Outlet temperatures and COP for a given step of the inlet temperature (return). Dependency on mass flow at A12 condition. Simulation output.***

4.1.1.2 Validation

For the validation process of the heat pump model, EN 14511:2012 and EN 14825:2012 measured data was available (Table 2 and Table 3). Again, $\Delta T_{cond,rated}$ was chosen to be 5 ºC while $\Delta T_{evap,rated}$ = 10 ºC, both at A7/W35 rating conditions. Air temperature drop at the evaporator was assumed to be at a constant level of 5 ºC. Mass flow at each measured operating point is known and has been set accordingly in the simulation.

Quantity	Origin of data	Unit	Rating condition			
			A-7	A2	A7	A12
Inlet temperature	measurement/ simulation	ºC	30.6	27.8	25.7	23.5
Outlet temp.	measurement	ºC	34.0	32.1	30.5	28.8
	simulation	ºC	34.1	32.1	30.6	28.9
Heating capacity	measurement	kW	7.04	8.98	10.03	11.16
	simulation	kW	7.34	9.08	10.26	11.24
el. Power	measurement	kW	2.35	2.28	2.11	2.01
	simulation	kW	2.36	2.29	2.09	1.98

COP	deviation from measurement	%	+3.7	+0.8	+3.6	+2.2

Tab. 5: ***Comparison of measured and simulated performance data for average climate, low temperature application acc. EN 14825:2013 (0.5 kg/s mass flow). Maximum deviation of COP is 3.7 %, which lies well within measurement uncertainty.***

Quantity	Origin of data	Unit	Rating condition			
			A-7	A2	A7	A12
Inlet temp.	measurement/ simulation	ºC	46.1	38.2	33.5	29.0
Outlet temp.	measurement	ºC	52.0	45.4	41.7	37.9
	simulation	ºC	51.8	45,3	41.6	37.9
Heating capacity	measurement	kW	7.24	8.99	10.30	11.25
	simulation	kW	7.09	8.90	10.14	11.15
el. Power	measurement	kW	3.08	2.78	2.52	2.35
	simulation	kW	3.06	2.83	2.56	2.36
COP	deviation from measurement	%	-1.3	-2.9	-3.2	-1.5

Tab. 6: ***Comparison of measured and simulated performance data for average climate, high temperature application acc. EN 14825:2013 (0.3 kg/s mass flow). Maximum deviation of COP is 3.2 %, which lies well within measurement uncertainty.***

From the last row of the tables above, the maximum deviation of simulated COP values from the measured ones is constantly well below 4 %. Thus, the simulation shows good correlation with measured values, especially when keeping in mind that measurement uncertainty according to EN 14511 is 6 % for COP and component tolerances may even be higher [15]. The inclusion of power transmission depending temperature drop/rise across condenser/evaporator in the model leads to somewhat better results at higher flow temperatures, but does show nearly no difference elsewhere.

4.1.2 Variable speed heat pumps

The fixed compressor speed heat pump model described in the previous section has been extended to a variable speed type. The input parameter controlling part load is defined in accordance with standard calculations as the capacity ratio X of heat demand (=effective heat output) and the maximum capacity of the heat pump at the same temperature levels (source inlet, sink outlet). In system simulations presented later, X is an input value to the heat pump from the heating controller.

The decreased heating capacity at part load is modelled as a scaling of the heating capacity at full speed as:

$$\dot{Q} = \dot{Q}_{max} \cdot X \quad (15)$$

$$X = 0.3 \ldots 1.0 \quad (16)$$

As there usually is a lower limit of compressor speed, another parameter (X_{min}) taking control of this minimum reachable speed is introduced to the model. In (16) it has been set to X_{min} = 0.3. For part load values $X < X_{min}$ fed to the model, the factor saturates internally at X = 0.3 and the heat pump switches to on/off cycling mode again as rotational speed and associated capacity of the compressor cannot be reduced any further. This is why the heat output starts to rise again at higher outdoor air temperatures, as shown in figure 7. The figure shows the dependency of the (part load) heating capacity from source temperature, it has been plotted for the heating curve of the low temperature application in the average climate as defined by EN 14825.

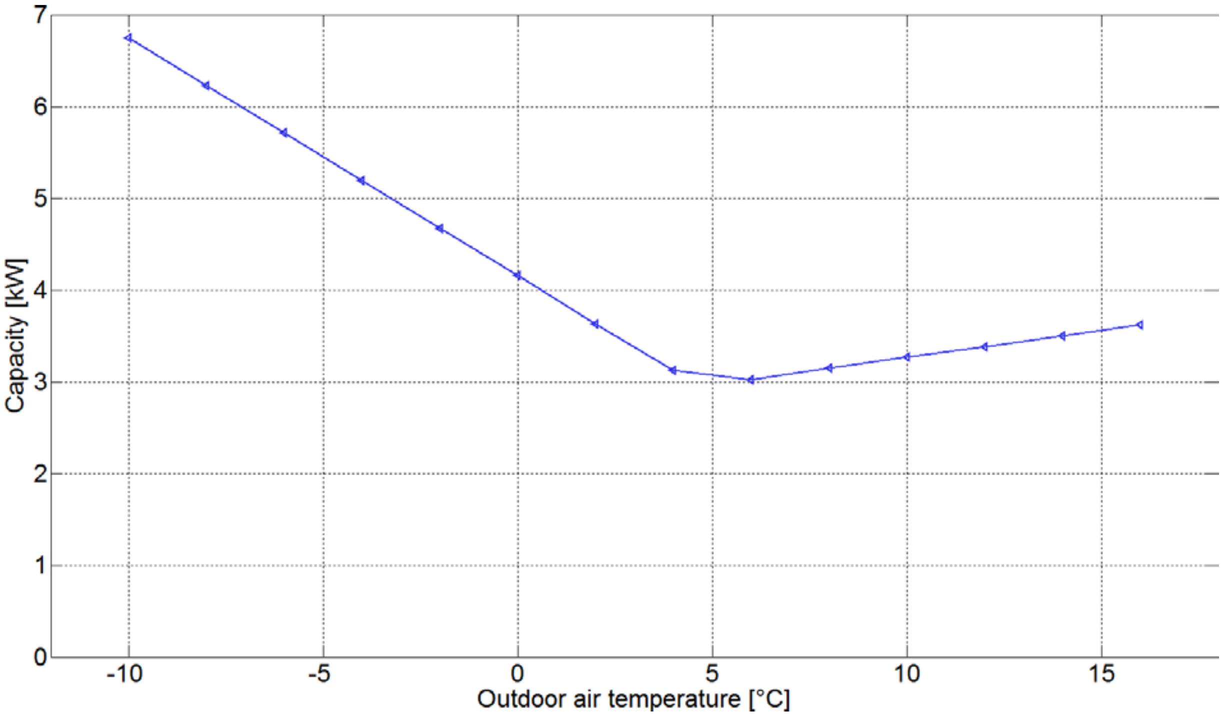


Fig. 7: ***Heating capacity vs. outdoor air temperature of the modeled variable compressor speed heat pump (air-to-water). Monovalent operation, 35 ºC design flow temperature, variable outlet temperature, EN 14825 average climate. The figure depicts capacity in compressor on states (no on/off cycling) rather than real operation, which explains the rising branch for T > 6 ºC when rotational speed cannot be reduced any further.***

When heating capacity is decreased as a consequence of lower compressor speed, several components contribute to the degradation (or improvement) of efficiency compared to full speed. One effect is already included in the model as a direct result of the heat pump modelling described in section 4.1.1: The modelling of the evaporator/condenser is such, that

COP rises at part load when power transmission at the heat exchangers is lower than at full capacity. Due to resulting lower temperature "losses" across the heat exchangers (compare formula 13) the theoretical COP rises when compared to maximum capacity situation: As capacity decreases, temperature drop (rise) across evaporator (condenser) linearly decreases (ΔT_{cond} in figure 8) as a consequence of the lower power transmission, giving higher (lower) refrigerant temperatures and therefore a lower temperature lift. At the condenser, there is a contrary effect which is also shown in the example of figure 8: Power transmission also influences mean temperature levels at the external sides of heat exchangers. When the outlet temperatures and flow rates are held constant -the situation that arises during heat pump rating-, the mean temperature at the condenser rises when the heat pump capacity is decreased. Taken alone, this would give a higher refrigerant temperature and therefore a higher temperature lift. The combination of the two effects described can be expressed as (at the sink side):

$$T_{sink,ref}(\dot{Q}) = T_{sink,out} - \frac{\dot{Q}_{actual}}{2 \cdot \dot{m} \cdot c} + \Delta T_{rated} \frac{\dot{Q}_{actual}}{\dot{Q}_{rated}} \quad (17)$$

The dependency of T_{ref} and therefore COP from heat pump capacity $\dot{Q}$ according to (17) results in overall lower refrigerant temperatures at the condenser when capacity decreases (Figure 8). Similarly, the refrigerant temperature at the source side rises, and therefore overall higher COPs are reached at part load conditions. Figure 9 shows this COP increase from simulations. It saturates at +30 % (X < 0.3) when compared to maximum capacity COP (X = 1).

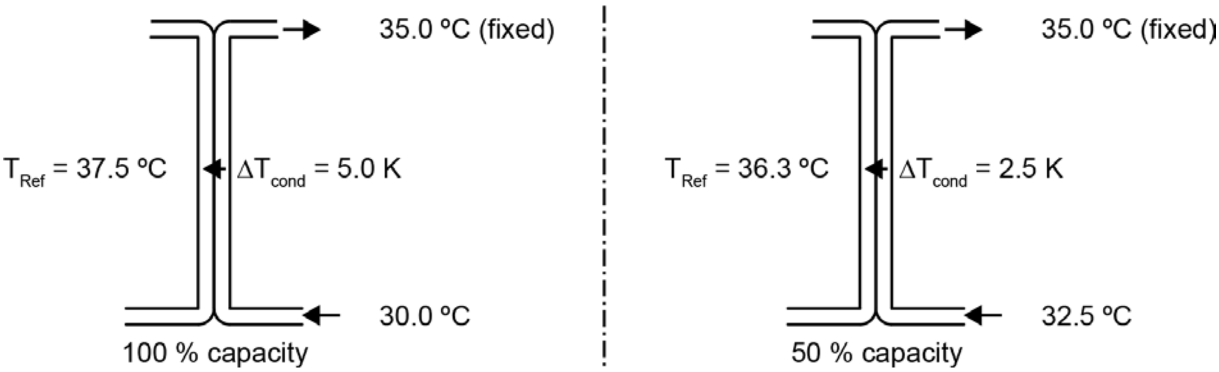


Fig. 8: ***Temperature levels at the condenser when capacity changes. Refrigerant temperature decreases with heating capacity due to reduced "transmission losses" ΔT_{cond} (arbitrary example).***

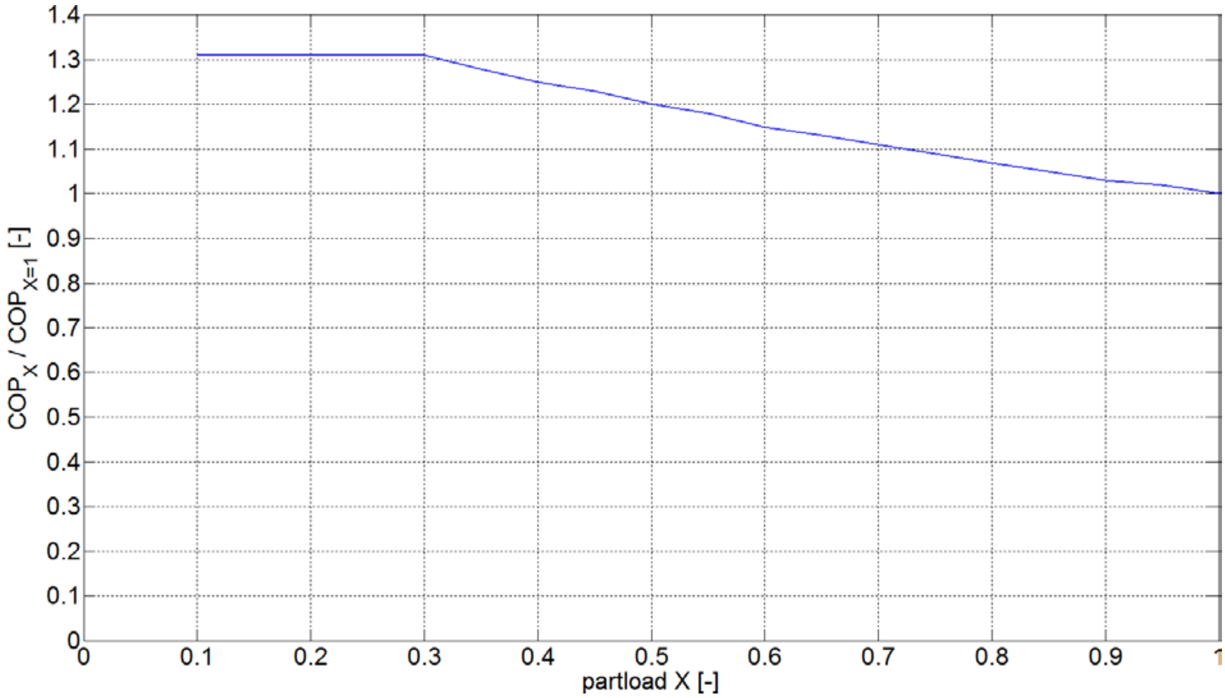


Fig. 9: ***COP at part load X normalized to full load (X=1). Besides capacity decrease, no other effects are included in the simulation model. From shifts in temperature levels, the COP will rise when X is decreased. If X < 0.3, capacity is no further reduced (A12/W35 condition) and the COP_X / $COP_{X=1}$ ratio remains constant. Simulation output.***

From datasheet values [16] of an air-to-water heat pump, the COP evolution at part load has been compared to the output of the simulation model in figure 9. The measured performance values have been taken from an inverter controlled Viessmann Vitocal 300-A AWO-AC 301.B11 heat pump with a rated capacity of 7.2 kW (A7/W35). Figure 10 shows rated capacity at a constant outlet temperature of 35 ºC. The depicted load curve is taken along the control range of the compressor, which is defined by A-5 and A2 points. At outdoor air temperatures of 7 ºC and above, the lower limit of compressor speed is reached and on/off control sets in again to follow the reduced load as good as it gets.

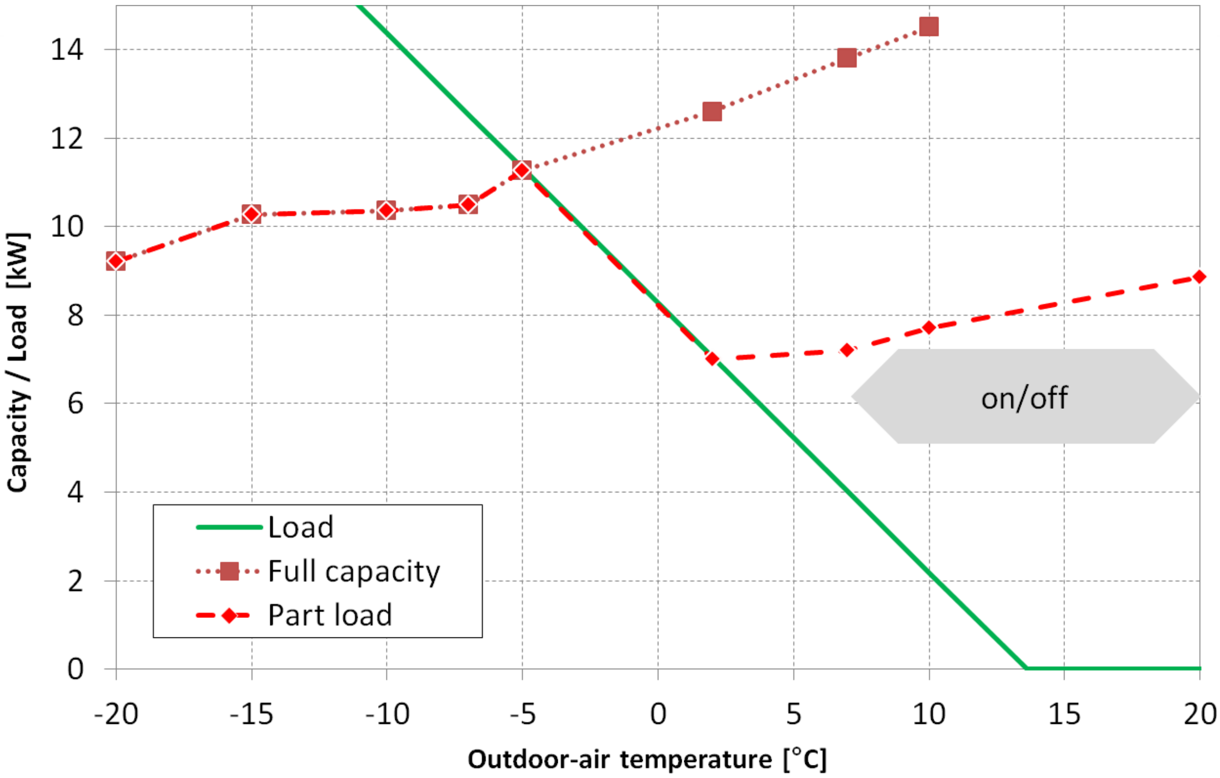


Fig. 10: ***Full and part load capacities of an exemplary inverter controlled air-to-water heat pump (W35 condition) [16]. Assumed load follows controllable capacity range.***

From this capacity data and corresponding COP values, the ratio COP_X / $COP_{X=1}$ has been plotted vs. X in analogy to simulation output in figure 9. While figure 9 has been created using a constant source temperature of A12, datasheet values as depicted in figure 10 are available for different source inlet temperatures -corresponding to a specific part load situation- only. Theoretical COP_X / $COP_{X=1}$ ratios from simulation outputs have therefore been evaluated in accordance to these datasheet conditions and are again shown in figure 11 below. The ratio rises at intermediate X values and flattens out again at low part load situations $X < 0.3$, when on/off cycling sets in. The highest COP gains (COP_X / $COP_{X=1} > 1$) are in the range of 1.22 … 1.27 at part load compared to full capacity states from rated data and 1.28 ... 1.31 from the Simulink model (when X=0 condition is omitted).

Though also describing COP change at part load, the plots from figures 9 and 11 cannot directly be compared to the f_X curve from ISO 13612-2: COP_X / $COP_{X=1}$ is the ratio of operational COPs, that is at steady states when the compressor is running. The $f_x(X)$ function as defined by ISO includes efficiency losses due to compressor off states in on/off cycling modes and thus always goes to zero when $X \rightarrow 0$ (no heat demand).

The $f_x(X)$ function as defined by ISO 13612-2 has finally also been evaluated for the Viessmann heat pump, assuming a compressor off power draw of 46 W. The resulting graph is shown below in figure 12.

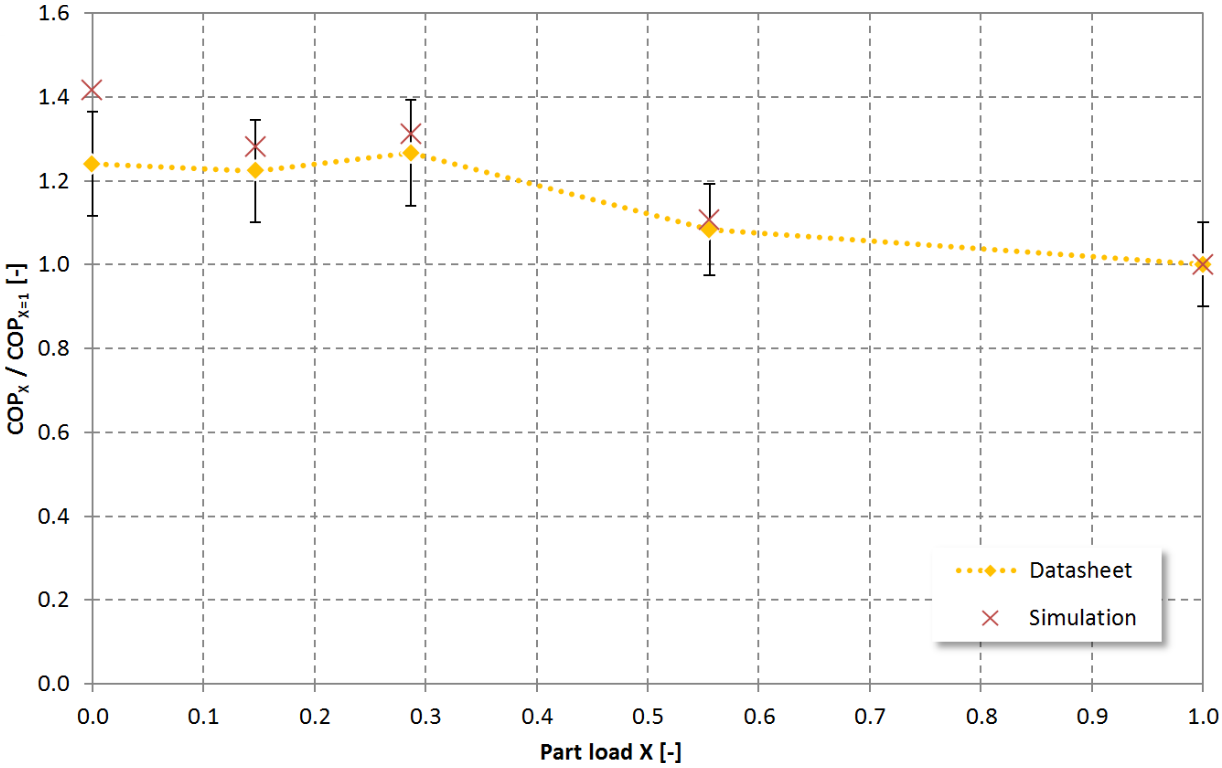


Fig. 11: ***COP_X / $COP_{X=1}$ ratio from datasheet values of an air-to-water heat pump. Comparison to simulation output of the modelled heat pump. Based on load curve from figure 10.***

While the "curves" in figures 11 and 12 are identical for $X \geq 0.56$, the f_x falloff in figure 12 becomes evident at lower part load when the minimum compressor speed is reached and the machine begins to cycle between on and off states ($X \leq 0.29$).

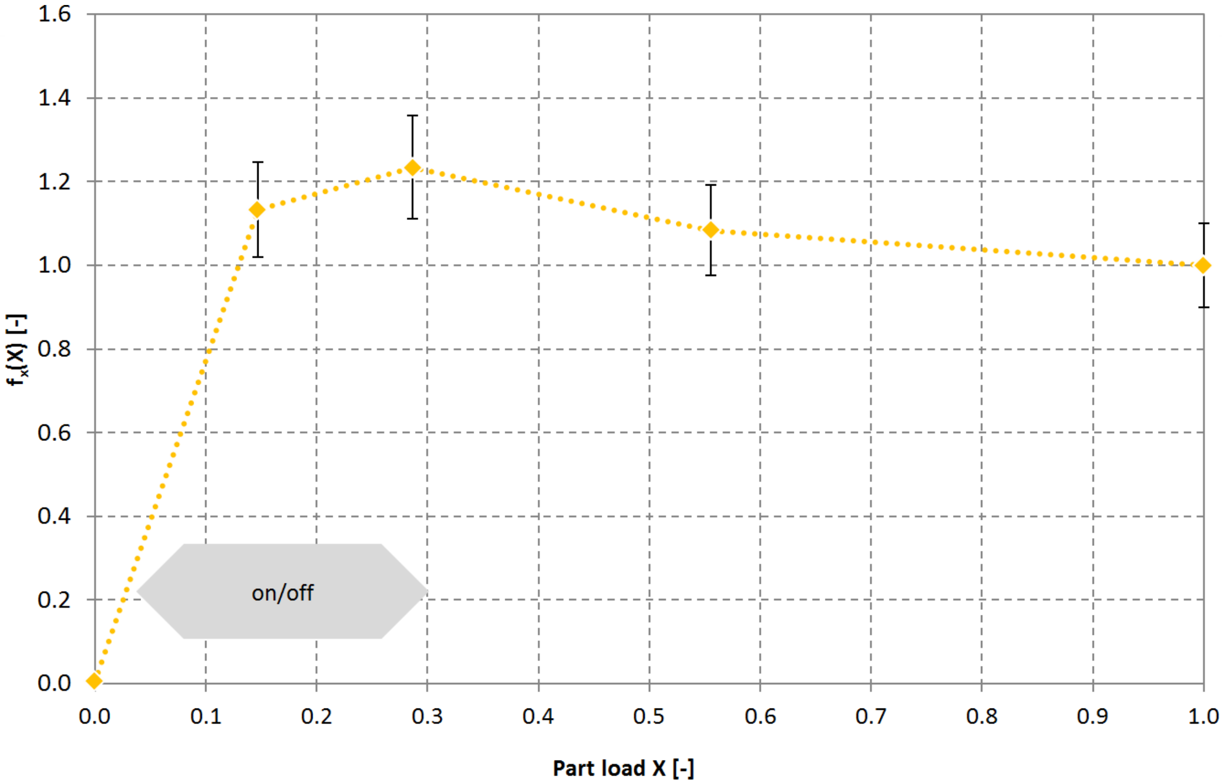


Fig. 12: ***$f_X(X)$ vs. X as defined by ISO 13612-2 for the inverter controlled Viessmann Vitocal 300-A device. Assumptions: Load curve from figure 10, compressor "off" power 46 W. Calculated from datasheet values.***

The evaluations above show, that the simulation model already shows good correlation with the behaviour of a market-available air-to-water heat pump with compressor speed control. Despite neglecting other contributions to COP change at part load, such as defrosting cycles or fan efficiency, also yearly simulations from the sections below indicate a good correlation with available datasheet values. Consequently, no further modifications were made to the variable speed heat pump model. The finally achievable COP values at various part load conditions X and three different outlet temperatures are depicted in 13. All curves were evaluated for a fixed outlet temperature of 35 ºC and a rating mass flow of 0.49 kg/s.

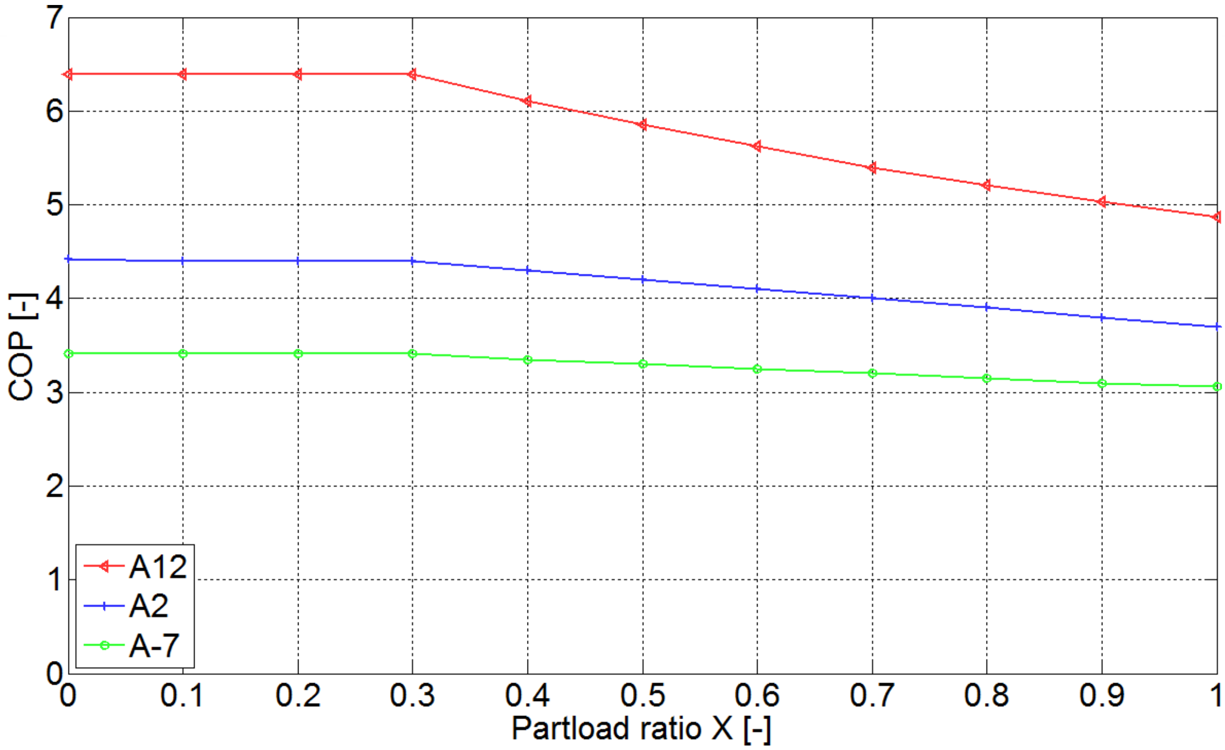


Fig. 13: COP at a constant 35 ºC outlet temperature level, fixed mass flow of 0.488 kg/s and varying air-source temperature (A12, A2 and A-7) as well as part load ratio X. Simulation output.

5 System boundaries and efficiency definitions

In the following chapters, results from efficiency simulations and comparative assessment methods for standardisation purposes will be presented. In order to unambiguously compare such rating numbers, a clear indication of the system boundaries chosen is essential, especially when comparing different calculation methods from various sources (e.g. standards). This chapter describes the boundaries and abbreviations used for the evaluation of heat pump efficiency numbers in this study. As it focuses on air-to-water heat pumps in heating mode, that is space heating and sanitary hot water preparation, no definitions for cooling modes are given. In short, the seasonal efficiency numbers used here are SPF_{HP} when COP boundaries are applied, SPF_{GEN} for the efficiency of the whole heat generator system (including circulation pumps and additional heaters such as electrical backup or solar thermal collectors) and SPF_{SYS} when the whole system is within the system boundaries, the main differentiation from SPF_{GEN} being additional heat storages. Before going into details, figure 14 shows a simplified schematic of the components of a heat pump system for space heating and/or sanitary water preparation.

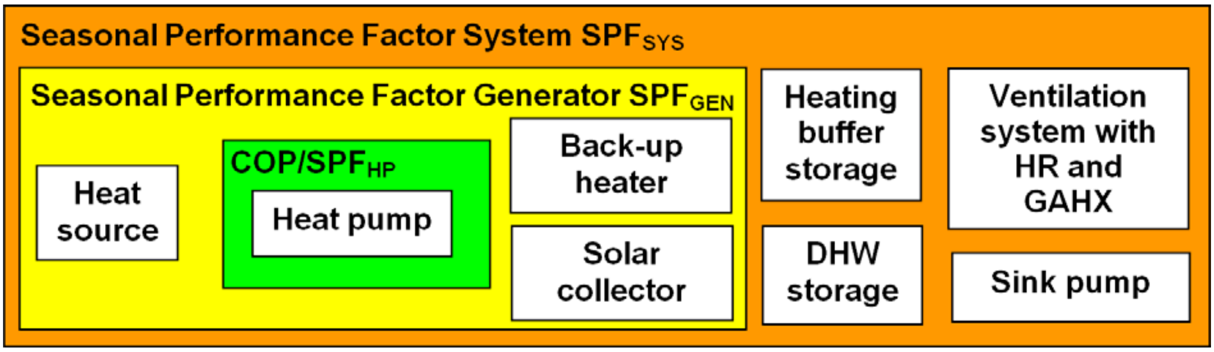


Fig. 14: System boundaries and efficiency definitions for heat pump systems.

The coefficient of performance (COP) is a momentary quantity which expresses the efficiency of a heat pump in a well-defined state of operation (steady state condition):

$$COP = \frac{\text{heat generated by the heat pump}}{\text{electrical energy consumed by the heat pump}} \tag{18}$$

The COP is defined according to EN 14511 and includes compressor, any defrost equipment required at the source side of air-to-water heat pumps as well as contributions of the control electronics, the crankcase heater and portions of the energy demand of ventilator/circulation pumps on both, source and sink side as far as required to overcome internal pressure losses.

The seasonal performance factor of the heat pump (SPF_{HP}) includes the same contributions as COP, but is evaluated from seasonal energy amounts. SPF_{HP} is defined as the overall generated heat by the heat pump divided by the amount of electrical energy required for this heat production:

$$SPF_{HP} = \frac{\text{heat generated by the heat pump}}{\text{electrical energy consumed by the heat pump}} \tag{19}$$

The generator system performance factor SPF_{GEN} includes all components required for the heat generation, but not its distribution. Besides the heat pump itself, all circulation pumps on the source side as well as any type of backup heaters (if applicable) are within the SPF_{GEN} boundaries:

$$SPF_{GEN} = \frac{\text{heat generated by all generators}}{\text{electrical energy consumed by the generator system}} \tag{20}$$

The boundaries for the definition of the system performance factor SPF_{SYS} finally includes all components of the whole heat generation, storage and distribution system. Differences between SPF_{GEN} and SPF_{SYS} are mainly storage losses from heating and/or DHW buffers, but SPF_{GEN} boundaries also includes circulation pumps on the distribution side (e.g. floor emission system).

$$SPF_{SYS} = \frac{\text{heat supplied to user}}{\text{overall consumed electricity}} \tag{21}$$

Though hinted in figure 14, mechanical ventilation systems are not in the scope of this study. Air exchange in the building used as a load for simulations and standard calculations presented here is achieved by means of a natural ventilation (properly operated windows).

6 Combined Systems

6.1 System setup

Based on the heat pump modelling described in chapter 4, whole heating system setups were chosen to inspect the behaviour of combined systems in simulations. Combined systems are here defined as systems relying on multiple heat generators. The following combinations were selected for a closer examination:

- Heat pump only (Fig. 15)
- Heat pump and solar thermal for DHW preparation (Fig. 16)
- Heat pump and solar thermal for space heating and DHW preparation (Fig. 17)

All systems have been simulated once with the on/off type heat pump and once with the variable speed heat pump type. Again, the heat source of the heat pump is outdoor air, the heating system is water based, as is typical for Switzerland. No direct electric backup heater has been applied.

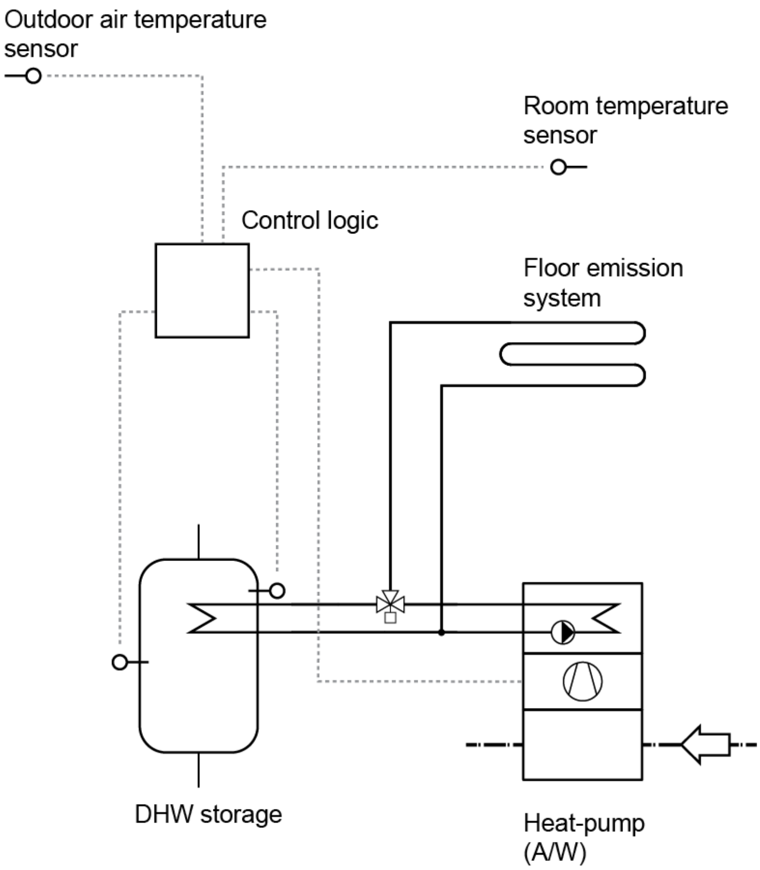


Fig. 15: Schematic of the heat pump only system (without solar thermal support)

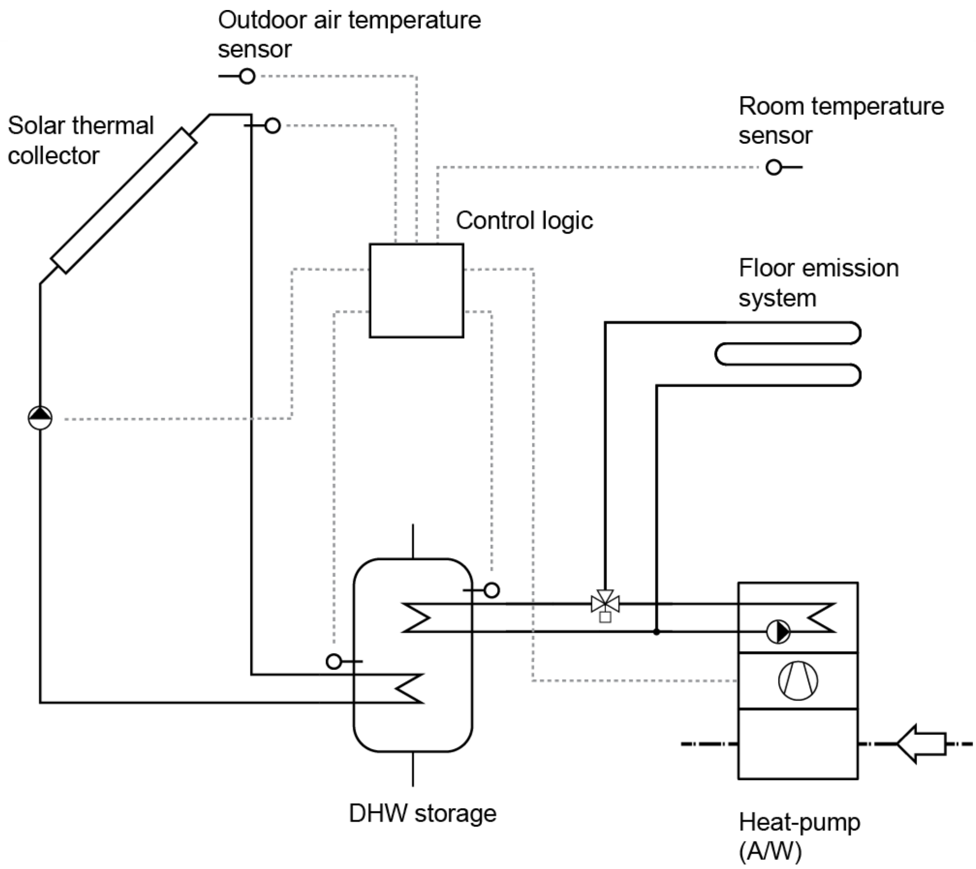


Fig. 16: Schematic of the heat pump system with solar thermal support for sanitary water preparation

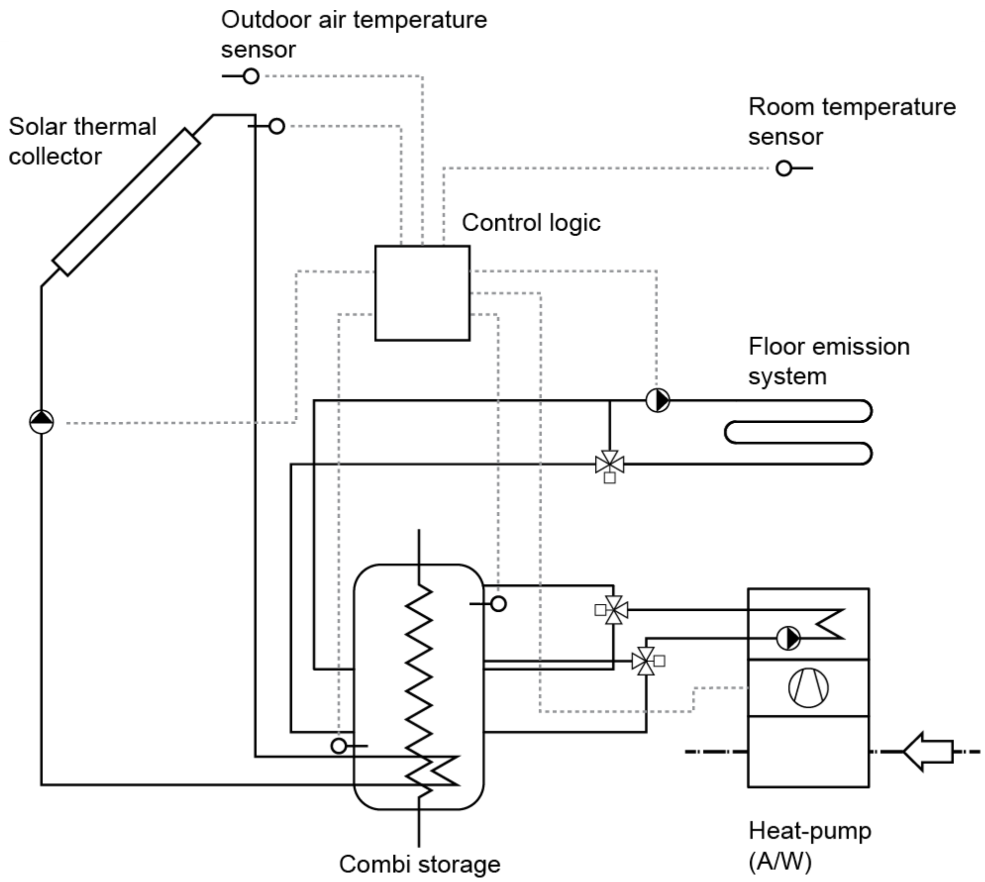


Fig. 17: ***Schematic of the heat pump system with solar thermal support for sanitary water preparation and space heating***

The building model used as load corresponds to a simplified single-node version of the single family house "SFH45" which is described in [7]. Net floor area of this moderate sized house is 140 m^2, simulated design heat load is 4.5 kW. The building insulation corresponds to a typical state-of-the art building required by Swiss regulations, rather than an extremely well insulated one. It has been located in Bern (climate of Bern-Liebefeld, design temperature -7 ºC), representing the most densely populated area of Switzerland. It therefore stands for a typical residential building found in Switzerland. Heat transfer to the rooms is accomplished by means of a low temperature underfloor heating system operated at 35 ºC flow temperature (design condition). Compressor off power consumption of the heat pump is set to 46 W, as is throughout this work. This is the only contribution to standby losses, which are also evaluated from simulations. The heat pump controller is never switched off during the year, as domestic hot water production is also in the scope of this work.

For the solar assisted systems a quality flat plate collector (η_0 = 0,745; a_1 = 3,61 W/m^{2}K; a_2 = 0,0082 W/m^{2}K^2) has been chosen. The collector field is oriented southwards at an elevation of 40 degrees. For reasons of simplicity and reliability in operation, a high flow system (60 litres/m^2/h) was implemented in the simulation models. This is regarded as the most common system in Switzerland. The system relying on solar heat for DHW preparation and space heating is set up with a combi storage tank, which includes a large heat exchanger for instant DHW heating, while systems with solar DHW support only use an ordinary heat storage with two internal heat exchangers for the heat pump (top) and the solar collector field (bottom). To proof robustness against variations in system dimensioning, the solar systems have been laid out in three different ways each, the main difference being the area of the collector field. While DHW storage tank size is left independent of collector field size, the combi storage tank volume varies with the number of collectors. The variations of all simulated systems are shown in table 7.

	Unit	Configuration						
Solar thermal support	-	-	DHW			HDHW		
Figure	-	15	16			17		
No. of collectors	-	-	1	2	3	4	6	8
Solar thermal collector field[a]	m^2	-	2.34	4.68	7.02	9.36	14.04	18.72
DHW storage	Litres	300	500	500	500	800[b]	1000[b]	1200[b]

[a]aperture area

[b]combi storage

Tab. 7: Configuration of all simulated systems.

6.1.1 Control strategy: Domestic hot water

DHW generation by the heat pump is enabled for a limited time only (4h per day), separated in two time slots of 2 h each. The heat pump switches on when the temperature in the DHW storage falls below 45 ºC. Due to a 5 K "off" hysteresis, set point temperature in the storage is 50 ºC. Regardless of heat pump type, the compressor is always operated at full speed for DHW preparation.

DHW drawings occur 3 times per day at a 2:1:2 volume ratio, that is drawings in the morning and evening hours are twice as much as at noon. Cold water supply temperature follows a sinusoidal curve throughout the year with a mean temperature of 10 ºC and an amplitude of 3 ºC. A hot water volume of 155 l/day is consumed by the user. This daily drawing amount is chosen such, that the total amount of energy matches default values given by local regulations (See section 7.1).

6.1.2 Control strategy: Space heating - fixed speed

Space heating operation is controlled via a reference room temperature sensor, represented by the single node of the building model: The heating system is activated as soon as the room temperature drops below 20 ºC and is deactivated when it reaches this value again. The thermal inertia of the underfloor heating system and building structure is high enough to prevent fast on/off switching. There are no further control instances, such as heat pump flow/return temperature sensing. Except for the system with solar assisted space heating, the heat pump is directly coupled to the floor emission system, without any additional heat storage. Where a combi storage is used -that is in systems with solar assisted space heating-, the underfloor heating is fed from this source. Space heating can then be supplied without interruption, while it is interrupted during DHW preparation in all other systems (DHW priority mode). The duration of this interruption is limited to the times when the heat pump effectively operates in DHW mode, that is typically less than an hour per day. This strategy allows tight control of room temperatures which are mostly held within 19 ºC ... 21 ºC throughout the heating season, as depicted in figure 18 for the heat pump only system.

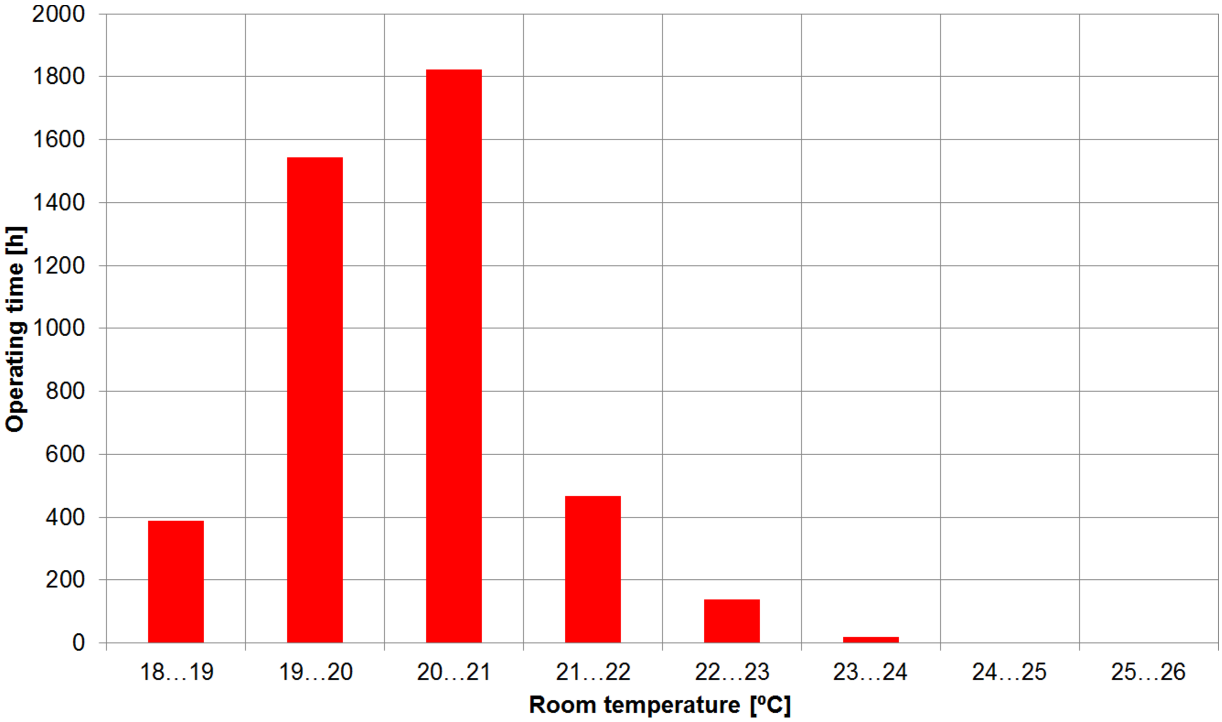


Fig. 18: Distribution of room temperatures from October - March. Simulation output, heat pump only system. Average room temperature is 20.15 ºC.

6.1.3 Control strategy: Space heating - variable compressor speed

For variable capacity heat pumps, the same control strategy as for fixed speed types is applied (reference room temperature), with the exemption that heating power now can be adjusted. The advantage of variable speed heat pumps is their ability to permanently adjust the heat output to actual demand. Ideally, flow temperatures could follow exactly a given heating curve, without overtemperature as it occurs with on/off heat pumps. The control strategy chosen here is to adapt compressor speed via a linear function of the actual outdoor temperature with breakpoints at -7 ºC (design load; 100%) and +20 ºC (no heat load; 0%). As mentioned in the modeling section (chapter 4.1.2), minimum capacity is internally set to 30 %. The definition of the heat pump input variable X as the ratio of effective heat output and maximum heat pump capacity is not exactly a linear function of the load (represented by the outdoor air temperature). It does therefore not match the definition of the heating controller output (fraction of full load) as described before. The compressor speed would therefore always a bit higher than required (see figure 19).

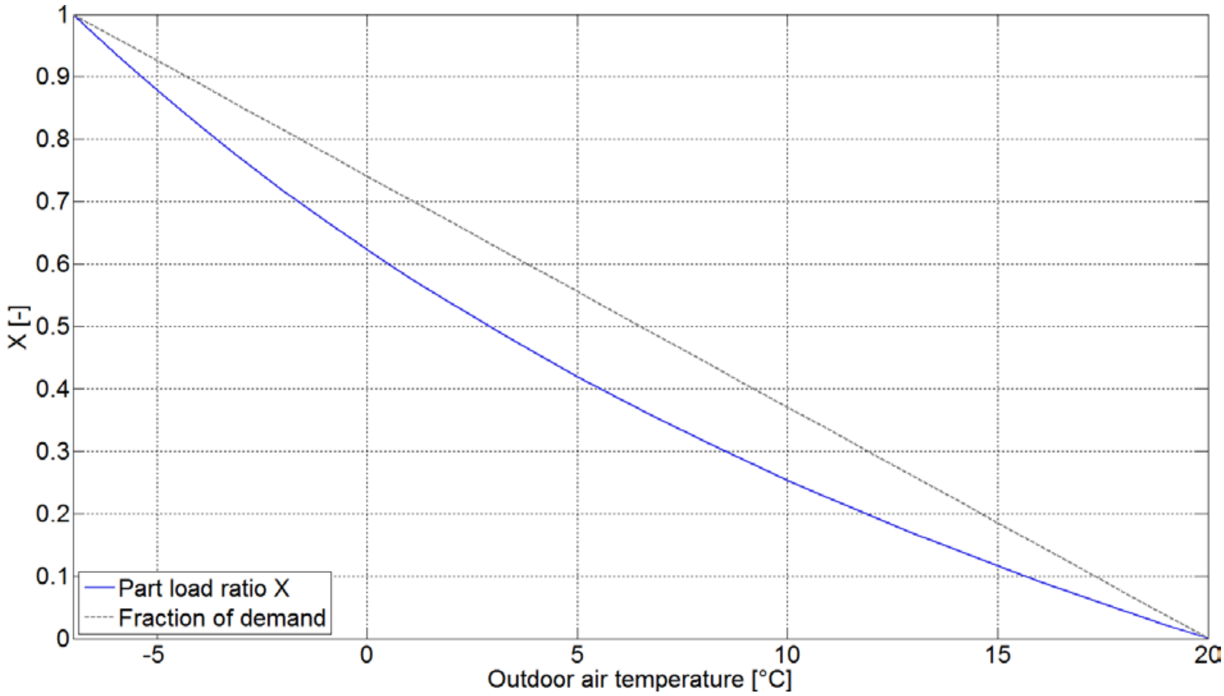


Fig. 19: ***Fraction of heating demand as an output of the heating controller and part load ratio as an input of the heat pump. The simple "linear" heating controller never exactly matches the theoretically required part load ratio X, part load capacity is always higher than what would be required.***

Simulations with a perfect matching of the heating controller output and the definition of X (heat pump input), that is a dependency of the controller output from outdoor air temperature which exactly follows the part load ratio curve of the heat pump show an improvement in the performance factor of 0.2 points in space heating mode $SPF_{HP,H}$ (Heat pump only system). Sanitary water preparation is not affected, but the improved heating performance gives also a higher "overall" $SPF_{HP,HDHW}$ -see table 8. The table also shows that the performance disparity diminishes when standby operation is included in the evaluation ($SPF_{HP,HDHW,sb}$)

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$	$SPF_{HP,HDHW,sb}$
Linear function	4.6	3.2	4.1	3.7
Perfectly matched	4.8	3.2	4.3	3.8

Tab. 8: ***Matching between heating curve (heating controller output) and heat pump input (part load ratio X): Heat pump performance can be increased by 2 % if the controller is perfectly matched to the heat pump. Heat pump only system.***

Despite the improvement in annual performance that can be achieved by fitting heating controller output to the heat pump behaviour, the simple linear controller output function has been deliberately chosen for further simulations with variable compressor speed heat pumps. This decision has been made as an average rather than a best case situation shall be examined.

With this control strategy, the indoor room temperatures are held mostly within a range of 19 ºC ... 21 ºC (over 80% of the time) of the winter season, although with a little shift of -

0.3 ºC of the mean temperature, which is 19.9 ºC for the system with the variable speed heat pump instead of 20.2 ºC for the of/off controlled heat pump type (Heat pump only system). These temperatures almost perfectly match those used in standard calculations for regulatory purposes, which ensures a fair comparison to be carried out in chapter 7.

6.2 Energy flows

From simulation outcomes, detailed data energy and performance of the building and the technical equipment is available. To give an impression of the relations of energy flows, this data has been processed for the system with two solar thermal collectors, used for sanitary hot water preparation alone. Figure 20 shows the energy flows of this heat supply system in a Sankey diagram.

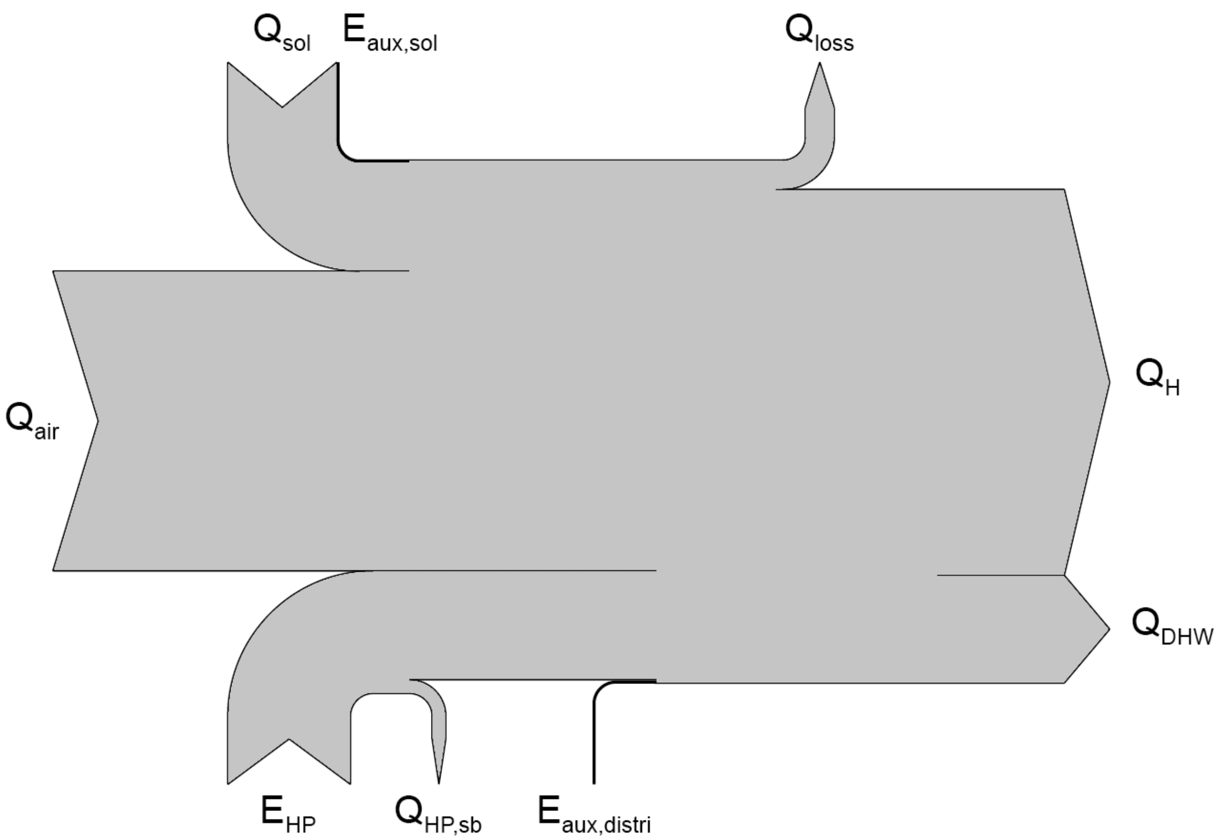


Fig. 20: Sankey diagram of heat generator subsystem. Annual energy flows. Typical system with solar thermal support for sanitary hot water (2 collectors). Simulation outcome.

More than three quarter of the yearly supplied heat is needed to cover demand for space heating (Q_H) -see table 9. The performance of the combined space heating and domestic hot water operation is hence dominated by the space heating efficiency.

Quantity	Symbol	Unit	
Hot water drawing	V_{DHW}	l/day	155
Sanitary hot water	Q_{DHW}	kWh	2266
Space heating	Q_{H}	kWh	8549
Overall	Q_{HDHW}	kWh	10815

Tab. 9: Hot water drawing (daily) and annual amount of heat flow to the end user.

Nevertheless, the SPF of the whole generator system -including heat pump and solar system- is improved by 0.9 when compared to the heat pump system alone (Tab. 10). The evaluation of the sanitary hot water production gives $SPF_{GEN,DHW}$ = 12.0 (without standby, as allocation of standby energy to different operationg modes is not obvious), it's a vast improvement over the the heat pump only system ($SPF_{GEN,DHW}$ = $SPF_{HP,DHW}$ = 3.2).

System	Heat pump only	Heat pump & solar thermal for DHW
$SPF_{GEN,HDHW}$	3.7	4.7

Tab. 10: Efficiency numbers of heat pump only and combined system.

Scrutinizing the solar subsystem, the two collectors with an overall aperture area of 4.68 m^2 deliver a yearly energy amount of 2369 kWh (Tab. 11) to the sanitary hot water storage. An aperture area specific yield of around 500 kWh/m^2 is reached -a typical value for such a subsystem in Switzerland's climate. The circulation pump consumes 40 kWh of electricity for this solar yield.

Quantity	Symbol	Unit	
Yield	Q_{sol}	kWh/a	2369
Specific yield	$Q_{sol,spec}$	kWh/m^2/a	506
Energy consumption circulation pump	$E_{aux,sol}$	kWh/a	40

Tab. 11: Yield and expenses of solar subsystem.

The heat pump's heat output is 9061 kWh while it consumes 2758 kWh of electricity, whereof 355 kWh are "leaked" in standby mode (Tab. 12) . This amount of energy is converted into heat which is considered a loss ($Q_{HP,sb}$). Further heat losses are caused by the hot water storage, which is the DHW tank. The electricity consumption of the distribution pumps is 24 kWh, slightly below 1 % of heat pump consumption. Its contribution to the heat delivery of the generator is neglected in the analyses.

Quantity	Symbol	Unit	
Electricity consumption (without standby)	E_{HP}	kWh	2403
Standby energy consumption	$E_{HP,sb}$	kWh	355
Heat for space heating	Q_H	kWh	8551
Heat for sanitary hot water	Q_{DHW}	kWh	510
Energy consumption distribution pumps	$E_{aux,distri}$	kWh	24
Storage losses (DHW)	$Q_{DHW,loss}$	% of input	22
Storage losses (H)	$Q_{H,loss}$	% of input	n.a.

Tab. 12: Amount of energy flow of the heat pump and storage subsystems

6.3 Performance results from simulations

Results of yearly heat pump efficiency for all system simulations with the on/off type heat pump are summarized in table 13 below. As performance number, SPF_{HP} with COP boundary conditions has been evaluated, as this is this the quantity usually required for energy evaluations. Naturally, performance of the whole generator system, including the solar thermal collector field is considerably higher, as the solar heat is gained at the expense of an additional circulation pump only. An example of the order of magnitude is given in chapter 6.2. Indices "H", "DHW" and "HDHW" are used to designate the modes "space heating" (H), "domestic hot water preparation" (DHW) and both combined (HDHW). As for the systems with solar thermal support, the typical dimensioning of the solar collector field size has been chosen for the setup of table 13, that is systems with two collectors for solar assisted DHW preparation and six for DHW and space heating support. $SPF_{HP,H}$ $SPF_{HP,DHW}$ and $SPF_{HP,HDHW}$ do not include standby mode (compressor off times). An obvious assignment of standby energy to the operating modes "heating" or "DHW preparation" does not exist. Therefore, overall standby consumption is included in the combined SPF_{HP} designated "$SPF_{HP,HDHW,sb}$" only.

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$	$SPF_{HP,HDHW,sb}$
Including standby	no	no	no	yes
no ST	3.9	3.2	3.7	3.3
ST for DHW	3.9	2.6	3.8	3.3
ST for DHW & SH	3.7	2.7	3.5	3.0

Tab. 13: Simulation results for on/off type air-to-water heat pump system combinations

As expected, $SPF_{HP,DHW}$ is much lower than the performance for space heating operation ($SPF_{HP,H}$) due to the higher temperature lift during hot water preparation (Maximum outlet temperature of 57 ºC). As the numbers for the combined efficiency ($SPF_{HP,HDHW}$ and

$SPF_{HP,HDHW,sb}$) show, the inclusion of standby operation decreases the overall performance by more than 10 %. This is partly to the assumption of a somewhat high standby power consumption (46 W) but also a fact of high standby times. An analysis of standby mode is given in chapter 7.5.

The same evaluations of the heat pump efficiency with a speed controlled compressor are shown in Table 14. Abbreviations and subscripts are identically chosen as described for on/off heat pumps above.

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$	$SPF_{HP,HDHW,sb}$
Including standby	no	no	no	yes
no ST	4.6	3.2	4.1	3.7
ST for DHW	4.4	2.6	4.3	3.7
ST for DHW & SH	4.1	2.7	3.8	3.1

Tab. 14: Simulation results for compressor speed controlled air-to-water heat pump system combinations

Comparing efficiency values of tables 13 and 14, the efficiency numbers for speed controlled heat pumps are at best 18 % higher compared to conventional on/off types (heating mode) and identical for DHW preparation. Both results were expected, as outlet temperatures in space heating mode are lower for speed controlled heat pumps with a variable outlet while DHW mode operation is identically implemented for both heat pump types. While average outlet temperature of the inverter controlled heat pump in heating mode is 31.0 ºC, it is 35.2 ºC for the on/off type (system without solar thermal support). Figure 21 depicts the distribution of the flow/outlet temperatures. This lower temperature alone would roughly explain 13 % of efficiency increase by a common rule of thumb (3 % efficiency variation per degree centigrade of sink temperature change), which is a little bit less of improvement than what the output of the simulation ($SPF_{HP,H}$ for system with "no ST") shows.

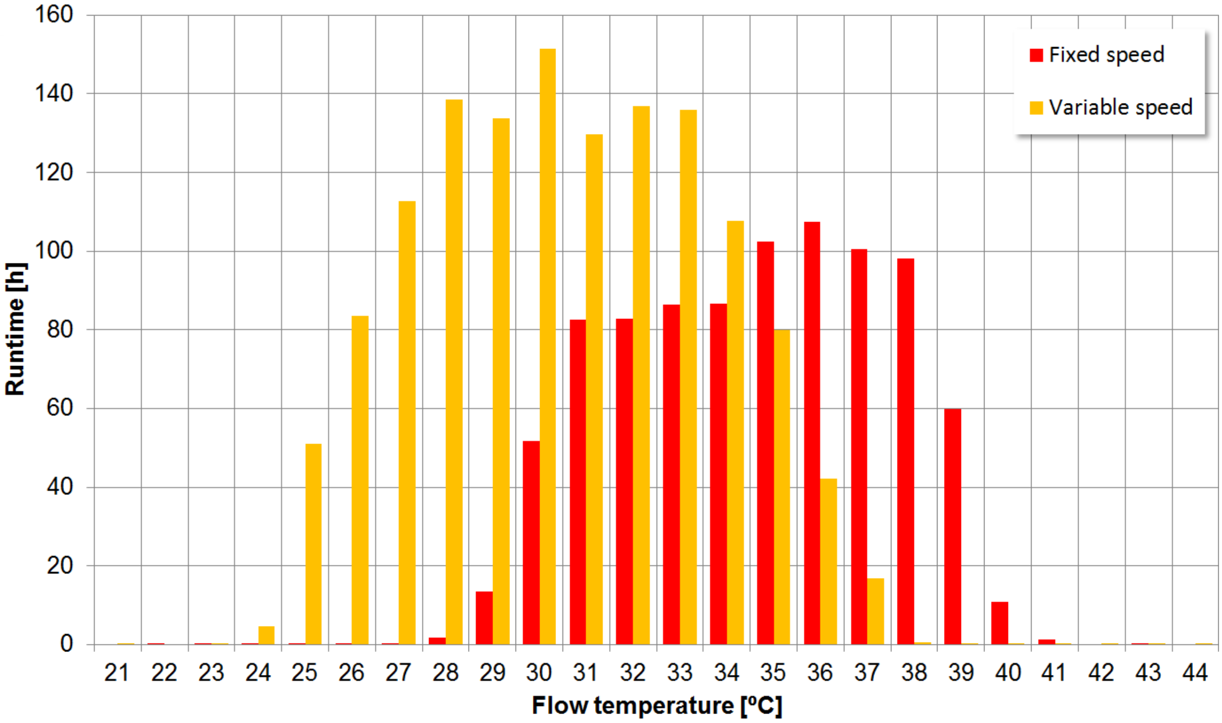


Fig. 21: ***Distribution of heat pump flow / outlet temperatures in space heating mode. Variable speed vs. fixed speed type. Simulation output.***

As can be seen from the higher absolute frequencies in figure 21, run times of the variable speed heat pump are increased by 50 % (888 h/year vs. 1328 h/year) in space heating mode, as most often the capacity in operation is lower for speed controlled heat pump units.

While the results so far have been presented for the typical dimensioning of the solar collector fields (where applicable), all variants from table 7 have been simulated. For on/off controlled heat pumps, the outcome of the additional setups with an extreme dimensioning of the solar collector field size match the efficiency numbers of the systems regarded as "typical" from table 13. For variable speed heat pumps however, there is a minor variation in $SPF_{HP,HDHW}$ and $SPF_{HP,HDHW}$ between the different setups as shown in table 15. It is concluded, that SPF outcomes are robust against variations in the dimensioning, which is a desirable feature for standard calculations (see section 7).

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$	$SPF_{HP,HDHW,sb}$
Including standby	no	no	no	yes
no ST	4.6	3.2	4.1	3.7
ST for DHW	4.4	2.5 … 2.6	4.2 ... 4.3	3.6 ... 3.7
ST for DHW & SH	4.1	2.7	3.7 … 3.8	3.1 ... 3.2

Tab. 15: ***Simulation results for variable compressor speed type air-to-water heat pump system combinations. Seasonal performance of heat pump in systems with other than typical collector count (range).***

6.4 Estimation of uncertainties

Simulation outcomes for space heating efficiency from chapter 6.2 show little variance only, $SPF_{HP,H}$ differences are always below 11 % or 0.5 SPF_{HP} points between systems with the same heat pump type (on/off resp. variable speed). Performance of the variable speed type heat pump vs. on/off controlled is higher by 18 % or 0.7 SPF_{HP} points at most. These differences are not much higher than tolerances in components and uncertainties in heat pump rating measurements. From EN 12900:2012, compressor COP_r (defined as ratio of refrigerating capacity to the power absorbed) may be 10 % lower than the declared value. From [11], the seasonal space heating energy efficiency η_s as defined by EUs Ecodesign requirements may not be lower by more than 8 % of the declared value, which translates into a SCOP uncertainty of 0.3 for an air-to-water heat pump with an SCOP of 4.0. While SCOP expresses a seasonal performance which also includes measurements in compressor off states, EN 14511 defines the maximum measurement uncertainty of single rating points, which are evaluated at steady state conditions. The uncertainty of EN 14511 COP measurements may not surpass 6 %, which is ±0.2 for a COP of 4.0. While tolerances of components and measurement uncertainties at the laboratory level are well defined, the load side can have a much higher impact on the seasonal efficiency in real applications. The question arises, how much the influence of the user side is, that is how precise simulation outputs can represent the reality. As an evidence of this, the influence of uncertainties of the input parameters on the simulation results has been examined in a Monte Carlo analysis: Usually, all simulation input parameters such as setpoint temperatures to control the heat pump are assumed to be precise, while in reality these values are always afflicted by a certain amount of uncertainty -in the given example the measurement uncertainty of the temperature sensors. This may lead to somewhat higher or lower than the setpoint temperature levels, which finally ends up in lower or higher seasonal efficiency values. When going from simulations to real buildings, such uncertainties can never be overcome. Therefore, even in a perfectly built house there will most probably always be some deviation between simulated and measured efficiency values. The analysis given here shows how precise a simulation can be when accounting these uncertainties of input parameters, ignoring all other real world influences such as actual build quality, actual weather conditions or user behaviour. The given numbers therefore show an upper limit of the precision that can be achieved. Fig. 22 shows the $SPF_{HP,HDHW}$ distribution from 32 simulation runs of a heat pump system with solar thermal support for DHW preparation. The analysis shows, that the mean of the seasonal performance is at 3.29, but it may lie anywhere between 3.11 and 3.47, which is a ±6 % uncertainty (P = 99%).

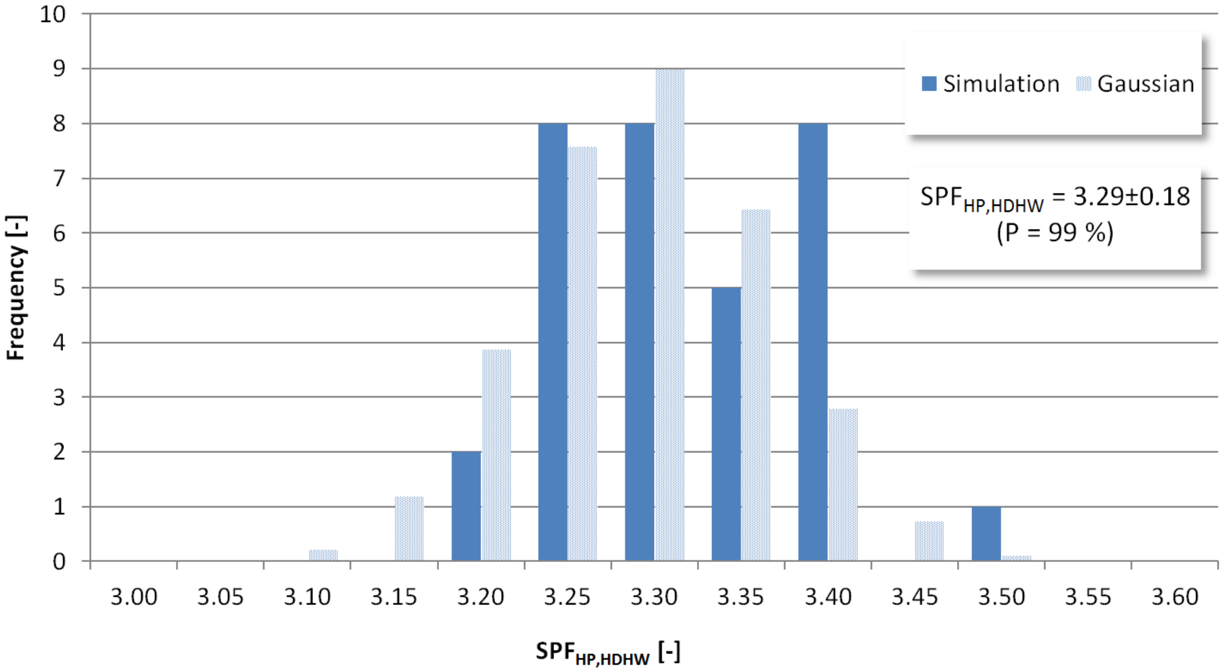


Fig. 22: ***Distribution of SF$_{HP,HDHW}$ in a Monte-Carlo analysis. The figure shows simulation output as well as how a corresponding ideal Gaussian distribution would look. Number of simulation runs: 32.***

The results cannot be directly compared to those given in the other sections of this study, due to a different system setup and simplified heat pump modelling to speed up simulation times. For the analysis, system parameters such as heat-transmission coefficient of walls (U-values), energy gain through windows (g-value), mass flow rate of pumps, temperature setpoints and so on have been randomly varied by an amount of ±5% at most, each in such a way that the probability distribution of the input parameter values follow a normal distribution (Fig. 23 / Tab. 16). Having a special look at the heat pump, compressor tolerances have not been included in this analysis.

There is a risk that the user would intervene when a certain combination of input parameters accidentally would influence comfort by a greater extent, say by too low room temperatures. This may limit the bandwidth of SPF variance. However, room temperatures varied by 0.15 °C only throughout all these simulations, the user would barely have noticed any change.

As a conclusion of this outcome, the simulation results presented here should be understood as a typical value with an uncertainty of about ±0.2 SPF points.

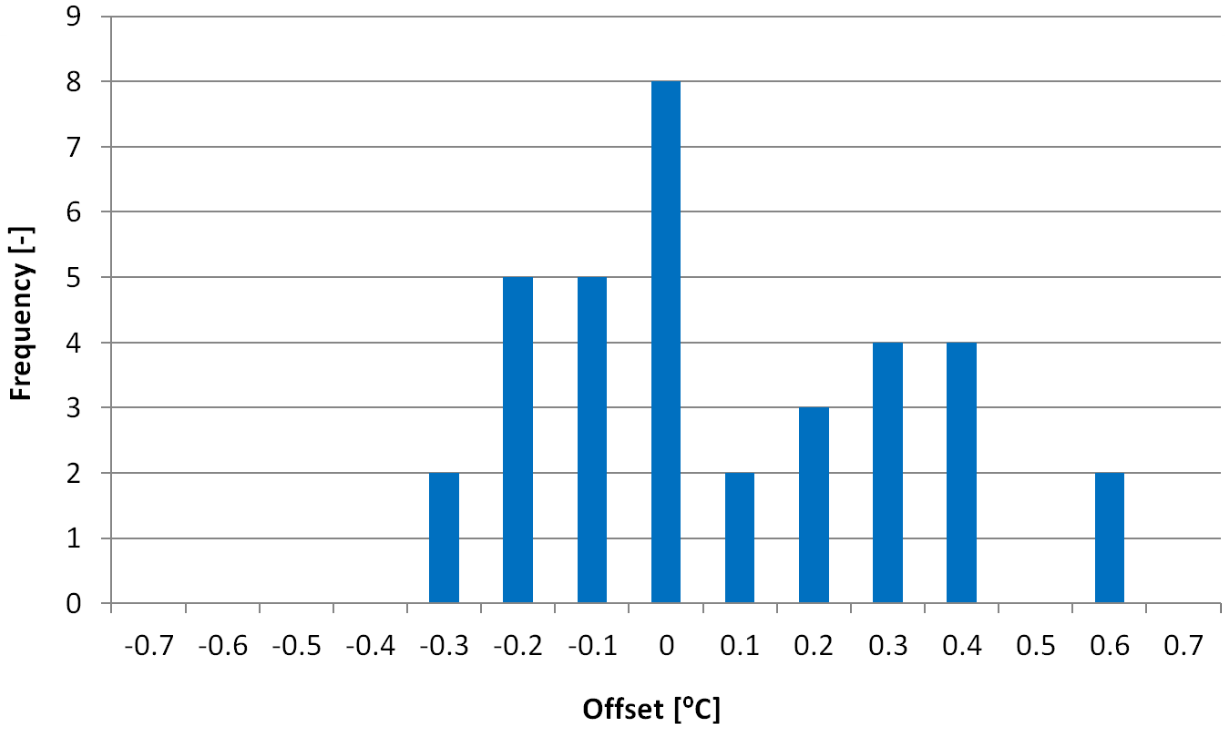


Fig. 23: Example of input parameter variation: Distribution of DHW storage temperature sensor offset. Depending on this offset, DHW storage temperatures are slightly changed in every simulation run.

Parameter	Unit	μ	σ
Storage Insulation	W/m^2	0.60	0.015
Daily DHW consumption	l	130	3.25
Building insulation (walls)	W/m^2	0.197	0.005
Building insulation (roof)	W/m^2	0.286	0.007
Building insulation (windows)	W/m^2	1.50	0.04
Windows g-value	W/m^2	0.622	0.016
Ventilation rate	h^{-1}	0.4	0.01
Mass flow solar thermal	$l/h/m^2$	50	1.25
Internal gains	W/m^2	2.56	0.01
Mass flow heating system	l/h	830	21
Heat pump capacity	kW	-	$0.06\dot{Q}$
Temperature sensors (storage, underfloor heating)	ºC	-	0.3

Tab. 16: Simulation input for the Monte Carlo analysis: Varied parameters, expectation values and standard deviation.

7 Standard calculations

Simulation results from chapter 6 have been compared to standard calculations which usually are required by regulations to show how well these simplified assessment tools reflect the behaviour of combined systems which are based on heat pumps as a primary heat source. Therefore, all calculations have been carried out for the systems with and without solar support as described in chapter 6.1, once with on/off controlled heat pump types and once with variable compressor speed heat pump types. Required input values from the heat pump -such as rating data- has been generated by simulations in a virtual test rig with the heat pumps modelled for Matlab/Simulink (Chapter 4). This guarantees the best possible consistency between simulations and standard calculations. As variable capacity heat pumps are not already fully implemented in all standards, a method on how to handle these types is also given.

7.1 Boundary conditions

Standard calculations have been performed for all three systems described in chapter 6.1. The selection of applied standards is based on the overview from chapter 3. Again, as far as within scope of each standard, the evaluations include space heating as well as sanitary hot water preparation. The following standards/calculation tools were used

- SIA 384/3
- WPesti
- EN 14825

The Swiss standard for energy calculations of heat pumps is named SIA 384/3. The WPesti tool is available as a MS Excel spreadsheet and has been added to the comparison as it is the tool accepted by Swiss authorities for regulatory purposes. It's available in version 8 now, The tool is therefore well established amongst building equipment designers and installers in Switzerland. Efficiency calculation from WPesti is based on a BIN method, the exact calculation procedure is decribed in [17]. Since SIA 384/3 was published, the WPesti calculation tool implements this standard. If not available, WPesti as well as SIA 384/3 estimate design heat load based on yearly heat demand calculations for the building under investigation (Swiss standard SIA 380/1).

EN 14825 deserves a closer look. It has become a major role in European efficiency requirements for heat pumps, it is the basis of EU's energy labelling and Ecodesign directives No. 811/2013 and No. 813/2013. These efficiency calculations have therefore also been added to the comparison, though applicable for space heating only. While SIA 384/3 and WPesti tools rely on given data for a specific object in a climate that can be chosen by the user, EN 14825 is mainly made for the assessment of heat pump equipment itself and therefore is based on some standard assumptions which represent typical operating situations as they occur throughout Europe. The standard defines four fixed heating curves in each of three climate conditions ("colder", "average", "warmer") to choose from. The calculations that have been carried out here are based on the "low temperature application" in the "average" climate of Strasbourg (France). Low temperature application is defined by a heat pump outlet temperature of 35 ºC at design heat load. As a "variable" output heat pump type has been chosen for the typical application, the predefined heating curve according to table 17 (top row) was applied. The given temperatures are understood as the temperature to be fed to the heating system, they correspond to the outlet temperatures of an ideal heat pump. For variable speed heat pumps, this is the outlet temperature to be hold for rating at each outdoor air temperature condition. For on/off controlled heat pumps, outlet temperatures must be higher, to compensate off times when on/off cycling. These overtemperatures of the outlet are handled by a calculation method which is outlined in annex D of EN 14825:2013, some more details on this issue are given in chapter 7.2. The procedure of evaluating rating outlet temperatures has to be carried out for fixed compressor speed heat pumps as well as for variable compressor speed types for rating points where minimum capacity is higher than demand (this will typically affect rating points A7 and A12).

Table 18 below shows the relevant boundary conditions for the standard calculations. They apply as well to system simulations carried out in chapter 6. Except for EN 14825, all calculations were performed at the local climate of Bern-Liebefeld, as was selected for simulations in chapter 6. For EN 14825 performance calculations, the climate "average" corresponds best to the situation in Bern-Liebefeld. As required for SIA 384/3 and WPesti calculations, space heating energy demand Q_H, internal and external heat gains and transmission losses through the building envelope were also calculated. The corresponding calculation is based on Swiss standard SIA 380/1, it has been carried out for the "SFH45" building that was used in simulations as well. That's why there is a colum in table 18 that is named accordingly. This data is solely required for the estimation of design heat load of the respective tools. But the procedure of SIA 380/1 carried out for a constant external temperature of -7 ºC and neglecting all gains also allows an estimation of design heat load. The value given here is evaluated for comparative reasons only, as it is a quick alternative to the complete design heat load evaluation from SIA 384.201 standard.

As for design heat load value from EN 14825 in table 18, this value has no correlation to the building. The value given here is the so called "rated heat output" of the heat pump, in the EN 14825 framework it is calculated from rating data of the heat pump, the chosen bivalence point and the fixed design condition (outdoor air temperature of -10 ºC in the average climate of Strasbourg) and includes an electric backup heater (if applicable). As required by Swiss regulations, bivalence point has been set to design temperature (monovalent operation). The rated heat output as given here is therefore the capacity of the heat pump (without any backup) at the A-10/W35 rating point. While these values do no perfectly match those derived from the building and local climate, they are intentionally not altered. Though it would be possible to do so, heat pump rating within the framework of the European Ecodesign and energy labelling directives is based on such given, fixed values. For end users, planners and installers, this is this data that will be available. The comparison conducted here therefore shows how well generic Ecodesign performance data will fit the actual situation of a typical residential building.

	A-10	A-7	A2	A7	A12
Variable outlet temperature	35 ºC	34 ºC	30 ºC	27 ºC	24 ºC
Fixed outlet temperature	35 ºC	35 ºC	35 ºC	35 ºC	35 ºC

Tab. 17: Heating curves according to EN 14825 (low temperature application, "average" climate). Temperatures correspond to ideal outlet of variable compressor speed heat pump.

	SIA 380/1	SIA 384/3	WPesti	384.201	EN 14825	Simulation
Climate	Bern-Liebefeld	Bern-Liebefeld	Bern-Liebefeld	Bern-Liebefeld	"average" (Strass-bourg)	Bern-Liebefeld
Heating energy demand	195 MJ/m^2	n.a.	n.a.	n.a.	n.a.	n.a.
Used heat gains (internal & external)	192 MJ/m^2	n.a.	n.a.	n.a.	n.a.	n.a.
Transmission losses	279 MJ/m^2	n.a.	n.a.	n.a.	n.a.	n.a.
Air change rate	0.4 h^{-1}	(0.4 h^{-1})	(0.4 h^{-1})	0.4 h^{-1}	n.a.	0.4 h^{-1}
Design temperature	-7 ºC	-7 ºC	-7 ºC	-7 ºC	-10 ºC	-7 ºC
Load @ A-7	4.9 kW	5.1 kW	5.1 kW	4.6 kW	6.8 kW[a]	4.5 kW
Compressor "off" power	n.a.	46 W	n.a.	n.a.	46 W	46 W
Setpoint indoor temperature	20 ºC	(20 ºC)	(20 ºC)	20 ºC	20 ºC	20 ºC[c]
Temperature control	Reference room	(Reference room[b])	(Reference room [b])	n.a.	n.a.	Reference room
Cold water temperature	n.a.	10 ºC	-	n.a.	n.a.	10±3 ºC sinusoidal
DHW setpoint (at tap)	n.a.	45 ºC	45 ºC	n.a.	n.a.	45 ºC
DHW demand (user side)	2333 kWh	2333 kWh	2333 kWh	n.a.	n.a.	2333 kWh

[a]Rated heat output, evaluated for A-10

[b]as based on SIA380/1

[c]effectively leading to 19 … 21 ºC indoor temperature during winter

Tab. 18: ***Boundary conditions for simulations and standard calculations of seasonal performance (SPF) of the heat pump***

There are some noteworthy observations: Heat load according to the detailed 384.201 evaluation is 4.6 kW, the corresponding value obtained from a simulation is 4.5 kW - essentially the same value. As with simulations, room temperature is controlled via a single sensor, positioned in a reference room. WPesti calculation tool as well as SIA 384/3 derive heat load from yearly heating demand calculations according to SIA 380/1, where a correction of +1 ºC for the room temperature is applied. This overtemperature accounts for the less than perfect control strategy (reference room temperature), as some rooms may overheat. Therefore, these heat load values derived from SIA 380/1 calculations alone are somewhat higher.

For all calculations, air change rate has been set to 0.4 h^{-1}. If applicable, standby power consumption of the heat pump is set to 46 W, independently of its type, the same value that has been chosen for system simulations in chapter 6. Setpoint of the room temperature is 20 ºC, as this value is frequently used for regulatory purposes. It is also implicitly anchored in the heating curves from EN 14825.

To fulfill local regulation requirements, the heat pump is operated without any backup heater, such as direct-electric backup (Bivalence point = Design temperature).

7.2 Bin method

All standard calculations scrutinized rely on a bin method, where climate data is divided up into outdoor air temperature classes (bins). All methods use temperature classes of 1 ºC width. From climate data, the duration of each temperature bin throughout the year is known. One of the differences between standards is how efficiency values of the heat pump are estimated for each temperature class. On/off operated air-to-water heat pumps usually suffer from overtemperature at the outlet when capacity increases due to higher source temperatures (typically mild days in spring and fall). This effect reduces efficiency and is handled in different ways. The approach of EN 14825 uses a method (described in appendix D of EN 14825) which assumes a time averaged mean temperature of the heat pump outlet that ideally suits the heating curve. While the compressor is "on", the outlet temperature must be higher to compensate for "off" times. As can be seen from tables 5 and 17, the outlet overtemperature in "on" mode at A12 condition is a rather moderate 4.8 ºC, evaluated for low temperature application and the heat pump model used for the measurement. This is a consequence of the method which is based on heat pump ratings carried out with a fixed water flow rate determined from the A7/W35 (ΔT = 5 ºC at the sink side) condition. The typically high capacity at A7 condition -when compared to design temperature of the building- requires rather high flow rates and therefore gives modest overtemperatures only, independent of source condition. One of the main differences in the bin setup between European EN 14825 and Swiss SIA 384/3 standards is that SIA 384/3 also includes the design mass flow of the heating distribution system rather than relying solely on mass flow rates from heat pump testing. For the heat pump regarded here, a mass flow of 0.49 kg/s is obtained from EN 14825. The setup of the building and heating system as described in chapter 6.1 "System setup" includes an underfloor heating system for the distribution of the generated heat to the rooms. The design temperatures of this heating system are 35 ºC / 30 ºC (flow/return) at a design outdoor air temperature of -7 ºC. A quick calculation gives a mass flow of 0.23 kg/s or less than half of the mass flow at rating condition from EN 14825.

In the SIA 384/3 procedure, the complete evaluation of heat pump operating parameters COP and capacity for each bin is performed as follows: At first, the return temperatures $T_{sink,in}$ of the heating distribution system are evaluated for each bin. The assumption made here is a linear function connecting outdoor air and return temperatures from the heating system. For the building under examination here, it was defined by the points A-7/W30 and A20/W20, where "A" stands for outdoor air temperature and "W" for the associated return temperature (see figure 24). The outlet temperature and capacity of the heat pump are depending variables:

$$\dot{Q} = \dot{Q}(T_{sink,out}) \tag{22}$$

Both are evaluated from $T_{sink,in}$ in an iterative process, the function (22) can be established from rating data (e.g. linear interpolation). As a starting point for the iteration process in each bin, heat pump data from the nearest temperature bin with known rating values is taken and adjusted from mass flow at rating conditions to the mass flow of the distribution system. SIA 384/3 then estimates outlet temperature and capacity from the iteration

$$T_{sink,out,n+1} = T_{in} + \frac{\dot{Q}_n(T_{sink,out,n})}{\dot{m} \cdot c} \tag{23}$$

$\dot{Q}_n(T_{sink,out,n})$	capacity at the outlet temperature $T_{sink,out}$ of iteration n
$\dot{m}$	mass flow rate of the heating system

In every iteration loop (n+1), "new" values for $T_{sink,out}$ are derived from the heating capacity obtained in the previous iteration step (n). Once temperature levels are known, the COP is evaluated in each iteration step. It is assumed that the exergetic efficiency in each bin is equal to the value evaluated at the nearest rating point (exergetic efficiency is considered as locally constant value):

$$COP_n = COP_{rated} \cdot k_n \tag{24}$$

$$k_n = \left(\frac{T_{sink,out,bin} - \frac{\Delta T_{sink,bin}}{2} + \Delta T_{cond}}{T_{sink,out,rated} - \frac{\Delta T_{sink,rated}}{2} + \Delta T_{cond}} \cdot \frac{T_{sink,out,rated} - \frac{\Delta T_{sink,rated}}{2} + \Delta T_{Cond} - T_{source,in,rated} + \frac{\Delta T_{source,rated}}{2} + \Delta T_{evap}}{T_{sink,out,bin} - \frac{\Delta T_{sink,bin}}{2} + \Delta T_{Cond} - T_{source,in,bin} + \frac{\Delta T_{source,bin}}{2} + \Delta T_{evap}} \right)_n$$

$\Delta T_{cond} = 4K$	fixed value from SIA 384/3 for refrigerant/water heat exchangers
$\Delta T_{evap} = 8K$	fixed value from SIA 384/3 for air/refrigerant heat exchangers
$\Delta T_{source,rated} = \Delta T_{source,bin} = 6K$	fixed value from SIA 384/3 for air - source heat pumps

Electric power consumption $P_{el,n}$ is estimated by a linear interpolation/extrapolation of known rating points at given outdoor air and heat pump outlet temperatures. Again, P_{el} is evaluated in each iteration loop from actual temperature $T_{sink,out,n}$. Finally, capacity is obtained from COP and P_{el} as

$$\dot{Q}_n = COP_n \cdot P_{el,n} \tag{25}$$

As an example of resulting temperature levels, in the A12 bin, an (operational) temperature at the outlet of the heat pump of 37.7 ºC is obtained from SIA 384/3, which is considerably higher than the EN 14825 result of 28.8 ºC. An overview of the outlet temperatures from

SIA 384/3 and EN 14825 calculations with the modelled fixed compressor speed heat pump is given below (table 19 and figure 24).

Condition	A-10	A-7	A2	A7	A12
Heating curve EN 14825	35 ºC	34 ºC	30 ºC	27 ºC	24 ºC
Outlet from EN 14825	-	34.0 ºC	32.1 ºC	30.5 ºC	28.8 ºC
Outlet from SIA 384/3	-	37.7 ºC	36.4 ºC	35.8 ºC	35.6 ºC

Tab. 19: ***Operational outlet temperatures for on/off controlled heat pumps (low temperature application, "average" climate). The first row shows the ideally required outlet temperatures for continuous operation (heating curve) from EN 14825. The following rows show temperatures reached due to cyclic on/off operation.***

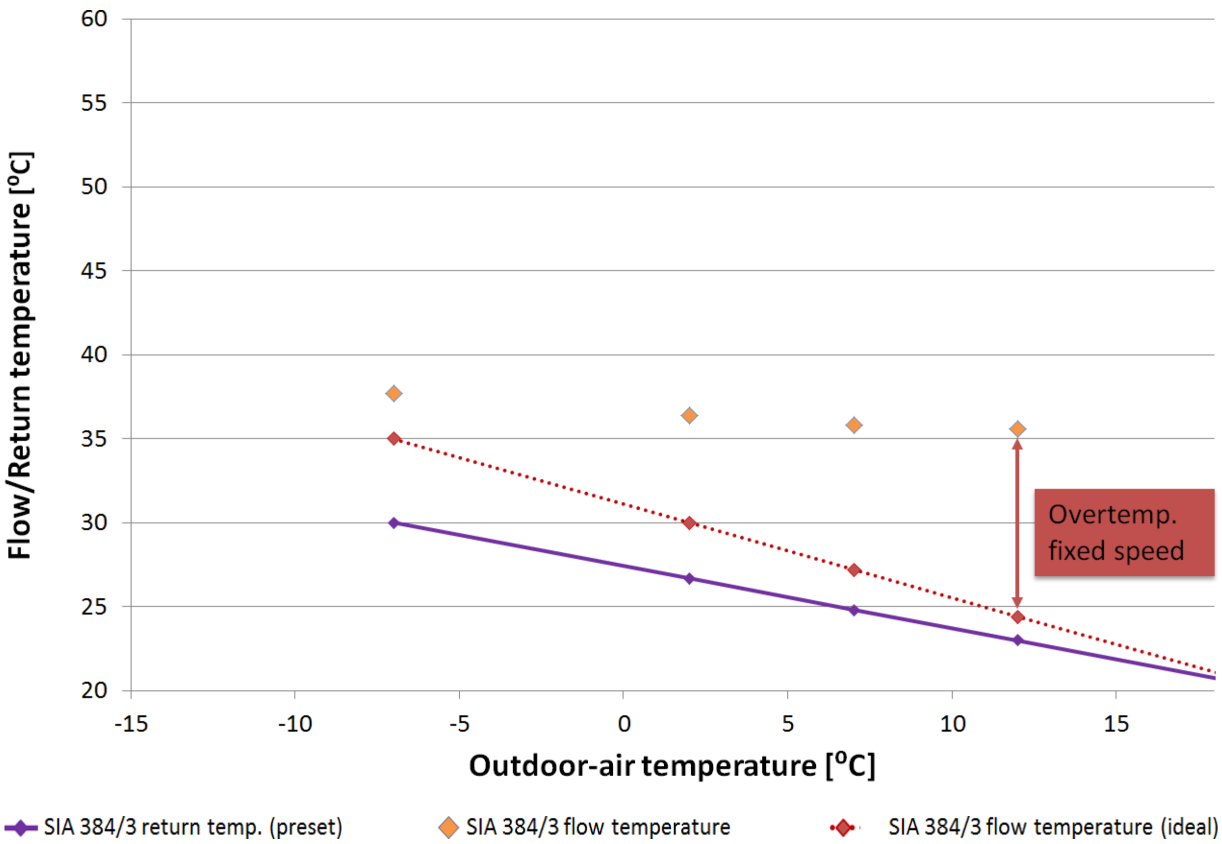


Fig. 24: ***SIA 384/3 heating curve: Return temperatures are preset (depending on heat emission system design), flow temperatures are then calculated from heat pump capacity and mass flow. Resulting overtemperatures due to on/off cycling are typically higher than values obtained from EN 14825.***

For variable speed compressor types, the outlet temperatures should ideally exactly be those given by the preset heating curve (within EN 14825 those given in first row of table 19 denoted as "heating curve"). Nevertheless, a small amount of overtemperature is inevitable at least for those operating points where minimum compressor speed is reached and no further reduction in capacity is possible. As the handling of variable speed compressor types is actually not described by SIA 384/3, a closer examination will be given in section 7.4.

While this chapter gave a deeper insight in the bin method as performed by SIA 384/3, the results of standardized annual efficiency calculations for all examined standards are presented in the following sections. Chapter 7.3 is about on/off controlled types, whereas chapter 7.4 presents results from systems with variable compressor speed heat pumps.

7.3 On/Off heat pumps

Figure 25 below shows the comparison of calculated and simulated heat pump efficiency in space heating mode $SPF_{HP,H}$ for the heat pump only system (without solar thermal support). As the WPesti tool is primarily made for building designers, the spreadsheet does not give detailed access to intermediate results such as electricity consumption in space heating mode alone. The space heating efficiency numbers as an outcome from WPesti calculations are empirically estimated values from the tool itself. Details about the WPesti modelling can be found in [17].

EN 14825 anyhow does cover space heating only, while SIA 384/3 as well as simulations allow a separate evaluation of space heating and sanitary hot water modes.

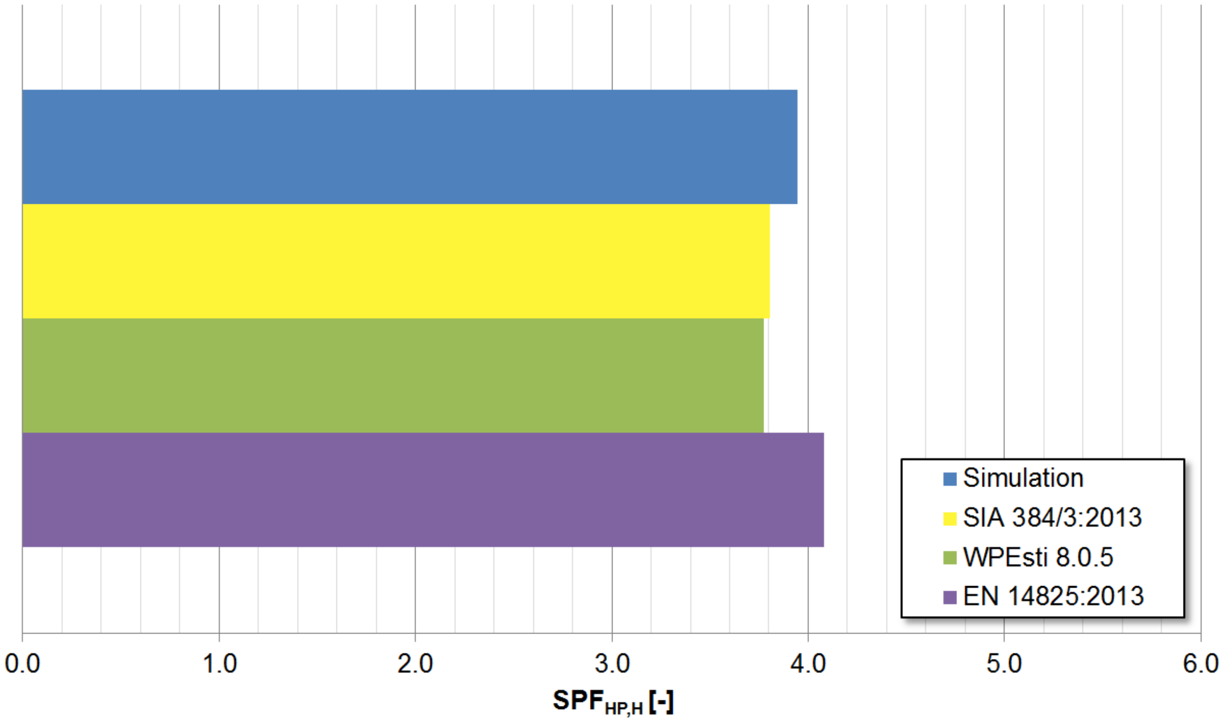


Fig. 25: ***Comparison of $SPF_{HP,H}$ calculations for heating mode. Heat pump only system. Calculations made for climate of Bern-Liebefeld, except EN 14825 (Strasbourg/France).***

The handling of electricity consumption in compressor "off" states is somehow different in all standards, which makes a fair comparison impossible. Therefore, none of the numbers given here do include any kind of standby electricity, be it during compressor off times while on/off cycling, be it from standby times when there is no heat demand. The influence of these "losses" will be discussed separately in chapter 7.5.

If the simulation is assumed to be the reference, WPesti outcome in heat pump space heating performance $SPF_{HP,H}$ as depicted in figure 25 is 3.1 % lower than the simulation result. The same applies to the SIA 384/3 evaluation. The result from EN 14825 is 4.6 % higher, but is based on another climate and a generic heating curve rather than actual data of the building -see also chapter 7.4.4. Nevertheless, a close correlation of the performance data can be observed. The deviation lies in the range of measurement uncertainty and is therefore considered as low.

As EN 14825 is made for heat pump assessment only (without solar thermal support), it cannot be applied to the evaluation of combined systems. It is therefore left out from the following discussion of solar-assisted heat generation (typical dimensioning). From SIA 384/3 and WPesti, the comparison with simulations for those systems with solar thermal support show nearly identical results. There is one exemption of a larger deviation of -12.8% in heating efficiency resulting from WPesti calculation of the system with solar thermal support for DHW preparation only (Figure 26). Simulation outcome is $SPF_{HP,H}$=3.9, the corresponding value from WPesti is 3.4. The discrepancy in space heating mode is compensated by sanitary water preparation, where WPesti overestimates the efficiency of the heat pump by 23 % or 0.6 SPF "points". When both operating modes are evaluated in combination, the seasonal performances $SPF_{HP,HDHW}$ show a nearly identical outcome when compared to simulations (see figure 28 in the following paragraphs).

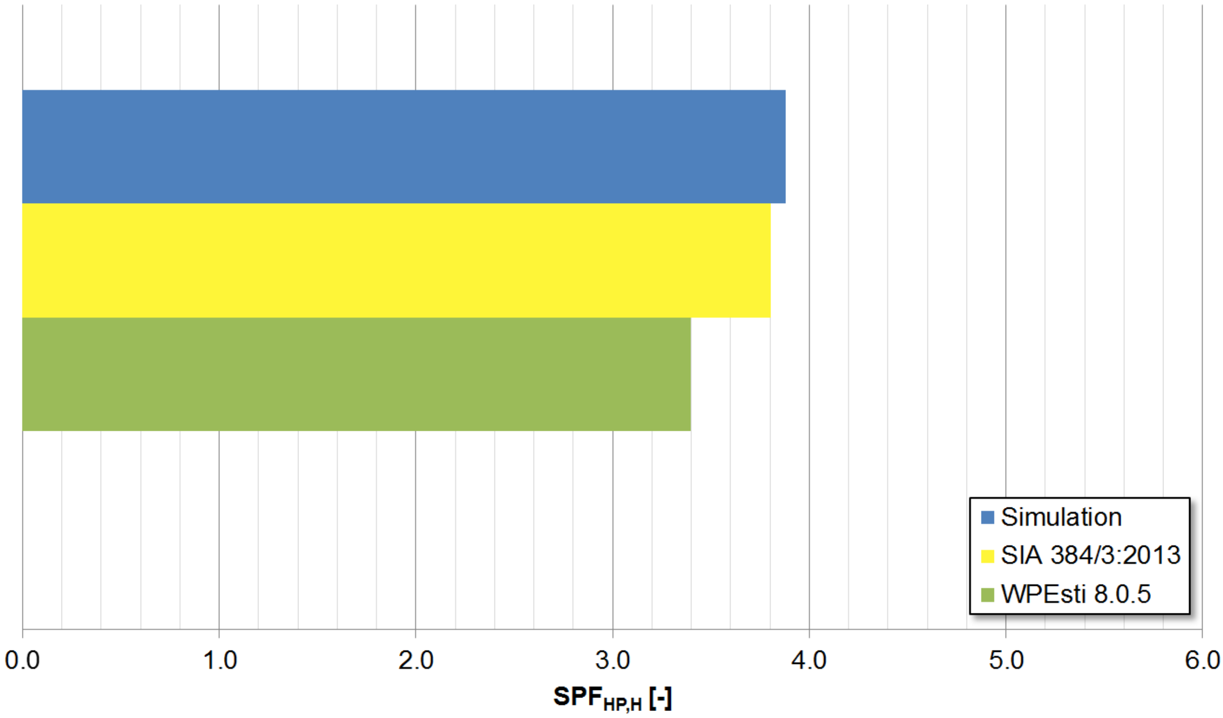


Fig. 26: ***Comparison of $SPF_{HP,H}$ outcome from standard calculations and simulations. System with solar thermal support for sanitary water preparation, climate of Bern-Liebefeld, two solar thermal collectors. WPesti estimates $SPF_{HP,H}$ via an empirical allocation to space heating mode from overall performance $SPF_{HP,HDHW}$, which explains the deviation from simulation.***

As mentioned above, the space heating efficiency as well as sanitary water preparation efficiency evaluated via WPesti are based on an empiric estimation from overall performance. Therefore, no further examination is carried out.

When comparing DHW mode in figure 27, seasonal performance from SIA 384/3 is 15.9 % higher compared to the simulation. Again, the evaluation shown in figure 27 is made for the system without solar thermal support. Closer examination of the sanitary water generation shows that mean flow temperature in the simulation is 45.1 ºC - just a tenth higher than the default value given by SIA 384/3. While return temperatures (heat pump inlet) during hot water preparation also almost exactly match (34.8 ºC in simulation vs. 34.3 ºC from SIA 384/3), the explanation in the $SPF_{HP,DHW}$ difference lies in the thermal intertia of the condensor. A typical sanitary hot water production cycle takes about 30 minutes, starts with 20 ºC heat pump outlet temperature and ends at latest when 57 ºC outlet is reached (to prevent overheating of the heat pump), see table 20. With 40 kJ/K overall thermal inertia on the hot side, about 10 % of the generated heat is consumed to heat up the thermal mass of the evaporator and surrounding circuitry. This heat usually will be "lost" (or recovered in subsequent space heating mode), but does not contribute to DHW preparation. As a consequence, efficiency for a DHW production cycle is decreased when compared to a calculation based on steady state conditions alone.

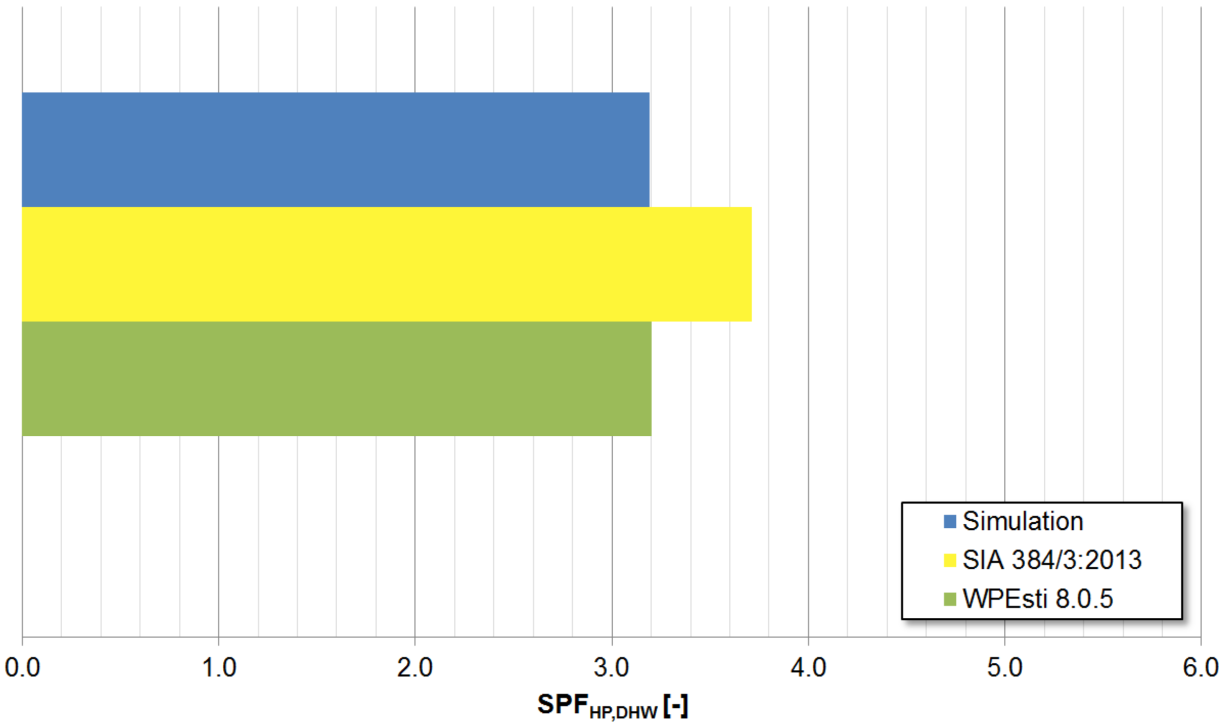


***Fig. 27:** Comparison of SPF_{HP} calculations for DHW mode. Heat pump only system. The purely steady-state calculation of SIA 384/3 explains the higher $SPF_{HP,DHW}$ outcome in this calculation.*

Yet standard calculations are based on such steady state rating points from the lab, which is why efficiency is somewhat overestimated here. The situation may be best explained by looking again at the example from table 20: Runtimes in DHW mode in the 3 ºC outdoor-air temperatur bin are a maximum in SIA 384/3 calculation. This bin is therefore considered as representative and has been chosen for demonstration. Derived from steady state simulations, at 3 ºC source temperature, the capacity of the heat pump is about 9 kW. At an outlet temperature of 45 ºC (for DHW preparation from SIA 384/3), the simulated steady state COP of the heat pump is 3.3. From the simulation results of a DHW preparation cycle in table 20, the average COP in DHW mode is 2.8. The average source temperature is 3.1 ºC (inlet) and the mean sink temperature is 44.5 -closely matching the steady state conditions. Nevertheless, "dynamic" COP in this specific case is 15 % lower than the value from steady state rating values.

$T_{outdoor\text{-}air}$	T_{outlet}	T_{inlet}	COP	Time
ºC	ºC	ºC	-	min
3.2	20.0	17.0	1.4	0
3.2	42.8	31.8	3.4	5
3.1	46.6	36.2	3.0	10
3.1	49.9	38.5	3.1	15
3.1	52.8	42.3	2.7	20
3.0	55.4	45.9	2.3	25

Tab. 20: Dynamics of a DHW preparation cycle on a randomly chosen day in February (simulation output).

Simulation runs without any thermal inertia at the sink side show almost exactly the $SPF_{HP,DHW}$ that is determined by standard calculation (from simulation without thermal inertia at the hot side $SPF_{HP,DHW}$ = 3.7, the same value as obtained from SIA 384/3). As a conclusion, it ia advisable to bear dynamic effects during hot water preparation in mind when the standard is revised. All the more, as the weight of sanitary water preparation on overall performance increases with building insulation quality.

So far, space heating and DHW preparation modes have been evaluated separately. Figure 28 shows the results of the combined space heating/sanitary water preparation operation for all three systems described in 6.1, that is heat pump as unique heat generator, heat pump with solar thermal support for DHW preparation (1-3 collectors) and heat pump with solar thermal support for DHW and space heating (4-8 collectors). As seen before, higher efficiency values from SIA 384/3 calculation are a result of overestimating $SPF_{HP,DHW}$ by neglecting dynamic effects. However, it does barely show up in the efficiency of the combined space heating and DHW operation, as much more of the heat is generated for space heating purposes (9282 kWh vs. 2333 kWh in the system without solar thermal support). The influence of sanitary hot water production on overall performance is therefore rather small. Similarly, a slight improvement of SPF can be observed when going from one to three collectors: The larger the collector field size, the less hot water is produced by the heat pump and the less is its influence on overall performance.

The largest deviation in combined operation occurs in the system with solar thermal support for space heating and DHW preparation, where WPesti outcome is always a little lower than simulation results.

Despite the mentioned drawbacks, variations between all three calculation methods are all within $\Delta SPF_{HP,HDHW}$ = ±0.15 or slightly below 4 % (referenced to simulation values). Again decided from admissible measurement uncertainties, the Monte Carlo examinations of chapter 6.4 and for the boundary conditions chosen, the bin method can be considered as an adequate method of energy assessment for combined systems consisting of on/off controlled heat pumps and an additional solar thermal heat generator.

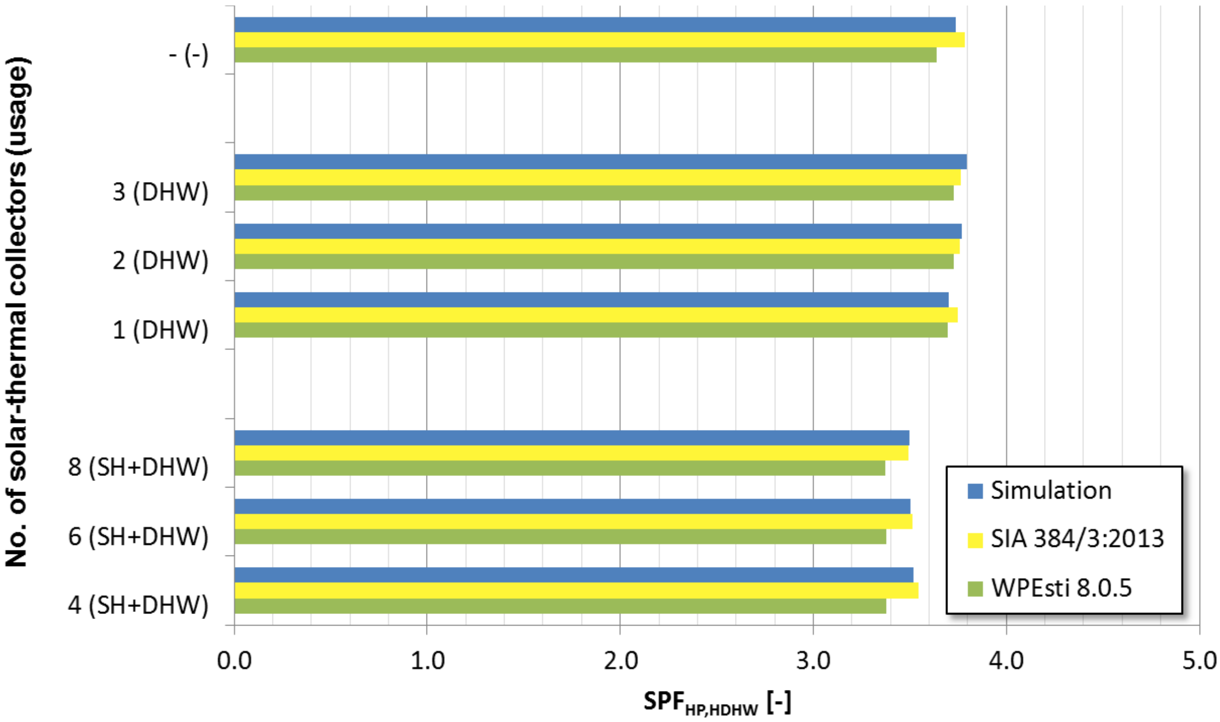


Fig. 28: ***Comparison of $SPF_{HP,HDHW}$ calculations for combined space heating and DHW. All systems, fixed speed heat pump.***

7.4 Variable speed heat pumps

The same comparisons as for on/off controlled heat pumps in previous chapter 7.3 have been carried out for variable capacity types. While EN 14825 is specifically made for part load situations and variable speed compressor heat pumps, SIA 384/3 is not yet. Therefore, some methods how to handle such heat pump types have been investigated. In a first step, several implementations of an algorithm for compressor speed controlled heat pumps have been implemented in the bin setup procedure of SIA 384/3 that is described in chapter 7.2. Three of these options are presented in the following subsections:

1) Using "plain" rating data from EN 14825, regardless of real design of the heating system (e.g. design mass flow, design heating curve)
2) Using simulated testing results, based on the (designed) operating conditions of the actual heating system
3) Using modified EN 14825 data, that is altered rating data to better fit on-site conditions

In all cases, input data for the assessment of the heat pump performance for DHW generation is not affected in any way. It is assumed that sanitary water is always produced at full compressor speed, therefore yearly efficiency values $SPF_{HP,DHW}$ do not change for speed controlled devices.

7.4.1 Plain EN 14825 data

Ecodesign requirements for heat pumps have been published in EU's official journal in September 2013. They must be fulfilled throughout the European Union, from September 2015 on. Also energy labelling of heat pumps will additionally be required -both requirements are based on EN 14825 rating data. As an accompanying measure, besides computed yearly performance data such as SCOP or η_s, the EN 14825 data basis -that is single rating points as well as power consumption in non-active operating modes- shall be publicly published. Hence it is assumed that such performance values will be commonly available and can serve as a basis for any energy assessment procedures of heat pump systems. But these values are still lab-based and do not necessarily reflect "real" operating conditions (temperature levels, mass flow rate). Nevertheless, due to the expected wide availability, these values are considered as ideal for energy efficiency assessments such as SIA 384/3 seasonal performance factor calculation.

The most simple approach for an SPF calculation is to use such test rig data as is. It is the method described in the following. EN 14825 rating data has therefore been generated by simulating a test rig for the modelled heat pump. Again, the bivalence point for the virtual heat pump testing was set to -10 ºC (design temperature for average climate) as would be done for a monovalent heat pump layout according to Ecodesign guidelines. Accordingly, part load fractions (demand side) for each rating point have been taken from EN 14825 (design heat load/maximum demand is given by the heat pump capacity and the selection of the bivalence point). With the exemption of choosing a heat pump model that fulfils minimum capacity requirements of the building and an EN 14825 temperature application (that is one of the predefined heating curves from EN 14825) which fits to the distribution system, there is no further correlation between the heat pump an its final application. For the SFH45 building used here, a low temperature application in the average climate was chosen. Rating data from the virtual test rig is summarized in table 21. At A7 and A12 conditions, even the variable compressor speed heat pump begins to cycle on/off, as minimum capacity is reached. Operating outlet temperatures at these points are slightly higher than those of the given heating curve, the overtemperature calculation follows the procedure of annex D in EN 14825.

Condition	Unit	A-10	A-7	A2	A7	A12
COP (operational)	-	2.9	3.2	4.9	7.1	9.4
COP (cycling)[a]	-	2.9	3.2	4.9	6.9	7.3
thermal capacity (Part load)	kW	6.8	5.9	3.6	3.1	3.4
floor emission system temperature	ºC	35	34	30	27	24
outlet temperature	ºC	35	34	30	27.4	25.2
on/off cycling	-	no	no	no	yes	yes

[a]EN 14825 COPbin

Tab. 21: Simulated COP values of the virtual heat pump according to EN 14825 (low temperature application, average climate)

As mentioned above, such data is expected –as lab rated– to be commonly available in the future. Here, the operational COP data is used as an input to the corresponding outdoor-air temperature bins in the SIA 384/3 yearly performance calculation without any further processing.

For outdoor-temperature bins where no rating data is available, it is generated by a linear interpolation (or extrapolation for higher outdoor temperatures) from next neighbour values, as explained in 7.2. As the virtual test data from table 21 is evaluated under part load conditions, not much overtemperature due to mismatch of demand and capacity is expected in the application, leaving alone some inaccuracies due to discrepancy of mass flow and heating power demand during test conditions and operating conditions. Results of SPF calculation outcomes from this most simple approach will be presented in the following sections where all methods are compared.

7.4.2 Simulated data

Standard heat pump performance tests are carried out at a constant mass flow rate which is determined via a given temperature lift of the external medium at the A7 condition. For the modelled heat pump type, this mass flow rate is 1.76 m^3/h (0.49 kg/s) in the low-temperature application (EN 14825).

While in the section above the "worst case method" (most simple method) for the SPF calculation has been applied, the idea of the method presented here is to get the best possible SPF estimation following the SIA 384/3 path. Therefore, the evaluation of bin COP is again based on simulated data rather than a somehow simplified estimation derived from heat pump rating values. Contradictory to the approach described under "Plain EN 14825 data" in section 7.4.1, the virtual heat pump rating has been carried out at the lower mass flow rate required by the heating system (derived from ΔT_{sink} = 5 K at design temperature A-7) and preset return temperatures (as would typically be used to control the heat pump). The design mass flow rate of the heat emission system is 0.83 m^3/h (0.23 kg/s), just about half of the nominal EN 14825 flow. Therefore and due to the slight overdimensioning of the heat pump, outlet temperatures are a bit higher than what they ideally would be (see table 22). Design load temperature has also been adjusted to the climate of Bern-Liebefeld, which is $T_{designh}$ = -7 ºC instead of EN 14825 default value of $T_{designh}$ = -10 ºC. This can be observed in the table below as higher capacities (when compared to table 21): Capacity at A-7 is now 7.3 kW (full capacity) instead of 5.9 kW (part load capacity) from EN 14825. Zero-capacity setpoint has been chosen to be 20 ºC (0 kW demand), leaving some room for DHW preparation and utility lock times, which are typically 2 - 4 hours per day.

Condition	Unit	A-10	A-7	A2	A7	A12
COP	-	-	3.0	4.6	6.4	7.2
Thermal capacity (Part load)	kW	-	7.3	4.9	3.5	3.4
Ideal outlet temperature (heating curve)	ºC	-	35.0	30.0	27.2	24.4
Outlet temperature	ºC	-	37.6	31.8	28.5	26.5
Inlet temperature	ºC	-	30.0	26.7	24.8	23.0

Tab. 22: ***Simulated COP values of the virtual heat pump according to EN 14825, with modifications of rating conditions (low temperature application, climate of Bern-Liebefeld)***

Comparative calculations have been carried out for the heat pump only system and the two methods of treating variable speed heat pumps described so far. The results of seasonal performance outcomes are summarized in table 23 below. Simulation outcomes considered as reference values are also listed in the table. As no changes have been applied to sanitary water production, $SPF_{HP,DHW}$ is expectedly equal for both versions. Differences show up in $SPF_{HP,H}$ and combined $SPF_{HP,HDHW}$ values.

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$
Simulation (From table 13)	4.6	3.2	4.1
SIA 384/3 (plain EN 14825)	4.8	3.7	4.5
SIA 384/3 (simulated data)	4.5	3.7	4.4

Tab. 23: ***Efficiency outcome from SIA 384/3 for inverter driven heat pump. Comparison of different calculation methods. System without solar thermal support.***

While using plain EN 14825 data as input to the SIA 384/3 bins gives $SPF_{HP,H}$ = 4.8 instead of 4.6 from simulation, the relative difference of +4.3 % is still regarded as very moderate. Using simulated rating data that fits on-site conditions is a kind of "best case" approach and leads to $SPF_{HP,H}$ = 4.5, which is 2.2 % lower than simulation outcome. The higher efficiency from pure EN 14825 data is explained by a heating curve which gives lower outlet temperatures than those from the application in the SFH45. From the SPF outcome with simulated data, it is concluded that the bin model is still valid for variable capacity heat pumps. However, such rating data that is perfectly adjusted to the heating system requirements is not available. A feasible alternative of adjusting rating data to on-site boundary conditions is therefore searched. The next section describes a possible way of how this can be accomplished within the SIA 384/3 framework.

7.4.3 Modified EN 14825 data

As rating data which perfectly fit the boundary condition of each individual project is not available, a way to modify EN 14825 performance values that are guided by generic heating curves was searched. The modifications proposed here do correspond the way of how the heat pump has been modelled in Matlab/Simulink: Heat pump capacity from EN 14825 rating is adjusted to the presumed operation, assuming it can perfectly follow the preset heating curve (defined as load). From capacity/load, associated flow rates and temperatures, Carnot efficiencies are calculated for both, rating and operating conditions. Again, temperature "loss" across heat exchangers are taken proportional to actual power transmitted. At each rating temperature, the exergetic efficiency is assumed to be identical for rating and operating conditions, which allows the calculation of electric power P_{el} consumption from

$$P_{el} = \frac{\dot{Q}_{demand}}{COP} \tag{26}$$

where the COP is evaluated from (24). As part load capacity is assumed to equal demand, it is known and no iteration of the capacity/outlet temperature is performed. The following steps are carried out at each rated outdoor temperature condition, an example is given for A2. Starting points are EN 14825 rating values:

- From rated capacity and rating flow rate, $\Delta T_{sink,out,rated}$ and $\Delta T_{cond,rated}$ are evaluated for the part load situation. EN 14825 rating flow is determined at A7 and full capacity with ΔT_{sink} = 5 K. $\Delta T_{cond,rated}$ at the A7 condition and full capacity (10.2 kW) was set to 5 K. Part load values at A2 are then derived as:

Mass flow	$\dot{Q}$	$\Delta T_{cond,rated}$	$\Delta T_{sink,out,rated}$
EN 14825 rating	EN 14825 rating	calculated	calculated
0.49 kg/s	3.6 kW	1.8 K	1.8 K

- From heating curve, required part load capacity of the heat pump is calculated. It's assumed that the heat pump perfectly matches the design curve (here defined by A-7 and A20 points), that is heat output matches demand.

	A-7	A20	A2
	design value	design value	interpolated
Capacity/demand	7.3 kW[a]	0 kW	4.9 kW
Return temp.	30 °C	20 °C	26.7 °C

[a] Heat pump data

- The design mass flow of the heating system is known. From part load capacity evaluated above, $\Delta T_{cond,operation}$ and $\Delta T_{sink,out,operation}$ at operating condition are then evaluated.

Mass flow	$\dot{Q}$	$\Delta T_{cond,operation}$	$\Delta T_{sink,out,operation}$
design value	from above	calculated	calculated
0.23 kg/s	4.9 kW	2.4	5.1

- $COP_{operation}$ at operating condition is evaluated via the assumption of constant exergetic efficiency. Temperatures at refrigerant level are known from predefined heating curves (EN 14825 rating/SIA 384/4 design) and above evaluations. From (24):

COP_{rated}	$T_{flow,rated}$	$T_{return,384/3}$	$COP_{operation}$
EN 14825 rating	EN 14825 rating	design value	calculated
4.4	34	26.7	4.7

- From COP and part load, electricity consumption is derived via (26)

One difficulty lies in low demand situations, when on/off cycling sets in. If heating demand is lower than minimum capacity of the heat pump, the latter value shall be taken. When cycling, $T_{sink,out,rated}$ is also altered from EN 14825 tables, but it is hardly possible to conclude on this rating outlet temperature (during on "states") from product datasheets. Fortunately, the difference between these overtemperatures and the EN 14825-given mean values of the floor emission system temperature are rather low (1… 2 ºC) and the discussed part load situations occur only with low runtimes. The influence of this overtemperature is therefore neglected, for the evaluation of (24), $T_{sink,out,rated}$ may be chosen as given in EN 14825.

Once all required values are known for each rating bin, values at intermediate outdoor temperature classes are simply interpolated (or extrapolated, if required). The method has been successfully tested out and results are given in table 24.

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$
Simulation (From table 13)	4.6	3.2	4.1
SIA 384/3 (modified EN 14825)	4.6	3.7	4.4

Tab. 24: ***Efficiency outcome from SIA 384/3 for inverter driven heat pump. Efficiency values derived from modified EN 14825 rating input. System without solar thermal support.***

The $SPF_{HP,H}$ value established via the described path is equal to the simulated value. From reasons explained in chapter 7.3, sanitary water preparation performance is still higher than simulation output.

While the results presented so far originate from the heat pump only system, the same procedures have been carried out for the system with solar-thermal support (for DHW preparation, 2 collectors as well as for DHW and space heating with a typical configuration of 6 collectors). Tables 25 and 26 list the results.

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$
SIA 384/3 (modified EN 14825)	4.6	3.2	4.5

***Tab. 25:** Efficiency outcome from SIA 384/3 for inverter driven heat pump. Typical system with solar thermal support for DHW preparation (2 collectors).*

	$SPF_{HP,H}$	$SPF_{HP,DHW}$	$SPF_{HP,HDHW}$
SIA 384/3 (modified EN 14825)	4.2	3.1	4.1

***Tab. 26:** Efficiency outcome from SIA 384/3 for inverter driven heat pump. Typical system with solar thermal support for DHW and space heating (6 collectors, combi storage)*

In both cases, the modified SIA 384/3 overestimates heating efficiency, by 0.2 (system with solar thermal support for DHW preparation) and by 0.1 (system with solar thermal support for DHW and space heating). Nevertheless, using pure 14825 data would give even higher SPF values. With regard to the thoughts about uncertainties from chapter 6.4 it is concluded that the simple modification of EN 14825 rating data is an adequate approach for standard calculations.

7.4.4 System comparison

In this chapter, yearly simulations and standard calculations are compared for variable compressor speed heat pumps. As concluded from the sections above, modified EN 14825 has been used to carry out the SIA 384/3 standard calculations. As the WPesti tool does not yet allow to input data for variable heat pump outlet temperatures, the tool has not been used for variable compressor speed units. The comparison for space heating efficiencies $SPF_{HP,H}$ of the heat pump only system is shown in figure 29. The difference between the SIA 384/3 and the simulation is negligible, as pointed out in 7.4.2.

EN 14825 gives a very high efficiency of $SPF_{HP,H}$ = 5.2. There are various contributions explaining this: First, the evaluation has been carried out without any cycling or standby losses as would normally be done for Ecodesign assessment. The amount of these non-operating modes reduces the SPF to 4.8, which is a realistic value for speed controlled heat pumps see also chapter 8.1. It would however not be comparable to (SIA 384/3 and simulation). Then, the discrepancy in climate data shows up: EN 14825 calculation is based on the climate of Strasbourg, simulations and SIA 384/3 calculation uses data from Bern-Liebefeld. The operating hours in each outdoor-air temperature bin are shown in figure 30 as relative frequencies. For EN 14825, a shift towards higher outdoor temperatures can be observed when compared to Bern-Liebefeld. Conditions a low part loads with high COPs are much more present in the Ecodesign evaluation. Would EN 14825 distribution follow the shape of SIA 384/3, the outcome would again be $SPF_{HP,H}$ = 4.8 instead of $SPF_{HP,H}$ = 5.2. The influence of climate data is much more pronounced for variable compressor speed heat pumps than for on/off controlled devices, where the $SPF_{HP,H}$ difference is only 0.2. In either situation, a part of the higher SPF outcome from EN 14825 can be explained by the different climate conditions the calculation is applied.

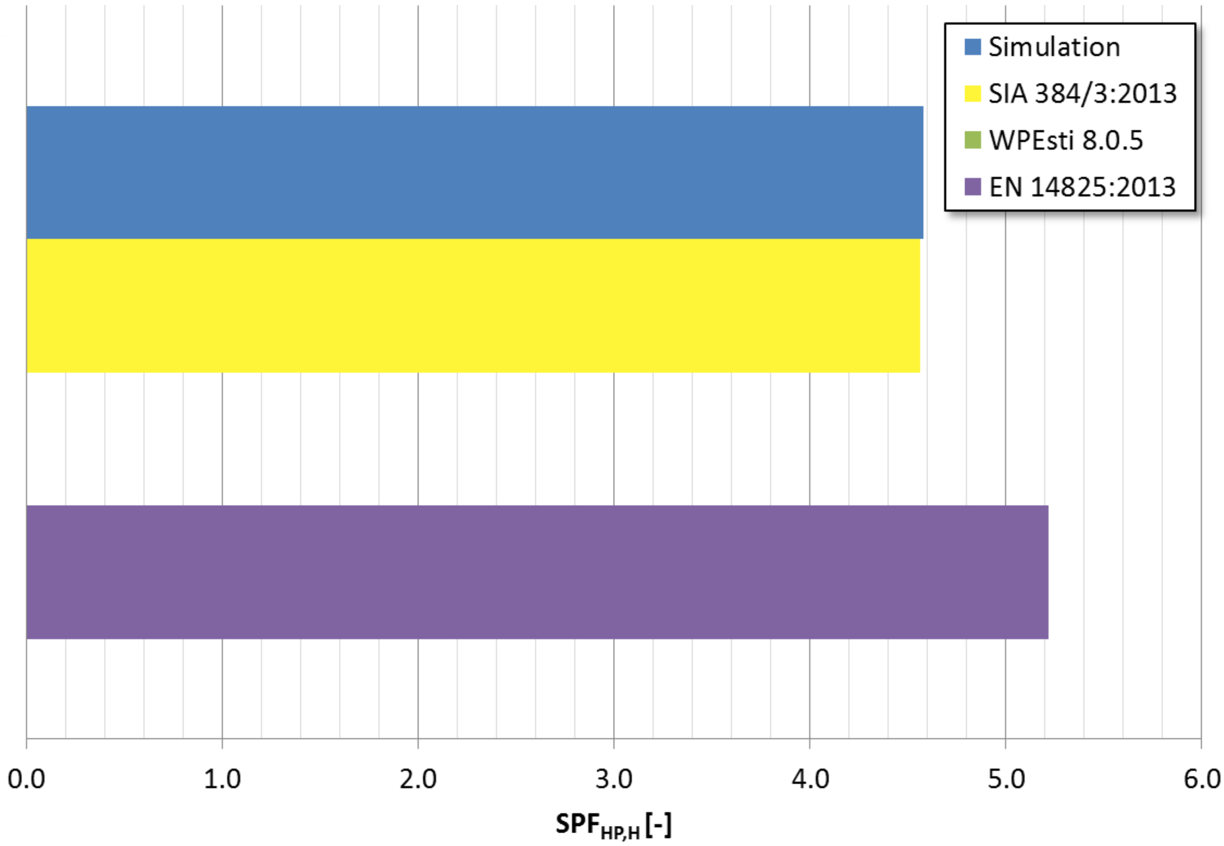


Fig. 29: Comparison of SPF_{HP} calculations for space heating mode. Calculations made for climate of Bern-Liebefeld, except EN 14825 (Strasbourg/France). Variable compressor speed heat pump.

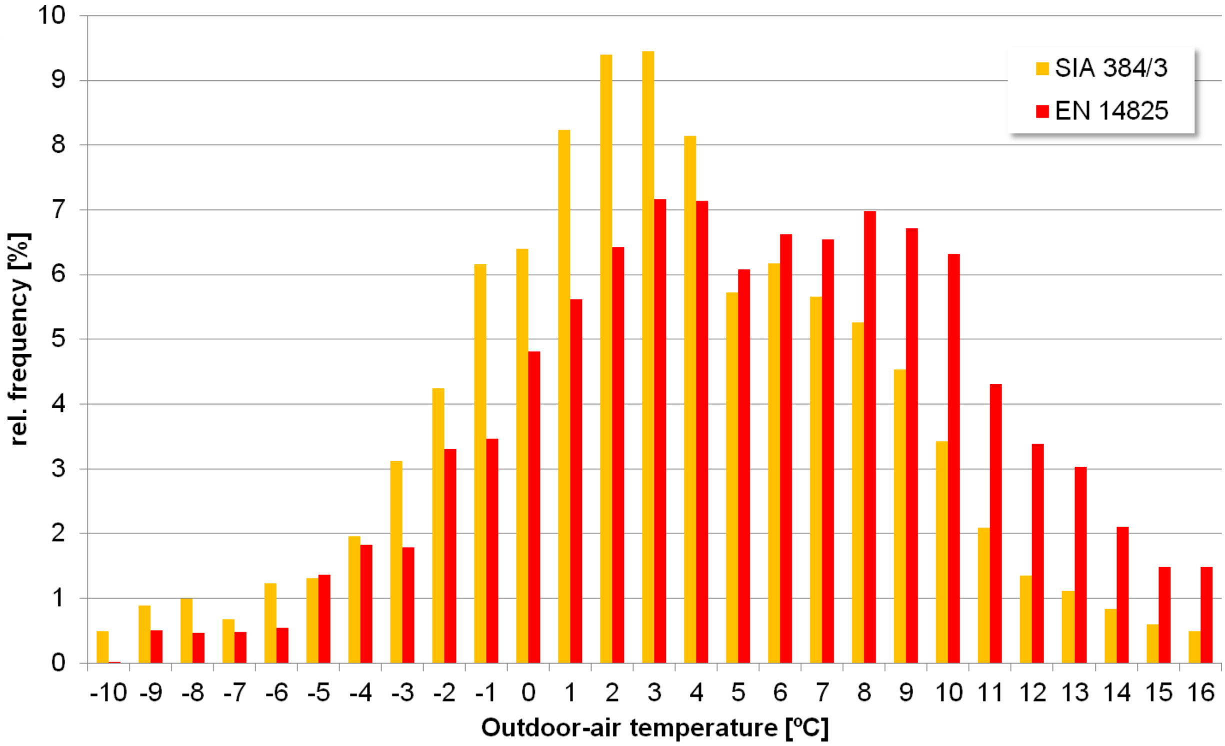


Fig. 30: ***Frequency of heat pump operating hours in space heating mode from SIA 384/3 (Bern-Liebefeld/Switzerland) and EN 14825 (Strasbourg/France).***

The comparison of the overall performance $SPF_{HP,HDHW}$ for all systems and configurations is depicted in figure 31. For on/off controlled types, space heating performance $SPF_{HP,H}$ from SIA 384/3 was consistently underestimated by a tiny amount, but compensated by higher $SPF_{HP,DHW}$ such that performance of both modes combined ($SPF_{HP,HDHW}$) almost perfectly matched simulation output. Now, the $SPF_{HP,HDHW}$ numbers from the standard calculation (SIA 384/3) are consistently higher, deviations from simulation are in the range from +3.8 % up to +9.5 %. The overestimation is a consequence of the higher $SPF_{HP,DHW}$ outcomes but even more due to higher heating efficiencies (systems with solar thermal support).

Still deviation from reference values (simulations) is kept in a range that can be accepted, just slightly above uncertainties accepted for heat pump rating according to Ecodesign (8%). Further, as will be shown in the next section, influence of standby power might be in the same range or even higher.

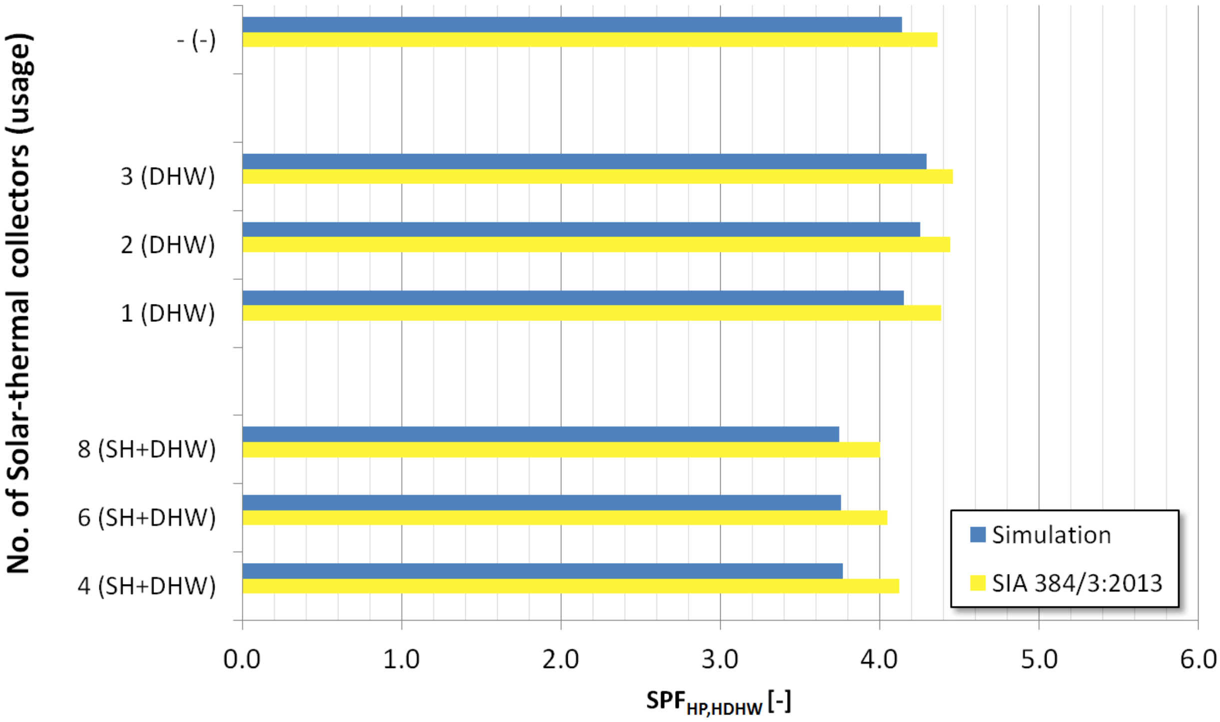


Fig. 31: ***Comparison of $SPF_{HP,HDHW}$ calculations for combined space heating and DHW. All systems, variable speed heat pump.***

7.5 Standby power

In the preceding sections, evaluations have been made excluding standby power or losses due to on/off cycling from calculations. Nevertheless, standard calculations allow the evaluation of these electricity consumptions. By default, Ecodesign explicitly requires such an assessment. These compressor "off" modes have a strong impact on yearly efficiency numbers, which will be demonstrated here. Refer to table 27 for a quick overview. Again, the heat pump only system is used for this purpose. From simulations, heat pump runtimes are 888 h and 301 h for space heating respectively domestic hot water preparation for fixed speed heat pumps. From this, standby time is 7571 h per year. At 46 W standby power consumption, this is approximately 349 kWh or 11 % of overall electricity consumption. That amount reduces seasonal performance factor $SPF_{HP,HDHW}$ from 3.7 by 0.4 "SPF-points" to 3.3.

From SIA 384/3, heat pump operating hours are 1004 in space heating modes, and 226 for DHW preparation. Yearly standby hours sum up to 7530. Again at 46 W standby power, $SPF_{HP,HDHW}$ decreases by 11 %, which matches the value obtained from simulations.

The prove of Ecodesign requirements is feasible for space heating mode only, excluding DHW. From Ecodesign, standby hours are 3850 h. At 46 W standby power, the yearly energy consumption is 177 kWh. "Off" states while on/off cycling when demand is lower than minimum capacity are already included in COP degradation. This explains the low standby hours and related energy consumption when compared to the values obtained above. SCOP decreases from 4.1 down to 3.5 when standby is included -an efficiency reduction of 13 %.

Concerning variable speed heat pumps, compressor runtimes for space heating drastically increase, while there is not much change in DHW preparation modes. From simulations, runtimes in space heating mode increase to 1328 h, while from SIA 384/3 calculations runtimes almost double, from 1004 h (on/off controlled) to 1985 h (variable capacity). This reduces standby times which account for standby losses. A value of 329 kWh resp. 301 kWh standby energy consumption results for this heat pump type in simulations resp. SIA 384/3 calculations. As overall electricity is also decreased for variable speed heat pumps, the contribution of standby modes is 12 % (simulations) respectively 11 % (SIA 384/3). There is no change from on/off heat pump types in the Ecodesign framework, standby hours are still 3850 h by definition. Thus, standby electricity consumption is again 177 kWh, $SPF_{HP,H}$ is reduced by 11 %.

As a conclusion of this examination, standby power becomes an important contribution to overall efficiency when heating demand and accompanying electricity consumption is reduced, as is the case for modern low energy demand buildings. In the evaluations of the heating system for the SFH45 building used here, seasonal performance is reduced by 10 - 15 %. It should therefore be included in any kind of heat pump assessment, necessary data will become available from European Ecodesign requirements.

	HP type	Standby time	Standby energy	$SPF_{HP,HDHW}$	$SPF_{HP,HDHW,sb}$
Simulations	on/off	7571	349 kWh	3.7	3.3
SIA 384/3	on/off	7530	343 kWh	3.8	3.3
EN 14825[a] (Ecodesign)	on/off	3850[b]	177[b] kWh	4.1	3.5
Simulations	variable speed	7131	329 kWh	4.1	3.7
SIA 384/3	variable speed	6094	301 kWh	4.4	3.9
EN 14825[a] (Ecodesign)	variable speed	3850[b]	177[b] kWh	4.7	4.2

[a]space heating only ($SPF_{HP,H}$)

[b]not including "off" states when on/off cycling

Tab. 27: Standby energy consumption: Simulation results for on/off type air-to-water heat pump system combinations

8 Ecodesign and the Swiss market

EU's Regulations 811/2013 and 813/2013 about Ecodesign an energy labelling of space heaters were published in the Official Journal of the European Union on 6th of September 2013. Underlying standard EN 14825:2013 covers all types of heat pumps, on/off controlled as well as variable speed types. While data for variable speed units are still rather seldom available, EN 14511 data for fixed-speed controlled heat pumps are publicly available, for instance through publications of the Swiss WPZ testing facility [18]. This data has been used to calculate energy classification of the heat pumps listed (Fig. 32 through 34), considering these datasets as representative for the Swiss heat pump market. Overall 139 datasets have been evaluated, whereof a majority is based on heat pumps using brine (68 units) or water (35 units) as a source of heat. Roughly 25 % of all datasets are from air-to-water heat pumps. While overall data quality is good and consistent, some values (power consumption in compressor off states and performance at rating points A2/W55, A12/W55 und A12/W35 for air-to-water heat pumps) had to be estimated from other rating points. A residual electricity consumption of 46 Watts has been assumed for all compressor off modes from EN 14825 except for crankcase heater power which was neglected. Unknown efficiency data at rating points A2/W55, A12/W55 and A12/W35 was estimated by linear interpolation or extrapolation from nearest neighbours.

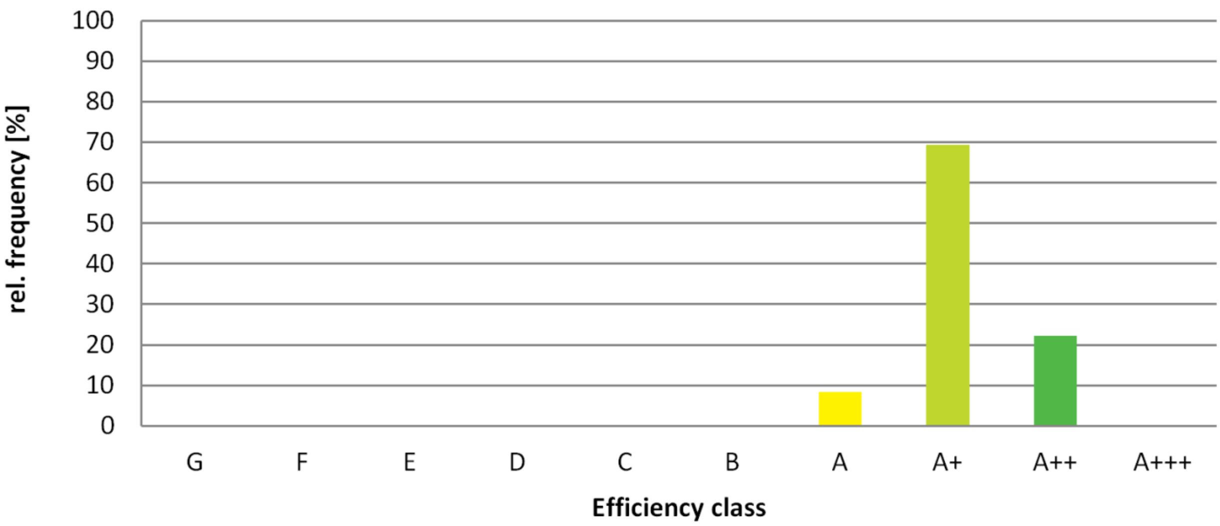


Fig. 32: ***Classification of actual air-to-water heat pumps on the Swiss market (Estimation, data from WPZ Bulletin [18]). Class A and higher fulfill Ecodesign requirements for low temperature (LT) application.***

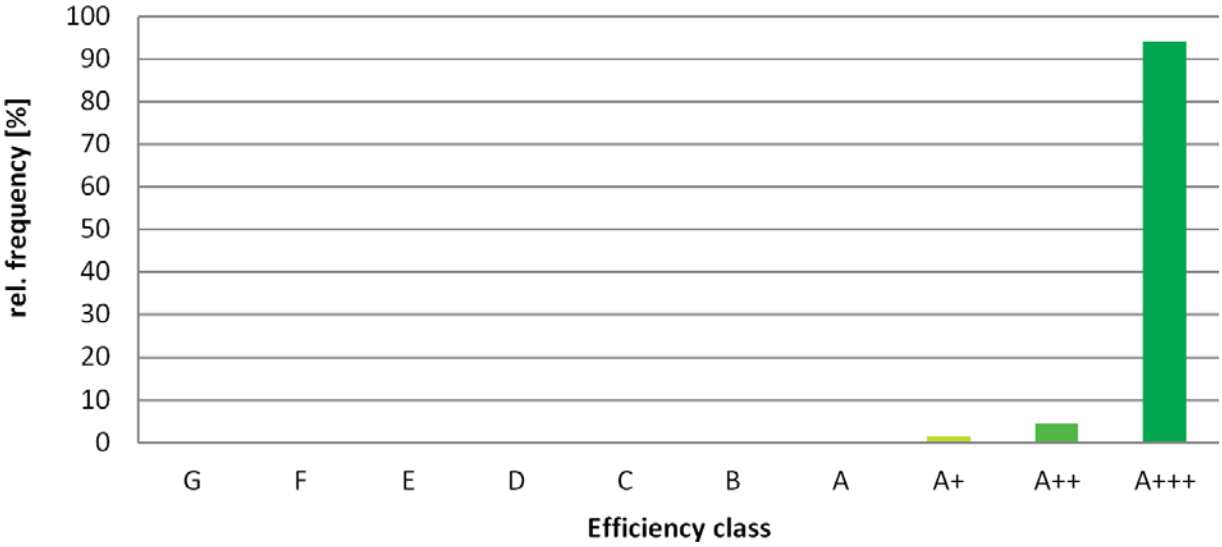


Fig. 33: ***Classification of actual brine-to-water heat pumps on the Swiss market (Estimation, data from WPZ Bulletin [18]). Evaluation for low temperature (LT) application. Generally, brine/water und water/water heat pumps reach high efficiency classes.***

While the figures above show classification of air/water and brine/water heat pumps, figure 34 (below) shows primary energy efficiency η_s according to EU definitions (primary energy factor for electricity 2.5) .

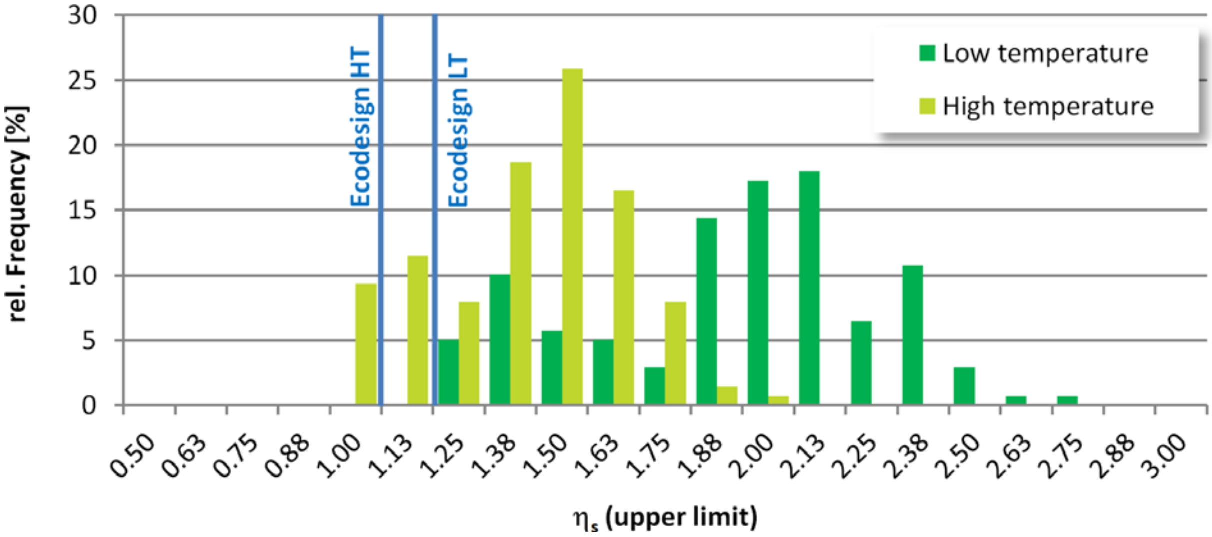


Fig. 34: ***Distribution of space heating energy efficiency η_s of heat pumps available on the swiss market (Estimation). All heat sources. Evaluation of data from WPZ. 9% of the units fail fulfilling the Ecodesign requirements 2015 for high temperature application.***

The evaluation shows that all regarded heat pumps will fulfill Ecodesign requirements 2015 for low temperature applications while 9 % of all units fail to do so for high temperature application (all air-to-water units). For the majority of the heat pumps that do not fulfill Ecodesign in this evaluation, the deviation between the calculated and the required η_s is lower than the 8 % accepted by Ecodesign regulations. These units would therefore still comply with Ecodesign requirements if they were declared to reach the minimum required efficiency of η_s. Only two units would definitely fail as their calculated efficiency is 91 %.

From September 2017 on, higher primary energy efficiency values will be required: 110 % for high temperature application (55 ºC flow temperature) and 125 % for low temperature (35 ºC flow temperature) application. From the evaluated WPZ dataset, 7 units would fail to comply with the low temperature requirements, all of these are outdoor-air source types. While only three brine/water heat pumps would not fulfill 2017 Ecodesign for high temperature application, 81 % of all air/water types would fail. Even if these 81 % were again declared to fulfill minimum requirements, still 20 units would definitively fail.

From these considerations, Ecodesign requirements will most possibly have an impact on air-to-water heat pump types, while brine-source and water-source units will barely be affected -they mostly reach Ecodesign already by now.

8.1 Classification of exemplary heat pump

The exemplary modelled air to water heat pump has been used to calculate classification according to energy labelling directive of the European Union. Again, compressor "off" power consumption was set to 46 W an the heat pump was chosen to operate without any additional backup heater. It is not equipped with a crankcase heater, its operation would further reduce efficiency (3850 h yearly runtime). From these boundary conditions, operational SCOP (designated $SCOP_{on}$ and accounting active heating operation only) for the on/off controlled heat pump is $SCOP_{on}$ = 3.9 while energy consumption during non heating times reduces efficiency to SCOP = 3.7. As the primary energy factor for electrical energy is CC = 2.5 and a supplementary penalty for the control logic of -3 % has to be applied, the primary energy efficiency η_s of the on/off heat pump is calculated as 145 %. From this value, the heat pump would reach an A+ classification in the low-temperature application regime of the average climate.

Following the same path for the variable speed heat pump, the operational SCOP is $SCOP_{on}$ = 5.1. Independent of heat pump type (on/off controlled or variable speed), non-operating hours are given from regulations. The standby energy consumption is therefore exactly the same for both types. For variable speed type compressors, the seasonal performance is reduced from $SCOP_{on}$ = 5.1 to SCOP = 4.80. The relevant number, η_s is calculated as η_s = 188 %. The efficiency increase of 26 % means a hop of two classes: The variable speed pump reaches A+++ classification.

heat pump type	SCOP	η_s	class
on/off	3.7	145 %	A+
variable speed	4.8	188 %	A+++

Tab. 28: Simulation results for on/off type air-to-water heat pump system combinations

Lately, first data evaluated according to Ecodesign has been published by manufacturers. While it's not possible to conclude on SCOP or η_s from classification, a coarse judgement is feasible. Exemplary datasheets [19][21] for air-to-water heat pumps show a classification of A (on/off type) and A++ (variable speed). Each type is one class higher than the exemplary types modelled for this study. A-class corresponds to an SCOP of 2.95 … 3.15, A++ is reached for SCOP = 3.20 … 3.83. The heat pumps modelled here show fairly good classification. For the comparison carried out here, absolute values are not that important, a match of relative values is much more important to keep results comparable.

9 Conclusion and outlook

The research made for this study shows, that the well-established temperature class or bin method is still usable for the efficiency assessment of combined heat-pump / solar-thermal heat generator systems, be it in combination with conventionally on/off controlled heat pumps, be it with modern variable compressor speed heat pumps. Swiss standard SIA 384/3 describes a calculation model where solar gains are used by priority to cover heat demand, which shows -in combination with a bin model- a very good match with simulation output. Deviations of annual heat pump performance between simulations -which are considered as the reference- and simplified standard calculations stay below 10 %, where the highest mismatch is observed for variable compressor speed heat pump types. The semi-empirical heat-pump modelling carried out for this study shows an advantage of variable speed compressors versus on/off controlled types of 11 % or 0.4 SPF points in combined space heating/sanitary hot water generation mode.

Further examinations also showed that dynamic effects during hot water generation shall not be neglected in simplified calculation models such as the bin method. Without further corrections, a bin model based on steady-state rating data overestimates heat pump efficiency $SPF_{HP,DHW}$ of domestic hot water preparation by about 15 % or 0.5 SPF "points", which is the highest deviation between standard calculation and simulation that was found in this study. The authors find it advisable to amend existing standards where this drawback occurs, as sanitary hot water preparation nowadays contributes about half of the heat demand of a typical high-performance residential building in central Europe.

All calculations carried out identify standby power as an important consumer of electricity. From the calculations carried out here, SPF is reduced by 10 - 15 % or 0.6 SPF "points" at most by standby electricity consumption. While not essential, it is at least advisable to regard these non-operating states in heat pump assessments as well.

Regarding European Ecodesign requirements, setting minimum performance standards and energy labelling as a user information instrument can be a helpful measure to increase performance of heat generators. Labelling of individual units can give an indication to the end-consumer of how well a heating system potentially can be, especially if different technologies are compared. However, it should be kept in mind that a highly efficient heat generator alone does not make an efficient system per se. Too many parameters contribute to a successful implementation of a high performance heating system. As the evaluation made shows, only few units that are available on the Swiss market will be affected by Ecodesign, the minimum performance limit could easily be set higher.

Associated with the Ecodesign requirements, there is an obligation to publish various performance data in a standardized manner, which comes in very handy for the design and precise assessment of individually composed systems in varying applications. The bin model was found to be suitable for variable compressor speed heat pumps too, when such data can be used to set up performance in each temperature bin. As these analyses show, while simulations are the most flexible variants, relatively simple an end-user friendly bin models can also satisfy the requirements of assessing (combined) heat pump systems.

Future standardisation works however (such as SIA 384/3 revisions) and calculation processes (e.g. WPesti) should make use of such data made available via European Ecodesign directive.

Last but not least, the authors of this study like to express their gratitude to the Swiss Federal Office of Energy (SFOW), especially to the program managers for the funding and the helpful guidance in the Annex 39.

10 Symbols

10.1 Symbols and abbreviations

A/W	Air to water (heat pump)
c	(Specific) heat capacity
CC	Degradation coefficient (from EN 14825:2013)
COP	Coefficient of performance
COPd	declared COP
COPbin	Bin specific COP
Cru	Capacity ration (from EN 14825:2013)
f	Part load factor
η	Efficiency
η_s	Seasonal space heating energy efficiency (from EU directives No. 811/2013 and 813/2013)
μ	Expectation value
$\dot{m}$	Mass flow
out	Outlet
P	Probability
Q	Heat
$\dot{Q}$	(Heat) capacity
σ	Standard deviation
SCOP	Seasonal coefficient of performance (from EN 14825:2013)
SEER	Seasonal energy efficiency ratio (from EN 14825:2013)
SPF	Seasonal performance factor
ST	Solar thermal
T	Temperature
V	Volume
X	Part load ratio

10.2 Subscripts

actual	(at) actual operation conditions
air	Air
ambient	Ambient
aux	Auxiliary
cond	Condenser
design	Design condition
designh	Design conditions for heating (from EN 14825:2013)
DHW	Domestic hot water mode

distri	(heat) Distribution
el	Electric
evap	Evaporator
ex	Exergetic
GEN	Generator
H	(Space) Heating mode
HP	Heat pump
HDHW	Space heating and domestic hot water
inlet	Inlet
loss	Losses
max	Maximum
min	Minimum
operation	Operating conditions
outlet	Outlet
r	Refrigerating (from EN 12900:2013)
rated	(at) rating conditions
ref	Refrigerating / Refrigerant
sb	Standby
sink	Sink (hot) side of heat pump
sol	Solar
source	Source (cold) side of heat pump
spec	Specific
SYS	System
X	(at) part load ratio

11 References

[1] Official Journal of the European Union, L285, 31 October 2009, **DIRECTIVE 2009/125/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 21 October 2009 establishing a framework for the setting of ecodesign requirements for energy-related products**, Publications Office of the European Union, Luxembourg, 2009, LU

[2] European Commission, **2030 Framework for climate and energy, Outcome of the October 2014 European Council**, Presentation found on http://ec.europa.eu/clima/policies/2030/documentation_en.htm (17.12.2014)

[3] Schweizerische Eidgenossenschaft, **Energiegesetz (EnG)**, Entwurf vom 4. September 2013, Bern, 2013, CH

[4] **EN 14511:2012, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling**, CEN, Brussels, 2012, BE

[5] The MathWorks GmbH, **Matlab/Simulink – Product Information**, Bern, 2005, CH, http://www.mathworks.ch

[6] **EN 14825:2013, Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling - Testing and rating at part load conditions and calculation of seasonal performance**, CEN, Brussels, September 2013, BE

[7] R**.**Dott et al., **The Reference Framework for System Simulations of the IEA SHC Task 44 / HPP Annex 38, Part B: Buildings and Space Heat Load, A technical report of subtask C, Report C1 Part B**, Muttenz, 2013, CH

[8] **SIA 384.342:2008, SN EN 15316-4-2:2008, Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4.2: Space heating generation systems, heat pump systems**, Schweizerischer Ingenieur- und Architektenverein (SIA), Zürich, 2008, CH

[9] **SIA 384/3, Heizungsanlagen in Gebäuden – Energiebedarf**, Schweizerischer Ingenieur- und Architektenverein (SIA), Zürich, 2013, CH

[10] **Commission Delegated Regulation (EU) No 811/2013 of 18 February 2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of space heaters, combination heaters, packages of space heater, temperature control and solar device and packages of combination heater, temperature control and solar device**, Official Journal of the European Union, L239, 6. September 2013, Publications Office of the European Union, Luxembourg, September 2013, LU

[11] **Commission Regulation (EU) No 813/2013 of 2 August 2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters**, Official Journal of the European Union, L239, 6. September 2013, Publications Office of the European Union, Luxembourg, September 2013, LU

[12] **ISO 13612-2, Heating and cooling systems in buildings — Method for calculation of the system performance and system design for heat pump systems — Part 2: Energy calculation**, ISO, Geneva, 2014, CH

[13] **CARNOT Blockset**, Version 5.0 beta, Solar-Institut Jülich, 2011 , DE

[14] K. Schwamberger, **Modellbildung und Regelung von Gebäudeheizungsanlagen mit Wärmepumpen**, Fortschrittsberichte VDI Reihe 6, Nr. 263, VDI Verlag Düsseldorf, 1991, DE

[15] **SN EN 12900:2013, Refrigerant compressors - Rating conditions, tolerances and presentation of manufacturer's performance data**, SNV, Winterthur, 2013, CH

[16] Viessmann Deutschland GmbH, **Planungsunterlagen für Wärmepumpen**, Ausgabe 5/2014, Allendorf, 2014, DE

[17] A. Huber, M. Stalder, **Rechenmethode WPesti: Handbuch**, 16.2.2013, CH

[18] Wärmepumpen-Testzentrum WPZ, **WPZ Bulletin Ausgabe 01 - 2013 vom 9.10.2013**, WPZ, Buchs, 2013, CH

[19] **Commission communication in the framework of the implementation of Commission Regulation (EU) No 813/2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters and of Commission Delegated Regulation (EU) No 811/2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of space heaters, combination heaters, packages of space heater, temperature control and solar device and packages of combination heater, temperature control and solar device**, Official Journal of the European Union, C207, 3 July 2014, Publications office of the European Union, Luxembourg, July 2014, LU

[20] Stiebel Eltron, **Produktedatenblatt WPL 10 AC**, STIEBEL ELTRON GmbH & Co. KG, undated, Holzminden, DE

[21] Stiebel Eltron, **Produktedatenblatt WPL 15/25 I(S)-2**, STIEBEL ELTRON GmbH & Co. KG, undated, Holzminden, DE

[22] **Directive 2010/30/EU of the European Parliament and of the Council of 19 May 2010 on the indication by labelling and standard product information of the consumption of energy and other resources by energy-related products**, Official Journal of the European Union, L153, 18 June 2010, Publications Office of the European Union, Luxembourg, June 2010, LU

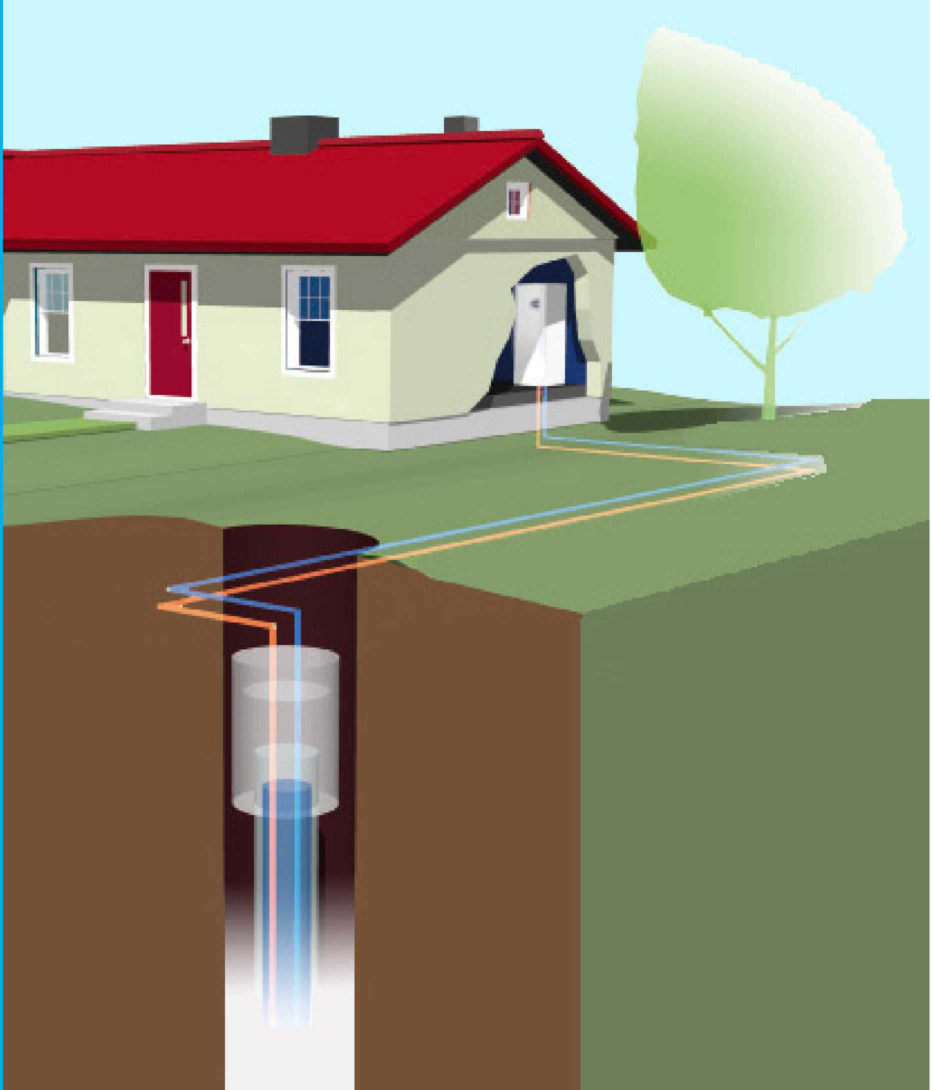
[23] **Commission Delegated Regulation (EU) No 812/2013 of 18 February 2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of water heaters, hot water storage**

tanks and packages of water heater and solar device, Official Journal of the European Union, L239, 6. September 2013, Publications Office of the European Union, Luxembourg, September 2013, LU

[24] **Commission Regulation (EU) No 814/2013 of 2 August 2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for water heaters and hot water storage tanks**, Official Journal of the European Union, L239, 6. September 2013, Publications Office of the European Union, Luxembourg, September 2013, LU

[25] Th. Afjei, **Scrollverdichter mit Drehzahlvariation, Abhandlung zur Erlangung des Titels Doktor der Technischen Wissenschaften der Eidgenössischen technischen Hochschule Zürich**, Zürich, 1993, CH

10.5 Appendix VI – National report Finland



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Renewable energy production of Finnish heat pumps

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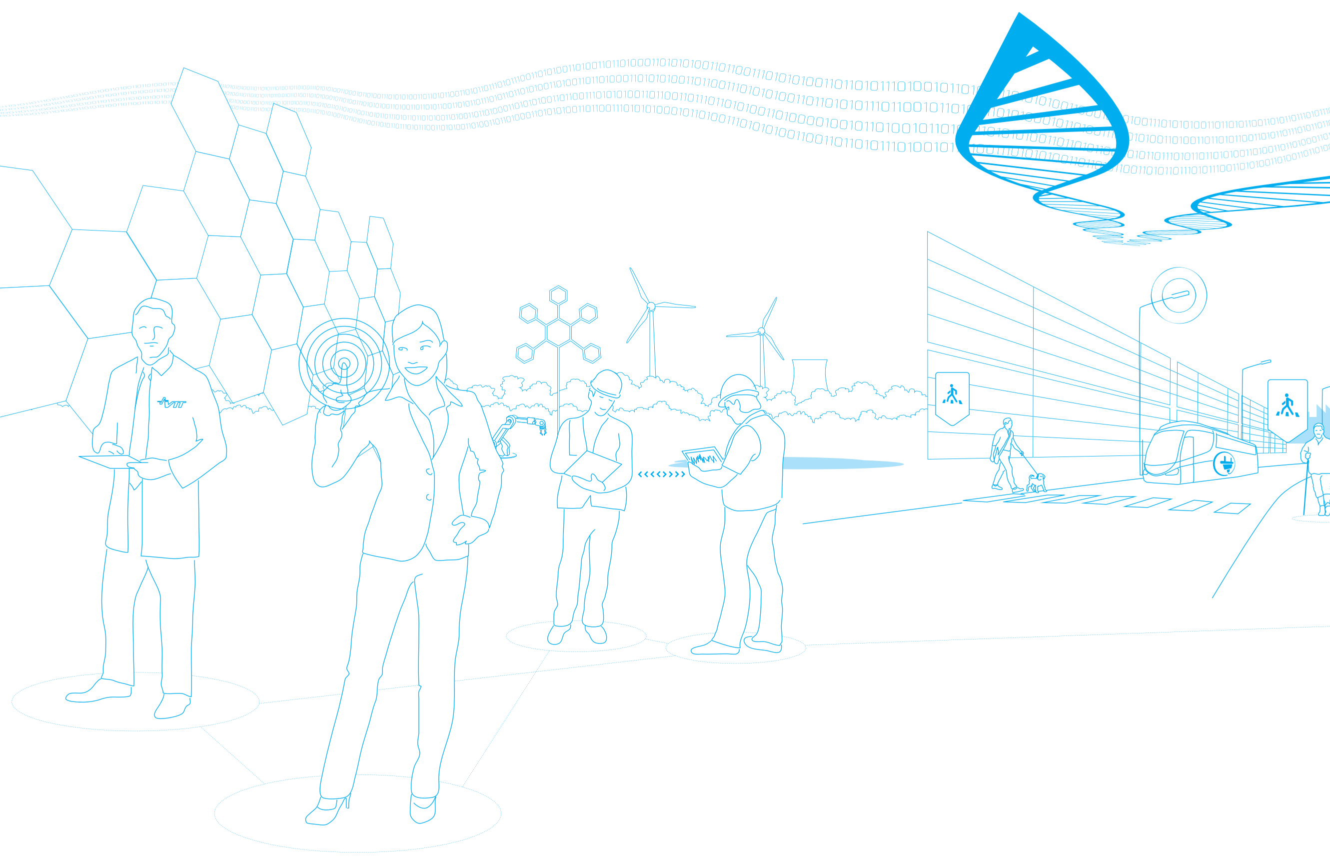

VTT TECHNOLOGY 164

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ISBN 978-951-38-8141-2 (URL: http://www.vtt.fi/publications/index.jsp)

VTT Technology 164

ISSN-L 2242-1211
ISSN 2242-122X (Online)

JULKAISIJA – UTGIVARE – PUBLISHER

VTT
PL 1000 (Tekniikantie 4 A, Espoo)
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Puh. 020 722 111, faksi 020 722 7001

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Renewable energy production of Finnish heat pumps
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Suomalaisten lämpöpumppujen uusiutuvan energian tuotto. SPF-hankkeen loppuraportti.
Ari Laitinen, Pekka Tuominen, Riikka Holopainen, Pekka Tuomaala, Juha Jokisalo, Lari Eskola & Kai Sirén.
Espoo 2014. VTT Technology 164. 90 p. + app. 30 p.

Abstract

The SPF project defined a national hourly seasonal performance factor calculation method for air to air heat pumps, air to water heat pumps, ground source heat pumps and exhaust air heat pumps in co-operation with international Annex 39 work.

The energy use of the Finnish building stock was estimated using standard building types further adapted to different decades: a detached house, an apartment building, an office building and a summer cottage. The energy use of these standard building types was calculated with different heat pump types leading to energy saving and renewable energy use of the heat pumps in different buildings.

The current and future cumulative energy consumption of the building stock was modelled using the REMA model developed at VTT. The future effects of heat pumps on the energy use and emissions of the Finnish building stock were modelled comparing with the REMA model a conservative Business as Usual scenario with a Heat Pump scenario.

Keywords seasonal performance factor, heat pump, energy saving, renewable energy

Suomalaisten lämpöpumppujen uusiutuvan energian tuotto
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Tiivistelmä

SPF-hankkeessa määriteltiin kansallinen tunneittainen kausihyötysuhteen laskentamenetelmä ilmalämpöpumpuille, ilma-vesilämpöpumpuille, maalämpöpumpuille sekä poistoilmalämpöpumpuille yhteistyössä kansainvälisen Annex 39 -ohjelman kanssa.

Suomen rakennuskannan energiankulutusta arvioitiin eri vuosikymmenille määriteltyjen tyyppirakennusten avulla, jotka edustivat pientaloa, kerrostaloa, toimistotaloa sekä vapaa-ajan rakennusta. Näille tyyppirakennuksille arvioitiin energiansäästöpotentiaali ja uusiutuvan energian tuotto eri lämpöpumppuvaihtoehdoilla.

Koko rakennuskannan nykyistä ja kumulatiivista energiankulutusta arvioitiin VTT:n kehittämällä REMA-mallilla. Lämpöpumppujen tulevaa vaikutusta suomalaisen rakennuskannan energiankulutukseen ja päästöihin arvioitiin vertaamalla perinteistä Business as Usual -skenaariota lämpöpumppuskenaarioon, joka kuvaa lämpöpumppujen nopeampaa yleistymistä.

Avainsanat seasonal performance factor, heat pump, energy saving, renewable energy

Preface

The object of the SPF project was to define a national seasonal performance factor calculation for heat pumps in co-operation with Annex 39 work. The other goal was to estimate the energy saving and renewable energy potential of heat pumps on the Finnish building stock. The project duration was from 1.3.2011 to 31.12.2013.

The project was financed by the Finnish ministry of employment and the economy, the Finnish ministry of the environment and SITRA, Finnish Innovation Fund. The work was performed by VTT Technical Research Centre of Finland (coordinator), Aalto University and the Finnish heat pump association SULPU.

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Appendices

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List of symbols

b	[°C]	constant term of the control curve
$c_{p,air}$	[kJ/kg,K]	specific heat capacity of air
$c_{p,wv}$	[°C]	specific heat capacity of water vapour
$COP_{DSW}(t)$	[-]	hourly heat pump COP for DSW heating
COP_N	[-]	measured heat pump COP in test point
$COP_{spaces}(t)$	[-]	hourly heat pump COP for space heating
COP_T	[-]	theoretical heat pump COP
$COP_T(t)$	[-]	theoretical hourly COP of the heat pump
E_{res}	[kWh/a]	annual renewable energy production
$f_T(t)$	[-]	loss factor of the compressor
h	[kJ/kg]	enthalpy
$h_1(t)$	[kJ/kg]	outlet air enthalpy before the condenser
$h_2(t)$	[kJ/kg]	waste air enthalpy after the condenser
H_{HP}	[h]	equivalent running time at full load
k	[-]	slope of the control curve
L_w	[kJ/kg]	evaporation heat for water in 0°C
P_{aux}	[kW]	electricity consumption of the auxiliary devices
P_{rated}	[GW]	installed nominal power of heat pumps
$q_{HP,DSW}(t)$	[kWh]	DSW heating energy production
$q_{HP,spaces}(t)$	[kWh]	space heating energy production
$q_{HP,spaces,max}(t)$	[kWh]	maximum space heating energy production
Q	[m³/s]	nominal flow of air or fluid

$Q_{additional}$	[kWh]	additional heating energy
$Q_{heating,DSW}$	[kWh]	DSW heating energy demand
$Q_{heating,spaces}$	[kWh]	space heating energy demand
$Q_{HP,DSW}$	[kWh]	DSW heating energy production of the heat pump
$Q_{HP,spaces}$	[kWh]	space heating energy production of the heat pump
$Q_{outlet}(t)$	[m³/s]	outlet air flow
Q_{usable}	[kWh/a]	annual heating energy production of a heat pump
S_{Hdd}	[Kh]	heating degree day
t_{DSW}	[h]	DSW heating time of heat pump during one time step
$t_{space,max}(t)$	[h]	space heating time of heat pump during one time step
$t_{time\text{-}step}$	[h]	calculation time step
T_{dim}	[°C]	dimensioning outdoor temperature for space heating
T_{DSW}	[°C]	DSW temperature
T_{fluid}	[°C]	fluid temperature leaving the heat collection circuit
T_{Hdd}	[°C]	indoor temperature representing the heating degree day
$T_{HP,max}$	[°C]	maximum water temperature the heat pump can deliver
T_{HSo}	[°C]	heating source temperature
T_{HSy}	[°C]	building heating system temperature
T_{indoor}	[°C]	indoor air temperature
T_{iw}	[°C]	temperature of the inlet water leaving the condenser
$T_{iw,max}$	[°C]	max. inlet water temp. in dimensioning outdoor temp.
$T_{iw,min}$	[°C]	minimum inlet water temperature
$T_{outdoor}(t)$	[°C]	hourly outdoor temperature
$T_{outdoor,iw,min}$	[°C]	outdoor temp. representing minimum inlet water temp.
T_{SW}	[°C]	cold service water temperature
$T(t)$	[°C]	outlet air or waste air temperature
$w_{HP}(t)$	[kWh	hourly electricity consumption of a heat pump]
W_{aux}	[kWh]	electricity consumption of auxiliary devices of heat pump
W_{HP}	[kWh]	electricity consumption of the heat pump
$x(t)$	[kg/kg]	absolute humidity of outlet air or waste air

Greek symbols:

$\beta_{HP,DSW}(t)$	[-]	hourly load power ratio for DSW
$\beta_{HP,spaces}(t)$	[-]	hourly load power ratio for space heating
ΔP_e	[Pa]	static ductwork or pipework pressure loss
Δt	[h]	auxiliary device usage time during calculation period
$\phi_{DSW}(t)$	[kW]	hourly DSW heating power demand of the building
$\phi_{HPc}(t)$	[kW]	condenser power of the heat pump
$\phi_{HP,DSW}(t)$	[kW]	hourly DSW heating power of the heat pump
$\phi_{HP,max}(t)$	[kW]	maximum heating power of the heat pump in test point
$\phi_{HP,spaces}(t)$	[kW]	hourly space heating power demand of the heat pump
$\phi_{spaces}(t)$	[kW]	hourly space heating power demand of the building
η	[-]	fan or pump efficiency rate
ρ	[kg/m³]	outlet air density

Abbreviations:

DSW	Domestic Service Water
HP	heat pump

1. Introduction

The effectiveness of a heat pump can be described by means of Coefficient Of Performance (COP) and Seasonal Performance Factor (SPF). COP represents how much heat power a heat pump delivers in relation to the electricity demand of the compressor and electrical devices, and it is calculated by dividing the delivered heat power with the electricity consumption of the heat pump. COP is calculated at single operation conditions and at full capacity, even though these conditions do not always reflect the real performance of heat pumps in practical operation in heating systems. Heat pumps mainly operate intermittently or at reduced capacity (through capacity control) in climatic conditions that differ from the standard rating conditions.

SPF is defined by dividing the energy output of a system by the energy used for the production and therefore it presents better the actual operation and annual savings of the heat pump. The influence of part load or variable capacity on SPF is currently not fully covered by existing methods for calculation of SPF (Annex 39 legal text). According to the European Heat Pump Association, the standard EN14511 does not specify the calculation method for SPF and therefore it is important to clearly define the system boundary.

The first main object of the SPF project was to define a Finnish SPF calculation method for heat pumps in co-operation with international Annex 39 work. The other main object was to estimate the current and future energy saving and renewable energy use potential of heat pumps on the Finnish building stock.

2. Heat pump volume scenarios

2.1 Heat pump stock and capacity prognosis (domestic heat pumps)

The year 2010 numbers of installed heat pumps in this chapter are based on the sales statistics maintained by the Finnish Heat Pump Association (Sulpu).

Air-air heat pumps are dominating the markets in terms of sales numbers. Traditionally, they have been a cost-effective way of retrofitting electrically heated houses. They have also been installed in a large number of summer cottages, where they provide a complement to electric heating. It should be noted that air/air heat pumps are sold through a number of channels: builder's merchants, mail-order firms, and web stores. Since there are no sales statistics for these channels, the sales numbers for air/air heat pumps can only be estimated.

Though the installed number of ground source heat pumps is much lesser than air/air heat pumps they play important role in installed capacity in the use of renewable energy. Ground source heat pumps will also play important role in the future in the new building sector as well as in retrofitting of old oil and electrical heating systems.

Exhaust air heat pumps have a moderate market share, but one which will probably decrease in the future. They have been installed in some new-building construction. However, the standard type of these heat pumps does not comply with the new, more stringent building regulations. They may be used in the most well-insulated new buildings, or may replace old exhaust air heat pumps.

Recently, air/water heat pumps have been introduced on the Finnish retrofit market but their current share of the market is marginal and the development is seen moderate.

The total number of heat pumps is estimated to reach about 950 000 units by 2020. These estimations take into account that part of the heat pump sales are replacements, thus the total heat pump sales will be higher than the numbers shown in Figure 1 and Table 1. The cumulative number of heat pumps is calculated based on the following life time expectations: air/air heat pumps 10 years, ground source heat pumps 20 years, air/water and exhaust heat pumps 15 years. The calculated heat pump heating capacity 2010 and estimations for 2016 and

2020 are presented in Table 2. The scenario of the heating capacity of the heat pumps by Sulpu is presented in Figure 2.

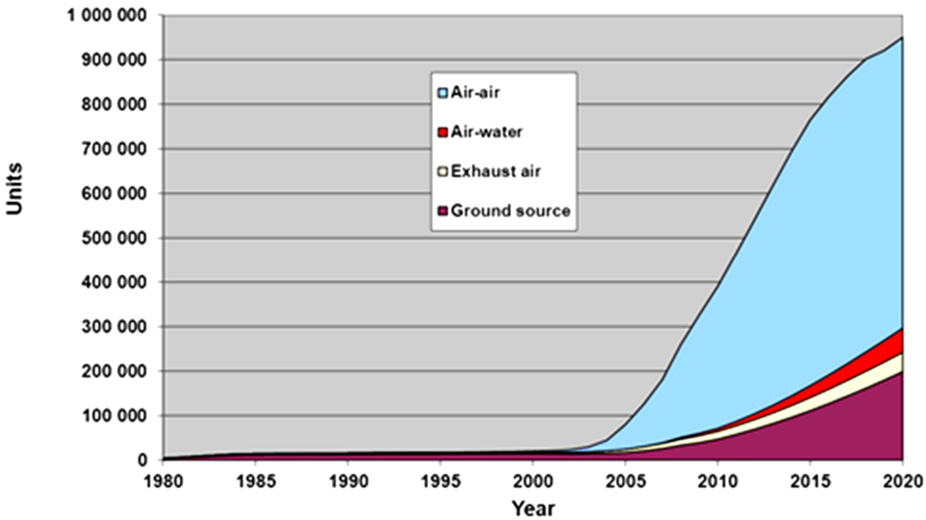


Figure 1. Scenario of the heat pump stock (Sulpu).

Table 1. Number of heat pumps in 2010 and estimated numbers in 2016 and 2020.

Year	Ground source	Air-air	Air-water	Exhaust air	Total
2010	47 390	319 501	6 326	17 533	390 750
2016	127 440	626 098	31 526	32 383	817 447
2020	199 190	653 821	54 326	43 207	950 544

Table 2. Heating capacity of heat pumps in 2010 and estimated capacities in 2016 and 2020.

Year	Ground source, kW	Air-air, kW	Air-water, kW	Exhaust air, kW	Total, MW
2010	564 415	1 520 825	73 255	58 911	2 217
2016	1 517 810	2 980 226	365 071	108 807	4 972
2020	2 372 353	3 112 188	629 095	145 176	6 259

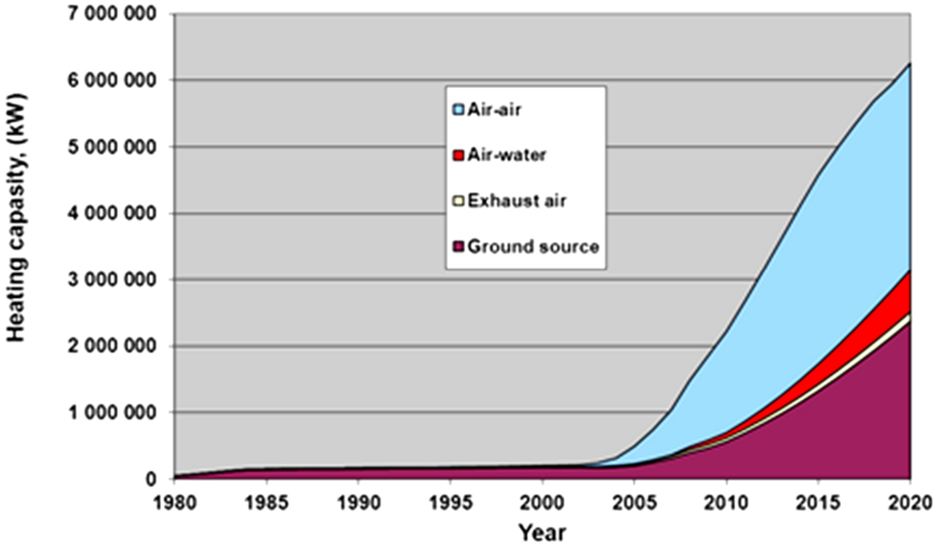


Figure 2. Scenario of the heating capacity of the heat pumps (Sulpu).

2.2 Heat pump stock (non-domestic heat pumps)

The amounts of large heat pumps used for industrial purposes have not been collected before in Finland. Finnish heat pump manufacturers were asked to give data about over 100 kW heat pumps which they have delivered during the last 10 years. Answers were received from four manufacturers. The total number of heat pumps was 95 and the average heating power of the pumps was 464 kW. At least 25 pumps were used for cooling and their average cooling power was 634 kW. The average yearly usage time was 5 525 hours and the average SPF 4.2.

The heat source was given for 25 heat pumps. Six of these 25 pumps utilized renewable energy for heat production: one from water and five from the ground. The total annual renewable heat production of these pumps was calculated with Eq. (1.1) to be around 12 GWh if they are estimated to operate with full power.

2.3 Heat pump distribution by building type

The distribution of heat pumps by building type has been estimated by the statistics of Rakennustutkimus RTS Oy. According to the obtained data based on the statistics of the year 2010 (RTS) the percentage of heat pumps in different building types were calculated (Table 3).

Table 3. Percentage distribution of heat pumps by building type in 2010.

Percentage distribution of heat pumps in 2010					
	Detached houses, %	**Attached houses, %**	**Blocks of flats, %**	**Free time residences, %**	**Total, %**
Ground source	97.90	0.21	0.04	1.85	100
Exhaust air	85.00	10.00	0.00	5.00	100
Air/air	75.76	12.12	4.55	7.58	100
Air/water	100.00	0.00	0.00	0.00	100

These distributions of heat pumps in different building types (Table 3) were used in estimating the number of heat pumps in different building types in 2010 (Table 4), 2016 (Table 5) and 2020 (Table 6). Also the building stock was made to correspond the figures of 2010.

Table 4. Number of heat pumps by building type in 2010.

Number of heat pumps in 2010					
	Detached houses	**Attached houses**	**Blocks of flats**	**Free time residences**	**Total number**
Building stock, 1000 numbers	1126	380	1221	580	3307
Ground source	46395	101	19	875	47390
Exhaust air	14903	1753	0	877	17533
Air/air	242046	38727	14523	24205	319501
Air/water	6326	0	0	0	6326
Total number of heat pumps	309670	40581	14542	25957	390750

Table 5. Number of heat pumps by building type in 2016.

Number of heat pumps in 2016					
	Detached houses	**Attached houses**	**Blocks of flats**	**Free time residences**	**Total number**
Building stock, 1000 numbers	1126	380	1221	580	3307
Ground source	124763	271	52	2354	127440
Exhaust air	27526	3238	0	1619	32383
Air/air	474317	75891	28459	47432	626098
Air/water	31526	0	0	0	31526
Total number of heat pumps	658132	79400	28511	51405	817447

Table 6. Number of heat pumps by building type in 2020.

Number of heat pumps in 2020					
	Detached houses	**Attached houses**	**Blocks of flats**	**Free time residences**	**Total number**
Building stock, 1000 numbers	1126	380	1221	580	3307
Ground source	195007	423	81	3679	199190
Exhaust air	36726	4321	0	2160	43207
Air/air	495319	79251	29719	49532	653821
Air/water	54326	0	0	0	54326
Total number of heat pumps	310	83995	29800	55372	950544

3. Simplified seasonal performance factor calculation method

3.1 Adaptation of the Eurostat method to national conditions

The simplified calculation method was used for the calculation of renewable energy production and energy saving potential of the heat pumps used in Finland for the National Renewable Energy Action Plan (NREEAP) and the National Energy Efficiency Action Plan (NEEAP). NREEAPs are roadmaps of all EU Member States for reaching its legally binding 2020 target for the share of renewable energy in their final energy consumption. NEEAPs include a an indicative energy savings target for the Member States, obligations on national public authorities as regards energy savings and energy efficient procurement, and measures to promote energy efficiency and energy services.

The simplified SPF calculation method is based on the method presented in renewable energy directive (Annex VII) by EUROSTAT, DG ENERGY and Industry. The renewable energy production E_{res} (kWh/a) of a heat pump, the portion of the heat pump energy output which is accepted as renewable, is calculated as

$$E_{res} = Q_{usable}\left(1 - \frac{1}{SPF}\right) \tag{1.1}$$

where Q_{usable} is the annual heating energy production of the heat pump (kWh/a) and SPF is the seasonal performance factor.

Q_{usable} is defined as the installed capacity of a heat pump (kW) multiplied with $Q_{usablefactor}$ (h/a). Table 7 presents the values for SPF and $Q_{usablefactor}$ suggested by EUROSTAT (DG ENERGY). These values were examined in the SPF project by VTT and new values suggested for calculation of the renewable energy production and energy saving potential of the heat pumps in Finland (Table 8).

Table 7. SPF and $Q_{usablefactor}$ values suggested by EUROSTAT.

Heat source	SPF			$Q_{usablefactor}$		
	warm climate	average climate	**cold climate**	warm climate	average climate	**cold climate**
Air	3.5	3.3	**3.0**	1 000	1 500	**2 500**
Water	3.6	3.5	**3.0**	1 150	1 600	**3 500**
Ground	3.7	3.6	**3.0**	1 150	1 600	**3 500**

Table 8. SPF and $Q_{usablefactor}$ values by VTT suggested to be used as national values in Finland.

Heat pump type	SPF, cold climate	$Q_{usablefactor}$, cold climate
Air-air	3.0	1 500
Air-water	2.0	2 000
Exhaust air	2.0	3 500
Ground	3.0	2 500

Justifications for the values in Table 8 are presented below for different heat pump types.

<u>Ground source heat pumps</u>

According to the statistics of Sulpu the average installed heating capacity of one heat pump is 11.9 kW. This capacity gives the yearly energy consumption of 41.7 MWh per one heat pump using the values of the proposal by EUROSTAT. This energy consumption was considered to be too high because most of the ground source heat pumps are installed in single family houses. The heating energy demand of a single family house ranges widely, but exceeds 30 MWh/year only in old buildings built before 1980.

In the future the multifamily installations (terraced and apartment buildings) will gain popularity but at the same time the energy demand of the building stock will decrease. A $Q_{usablefactor}$ value of 2500 instead of the EUROSTAT value of 3000 was proposed to have more realistic estimates for the renewable energy use of ground source heat pumps. This value is based on the yearly energy demand of 30 MWh per unit. The SPF value 3 proposed by EUROSTAT was considered to be at a right level compared to the literature values.

<u>Air/air heat pumps</u>

According to the statistics of Sulpu the average heating capacity of one air/air heat pump is 4.8 kW. With this heating capacity and the $Q_{usablefactor}$ value of 2500 proposed by EUROSTAT the yearly heating energy production of an air/air hest pump is 11.9 MWh. This annual heating energy production is considerably higher than the empirical (Elvari, VTT) and theoretical studies (Aalto, VTT) made for Finnish air/air heat pumps have shown. For example, the field studies made in Elvari project suggest an average energy production of 2.5 MWh/year per one air/air heat pump and the theoretical studies imply average heating energy productions between 3–7 MWh/year depending on the heating demand of the building.

The suggested national value 1500 for the $Q_{usablefactor}$ is based on the average heating capacity based on the statistics of the Finnish Heat Pump Association Sulpu and the estimated yearly energy production of 5 MWh/year which is a compromise of the experiences from field measurements and theoretical studies stated above. The SPF value 3 proposed by EOROSTAT was considered to be at the same level as the laboratory tests made by VTT and the Finnish magazine Tekniikan Maailma have shown.

Air/water heat pumps

Air/water heat pumps are rather new in the Finnish heat pump markets so there are few experiences from them. In this study the energy production of one air-water heat pump was estimated to be 24 MWh/year which is 80% of the energy production of a ground source heat pump. This assumption was proposed by the Finnish Heat Pump Association Sulpu. An energy production of 24 MWh/year together with the statistical average heating capacity of 11.6 kW, also based on the experiences by Sulpu, the national value for $Q_{usablefactor}$ is 2000. This is substantially lower that the value 3000 proposed by EUROSTAT.

The SPF value of 3 given by EUROSTAT was considered to be too high compared to the laboratory tests made by Tekniikan Maailma and experiences by Sulpu. Therefore a national SPF value of 2 is proposed in this report.

Exhaust air heat pumps

In the case of exhaust air heat pumps it is proposed to use a higher national value 3500 for $Q_{usablefactor}$ instead of the suggested 3000 by EUROSTAT. The higher value is based on experiences from literature and Sulpu which suggest the average heating energy production of one exhaust air heat pump to be around 12 MWh/year per heat pump and the statistical average heating capacity 3.4 kW.

The SPF value of 3 proposed by EUROSTAT was considered to be too high compared to the experiences by Aalto and literature, therefore a lower national value of 2 was proposed in this study.

3.2 Energy saving estimations using a simplified SPF calculation method

So called KETO-calculation was performed by VTT for the Ministry of employment and the economy in autumn 2013. KETO (kansallinen energiatehokkuusohjelma) is national implementation plan of EED directive (Directive 2012/27/EU). KETO-calculation for heat pumps covered the energy saving estimations for years 2010–2020. The number of heat pumps in this calculation were based on updated heat pump scenarios (chapter 2) of SULPU. The average nominal powers were based on SULPU's estimation from year 2010 and expert estimation on the future development until year 2020. Based on the sales statistics (2010 -> 2012) the average power of all heat pump types are increasing and number of big installations, that are not included in the statistics, are increasing. Reasonable estimate for the increasing average power is 20% from year 2010 until 2020.

Energy saving potential was calculated based on the official instructions of the European Union[1]. The use of renewable energy was calculated with Eq. (1.1) based the method presented in the annex VII of the directive 2009/28/EY. The estimated total heat produced by the heat pumps, Q_{usable}, was calculated with multiplying the equivalent running time at full load with the installed nominal power of heat pumps regarding the life time of each heat pump type:

$$Q_{usable} = H_{HP} * P_{rated} \tag{1.2}$$

where

Q_{usable}	is the estimated total heat produced by the heat pumps, GWh
H_{HP}	is the equivalent running time at full load, h
P_{rated}	is the installed nominal power of heat pumps regarding the life time of each heat pump type, GW.

The equivalent running times H_{HP} and seasonal performance factors SPF suitable for the Finnish climate are presented in Table 9. These values are defined in the official instructions[1] and therefore vary from the values presented in Table 8. The life times of the heat pumps in the cumulative calculations were: 20 years for ground source heat pumps, 15 years for air to water heat pumps and 10 years for air to air heat pumps.

[1] The Official Journal of the European Union (6.3.2013): Commission decision, given on March 1 2013, on directions for member states for calculation of the share of renewable energy produced with different heat pump technologies according to the 5th article of the European Parliament and Commission directive 2009/28/EY (information number C(2013) 1082)

Table 9. Equivalent yearly running times (HHP) and average seasonal performance factors (SPF) for Finnish climate.

Heat pump	Equivalent yearly running time H_{HP}, h	Average seasonal performance factor, SPF
Air to air	1970	2.5
Air to water	1710	2.5
Exhaust air	600	2.5
Ground source	2470	3.5

Calculated cumulative energy savings of new heat pump installations for years 2010–2020 are presented in Table 10.

Table 10. Estimated total energy saving potential (E_{RES}) of new heat pump installations.

Heat pump	Cumulative saving between 2010–2013, TWh	Cumulative saving between 2014–2016, TWh	Cumulative saving between 2017–2020, TWh	Cumulative saving between 2014–2020, TWh
Air to air	7.567	3.909	0.322	4.231
Air to water	0.381	0.576	0.417	0.993
Exhaust air	0.066	0.053	0.034	0.087
Ground source	9.464	6.788	5.409	12.197
Total	**17.478**	**11.325**	**6.182**	**17.508**

Besides new installations energy saving can also be achieved when old poorly working heat pumps are replaced by new better technology. The estimated energy saving with these replacement installations is presented in Table 11. These calculations are based on very conservative improvement estimations of the seasonal performance factors: ground source SPF from 2.5 to 3.5, air to air SPF from 2.0 to 2.5, air to water SPF from 2.0 to 2.5 and exhaust air SPF from 2.0 to 2.5.

Table 11. Estimated total energy saving potential (E_{RES}) of replacement installations.

Heat pump	**Cumulative saving between 2010–2013 TWh**	**Cumulative saving between 2014–2016 TWh**	**Cumulative saving between 2017–2020 TWh**	**Cumulative saving between 2014–2020 TWh**
Air to air	0.031	0.286	0.549	0.835
Air to water	0.000	0.000	0.023	0.023
Exhaust air	0.001	0.000	0.000	0.001
Ground source	0.026	0.007	0.015	0.022
Total	**0.063**	**0.294**	**0.587**	**0.881**

4. Detailed seasonal performance factor calculation method

The detailed SPF calculation method was developed by Aalto University. The Finnish version of the detailed calculation method is presented in Appendix 1.

4.1 Calculation principles and boundaries

The hourly calculation method presented here can be used for heating energy calculation of air-air-, air-water-, exhaust air and ground source heat pumps. The calculation method cannot be used for power dimensioning of a heat pump. Calculation can be performed also with other time steps than one hour, even though the time step used in this report is one hour.

Measured performance values for the heat pump are needed for calculation input values at least in one test point. In the calculation an on-off controlled heat pump operates always with its nominal power. Evaporator and condenser powers of on-off controlled heat pumps are assumed to change according to temperatures and COP when the performance values of the heat pump are known in one test point. The calculation is more accurate if performance values are available from several test points. The condenser power of an on-off controlled heat pump is dependent on evaporator power and COP. The true COP is directly proportional to the theoretical COP value. The heating power of all power-controlled heat pumps is defined according to the hourly heating power demand of the building.

The calculation method does not take into account the heat storage ability of the domestic service water accumulator. The calculation method assumes that the heat pump heats up both domestic service water and spaces in turns so that heating up the domestic service water is the primary function.

4.2 Calculation description

4.2.1 Air-air heat pump

The SPF calculation for an air-air heat pump is performed according to following steps:

1. Definition of following input values for the calculation:
    a. Outdoor air temperature throughout the year and the calculation time step (chapter 4.3.1)
    b. Heating power or heating energy demand for space heating (chapter 4.3.3)
    c. Inlet air temperature from the heat pump or indoor air temperature set point if the inlet air temperature is not known (chapter 4.3.4)
    d. Measured COP, heating power and temperatures at least in one test point for the heat pump (chapter 4.3.4)
    e. Lowest outdoor temperature, where the heat pump can be operated (chapter 4.3.4)
2. Calculation of space heating power demand for each time step (if not available as input value) (chapter 4.5.2)
3. Calculation of heat pump COP for each time step (chapter 4.5.4)
4. Calculation of the effect of partial power on COP for each time step if the heat pump is power controlled (chapter 4.5.5)
5. Calculation of the heating power of the heat pump for each time step (chapter 4.5.10)
6. Calculation of the space heating energy demand for each time step (chapter 4.5.13)
7. Calculation of the space heating energy which the heat pump can deliver for each time step and the annual delivered space heating energy (chapter 4.5.13)
8. Calculation of the electricity consumption of the heat pump for each time step and the annual electricity consumption (chapter 4.5.14)
9. Calculation of the heat pump SPF (chapter 4.5.16)
10. Calculation of the additional space heating demand (chapter 4.5.17)

4.2.2 Air-water heat pump

The SPF calculation for an air-water heat pump is performed according to following steps:

1. Definition of following input values for the calculation:
   a. Outdoor air temperature throughout the year and the calculation time step (chapter 4.3.1)
   b. DSW (domestic service water) heating power or heating energy demand and DSW temperature (chapter 4.3.2)
   c. Heating power or heating energy demand for space heating and heating up the ventilation inlet air (chapter 4.3.3)
   d. Maximum and minimum temperatures for inlet water, dimensioning outdoor temperature and the outdoor temperature where the inlet water temperature is equal to the minimum temperature (chapter 4.3.3)
   e. Measured COP, heating power and temperatures at least in one test point for the heat pump (chapter 4.3.4)
   f. Highest DSW heating up temperature with the heat pump without additional heating demand for DSW (chapter 4.3.4)
   g. Lowest outdoor temperature, where the heat pump can be operated (chapter 4.3.4)
2. If the heat pump is used only for DSW heating, following steps are calculated:
   a. Calculation of DSW heating demand for each time step (if not available as input value) (chapter 4.5.1)
   b. Calculation of heat pump COP for DSW heating for each time step (chapter 4.5.4)
   c. Calculation of the DSW heating power of the heat pump for each time step (chapter 4.5.6)
   d. Calculation of the effect of partial power on COP for DSW heating for each time step if the heat pump is power controlled (chapter 4.5.5)
   e. Calculation of the DSW heating time during each time step (chapter 4.5.8)
   f. Calculation of the DSW heating energy which the heat pump can deliver for each time step and the annual delivered DSW heating energy (chapter 4.5.9)

3. If the heat pump is used also for space heating and/or heating up the ventilation inlet air besides DSW heating, following steps are calculated:
    a. Calculation of heating power demand for space heating and heating up the ventilation inlet air for each time step (if not available as input value) (chapter 4.5.2)
    b. Calculation of heat pump COP for space heating and heating up the ventilation inlet air for each time step (chapter 4.5.4)
    c. Calculation of the heating power of the heat pump for space heating and heating up the ventilation inlet air for each time step (chapter 4.5.10)
    d. Calculation of the effect of partial power on COP for space heating and heating up the ventilation inlet air for each time step if the heat pump is power controlled (chapter 4.5.5)
    e. Calculation of the heating time available for space heating and heating up the ventilation inlet air during each time step step (chapter 4.5.12)
    f. Calculation of the heating energy demand of space heating and heating up the ventilation inlet air during each time step (chapter 4.5.13)
    g. Calculation of the heating energy which the heat pump can deliver for space heating and heating up the ventilation inlet air during each time step and the annual delivered heating energy deliver for space heating and heating up the ventilation inlet air (chapter 4.5.13)
4. Calculation of the electricity consumption of the heat pump for DSW heating, space heating and heating up the ventilation inlet air during each time step and the annual electricity consumption (chapter 4.5.14)
5. Calculation of, if needed, the electricity use of auxiliary devices not included in the measured COP value of the heat pump (chapter 4.5.15)
6. Calculation of the heat pump SPF (chapter 4.5.16)
7. Calculation of the additional space heating demand (chapter 4.5.17).

4.2.3 Ground source heat pump

The SPF calculation for a ground source heat pump is performed according to the same steps as for an air-water heat pump with the only difference of needing also the return temperature of the fluid from the heat collection circuit as an input value and using the fluid temperature entering the evaporator in the COP calculation.

4.2.4 Exhaust air heat pump

The SPF calculation for an exhaust air heat pump is performed according to following steps:

1. Definition of following input values for the calculation:
   a. Outdoor air temperature throughout the year if the hourly heating power demand for space heating and heating up the ventilation inlet air is not known and the calculation time step (chapter 4.3.1)
   b. DSW heating power of heating energy demand and DSW temperature (chapter 4.3.2)
   c. Calculation of heating power and heating energy demand for space heating and heating up the ventilation inlet air (chapter 4.3.3)
   d. Maximum and minimum temperatures for inlet water, dimensioning outdoor temperature and the outdoor temperature where the inlet water temperature is equal to the minimum temperature (chapter 4.3.3)
   e. Measured COP, heating power and temperatures at least in one test point for the heat pump (chapter 4.3.4)
   f. Highest DSW heating up temperature with the heat pump without additional heating demand for DSW (chapter 4.3.4)
   g. Temperature and humidity of the outlet air and the lowest temperature of the waste air (chapter 4.3.4)
2. If the heat pump is used for DSW heating, following steps are calculated:
   a. Calculation of DSW heating demand for each time step (if not available as input value) (chapter 4.5.1)
   b. Calculation of heat pump COP for DSW heating for each time step (chapter 4.5.4)
   c. Calculation of the DSW heating power of the heat pump for each time step (chapter 4.5.7)
   d. Calculation of the effect of partial power on COP for DSW heating for each time step if the heat pump is power controlled (chapter 4.5.5)
   e. Calculation of the DSW heating time during each time step (chapter 4.5.8)
   f. Calculation of the DSW heating energy which the heat pump can deliver for each time step and the annual delivered DSW heating energy (chapter 4.5.9)

3. If the heat pump is used also for space heating and/or heating up the ventilation inlet air besides DSW heating, following steps are calculated:
   a. Calculation of heating power demand for space heating and heating up the ventilation inlet air for each time step (if not available as input value) (chapter 4.5.2)
   b. Calculation of heat pump COP for space heating and heating up the ventilation inlet air for each time step (chapter 4.5.4)
   c. Calculation of the heating power of the heat pump for space heating and heating up the ventilation inlet air for each time step (chapter 4.5.11)
   d. Calculation of the effect of partial power on COP for space heating and heating up the ventilation inlet air for each time step if the heat pump is power controlled (chapter 4.5.5)
   e. Calculation of the heating time available for space heating and heating up the ventilation inlet air during each time step (chapter 4.5.12)
   f. Calculation of the heating energy demand of space heating and heating up the ventilation inlet air during each time step (chapter 4.5.13)
   g. Calculation of the heating energy which the heat pump can deliver for space heating and heating up the ventilation inlet air during each time step and the annual delivered heating energy deliver for space heating and heating up the ventilation inlet air (chapter 4.5.13)
4. Calculation of the electricity consumption of the heat pump for DSW heating, space heating and heating up the ventilation inlet air during each time step and the annual electricity consumption (chapter 4.5.14)
5. Calculation of, if needed, the electricity use of auxiliary devices not included in the measured COP value of the heat pump (chapter 4.5.15)
6. Calculation of the heat pump SPF (chapter 4.5.16)
7. Calculation of the additional space heating demand (chapter 4.5.17)

4.3 Calculation input values

4.3.1 Weather data

Following input data is needed for the calculation:

- Hourly outdoor temperature $T_{outdoor}(t)$, °C.
- Time step for calculation and weather data $t_{time\text{-}step}$, h.

As hourly weather data e.g. the annual values of the Finnish Building Code part D3 (2012) can be used. This data is available at the web pages of the Finnish

Meteorological Institute [http://ilmatieteenlaitos.fi/rakennusten-energialaskennan-testivuosi]. The calculation can be performed with also other time steps than one hour, but in this report one hour is used as the time step. The calculation time step $t_{time\text{-}step}$ must be equal to the time step of the weather data. It is worth noticing that with a longer time step the accuracy of the calculation decreases.

4.3.2 DSW heating demand

If the heat pump is used for DSW heating, following data is needed as input values:

- Hourly DSW heating power demand $\phi_{DSW}(t)$, kW.
- Annual DSW heating energy demand $Q_{heating,DSW}$, kWh/a, if the hourly heating power demand $\phi_{DSW}(t)$is not known.
- DSW temperature T_{DSW}, °C.
- Temperature of the cold service water T_{SW}, °C.

The transfer and storage heat losses are included in the DSW heating power $\phi_{DSW}(t)$ and heating energy demand $Q_{heating,DSW}$. The hourly DSW heating power demand $\phi_{DSW}(t)$ can be calculated using a dynamical simulation program or it can be estimated using the method presented in chapter 4.5.1. The annual DSW heating energy demand $Q_{heating,DSW}$ can be calculated using e.g. a method according to Part D5 (2012) of the Finnish Building Code.

If solar heat collectors are utilized in DSW heating besides the heat pump, the annual heating energy produced by the solar heat collectors and the heating power which can be utilized in DSW heating must be substituted from the annual heating energy demand $Q_{heating,DSW}$ and the hourly heating power demand $\phi_{DSW}(t)$. The hourly heating power produced with the solar heat collectors can be calculated using dynamical simulation programs such as IDA-ICE 4.5 and TRNSYS 17 taking into account the hourly DSW consumption in the simulated building.

4.3.3 DSW heating demand

Following input data concerning the heating demand of the building is needed when:

- The heat pump is used for space heating:
  - Hourly space heating power demand $\phi_{spaces}(t)$, kW.
  - Annual space heating energy demand $Q_{heating,spaces}$, kWh/a, if the hourly space heating power demand $\phi_{spaces}(t)$ is not known.
- The heat pump is used for the after-heating of the spaces and ventilation inlet air:

- The sum of the hourly after-heating power demands for space heating and ventilation inlet air heating $\phi_{spaces,ventilation}(t)$, kW.
- The sum of the hourly after-heating energy demands for space heating and ventilation inlet air heating $Q_{heating,spaces,ventilation}$, kWh/a if the hourly after-heating power demand $\phi_{spaces,ventilation}(t)$ is not known.

The heating distribution and storage heat losses are included in the heating power and heating energy demand of space heating and ventilation inlet air heating. The annual heating energy demand for space heating and ventilation inlet air heating can be calculated using e.g. a method according to Part D5 (2012) of the Finnish Building Code. The hourly heating power demand of space heating and ventilation inlet air heating is primarily calculated using a dynamical simulation program or it can be estimated using a method presented in chapter 4.5.2.

For reasons of simplification, in the following text and formulas of this report the space heating includes also the heating of the ventilation inlet air. The calculation method can anyhow be directly utilized also in cases where the heat pump heats up also the ventilation inlet air besides of spaces by using the sum of the heating energy $Q_{heating,spaces,ventilation}$ and heating power $\phi_{spaces,ventilation}$ for space heating and ventilation inlet air instead of the heating energy $Q_{heating,spaces}$ and heating power $\phi_{spaces}(t)$ for space heating.

If solar heat collectors are utilized in space heating besides the heat pump, the annual heating energy and the heating power produced by the solar heat collectors must be substituted from the annual space heating energy demand $Q_{heating,spaces}$ and the hourly space heating power demand $\phi_{spaces}(t)$. The hourly heating power and heating energy produced with the solar heat collectors can be calculated using e.g. methods presented in chapter 4.3.2.

If the heat pump is linked to a water-circulated heating distribution network, following input data are needed according to the heating distribution network:

- Dimensioning outdoor temperature Tdim, °C used for the calculation of the heating power of the building.
- Maximum temperature of the inlet water Tiw,max (°C) with the dimensioning outdoor temperature Tdim, °C.
- Minimum temperature of the inlet water Tiw,min (°C)
- The minimum outdoor temperature Toutdoor,iw,min where the inlet water temperature reaches its minimum value Tiw,min

4.3.4 Heat pump

Product data measured e.g. according to standards SFS-EN 14511-3, SFS-EN 16147 or SFS-EN 14825 are used as input data for the heat pump. Following input data is needed for the calculation:

- The measured COP_N for an on-off controlled heat pump at least in one test point.
- The measured COP_N for a power controlled heat pump with maximum power and at least in one test point representing the partial power operation.
- Maximum heating power $P_{HP,max}$ (kW) produced by an air-air, air-water or ground source heat pump in those test points where the COP_N has been measured.
- Temperatures of those test points where the COP_N has been measured.
- The maximum water heating temperature, $T_{HP,\ max}$ (°C) of the heat pump without utilization of additional heating.

The maximum water heating temperature, $T_{HP,\ max}$ (°C) of the heat pump without utilization of additional heating. The temperature levels of the test points have been defined e.g. in standard SFS-EN 14511-2. The input values mentioned before are given depending on the heat pump type at least in one test point, e.g:

- Air-air heat pump: $T_{outdoor}$ /T_{ndoor} = (+7/+20°C)
- Air-water heat pump: $T_{outdoor}$/T_{iw} = (+7/+35°C)
- Ground source heat pump: T_{fluid}/T_{iw} = (0/+35°C)
- Exhaust air heat pump: T_{indoor}/T_{iw} = (+20/35°C).

where

$T_{outdoor}$	outdoor air temperature, °C
T_{indoor}	indoor air temperature, °C
T_{iw}	temperature of the inlet water leaving the condenser and entering the heating distribution network, °C
T_{fluid}	temperature of the fluid leaving the heat collection circuit and entering the evaporator, °C.

If input data in these test points is not available, also other test points mentioned in e.g. standard SFS-EN 14511-2 can be utilized. It is worth underlining that the accuracy of the calculation increases when several test points are utilized. Following heat pump specific input data are also needed:

- Air-air heat pump:
  - o The in-blast air temperature of the heat pump T_{ib} (°C) (the temperature of the air heated flowing through the condenser) or the set point indoor air temperature of the indoor air T_{indoor} (°C), if the in-blast temperature is not known.
  - o The minimum outdoor temperature according to the recommendation of the heat pump manufacturer $T_{outdoor,min}$ (°C), where the heat pump can be operated.

- Air-water heat pump:
  - o The minimum outdoor temperature according to the recommendation of the heat pump manufacturer $T_{outdoor,min}$ (°C), where the heat pump can be operated.
- Ground source heat pump:
  - o The return temperature of the fluid from the heat collection circuit and entering the evaporator T_{fluid} (°C). If more detailed information, e.g. case-specific data calculated with a heat collection circuit dimensioning program, can be utilized, the time-dependency of the fluid temperature can be taken into account in the calculation using the hourly fluid temperature from the heat collection circuit. If more detailed information is not available the average monthly or yearly return temperature can be used.
- Exhaust air heat pump:
  - o Outlet air flow from the building Q_{outlet} (m^3/s) entering the evaporator of the heat pump.
  - o Outlet air temperature $T_{outlet}(t)$, °C. Hourly outlet air temperatures calculated using e.g. a dynamical simulation program can be used as inlet values. In other case the constant indoor air set point temperature T_{indoor} (°C) can be used as the outlet air temperature.
  - o The absolute humidity of the outlet air x_{outlet} (kg/kg). If hourly outlet air temperatures are used as an input values, also hourly values for the absolute humidity x_{outlet} must be used. Hourly absolute humidities can be calculated using e.g. dynamical simulation programs. If the absolute humidity is not known, it can be calculated e.g. using the method presented in Annex 2 (chapter 4.6) with the relative humidity and temperature of the outlet air as input values.
  - o Minimum waste air temperature after the evaporator $T_{waste,min,}$ °C according to the information given by the heat pump manufacturer.

4.4 Calculation results

Following results are given by the SPF calculation method:

- Heating energy produced by the heat pump.
- Electricity consumption of the heat pump.
- SPF of the heat pump.
- Additional heating energy demand if the heat pump cannot produce all of the heating energy demand.

4.5 Calculation

4.5.1 DSW heating power demand

Primarily the hourly DSW heating power demands $\phi_{DSW}(t)$ calculated according to hourly DSW consumptions should be used as the inlet values for the calculation. If the hourly DSW heating power demands $\phi_{DSW}(t)$ are not known, the heating power can be estimated according to the annual DSW heating energy demand $Q_{heating,DSW}$ (D5, 2012) if the heating power demand is assumed to be constant

$$\phi_{DSW} = \frac{Q_{heating,DSW}}{t_{DSW}} \tag{4.1}$$

where

t_{DSW} yearly usage time of DSW, h.

If the DSW temperature T_{DSW} is higher than the maximum water temperature $T_{HP,max}$ that the heat pump can deliver without additional heating, the portion of the additional heating is substituted from the total heating power demand calculated according to DSW consumption or the annual heating energy demand calculated with Eq. 4.1. In this case the corrected heating power demand $\phi_{DSW}(t)$ calculated with Eq. 4.2 is used as the DSW heating power demand to be produced by the heat pump.

$$\phi_{DSW}(t) = \phi_{DSW}(t)\left(1 - \frac{T_{DSW} - T_{HP,max}}{T_{DSW} - T_{SW}}\right) \tag{4.2}$$

where

T_{DSW} DSW temperature, °C
$T_{HP,max}$ maximum water temperature that the heat pump can deliver, °C
T_{SW} temperature of cold service water, °C.

4.5.2 Space heating power demand

If the hourly space heating power demand of the building $\phi_{spaces}(t)$ is not known, it can be calculated based on the annual space heating energy demand $Q_{heating,spaces}$ using the following equation

$$\phi_{spaces}(t) = \frac{Q_{heating,spaces}}{S_{Hdd}}\left(T_{Hdd} - T_{outdoor}(t)\right)$$, when THdd > Toutdoor (4.3)

$$\phi_{spaces}(t) = 0$$, when THdd ≤ Toutdoor.

where

S_{Hdd} heating degree day, Kh
T_{Hdd} indoor temperature representing the heating degree day, °C
$T_{outdoor}(t)$ hourly outdoor temperature, °C.

The space heating power demand or space heating energy demand of the equation (4.3) are based on the heating power or the energy demand of the zones which are located in the sphere of influence of the air-air heat pump.

The heating degree day S_{Hdd} can be calculated with the following equation using e.g. 17°C for the indoor temperature representing the heating degree day. In this temperature is it assumed that the indoor heat loads are enough to heat up the indoor air from 17°C to the set point temperature.

The space heating power demand or space heating energy demand of the equation (4.3) are based on the heating power or the energy demand of the zones which are located in the sphere of influence of the air-air heat pump.

The heating degree day S_{Hdd} can be calculated with the following equation using e.g. 17°C for the indoor temperature representing the heating degree day. In this temperature is it assumed that the indoor heat loads are enough to heat up the indoor air from 17°C to the set point temperature.

$$S_{Hdd} = \sum\left(T_{Hdd} - T_{outdoor}(t)\right)\cdot t_{timestep} \tag{4.4}$$

where

$T_{outdoor}(t)$ hourly outdoor temperature, °C.
$t_{timestep}$ calculation time step, h.

4.5.3 Heat distribution network temperature and in-blast temperature

The hourly inlet water temperature of the heat distribution network $T_{iw}(t)$ can be calculated with following equations.

$$T_{iw}(t) = T_{iw,max}$$, when $T_{outdoor}(t) \le T_{dim}$ (4.5)

$$T_{iw}(t) = k \cdot T_{outdoor}(t) + b$$, when $T_{outdoor}$ is between $T_{dim} < T_{outdoor}(t) < T_{outdoor,iw,min}$ (4.6)

$$T_{iw}(t) = T_{iw,min}\text{, when } T_{outdoor}(t) \geq T_{outdoor,iw,min} \tag{4.7}$$

where

$T_{iw,max}$	maximum inlet water temperature in the dimensioning outdoor temperature, °C
k	slope of the control curve, -
$T_{outdoor}(t)$	hourly outdoor temperature, °C
b	constant term of the control curve, °C
$T_{iw,min}$	mimimum inlet water temperature, °C.

The slope of the control curve can be calculated with following equation

$$k = \frac{T_{iwv,max} - T_{iw,min}}{T_{dim} - T_{outdoor,iw,min}} \tag{4.8}$$

where

$T_{iw,max}$	maximum inlet water temperature in the dimensioning outdoor temperature, °C
$T_{iw,min}$	mimimum inlet water temperature, °C
T_{dim}	dimensioning outdoor temperature for the heating of the building, °C
$T_{outdoor,iw,min}$	outdoor temperature representing the minimum inlet water temperature, °C.

The constant term of the control curve b can be calculated with the following equation

$$b = T_{iw,max} - k \cdot T_{dim} \tag{4.9}$$

where

k	slope of the control curve, -.

If the value of the inlet water temperature $T_{iw}(t)$ calculated with the equations (4.5–4.7) is higher than the maximum water temperature $T_{HP,max}$ that the heat pump can deliver without additional heating, the value of the exceeding inlet water temperatures is $T_{HP,max}$. Figure 3 presents an example about inlet water temperatures for different outdoor temperatures.

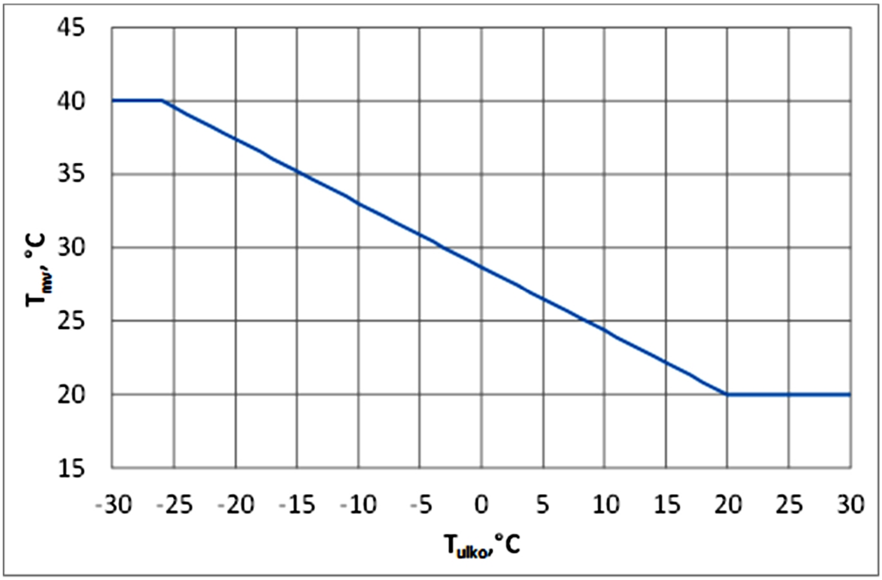


Figure 3. Example about an inlet water temperature control curve (T_{mv} = T_{iw} and T_{ulko} = $T_{outdoor}$).

The in-blast temperature T_{ib} (°C) of the air-air heat pump is the temperature of the heated-up air leaving the condenser. If the in-blast temperature is not known, it can be estimated according to the indoor air set point temperature T_{indoor}. The in-blast temperature can be estimated to be 15°C warmer that the indoor air set point temperature if more detailed knowledge is not available.

4.5.4 Heat pump COP

The hourly COP(t) of the heat pump is calculated with the following equation

$$COP(t) = f_T(t) \cdot COP_T(t) \tag{4.10}$$

where

$f_T(t)$	loss factor of the compressor, -
$COP_T(t)$	theoretical hourly COP of the heat pump, -.

The effect of partial power on the COP of a power-controlled heat pump is taken into account using the method presented in chapter 4.5.5.

The loss factor $f_T(t)$ of Eq. (4.10), which takes into account the heating process losses of the heat pump is calculated with Eq. (4.11):

$$f_T(t) = \frac{COP_N}{COP_T} \tag{4.11}$$

where

COP_N	measured heat pump COP, -
COP_T	theoretical heat pump COP, -.

If the heat pump COP_N has been measured only in one test point the loss factor is assumed to be constant throughout the whole calculation period (e.g. a year). If the COP has been measured in several test points the loss factor can be calculated for each test point with Eq. (4.11). The loss factor values $f_T(t)$ between different test points can be linearly interpolated in sections using the method presented in Annex 1 (chapter 4.6). For lower or higher temperatures than the test point temperatures the nearest measured loss factor value can be used.

The hourly theoretical $COP_T(t)$ in Eq. (4.10) is calculated with Eq. (4.12) using heating source and building heating system temperatures $T_{HSo}(t)$ and $T_{HSy}(t)$ defined for each time step. The theoretical heat pump COP_T is calculated with Eq. (4.12) using constant heating source and building heating system temperatures.

$$COP_T = \frac{T_{HSy}}{T_{HSy} - T_{HSo}} \tag{4.12}$$

where

T_{HSy}	building heating system temperature, K
T_{HSo}	heating source temperature, K.

The DSW temperature T_{DSW} is used as the building heating system temperature T_{HSy} in calculation of the COP for DSW heating. Accordingly the inlet water temperature $T_{iw}(t)$ is used as the building heating system temperature T_{HSy} for a water-circulated network and the in-blast temperature T_{ib} as the building heating system temperature T_{HSy} for an air-air heat pump (see chapter 4.5.3) when calculating the COP for space heating. If the DSW or inlet water temperature is higher than the maximum water temperature that the heat pump can deliver, $T_{HP,max}$, the maximum deliver temperature $T_{HP,max}$ is used in Eq. (4.12) as the building heating system temperature T_{HSy}.

Depending on the heat pump type following temperatures are used as the heating source temperature T_{HSy} in Eq. (4.12):

- Air-air- and air-water heat pump: Outdoor air temperature Toutdoor(t).
- Ground source heat pump: Temperature of the fluid leaving the heat collection circuit and entering the evaporator $T_{fluid}(t)$.
- Exhaust air heat pump: Minimum waste air temperature $T_{waste,min}$.

4.5.5 Heat pump COP with partial power

The effect of a partial power load on the COP of a power-controlled heat pump can be taken into account if the COP has been measured both with a maximum power and at least in one test point representing partial power.

The hourly load power ratios $\beta_{HP,DSW}(t)$ ja $\beta_{HP,spaces}(t)$ are calculated for DSW and space heating with following equations

$$\beta_{HP,lkv}(t) = \frac{\phi_{DSW}(t)}{\phi_{HP,DSW}(t)} \tag{4.13}$$

where

$\phi_{DSW}(t)$ hourly DSW heating power demand, kW (see chapter 4.5.1)

$\phi_{HP,DSW}(t)$ hourly maximum DSW heating power demand of the heat pump, kW (see Eq. 4.15).

$$\beta_{HP,spaces}(t) = \frac{\phi_{spaces}(t)}{\phi_{HP,spaces}(t)} \tag{4.14}$$

where

$\phi_{spaces}(t)$ hourly space heating power demand, kW (see chapter 4.5.2)

$\phi_{HP,spaces}(t)$ hourly maximum space heating power demand of the heat pump, kW (see Eq. 4.22)

Hourly DSW and space heating COP values $COP_{DSW}(t)$ and $COP_{spaces}(t)$ can be interpolated between known power ratios e.g. with the method presented in Annex 1 (chapter 4.6). If the COP values have been measured for e.g. power ratios 1.0 and 0.5, temperature corrections are first calculated for these COP values with Eq. (4.10) and then partial power corrections are performed by interpolating the corrected COP values between power ratios 0.5 and 1.0. If the power ratio is momentarily lower than 0.5 the COP representing the power ratio 0.5 can be used in calculation.

4.5.6 Heating power of air-water- and ground source heat pump in DSW heating

If the maximum heating power $\phi_{HP,max}$ of an on-off- or power controlled air-water- or ground source heat pump has been measured only in one test point, its hourly maximum DSW heating power $\phi_{HP,DSW}(t)$ is calculated with following equation

$$\phi_{HP,DSW}(t) = \phi_{HP,max} \frac{COP_{DSW}(t)}{COP_N} \tag{4.15}$$

where

$\phi_{HP,max}$	maximum heating power of the heat pump in test point, kW
$COP_{DSW}(t)$	hourly heat pump COP in DSW heating, -
COP_N	measured heat pump COP in test point, -.

The $COP_{DSW}(t)$ is calculated with Eq. (4.10) using the defined DSW temperature T_{DSW} as the input value. If the DSW temperature is higher than the maximum water temperature $T_{HP,max}$ that the heat pump can deliver without additional heating, the $COP_{DSW}(t)$ is calculated using the temperature $T_{HP,max}$.

The hourly heating power $\phi_{HP,DSW}(t)$ of a power-controlled heat pump is equal to the hourly DSW heating power demand $\phi_{DSW}(t)$ (see chapter 4.5.1) if $\phi_{DSW}(t)$ is smaller than $\phi_{HP,DSW}(t)$. If the DSW heating power demand $\phi_{DSW}(t)$ is larger than the DSW heating power $\phi_{HP,DSW}(t)$ calculated with Eq. (4.15), the value calculated with Eq. (4.15) is used as the hourly DSW heating power of the heat pump.

If the maximum heating power of an on-off- or power-controlled air-water- or ground source heat pump has been measured in several test points, the maximum heating power values between different test points can be linearly interpolated in sections using the method presented in Annex 1 (chapter 4.6). The interpolated values can be utilized between test points instead of the heating power calculated with Eq. (4.15). The maximum heating power in lower or higher temperatures than the test point temperatures can be calculated with Eq. (4.15) using the heating power of the nearest measured test point as the maximum heating power $\phi_{HP,max}$.

4.5.7 DSW heating power of an exhaust air heat pump

The hourly DSW heating power $\phi_{HP,DSW}(t)$ of an on-off-controlled exhaust air heat pump is calculated with following equation

$$\phi_{HP,DSW}(t) = \phi_{HPc}(t) \frac{COP_{DSW}(t)}{COP_{DSW}(t) - 1} \tag{4.16}$$

where

$\phi_{HPc}(t)$	condenser power of the heat pump, kW
$COP_{DSW}(t)$	hourly heat pump COP for DSW heating, -.

The hourly DSW heating power $\phi_{HP,DSW}(t)$ of a power-controlled exhaust air heat pump is equal to the hourly DSW heating power demand $\phi_{DSW}(t)$ (see chapter 4.5.1) if $\phi_{DSW}(t)$ is smaller than $\phi_{HP,DSW}(t)$. If the DSW heating power demand $\phi_{DSW}(t)$ is larger than the DSW heating power $\phi_{HP,DSW}(t)$ calculated with Eq. (4.16), the value calculated with Eq. (4.16) is used as the hourly DSW heating power of the heat pump.

The condenser power of the exhaust heat pump, used in Eq. (4.16), is calculated with following equation

$$\phi_{HPc}(t) = Q_{outlet}(t) \cdot \rho \left(h_1(t) - h_2(t)\right) \tag{4.17}$$

where

$Q_{outlet}(t)$	outlet air flow, m^3/s
ρ	outlet air density, kg/m^3
$h_1(t)$	outlet air enthalpy (before the condenser), kJ/kg
$h_2(t)$	waste air enthalpy (after the condenser), kJ/kg.

The outlet and waste air enthalpies can be calculated with following equation

$$h(t) = c_{p,air} \cdot T(t) + x(t) \cdot \left(L_w + c_{p,wv} \cdot T(t)\right) \tag{4.18}$$

where

$c_{p,air}$	specific heat capacity of air, kJ/kg,K
$T(t)$	outlet air or waste air temperature, °C
$x(t)$	absolute humidity of outlet air or waste air, kg/kg
L_w	evaporation heat for water in 0°C, kJ/kg
$c_{p,wv}$	specific heat capacity of water vapour, kJ/kg,K.

In Eq. (4.18) the value 1.006 kJ/kg,K can be used for the specific heat capacity of air, 1.85 kJ/kg,K for the specific heat capacity of water vapour and 2502 kJ/kg for the evaporation heat for water.

For calculation of the outlet air enthalpy the outlet air temperature is used as T(t) and the absolute humidity of outlet air is used as x(t) in Eq. (4.18). If the absolute humidity of outlet air is not known, it can be calculated by means of the relative humidity $RH_{outlet}(t)$ and temperature $T_{outlet}(t)$ of outlet air using the method presented in Annex 2 (chapter 4.6).

For calculation of the waste air enthalpy with Eq. (4.18) the minimum waste air temperature $T_{waste,min}$ according to the information given by the heat pump manufacturer can be used as T(t). The absolute humidity of waste air can be calculated using the method presented in Annex 2 (chapter 4.6) with 100% as the relative humidity RH_{waste} and T_{waste} as the temperature.

4.5.8 DSW heating time

The DSW heating time, t_{DSW}, during one time step is calculated with following equation

$$t_{DSW}(t) = t_{timestep}\,\frac{\phi_{DSW}(t)}{\phi_{HP,DSW}(t)}\text{, when } \phi_{DSW} < \phi_{HP,DSW} \tag{4.19}$$

$$t_{DSW}(t) = t_{timestep}\text{, when } \phi_{DSW} \geq \phi_{HP,DSW}$$

where

$t_{timestep}$	calculation time step, h
$\phi_{DSW}(t)$	DSW heating power demand, kW (see chapter 4.5.1)
$\phi_{HP,DSW}(t)$	DSW heating power of the heat pump, kW (see chapters 4.5.6 and 4.5.7).

4.5.9 DSW heating energy

The DSW heating energy production of the heat pump during one calculation time step is calculated with following equation

$$q_{HP,DSW}(t) = \phi_{HP,DSW}(t) \cdot t_{DSW} \tag{4.20}$$

where

$\phi_{HP,DSW}(t)$	DSW heating power of the heat pump, kW (see chapters 4.5.6 and 4.5.7)
t_{DSW}	DSW heating time by the heat pump during one time step, h (see chapter 4.5.8).

For an air-water heat pump the hourly DSW heating energy production $q_{LP,lkv}(t)$ is calculated only for those time steps where the outdoor air temperature $T_{outdoor}(t)$ is higher than the lowest utilization temperature for the heat pump, $T_{outdoor,min}$, defined as an input value.

The annual DSW heating energy production $Q_{HP,DSW}$ is the sum of hourly DSW heating energies $q_{HP,DSW}(t)$ as presented in the following equation

$$Q_{HP,DSW} = \sum q_{HP,DSW}(t). \tag{4.21}$$

4.5.10 Space heating power of an air-air, air-water and ground source heat pump

If the maximum space heating power $\phi_{HP,max}$ of an on-off- or power-controlled air-air, air-water or ground source heat pump has been measured only in one test point, the hourly maximum space heating power $\phi_{HP,spaces}(t)$ of these heat pump types is calculated with following equation

$$\phi_{HP,spaces}(t) = \phi_{HP,max} \frac{COP_{spaces}(t)}{COP_N} \tag{4.22}$$

where

$\phi_{HP,max}$	maximum heating power in the test point, kW
$COP_{spaces}(t)$	hourly heat pump COP for space heating, -
COP_N	measured heat pump COP in test point, -.

The heat pump $COP_{spaces}(t)$ is calculated with Eq. (4.10) using the hourly heat source temperature and the inlet water temperature $T_{iw}(t)$ of the heating distribution network or the in-blast temperature T_{ib} (air-air heat pump) as input values. If the value of the inlet water temperature is higher that the maximum water temperature that the heat pump can deliver without additional heating, the $COP_{spaces}(t)$ is calculated by using the temperature $T_{HP,max}$

The hourly space heating power $\phi_{LP,tilat}(t)$ of a power-controlled heat pump is equal to the hourly space heating power demand $\phi_{spaces}(t)$ (see chapter 4.5.2) if $\phi_{spaces}(t)$ is smaller than $\phi_{HP,spaces}(t)$. If the space heating power demand $\phi_{spaces}(t)$ is larger than the space heating power $\phi_{HP,spaces}(t)$ calculated with Eq. (4.22), the value calculated with Eq. (4.22) is used as the hourly space heating power of the heat pump.

If the maximum heating power $\phi_{HP,max}$ of an on-off- or power-controlled air-air-, air-water- or ground source heat pump has been measured in several test points, the maximum heating power values between different test points can be linearly interpolated in sections using the method presented in Annex 1 (chapter 4.6). The interpolated values can be utilized between the test points instead of the heating power calculated with Eq. (4.22). The maximum heating power in lower or higher temperatures than the test point temperatures can be calculated with Eq. (4.22) using the heating power of the nearest measured test point as the maximum heating power $\phi_{HP,max}$.

4.5.11 DSW heating time

The hourly space heating power $\phi_{HP,spaces}(t)$ of an on-off-controlled exhaust air heat pump is calculated with following equation

$$\phi_{HP,spaces}(t) = \phi_{HPc}(t) \frac{COP_{spaces}(t)}{COP_{spaces}(t) - 1} \tag{4.23}$$

where

$\phi_{HPc}(t)$ condenser power of the heat pump, kW
$COP_{spaces}(t)$ hourly heat pump COP for space heating, -.

The exhaust air heat pump condenser power of Eq. (4.23) is equal to the exhaust air heat pump condenser power calculated in chapter (4.5.7).

The hourly space heating power $\phi_{HP,spaces}(t)$ of a power-controlled exhaust air heat pump is equal to the hourly space heating power demand $\phi_{spaces}(t)$ (see chapter 4.5.2) if $\phi_{spaces}(t)$ is smaller than $\phi_{HP,spaces}(t)$. If the space heating power demand $\phi_{spaces}(t)$ is larger than the space heating power $\phi_{HP,spaces}(t)$ calculated with Eq. (4.23), the value calculated with Eq. (4.23) is used as the hourly space heating power of the exhaust air heat pump.

4.5.12 Space heating time

The space heating time, $t_{space,max}(t)$ during one time step is calculated with following equation

$$t_{spaces,max}(t) = t_{timestep} - t_{DSW}(t) \tag{4.24}$$

where

$t_{timestep}$ calculation time step, h
$t_{DSW}(t)$ DSW heating time during one time step, h (see chapter 4.5.8).

Eq. (4.24) can be used for a heat pump, which heats up both DSW and spaces. For a heat pump only heating up the spaces, the total length of the time step can be used for space heating.

4.5.13 Space heating energy

The space heating energy production qspaces(t) of the heat pump during one calculation time step is calculated with following equation

$$q_{spaces}(t) = \phi_{spaces}(t) \cdot t_{timestep} \tag{4.25}$$

where

$\phi_{spaces}(t)$ hourly space heating power demand, kW (see chapter 4.5.2)
$t_{timestep}$ calculation time step, h.

The maximum space heating energy $q_{HP,spaces,max}(t)$ that the heat pump can produce during one time step is calculated with following equation

$$q_{HP,spaces,max}(t) = \phi_{HP,spaces}(t) \cdot t_{spaces,max}(t) \tag{4.26}$$

where

$\phi_{HP,spaces}(t)$	space heating power, kW (see chapters 4.5.10 and 4.5.11)
$t_{spaces,max}(t)$	maximum space heating time during one time step, h (see chapter 4.5.12).

For a heat pump used only for space heating the maximum space heating time $t_{spaces,max}(t)$ is equal to the calculation time step.

The space heating energy production $q_{HP,spaces}(t)$ during one calculation time step is equal to the maximum space heating energy $q_{HP,spaces,max}(t)$ calculated with Eq. (4.26) if the space heating energy demand $q_{spaces}(t)$ (Eq. 4.25) is larger than the $q_{HP,spaces,max}(t)$ according to Eq. (4.27). In other case the space heating energy production is equal to the space heating energy demand (Eq. 4.28).

$$q_{HP,spaces}(t) = q_{HP,spaces,max}(t) \text{, when } q_{spaces} > q_{HP,spaces,max} \tag{4.27}$$

$$q_{HP,spaces}(t) = q_{spaces}(t) \text{, when } q_{spaces} \leq q_{HP,spaces,max} \tag{4.28}$$

where

$q_{HP,spaces,max}(t)$	maximum heating energy production during one calculation time step, kWh
$q_{spaces}(t)$	space heating energy demand during one calculation time step, kWh.

For an exhaust air heat pump the space heating energy production $q_{HP,spaces}(t)$ is calculated only for those time steps where the outdoor air temperature $T_{outdoor}(t)$ is higher than the lowest utilization temperature for the exhaust air heat pump, $T_{outdoor,min}$, defined as an input value.

The annual space heating energy production $Q_{HP,spaces}$ is the sum of the space heating energies produced during each time step $q_{HP,spaces}(t)$:

$$Q_{HP,spaces} = \sum q_{HP,spaces}(t) \tag{4.29}$$

4.5.14 Electricity consumption of the heat pump

The hourly electricity consumption of a heat pump $w_{HP}(t)$ including also the electricity consumption of the compressor and those auxiliary devices included in the electricity consumption of the heat pump in a test situation is calculated with the following equation

$$w_{HP}(t) = \frac{q_{HP,DSW}(t)}{COP_{DSW}(t)} + \frac{q_{HP,spaces}(t)}{COP_{spaces}(t)} \tag{4.30}$$

where

$q_{HP,DSW}(t)$	hourly DSW heating energy production, kWh
$COP_{DSW}(t)$	hourly heat pump COP for DSW heating, -
$q_{HP,spaces}(t)$	hourly space heating energy production, kWh
$COP_{spaces}(t)$	hourly heat pump COP for space heating, -.

The annual electricity consumption of the heat pump W_{HP} is the sum of the as presented in the following equation:

$$W_{HP} = \sum w_{HP.}(t) \tag{4.31}$$

4.5.15 Electricity consumption of the auxiliary devices

According to standards SFS-EN 14511-3 and SFS-EN 14825, the measured electricity consumption of the heat pump and the heat pump COP_N include the compressor electricity consumption, electricity consumption for the melting of the evaporator and a part of the electricity consumption of the auxiliary devices of the heat pump. The measured electricity consumption of the auxiliary devices includes the total electricity consumption of all control and protective devices and the electricity consumption of the fans and pumps transferring air or fluid inside of the heat pump unit. Thus the electricity consumption of the fans and pumps transferring air or fluid in ducts or pipes outside of the heat pump unit is not included in the electricity consumption measured in test conditions.

The electricity consumption of the fans and pumps, not included in the COP values measured according to the above-mentioned standards, is separately taken into account in the electricity consumption of the auxiliary devices, W_{aux}, depending on the heat pump type. The electricity consumption of the fans of the exhaust air heat pump is calculated into the electricity consumption of the auxiliary devices. Accordingly the pumping electricity consumption of the ground source heat pump heat collection circuit is taken into account to the extent of the pipe network outside of the heat pump unit. However, the pumping electricity consumption of the heat distribution network is not taken into account in the electricity consumption of the auxiliary devices of the heat pump, but it is instead taken into account in the electricity consumption calculation of the auxiliary devices of the heating distribution system e.g. according to part D5 (2012) of the Finnish Building Code.

The electricity consumption of the auxiliary devices of the heat pump, W_{aux}, not included in the electricity consumption of the heat pump and the measured COP_N values can be calculated with the following equation

$$W_{aux} = P_{aux}\Delta t \tag{4.32}$$

where

P_{aux}	the electricity consumption of the auxiliary devices of the heat pump not included in the measured heat pump COP value, kW
Δt	usage time of the auxiliary devices during the calculation period, h.

The electricity power P_{aux} of the auxiliary devices of the heat pump is calculated with the following equation

$$P_{aux} = \frac{Q \cdot \Delta P_e}{\eta} \tag{4.33}$$

where

Q	the nominal flow of the air or fluid Q, m³/s
ΔP_e	the static pressure loss of the ductwork or pipework outside the heat pump unit, Pa
η	the fan or pump efficiency rate, -.

4.5.16 Heat pump SPF

The SPF of the heat pump is defined according to following equation

$$SPF = \frac{Q_{HP,DSW} + Q_{HP,spaces}}{W_{HP} + W_{aux}} \tag{4.34}$$

where

$Q_{HP,DSW}$	annual DSW heating energy production of the heat pump, kWh (chapter 4.5.9)
$Q_{HP,spaces}$	annual space heating energy production of the heat pump, kWh (chapter 4.5.13)
W_{HP}	annual electricity consumption of the heat pump, kWh (chapter 4.5.14)
W_{aux}	annual electricity consumption of the auxiliary devices of the heat pump, kWh (chapter 4.5.15).

4.5.17 Additional heating energy

If the heat pump is not able to produce all the space and DSW heating energy demand, additional heating is needed. This additional heating energy can be produced e.g. by means of an electric resistance inside of the hot-water tank or by using another heating system. The additional heating energy demand is calculated with the following equation

$$Q_{additional} = Q_{heating,DSW} + Q_{heating,spaces} - Q_{HP,DSW} - Q_{HP,spaces} \quad (4.35)$$

where

$Q_{heating,\ DSW}$	DSW heating energy demand, kWh (see chapter 4.3.2)
$Q_{heating,\ spaces}$	space heating energy demand, kWh (see chapter 4.3.3)
$Q_{HP,\ DSW}$	DSW heating energy production of the heat pump, kWh (see chapter 4.5.9)
$Q_{HP,\ spaces}$	space heating energy production of the heat pump, kWh (see chapter 4.5.13).

4.6 Annexes for the detailed calculation method

4.6.1 Annex 1

The value of the variable X depending on the function A(X) can be linearly interpolated between two known points X_i and X_{i+1} when the values of the variable a known in the points $A(X_i)$ and $A(X_{i+1})$. The values of the variable A(X) between these two points can be calculated with the following equation (L1):

$$A(X) = A(X_i) + \frac{A(X_{i+1}) - A(X_i)}{X_{i+1} - X_i}(X - X_i) \quad (L1)$$

where

$A(X_i)$	value of function A with the variable X in point i
$A(X_{i+1})$	value of function A with the variable X in point i+1
X_i	value of the variable X in point i
X_{i+1}	value of the variable X in point i+1.

4.6.2 Annex 2

The absolute humidity of air can be calculated by means of the relative humidity of the air and the air temperature with following equations.

The partial water vapour pressure of air ph (t) can be calculated with the following equation

$$p_h(t) = \frac{RH(t) \cdot p_{hs}(t)}{100} \qquad \text{(L2)}$$

where

RH(t)	relative humidity of air, %
$p_{wvs}(t)$	saturation pressure of water vapour, kPa.

The saturation pressure of water vapour $P_{wvs}(t)$ of Eq. (L2) can be estimated e.g. with the following equation

$$p_{wvs}(t) = \frac{\exp\left(77.345 + 0.0057 \cdot T(t) - \dfrac{7235}{T(t)}\right)}{1000 \cdot T(t)^{8.2}} \qquad \text{(L3)}$$

where

T	air temperature, K.

The absolute humidity of air x(t) can be calculated with the following equation

$$x(t) = 0.622 \frac{p_{wv}(t)}{p - p_{wv}(t)} \qquad \text{(L4)}$$

where

$p_{wv}(t)$	partial pressure of the water vapour, kPa
p	total air pressure, kPa.

The normal pressure 101.3 kPa of air can be used as the total air pressure in Eq. (L4).

4.7 Comparing the detailed calculation method with heat pump manufacturers estimations

The detailed SPF calculation method was validated and further improved by VTT utilizing an excel application. In the future it would be beneficial to further validate the method by comparing the results with real measurement values from relevant case studies.

Heat pump manufacturers operating in Finland were asked to dimension heat pumps for two case studies to compare these dimensioning results to the results obtained with the detailed SPF calculation method:

- standard building type B representing a detached house built between 1960–70
- standard building type D3 representing a passive house level detached house.

More information about these houses is given in chapter 5. Five heat pump manufacturers operating in Finland were asked to dimension heat pumps for the heat pump types belonging to their sortiment for both houses. The weather file to be used was Jyväskylä in middle Finland. The heating demands, air temperatures and temperatures of the heating distribution system were given as background information for the dimensioning. Dimensioning data was received from two heat pump manufacturers. Tables 12–15 show this data compared with values calculated with the detailed SPF calculated method.

Table 12. Results for standard building type B, ground source heat pump.

	Heat pump manufacturer 1	**Heat pump manufacturer 2**	**SPF-calculation method**
Energy production of the heat pump, kWh/a	30 184	26 655	30 100
Additional heating energy demand, kWh	300	845	500
Electricity use of the heat pump, kWh/a	10 780	8 736	7 900
SPF of the heat pump	2.8	3.05	3.8

Table 13. Results for standard building type B, air to water heat pump.

	Heat pump manufacturer 2	**SPF-calculation method**
Energy production of the heat pump, kWh/a	22 985	28 400
Additional heating energy demand, kWh	4 515	2 200
Electricity use of the heat pump, kWh/a	9 256	10 600
SPF of the heat pump	2.48	2.7

Table 14. Results for standard building type D3, ground source heat pump.

	Heat pump manufacturer 1	**Heat pump manufacturer 2**	**SPF-calculation method**
Energy production of the heat pump, kWh/a	8 900	5 750	9 000
Additional heating energy demand, kWh	0	150	0
Electricity use of the heat pump, kWh/a	2 781	2 005	2 300
SPF of the heat pump	3.2	2.87	3.9

Table 15. Results for standard building type D3, air to water heat pump.

	Heat pump manufacturer 2	**SPF-calculation method**
Energy production of the heat pump, kWh/a	5 521	8 800
Additional heating energy demand, kWh	379	100
Electricity use of the heat pump, kWh/a	2 335	2 900
SPF of the heat pump	2.36	3.0

5. Standard building types

5.1 Classification and structures

The energy use of the Finnish building stock was estimated using standard building types further adapted to different decades: a detached house, an apartment building, an office building and a summer cottage. The building types were based on the building types utilized in the “Sustainable Energy” project by Aalto University (Figure 4 – Figure 6).

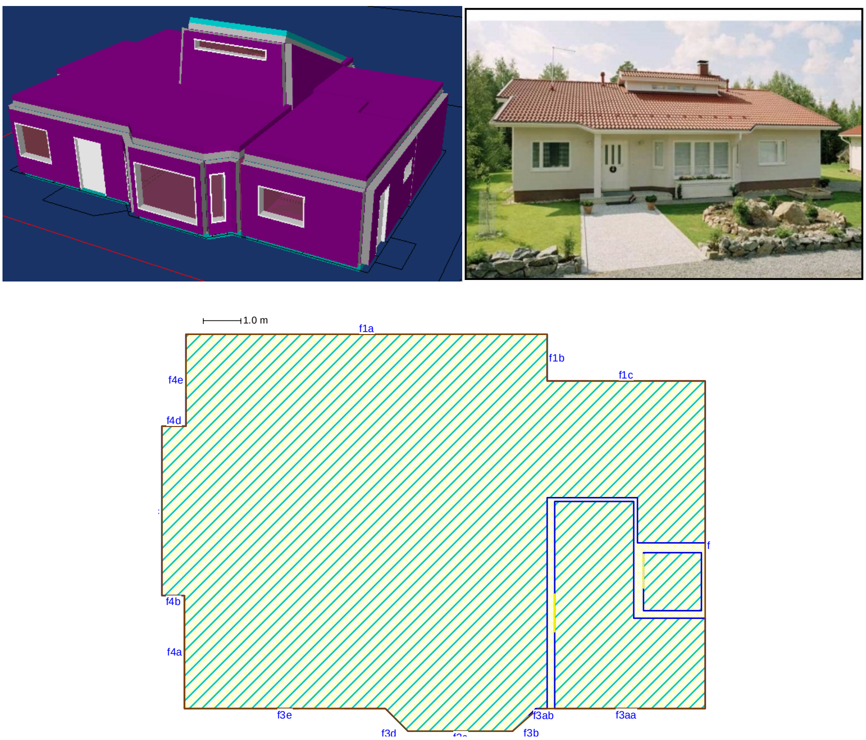


Figure 4. Detached house and recreational cottage building.

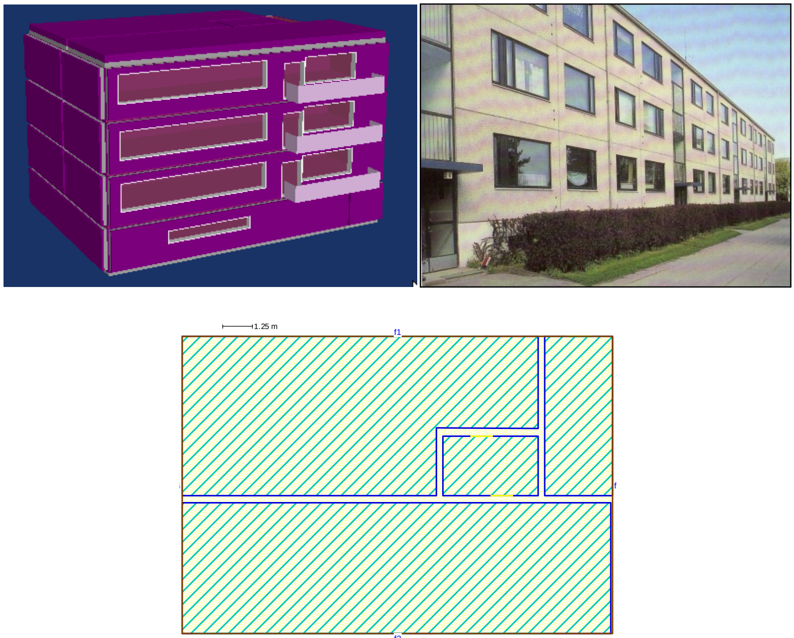


Figure 5. Apartment house.

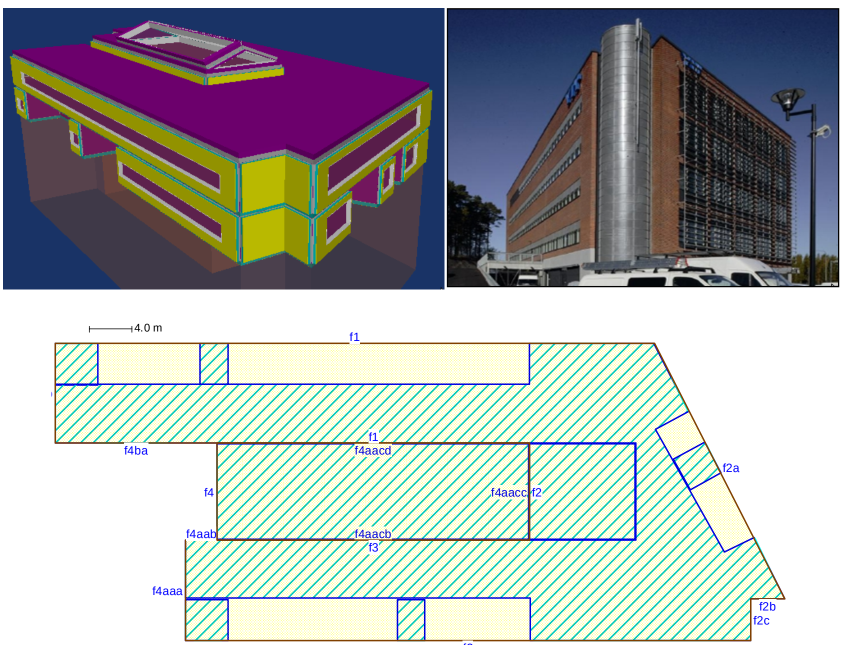


Figure 6. Office building.

The living areas of the standard building types were following: detached house 134 m^2, apartment house 814 m^2 and cottage 134 m^2. The net area of the office building was 2 695 m^2.

The standard building types were further divided to subgroups A, B, C1, C2, D1, D2 and D3 representing the construction styles and building energy use of different decades (Table 16). The specific parameters for different subgroups were selected based on various previous studies. These parameters are presented in following tables.

Table 16. Subgroups of standard building types.

Subgroup	**Building year (and energy demand)**
A	before 1960
B	1960–1979
C1	1980–2000
C2	2001–2010
D1	after 2010 (energy demand fulfilling Finnish building code 2012 Part D5)
D2	after 2010 (low-energy building)
D3	after 2010 (very low-energy building, passive house)

Table 17. Air-tightness of the building envelope n50, 1/h.

Sub-group	**Standard building type**			
	Detached house	Apartment house	Office building	Cottage
A	*	*	*	*
B	*	2.3 [a]	2.3 [a]	*
C1	4.0 [d]	1.0	1.5	7.9 [b]
C2	3.5 [d]	0.9 [e]	0.9 [f]	5.8 [c]
D1	2.0	0.7	0.5	5.8 [c]
D2	0.8	0.6	0.5	0.8
D3	0.6	0.6	0.5	0.6

* For houses with natural ventilation the air-leakage through the envelope is included in the air-change rate. Sources: [a] Polvinen et al., 1983, [b] Dyhr, 1993, [c] Vinha et al., 2009, [d] Vinha et al., 2009 and Polvinen et al., 1983, [e] Vinha et al., 2009, [f] Suomela, 2010 and Eskola et al., 2009.

Table 18. Heat loss values (U-values) of the building structures, W/m2K. OW = outer wall, UF = upper floor, BF = base floor, W = window.

Sub-group	Standard building type			
	Detached house	Apartment house	Office building	Cottage
A	OW = 0.69 [a] UF = 0.41[a] BF = 0.48 W = 2.2[a]	OW = 0.83[a] UF = 0.42[a] BF = 0.48 W = 2.2[a]	OW = 0.83[a] UF = 0.42[a] BF = 0.48 W = 2.2[a]	as detached house of A
B	OW = 0.42[a] UF = 0.24[a] BF = 0.48 W = 2.2[a]	OW = 0.47[a] UF = 0.29[a] BF = 0.48 W = 2.2[a]	OW = 0.47[a] UF = 0.29[a] BF = 0.48 W = 2.2[a]	as detached house of A
C1	OW = 0.28[b] UF = 0.22[b] BF = 0.36[b] W = 1.6[a]	OW = 0.28[b] UF = 0.22[b] BF = 0.36[b] W = 1.6[a]	OW = 0.28[b] UF = 0.22[b] BF = 0.36[b] W = 1.6[a]	as detached house of B
C2	OW = 0.25[c] UF = 0.16[c] BF = 0.25[c] W = 1.4[c]	OW = 0.25[c] UF = 0.16[c] BF = 0.25[c] W = 1.4[c]	OW = 0.25[c] UF = 0.16[c] BF = 0.25[c] W = 1.4[c]	as detached house of C1
D1	OW = 0.17[d] UF = 0.09[d] BF = 0.16[d] W = 1.0[d]	OW = 0.17[d] UF = 0.09[d] BF = 0.16[d] W = 1.0[d]	OW = 0.17[d] UF = 0.09[d] BF = 0.16[d] W = 1.0[d]	according to Finnish building code, part C3 (2010), log wall U-value 0.4
D2	OW = 0.14, UF = 0.08, BF = 0.12, W = 0.9[e]			as detached house of D2
D3	OW = 0.08 UF = 0.07 BF = 0.09, W = 0.7[f]			as detached house of D3

Sources: [a] Nykänen & Heljo, 1985, [b] C3, 1985, [c] C3, 2003, [d] C3, 2010, [e] Peuhkuri & Pedersen, 2010, [f] Nieminen et al., 2007.

Table 19. Set indoor temperatures, °C.

Sub-group	Standard building type			
	Detached house [a]	Apartment house	Office building	Cottage
A	21.0	22.0 [b]	21.5	21.0
B	21.0	22.0 [b]	21.5	21.0
C1	21.0	22.0 [b]	21.5	21.0
C2	21.0	21.5 [b]	21.5	21.0
D1, D2, D3	21.0	21.0 [c]	21.5	21.0

[a] bathroom and sauna set temperatures 21 °C, [b] cellar and staircase set temperatures 19.0 °C, WC and bathroom set temperatures 23 °C, [c] cellar and staircase set temperatures 17.0 °C, WC and bathroom set temperatures 23 °C.

Table 20. Ventilation systems. NV = natural ventilation, ME = mechanical exhaust ventilation, MSE = mechanical suppy and exhaust ventilation, HR = heat recovery (yearly efficiency rate).

Sub-group	Standard building type			
	Detached house	Apartment house	Office building	Cottage
A	NV	NV	NV	NV
B	NV	ME	ME	NV
C1	ME	ME	MSE + HR (50%)	ME
C2	MSE + HR (60%)	MSE + HR (60%)	MSE + HR (80%)	ME
D1	MSE + HR (60%)	MSE + HR (60%)	MSE + HR (80%)	MSE + HR (60%)
D2	MSE + HR (80%)	MSE + HR (80%)	MSE + HR (80%)	MSE + HR (80%)
D3	MSE + HR (85%)	MSE + HR (85%)	MSE + HR (85%)	MSE + HR (80%)

Table 21. Air-change rate, 1/h.

Sub-group	Standard building type			
	Detached house	Apartment house	Office building	Cottage
A	0.41 [a]	0.62 [a]	0.62 [a]	0.41*
B	0.41 [a]	0.43 [b]	0.43 [b]	0.41*
C1	0.46 [a]	0.5 [d]	0.5 [d]	0.46*
C2	0.40 [c]	0.56 [c]	0.5 [e]	0.40*
D1	0.5 [f]	0.5 [f]	0.5 [f]	0.5*
D2	0.5 [f]	0.5[f]	0.5 [f]	0.5*
D3	0.5[f]	0.5[f]	0.5 [f]	0.5*

* During usage time, in other times only air-leakage through the envelope. Sources: [a] Ruotsalainen, 1992, [b] Dyhr, 1993, [c] Vinha et al., 2005 and Vinha et al., 2009, [d] Finnish building code: part D2, 1987, [e] Finnish building code: part D2, 2003, [f] Finnish building code: part D2, 2010.

Table 22. Warm service water consumption.

Sub-group	Standard building type			
	Detached house	Apartment house	Office building	Cottage
A	42 dm^3/ pers,day [a]	64 dm^3/ pers,day [b]	100 dm^3/rm^2, a [c]	According to the usage profile
B	42 dm^3/ pers,day [a]	62 dm^3/ pers,day [b]	— II —	— II —
C1	42 dm^3/ pers,day [a]	59.2 dm^3/ pers,day [b]	— II —	— II —
C2	42 dm^3/ pers,day [a]	57.6 dm^3/ pers,day [b]	— II —	— II —
D1, D2, D3	42 dm^3/ pers,day [a]	56 dm^3/ pers,day [b]	— II —	— II —

Sources: [a] Motiva, 2009, [b] Virta & Pylsy, 2011, [c] Finnish Building Code, Part D5, 2007.

The lighting and device electricity use of the different standard building types was estimated according to the Finnish Building Code, part D3 (2012).

The electricity use of the following systems are based on the the Finnish Building Code, part D5 (2012). The electricity use of courtyard lighting was 2 kWh/m^2 for detached houses, apartment houses and office buildings. The parking place heating was 150 kWh/parking place for detached houses, apartment houses and office buildings. The annual electricity use of the elevator was 23 kWh/resident for the apartment house and 2 000 kWh/elevator for the office building with 4 elevators. The electricity use for sauna heating is presented in Table 23. The apartment house had 8 flats with 3 residents per a flat. Saunas were heated once a week.

Table 23. Electricity use for sauna heating.

Sub group	Standard building type			
	Detached house	Apartment house	Office building	Cottage
A	Apartment sauna 8 kWh/ one heating	Building sauna 410 kWh/flat,a	-	-
B	Apartment sauna 8 kWh/ one heating	Building sauna 410 kWh/flat,a	-	-
C1	Apartment sauna 8 kWh/ one heating	Apartment sauna 8 kWh/ one heating	-	-
C2	Apartment sauna 8 kWh/ one heating	Apartment sauna 8 kWh/ one heating	-	-
D1, D2, D3	Apartment sauna 8 kWh/ one heating	Apartment sauna 8 kWh/ one heating	-	-

Source: Finnish Building Code, Part D5 (2007).

5.2 Energy demands of standard building types

The energy demands of the different standard building types and subgroups were simulated using IDA ICE dynamical simulation program with the test weather data 2012 of Jyväskylä, Central Finland (Kalamees et al., 2012). The heating and cooling energy use of the detached house is presented in Table 24 and the electricity use in Table 25. For the apartment house the heating and cooling energy use is presented in Table 26 and the electricity use in Table 27. The heating and cooling energy use of the office building is presented in Table 28 and the electricity use in Table 29. For the cottage the heating and cooling energy use is presented in Table 30 and the electricity use in Table 31.

Table 24. Detached house: heating and cooling demand.

Sub-group	Dimensioning power, W/m²		Heating and cooling energy net demand, kWh/m²			
	Space heating	Inlet air heating	Space heating	Space cooling	Inlet air heating	Warm service water heating
A	96	0	242	0	0	21
B	78	0	189	0	0	21
C1	66	0	157	0	0	21
C2	45	8	98	0	10	21
D1	34	11	68	2	9	21
D2	30	8	53	3	2	21
D3	24	8	38	4	2	21

Table 25. Detached house: specific electricity demand, kWh/m^2.

Sub-group	Device electricity	Lighting	Courtyard lighting	Fan electricity	Parking place electricity	Sauna
A	23	7	2	0	2	3
B	23	7	2	0	2	3
C1	23	7	2	2	2	3
C2	23	7	2	5	2	3
D1	23	7	2	6	2	3
D2	23	7	2	6	2	3
D3	23	7	2	6	2	3

Table 26. Apartment house: heating and cooling demand.

Sub-group	Dimensioning power, W/m²		Heating/cooling energy net demand, kWh/m²			
	Space heating	Inlet air heating	Space heating	Space cooling	Inlet air heating	Warm service water heating
A	87	0	200	0	0	49
B	60	0	125	0	0	47
C1	39	0	51	0	0	45
C2	24	16	22	0	17	44
D1	18	15	11	0	15	43
D2	17	12	8	0	4	43
D3	14	12	4	0	3	43

Table 27. Apartment house: specific electricity demand, kWh/m^2.

Sub-group	Devices	Lighting	Courtyard lighting	Fans	Parking places	Elevator	Saunas
A	30	10	2	0	2	2	0.1
B	30	10	2	1	2	2	0.1
C1	30	10	2	1	2	2	0.1
C2	30	10	2	7	2	2	0.5
D1	30	10	2	7	2	2	0.5
D2	30	10	2	7	2	2	0.5
D3	30	10	2	7	2	2	0.5

Table 28. Office building: heating and cooling demand.

Sub-group	Dimensioning power, W/m^2		Heating/cooling energy net demand, kWh/m^2			
	Space heating	Inlet air heating	Space heating	Space cooling	Inlet air heating	Warm service water heating
A	96	0	232	0	0	6
B	67	0	135	0	0	6
C1	47	54	105	12	27	6
C2	44	45	52	20	5	6
D1	36	47	41	16	6	6
D2	32	47	33	23	6	6
D3	29	47	25	27	5	6

Table 29. Office building: specific electricity demand, kWh/m^2.

Sub-group	Devices	Lighting	Courtyard lighting	Fans	Parking places	Elevators
A	30	22	2	0	9	3
B	30	22	2	2	9	3
C1	30	22	2	12	9	3
C2	30	22	2	12	9	3
D1	30	22	2	13	9	3
D2	30	22	2	14	9	3
D3	30	22	2	13	9	3

Table 30. Cottage: heating and cooling demand.

Sub-group	Dimensioning power, W/m^2		Heating/cooling energy net demand, kWh/m^2			
	Space heating	Inlet air heating	Space heating	Space cooling	Inlet air heating	Warm service water heating
A	108	0	94	0	0	11
B	108	0	91	0	0	11
C1	88	0	63	0	0	11
C2	88	0	61	0	0	11
D1	51	13	39	0	1	11
D2	33	10	22	0	0	11
D3	27	10	17	0	0	11

Table 31. Cottage: specific electricity demand, kWh/m^2.

Subgroup	Devices	Lighting	Pumps and fans
A	30	22	0
B	30	22	2
C1	30	22	12
C2	30	22	12
D1	30	22	13
D2	30	22	14
D3	30	22	13

5.3 Energy use in the Finnish building stock with standard building types

The cumulative energy consumption of the standard building types was calculated based on the modelled development of the building stock using the REMA model developed at VTT. The model is described here briefly, a more detailed description of the model is available in Tuominen et al. (2014). The simulated energy demand results of each standard building type and subgroup were used as an input for the REMA model to calculate the total energy consumption of the building stock in each year, taking into consideration the estimated changes in the future development of the building stock. The building stock for year 2010 is presented in Table 32.

Table 32. Modelled size of building stock in 2010 according to the REMA model, in millions of m^2. Inconsistences in totals are due to rounding.

Construction year	Detached homes	Apartment buildings	Office buildings	Cottages
1980–2010	57	49	47	5
1960–1979	40	51	30	6
–1960	50	19	24	10
Total	147	119	100	21

The model uses the built area of each building type to multiply the specific energy demands presented in section 5.2 to achieve an estimate of annual cumulative consumption, for each year, producing a time series. This calculation allows the estimation of the consumption of delivered energy throughout the building stock. However, the REMA model also contains a simplified model of energy production, which allows the estimation of primary energy consumption and CO_2-emissions for various scenarios.

Besides national building statistics, the forecast development of the building stock is also based on previous VTT predictions of future development of the stock, which are further based on the theory of Rank Bo about the effects of economic and social factors on the building stock. For new construction, the estimates are largely based on long term observations and, for residential construction, also on statistical population projections. The projected figures for new construction, as well as the removal of old buildings from the stock, vary depending on the type of building and the time period in question. The beginning of the modelling was set in the year 2010 to allow comparing the modelled results to statistics and previous studies. Figure 7 shows the results of the REMA modelling compared with statistics and selected previous studies. The modelled results fit reasonably well within the variation present in the literature.

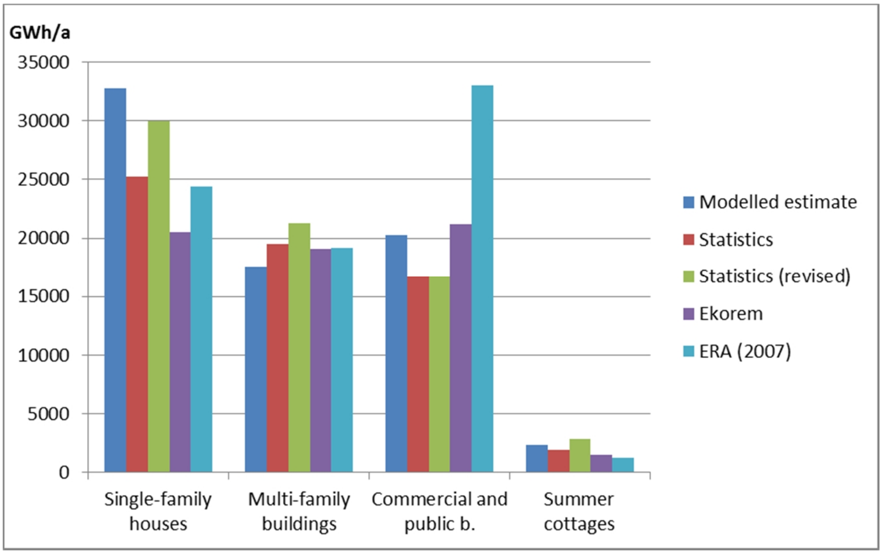


Figure 7. Calculated heating energy consumption of the standard building types compared to previous estimations from Statistics, Ekorem model and ERA (2007).

6. Energy saving potential and renewable energy use of the heat pumps in Finland

The method used to estimate the energy saving and renewable energy use potentials of the heat pumps on the Finnish building stock was based on the description of the building stock as standard building types and on the calculation of the thermal characteristics of each heat pump type on these typical buildings. The descriptions and energy consumption calculations of the type buildings are presented in chapter 4.

In this chapter the heat pump characteristics, calculated energy savings and use of renewable energy are presented for the standard building types of detached house, apartment house and cottage. Finally, the energy saving potential and reduction of emissions of heat pumps on the Finnish building stock is estimated.

6.1 Energy use of the standard type buildings with heat pumps

To estimate the energy saving and renewable energy potential of heat pumps on the Finnish building stock the behaviour of different heat pumps on the type buildings, described in chapter 5, were modelled. Seasonal performance factors for different heat pump types were calculated with the detailed SPF calculation method developed in this project (chapter 4). The calculated heat pump types were:

1. ground source heat pump
2. air to water heat pump
3. air to air heat pump
4. exhaust air heat pump.

Following assumptions were made in the calculations:

- individual characteristics of different heat pumps represents the best available technology of today
- dimensioning of the heat pumps was chosen to be both realistic and yet to deliver a close to maximal heat production
- temperature of the heating system was chosen to represent the prevailing system and thermal dimensioning values of each type building.

Energy consumption simulations and heat pump calculations were carried out with Jyväskylä[2] weather data. Heat pump calculations were performed for three standard building types: detached house, apartment house and cottage.

6.1.1 Detached houses

Ground source heat pump

The ground source heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology.

Table 33. Ground source heat pump in detached house: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	0 ºC	35 ºC	5.0
hot water production	0 ºC	52 ºC	2.7

The temperature of the ground source was modelled as a cosine function with yearly average value of 0 ºC and peak-to-peak amplitude of 2 ºC (Figure 8). The warmest month was presupposed to be August.

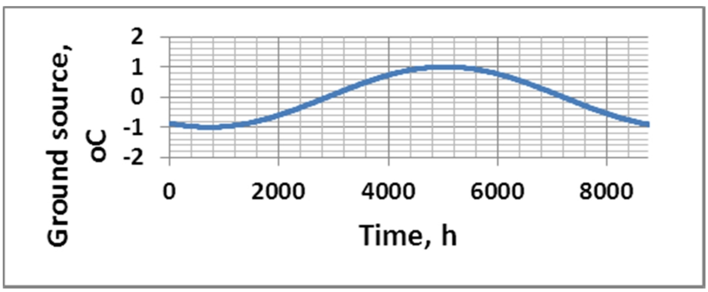


Figure 8. Ground source temperature over one year period.

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC.

[2] http://www.ym.fi/fi-FI/Maankaytto_ja_rakentaminen/Lainsaadanto_ja_ohjeet/Rakentamismaarayskokoelma

The heating power varied from 14 kW (building type A) to 7 kW (building type D3) at declared operation temperatures of 0 ºC / +35 ºC. The results of the yearly calculation are presented in Table 34.

Table 34. Ground source heat pump in detached house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A(1	37.5	9.7	28.1	3.9	0.8
B(1	30.1	7.9	22.4	3.8	0.5
C1(1	25.9	5.6	20.5	4.6	0.0
C2(2	18.9	4.1	14.9	4.6	0.0
D1(3	14.6	3.4	11.3	4.3	0.0
D2(4	11.1	2.6	8.6	4.2	0.0
D3(5	9.0	2.3	6.8	3.9	0.0

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating,(4Low energy building, floor heating, (7Passive house, floor heating

Air to water heat pump

Air to water heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology. The outdoor temperature used was based on the weather data for Jyväskylä.

Table 35. Air to water heat pump in detached house: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	7 ºC	35 ºC	4.2
hot water production	7 ºC	47 ºC	3.3

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power varied from 12 kW (building type A) to 6 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. The results of the yearly calculation are presented in Table 36.

Table 36. Air to water heat pump in detached house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A(1	33.7	12.2	21.5	2.8	4.6
B(1	28.4	10.6	17.9	2.7	2.2
C1(1	24.0	9.0	15.1	2.7	1.9
C2(2	18.2	5.6	12.7	3.3	0.7
D1(3	14.0	5.1	9.1	2.8	0.5
D2(4	10.9	3.5	7.5	3.1	0.3
D3(5	8.8	2.9	6.0	3.0	0.1

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating, (4Low energy building, floor heating , (5Passive house, floor heating

Air to air heat pump

Air to air heat pumps were modelled with three performance points with different loads. The performance factors were chosen to represent the best practice of the available technology.

Table 37. Air to air heat pump in detached house: performance points.

Load	Temperature of heat source	Temperature of heating	COP
100%	7 ºC	35 ºC	3.6
75%	7 ºC	35 ºC	4.1
50%	7 ºC	35 ºC	5.4

The outdoor temperature was based on the weather data for Jyväskylä[3]. The heating power varied from 8 kW (building type A) to 3 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. It was presumed that the effective floor area of the heat pump is 50% of the total floor area and that the air heat pump raises the indoor air temperature by 2 ºC of the working area. The results of the yearly calculation are presented in Table 38.

3 http://www.ym.fi/fi-FI/Maankaytto_ja_rakentaminen/Lainsaadanto_ja_ohjeet/Rakentamismaarayskokoelma

Table 38. Air to air heat pump in detached house: results.

Building type	Heat produced by heat pump for space heating, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use[1], MWh/a	SPF	Back-up heat required, MWh/a
A	18.6	5.6	11.5	3.3	19.7
B	14.4	4.4	8.8	3.3	16.2
C1	12.2	3.7	7.6	3.3	13.8
C2	7.7	2.3	4.9	3.3	11.2
D1	5.2	1.6	3.3	3.3	9.3
D2	4.1	1.3	2.6	3.2	7.0
D3	3.0	1.0	1.9	3.1	6.0

[1] The renewable energy is the virtual value i.e. the increased energy consumption of higher room air temperature has been taken into account.

Exhaust air heat pump

Exhaust air heat pumps were modelled with three performance points: 1) heating at full power, 2) heating at 46% power and 3) hot water production. The performance factors were chosen to represent the best practice of the available technology.

Table 39. Exhaust air heat pump in detached house: operation modes.

Operation mode	Temperature of exhaust air	Temperature of heating	COP
heating, 100% load	-9 ºC	35 ºC	3.15
heating, 46% load	-9 ºC	35 ºC	4.7
hot water production	-9 ºC	50 ºC	2.5

The humidity of the indoor air was presumed to be the same as the outdoor air humidity. Weather data of Jyväskylä was used as the climate data. The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power of the exhaust air heat pump is determined by the exhaust air flow rate and it was 2,8 kW for each case at declared operation temperatures of -9 ºC / +35 ºC. The exhaust air flow rate was 42 l/s. The results of the yearly calculation are presented in Table 40.

Table 40. Exhaust air heat pump in detached house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Energy saving / Renewable energy[1] MWh/a	SPF	Back-up heat required, MWh/a
A	17.5	6.4	11.4 / 4.2	2.8	20.7
B	16.7	6.1	10.8 / 3.7	2.7	13.9
C1	15.1	4.8	10.5 / 3.3	3.1	10.9
C2	14.4	4.4	10.2 / 3.1	3.2	8.2
D1	13.5	4.2	9.5 / 2.3	3.3	4.9
D2	12.3	3.7	8.8 / 2.0	3.3	3.0
D3	11.2	3.4	8.0 / 1.6	3.3	1.9

[1] Energy saving means the total energy from exhaust air (=evaporator energy) and renewable energy means energy saving deducted by the ventilation heat demand.

6.1.2 Apartment houses

<u>Ground source heat pump</u>

Ground source heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology. The temperature of the ground source was modelled as a cosine function with yearly average value of 0 ºC and peak-to-peak amplitude of 2 ºC (Figure 8). The warmest month was taken to be August.

Table 41. Ground source heat pump in apartment house: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	0 ºC	35 ºC	4,8
hot water production	0 ºC	52 ºC	2,7

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power varied from 80 kW (building type A) to 30 kW (building type D3) at declared operation temperatures of 0 ºC / +35 ºC. The results of the yearly calculation are presented in Table 42.

Table 42. Ground source heat pump in apartment house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A(1	232.4	64.5	169.0	3.6	6.6
B(1	160.2	46.7	114.3	3.4	3.7
C1(1	97.2	30.9	66.8	3.1	0.1
C2(2	82.5	26.2	56.8	3.2	0.1
D1(3	65.6	20.8	45.2	3.2	0.0
D2(4	52.8	18.2	34.9	2.9	0.0
D3(5	47.8	17.3	30.8	2.8	0.0

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating, (4Low energy building, floor heating, (5Passive house, floor heating

Air to water heat pump

Air to water heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology. The outdoor temperature was based on the weather data for Jyväskylä.

Table 43. Air to water heat pump in apartment house: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	7 ºC	35 ºC	4.2
hot water production	7 ºC	50 ºC	3.3

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power varied from 70 kW (building type A) to 25 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. The results of the yearly calculation are presented in Table 44.

Table 44. Air to water heat pump in apartment house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A(1	222.2	68.1	154.4	3.3	16.8
B(1	159.1	51.2	108.2	3.1	4.7
C1(1	93.6	31.5	62.4	3.0	3.6
C2(2	80.8	26.8	54.2	3.0	1.8
D1(3	64.6	20.7	44.1	3.1	1.0
D2(4	52.4	17.7	34.9	3.0	0.4
D3(5	47.4	16.5	31.1	2.9	0.4

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating, (4Low energy building, floor heating, (5Passive house, floor heating

Air to air heat pump

Air to air heat pumps were modelled with three performance points with different loads. The performance factors were chosen to represent the best practice of the available technology. The outdoor temperature was based in the weather data for Jyväskylä[4].

Table 45. Air to air heat pump in apartment house: performance points.

Load	Temperature of heat source	Temperature of heating	COP
100%	7 ºC	35 ºC	3.6
75%	7 ºC	35 ºC	4.1
50%	7 ºC	35 ºC	5.4

The heating power of one heat pump varied from 4 kW (building type A) to 3 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. It was presumed that the effective floor area of the heat pump is 75% of the total floor area and that the air heat pump raises the indoor air temperature by 2 ºC of the working area. The results of the yearly calculation are presented in Table 46.

[4] http://www.ym.fi/fi-FI/Maankaytto_ja_rakentaminen/Lainsaadanto_ja_ohjeet/Rakentamismaarayskokoelma

Table 46. Air to air heat pump in apartment house in one dwelling: results.

Building type	Heat produced by heat pump for space heating, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A	12.4	4.0	7.3	3.1	7.5
B	8.1	2.5	5.0	3.3	5.5
C1	3.8	1.1	2.4	3.3	4.3
C2	1.8	0.5	1.2	3.3	5.1
D1	0.7	0.3	0.5	2.6	4.7
D2	0.5	0.2	0.4	2.4	3.9
D3	0.3	0.2	0.2	1.8	3.7

Exhaust air heat pump

Exhaust air heat pumps were modelled with two performance point: 1) heating and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology.

Table 47. Exhaust air heat pump in apartment building: operation modes.

Operation mode	Temperature of exhaust air	Temperature of heating	COP
heating	-9 ºC	35 ºC	3.5
hot water production	-9 ºC	50 ºC	2.5

The humidity of the indoor air was presumed to be the same as the outdoor air humidity. Weather data of Jyväskylä[5] was used as the climate data. The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power of the exhaust air heat pump is determined by the exhaust air flow rate and it was 19,5 kW for each case at declared operation temperatures of -9 ºC / +35 ºC. The exhaust ai flow rate was 300 l/s. The results of the yearly calculation are pre-

5 http://www.ym.fi/fi-FI/Maankaytto_ja_rakentaminen/Lainsaadanto_ja_ohjeet/Rakentamismaarayskokoelma

sented in Table 48. Exhaust air heat pumps were supposed to be installed only to building types with no exhaust air heat recovery devices.

Table 48. Exhaust air to water heat pump in apartment house: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Energy saving / Renewable energy[1] MWh/a	SPF	Back-up heat required, MWh/a
A	126.9	44.4	82.8 / 30.8	2.9	112.2
B	114.1	41.0	73.5 / 21.9	2.8	49.7
C1	87.9	33.3	55.0 / 9.6	2.6	9.4

[1] Energy saving means the total energy from exhaust air (=evaporator energy) and renewable energy means energy saving deducted by the ventilation heat demand.

6.1.3 Cottages

Ground source heat pump

Ground source heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology. The temperature of the ground source was modelled as cosine function with yearly average value of 0 ºC and peak-to-peak amplitude of 2 ºC (Figure 8). The warmest month was taken to be August.

Table 49. Ground source heat pump in cottage: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	0 ºC	35 ºC	5.0
hot water production	0 ºC	52 ºC	2.7

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power varied from 8 kW (building type A) to 4 kW (building type D3) at declared operation temperatures of 0 ºC / +35 ºC. The results of the yearly calculation are presented in Table 50.

Table 50. Ground source heat pump in cottage: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A(1	13.8	3.8	10.1	3.7	1.5
B(1	13.4	3.7	9.8	3.6	1.8
C1(1	10.1	2.8	7.4	3.6	0.6
C2(2	10.0	2.6	7.5	3.8	0.5
D1(3	7.0	1.7	5.4	4.2	0.2
D2(4	4.5	1.2	3.4	3.7	0.0
D3(5	3.7	1.1	2.8	3.5	0.0

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating, (4Low energy building, floor heating, (5Passive house, floor heating

Air to water heat pump

Air to water heat pumps were modelled with two performance points: 1) heating at full power and 2) hot water production. The performance factors were chosen to represent the best practice of the available technology. The outdoor temperature was based on the weather data for Jyväskylä.

Table 51. Air to water heat pump in detached house: operation modes.

Operation mode	Temperature of heat source	Temperature of heating	COP
heating	7 ºC	35 ºC	4,2
hot water production	7 ºC	47 ºC	3,3

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power varied from 8 kW (building type A) to 4 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. The results of the yearly calculation are presented in Table 53.

Table 52. Air to water heat pump in cottage: results.

Building type	Heat produced by heat pump for space heating and domestic hot water MWh/a	Electricity use of heat pump MWh/a	Renewable energy use MWh/a	SPF	Back-up heat required MWh/a
A(1	12.4	4.9	7.6	2.5	2.9
B(1	12.0	4.7	7.3	2.5	3.2
C1(1	9.3	3.7	5.7	2.5	1.4
C2(2	8.6	3.2	5.5	2.7	1.9
D1(3	6.6	2.2	4.5	3.0	0.6
D2(4	4.2	1.5	2.8	2.9	0.3
D3(5	3.6	1.3	2.4	2.8	0.2

(1Old house, radiators, (2Old house, floor heating, (3New building, floor heating, (4Low energy building, floor heating, (5Passive house, floor heating

Air to air heat pump

Air to air heat pumps were modelled with three performance points with different loads. The performance factors were chosen to represent the best practice of the available technology. The outdoor temperature was based on the weather data for Jyväskylä.

Table 53. Air to air heat pump in cottage: performance points.

Load	Temperature of heat source	Temperature of heating	COP
100%	7 ºC	35 ºC	3.6
75%	7 ºC	35 ºC	4.1
50%	7 ºC	35 ºC	5.4

The heating power varied from 6 kW (building type A) to 4 kW (building type D3) at declared operation temperatures of +7 ºC / +35 ºC. It was presumed that the effective floor area of the heat pump is 50% of the total floor area and that the air heat pump raises the indoor air temperature by 2 ºC of the working area. The results of the yearly calculation are presented in Table 54.

Table 54. Air to air heat pump in cottage: results.

Building type	Heat produced by heat pump for space heating, MWh/a	Electricity use of heat pump, MWh/a	Renewable energy use, MWh/a	SPF	Back-up heat required, MWh/a
A	6.9	2.3	4.1	3.0	8.4
B	6.7	2.3	4.0	2.9	8.5
C1	4.7	1.6	2.9	3.0	6.0
C2	4.6	1.5	2.8	3.0	5.9
D1	2.7	1.0	1.7	2.8	4.4
D2	1.5	0.6	1.0	2.5	3.0
D3	1.1	0.5	0.7	2.3	2.6

Exhaust air heat pump

Exhaust air heat pumps were modelled with three performance points: 1) heating at full power, 2) heating at 46% power and 3) hot water production. The performance factors were chosen to represent the best practice of the available technology. The humidity of the indoor air was presumed to be the same as the outdoor air humidity. Weather data of Jyväskylä was used as the climate data.

Table 55. Exhaust air heat pump in cottage: operation modes.

Operation mode	Temperature of exhaust air	Temperature of heating	COP
heating, 100% load	-9 ºC	35 ºC	3.15
heating, 46% load	-9 ºC	35 ºC	4.7
hot water production	-9 ºC	50 ºC	2.5

The dimensioning supply temperature of the heating network varied between the building types from +70 ºC (building type A with radiators) to +35 ºC (building type D3 with floor heating). Linear weather compensation of the supply temperature was used in each case. The temperature of the hot water demand was +55 ºC. The heating power of the exhaust air heat pump is determined by the exhaust air flow rate and it was 2,8 kW for each case at declared operation temperatures of -9 ºC / +35 ºC. The exhaust air flow rate was 42 l/s. The results of the yearly calculation are presented in Table 56.

Table 56. Exhaust air to water heat pump in cottage: results.

Building type	Heat produced by heat pump for space heating and domestic hot water, MWh/a	Electricity use of heat pump, MWh/a	Energy saving /Renewable energy[1] MWh/a	SPF	Back-up heat required, MWh/a
A	9.6	3.6	6.2 / 1.5	2.7	5.6
B	9.4	3.5	6.1 / 1.4	2.7	5.7
C1	7.6	2.8	5.0 / 1.1	2.7	3.0
C2	7.4	2.6	5.0 / 1.0	2.8	3.1

[1]Energy saving means the total energy from exhaust air (=evaporator energy) and renewable energy means energy saving deducted by the the ventilation heat demand.

6.2 Energy saving and renewable energy use by building type

Tables 57–59 present energy saving and renewable energy use by building type for years 2010, 2016 and 2020 in accordance with the heat pump (HP) scenario presented in section 6.3.

Table 57. Renewable energy use and energy savings by building type in 2010.

Energy savings and renewable energy produced[6] in 2010, GWh				
	Detached houses	**Apartment buildings**	**Free time residences**	**Total**
Ground source	936	2	3	941
Air/water	107	10	0	117
Air/air	1803	184	29	2016
Exhaust air	155	0	2	157
Total	3001	196	34	3231

[6] Renewable energy for other than exhaust air heat pumps. With exhaust air heat pumps only part of the savings can be considered renewable energy, see section 6.1.

Table 58. Renewable energy use and energy savings by building type in 2016.

Energy savings and renewable energy produced[6] in 2016, GWh				
	Detached houses	**Apartment buildings**	**Free time residences**	**Total**
Ground source	2434	5	6	2445
Exhaust air	579	11	0	590
Air/air	2960	670	42	3672
Air/water	306	77	3	386
Total	6279	763	51	7093

Table 59. Renewable energy use and energy savings by building type in 2020.

Energy savings and renewable energy produced[6] in 2020, GWh				
	Detached houses	**Attached houses**	**Free time residences**	**Total**
Ground source	3360	7	8	3375
Exhaust air	890	18	0	908
Air/air	3726	842	49	4617
Air/water	374	109	4	487
Total	8350	976	61	9387

6.3 Effects of heat pumps on the energy use and emissions of the Finnish building stock

The effects of heat pumps on the energy use in the building stock were modelled using the REMA model. REMA, described in section 5.3, was used to model the future energy use in the Finnish building stock. As a starting point, the model has a conservative Business as Usual or BAU scenario, where the present trends in the development of the building stock, including heat pumps, are assumed to continue in the future but taking into account known changes in building regulation. For the purposes of this project, another scenario called Heat Pump or HP scenario was calculated based on the results of the calculations concerning heat pump use in the type buildings, presented in section 6.1, and the calculation concerning the increase of heat pump use in the future, presented in section 6.2. The HP scenario and the calculations used to produce it are intented to present a possible development path where heat pumps are used in a rather large scale and they are installed and operated at close to optimum settings. This scenario is seen as technically possible but dependent on future decisions.

REMA model comprises the whole building stock, allowing the inclusion of factors such as the different energetic properties of different types and ages of buildings, the replacement of various alternative heat sources with heat pumps and differentiating heat pumps retrofitted in old buildings from the ones installed in new buildings. REMA model also includes a simplified model of the energy production infrastructure that allows the assessment of the effects on CO_2 emissions.

The calculation of the scenario starts from year 2010 and ends on year 2020 based on the data available in Table 4 and Table 6. This means that some of the results will differ from the baseline scenario already in past years, namely 2010–2013. As the fast increase in the number of heat pumps in the recent years has brought uncertainty in their effects on energy use in the building stock, the results for 2013 are presented in Table 60 together with the forecast for the year 2020 in Table 61. It should be noted that the difference shown in these tables shows the difference between BAU and HP scenarios. As BAU scenario also includes a modest amount of heat pumps, the figures should not be interpreted to show the total effect of all heat pumps. For such figures, the reader is directed to section 6.4. The differences between the scenarios presented here represent the difference of an accelerated adoption of heat pumps as opposed to a continuation of a business as usual scenario.

Table 60. Modelled energy use for heating in 2013 (GWh).

	District heat	**Oil**	**Wood**	**Electricity**	**Total**
BAU scenario	28960	16196	17062	10553	72771
HP scenario	28957	14788	16539	9364	69648
Difference	-3	-1408	-523	-1189	-3123

Table 61. Modelled energy use for heating in 2020 (GWh).

	District heat	**Oil**	**Wood**	**Electricity**	**Total**
BAU scenario	28608	14785	16006	11086	70485
HP scenario	28598	11510	15086	10022	65216
Difference	-10	-3275	-920	-1064	-5269

The results indicate that in 2013 total modelled use of heating energy in the HP scenario would be 72 800 GWh, of which electricity accounts for 10 600 GWh. This is 3100 GWh less in total and 1200 GWh less in electricity than was projected in the

BAU scenario, the difference being due to a faster than anticipated increase in the number of heat pumps. By 2020 the numbers are forecast to be 70 500 GWh of total heating energy in the HP scenario, of which 11 100 GWh is electricity, and which is 5300 GWh lower in total and 1100 GWh lower in electricity than the BAU.

Table 62 and Table 63 show the results for modelled CO_2 emissions. The said differences in energy consumption cause the emissions to be 800 kilotonnes lower in 2013 and 1300 kilotonnes lower in 2020 in the HP scenario compared to BAU.

Table 62. Modelled CO_2 emissions from the building stock in 2013 (KT). DHW stands for domestic hot water.

	Space heating	**DHW**	**Electricity (non-heating)**	**Total**
BAU scenario	12573	2023	4916	19511
HP scenario	11806	2023	4916	18745
Difference	-767	0	0	-767

Table 63. Modelled CO_2 emissions from the building stock in 2020 (KT). DHW stands for domestic hot water.

	Space heating	**DHW**	**Electricity (non-heating)**	**Total**
BAU scenario	11792	2084	4811	18688
HP scenario	10554	2013	4811	17379
Difference	-1238	-71	0	-1309

A time series of the development of energy consumed and CO_2 emissions caused in the building stock can be seen in Figure 9. It should be noted that the modelling after 2020 is not accurate but can be used as an indicator of the rought direction of the development. It can be seen in the figure that both the energy use and CO_2 emissions are on a likely path of reductions in any case because new regulations require much lower energy consumption from buildings. Over time new buildings and buildings renovated to new standards replace older, more energy consuming buildings causing the downward trend. The effect of heat pumps in the HP scenario is to slightly increase the reductions and also quicken their pace in the near future. This is a desirable development as reductions in CO_2 emissions taking place soon are preferable to later reductions due to the urgency in the mitigation of the climatic effects.

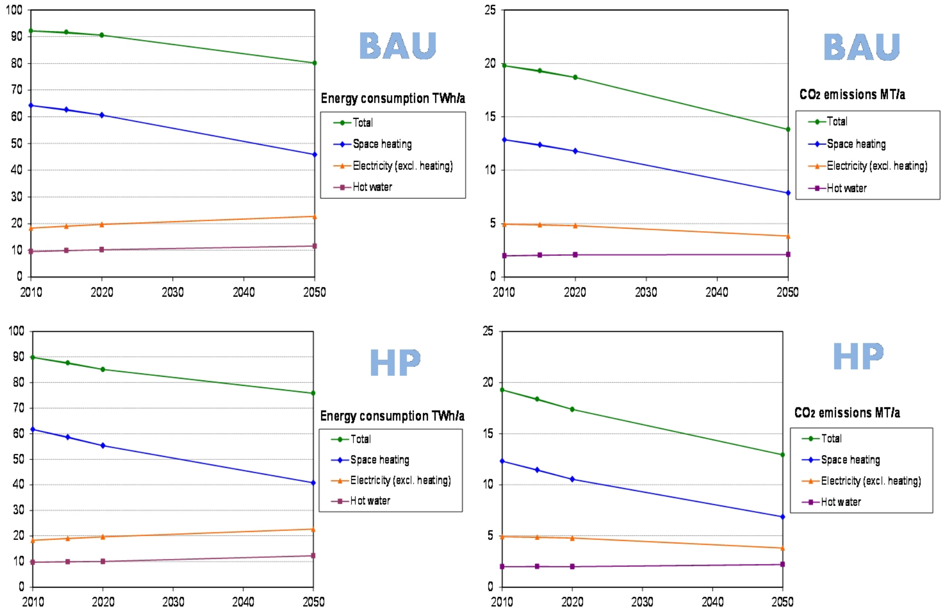


Figure 9. The development of energy use and CO_2 emissions in the two scenarios.

6.4 Renewable energy use and energy savings by heat pumps

Figure 10 presents energy savings for the different heat pump type in accordance with the heat pump (HP) scenario presented in section 6.3. This is approximate also for renewable energy use, except for exhaust air pumps where only part of the savings can be considered renewable energy, for more discussion see section 6.1. Table 64 summarizes the estimated use of renewable energy and energy savings of the heat pumps.

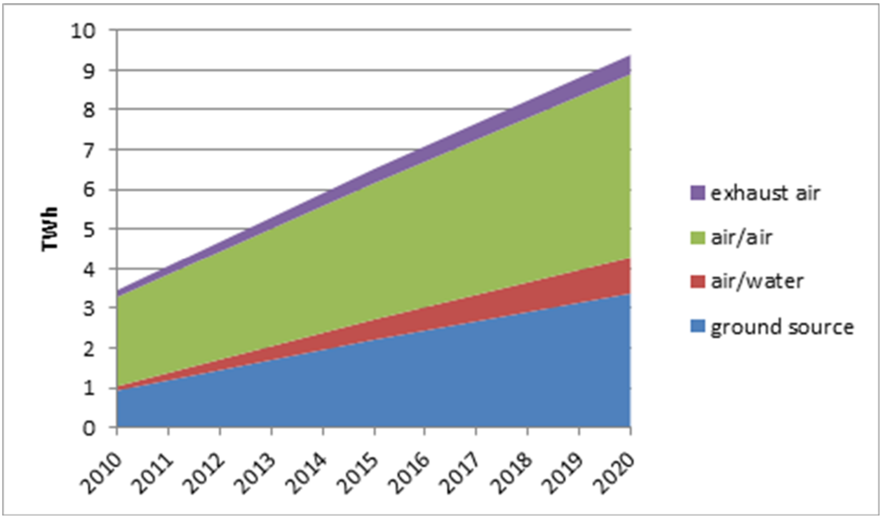


Figure 10. Estimate of energy saved by heat pumps in 2010–2020. For heat pumps other than exhaust air, it represents also renewable energy production.

Table 64. Summary of the estimated use of renewable energy and energy savings of the heat pumps.

Year	Renewable energy, GWh	Energy savings, GWh
2010	3128	3231
2016	6903	7093
2020	9133	9387

7. Annex co-operation

One task of the project was participating in IEA Annex 39 “A common method for testing and rating of residential HP and AC annual/seasonal performance”.

The background for Annex 39 was the demand for a common SPF calculation method for fair comparison between different types of heat pump systems as well as fair comparison with other competing technologies using fossil fuels. A common SPF method could also later be incorporated in different labelling, rating and certification schemes. The common method should be a transparent and harmonised method for calculation of heat pump system SPF and based on repeatability and reliable test data from laboratory measurements.

Heat pumps using aerothermal, geothermal or hydrothermal energy as a source are defined as renewable in the European RES Directive if the SPF is above a specific value. The renewable energy production E_{res} (kWh/a) of a heat pump is calculated as a function of SPF and the annual heating energy production of the heat pump (Q_{usable}), see Chapter 3. The Annex Commission has established in 2013 guidelines on how Member States shall estimate the values of Qusable and SPF for the different heat pump technologies and applications, taking into consideration differences in climatic conditions, especially very cold climates. From a European point of view, it was therefore very important and urgent to define a common standard for SPF calculation at the time when the Annex 39 was prepared.

The legal text for Annex 39 also states that “The development of heat pump standards differs between Asia, North America and Europe and there is a large number of national standards for both testing and calculation of SPF. The heat pump manufacturers would need common testing methods and common SPF methods to simplify the export of heat pumps to different countries. The end users need reliable information in the selection procedure both between different heat pumps as well as in comparing heat pumps with other competing technologies.”

A common SPF calculation method is not easy to define because of different building standards and heat distribution systems. A real value of the SPF should be calculated for each specific installation, from field measured data. A simplified general approach would be making the calculations for one specific building in one specific climate, or to define a limited number of regions with typical climate and buildings.

Annex 39 was coordinated by SP Technical Research Institute of Sweden. Other participants besides VTT Technical Research Centre of Finland and Aalto University were Oak Ridge National Laboratory (ORNL) and Air-Conditioning, Heating, and Refrigeration Institute (AHRI) from USA, Kungliga Tekniska Högskolan (KTH) from Sweden, Austrian Institute of Technology (AIT), EDF from France, Fraunhofer ISE from Germany, HPTCJ from Japan, Korea Institute of Energy Research (KIER) from South Korea and FHNW from Swizerland.

The objects of Annex 39 were to

1. Establish common calculation methods for SPF using a generalised and transparent approach, fair comparison between different heat pump types and comparison between different competing technologies, such as pellet boilers and gas boilers.
2. Establish comprehensive test methods based on further development of existing test standards. The test standards should include test conditions needed for the future SPF calculations.

The Annex work consisted of following tasks

1. Survey and evaluation of existing testing methods and calculation methods for SPF
2. Matrix definition of needs for testing and calculation methods
3. New calculation method for SPF/ Commonly accepted definitions on how SPF is calculated
4. Identify improvements to existing test procedures
5. Validation of SPF method
6. Development of an alternative method to evaluate heat pump performance
7. Communication to stakeholders.

The outcome from the project was meant to be a proposal for a common transparent SPF calculation method for domestic heat pumps including heating, cooling and domestic hot water production. However, a common approach was not found in the Annex 39, instead of this the Annex ended up into listing possible ways of SPF calculation.

8. Summary and conclusions

The main objects of the SPF project were to define a Finnish SPF calculation method for heat pumps in co-operation with international Annex 39 work and to estimate the current and future energy saving and renewable energy use potential of heat pumps on the Finnish building stock.

The developed hourly SPF calculation method can be used for heating energy calculation of air-air-, air-water-, outlet air and ground source heat pumps but not for power dimensioning of a heat pump. Calculation can be performed also with other time steps than one hour. Measured performance values for the heat pump are needed for calculation input values at least in one test point but the calculation is more accurate if performance values are available from several test points. The calculation method does not take into account the heat storage ability of the domestic service water accumulator. The calculation method assumes that the heat pump heats up both domestic service water and spaces in turns so that heating up the domestic service water is the primary function.

The energy use of the Finnish building stock was estimated using standard building types further adapted to different decades: a detached house, an apartment building, an office building and a summer cottage. The energy use of these standard building types was calculated with different heat pump types leading to energy saving and renewable energy use of the heat pumps.

The cumulative energy consumption of the standard building types was calculated based on the modelled development of the building stock using the REMA model developed at VTT. The simulated energy demand results of each standard building type and subgroup were used as an input for the REMA model to calculate the total energy consumption of the building stock in each year, taking into consideration the estimated changes in the future development of the building stock.

The future effects of heat pumps on the energy use in the Finnish building stock were modelled comparing with the REMA model a conservative Business as Usual or BAU scenario with a Heat Pump or HP scenario. In the BAU scenario the present trends in the development of the building stock were assumed to continue in the future but the known changes in building regulation were taken into account. HP scenario was calculated based on the results of the calculations concerning heat pump use in the type buildings and the calculation concerning the increase of heat pump use in the future. The HP scenario presented a possible development path where heat pumps are used in a rather large scale and they are installed and operated at close to optimum settings. This scenario is seen as technically possible but dependent on future decisions.

The HP scenario would have 3100 GWh less total heating energy use and 1200 GWh less electricity use than the BAU scenario in 2013. In 2020 the differences would be 5300 GWh lower total heating energy use and 1100 GWh lower electricity use in HP scenario compared to the BAU scenario. The CO_2 emissions of the HP scenario would be 800 kilotonnes lower in 2013 and 1300 kilotonnes lower in 2020 compared to the BAU scenario.

Acknowledgements

The project was financed by the Finnish ministry of employment and the economy, the Finnish ministry of the environment and SITRA.

References

Directive 2012/27/EU of the European Parliament and of the Council of 25 October 2012 on energy efficiency, amending Directives 2009/125/EC and 2010/30/EU and repealing Directives 2004/8/EC and 2006/32/EC.

Dyhr, R. 1993. Asuinrakennusten ilmanvaihtoselvitys. Helsingin kaupunki.

Eskola, L., Kurnitski, J. and Palonen, J. 2009. Building leakage rate measurement with building's own air handling unit. Healthy Buildings 2009, Syracuse, NY, USA, 13–17 September 2009. 6 p. [in Finnish]

Kalamees, T., Jylhä, K., Tietäväinen, H., Jokisalo, J., Ilomets, S., Hyvönen, R. and Saku, S. 2012. Development of weighting factors for climate variables for selecting the energy reference year according to the EN ISO 15927-4 standard. Energy and Buildings, 47, 53–60.

Motiva Oy. 2009. Pientalon lämmitysjärjestelmät. Helsinki. www.motiva.fi/files/2701/Pientalon_lammitysjarjestelmat.pdf [in Finnish]

Nieminen, J., Jahn, J. and Airaksinen, M. 2007. Passiivitalon rakennesuunnittelu. VTT Technical Research Centre of Finland, Espoo, Finland. [in Finnish] http://northpass.vtt.fi/Finland/Documents/5Rakennesuunnittelu.pdf.

Nykänen, V. and Heljo, J. 1985. Rakennusten energiatalous Suomessa (Energy performance of buildings in Finland). Ministry of Trade and Industry, Helsinki, Finland. [in Finnish]

Peuhkuri, R. and Pedersen, S. 2010. Application of the local criteria/standards and their differences for very low-energy and low energy houses in the participating countries. NorthPass Report. Uploadable in www.northpass.eu.

Polvinen, M., Kauppi, A., Saarimaa, J., Haalahti, P. and Laurikainen, M. 1983. Rakennusten ulkovaipan ilmanpitävyys. (Air tightness of the buildings.) Tutkimuksia 215/1983. VTT, Espoo, Finland. [in Finnish]

Ruotsalainen, R. 1992. Indoor climate and the performance of ventilation in Finnish residences. Indoor Air, 2, 137–145.

SFS-EN 14511-2. Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling. Part 2: Test conditions.

SFS-EN 14511-3. Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling. Part 3: Test methods.

SFS-EN 14825. Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling. Testing and rating at part load conditions and calculation of seasonal performance.

Suomela, M. 2010. Toimistotalon ilmatiiviyden mittaaminen. (Measurement of the airtightness of office buildings.) Bachelor thesis 96. Seinäjoki University of Applied Sciences, Seinäjoki, Finland. [in Finnish]

Tuominen, P., Holopainen, R., Eskola, L. and Jokisalo, J. 2014. Calculation method and tool for assessing energy consumption in the building stock. Building and Environment, 75, 153–160.

Vinha, J., Korpi, M., Kalamees, T., Eskola, L., Palonen, J., Kurnitski, J., Valovirta, I., Mikkilä, A. and Jokisalo, J. 2005. Puurunkoisten pientalojen kosteus- ja lämpötilaolosuhteet, ilmanvaihto ja ilmatiiviys. (Indoor temperature and humidity conditions, ventilation and airtightness of Finnish timber-framed detached houses.) Report 131. Tampere University of Technology, Institute of Structural Engineering, Tampere, Finland. 102 p. + app. 10 p. [in Finnish]

Vinha, J., Korpi, M., Kalamees, T., Jokisalo, J., Eskola, L., Palonen, J., Kurnitski, J., Aho, H., Salminen, M., Salminen, K. and Keto, M. 2009. Asuinrakennusten ilmanpitävyys, sisäilmasto ja energiatalous. (Airtighness, indoor climate and energy efficiency of residential buildings.) Report 140. Tampere University of Technology, Institute of Structural Engineering, Tampere, Finland.

Virta, J. and Pylsy, P. 2011. Taloyhtiön energiakirja. Kiinteistöalan kustannus, Helsinki, Finland. [in Finnish]

Part C3. 1985. Finnish Building Code (Thermal insulation of buildings). Suomen rakentamismääräyskokoelma, Rakennusten lämmöneristys. Regulations 1985. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part C3. 2003. Finnish Building Code (Thermal insulation of buildings). Suomen rakentamismääräyskokoelma, Rakennusten lämmöneristys. Regulations 2003. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part C3. 2010. Finnish Building Code (Thermal insulation of buildings). Suomen rakentamismääräyskokoelma, Rakennusten lämmöneristys. Regulations 2010. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D2. 1987. Finnish Building Code (Indoor Climate and Ventilation of Buildings). Suomen rakentamismääräyskokoelma, Rakennusten sisäilmasto ja ilmanvaihto. Regulations and guidelines 1987. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D2. 2003. Finnish Building Code (Indoor Climate and Ventilation of Buildings). Suomen rakentamismääräyskokoelma, Rakennusten sisäilmasto ja ilmanvaihto. Regulations and guidelines 2003. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D2. 2010. Finnish Building Code (Indoor Climate and Ventilation of Buildings). Suomen rakentamismääräyskokoelma, Rakennusten sisäilmasto ja ilmanvaihto. Regulations and guidelines 2010. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D3. 2012. Finnish Building Code (Energy efficiency of buildings). Suomen rakentamismääräyskokoelma, Rakennusten energiatehokkuus. Regulations and guidelines 2012. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D5. 2007. Finnish Building Code (Calculation of energy consumption and heating power of buildings). Suomen rakentamismääräyskokoelma, Rakennuksen energiankulutuksen ja lämmitystehontarpeen laskenta. Guidelines 2007. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Part D5. 2012. Finnish Building Code (Calculation of energy consumption and heating power of buildings). Suomen rakentamismääräyskokoelma, Rakennuksen energiankulutuksen ja lämmitystarpeen laskenta. Guidelines 2012. Ministry of the Environment, Helsinki, Finland. [in Finnish]

Appendix A: Detailed calculation method in Finnish

Lämpöpumppujen energialaskentamenetelmä

1. Määritelmiä

Aika-askel	Laskentahetkien välinen vakioaika, joka on tässä laskentamenetelmässä vapaasti valittavissa Tässä julkaisussa aika-askel on oletusarvoisesti yksi tunti. Laskentamenetelmän yhtälöt ratkaistaan kullakin aika-askeleella.
Lisälämmitys	Lämpöpumpun tuottaman lämpöenergian ja rakennuksen lämmitysenergian välinen erotus, jos lämpöpumppu ei pysty tuottamaan tarvittavaa lämpötehoa. Lisälämmitystarve aiheutuu joko lämpöpumpun toimintaan liittyvistä lämpötilarajoituksista (vrt. toimintarajalämpötila ja ylärajalämpötila) tai lämpöpumpun osatehomitoituksesta, jolloin lämpöpumppu ei tuota rakennuksen lämmitystehon tarvetta mitoitustilanteessa.
Mitoitusulkolämpötila	Rakennusten lämmitysjärjestelmän lämmitystehon mitoituksessa käytettävä ulkoilman vakiolämpötila, joka määritellään Suomessa säävyöhykekohtaisesti Suomen rakentamismääräyskokoelmassa D3 (2012).
On-off-säätöinen lämpöpumppu	
	On-off-säätöisissä lämpöpumpuissa kompressori kytkeytyy päälle ja toimii nimellistehollaan, kunnes lämpöpumpun säätöjärjestelmä havaitsee, että haluttu lämpötila on saavutettu, jolloin kompressori pysähtyy.
SPF-luku	Lämpöpumpun vuoden keskimääräinen lämpökerroin, joka on lämpöpumpulla tuotetun vuotuisen energian suhde lämpöpumpun sekä lämpöpumpun apulaitteiden vuotuiseen sähkönkulutukseen.
Tehosäätöinen lämpöpumppu	
	Tehosäätöinen (invertteri- tai kapasiteettisäätöinen) lämpöpumppu voi toimia osateholla lämmöntarpeen ollessa pienempi kuin lämpöpumpun tuottama suurin mahdollinen lämmöntuotto toimintalämpötilassa.

Teoreettinen lämpökerroin

Lämpöpumpun hetkellinen lämpökerroin, joka määräytyy pelkästään rakennuksen lämmitysjärjestelmän lämpötilan ja lämmönlähteen lämpötilan perusteella. Teoreettisessa lämpökertoimessa ei ole otettu huomioon kompressorin häviöitä.

2. Laskentamenetelmä

2.1 Laskentaperiaatteet ja rajaukset

Tässä julkaisussa esitettävää tunneittaista laskentamenetelmää voidaan käyttää lämmityskäytössä olevien ilma-ilma-, ilma-vesi-, poistoilma- ja maalämpöpumppujen energialaskentaan. Laskentamenetelmällä ei voida tehdä lämpöpumppujen tehomitoitusta. Laskenta voidaan suorittaa haluttaessa myös muulla kuin tunnin aika-askeleella, mutta tässä julkaisussa aika-askeleen pituutena käytetään yhtä tuntia.

Laskennan lähtötiedoiksi tarvitaan lämpöpumpun mitattuja tuotetietoja vähintään yhdessä testauspisteessä. Laskennassa on-off-säätöinen ilma-ilma-, ilma-vesi- ja maalämpöpumppu toimivat aina nimellisteholla. On-off-säätöisten lämpöpumppujen höyrystimen ja lauhduttimen tehojen oletetaan muuttuvan lämpötilojen ja niiden myötä lämpökertoimen mukaan, kun lämpöpumpun tuotetiedot tunnetaan yhdessä testauspisteessä. Laskentaa voidaan tarkentaa, mikäli lämpöpumppujen tuotetietoja on käytettävissä useasta testauspisteestä. On-off-säätöisen poistoilmalämpöpumpun lauhduttimen teho riippuu höyrystimen tehosta ja lämpökertoimesta. Lämpöpumppujen todellinen lämpökerroin on suoraan verrannollinen teoreettisen lämpökertoimen arvoon. Kaikkien tehosäätöisten lämpöpumppujen lämmitysteho määräytyy rakennuksen tunneittaisen lämmitystehontarpeen mukaan.

Laskentamenetelmä ei ota huomioon lämminvesivaraajan lämmönvarastointikyvyn vaikutusta lämpöpumpun toimintaan. Laskennassa oletetaan, että käyttövettä ja tiloja lämmittävä lämpöpumppu lämmittää käyttövettä ja tiloja vuorotellen niin, että käyttövettä lämmitetään ensisijaisesti.

2.2 Laskennan kuvaus

2.2.1 Ilma-ilmalämpöpumppu

1. Määritetään seuraavat laskennan lähtötiedot:
    a. Vuoden ulkolämpötila sekä laskennan aika-askeleen pituus (luku 2.3.1).
    b. Tilojen lämmitystehon tai energian tarve (luku 2.3.3).
    c. Lämpöpumpun sisäänpuhalluslämpötila tai tilojen lämmityksen asetusarvo, mikäli sisäänpuhalluslämpötilaa ei tunneta (luku 2.3.4).

   d. Lämpöpumpun mitattu lämpökerroin, lämmitysteho sekä lämpötilat vähintään yhdessä testauspisteessä (luku 2.3.4).
   e. Matalin ulkolämpötila, jossa lämpöpumppua voidaan käyttää (luku 2.3.4).
2. Lasketaan tilojen lämmitystehon tarve kullakin aika-askeleella, mikäli sitä ei ole saatavilla lähtötietona (luku 2.5.2).
3. Lasketaan lämpöpumpun lämpökerroin kullakin aika-askeleella (luku 2.5.4).
4. Lasketaan osatehon vaikutus lämpökertoimen arvoon kullakin aika-askeleella, mikäli lämpöpumppu on tehosäätöinen (luku 2.5.5).
5. Lasketaan lämpöpumpun lämmitysteho kullakin aika-askeleella (luku 2.5.10).
6. Lasketaan tilojen lämmitysenergian tarve kullakin aika-askeleella (luku 2.5.13).
7. Lasketaan tilojen lämmitysenergia, jonka lämpöpumppu pystyy tuottamaa kullakin aika-askeleella ja koko vuoden aikana (luku 2.5.13).
8. Lasketaan lämpöpumpun sähköenergia kullakin aika-askeleella sekä vuotuinen sähköenergian kulutus (luku 2.5.14).
9. Lasketaan lämpöpumpun SPF-luku (luku 2.5.16).
10. Lasketaan tarvittava tilojen vuotuinen lisälämmitysenergia (luku 2.5.17).

2.2.2. Ilma-vesilämpöpumppu

Ilma-vesilämpöpumpun laskenta tehdään seuraavien vaiheiden mukaisesti:

1. Määritetään seuraavat laskennan lähtötiedot:
   a. Vuoden ulkolämpötila sekä laskennan aika-askeleen pituus (luku 2.3.1).
   b. Käyttöveden lämmitystehon tai lämmitysenergian tarve sekä lämpötila (luku 2.3.2).
   c. Tilojen ja ilmanvaihdon lämmitysenergian ja tehon tarve (luku 2.3.3).
   d. Menoveden maksimi- ja minimilämpötila, mitoitusulkolämpötila ja ulkolämpötila, jolla menoveden lämpötila vastaa minimilämpötilaa (luku 2.3.3).
   e. Lämpöpumpun mitattu lämpökerroin, lämmitysteho sekä lämpötilat vähintään yhdessä testauspisteessä (luku 2.3.4).
   f. Korkein lämpötila, johon lämpöpumppu pystyy lämmittämään käyttövettä ilman lisälämmityksen tarvetta (luku 2.3.4).
   g. Matalin ulkolämpötila, jossa lämpöpumppua voidaan käyttää (luku 2.3.4).
2. Mikäli lämpöpumppu lämmittää käyttövettä, lasketaan seuraavat vaiheet:
   a. Lasketaan käyttöveden lämmitystehon tarve kullakin aika-askeleella, mikäli sitä ei ole saatavilla lähtötietona (luku 2.5.1).

b. Lasketaan lämpöpumpun lämpökerroin käyttöveden lämmityksessä kullakin aika-askeleella (luku 2.5.4).

c. Lasketaan lämpöpumpun lämmitysteho käyttöveden lämmityksessä kullakin aika-askeleella (luku 2.5.6).

d. Lasketaan osatehon vaikutus käyttöveden lämmityksen lämpökertoimen arvoon kullakin aika-askeleella, mikäli lämpöpumppu on tehosäätöinen (luku 2.5.5).

e. Lasketaan käyttöveden lämmitysaika kunkin aika-askeleen aikana (luku 2.5.8).

f. Lasketaan käyttöveden lämmitysenergia, jonka lämpöpumppu pystyy tuottamaa kullakin aika-askeleella ja koko vuoden aikana (luku 2.5.9).

3. Mikäli lämpöpumppu lämmittää käyttöveden lisäksi myös tiloja ja/tai ilmanvaihtoa, lasketaan seuraavat vaiheet:

a. Lasketaan tilojen ja ilmanvaihdon lämmitystehon tarve kullakin aika-askeleella, mikäli sitä ei ole saatavilla lähtötietona (luku 2.5.2).

b. Lasketaan lämpöpumpun lämpökerroin tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella (luku 2.5.4).

c. Lasketaan lämpöpumpun lämmitysteho tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella (luku 2.5.10).

d. Lasketaan osatehon vaikutus tilojen ja ilmanvaihdon lämmityksen lämpökertoimen arvoon kullakin aika-askeleella, mikäli lämpöpumppu on tehosäätöinen (luku 2.5.5).

e. Lasketaan aika, joka on käytettävissä tilojen ja ilmanvaihdon lämmitykseen kullakin aika-askeleella (luku 2.5.12).

f. Lasketaan tilojen ja ilmanvaihdon lämmitysenergian tarve kullakin aika-askeleella (luku 2.5.13).

g. Lasketaan tilojen ja ilmanvaihdon lämmitysenergia, jonka lämpöpumppu pystyy tuottamaa kullakin aika-askeleella ja koko vuoden aikana (luku 2.5.13).

4. Lasketaan lämpöpumpun sähköenergia käyttöveden, tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella sekä vuotuinen sähköenergian kulutus (luku 2.5.14).

5. Lasketaan tarvittaessa niiden apulaitteiden sähkönkulutuksen osuus, joka ei sisälly lämpökertoimen mitattuun arvoon (luku 2.5.15).

6. Lasketaan lämpöpumpun SPF-luku (luku 2.5.16).

7. Lasketaan tarvittava tilojen vuotuinen lisälämmitysenergia (luku 2.5.17).

2.2.3 Maalämpöpumppu

Maalämpöpumpun laskenta tehdään samojen vaiheiden mukaisesti kuin ilma-vesilämpöpumpun (ks. luku 2.2.2) sillä erotuksella, että lähtötietona tarvitaan lisäksi lämmönkeruupiiristä tulevan nesteen lämpötila (luku 2.3.4) ja lämpökertoimen laskennassa käytetään keruupiiriltä höyrystimelle virtaavan nesteen lämpötilaa.

2.2.4 Poistoilmalämpöpumppu

Poistoilmalämpöpumpun laskenta tehdään seuraavien vaiheiden mukaisesti:

1. Määritetään seuraavat laskennan lähtötiedot:
   a. Vuoden ulkolämpötila, mikäli tilojen ja ilmanvaihdon tunneittaista lämmitystehontarvetta ei tunneta, sekä laskennan aika-askeleen pituus (luku 2.3.1).
   b. Käyttöveden lämmitystehon tai lämmitysenergian tarve sekä lämpötila (luku 2.3.2).
   c. Tilojen ja ilmanvaihdon lämmitysenergian ja tehon tarve (luku 2.3.3).
   d. Menoveden maksimi- ja minimilämpötila, mitoitusulkolämpötila ja ulkolämpötila, jolla menoveden lämpötila vastaa minimilämpötilaa (luku 2.3.3).
   e. Lämpöpumpun mitattu lämpökerroin sekä lämpötilat vähintään yhdessä testauspisteessä (luku 2.3.4).
   f. Korkein lämpötila, johon lämpöpumppu pystyy lämmittämään käyttövettä ilman lisälämmityksen tarvetta (luku 2.3.4).
   g. Poistoilman lämpötila, kosteus sekä jäteilman matalin lämpötila (luku 2.3.4).
2. Mikäli lämpöpumppu lämmittää käyttövettä, lasketaan seuraavat vaiheet:
   a. Lasketaan käyttöveden lämmitystehon tarve kullakin aika-askeleella, mikäli sitä ei ole saatavilla lähtötietona (luku 2.5.1).
   b. Lasketaan lämpöpumpun lämpökerroin käyttöveden lämmityksessä kullakin aika-askeleella (luku 2.5.4).
   c. Lasketaan poistoilmalämpöpumpun lämmitysteho käyttöveden lämmityksessä kullakin aika-askeleella (luku 2.5.7).
   d. Lasketaan osatehon vaikutus käyttöveden lämmityksen lämpökertoimen arvoon kullakin aika-askeleella, mikäli lämpöpumppu on tehosäätöinen (luku 2.5.5).
   e. Lasketaan käyttöveden lämmitysaika kunkin aika-askeleen aikana (luku 2.5.8).

   f. Lasketaan käyttöveden lämmitysenergia, jonka lämpöpumppu pystyy tuottamaa kullakin aika-askeleella ja koko vuoden aikana (luku 2.5.9).

3. Mikäli lämpöpumppu lämmittää käyttöveden lisäksi myös tiloja ja/tai ilmanvaihtoa, lasketaan seuraavat vaiheet:

   a. Lasketaan tilojen ja ilmanvaihdon lämmitystehon tarve kullakin aika-askeleella, mikäli sitä ei ole saatavilla lähtötietona (luku 2.5.2).

   b. Lasketaan lämpöpumpun lämpökerroin tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella (luku 2.5.4).

   c. Lasketaan poistoilmalämpöpumpun lämmitysteho tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella (luku 2.5.11).

   d. Lasketaan osatehon vaikutus tilojen ja ilmanvaihdon lämmityksen lämpökertoimen arvoon kullakin aika-askeleella, mikäli lämpöpumppu on tehosäätöinen (luku 2.5.5).

   e. Lasketaan aika, joka on käytettävissä tilojen ja ilmanvaihdon lämmitykseen kullakin aika-askeleella (luku 2.5.12).

   f. Lasketaan tilojen ja ilmanvaihdon lämmitysenergian tarve kullakin aika-askeleella (luku 2.5.13).

   g. Lasketaan tilojen ja ilmanvaihdon lämmitysenergia, jonka lämpöpumppu pystyy tuottamaa kullakin aika-askeleella ja koko vuoden aikana (luku 2.5.13).

4. Lasketaan lämpöpumpun sähköenergia käyttöveden, tilojen ja ilmanvaihdon lämmityksessä kullakin aika-askeleella sekä vuotuinen sähköenergian kulutus (luku 2.5.14).

5. Lasketaan tarvittaessa niiden apulaitteiden sähkönkulutuksen osuus, joka ei sisälly lämpökertoimen mitattuun arvoon (luku 2.5.15).

6. Lasketaan lämpöpumpun SPF-luku (luku 2.5.16).

7. Lasketaan tarvittava tilojen vuotuinen lisälämmitysenergia (luku 2.5.17).

2.3 Laskennan lähtötiedot

2.3.1 Säätiedot

Laskennan lähtötiedoksi tarvitaan seuraavia tietoja:

- Vuoden tunneittainen ulkoilman lämpötila $T_{ulko}(t)$, °C.
- Laskennan sekä säätietojen aika-askeleen pituus $t_{aika\text{-}askel}$, h.

Säätietoina voidaan käyttää esimerkiksi Suomen RakMk D3:n (2012) mukaisia tunneittaisia energialaskennan säätietoja, jotka ovat saatavilla esimerkiksi Ilmatieteenlaitoksen www-sivuilla [http://ilmatieteenlaitos.fi/rakennusten-energialaskennan-testivuosi]. Laskenta voidaan tehdä haluttaessa myös muulla kuin tunnin aika-askeleella, mutta tässä julkaisussa aika-askeleen pituudeksi oletetaan yksi tunti. Laskennan aika-askeleen pituuden $T_{aika\text{-}askel}$ tulee vastata säätietojen aika-askeleen pituutta, jolloin tunneittaisia säätietoja käytettäessä aika-askeleen pituus on 1 h. On syytä huomata, että aika-askeleen pituuden kasvattaminen lisää laskennan epätarkkuutta.

2.3.2 Käyttöveden lämmitystarve

Mikäli lämpöpumppua käytetään käyttöveden lämmitykseen, laskennan lähtötiedoksi tarvitaan seuraavia lähtötietoja:

- Käyttöveden tunneittainen lämmitystehon tarve $\phi_{lkv}(t)$, kW.
- Käyttöveden lämmityksen vuotuinen lämpöenergian tarve $Q_{lämmitys,lkv}$, kWh/v, mikäli tunneittaista lämmitystehon tarvetta $\phi_{lkv}(t)$ ei tunneta.
- Käyttöveden lämpötila T_{lkv}, °C.
- Kylmän käyttöveden lämpötila T_{kv}, °C.

Käyttöveden lämmitysteho $\phi_{lkv}(t)$ ja lämmitysenergian tarve $Q_{lämmitys,lkv}$ sisältävät siirron ja varastoinnin lämpöhäviöt. Käyttöveden tunneittainen lämmitystehon tarve $\phi_{lkv}(t)$ voidaan laskea esimerkiksi dynaamisella simulointiohjelmalla tai se voidaan arvioida luvussa 2.5.1 esitettävän menetelmän avulla. Vuotuinen käyttöveden lämmitysenergiantarve $Q_{lämmitys,lkv}$ voidaan laskea esimerkiksi Suomen RakMk D5:n (2012) mukaisella laskentamenetelmällä.

Mikäli käyttöveden lämmityksessä käytetään lämpöpumpun lisäksi myös aurinkokeräimiä, tulee vuotuisesta lämpöenergian tarpeesta $Q_{lämmitys,lkv}$ sekä tunneittaisesta lämmitystehon tarpeesta $\phi_{lkv}(t)$ vähentää aurinkokeräimien tuottama lämmitysenergia sekä lämmitysteho, joka pystytään hyödyntämään käyttöveden lämmityksessä. Aurinkokeräinten tuottama tunneittainen lämmitysteho voidaan laskea dynaamisella simulointiohjelmalla, esimerkiksi (IDA-ICE 4.5, TRNSYS 17) ottaen huomioon käyttöveden tunneittainen kulutus simuloitavassa rakennuksessa.

2.3.3 Tilojen ja ilmanvaihdon lämmitystarve

Rakennuksen lämmitystarpeen osalta tarvitaan seuraavia lähtötietoja, kun:

- Lämpöpumppua käytetään tilojen lämmitykseen:
  - Tilojen lämmityksen tunneittainen lämmitystehon tarve $\phi_{tilat}(t)$, kW.
  - Tilojen lämmityksen vuotuinen lämpöenergian tarve $Q_{lämmitys,tilat}$, kWh/v, mikäli tunneittaista lämmitystehon tarvetta $\phi_{tilat}(t)$ ei tunneta.

- Lämpöpumppua käytetään tilojen ja ilmanvaihdon jälkilämmitykseen:
  - Tilojen ja ilmanvaihdon lämmityksen yhteenlaskettu tunneittainen lämmitystehon tarve $\phi_{tilat,iv}(t)$, kW.
  - Tilojen ja ilmanvaihdon jälkilämmityksen yhteenlaskettu vuotuinen lämpöenergian tarve $Q_{lämmitys,tilat,iv}$, kWh/v, mikäli tunneittaista lämmitystehon tarvetta $\phi_{tilat,iv}(t)$ ei tunneta.

Tilojen ja ilmanvaihdon lämmitysenergian sekä tehon tarve sisältää lämmön luovutuksen, jakelun ja varastoinnin häviöt. Vuotuinen tilojen ja ilmanvaihdon lämmitysenergian tarve voidaan laskea esimerkiksi Suomen RakMk D5:n (2012) mukaisella laskentamenetelmällä. Tilojen ja ilmanvaihdon tunneittainen lämmitystehon tarve lasketaan ensisijaisesti dynaamisella simulointiohjelmalla tai se voidaan arvioida luvussa 2.5.2 esitettävän menetelmän avulla.

Jatkossa ilmanvaihdon lämmitystä ei yksinkertaisuuden vuoksi mainita erikseen, vaikka lämpöpumppua käytettäisiin sekä tilojen että ilmanvaihdon lämmitykseen, vaan julkaisun tekstissä ja kaavoissa mainitaan vain tilojen lämmitys. Laskentamenetelmää voidaan kuitenkin täyttää suoraan myös tapauksiin, joissa lämpöpumppu lämmittää tilojen ohella myös ilmanvaihdon tuloilmaa käyttämällä tilojen lämmitysenergian $Q_{lämmitys,tilat}$ ja tehon $\phi_{tila}(t)$ sijaan tilojen ja ilmanvaihdon yhteenlaskettua lämmitysenergiaa $Q_{lämmitys,tilat,iv}$ ja tehoa $\phi_{tila,iv}(t)$.

Mikäli aurinkokeräimiä käytetään lämpöpumpun ohella tilojen lämmitykseen, tulee vuotuisesta lämpöenergian tarpeesta $Q_{lämmitys,tilat}$ sekä tunneittaisesta lämmitystehon tarpeesta $\phi_{tila}(t)$ vähentää aurinkokeräimien tuottama lämmitysenergia sekä lämmitysteho, jotka lasketaan esim. kohdassa 2.3.2 mainituilla menetelmillä.

Mikäli lämpöpumppu on kytketty vesikiertoiseen lämmönjakoverkostoon, tarvitaan lämmönjakoverkoston osalta seuraavia lähtötietoja:

- Menoveden maksimilämpötila $T_{mv,max}$ (°C) mitoitusulkolämpötilalla $T_{mit,}$ °C.
- Rakennuksen lämmityksen mitoitusulkolämpötila $T_{mit,}$ °C.
- Menoveden minimilämpötila $T_{mv,min}$, °C.
- Ulkolämpötila $T_{ulko,mv,min}$, jolla ja jota korkeammilla ulkolämpötiloilla menoveden lämpötila saa arvon $T_{mv,min.}$

2.3.4 Lämpöpumppu

Lähtötietoina käytetään lämpöpumpun tuotetietoja, jotka on mitattu esimerkiksi standardien SFS-EN 14511-3, SFS-EN 16147 tai SFS-EN 14825 mukaisesti.

Laskennassa tarvitaan seuraavia lähtötietoja:

- On-off-tyyppisen lämpöpumpun mitattu lämpökerroin COP_N vähintään yhdessä testauspisteessä.
- Tehosäätöisen lämpöpumpun mitattu lämpökerroin COP_N maksimiteholla sekä vähintään yhdessä osatehoa vastaavassa testauspisteessä.

- Ilma-ilma-, ilma-vesi- ja maalämpöpumpun tuottama maksimilämmitysteho $P_{LP,max}$ (kW) niissä testauspisteissä, joissa COP_N on mitattu.
- Niiden testauspisteiden lämpötilat, joissa COP_N on mitattu.
- Korkein lämpötila, $T_{LP,\ max}$ (°C), johon lämpöpumppu pystyy lämmittämään vettä ilman lisälämmityksen käyttöä.

Testauspisteiden lämpötilatasot on määritetty esim. standardissa SFS-EN 14511-2. Em. lähtötiedot annetaan lämpöpumpputyypistä riippuen vähintään yhdessä testauspisteessä, joita ovat esimerkiksi

- ilma-ilmalämpöpumppu: T_{ulko} /$T_{sisä}$ = (+7/+20 °C)
- ilma-vesilämpöpumppu: T_{ulko}/T_{mv} = (+7/+35 °C)
- maalämpöpumppu: T_{liuos}/T_{mv} = (0/+35 °C)
- poistoilmalämpöpumppu: $T_{sisä}$/T_{mv} = (+20/35 °C),

missä

T_{ulko}	ulkoilman lämpötila, °C
$T_{sisä}$	sisäilman lämpötila, °C
T_{mv}	lauhduttimelta lämmönjakoverkostoon virtaavan menoveden lämpötila, °C
T_{liuos}	lämmönkeruupiiristä höyrystimelle virtaavan liuoksen lämpötila, °C.

Mikäli lähtötietoja ei ole saatavissa em. testauspisteissä, voidaan käyttää myös muita esim. standardissa SFS-EN 14511-2 mainittuja testauspisteitä. On syytä korostaa, että laskentatulos on sitä luotettavampi, mitä useammassa testauspisteessä mitattuja lähtötietoja laskennassa käytetään.

Lisäksi tarvitaan seuraavia lämpöpumppukohtaisia lähtötietoja:

- Ilma-ilmalämpöpumppu:
  - Lämpöpumpun sisäänpuhalluslämpötila T_{sp} (°C) (ilman lämpötila, johon lämpöpumppu lämmittää lauhduttimen läpi virtaavan ilman) tai tilojen lämmityksen asetusarvo $T_{sisä}$ (°C), mikäli sisäänpuhalluslämpötilaa ei tunneta.
  - Valmistajan suosituksen mukainen matalin ulkolämpötila $T_{ulko,min}$ (°C), jossa lämpöpumppua voidaan käyttää.
- Ilma-vesilämpöpumppu:
  - Valmistajan suosituksen mukainen matalin ulkolämpötila $T_{ulko,min}$ (°C), jossa lämpöpumppua voidaan käyttää.

- Maalämpöpumppu:
  - Lämmönkeruupiiristä höyrystimelle tulevan liuoksen lämpötila T_{liuos} (°C). Mikäli liuoksen lämpötilasta on käytettävissä tarkempaa esim. lämmönkeruupiirin mitoitusohjelmistolla laskettua tapauskohtaista tietoa, voidaan liuoksen lämpötilan aikariippuvuus ottaa laskennassa huomioon käyttäen keruupiiristä tulevan liuoksen tunneittaista lämpötilaa. Mikäli tarkempaa tietoa ei ole käytettävissä, voidaan laskennassa käyttää kuukauden tai vuoden keskimääräistä keruupiiristä tulevan liuoksen lämpötilaa.
- Poistoilmalämpöpumppu:
  - Lämpöpumpun höyrystimelle tuleva rakennuksen poistoilmavirta Q_{poisto}, m³/s.
  - Poistoilman lämpötila $T_{poisto}(t)$, °C. Lähtötietona voidaan käyttää esimerkiksi dynaamisella simulointiohjelmalla laskettua tunneittaista poistoilman lämpötilaa. Muussa tapauksessa poistoilman lämpötilana voidaan käyttää tilojen lämmityksen asetusarvon mukaista vakiolämpötilaa $T_{sisä}$, °C.
  - Poistoilman absoluuttinen kosteus x_{poisto} (kg/kg). Mikäli laskennassa käytetään tunneittaista poistoilman lämpötilaa, tulee poistoilman absoluuttinen kosteus x_{poisto} määritellä myös tunneittain. Tunneittainen absoluuttinen kosteus voidaan laskea esimerkiksi dynaamisilla simulointiohjelmilla. Mikäli absoluuttista kosteutta ei tunneta, se voidaan laskea esimerkiksi liitteessä 2 esitetyllä menetelmällä käyttäen lähtötietona poistoilman suhteellista kosteutta sekä lämpötilaa.
  - Lämpöpumppuvalmistajan ilmoittama jäteilman minimilämpötila höyrystimen jälkeen $T_{jäte,min,}$ °C.

2.4 Laskentatulokset

- Laskentamenetelmä antaa seuraavat tulokset:
- Lämpöpumpun tuottama lämmitysenergia.
- Lämpöpumpun sähköenergian kulutus.
- Lämpöpumpun SPF-luku.
- Lisälämmitysenergia, mikäli lämpöpumppu ei pysty tuottamaan kaikkea tarvittavaa lämmitysenergia.

2.5 Laskenta

2.5.1 Käyttöveden lämmitystehon tarve

Laskennassa tulee käyttää ensisijaisesti tunneittaisen käyttöveden kulutuksen avulla laskettua tunneittaista lämmitystehon tarvetta $\phi_{lkv}(t)$.

Mikäli rakennuksen käyttöveden tunneittaista lämmitystehon tarvetta $\phi_{lkv}(t)$ ei tunneta, se voidaan arvioida esimerkiksi laskennan lähtötiedoissa määritetyn vuotuisen käyttöveden lämmitysenergian tarpeen $Q_{lämmitys,lkv}$ (Suomen RakMk D5, 2012) avulla, mikäli lämmitystehon tarve oletetaan vakioksi

$$\phi_{lkv} = \frac{Q_{lämmitys,lkv}}{t_{lkv}} \tag{1}$$

missä

t_{lkv}	käyttöveden käyttöaika vuodessa, h.

Mikäli käyttöveden lämpötila T_{lkv} on suurempi kuin lämpötila $T_{LP,max}$, johon lämpöpumppu pystyy lämmittämään käyttövettä ilman lisälämmitystä, vähennetään lämpötilarajoituksesta johtuva lisälämmitystarpeen osuus käyttöveden kulutuksen tai kaavan (1) avulla lasketusta lämmitystehontarpeesta kaavan (2) avulla. Tällöin käyttöveden lämmitystehontarpeena, joka pyritään kattamaan lämpöpumpun avulla, käytetään kaavalla (2) korjattua tehontarvetta $\phi_{lkv}(t)$.

$$\phi_{lkv}(t) = \phi_{lkv}(t)\left(1 - \frac{T_{lkv} - T_{LP,max}}{T_{lkv} - T_{kv}}\right) \tag{2}$$

missä

T_{lkv}	käyttöveden lämpötila, °C
$T_{LP,max}$	lämpötila, johon lämpöpumppu pystyy lämmittämään vettä, °C
T_{kv}	kylmän käyttöveden lämpötila, °C.

2.5.2 Tilojen lämmityksen tehontarve

Mikäli rakennuksen tilojen tunneittaista lämmitystehon tarvetta $\phi_{tilat}(t)$ ei tunneta, se voidaan laskea laskennan lähtötiedoissa määritetyn vuotuisen tilojen lämmitysenergian tarpeen $Q_{lämmitys,tilat}$ avulla käyttäen seuraavaa kaavaa:

$$\phi_{tilat}(t) = \frac{Q_{lämmitys,tilat}}{S_{Ts}}\left(T_{Ts} - T_{ulko}(t)\right), \text{ kun } T_{Ts} > T_{ulko}. \quad (3)$$

$$\phi_{tilat}(t) = 0, \text{ kun } T_{Ts} \leq T_{ulko}.$$

missä

S_{Ts}	astetuntiluku, Kh
T_{Ts}	astetuntilukua vastaava sisälämpötila, °C
$T_{ulko}(t)$	tunnittainen ulkoilman lämpötila, °C.

Tässä laskentamenetelmässä tilojen lämmitystehon tarpeena tai kaavassa (3) käytettävänä tilojen lämmitysenergian tarpeena käytetään vain niiden tilojen lämmitystehontarvetta tai lämmitysenergiaa, jotka ovat ilma-ilmalämpöpumpun vaikutuspiirissä.

Astetuntiluku S_{Ts} voidaan laskea seuraavan kaavan avulla käyttäen sisälämpötilan T_{Ts} arvona esimerkiksi 17 °C, jolloin sisäisten lämpökuormien oletetaan lämmittävän tiloja lämmityksen asetusarvoon asti:

$$S_{Ts} = \sum \left(T_{Ts} - T_{ulko}(t)\right) \cdot t_{aika-askel} \quad (4)$$

missä

$T_{ulko}(t)$	tunneittainen ulkoilman lämpötila,°C
$t_{aika-askel}$	laskennan aika-askel, h.

2.5.3 Lämmönjakoverkoston lämpötila ja sisäänpuhalluslämpötila

Menoveden tunneittainen lämpötila $T_{mv}(t)$ voidaan laskea seuraavien kaavojen avulla:

$$T_{mv}(t) = T_{mv,max}, \text{ kun } T_{ulko}(t) \leq T_{mit} \quad (5)$$

$$T_{mv}(t) = k \cdot T_{ulko}(t) + b, \text{ kun } T_{ulko} \text{ on välillä } T_{mit} < T_{ulko}(t) < T_{ulko,mv,min} \quad (6)$$

$$T_{mv}(t) = T_{mv,min}, \text{ kun } T_{ulko}(t) \geq T_{ulko,mv,min} \quad (7)$$

joissa

$T_{mv,max}$	menoveden maksimilämpötila mitoitusulkolämpötilalla, °C
k	säätökäyrän kulmakerroin, -

$T_{ulko}(t)$	tunnittainen ulkoilman lämpötila, °C
b	säätökäyrän vakiotermi, °C
$T_{mv,min}$	menoveden minimilämpötila, °C.

Kulmakerroin k voidaan laskea seuraavan kaavan avulla:

$$k = \frac{T_{mv,\max} - T_{mv,\min}}{T_{mit} - T_{ulko,mv,\min}} \quad (8)$$

missä

$T_{mv,max}$	menoveden maksimilämpötila mitoitusulkolämpötilalla, °C
$T_{mv,min}$	menoveden minimilämpötila, °C
T_{mit}	rakennuksen lämmityksen mitoitusulkolämpötila, °C
$T_{ulko,mv,min}$	menoveden min. lämpötilaa vastaava ulkolämpötila, °C.

Säätökäyrän vakiotermi b voidaan laskea seuraavan kaavan avulla:

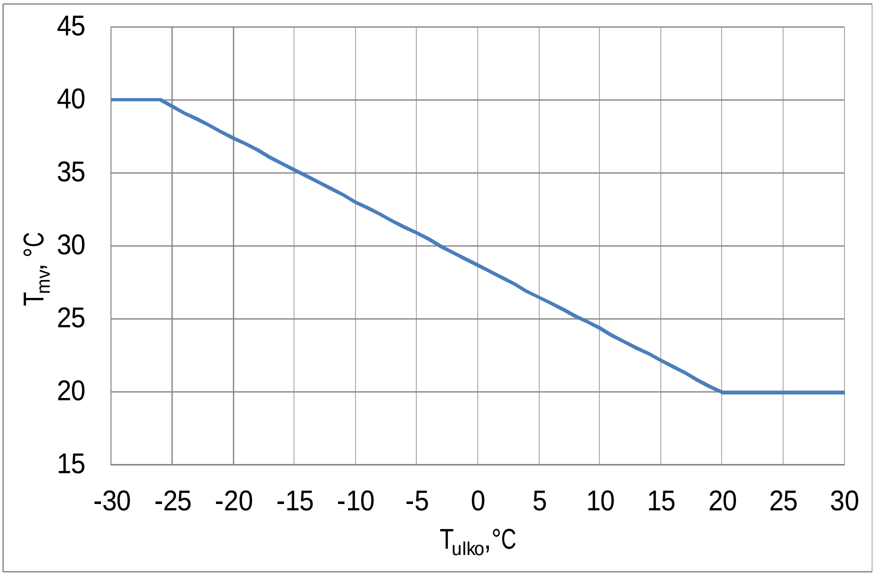
$$b = T_{mv,\max} - k \cdot T_{mit} \quad (9)$$

missä

k	säätökäyrän kulmakerroin, -.

Mikäli kaavoilla (5–7) laskettu menoveden lämpötilan $T_{mv}(t)$ arvo on suurempi kuin lämpötila $T_{LP,max}$, johon lämpöpumppu pystyy lämmittämään vettä ilman lisälämmitystä, käytetään laskennassa tämän lämpötilarajan ylittävien menoveden lämpötilojen arvoina lämpötilaa $T_{LP,max}$.

Kuvassa A1 on esimerkki menoveden lämpötilasta eri ulkolämpötilan arvoilla.



Kuva A1. Esimerkki menoveden lämpötilan säätökäyrästä.

Mikäli ilma-ilmalämpöpumpun sisäänpuhalluslämpötilaa T_{sp} (°C) (ilman lämpötila, johon lämpöpumppu lämmittää lauhduttimen läpi virtaavan ilman) ei tunneta, se voidaan arvioida tilojen lämmityksen asetusarvon $T_{sisä}$ perusteella. Sisäänpuhalluslämpötilan voidaan olettaa olevan 15 °C lämpimämpi kuin tilojen lämmityksen asetusarvon, ellei tarkempaa tietoa ole käytettävissä.

2.5.4 Lämpöpumpun lämpökerroin

Lämpöpumpun tunneittainen lämpökerroin COP(t) lasketaan seuraavan kaavan avulla:

$$COP(t) = f_T(t) \cdot COP_T(t) \qquad (10)$$

missä

$f_T(t)$	kompressorin häviökerroin, -
$COP_T(t)$	lämpöpumpun tunnittainen teoreettinen lämpökerroin, -.

Osatehon vaikutus tehosäätöisen lämpöpumpun lämpökertoimeen otetaan huomioon luvussa 2.5.5 esitettävän menetelmän mukaan.

Kaavassa (10) käytettävä, lämpöpumpun lämmitysprosessin häviöt huomioon ottava häviökerroin $f_T(t)$ lasketaan kaavan (11) avulla:

$$f_T(t) = \frac{COP_N}{COP_T} \tag{11}$$

missä

COP_N	lämpöpumpun mitattu lämpökerroin, -
COP_T	lämpöpumpun teoreettinen lämpökerroin, -.

Mikäli lämpöpumpun mitattu lämpökerroin COP_N tunnetaan vain yhdessä testauspisteessä, oletetaan, että häviökerroin on vakio koko laskentajakson (esim. vuosi) aikana. Mikäli mitattu lämpökerroin tunnetaan useammassa testauspisteessä, voidaan häviökerroin laskea kullekin testauspisteelle kaavan (11) avulla. Testauspisteiden lämpötilojen väliset häviökertoimen arvot $f_T(t)$ voidaan interpoloida lineaarisesti paloittain liitteessä 1 esitetyn menetelmän avulla. Tunnettuja testauspisteitä matalammilla tai korkeammilla lämpötiloilla häviökertoimen arvona voidaan käyttää lähimmän tunnetun häviökertoimen arvoa.

Kaavassa (10) käytettävä tunneittainen teoreettinen lämpökerroin $COP_T(t)$ lasketaan kaavan (12) avulla käyttäen jokaisella aika-askeleella määritettäviä lämmönlähteen $T_{LL}(t)$ ja rakennuksen lämmitysjärjestelmän lämpötiloja $T_{LJ}(t)$. Kaavassa (11) käytettävä lämpöpumpun teoreettinen lämpökerroin COP_T lasketaan testauspistettä vastaavilla lämmönlähteen ja rakennuksen lämmitysjärjestelmän vakiolämpötiloilla kaavan (12) avulla:

$$COP_T = \frac{T_{LJ}}{T_{LJ} - T_{LL}} \tag{12}$$

missä

T_{LJ}	rakennuksen lämmitysjärjestelmän lämpötila, K
T_{LL}	lämmönlähteen lämpötila, K.

Käyttöveden lämmityksen lämpökertoimen laskennassa käytetään lämmitysjärjestelmän lämpötilan arvona $T_{LJ}(t)$ käyttöveden lämpötilaa T_{lkv}. Vastaavasti tilojen lämmityksen lämpökertoimen laskennassa käytetään menoveden lämpötilaa $T_{mv}(t)$ vesikiertoisen lämmönjakojärjestelmän tapauksessa ja sisäänpuhalluslämpötilaa T_{sp} ilma-ilmalämpöpumpun tapauksessa (ks. luku 2.5.3). Mikäli käyttöveden tai menoveden lämpötilan arvo on suurempi kuin korkein lämpötila $T_{LP,max}$, johon lämpöpumppu pystyy lämmittämään vettä ilman lisälämmitystä, käytetään kaavassa (12) lämpötilaa $T_{LP,max}$.

Kaavassa (12) lämmönlähteen lämpötilana $T_{LL}(t)$ käytetään lämpöpumpputyypistä riippuen seuraavia lämpötiloja:

- Ilma-ilma- ja ilma-vesilämpöpumppu: ulkoilman lämpötila $T_{ulko}(t)$.

- Maalämpöpumppu: höyrystimelle virtaavan lämmönkeruunesteen lämpötila $T_{liuos}(t)$.
- Poistoilmalämpöpumppu: jäteilman minimilämpötila $T_{jäte,min}$.

2.5.5 Lämpöpumpun lämpökerroin osateholla

Osatehokuormituksen vaikutus tehosäätöisen lämpöpumpun lämpökertoimeen voidaan ottaa huomioon, mikäli mitattu lämpökertoimen arvo on tiedossa sekä maksimiteholla että vähintään yhdessä osatehoa vastaavassa testauspisteessä.

Tehosäätöisen lämpöpumpun kuormituksen tunneittaiset tehosuhteet $\beta_{LP,lkv}(t)$ ja $\beta_{LP,tilat}(t)$ lasketaan käyttöveden ja ilojen lämmitykselle seuraavien kaavojen avulla:

$$\beta_{LP,lkv}(t) = \frac{\phi_{lkv}(t)}{\phi_{LP,lkv}(t)} \tag{13}$$

missä

$\phi_{lkv}(t)$	käyttöveden tunneittainen lämmitystehon tarve, kW (ks. luku 2.5.1)
$\phi_{LP,lkv}(t)$	lämpöpumpun käyttöveden tunneittainen maksimilämmitysteho, kW (ks. kaava 15)

$$\beta_{LP,tilat}(t) = \frac{\phi_{tilat}(t)}{\phi_{LP,tilat}(t)} \tag{14}$$

missä

$\phi_{tilat}(t)$	tilojen tunneittainen lämmitystehon tarve, kW (ks. luku 2.5.2)
$\phi_{LP,tilat}(t)$	lämpöpumpun tilojen tunneittainen maksimilämmitysteho, kW (ks. kaava 22).

Tunneittaiset lämpökertoimen arvot käyttöveden ja tilojen lämmityksessä ($COP_{lkv}(t)$ ja $COP_{tilat}(t)$) voidaan interpoloida tunnettujen tehosuhteiden välillä esimerkiksi liitteessä 1 esitetyn menetelmän avulla. Jos mitatut lämpökertoimen arvot tunnetaan esimerkiksi tehosuhteen arvoilla 1,0 ja 0,5, lasketaan lämpökertoimille ensin lämpötilakorjaus kaavan (10) avulla, minkä jälkeen tehdään osatehokorjaus interpoloimalla tulokseksi saadut lämpökertoimen arvot tehosuhteen arvojen 0,5 ja 1,0 välillä. Mikäli tehosuhde on hetkellisesti pienempi kuin 0,5, voidaan laskennassa käyttää tällöin tehosuhdetta 0,5 vastaavaa lämpökertoimen arvoa.

2.5.6 Ilma-vesi- ja maalämpöpumpun lämmitysteho käyttöveden lämmityksessä

Mikäli on-off-säätöisen tai tehosäätöisen ilma-vesi- tai maalämpöpumpun maksimilämmitysteho $\phi_{LP,max}$ tunnetaan vain yhdessä testauspisteessä, lasketaan näiden lämpöpumpputyyppien tunneittainen käyttöveden lämmityksen maksimilämmitysteho $\phi_{LP,lkv}(t)$ seuraavan kaavan avulla:

$$\phi_{LP,lkv}(t) = \phi_{LP,max} \frac{COP_{lkv}(t)}{COP_N} \tag{15}$$

missä

$\phi_{LP,max}$	lämpöpumpun maksimilämmitysteho testauspisteessä, kW
$COP_{lkv}(t)$	lämpöpumpun tunneittainen lämpökerroin käyttöveden lämmityksessä, -
COP_N	lämpöpumpun mitattu lämpökerroin testauspisteessä, -.

Lämpökerroin $COP_{lkv}(t)$ lasketaan kaavalla (10) käyttäen lähtötietona määritettyä käyttöveden lämpötilaa T_{lkv}. Mikäli käyttöveden lämpötila on suurempi kuin lämpötila $T_{LP,max}$, johon lämpöpumppu pystyy lämmittämään käyttövettä ilman lisälämmitystä, lasketaan lämpökerroin $COP_{lkv}(t)$ käyttäen lämpötilaa $T_{LP,max}$.

Tehosäätöisen lämpöpumpun tunneittainen lämmitysteho $\phi_{LP,lkv}(t)$ on yhtä suuri kuin käyttöveden tunneittainen lämmitystehon tarve $\phi_{lkv}(t)$ (ks. luku 2.5.1), jos $\phi_{lkv}(t)$ on pienempi kuin $\phi_{LP,lkv}(t)$. Jos käyttöveden lämmitystehontarve $\phi_{lkv}(t)$ on suurempi kuin kaavalla (15) laskettu lämmitysteho $\phi_{LP,lkv}(t)$, käytetään tehosäätöisen lämpöpumpun tunneittaisena käyttöveden lämmitystehona $\phi_{LP,lkv}(t)$ kaavalla (15) laskettua arvoa.

Mikäli on-off-säätöisen tai tehosäätöisen ilma-vesi- tai maalämpöpumpun mitattu maksimilämmitysteho $\phi_{LP,max}$ tunnetaan useassa testauspisteessä, voidaan testauspisteiden väliset maksimitehon arvot interpoloida lineaarisesti paloittain liitteessä 1 esitetyn menetelmän avulla. Interpoloituja arvoja voidaan käyttää testauspisteiden välillä kaavalla (15) lasketun tehon sijaan. Lämpöpumpun maksimilämmitysteho tunnettujen testauspisteiden lämpötiloja matalammilla tai korkeammilla arvoilla voidaan laskea kaavan (15) avulla käyttäen lämmitystehon $\phi_{LP,max}$ arvona lähimmässä tunnetussa testauspisteessä mitattua lämmitystehoa.

2.5.7 Poistoilmapumpun lämmitysteho käyttöveden lämmityksessä

On-off-säätöisen poistoilmalämpöpumpun tunneittainen lämmitysteho $\phi_{LP,lkv}(t)$, joka käytetään käyttöveden lämmitykseen, lasketaan seuraavan kaavan avulla:

$$\phi_{LP,lkv}(t) = \phi_{LPh}(t)\frac{COP_{lkv}(t)}{COP_{lkv}(t)-1} \tag{16}$$

missä

$\phi_{LPh}(t)$	lämpöpumpun höyrystinteho, kW
$COP_{lkv}(t)$	lämpöpumpun tunneittainen lämpökerroin käyttöveden lämmityksessä, -.

Tehosäätöisen poistoilmalämpöpumpun tunneittainen lämmitysteho $\phi_{LP,lkv}(t)$ on yhtä suuri kuin käyttöveden tunneittainen lämmitystehon tarve $\phi_{lkv}(t)$ (ks. luku 2.5.1), jos $\phi_{lkv}(t)$ on pienempi kuin $\phi_{LP,lkv}(t)$. Jos käyttöveden lämmitystehontarve $\phi_{lkv}(t)$ on suurempi kuin kaavalla (16) laskettu lämmitysteho $\phi_{LP,lkv}(t)$, käytetään tehosäätöisen lämpöpumpun tunneittaisena käyttöveden lämmitystehona $\phi_{LP,lkv}(t)$ kaavalla (16) laskettua arvoa.

Kaavassa (16) käytettävä poistoilmalämpöpumpun höyrystinteho lasketaan seuraavalla kaavalla:

$$\phi_{LPh}(t) = Q_{poisto}(t)\cdot\rho\left(h_1(t)-h_2(t)\right) \tag{17}$$

missä

$Q_{poisto}(t)$	poistoilmavirta, m^3/s
ρ	poistoilman tiheys, kg/m^3
$h_1(t)$	poistoilman entalpia (ennen höyrystintä), kJ/kg
$h_2(t)$	jäteilman entalpia (höyrystimen jälkeen), kJ/kg.

Poistoilman ja jäteilman entalpia voidaan laskea seuraavan kaavan avulla:

$$h(t) = c_{pi}\cdot T(t) + x(t)\cdot\left(L_{ho} + c_{ph}\cdot T(t)\right) \tag{18}$$

missä

c_{pi}	ilman ominaislämpökapasiteetti, kJ/kg,K
$T(t)$	poistoilman tai jäteilman lämpötila, °C
$x(t)$	poistoilman tai jäteilman absoluuttinen kosteus, kg/kg
L_{ho}	veden höyrystymislämpö 0 °C lämpötilassa, kJ/kg
c_{ph}	vesihöyryn ominaislämpökapasiteetti kJ/kg,K.

Kaavassa (18) ilman ominaislämpökapasiteetin c_{pi} arvona voidaan käyttää 1,006 kJ/kg,K, vesihöyryn ominaislämpökapasiteetin c_{ph} arvona 1,85 kJ/kg,K ja veden höyrystymislämmön L_{hg} arvona 2502 kJ/kg.

Laskettaessa poistoilman entalpia kaavan (18) avulla, ilman lämpötilan T(t) ja absoluuttisen kosteuden x(t) arvoina käytetään poistoilman arvoja. Mikäli poistoilman absoluuttista kosteutta ei tunneta, se voidaan laskea poistoilman suhteellisen kosteuden $RH_{poisto}(t)$ ja lämpötilan $T_{poisto}(t)$ avulla käyttäen liitteessä 2 esitettyä menetelmää.

Laskettaessa jäteilman entalpia kaavan (18) avulla ilman lämpötilan T(t) arvona voidaan käyttää lämpöpumppuvalmistajan ilmoittaman jäteilman lämpötilan minimiarvoa $T_{jäte}$. Jäteilman absoluuttinen kosteus voidaan laskea liitteessä 2 esitetyllä menetelmällä käyttäen jäteilman suhteellisen kosteuden $RH_{jäte}$ arvona 100 %:a ja jäteilman lämpötilaa $T_{jäte}$.

2.5.8 Käyttöveden lämmitysaika

Aika t_{lkv}, jonka lämpöpumppu lämmittää käyttövettä yhden laskennan aika-askeleen aikana, lasketaan seuraavan kaavan avulla:

$$t_{lkv}(t) = t_{aika-askel} \frac{\phi_{lkv}(t)}{\phi_{LP,lkv}(t)} \text{, kun } \square_{lkv} < \square_{LP,lkv} \tag{19}$$

$$t_{lkv}(t) = t_{aika-askel} \quad \text{, kun } \square_{lkv} \geq \square_{LP,lkv}$$

missä

aika-askel	laskennan aika-askeleen pituus, h
$\phi_{lkv}(t)$	käyttöveden lämmitystehon tarve, kW (ks. luku 2.5.1)
$\phi_{LP,lkv}(t)$	lämpöpumpun lämmitysteho, joka käytetään käyttöveden lämmitykseen, kW (ks. luvut 2.5.6 ja 2.5.7).

2.5.9 Käyttöveden lämmitysenergia

Lämpöpumpun tuottama lämmitysenergia käyttöveden lämmitykseen yhden laskennan aika-askeleen aikana lasketaan seuraavan kaavan avulla:

$$q_{LP,lkv}(t) = \phi_{LP,lkv}(t) \cdot t_{lkv} \tag{20}$$

missä

$\phi_{LP,lkv}(t)$	lämpöpumpun lämmitysteho käyttöveden lämmityksessä, Kw (ks. luvut 2.5.6 ja 2.5.7)
t_{lkv}	aika, jonka lämpöpumppu lämmittää käyttövettä aika-askeleen aikana, h (ks. luku 2.5.8).

Ilma-vesilämpöpumpun osalta lämpöpumpulla tuotettu tunneittainen lämmitysenergia $q_{LP,lkv}(t)$ lasketaan vain niiden aika-askeleiden osalta, joiden aikana ulkolämpötila $T_{ulko}(t)$ on suurempi kuin lähtötiedoissa määritetty lämpöpumpun matalin käyttölämpötila $T_{ulko,min}$.

Lämpöpumpun tuottama vuotuinen lämmitysenergia käyttöveden lämmitykseen $Q_{LP,lkv}$ on tunneittaisen käyttöveden lämmitysenergian $q_{LP,lkv}(t)$ summa seuraavan kaavan mukaisesti:

$$Q_{LP,lkv} = \sum q_{LP,lkv}(t). \tag{21}$$

2.5.10 *Ilma-ilma-, ilma-vesi- ja maalämpöpumpun lämmitysteho tilojen lämmityksessä*

Mikäli on-off-säätöisen tai tehosäätöisen ilma-ilma-, ilma-vesi- tai maalämpöpumpun maksimilämmitysteho $\phi_{LP,max}$ tunnetaan vain yhdessä testauspisteessä, lasketaan näiden lämpöpumpputyyppien tunneittainen tilojen lämmityksen maksimilämmitysteho $\phi_{LP,tilat}(t)$ seuraavan kaavan avulla:

$$\phi_{LP,tilat}(t) = \phi_{LP,max} \frac{COP_{tilat}(t)}{COP_N} \tag{22}$$

missä

$\phi_{LP,max}$	lämpöpumpun maksimilämmitysteho testauspisteessä, kW
$COP_{tilat}(t)$	lämpöpumpun tunneittainen lämpökerroin tilojen lämmityksessä, -
COP_N	lämpöpumpun mitattu lämpökerroin testauspisteessä, -.

Lämpökerroin $COP_{tilat}(t)$ lasketaan kaavalla (10) käyttäen lähtötietona tunneittaista lämmönlähteen lämpötilaa sekä lämmönjakoverkoston menoveden lämpötilaa $T_{vm}(t)$ tai ilma-ilmalämpöpumpun tapauksessa sisäänpuhalluslämpötilaa T_{sp}. Mikäli menoveden lämpötila on suurempi kuin lämpötila $T_{LP,max}$, johon lämpöpumppu pystyy lämmittämään vettä ilman lisälämmitystä, lasketaan lämpökerroin $COP_{tilat}(t)$ käyttäen lämpötilaa $T_{LP,max}$.

Tehosäätöisen lämpöpumpun tunneittainen lämmitysteho $\phi_{LP,tilat}(t)$ on yhtä suuri kuin tilojen lämmitystehon tarve $\phi_{tilat}(t)$ (ks. luku 2.5.2), jos $\phi_{tilat}(t)$ on pienempi kuin $\phi_{LP,tilat}(t)$. Jos tilojen lämmitystehontarve $\phi_{tilat}(t)$ on suurempi kuin kaavalla (22) laskettu lämmitysteho $\phi_{LP,tilat}(t)$, käytetään tehosäätöisen lämpöpumpun tunneittaisena tilojen lämmitystehon arvona kaavalla (22) laskettua lämmitystehoa.

Mikäli on-off-säätöisen tai tehosäätöisen ilma-ilma-, ilma-vesi- tai maalämpöpumpun mitattu maksimilämmitysteho $\phi_{LP,max}$ tunnetaan useassa testauspisteessä, voidaan testauspisteiden väliset maksimitehon arvot interpoloida lineaarisesti paloittain liitteessä 1 esitetyn menetelmän avulla. Interpoloituja arvoja voidaan käyttää testauspisteiden välillä kaavalla (22) lasketun tehon sijaan. Lämpöpumpun maksimilämmitysteho tunnettujen testauspisteiden lämpötiloja matalammilla tai korkeammilla arvoilla voidaan laskea kaavan (22) avulla käyttäen lämmitystehon $\phi_{LP,max}$ arvona lähimmässä tunnetussa testauspisteessä mitattua lämmitystehoa.

2.5.11 Poistoilmalämpöpumpun lämmitysteho tilojen lämmityksessä

On-off-säätoisen poistoilmalämpöpumpun tunneittainen lämmitysteho $\phi_{LP,tilat}(t)$, joka käytetään tilojen lämmitykseen, lasketaan seuraavan kaavan avulla:

$$\phi_{LP,tilat}(t) = \phi_{LPh}(t)\frac{COP_{tilat}(t)}{COP_{tilat}(t)-1} \tag{23}$$

missä

$\phi_{LPh}(t)$	lämpöpumpun höyrystinteho, kW
$COP_{tilat}(t)$	lämpöpumpun tunneittainen lämpökerroin tilojen lämmityksessä, -.

Kaavassa (23) käytettävä poistoilmalämpöpumpun höyrystinteho on yhtä suuri kuin luvussa (2.5.7) laskettu poistoilmalämpöpumpun höyrystinteho.

Tehosäätöisen poistoilmalämpöpumpun tunneittainen lämmitysteho $\phi_{LP,tilat}(t)$ on yhtä suuri kuin tilojen lämmitystehon tarve $\phi_{tilat}(t)$ (ks. luku 2.5.2), jos $\phi_{tilat}(t)$ on pienempi kuin $\phi_{LP,tilat}(t)$. Jos tilojen lämmitystehontarve $\phi_{tilat}(t)$ on suurempi kuin kaavalla (23) laskettu lämmitysteho $\phi_{LP,tilat}(t)$ käytetään tehosäätöisen lämpöpumpun tunneittaisena tilojen lämmitystehon arvona kaavalla (23) laskettua lämmitystehoa.

2.5.12 Tilojen lämmitysaika

Aika, joka on käytettävissä tilojen lämmitykseen yhden aika-askeleen aikana $t_{tilat,max}(t)$, lasketaan seuraavan kaavan avulla:

$$t_{tilat,max}(t) = t_{aika-askel} - t_{lkv}(t) \tag{24}$$

missä

$t_{aika-askel}$	laskennan aika-askeleen pituus, h
$t_{lkv}(t)$	aika, jonka lämpöpumppu lämmittää käyttövettä aika-askeleen aikana, h (ks. luku 2.5.8).

Kaavaa (24) voidaan käyttää käyttövettä ja tiloja lämmittävän lämpöpumpun tapauksessa. Pelkästään tiloja lämmittävän lämpöpumpun tapauksessa tilojen lämmitykseen voidaan käyttää koko aika-askeleen pituus.

2.5.13 Tilojen lämmitysenergia

Tilojen lämmitykseen tarvittava lämmitysenergia $q_{tilat}(t)$ aika-askeleen aikana lasketaan seuraavan kaavan avulla:

$$q_{tilat}(t) = \phi_{tilat}(t) \cdot t_{aika-askel} \tag{25}$$

missä

$\phi_{tilat}(t)$	tilojen lämmityksen tunneittainen tehontarve, kW (ks. luku 2.5.2)
$t_{aika-askel}$	laskennan aika-askeleen pituus, h.

Maksimienergiamäärä $q_{LP,tilat,max}(t)$, jonka lämpöpumppu pystyy enimmillään tuottamaan aika-askeleen aikana tilojen lämmitykseen, lasketaan seuraavan kaavan avulla:

$$q_{LP,tilat,max}(t) = \phi_{LP,tilat}(t) \cdot t_{tilat,max}(t) \tag{26}$$

missä

$\phi_{LP,tilat}(t)$	lämpöpumpun lämmitysteho tilojen lämmityksessä, kW (ks. luvut 2.5.10 ja 2.5.11)
$t_{tilat,max}(t)$	aika, jonka lämpöpumppu voi enimmillään lämmittää tiloja aika-askeleen aikana, h (ks. luku 2.5.12).

Pelkästään tiloja lämmittävän lämpöpumpun tapauksessa $t_{tilat,max}(t)$ on yhtä suuri kuin aika-askeleen pituus.

Lämpöpumpun tuottama tilojen lämmitysenergia $q_{LP,tilat}(t)$ yhden laskennan aika-askeleen aikana on yhtä suuri kuin kaavalla (26) laskettu maksimilämmitysenergia $q_{LP,tilat,max}(t)$, jos tilojen lämmitysenergian tarve $q_{tilat}(t)$ (kaava 25) on suurempi kuin $q_{LP,tilat,max}(t)$ kaavan (27) mukaisesti. Muussa tapauksessa lämpöpumpun tilojen lämmitykseen tuottama energia on yhtä suuri kuin tilojen lämmitysenergian tarve kaavan (28) mukaisesti:

$$q_{LP,tilat}(t) = q_{LP,tilat,max}(t) \text{, kun } q_{tilat} > q_{LP,tilat,max} \tag{27}$$

$$q_{LP,tilat}(t) = q_{tilat}(t) \text{, kun } q_{tilat} \le q_{LP,tilat,max} \tag{28}$$

joissa

$q_{LP,tilat,max}(t)$ maksimilämmitysenergia, jonka lämpöpumppu voi tuottaa aika-askeleen aikana, kWh

$q_{tilat}(t)$ tilojen lämmitysenergian tarve aika-askeleen aikana, kWh.

Ulkoilmalämpöpumppujen osalta lämpöpumpulla tuotettu lämmitysenergia $q_{LP,tilat}(t)$ lasketaan vain niiden aika-askeleiden osalta, joiden aikana ulkolämpötila $T_{ulko}(t)$ on suurempi kuin lähtötiedoissa määritetty lämpöpumpun matalin käyttölämpötila $T_{ulko,min}$.

Lämpöpumpun tuottama lämmitysenergia tilojen lämmitykseen koko vuoden aikana $Q_{LP,tilat}$ lasketaan laskemalla yhteen jokaisen aika-askeleen aikana tuotettu tilojen lämmitysenergia $q_{LP,tilat}(t)$:

$$Q_{LP,tilat} = \sum q_{LP,tilat}(t) \tag{29}$$

2.5.14 Lämpöpumpun sähköenergia

Seuraavan kaavan avulla lasketaan lämpöpumpun tunneittainen sähköenergia $w_{LP}(t)$, joka sisältää kompressorin sekä niiden lämpöpumpun apulaitteiden kuluttaman sähköenergian, jotka sisältyvät testaustilanteessa lämpöpumpun sähkönkulutukseen:

$$w_{LP}(t) = \frac{q_{LP,lkv}(t)}{COP_{lkv}(t)} + \frac{q_{LP,tilat}(t)}{COP_{tilat}(t)} \tag{30}$$

missä

$q_{LP,lkv}(t)$ lämpöpumpun tuottama tunneittainen käyttöveden lämmitysenergia, kWh

$COP_{lkv}(t)$ lämpöpumpun tunneittainen lämpökerroin käyttöveden lämmityksessä, -

$q_{LP,tilat}(t)$ lämpöpumpun tuottama tunneittainen tilojen lämmitysenergia, kWh

$COP_{tilat}(t)$ lämpöpumpun tunneittainen lämpökerroin tilojen lämmityksessä, -

.

Lämpöpumpun sähköenergia W_{LP} vuoden aikana on lämpöpumpun tunneittaisen sähköenergian $w_{LP}(t)$ summa seuraavan kaavan mukaisesti:

$$W_{LP} = \sum w_{LP}(t) \,. \tag{31}$$

2.5.15 Lämpöpumpun apulaitteiden sähköenergia

Standardien SFS-EN 14511-3 ja SFS-EN 14825 mukaisesti mitattuun lämpöpumpun sähkönenergiankulutukseen ja lämpökertoimen COP_N arvoon sisältyvät kompressorin kuluttama sähköenergia, höyrystimen sulatukseen kuluva sähköenergia sekä osa lämpöpumppujen apulaitteiden kuluttamasta sähköenergiasta. Mitattuun apulaitteiden sähköenergiankulutukseen sisältyvät lämpöpumpun kaikkien säätö- ja suojalaitteiden kulutus sekä puhaltimien ja pumppujen kulutus lämpöpumppuyksikön sisällä tapahtuvan ilman tai nesteen siirron osalta. Tällöin lämpöpumpun puhaltimien tai pumppujen sähkönkulutus, joka käytetään ilman tai nesteen siirtoon lämpöpumppuyksikön ulkopuolisessa kanavistossa tai putkistossa, ei sisälly testausolosuhteissa mitattuun sähkönkulutukseen.

Lämpöpumpun puhaltimien ja pumppujen sähköenergiankulutus, joka ei ole mukana em. standardien mukaisesti mitatuissa lämpökertoimien arvoissa, otetaan erikseen huomioon apulaitteiden sähkönkulutuksen W_{apu} avulla lämpöpumpputyypistä riippuen. Poistoilmalämpöpumpun puhaltimien sähköenergiankulutus lämpöpumppuyksikön ulkopuolisen kanaviston osalta lasketaan mukaan apulaitteiden sähköenergian kulutukseen. Vastaavasti lasketaan mukaan maalämpöpumpun lämmönkeruupiirin pumppauksen sähköenergiankulutus lämpöpumppuyksikön ulkopuolisen putkiston osalta. Sen sijaan lämpöpumppuyksikön ulkopuolisen rakennuksen lämmönjakopiirin pumppaukseen käytettävää sähköenergiaa ei lasketa mukaan lämpöpumpun apulaitteiden sähköenergiaan, vaan se otetaan huomioon, kun lasketaan rakennuksen lämmönjakojärjestelmän apulaitteiden sähköenergiankulutusta esimerkiksi laskentamenetelmän Suomen RakMk D5 (2012) mukaisesti.

Lämpöpumpun apulaitteiden kuluttama sähköenergia W_{apu}, joka ei sisälly lämpöpumpun sähkönkulutuksen ja lämpökertoimen COP_N mitattuihin arvoihin, voidaan laskea kaavan (32) avulla:

$$\mathrm{W_{apu}} = \mathrm{P_{apu}} \Delta \mathrm{t} \tag{32}$$

missä

P_{apu}	lämpöpumpun apulaitteiden sähköteho, joka ei sisälly mitattuun lämpökertoimen arvoon, kW
Δt	apulaitteiden käyttöaika laskentajaksolla, h.

Lämpöpumpun apulaitteiden sähköteho P_{apu} voidaan laskea seuraavan kaavan avulla:

$$P_{apu} = \frac{Q \cdot \Delta P_e}{\eta} \tag{33}$$

missä

Q	ilman tai nesteen nimellisvirtaama Q, m³/s
ΔP_e	lämpöpumppuyksikön ulkopuolisen kanaviston tai putkiston staattinen painehäviö, Pa
η	puhaltimien tai pumppujen hyötysuhde,-.

2.5.16 Lämpöpumpun SPF-luku

Lämpöpumpun SPF-luku määritellään seuraavan kaavan avulla:

$$SPF = \frac{Q_{LP,lkv} + Q_{LP,tilat}}{W_{LP} + W_{apu}} \tag{34}$$

missä

$Q_{LP,lkv}$	lämpöpumpun tuottama vuotuinen käyttöveden lämmitysenergia, kWh (ks. luku 2.5.9)
$Q_{LP,tilat}$	lämpöpumpun tuottama vuotuinen tilojen lämmitysenergia, kWh (ks.luku 2.5.13)
W_{LP}	lämpöpumpun kuluttama sähköenergia, kWh (ks. luku 2.5.14)
W_{apu}	lämpöpumpun apulaitteiden kuluttama sähköenergia, kWh (ks. luku 2.5.15).

2.5.17 Lisälämmitysenergia

Mikäli lämpöpumppu ei pysty tuottamaan kaikkea tarvittavaa käyttöveden ja tilojen lämmitysenergiaa, tarvitaan lisälämmitystä. Lisälämmitys voidaan toteuttaa esimerkiksi lämminvesivaraajassa olevalla sähkövastuksella tai muulla lämmitysjärjestelmällä. Vuotuinen lisälämmitysenergian tarve lasketaan seuraavan kaavan avulla:

$$Q_{lisälämmitys} = Q_{lämmitys,lkv} + Q_{lämmitys,tilat} - Q_{LP,lkv} - Q_{LP,tilat} \tag{35}$$

missä

$Q_{lämmitys, lkv}$	käyttöveden lämmityksen lämpöenergian tarve, kWh (ks. luku 2.3.2)
$Q_{lämmitys, tilat}$	tilojen lämmityksen lämpöenergian tarve, kWh (ks. luku 2.3.3)
$Q_{LP, lkv}$	lämpöpumpun tuottama käyttöveden lämmitysenergia, kWh (ks. luku 2.5.9)
$Q_{LP, tilat}$	lämpöpumpun tuottama tilojen lämmitysenergia, kWh (ks. luku 2.5.13).

LÄHTEET:

D3. 2012. Rakennusten energiatehokkuus. Määräykset ja ohjeet 2012. Suomen rakentamismääräyskokoelma, Helsinki, Ympäristöministeriö.

D5. 2012. Rakennuksen energiankulutuksen ja lämmitystehon tarpeen laskenta, ohjeet 2012. Suomen rakentamismääräyskokoelma, Helsinki, Ympäristöministeriö.

IDA-ICE 4.5. http://www.equa-solutions.co.uk/ (viitattu 2.7.2012).

SFS-EN 14511-2. 2011. Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – Part 2: Test conditions.

SFS-EN 14511-3. 2011. Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors – Heating mode – Part 3: Testing and requirements for marking for sanitary hot water units.

SFS-EN 14825. 2012. Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling. Testing and rating at part load conditions and calculation of seasonal performance.

SFS-EN 16147. 2011. Heat pumps with electrically driven compressors. Testing and requirements for marking of domestic hot water units.

Trnsys 17. http://sel.me.wisc.edu/trnsys/ (viitattu 2.7.2012).

LIITE 1

Suureesta X riippuvan suureen A(X) arvo voidaan interpoloida lineaarisesti kahden tunnetun pisteen X_i ja X_{i+1} välillä, kun suureen arvot tunnetaan näissä pisteissä $A(X_i)$ ja $A(X_{i+1})$. Suureen arvot A(X) näiden kahden pisteen välillä voidaan laskea seuraavan kaavan (L1) avulla:

$$A(X) = A(X_i) + \frac{A(X_{i+1}) - A(X_i)}{X_{i+1} - X_i}(X - X_i)\,, \tag{L1}$$

missä

$A(X_i)$	suureen A arvo pisteessä i
$A(X_{i+1})$	suureen A arvo pisteessä i+1
X_i	suureen X pisteessä i
X_{i+1}	suureen X pisteessä i+1.

LIITE 2

Mikäli ilman absoluuttista kosteutta ei tunneta, se voidaan laskea ilman suhteellisen kosteuden ja lämpötilan avulla käyttäen seuraavia kaavoja:

Ilman vesihöyryn osapaine ph (t) voidaan laskea seuraavan kaavan avulla, missä

$$p_h(t) = \frac{RH(t) \cdot p_{hs}(t)}{100} \tag{L2}$$

missä

RH(t)	ilman suhteellinen kosteus, %
$p_{hs}(t)$	vesihöyryn kyllästymispaine, kPa.

Kaavassa (L2) tarvittava vesihöyryn kyllästymispaine $P_{hs}(t)$ voidaan laskea likimääräisesti esimerkiksi seuraavan kaavan avulla

$$p_{hs}(t) = \frac{\exp\left(77.345 + 0.0057 \cdot T(t) - \frac{7235}{T(t)}\right)}{1000 \cdot T(t)^{8.2}} \tag{L3}$$

missä

T	ilman lämpötila, K.

Ilman absoluuttinen kosteus x (t) voidaan laskea seuraavan kaavan avulla

$$x(t) = 0.622 \frac{p_h(t)}{p - p_h(t)} \tag{L4}$$

missä

$p_h(t)$	vesihöyryn osapaine, kPa
p	ilman kokonaispaine, kPa.

Kaavassa (L4) ilman kokonaispaineen p arvona voidaan käyttää ilman normaali-paineen arvoa 101.3 kPa.

Appendix B: Data used for NEEAP and NREAP calculation in 2013

The data below was generated in the project as preliminary results aimed for the NEEAP and NREAP calculation for the ministry of employment and the economy.

Table B1. Renewable energy use and energy savings by building type in 2010.

Renewable energy and energy savings in 2010, GWh					
	Detached houses	**Attached houses**	**Blocks of flats**	**Free time residences**	**Total**
Ground source	921	2	0	17	941
Exhaust air	88	10	0	5	103
Air/air	1152	184	69	115	1521
Air/water	73	0	0	0	73
Sum of renewable energy	2146	186	70	133	2535
Sum of energy savings	2234	197	70	138	2638

Table B2. Renewable energy use and energy savings by building type in 2016.

Renewable energy and energy savings in 2016, GWh					
	Detached houses	**Attached houses**	**Blocks of flats**	**Free time residences**	**Total**
Ground source	2477	5	1	47	2530
Exhaust air	162	19	0	10	190
Air/air	2258	361	135	226	2980
Air/water	365	0	0	0	365
Sum of renewable energy	5099	367	136	273	5875
Sum of energy savings	5261	386	136	282	6065

Table B3. Renewable energy use and energy savings by building type in 2020.

Renewable energy and energy savings in 2020, GWh					
	Detached houses	Attached houses	Blocks of flats	Free time residences	Total
Ground source	3871	8	2	73	3954
Exhaust air	216	25	0	13	254
Air/air	2358	377	141	236	3112
Air/water	629	0	0	0	629
Sum of renewable energy	6858	386	143	309	7695
Sum of energy savings	7074	411	143	322	7949

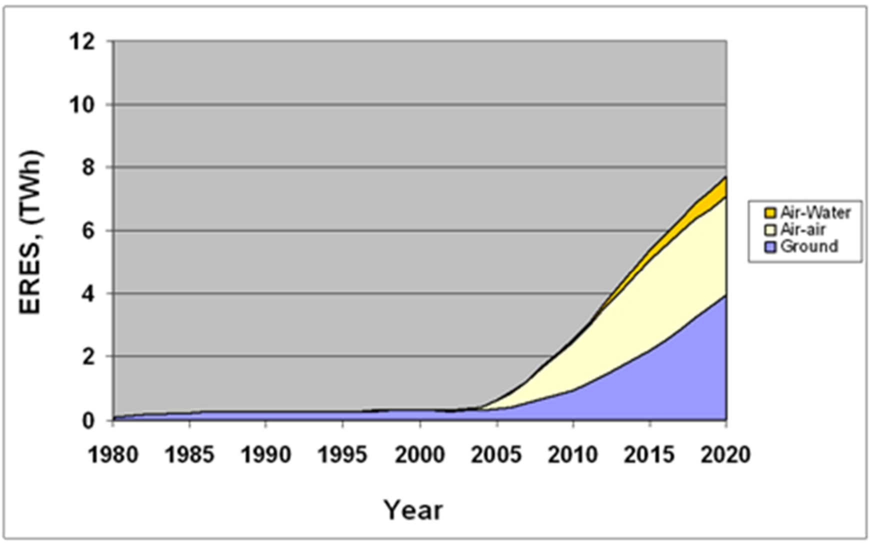


Figure B1. Prognosis of the use of the renewable energy by heat pumps by 2020.

Series title and number
VTT Technology 164

Title	**Renewable energy production of Finnish heat pumps** **Report of the SPF-project**
Author(s)	Ari Laitinen, Pekka Tuominen, Riikka Holopainen, Pekka Tuomaala, Juha Jokisalo, Lari Eskola & Kai Sirén
Abstract	The SPF project defined a national hourly seasonal performance factor calculation method for air to air heat pumps, air to water heat pumps, ground source heat pumps and exhaust air heat pumps in co-operation with international Annex 39 work. The energy use of the Finnish building stock was estimated using standard building types further adapted to different decades: a detached house, an apartment building, an office building and a summer cottage. The energy use of these standard building types was calculated with different heat pump types leading to energy saving and renewable energy use of the heat pumps in different buildings. The current and future cumulative energy consumption of the building stock was modelled using the REMA model developed at VTT. The future effects of heat pumps on the energy use and emissions of the Finnish building stock were modelled comparing with the REMA model a conservative Business as Usual scenario with a Heat Pump scenario.
ISBN, ISSN	IISBN 978-951-38-8141-2 (URL: http://www.vtt.fi/publications/index.jsp) ISSN-L 2242-1211 ISSN 2242-122X (Online)
Date	April 2014
Language	Finnish, English abstract
Pages	90 p. + app. 30 p.
Name of the project	SPF
Commissioned by	the Finnish ministry of employment and the economy, the Finnish ministry of the environment and SITRA, Finnish Innovation Fund
Keywords	Seasonal performance factor, heat pump, energy saving, renewable energy
Publisher	VTT Technical Research Centre of Finland P.O. Box 1000, FI-02044 VTT, Finland, Tel. +358 20 722 111

Julkaisun sarja ja numero
VTT Technology 164

Nimeke	**Suomalaisten lämpöpumppujen uusiutuvan energian tuotto** **SPF-hankkeen loppuraportti**
Tekijä(t)	Ari Laitinen, Pekka Tuominen, Riikka Holopainen, Pekka Tuomaala, Juha Jokisalo, Lari Eskola & Kai Sirén
Tiivistelmä	SPF-hankkeessa määriteltiin kansallinen tunnittainen kausihyötysuhteen laskentamenetelmä ilmalämpöpumpuille, ilma-vesilämpöpumpuille, maalämpöpumpuille sekä poistoilmalämpöpumpuille yhteistyössä kansainvälisen Annex 39-ohjelman kanssa. Suomen rakennuskannan energiankulutusta arvioitiin eri vuosikymmenille määriteltyjen tyyppirakennusten avulla jotka edustivat pientaloa, kerrostaloa, toimistotaloa sekä vapaa-ajan rakennusta. Näille tyyppirakennuksille arvioitiin energiansäästöpotentiaali ja uusiutuvan energian tuotto eri lämpöpumppuvaihtoehdoilla. Koko rakennuskannan nykyistä ja kumulatiivista energiankulutusta arvioitiin VTT:n kehittämällä REMA-mallilla. Lämpöpumppujen tulevaa vaikutusta suomalaisen rakennuskannan energiankulutukseen ja päästöihin arvioitiin vertaamalla perinteistä Business as Usual-skenaariota lämpöpumppujen nopeampaa yleistymistä kuvaavaan lämpöpumppuskenaarioon.
ISBN, ISSN	ISBN 978-951-38-8141-2 (URL: http://www.vtt.fi/publications/index.jsp) ISSN-L 2242-1211 ISSN 2242-122X (verkkojulkaisu)
Julkaisuaika	Huhtikuu 2014
Kieli	Suomi, englanninkielinen tiivistelmä
Sivumäärä	90 s. + liitt. 30 s.
Projektin nimi	SPF
Toimeksiantajat	TEM, YM, Sitra
Avainsanat	Seasonal performance factor, heat pump, energy saving, renewable energy
Julkaisija	VTT PL 1000, 02044 VTT, puh. 020 722 111

Renewable energy production of Finnish heat pumps

Final report of the SPF-project

ISBN 978-951-38-8141-2 (URL: http://www.vtt.fi/publications/index.jsp)
ISSN-L 2242-1211
ISSN 2242-122X (Online)

10.6 Appendix VII – National report Sweden

Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning

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SP Technical Research Institute of Sweden

Energy Technology

SP Report 2010:49

Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning

Roger Nordman, Kajsa Andersson, Monica Axell, Markus Lindahl

SP Technical Research Institute of Sweden

Abstract

In this project, results from field measurements of heat pumps have been collected and summarised. Also existing calculation methods have been compared and summarised. Analyses have been made on how the field measurements compare to existing calculation models for heat pumps Seasonal Performance Factor (SPF), and what deviations may depend on. Recommendations for new calculation models are proposed, which include combined systems (e.g. solar – HP), capacity controlled heat pumps and combined DHW and heating operation.

Key words: Heat pump, SPF, calculation model, field measurements

SP Sveriges Tekniska Forskningsinstitut
SP Technical Research Institute of Sweden

SP Report 2010:49
ISBN 978-91-86319-86-1
ISSN 0284-5172
Borås 2010

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Preface

This report summarize the findings from SP Technical Research Institute of Sweden in the joint KTH-SP project “Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning”, project P9 in the Effsys-2 research programme, financed by the Swedish Energy Administration and participating companies and organizations.
The project was set up so that SP and KTH performed separate parts of the projects, but with discussions and meetings in between.
The project parts are reported according to the parts stipulated in the application.

Sammanfattning

I denna rapport redovisas de delar av projektet ”Beräkningsmetoder för årsvärmefaktor för värmepumpsystem för jämförelse, systemval och dimensionering” som SP Sveriges tekniska forskningsinstitut svarat för. Projektet har genomförts av SP och KTH. KTH:s del av projektet redovisas i en separat rapportdel.

I en inledande del av projektet har förberedelser för ett IEA samarbete, samt gemensam övergripande projektplanering tillsammans med industriparterna utförts. IEA-projektet har godkänts att starta av styrelsen för IEA Heat Pump Programme, och ett första inledande möte har hållits.

SP har koordinerat samt sammanställt resultat av fältmätningar. Väl genomförda fältmätningar är en förutsättning för validering av olika beräkningsalgoritmer. Sammanställningen visar att det finns ett flertal utförda fältmätningar i Sverige under de senaste 20 åren, men få har gjorts med SPF som fokus, utan ofta har mätningarna gjorts med syfte att studera en viss teknikförändring, eller andra faktorer. Det har inte under de senaste 10 åren utförts någon stor mätning på värmepumpar liknande de välkända Fraunhofermätningarna eller FAVA-studien i Schweiz. Den enda studie som syftat till att mäta SPF är den som SP utfört. Detta kan ses som en brist i ett land där värmepumpar har ett så stort genomslag för uppvärmningen av bostäder.

En kravspecifikation för mätdata som behövs för att användas för validering har tagits fram.

En sammanställning av befintliga standardliknande beräkningsmetoder (existerande algoritmer) för SPF har gjorts. Syftet med analysen har varit att beskriva existerande algoritmer (modeller) samt kartlägga om nuvarande program (Annex 28, SP´s beräkningsprogram mm) innefattar alla typer av värmepumpsystem som finns på marknaden idag. En viktig del är att undersöka hur kombinerad drift dvs. tappvarmvatten och värme behandlas i modellerna. En annan fråga är huruvida olika typer av kapacitetsreglering behandlas. Sammanställningen har visat att det finns en stor brist bland förekommande program och metoder vad gäller att ta hänsyn till :

- Kombisystem, såsom sol-vp
- Kapacitetsreglerade system
- System med kombinerad varmvattentillverkning och uppvärmning

Existerande algoritmer har jämförts med resultat från fältmätningar. Från existerande fältmätningar har data tagits för att jämföra resultaten med befintliga metoder för att beräkna SPF. En analys av hur väl dessa metoder förmådde beräkna SPF för de studerade systemen har gjorts. Denna analys visar att resultaten från fältmätningarna ofta visar på högre SPF än vad som beräkningsmodellerna ger. Det finns flera orsaker till detta, bland annat att modellerna använder sig av konstant marktemperatur (som i förekommande fall är lägre än verklig marktemperatur), att modellerna använder en bivalent punkt som aldrig uppträtt i de verkliga mätningarna mm. Den gjorda jämförelsen visar på ett antal viktiga faktorer att studera vidare.

För att utveckla ett enkelt program för jämförelse av värmepumpsystem är det viktigt att begränsa beräkningarna till ett antal klimatzoner och ett antal typhus. Målet är att beräkningsmetoden skall kunna användas både nationellt och internationellt. I ett dimensioneringsprogram skall däremot stor frihet ges att definiera det specifika huset för att utförligt kunna studera de behov som finns för de specifika installationerna.

En ny beräkningsmetodik för SPF och årsenergibesparing baserad på, eller som ersättning för existerande algoritmer som input för nytt Annex inom IEA HPP och Europastandard (CEN) har diskuterats. Det gemensamma beräkningsprogrammet skall baseras på indata från gällande

Europeiska standarder (EN 14511) för kombinerad drift med värme och tappvarmvatten. Det skall även till fullo implementera rutiner för drift med kapacitetsreglerade värmepumpar (kompressorer och pumpar/fläktar).

Förslag till vad som bör ingå i ett nytt transparent gemensamt beräkningsprogram som kan användas för jämförelse och certifiering har getts. Industrigruppen menade tidigt att det viktiga i denna del är att ta fram de samband som bör implementeras i ett beräkningsprogram, men att de själva oftast skriver in-house kod som de kan implementera dessa samband i. Detta gör att förutsättningarna blir likartade, men att tillverkarna fortfarande kan ha sina specifika (ofta hemliga) indata själva.

1 Introduction

The existing calculation tools for 1) design and 2) comparison need to be further developed to show the potential with new technology such as capacity controlled systems and more efficient system for combined operation with space heating and domestic hot water production. The overall aim is to develop existing tools for future needs. The outcome from the calculation tools should be useable for calculation of environmental impact. The purpose is to compare existing tools for calculation of seasonal performance factor and annual energy savings in order to propose needs for further development. For validation of the calculation tools existing data from laboratory and field measurements will be used.

Seasonal Performance Factor, SPF, is a term used mainly for real installations, compared to the Coefficient of performance, COP, which is evaluated in controlled lab environment. How SPF is estimated depends on the situation under which it is evaluated, see Figure 1 below.

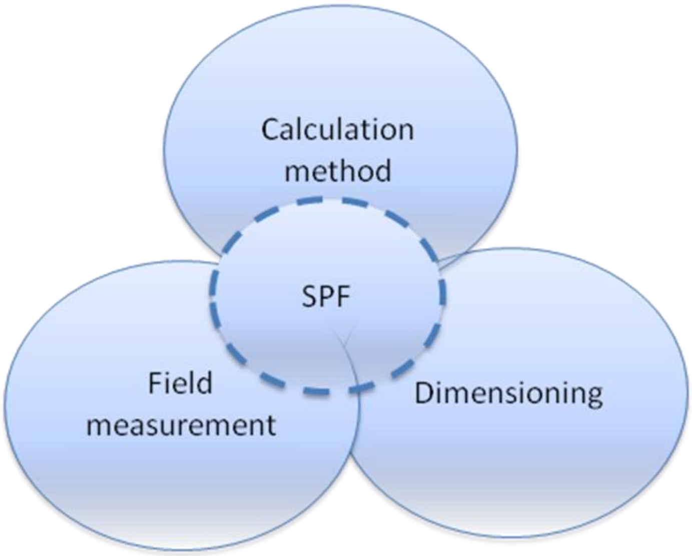


Figure 1. SPF can be determined in various ways, including field measurement, calculation methods and dimensioning software.

Based on lab measured performance data, SPF can be calculated according to calculation methods, that normally relates performance data in specific operating modes to annual climatic conditions, expressed as “bin models” where the number of hours in a year the temperature is between certain values are binned together. Model buildings are normally used to give annual heat demands and overall heat transfer resistances of the building.
For the installer of heat pumps, more specific details of the building must be prompted, as well as detailed data about the ground properties in the case of GSHP’s. Local climatic data is also used for estimating the heat demand. The climatic data contains a cold shock in order to dimension the heat pump capacity to extreme conditions that may occur during the lifetime of the installation. Other data such as the number of occupants, Domestic Hot Water (DHW) energy consumption is also normally entered in the software models for dimensioning.
To evaluate the real performance of the installed heat pump, field measurements are carried out to relate the useful heat produced to the energy input, often electrical power (but it could also be heat driven processes). The SPF of the heat pump is then often expressed as the ratio of the heat delivered to the heat distribution system (including DHW when relevant) to the electricity to operate the heat pump (including electricity to operate pumps and fans to bring the heat source to the heat pump).
The different level of detail given as input in the different stages of SPF calculation will lead to different SPF values. The main objective of this project is to identify what needs to be included in a new calculation method in order to better represent the real SPF of the heat pump in the building system.

2 Preparing an IEA HPP Annex on SPF

Preparations for an IEA annex on SPF have included preparatory meetings, and communication with research communities involved in the IEA HPP sphere. Meetings include a meeting during the ASHRAE winter Conference 2009 [1.1.1.1.11], NT meeting in Borås, September 2009 , and a Meeting in Paris march 5th, 2010 [2].
A draft legal text was prepared and circulated among interested parties and the executive committee in HPP. The draft legal text was discussed in the ExCo meetings in Rome, November 2009 and in Helsinki June 2010. In the Helsinki meeting it was suggested that the annex proposal for “Dynamic testing of heat pumps” should be integrated with the SPF annex. The kick-off meeting for the SPF Annex in June 30th- July 1st 2010 will discuss the possibility for this integration. The legal was just recently approved by the ExCo [3].
The preparation and starting up of the international Annex has taken much more time than expected, mainly due to constraints in timing and funding. However, on June 30 –July 1st, the kick-off meeting for the new annex is held in Albuquerque, New Mexico.

3 Summary of already performed field measurements.

In order to evaluate already made field measurements in Sweden, or made by Swedish manufacturers, meetings in the project discussed earlier made field measurements. The result is that there has been a large number of field measurements made during the last decades, see Appendix 1 and references [4-6], but few studies have had the specific goal to examine the SPF.
In order to make detailed analyses of the performance, also detailed data from the measurements are needed, and this was only available in two studies, the SP study ”Erfarenheter från fältutvärdering av fem bergvärmepumpar i Sjuhärad” and the Fraunhofer study “Heat Pump Efficiency” where a number of Swedish heat pump manufacturers participated with heat pump units. For Air-air heat pumps, only one study has been found [7]. These three studies are describes more in detail below.

3.1 Description of evaluated field measurements

3.1.1 Fraunhofer

The Fraunhofer-Institute for Solar Energy Systems ISE is running two large field monitoring project including approximately 200 heat pumps in total. The heat pump efficiency project includes approximately 110 installed heat pumps with a heating capacity of 5-10 kW. In the Replacement of Central Oil boilers with Heat Pumps in Existing Building Project 75 heat pumps are included. The heat pump types included are air to water, ground source and water to water heat pumps. In this study two heat pump producers, IVT and Nibe, have provided the project with data based on the field measurements in the Fraunhofer study.

3.1.1.1 Measured parameters

Table 1 gives an overview of the parameters normally measured in the Fraunhofer field measurements. Exactly what parameters tested might differ from test site to test site. For some test sites additional equipment are measured as well. Examples of such equipment are circulation pumps or control equipment.

Table 1. Measured parameters for brine to water heat pumps in the Fraunhofer study.

	Running time	Energy content	Energy consumption	Inlet temp.	Outlet temp.	Volume flow	Delivered heat during operation	Average power during operation
	(min)	(kWh)	(kWh)	(˚C)	(˚C)	(l/h)	(kW)	(W)
	Sum	Sum	Sum	Average	Average	Average		Average
Heat Pump, total	X		X				X	X
Compressor	X		X					X
Warm heat transfer medium circuit		X		X	X	X	X	
Cold heat transfer medium circuit (brine)		X		X	X	X	X	
Space heating circuit		X		X	X	X	X	
Domestic hot water circuit		X		X	X	X	X	
Supplementary heater			X					X
Measurement equipment			X					X
Pump, space heating circuit			X					X
Pump, warm heat transfer medium circuit			X					X
Pump, cold heat transfer medium circuit (brine)			X					X

For air to water heat pumps included in the study many of the measured parameters are the same. The data related to the cold heat transfer medium are replaced with data regarding fans in the outdoor unit. Additional the outdoor temperature and the humidity are measured for air to water heat pumps whereas it is not for the brine to water heat pumps.

Table 2. Measured parameters for air to water heat pumps in the Fraunhofer study.

	Running time	Energy content	Energy consumption	Inlet temp.	Outlet temp.	Volume flow	Delivered heat during operation	Average power during operation
	(min)	(kWh)	(kWh)	(˚C)	(˚C)	(l/h)	(kW)	(W)
	Sum	Sum	Sum	Average	Average	Average		Average
Heat Pump, total	X		X				X	X
Compressor	X		X					X
Warm heat transfer medium circuit		X		X	X	X	X	
Space heating circuit		X		X	X	X	X	
Domestic hot water circuit		X		X	X	X	X	
Supplementary heater			X					X
Measurement equipment			X					X
Pump, space heating circuit			X					X
Pump, warm heat transfer medium circuit			X					X
Fan			X					X

3.1.1.2 System boundaries

The system overview below shows the placement of the measurement equipment. The figure shows a general system, the real systems are many times more complicated and will not fit into the general description. In these cases additional meters are installed in order to be able to monitoring the system in a good way.

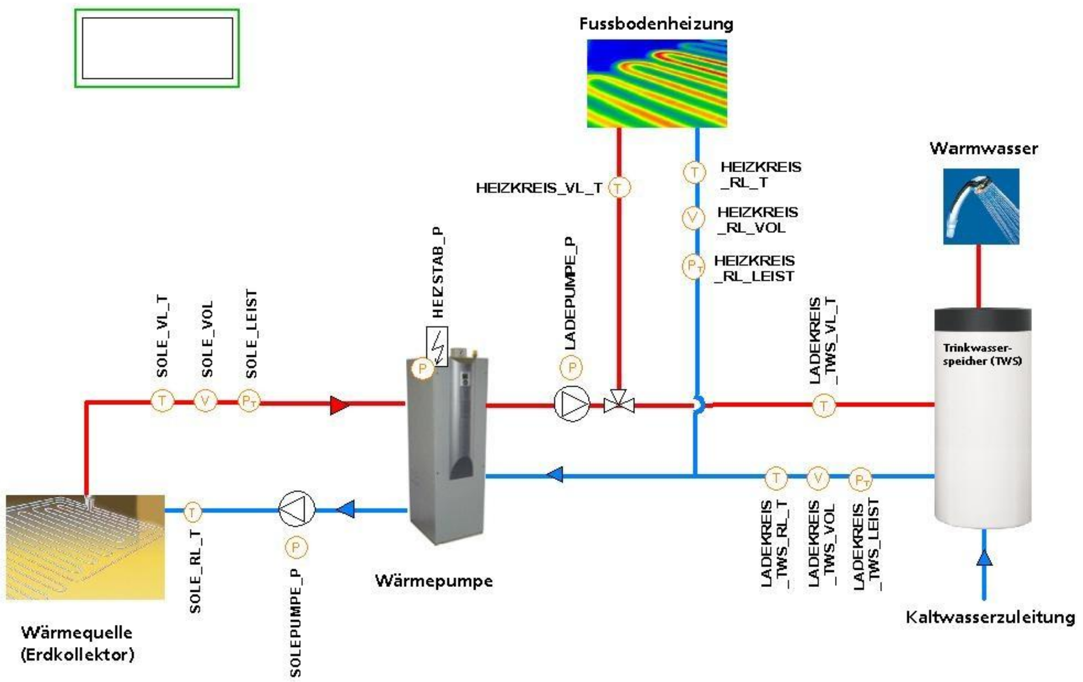


Figure 2. System overview, placement of measurement equipment.

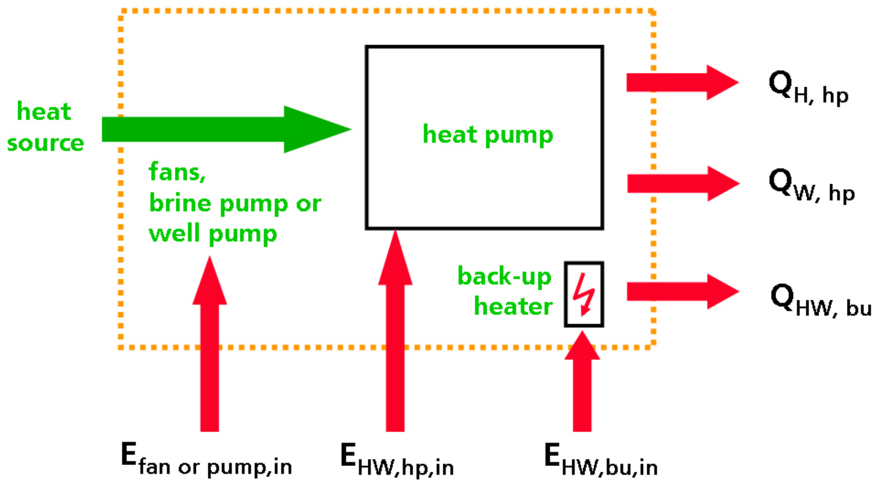


Figure 3. Schematic overview of used and delivered energy**.**

3.1.1.3 Sampling interval

The data are collected automatically and stored every minute. The stored data are remotely accessible by a GSM modem and transferred to the Fraunhofer Institute, followed by an automatic saving and sorting of the data. An automatic test of plausibility is also done, using specially made software.

The data used in this SPF project are presented as daily averages.

3.1.1.4 Measurement equipment

The meters are generally located at both the source and the heat side. For systems equipped with buffer tanks, the meters are installed before the tanks if possible. The meters are installed as close to the heat pump as possible but after the split of warm hot water transfer hot water into space heating and domestic water circuits. This in order to be able to measure energy amounts used for both space heating and domestic water separately.

Ultrasonic heat measuring device combined with data loggers are used to measure the produced heat. Temperatures, volume flows, amount of accumulated heat, electricity consumption of pumps and other equipment are measured by means data loggers.

3.1.1.5 Measurement uncertainty

No information about measurement uncertainty was provided in the Fraunhofer studies.

3.2 Measurement of ground source heat pumps

In 2003-2004 SP made a field measurements including five ground source heat pumps located in the Boras area. The study named "Årsmätningar av fem bergvärmeanläggningar i Sjuhärad" [6]. The measurements were performed from November 2003 to November 2004.

3.2.1 Measured parameters

The following parameters are measured:

- Thermal heat content, space heating
- Thermal heat content, tapped sanitary hot water
- Electricity consumption, total heat pump
- Electricity consumption, supplementary heater

- Indoor temperature
- Outdoor temperature
- Brine temperature, inlet (3 of 5 units)
- Brine temperature, outlet (3 of 5 units)
- Compressor, running time

Heat meters were installed between the space heating system and the heat pump, the same was done for the tapped sanitary hot water. Thereby internal heat losses were not measured. The meters was installed as close to the heat pump as possible in order to minimize the influence of these losses.

The electricity consumption of the supplementary heater was measured indirectly by measuring the running time and the instantaneous power for each efficiency step.

The indoor and outdoor temperatures were logged continuously. The indoor meter was placed centrally in the building with no influence of sunshine or other sources of interference. The outdoor meter was placed on the north or northeast façade.

3.2.1.1 Sampling interval

Table 3. Measured parameters and sampling interval

Thermal heat content, space heating	Once per week
Thermal heat content, tapped sanitary hot water	Once per week
Electricity consumption, total heat pump	Once per week
Electricity consumption, supplementary heater	Once per week
Indoor temperature	Every 20 minutes
Outdoor temperature	Every 20 minutes
Brine temperature, inlet (3 of 5 units)	Every 10 minutes
Brine temperature, outlet (3 of 5 units)	Every 10 minutes
Compressor, running time	Once per week

3.2.1.2 Measurement equipment

The measurement equipment used is listed in **Fel! Hittar inte referenskälla.**. The equipment used for measuring the brine temperature is not specified.

Table 4. Measurement equipment

Electrical energy	ABB Deltameter CBB 211700
Running time	Paladin
Electrical power	Siemens B4301
Heat meter	Siemens Ultraheat 2WR5151
Indoor temperature	Easy Log 24 RFT
Outdoor temperature	Easy Log 40 KH
Brine temperature	Not specified

3.2.1.3 Measurement uncertainty

No information about measurement uncertainty in the report.

3.3 Field measurement of air-to-air heat pumps

From March 2008 to February 2009 SP Technical Research Institute of Sweden made a field measurement of five air-to-air heat pumps in the Borås area. The results from the measurements are presented in SP report 2009:26 “Fältmätning av Luft/Luft värmepumpar I svenska småhus”. [7]

Electricity consumption and temperatures was logged continually and five performance tests were made during the year. The performance tests were planned to be made at different outdoor temperatures. Two test during spring and autumn and one during the winter. But due to the mild winter and divergence between the weather forecast and the actual weather conditions at the test site the planed dissemination was not reached. The performance test follows SP method no. 1721 [11].

3.3.1.1 Measured parameters

The following parameters are measured and logged continually:

- Electricity consumption, total to the building
- Electricity consumption, heat pump
- Electricity consumption, supplementary heat
- Indoor temperatures in tree rooms
- Outdoor temperatures
- Outdoor humidity

The following parameters are measured during the performance test due to SP method no. 1721:

- Airflow from indoor unit
- Air temperature before the indoor unit
- Air temperature after the indoor unit
- Electrical power, heat pump
- Air pressure

Figure 4. Measurement equipment due to SP Method no. 1721

3.3.1.2 Sampling interval

Table 5. Measured parameters and sampling interval

Electricity consumption, total to the building	
Electricity consumption, heat pump	Every 5 minutes
Electricity consumption, supplementary heat	Every 5 minutes
Indoor temperature	Every 20 minutes
Outdoor temperature	Every 20 minutes
Thermal heat content, space heating	5 measurements
Electricity consumption, heat pump	5 measurements

3.3.1.3 Measurement equipment

Table 6. Measurement equipment

Electricity consumption	ABB Deltameter CBB 211700
Logger pulse	Easy Log 40 IMP
Logger air temperature and humidity	Easy Log 24 RTF
Logger outdoor temperature	Easy Log 40 KH
Flow meter, air	VEAB
Air pressure meter	Testo 511
Temperature meters	PT100
Pressure meter	
Data logger	
Meter electrical power	

3.3.1.4 Measurement uncertainty

If the demands stated in SP method no. 1721 is fulfilled the Coefficient of performance (COP) can be calculated with an uncertainty lower than ±10%. The yearly delivered heat from the heat pump can be calculated with an uncertainty of ±20%.

The results presented follow the standard SP 1721. The capacity of the heat pump is measured during stable conditions and is not including any defrost cycle. Thereby the results for the SPF are based on data from the heat pump running at stable conditions, which will lead to an overestimation of the SPF. For COP calculations uncertainty will be smaller, since the output of heat is more or less proportional to the electricity consumption.

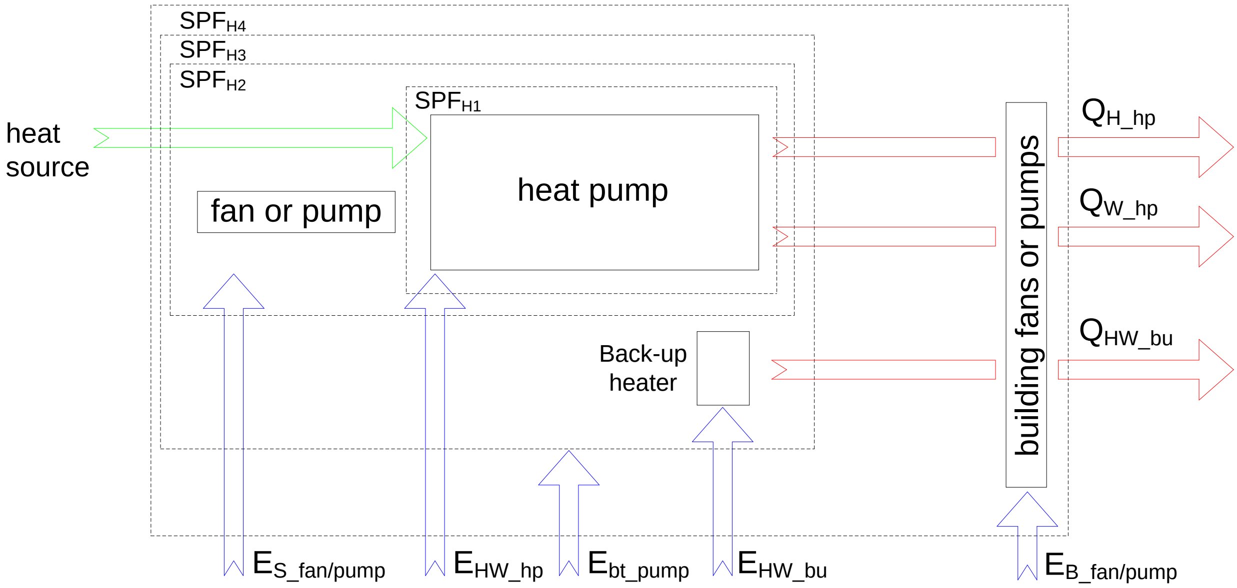
During the field measurements the conditions under a whole cycle was measured for internal use. But due to problems to have equivalent measurement conditions at all test sites it was decided to not include this information in the report.

4 Minimum required measured parameters in field measurements

The SPF-value can be calculated for different levels of the heating system. The level is described by defined system boundaries. This project relates to four different system boundaries developed in the SEPEMO EU project. The system boundaries are more detailed described in section 8.4.

The system boundaries are named SPF_1-SPF_4, each number describing its own system boundary. Different system boundaries mean different requirements of data to be measured. Before performing field measurements it must be clear what SPF level that is to be measured.

The figure below shows the different system boundaries developed in SEPEMO. SPF_1 includes SPF for the heat pump itself *only*. SPF_2 also includes heat source pumps and fans, the equipment to make the heat source available for the heat pump. SPF_3 also includes auxiliary heating, back up heating. SPF_4 includes heat sink equipment like fans or liquid pumps, to make the heat available in the house.



The required measurements differ between different types of heat pumps. The required measurements related to each type of heat pump is shown in Table 7.

Table 7. Minimum results for different heat pump types.

		A/W	DX/W	B/W	W/W	A/A
Electric energy input - total	kWh	x	x	x	x	x
Electric energy input backup heater	kWh	x	x	x	x	x
Electric energy input pumps/fans heat source side	kWh	x	x	x	x	
Electric energy input pumps/fans heat sink side	kWh	x	x	x	x	
Energy output heating / cooling	kWh	x	x	x	x	x
Energy output DHW	kWh	x	x	x	x	optional
SPF according the system boundaries	-	x	x	x	x	x
Average supply temperature heat sink*	°C	x	x	x	x	x
Average return temperature heat sink*	°C	x	x	x	x	x
Average supply temperature DHW*	°C	x	x	x	x	Optional
Average return temperature DHW*	°C	x	x	x	x	Optional
Average supply temperature heat source*, 1	°C			x	x	
Average return temperature heat source*, 1	°C			x	x	
Average outdoor temperature*	°C	x	x	x	x	x
Average indoor temperature*	°C	x	x	x	x	x
Outdoor humidity	%	x				x

*During heating season (operating season). 1Ground temperature should me measured in direct expansion systems

The performance of air to air heat pumps is measured according to SP method 1721. This method is more detailed explained in section 5.2. The boundary condition that is used in this method differs from the boundaries stated in the figure above. This method includes separate measurements of the auxiliary heater and the total electrical input to the heat pump, the fans in the indoor and outdoor unit included. For an air to air heat pump the auxiliary heating is not a part of the heat pump system, but a part of the building that is to be heated. The energy used for auxiliary heating should be measured in order to be able to calculate the energy cover ratio from the heat pump. The DHW production is also outside the heat pump system regarding air to air heat pumps. These parameters are optional to measure, but are interesting for information purposes.

4.1 Minimum results for the different SPF levels

The minimum result from the measurements according to each SPF level is stated in Table 8 below. Some parameters are necessary to measure in order to get data for the SPF equations, while some parameters are necessary to measure in order to understand the operating conditions for the heat pump and to be able to read and compare the results from different systems. The energy output can be measured either by using an energy meter or by measuring the supply and return temperatures together with the liquid flow.

Table 8. Required measurements for meeting the SPF levels according to SEPEMO.

		SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}
electric energy input heat sink auxiliary	kWh	-	-	-	x
electric energy input backup heater	kWh	-	-	x	x
electric energy input heat source auxiliary	kWh	-	x	x	x
electric energy input - total	kWh	x	x	x	x
energy output heat	kWh	x	x	x	x
energy output DHW	kWh	x	x	x	x
supply temperature (heat sink)	°C	x	x	x	x
return temperature (heat sink)	°C	x	x	x	x
supply temperature (heat source)	°C	x	x	x	x
return temperature (heat source)	°C	x	x	x	x
outdoor temperature	°C	x	x	x	x
outdoor humidity	%	x	x	x	x
indoor temperature	°C	x	x	x	x

4.2 Additional measurements

There are also parameters that can be measured that are not necessary for the calculation of SPF, but can be usable for other purposes, for example in an energy balance over the heat pump system or for information purposes. The storage losses of the storage tank can also be calculated by using extra measurements. Examples of extra measuring points are displayed in Table 9 below.

Table 9. Optional measurements

		SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}
energy output heat source	kWh	x	x	x	x
energy output into DHW storage	kWh	x	x	x	x
pressure difference, heat source	Pa	x	x	x	x
pressure difference, heat sink	Pa	x	x	x	x

4.3 Data acquisition system

The data must be recorded with a system that interfaces the sensors to a data acquisition system that can handle the necessary number of inputs from the entire sensor set.

5 Studied methods for field measurement

The relevant methods for field measurements that are studied in this project are three Nordtest methods (NT VVS) and one SP method:

- Large heat pumps - Field testing and presentation of performance (NT-VVS076)
- Refrigeration and heat pump equipment - General conditions regarding field testing and presentation of performance (NT-VVS115)
- Refrigeration and heat pump equipment - Check-ups and performance data inferred from measurements in the refrigerant system (NT-VVS116)
- Prestandaprovning av luft/luft värmepumpar i fält (SP metod nr 1721)

5.1 NT VVS methods

The NT VVS methods intend to cover the need of capacity- and functional controls and measurements for heat pumps in field applications in four different levels.
The methods states recommendations of how the measurements of temperature, flowrates, pressures and pressures differences shall be performed. In appendix estimations of measured uncertainties are given for all measured quantities with examples. The stated uncertainties for measurement given are:

- Level 1 < 5% capacity measurement
- Level 2 < 10% capacity measurement
- Level 3 < 15% capacity control

Table 10. Example of maximum permissible deviation from the mean value. Taken from the NT VVS 115-method.

Temperature, flowrate	maximum permissible deviation from the mean value (±)	
	Level 1	Level 2 and 3
Temperature of heat transfer medium, cold side	0.5 K	1 K
Flowrate of heat transfer medium, cold side	5%	10%
Temperature of heat transfer medium, hot side	1 K	2 K
Flowrate of heat transfer medium, hot side	5%	10%

The system boundaries are specified in each method. The measurements can either be carried out for the single heat pump or for the larger system, the plant.

Method NT-VVS 076 recommends that operating conditions are those for which the heat pump performance data has been guaranteed. NT-VVS 115 and NT-VVS 116 do not have recommendations. The thermal power output is decided by measuring the flow rate and temperature rise of the hot side heat transfer medium. Thermal power input is determined by measuring the flow rate and the temperature drop of the cold side heat transfer medium. Heat meters can be used. In method 116 also refrigeration condensing and evaporating pressures and temperatures are measured.

If possible the plant/ heat pump must have operated under stable conditions, within the limits of stated maximum deviations, for at least 30 minutes before the measurements starts. The measurement period is at least 30 minutes and readings are taken at a maximum interval of 3 minutes.

If the heat pump operates during defrost conditions the measurements shall be carried out with defrosted heat exchanger surfaces, during the most stable 30 minute period possible. The performance test in NT-VVS 115 and NT-VVS 116 is carried out when the heat pump has attained regular frosting-defrosting sequence starting at least 10 minutes after a terminated defrost cycle. In method NT-VVS 076, the defrosting function is checked concerning its influence on heat pump performance during one complete frosting- and defrosting cycle.

Measuring instruments must have a certificate of calibration traceable to a national or international primary standard that is not older than 1 year at the moment of testing.

Equations for calculating COP and SPF are given. The SPF equations include also any supplementary heating and states that standby losses must be concerned.

5.2 SP method nr 1721

SP method nr 1721 is a field measurement method for field testing of electrically driven air to air heat pumps in heating or cooling mode. The method includes heating capacity, electric power input and coefficient of performance. Instructions of how the measurements shall be performed are stated. If the test is conducted in accordance with the measuring requirements of the method, the coefficient of performance can be determined with an uncertainty of measurement lower than 10%. The method is validated in a combination of laboratory and field measurements.

The system boundaries are specified in the method. The measurements can either be carried out for the single heat pump, the heat pump system or for the entire heat system, see Figure 5.

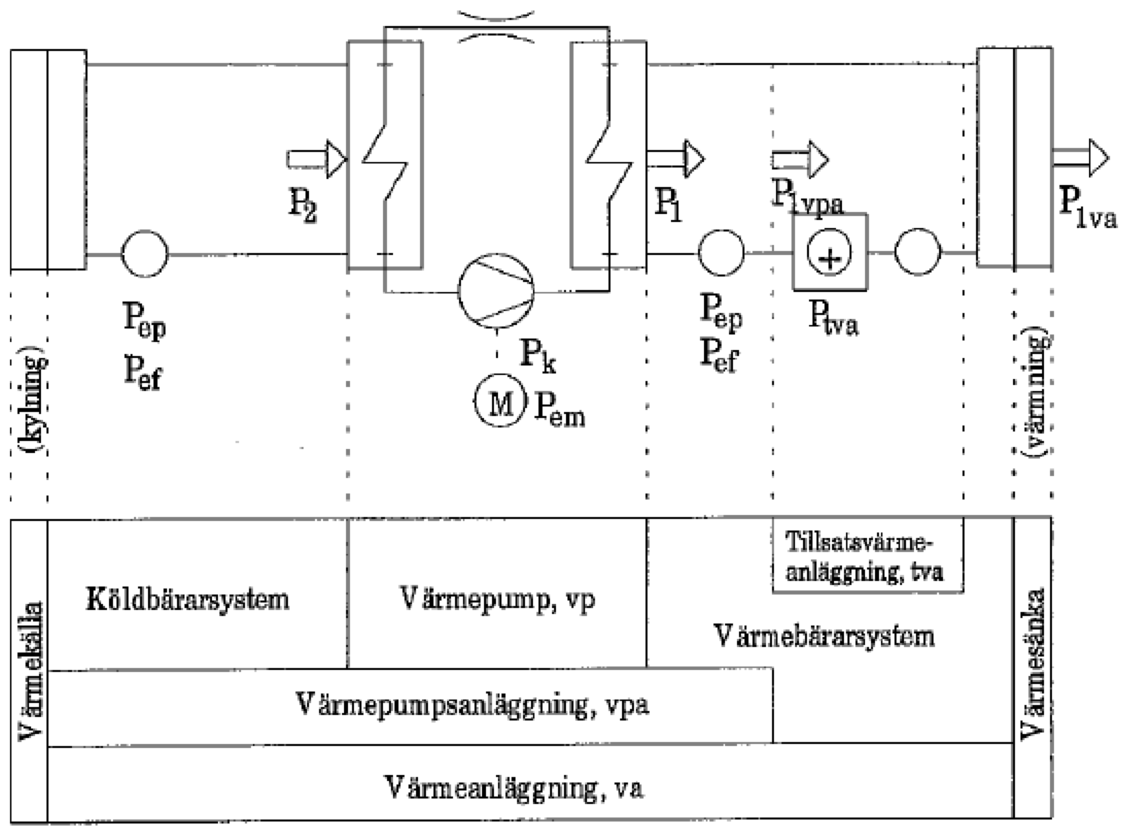


Figure 5. The figure shows the boundaries of the system for measuring the heat factors.

No examples or recommendations of operating points for the tests are stated in this method.

The total electricity consumption during the test is measured by attaching an electrical power meter or an integrated electrical energy meter to the supply cable of the heat pump.

The emitted heat effect is decided by measurements in the circulation flow. A volume- or a mass flow meter (installed according to the manufacturer's instructions) is used to measure the air flows in the heat transfer medium circuit. To minimize effects at the air flow, the meter is not allowed to affect the static pressure at the outflow of the heat pump more than ±3Pa. Therefore it is often necessary to include an extra fan.

The temperatures that shall be measured are: incoming cooling medium temperature, incoming heating medium and leaving heat transfer medium temperature. The temperature of the incoming cooling medium is measured by one sensor placed in the centre of the air intake. The temperature of the incoming heating medium is measured by at least four temperature sensors evenly spread over the air intake. The variation between the highest and lowest temperature indication shall be lower than 1 K. The temperature of the leaving heat transfer medium circuit is measured by at least four sensors evenly spread out at a point where the air is mixed. The mixing device is not allowed to affect the static pressure of the outflow of the heat pump more than ±3Pa, whereupon it is often necessary to include an extra fan. Heat exchange between the mixing device and the surroundings shall be taken into account. The variation between the highest and lowest temperature indication shall be lower than 1 K.

The data collection starts when "the plant" has operated at least five minutes at steady state conditions, within the required permissible deviations, see Table 11. The stability is controlled by continuous measuring at intervals shorter than 1/5 of the stability period, maximum one minute interval.

Table 11. Required permissible deviations for data collection in SP Method 1721.

Temperature, flow	Maximum permissible deviations from mean value
t_{vbin}	± 1K
t_{vbut}	± 1 K
q_{vvb}, q_{mvb}	± 5%

The sampling period shall be at least 10 minutes and the collection of data shall be either continuous register or measuring by intervals more frequent than 1/5 of the measuring period (<2min). The operation shall be stable also during the measurement period.

When the heat pump operates during conditions where frosting occurs, the capacity test is performed after a defrost period at the most stable 10-minutes period possible (at least five minutes after the defrost period).

6 Studied methods for calculation of SPF

The matrix below (Table 12) is a summary of the most important standards studied in the project. It is divided into different categories trying to sort out the content of the different standards. All AHRI standards mentioned above refers to ASHRAE standard 37 for the description the test method and requirements for testing. The purpose of the AHRI standards is to provide test and rating requirements, requirements for operating and the like for different kinds of heat pumps. The standards EN 255-3, prEN 255-3, TS14825 and prEN14825 all refers to the standard EN 14511 for requirements to fulfil the test method. For data input to the calculations of the calculation method EuP Lot 10 and to some extent EuP Lot 1 and EN15316-4-2, one is referred to the test results from standard EN 14511.

The first category "type of standard" shows whether the standard describes a test method for laboratory tests, for field tests and if it includes a calculation model for the calculation of seasonal performance factor.

The second category "type of heat pump" describes what kind of heat pumps that is included in the standard or test method.
The third category "Operation" describes the type of operation that is treated by the standard. The different types of operation can be heating mode, cooling mode or production of domestic hot water. The column called "combined operating" refers to the simultaneous production of heating and/or cooling and the production of domestic hot water. The last column within this category "part load conditions" shows if the standard includes the operation of the heat pump in part load.

The intention of the fourth category "requirements" is to show whether the standard has any requirements of testing to reach accurate test results. Typical requirements could be that steady state has to be reached before the measurements are performed, requirements of maximum deviations from the stated measurements and a largest permissible uncertainty of measurements of the tests. The last column within this category shows whether the standard gives any recommendations of how the measurements shall be performed, such as the placement of sensors.

Table 12. Matrix of existing methods for testing and measurement and evaluation of SPF for heat pumps.

	Type of standard			Type of heat pump			Operation					Requirements				Aspects in capacity calculations					Calculations of	
	Laboratory tests	Field tests	Calculation model for SPF	ASHP	GSHP/WSHP*	AIR/AIR	Heating	Cooling	Domestic hot water	Combined operating	Part load conditions	Steady state	Permissible deviations	Uncertainty of measurements	Test set up/ performance of measurement	Pumps and fans included	Defrost period	Standby losses	On/off cycles capacity regulation	Other	COP/EER	SPF/SEER
NT VVS 076				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 115				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 116				x	x	x	x	x				Δ	Δ	x	x						x	
SP 1721						x	x	x				x	x	x	x						x	
ASHRAE standard 37	x			x	x	x	x	x				x	x	x	x		x			x		
AHRI 210/240	x					x	x	x			x	x	x	Δ	x				x		x	x
AHRI 870-2005	x				x		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 390-2003	x					x	x	x			x	Δ	Δ	Δ	Δ		x				x	
AHRI 320-1998	x				x*		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 325-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
AHRI 330-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
EN14511	x			x	x	x	x	x				x	x	x	x	x	x					
prEN14511	x			x	x	x	x	x				x	x	x	x	x	x					
EN 255-3	x			x	x				x			x	x	x	x	x	x	x			x	
prEN 255-3	x			x	x				x			Δ	Δ	Δ	Δ	x	x	x			x	
TS14825	x			x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	
prEN14825	x		x	x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	x
EN15316-4-2			x	x	x	x	x		x	x	x	α	α	α	α	α	α	α	x	α		x
EuP Lot 1	x		x	x	x		x				x	x	x	x	x	x	?		x		x	x
EuP Lot 10	x		x			x	x	x		x	x	Δ	Δ	Δ	Δ	x	x		x		x	x

The sign “Δ” means that the standard refers to another standard where the requirements are fulfilled.

The sign “α“ means that the method is a calculation method that does not include requirements from a specified test method.

The fifth category “Aspects in capacity calculations” describes aspects that are taken into account in the capacity calculations. It describes whether liquid pumps and fans are included in the effective power absorbed by the unit. The “Defrost period” column describes whether the defrost periods are taken into account when measuring and calculating the capacity of the heat pump. The “standby losses” column means that standby losses are measured and taken into account when calculating the capacity of the heat pump. The NT-VVS 076 and NT-VVS 115 both mention that it is necessary to take standby losses into account when calculating the SPF, but there is no method of how to measure the losses. Both the standards for measuring the production of domestic hot water EN 255-3 and prEN 255-3 states methods of how to measure the standby losses, but the way of taking the standby losses into account when calculating the COP differs a lot between the standards. “On/off cycles and capacity regulation” shows whether the standard treats what kind of capacity regulation that is used by the heat pump. The last column “other” shows whether there are other important aspects apart from the earlier mentioned ones, which are taken into account in the capacity calculations. It shows that for some of the methods mentioned in the standard ASHRAE 37 adjustments of the line loss capacity and duct losses are made.

The last category “calculations of” describes the calculated outcome of the standard. The NT VVS standards provide simple equations of how to calculate SPF without a calculation model.

6.1 Other methods including calculation models

Besides the models mentioned above there are several other standards and models that can be used in order to find an appropriate model to calculate a seasonal performance factor. The ones studied in this project are shortly summarized in this chapter.

EN 15316-2-3 Heating systems in buildings – Method for calculation of system energy requirement and system efficiencies – Part 2-3: Space heating distribution systems
This method calculates the system thermal losses and the auxiliary energy demand of water based distribution system for heating circuits (primary and secondary), as well as the recoverable system thermal losses and the recoverable auxiliary energy. The calculations are related to a design effect and design heat load of the accounted zone (EN 12831). Correction factors are provided for a number of different conditions, these conditions can for example be corrections for the size of the building, for systems without outdoor temperature compensation, efficiency and part load. The method can be applied for any time step (hour, day, month or year).

EN 13790:2008, Energy performance of buildings – Calculation of energy use for space heating and cooling (ISO 13790:2008)
This standard provides a calculation method for the assessment of the annual energy use of buildings. Factors that are taken into account are for example the heat transfer by transmission and ventilation of the building when heated or cooled to constant internal temperature, contribution of internal and solar heat gains to the building energy balance and the annual energy use for heating and cooling.
There are two different main methods that are used by the standard, one where the heat balance is calculated during a sufficiently long time (one month or a season) and dynamic effects of the building are taken into account by an empirically determined gain and/or

loss utilization factor and one method where the heat balance is calculated over small time steps (typically one hour) and the heat stored in, and released from, the mass of the building is taken into account.

EN 12831 Heating systems in buildings – Method for calculation of the design heat load
This standard is used to calculate the design heat losses of a heated space; the result is then used to determine the design heat load at standard design conditions. The temperature distribution (air and design temperature) is assumed to be uniform. The climatic data that is used for the calculations are the external design temperature and the annual mean external temperature.

Factors taken into account are for example size of the building, type of building, activities inside the building, type of room, interior, building envelope and ventilation.

A number of standards/methods for the calculation of seasonal performance factor are investigated. Some of the methods only contain a calculation model while some of them also contain instructions of how to test the heat pumps. The calculation models that are studied in this project are prEN14825:2009 draft Nov 09, EN 15316-4-2:2008, EUP LOT 1 and EUP Lot 10.

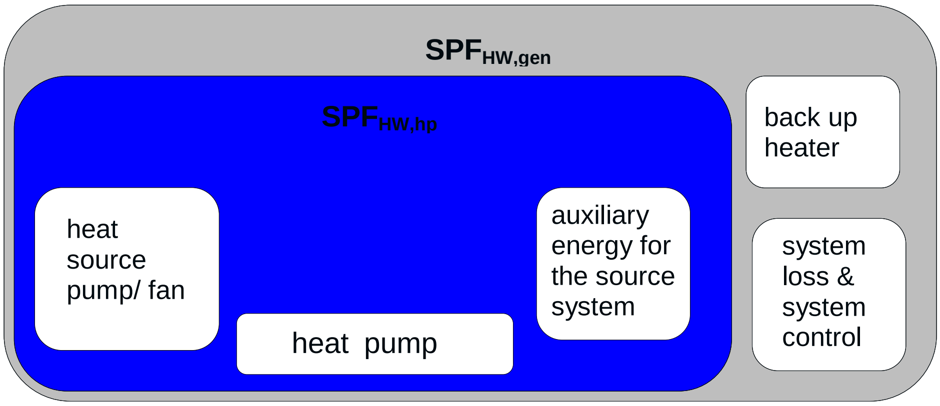
6.2 EN 15316-4-2:2008

Heating systems in buildings – method for calculation of system energy requirements and system efficiencies – Part 4-2: space heating generation systems, heat pump systems

15316-4-2 is a calculation model for the calculation of system energy requirements and system efficiencies. Input product data for the calculations, like heating capacity and COP are determined according to European or national test standards. The method treats calculations for space heating, production of sanitary hot water and combined operation of space heating and sanitary hot water production in either simultaneous or alternating operation. Presently there is no European standard for testing DHW production and space heating simultaneously; therefore a national standard shall be used instead. As an example in this standard calculations based on testing of a DHW cycle performed according to EN 255-3 during heating operation are done, see Annex D in EN 15316-4-2:2008.

<u>System boundaries</u>
The method takes into account different physical factors that can have impact on the SPF and required energy input. For example type of generator, type of heat pump, variation of heat source and sink temperature, effects of compressor working in part load (on-off, stepwise, variable speed units), and system thermal losses.

Losses due to ON/OFF cycling are considered small and negligible unless part load testing data or national values are available. If part load data is not available the stand-by auxiliary energy is considered enough for the degradation of COP in part load operation.



Input to the calculations
Two performance calculation methods for the generation subsystem are described corresponding to different applications (simplified or detailed estimation). The differences between the two methods are the required input data; the operating conditions taken into account and the calculation periods.

The simplified method
The considered calculation period is the heating season and the performance data is taken from tabulated values for fixed performance classes of the heat pump. Operating conditions are taken from typology of implementation characteristics, which means that they are not case specific. This method is in particular suitable when limited information of the generation subsystem exists.

The detailed method
This method is a temperature bin method where the specific operating conditions of each individual heat pump can be considered. The bins describe frequency of the outdoor temperature and the calculations are carried out with operating conditions for the heat pump that corresponds to the heat energy requirement of the space at each bin. The operating conditions of the bins are characterized by an operating point in the centre of each bin and in the calculations it is assumed that this point represents the operating conditions of the whole bin. The standard contains one example of climate; it represents the climate of Gelterkinden in Switzerland and span from -11°C-35°C with a resolution of one bin per K. Appendix A in EN 15316-4-2:2008 shows how to calculate bins using meteorological data for the actual spot. There are examples in the standard that uses only four bins, but with lower resolution, see figure 4 in EN 15316-4-2:2008. There are some criteria when choosing the bin resolution. The bins has to be evenly spread out over the operating range, operating points should be chosen at, or close to test points and the number of bins shall reflect the changes in heat source and sink temperatures. COP values and heat capacity can be interpolated from tested values to fit the bins.

The heat energy requirement of the distribution subsystem can be evaluated if the heat load for space heating and domestic hot water is known. The heat load for space heating is calculated based on cumulated heating degree hours which are defined by the difference between the outdoor air temperature and the indoor design temperature at the different bins. Analogously the DHW load depicted as constant daily profile can be cumulated.

Back up heaters can be accounted for, both for space heating and for sanitary hot water production. If no information about electrical back up heaters is given, an efficiency of 95% is used.

Input data for calculation with the bin method according to chapter 5.3.2 requires indoor design temperature, heat energy requirement of the space heating distribution subsystem according to EN 15316-2-3, type and controller setting of the heat emission system heat pump characteristics for heating capacity and COP according to test standards, results for part load operation according to prEN 14825, system configuration like back up heater calculated according to 15316-4-1 and installed heating buffer storage, power of auxiliary components (pumps etc.). It also requires input data for the DHW-production for example heat energy requirement of the distribution subsystem according to EN 15316-3-2 etc.

Output from the model

Two different seasonal performance factors can be calculated by using this model.

$SPF_{HW,\,gen}$ is the total seasonal performance factor of the generation subsystem. It includes the heat pump in space heating mode and production of sanitary hot water, the backup heater, the space heating distribution system and auxiliary energy.

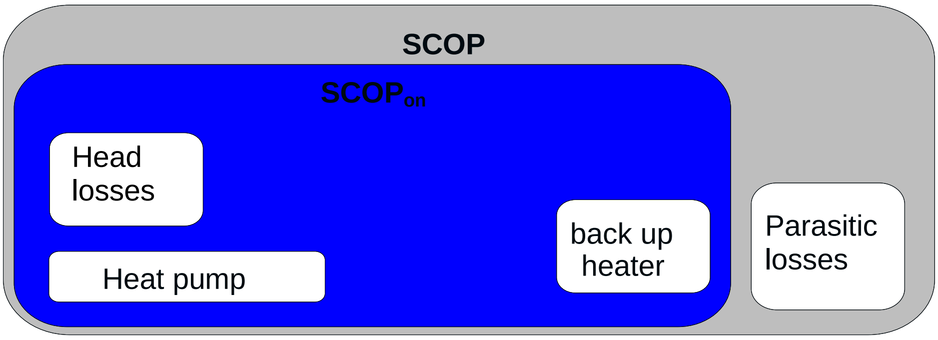
$SPF_{HW,\,hp}$ is the seasonal performance factor of the heat pump with regard to the heat produced by the heat pump. It includes the heat pump in space heating mode and production of sanitary hot water, the auxiliary energy input for the source system and the auxiliary energy for the heat pump in standby mode.

6.3 Ecodesign LOT 10

LOT 10 applies to “residential room conditioning” appliances (air conditioners and ventilation) with cooling power ≤12kW. It describes a calculation model for calculating the seasonal energy efficiency for operating in heating or cooling mode. This model will probably be replaced by prEN14825 within shortly.

System limits

The model can be used to calculate the seasonal performance factor for an air/air heat pump. The model does not include any losses from the house. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for.



Input to the calculations

To use the calculation model provided by the excel sheet the load profile of the building, P*designh* has to be selected. There are nine different sizes to chose between from size 3XS to XXL that spans from 1.1 kW to 19.2 kW. The function of the heat pump is set to either “heat only” or “heat and cool” and the type of heat pump is set to “split” or “multi-split”.

The model is a bin method with three different climates for the heating season and one for the cooling season. A table declares the number of bin hours occurring at each bin temperature, T_j, for each specific climate. The lowest temperature for each climate respectively is declared the design temperature, T_{design}. The part load ratio (of the building), pl_j, is calculated from the equation below:

$$Pl_j = \frac{(T_j - 16)}{(T_{design} - 16)}$$

The reference annual heating demand, Q_{HE} is decided in kWh for each climate as a product of P*designh* and the number of full load heating hours that corresponds to each climate.
Load fractions fracA, fracC and fracW indicate the fraction of the total heating demand (load) occurring in a specific bin at a specific climate. The fractions are given by:

$$frac_j = \frac{n_j * pl_j}{\sum_{j=1}^{40} n_j * pl_j}$$

Input to the calculations is the COP and capacity of the heat pump at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C(-15°C is only required for the colder climate). The heat pump should be tested at part load to deliver the required heat load of the building at each temperature level. At this point the paper version is not consistent. In one way it says that the capacity of the heat pump at each bin shall complete the energy demand of the building at the part load declared by the product of the annual reference heating demand, Q_{He}, and $frac_j$, but in one way it says that the energy demand is declared by the product of the part load ratio, Pl_j, and P*design.* However the excel sheet uses the first alternative and therefore care should be taken when deciding the operating points (the required effect at each temperature bin) for testing the heat pump. This alternative does not provide any effect balances. Since one house is chosen for the calculations the required effect at each outdoor temperature should be the same among the climates, but this is not the case.

In cases where the heating power supplied by the heat pump is not enough to cover the energy demand of the building in a specific bin, the difference is filled up by a backup heater with a declared capacity of COP=1. Deciding the part load from the product of Q_{He}, and $frac_j$, might result in an underestimated effect demand and therefore underestimate the required backup heating.

Instructions of how the heat pump shall be tested are given in the method for each type of operation respectively; fixed capacity units, staged capacity units and variable speed capacity units.

A degradation factor Cd, which is the efficiency loss per kW of output power when cycling the heat pump, is decided from a specific cycling test.

The energy consumption for the heat pump when operating in thermostat off mode, off mode and crankcase heater mode is decided in tests, but is only required for the calculation of SCOP.

The turndown ratio for heating, which is the lowest steady state over the maximum power and the binlimit, which is the lowest operating temperature of the heat pump, is used as input to both of the SCOP calculations.

<u>Output from the model</u>
This model is used to calculate two different seasonal performance factors:

COP_{ON} is a seasonal performance factor for the heat pump that includes electricity of the backup heater. COP_{ON} is calculated by the total electricity used by the heat pump and the backup heater over the total heat demand of the building.
(LhpC_tp*COPC_tp+resC_tp)/LhsysC_tp

SCOP is a seasonal performance factor which unlike COP_{ON}, also includes the electricity consumption of auxiliary energy for the heat pump operating in thermostat off mode, off mode and crankcase heater mode.

The energy of the backup heater is included in all seasonal performance factors that results from the excel-calculation sheet.

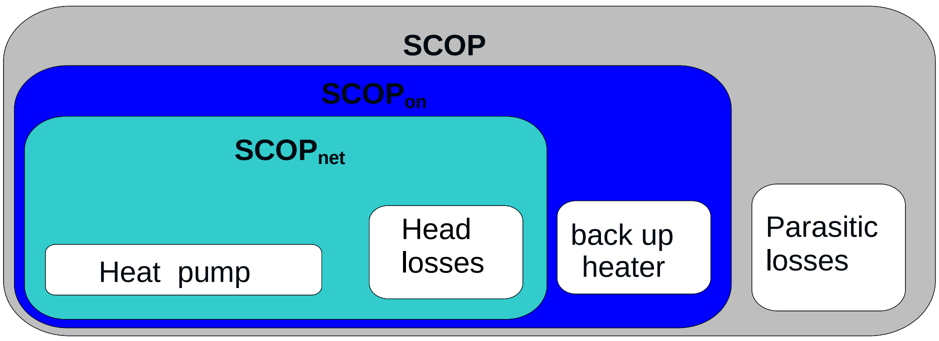
The annual electricity consumption split up in supplementary heating, heat pump operation and auxiliary heating is given from the calculations.

The annual carbon emission and label energy class is also result of the calculations.

6.4 PrEN14825

This is a standard under development that aims to cover the laboratory testing and a calculation model for SPF calculations for electric driven heat pumps. The heat pumps are tested at a number of different part load conditions (4-6) designed for heating or cooling the house to a set temperature of 16°C at different outdoor temperatures. Different test conditions are given for each type of heat pump.
This standard serves as an input for the calculation of the system energy efficiency in heating mode of specific heat pump systems in buildings, as stipulated in the standard EN15316-4-2:2008.

System limits
The model can be used to calculate the seasonal performance factor for air/air- ground source- and air source- heat pumps. The model does not include any losses from the house. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for. The system boundary in SPF 4 applies. (Data is treated according to EN14511 where the effect of heat sink pumps and ventilation fans is corrected to overcome the pressure differences of the heat pump.)



Input to the calculations
The calculation of the seasonal performance (SPF or SEER) is performed using a temperature bin method where each bin represents one degree Celsius and the number of bin hours occurring at the corresponding temperature is given. The cooling season is represented by one climate that span from 17°C-40°C while the heating season is represented by three different climates: one colder, one average and one warmer, that

span from -30°C-15°C, see Table 29 and 30 in prEN 14825:2009 draft Nov 09. Each climate corresponds to one design temperature and one design heat load of the building.

The heating/cooling demand and the number of bin hours for the different climates are determined as templates, taking different aspects into account; the climate, type of building and building characteristics, set point and set back settings and internal gains. Those aspects also decide the number of hours in which the heat pump works in active mode, thermostat off mode, standby mode, crankcase heater mode or off mode. The electricity consumptions at the different modes are determined from tests. These effects are called the parasitic losses.

Input to the calculations is the COP and capacity of the heat pump tested at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C (-15°C is only required for the colder climate). The heat pump shall be tested in equivalence with standard EN 14511:2007, with the same test methods, test set up, uncertainty of measurements and the way of evaluating data. The heat pump shall be tested at part load to deliver the required heat load of the building at each temperature level. Instructions of how the heat pump shall be tested by means of part load and type of operation; fixed capacity units, staged capacity units and variable speed capacity units, are given in this method.
The required part load for the building at the test points are given by:

$$Part\ load\ ratio = \frac{(T_j - 16)}{(T_{design} - 16)}$$

Where T_j is the outdoor (bin) temperature and T_{design} is the lower temperature limit of the selected climate.

If the declared capacities of a unit matching with the required heating/ cooling demand the corresponding COP/EER value is to be used. This may occur with staged capacity or variable speed capacity units. If the declared capacity is higher than the heating/cooling loads, the unit has to cycle on/off. Then a degradation factor (Cd (air/air or Water/air) or Cc (others)) has to be used to calculate the corresponding COP/EER values. Cd and Cc can be determined by testing; else a default value of 0.25 and 0.9 respectively is used.

The bivalent temperature, which is the lowest temperature when the heat pump can deliver 100% of the heat demand of the building, is necessary to use the excel sheet. The design heat demand of the building is a consequence of the stated bivalent temperature. The reference annual heating demand, kWh/a, is given by the product of the full load in heating P_{design} and the equivalent number of heating hours.

The operation limit of the heat pump is set to the lower temperature limit for which the heat pump can operate.

Output from the model
With above input the excel sheet gives two different SCOP: $SCOP_{NET}$ and $SCOP_{ON}$. $SCOP_{NET}$ is the seasonal performance factor for the heat pump, while $SCOP_{ON}$ also includes the electricity and heat delivered to the building from a backup heater.
The paper version of the standard also calculates a seasonal performance factor, SCOP that includes the parasitic losses of the heat pump. The effect from each operational mode is tested according to the standard while the corresponding operational hours for each mode respectively are found in a reference table.

6.5 EuP LOT 1 - Boiler testing and calculation method

This model is used to calculate the specific seasonal energy efficiency *etas* of a space heating boiler. The model contains possibilities to include several different types of space heating appliances in the efficiency calculations, such as boilers, heat pumps, electricity or solar systems. The types of heat pumps included in the model is air source and ground source heat pumps tested in either floor heating- or in radiator heating mode. The model only applies for space heating.

System limits

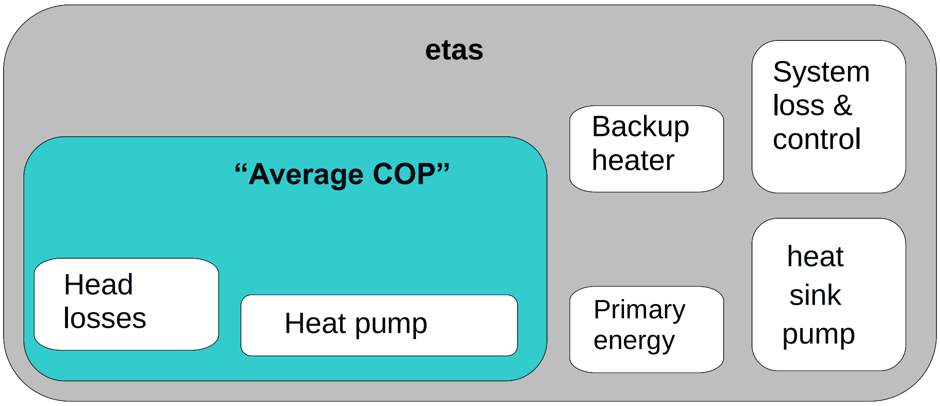
Heat pump data is taken from tests according to EN14511, therefore the head losses from heat source fans or liquid pumps are taken into account in the heat capacity and COP data. This model also includes the heat sink liquid pump.

The model takes into account the net space heating demand, Lh, of the house. The heat demand of the house is a consequence of the choice of the load profile and the so-called system losses Lsys. The size of Lsys depends on the characteristics of the boiler and the installation characteristics. The system losses include fluctuation losses, stratification losses, distribution losses, buffer losses and timer losses, which are set as a percentage that is depending on the heat demand.

The model also includes losses from control, auxiliary equipment and system buffer standing losses.

A back up heater is used to cover up the energy demand that the heat pump cannot deliver.

The electricity use in the model is accounted with the primary energy factor 2.5.



Input to the calculations

The test method for testing the heat pump refers to best testing practice e.g. EN 14511 (see document 7) except for some deviations. The test points are similar to the test points in EN14511:2007, but the temperatures of the return/feed temperature differs, se table IV.2 in the standard. In LOT 1 the temperature difference between T_{return} and T_{feed} gets larger the higher temperature of the T_{feed}. Only three test points are necessary to calculate the seasonal energy efficiency by using this model.

The calculation uses a temperature bin method to evaluate the seasonal energy efficiency, etas. There are three different climates to chose among, warmer (+2°C), average (-10°C) and colder (-22°C), see table I.1, LOT 10. Each bin describes the equivalent number of hours corresponding to the bin temperature with a resolution of one bin/K. Input data to the calculations can be either the test points given in this method or test points given in EN 14511.

The maximum heating capacity, Pmax, at the different climates is calculated from the heating capacity data obtained in the test. It is not possible to choose the size of the required heat load for the building, but is given by the model for each bin level based on the capacity of the heat pump. To meet the lower heat load requirements at the different bin levels, the heat pump is assumed to work in part load condition. The heat pump does not have to be tested in part load operation; instead the model uses a degradation factor, Cd, to calculate the COP when working in part load condition. Cd can either be obtained from tests or a default value, Cd=0.15, can be used.

For fixed capacity units the default is COPmin= 0.89*COP at power output Phpmin=0.5*Php.
For staged capacity units the default is COPmin= 0.975*COP at power output Phpmin=0.5*Php.
For variable capacity units the default is COPmin= COP at power output Phpmin=0.4*Php.

It is optional to choose whether the heat pump operates with night set back or not. The bin assumes constant night temperatures during night set back to +1°C, +6°C and 0°C for each climate respectively.

Other inputs to the calculations is type of heat pump, type of operation of the heat pump, type of control of the heat pump, type of heating (floor heating or radiator heating), minimum source operating temperature, the effect of auxiliary equipment and backup electricity heater.

Other possible energy sources can also be chosen, but this chapter only treats the heat pumps.

Output from the calculations
The model calculates the energy use and losses based upon constant fractions. The fraction of the energy use and the different losses is displayed by the model. A diagram shows the energy supply per temperature bin and how it is covered from different energy sources. The seasonal energy efficiency, etas, is calculated.

$Etas = Lh/Q_{tot}+cctrl$ *where* $Q_{tot}=Lh + L_{sys} + Q_{gen} + Q_{el}$

etas is the net space heating demand of the house over the sum of the generated heat of the system. Q_{tot} is the sum of the space heating demand (Lh), the losses from the heating system (L_{sys}), the primary energy losses of the energy input to the system (Q_{gen}) and the energy needed by the auxiliary equipment such as control and heat sink pumps (Q_{el}).

All electricity used by the heat pump and the backup heater is multiplied by a primary energy factor of 2.5. The model is not transparent. It is tricky to follow the outputs of the model since it consists from several excel-sheets and the information turns up all over. It is also difficult to understand all steps of the calculations. To be able to compare the results with field measurements and prEN14825 a value of SPF, the so called “average COP” (see the system boundaries) is calculated without the system losses. Average COP corresponds to $SCOP_{net}$ in prEN14825.

6.6 SP-method A3 528

SPA3 528 is a calculation program that is used to calculate the seasonal performance factor and energy saving over the year for houses having a defined heating requirement. It can be used for air/air heat pumps, air source heat pumps and ground source heat pumps. The heat loss from the house is defined in the program and given as the total loss factor, k-value, of the house [W/K]. The method can be used to calculate the energy requirement of a building with a k-value of either 109 W/K or 199W/K. A duration diagram of the outdoor temperature can be calculated from the mean annual temperature and together with the loss factor, the area under the duration curve gives the actual power requirement.

The heat pump is tested in accordance to EN 14511 at outdoor temperatures of -15°C, -7°C, +2°C and +7°C with an indoor temperature of +20°C. The heat pump is also tested in part load conditions according to CEN/TS 14825 at +7°C (75% and 50%) and at +2°C (50%). The lowest ambient temperature is assumed to be -15 °C and no heating is assumed to be required for ambient temperatures above +17 °C. The output data from the tests, thermal heat capacity and electrical input power, is used as input to the calculations.

7 Strengths and weaknesses with current methods

7.1 prEN14825

Strenghts

A strength of standard prEN14825 is that it includes all kinds of heat pumps (except exhaust air heat pumps). The model treats heat pumps both in heating and cooling operation. The fact that the heat pump is tested in exactly part load should result in more sufficient results compared to degradation coefficient etc. The model is foreseeable and quite easy to follow.

Weakness

The model is not completely clear with its definitions of part loads. The part load ratio for which the heat pump is to be tested is the part load energy demand of the building at the corresponding temperature bin. To perform the SPF calculations according to prEN14825 the heat pump is tested at a certain climate (A,W or C) and a certain heat load profile for the building. This means that the test data might not be suitable for another climate or another heat load.

It is also not completely transparent since it describes (ANNEX C) the reference heating/cooling demand and the number of hours in each operational mode (active mode, thermostat off mode, standby and crankcase heater mode) is decided from weighted climate, type of building, internal gains, set back setting and so on, but there is no reference that describes the calculations. Therefore it is not possible to recalculate the hours to fit specific needs. The climate hours that describes the temperature bins does not seem to be adjusted in any ways since it is the same hours that is used in Ecodesign LOT 1.

Another weakness is that the model does not include domestic hot water.

Possibility

The model could be developed so that it would be possible to decide the energy demand of the house. It could also be a possibility to fit the model to your own climate. Maybe the ground water temperature and thereby the bore hole temperature could be climate depending.
It should be obvious how interpolations or extrapolations of capacity and/or COP should be performed to avoid differences between users.

Risk

The performance of water/water heat pumps can be overestimated, especially at the cold climate, since they are only tested at +10°C at the cold side (in reality the ground water temperature can be lower than this). This can also be the case for other ground source heat pumps.

The degradation coefficient Cc might be a disadvantage for a ground source heat pump when default values are used. Cc =0.9 is a larger degradation of GSHP’s than what is shown in reality. There is a risk that the requirement of having heat pumps tested in part load might lead to extensive laboratory tests, which is costly. It is also difficult to get sufficient data from existing laboratory tests, since few heat pumps are tested in part loads.

7.2 EN 15316-4-2

Strength

This model is very wide and thorough in its content. It treats both room heating and tap water production. The model is adaptable to different climates and the resolution of the temperature bins can be chosen.

The model specifies the requirements and losses of the certain house and defines recoverable respectively unrecoverable energy.

It is not necessary to test the heat pump at the part loads, since there are default values that can be used.

The model can be used to calculate the SPF for the entire system with the building included or only for the heat pump.

Weakness

The strengths of this model could also turn out to be its weaknesses. The wideness of the model makes it complicated and twisty. There are too many aspects that are taken into account in the calculations. The standard refers to several other standards for calculations of losses and needs. The model requires large knowledge of the house.

The fact that default values can be used to calculate the operation in part load for the heat pumps can result in lower accurateness of the model.

The model does not treat operation in cooling mode.

Possibility

The model can be studied and give input to a new easier model.

Risk

There is a present danger of doing mistakes when using the model. The large amount of data that is taken into account will probably result in much estimation that will differ from case to case and will therefore result in incomparable outcome of the model. Also the same heat pump installation can probably give different results depending on the way it is calculated, (choosing method, input, accuracy and test points).

7.3 EuP LOT 1

In general, the Energy Using Products (E**u**P) Directive have broadened to include also Energy Related Products (E**r**P), but for the treatment in this report, we choose to use the term EuP, since heat pumps are energy using.

Strength

Test data from EN 14511 can be used in the calculations. The model provides default values to recalculate the test points to fit the part load of the heat pump for the different kinds of heat pumps (fixed capacity, staged capacity och variable capacity). The capacity and corresponding COP values are then interpolated between the temperature bins. However, the accurateness of the recalculation is unknown.

The model itself has suggested test points with a radiator curve (supply temperature) that is adjusted to the outdoor temperature. At colder outdoor temperatures the supply temperature is higher and at warmer outdoor temperatures the supply temperatures are lower.

The model can be used to calculate how to cover the energy need of the house by using different techniques, for example solar cells, heat pumps and fossil fuel. This is a good thought, but might not be interesting in this project (?).

Weakness

Unfortunately the model still contains bugs and technical mistakes in the equations and the way of thinking. It seems to be adjusted to boilers and bio boilers instead of heat pumps.

The model does not include a power balance, but is doing a temperature balance instead. This makes the distribution of the energy need and the required amounts of backup heat differ from the theoretical needed.

The model includes a decided fraction of heat loss that cannot be escaped from. For example if the heat pump does not use night set back a default penalty loss of 12% from the total delivered energy is subtracted. The losses from the apparatus and system operation are also decided in percentages.

At part load operation there is no change in the system flows. This does not seem right with controlled radiators. (Should the radiators be controlled or is it enough with a displacement/adjustment of the radiator curve?)

The night set back function uses the same night temperature all year around, which is not the case in reality.

It is not possible to choose the energy requirement of the house; instead the energy demand is an outcome of the capacity of the heat pump. If the heat pump is not monovalent also the fraction of backup heat is needed to decide the energy demand of the house.

GSHP's are treated unfairly when recalculating the operation data to part load operation. The ground source heat pumps are degraded by a factor 0.89 at 50 % of the delivered capacity. (The Cd factor, i.e. the on/off control, is overestimated for water borne systems)

Even though the program is transparent in the sense that all equations are reported in the model, it is very hard to understand and follow the calculations, and the program cannot be said to be transparent in the general sense. The interface of the program is not very friendly and can easily confuse the user. The model does not include tap water.

Possibility

Making the ground water and borehole temperature climate dependent might lead to results more sufficient to its actual installation spot.

Risk

The model is not adjusted to fit heat pumps and is disadvantaging heat pumps. Despite this the COP and capacity of water to water heat pumps can be overestimated since they are tested at +10°C at the cold side (this can also happen to ground source heat pumps, but probably not to the same extent).

7.4 EuP LOT 10

Strength

This model can be used both in heating and cooling mode and it has three different climates both for the cooling season and the heating season. The model has reference heating/cooling demands to choose between.

Weakness

The model takes only air to air heat pumps into account. In accordance to prEN14825 the test points for the heat pump has to be chosen specifically to fit the chosen climate and heat profile of the house.

In accordance to LOT 1 the model does not include an effect balance at each temperature bin. This results in that the heat demand of a house at a specific temperature bin is different at different climates and that the heat requirement of a backup heater is misleading.

The model does not seem to be entirely consistent, partly it is contradicting itself.

Possibility

To make the model usable at other spots it would be better to make it possible to use other climates. Now the model only provides a number of specified heat loads of the house. It would be useful to be able to freely choose the heat demand of the house. There is a risk though, that since the heat pump has to be tested in part load, it has to be tested at each specific heat requirement.

Other types of heat pumps could be included in the model. The model only provides the SPF (SCOP) with the backup heater included. For comparable reasons, it would be useful to include a SPF with backup heater excluded.

Risk

It is not obvious whether the excel model is compatible with the standard. There are also some calculations in the standard that seems to be incorrect.

8 Comparison of existing calculation methods and results from field measurements

8.1 Heat (and cooling-) demand of the house

This study is focused on heat pumps for indoor heating. The study is made in houses with different heat demand. The ground source heat pumps in this study are considered monovalent, but it is difficult to determine the actual energy demand of the house. When using the calculation models the required heat load of the house is decided by the capacity of the heat pump.

The studied air to air heat pump is not monovalent. The energy demand of the house with the heat pump installation was estimated in the field study. When using the calculation models the energy demand of the house were tried to be the same as in the field study.

8.2 Indoor climate

The indoor climate is expected to reach 20°C for all models. In the calculation models the heat pump is used to reach a temperature of 16°C. Internal gains are expected to contribute to the last temperature increase.

The actual indoor temperature has not been measured in the Fraunhofer field measurements. Thereby it is not possible to compare the real indoor temperatures with the temperatures estimated in the calculation models.

8.3 Outdoor climate

The outdoor climate follows the climate of the year. The calculation models use the same temperature climate when calculating SPF for the ground source heat pumps. The climate corresponds to a European average climate, Strasbourg, with the coldest temperature of -10°C.

The field measurements of the ground source heat pumps are carried out in Germany. The heat pumps installations used for the SPF calculations are spread over the country, from the Hamburg area in the north to Stuttgart in the south. In the calculation models the average climate is chosen as the climate mostly corresponding to the German.

The air to air heat pump installation is made in a climate that is similar to the “colder” climate. Therefore the colder climate is used in the calculation models when calculating SPF for the air to air heat pump.

8.4 Definition of SPF field measurement system boundaries

In the ongoing EU project “SEPEMO-Build (SEPEMO short)” four SPF’s with different system boundaries are defined. The definitions from the SEPEMO project have been used for calculating the SPF for the field measurements. The four defined SPF’s are:

SPF_1 includes only the heat pump unit itself. Thereby SPF_1 is identical to the average COP for the measured period.

$$SPF_{H1} = \frac{Q_{H\_hp} + Q_{W\_hp}}{E_{HW\_hp}}$$

SPF_2 consist of the heat pump unit and the equipment needed to make the heat source available the heat pump.

$$SPF_{H2} = \frac{Q_{H\_hp} + Q_{W\_hp}}{E_{S\_fan/pump} + E_{HW\_hp}}$$

SPF_3 represents the heat pump system SPF. SPF_3 includes the heat pump and the heat source pump as in SPF_2, but also the back up heater.

$$SPF_{H3} = \frac{Q_{H\_hp} + Q_{W\_hp} + Q_{HW\_bu}}{E_{S\_fan/pump} + E_{HW\_hp} + E_{HW\_bu}}$$

SPF_4 includes all parts relates to SPF_3, additionally SPF_4 also includes the distribution of the heat.

$$SPF_{H4} = \frac{Q_{H\_hp} + Q_{W\_hp} + Q_{HW\_bu} + Q_{DHW\_bu}}{E_{S\_fan/pump} + E_{HW\_hp} + E_{bt\_pump} + E_{HW\_bu} + E_{B\_fan/pump}}$$

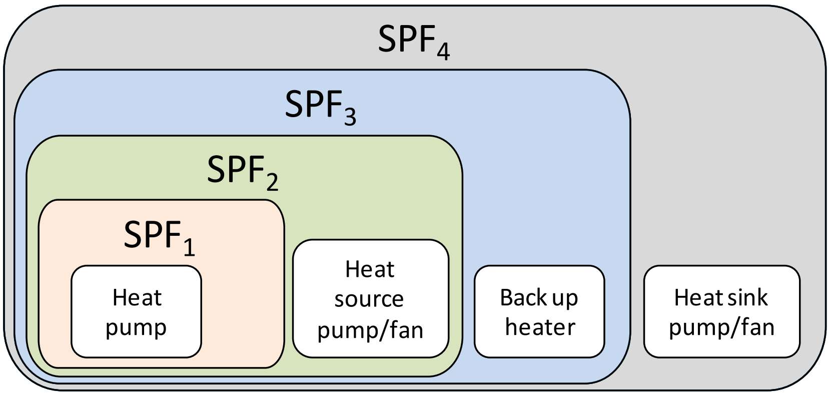


Figure 6 System boundaries for calculations of SPF

SPF_1 is normally measured on the brine/water sides of the evaporator/condenser, but it could also be measured directly in the refrigeration loop with e.g. the Climacheck equipment [10]. This requires measurement of the pressure and temperature of the refrigerant. This methodology is very efficient if the status/condition or diagnosis of the heat pump is to be evaluated, but generally in domestic heat pumps, the measurement is not easy to carry out since measurement sockets are not generally installed.

8.5 Calculation of SPF

In this study SPF is calculated for three of the four different system boundaries and categories by using data from the field measurements. The system boundaries used are SPF_2, SPF_3 and SPF_4 and are described in section 8.4. The categories are “heating only”, “heating and domestic hot water production” and “domestic hot water production”.

For facilities where the installed heat pump also is tested in a laboratory, the laboratory test results are used to calculate SPF by using the calculation models. This is the case for seven different ground source heat pumps and one air to air heat pump.

This chapter will explain how the calculations are performed and what assumptions are made for the different models.

Field measurements

Ground source heat pumps
All analyzed heat pump systems are installed in German single family houses with floor heating. The heat pump is more or less monovalent, only a very small amount of backup heat has been used during the year of measurements. The heat pumps in the study were all installed in new built houses during the years 2004-2008. The data used for the SPF calculations are based on field measurements carried out during one year, with one exception the SPF for site no. 1 is based on data measured from January to August.

The calculations of SPF's are based on the field measurements data from the Fraunhofer study. In the data we have received from the Fraunhofer study the total energy consumption for the heat pump system and its components is presented as well as the energy consumption divided into energy used for space heating and energy used for production of domestic hot water.

In this project we have not been able to evaluate exactly how these allocations have been made. For some of the studied installation sites a part (up to 20%) of the total electricity consumption has been allocated neither to space heating nor to the domestic hot water production. This is mainly the case for the electricity consumption. For the heat produced no energy gap is seen between the total energy production and the energy divided into space heating and domestic hot water.

The calculated SPF's in the study are based on the energy allocated to the space heating only, this in order to make the results comparable to the results from the calculation models in prEN14825 and Lot 1, which not include the production of domestic hot water.

Air to air heat pumps
The field measurement of the air to air heat pumps is carried out in single family houses located in the Borås area of Sweden. All houses in the study have electricity driven radiators for back up heating. The field measurements are based on SP method 1721. From the field measurements SPF_2 and SPF_3 has been calculated as described below.

The electricity consumed by the heat pump, W_{HP}, is measured continually while the produced space heating is measured at five "performance tests" done at different outdoor temperatures. During the performance tests the heating capacity of the heat pump is measured during stable conditions and is thereby not including any defrost period. Therefore the calculated COP for each test point is based on data from only a part of the operating cycle.

The total amount of heat produced during the total measuring period needs to be calculated based on the five performance tests. The calculations are made as follows:

$$Q_{HP\_year} = \sum W_{HP\_month} * COP_{average\_month}$$

COP is calculated from the performance tests at made at different outdoor temperatures. During the performance test both the electricity consumed (W_{HP}) and the produced heat (Q_{HP}) is measured and COP can be calculated as:

$$COP = \frac{Q_{HP\_test}}{W_{HP\_test}}$$

From the five performance tests the COP for the heat pump can be expressed as a function of the outdoor temperature. From this function an average COP for each month is calculated based on the average temperature for the month. Knowing the electricity produced each month by the heat pump the SPF_2 can be calculated:

$$SPF_2 = \frac{Q_{HP\_year}}{W_{HP\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year}}$$

The electricity consumed by the backup heaters is also measured, thereby SPF_3 is calculated as:

$$SPF_3 = \frac{Q_{HP\_year}}{W_{HP\_year} + W_{backup\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year} + W_{backup\_year}}$$

Due to lack of data from laboratory testing of the heat pump models included in the field measurement, this study has only been able to compare the SPF values from the field measurements with the SPF calculated with the calculation models prEN14825 and Lot 10 for one of the tested heat pumps.

prEN14825

Ground source heat pumps
When using prEN14825, data according to Table 13 has to be filled in. The chosen climate, "average" gives that Tdesign is -10°C. Tbivalent is the outdoor temperature where the capacity of heat pump covers the heat demand of the house. It is set to -10°C, to make the heat pump monovalent, like in the field study. TOL, the operation limit temperature, is set to -25°C. This temperature declares where the heat pump no longer can operate. The model calculates Pdesign as a result of Tbivalent and is the heat demand of the house at Tdesign.

Table 13. Input data for the prEN14825 calculation model.

T design	-10	°C
T bivalent	-10	°C
T OL	-25,00	°C
P design	8,81	kW

The test conditions for the heat pumps were taken from Table 20 in the standard, brine to water heat pump, average climate and low temperature application. The unit is assumed to be a fixed capacity unit with fixed outlet temperature. The heat pumps in the study where all tested in full load according to EN 14511. For the part load conditions the COP was calculated by using equation 12 in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The capacity and COP at Tbivalent and TOL is set to the maximum, while the COP for the delivered capacity at the different outdoor temperatures is calculated by using equations from the standard prEN14825. The default degradation factor where Cc=0.9 is used.

Air to air heat pumps
The data for SPF calculations regarding air to air heat pumps are taken from the field measurements. There are no laboratory data available for the heat pumps tested in the field study.

The colder climate is chosen for the calculations, since this climate is similar to the climate where the field installation is. The bivalent operation point of the heat pump is calculated by using SPA3528, which is another model for the calculation of SPF. The bivalent point is 0°C. The operation limit point is set to -20°C.

At -7°C the heat pump operates in full load to deliver heat to the house. At +2°C and at +7°C the heat pump operates in part load. COP for part load operation is interpolated by using linear interpolation between existing test points. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 50% and 57% of the heat pump capacity. At +12°C the required heat load is so small that the heat pump is assumed to cycle on/off. The capacity of this point is calculated by using equation 11 in the standard. The COP for the bivalent point is interpolated from test points in full load operation at +2°C and -7°C.

<u>*Lot 1*</u>
In Lot 1 there are some general inputs that has to be filled in into the excel sheet. The following inputs are used:

- Reduced setback: Yes
- Radiator (with setback): No
- Floor heat (24h): Yes
- Control: 4 – Weather ctrl BT
- Pump: 3 fixed speed
- Pump timer: 24h
- Buffer: No
- Tmino: -25°C

The only heat generator in use is heat pump. No back up heater is included in the calculations.

The default degradation factor, Cd= 0.15, is used. Default is also used for hpaux (=30W) and hpsb (=10W). The test conditions are taken from the reference test conditions in table V.3. in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The model recalculates the test data to fit with the test conditions of Lot 1 (table V.2.).

Data for part load operation is calculated from equations of "option B" at page 27 in the standard, where COP_{min}= 0.89*COP at power output Phpmin=0.5*Php for a fixed capacity unit.

From Lot 1 two different results are obtained, "etas" and "average COP". Etas are calculated by involving the primary energy factor of 2.5 which makes it difficult to compare with other calculated SPF. However, "average COP" corresponds to SPF 1.

<u>*Lot 10*</u>

Air to air heat pumps
The design load of the house is chosen to 8,5kW, which is the design load that best corresponds to the size of the house in the field measurement. The house in the field is installed in a climate, similar to "colder" climate, therefore "colder" is chosen. The test

points for the calculation are given in a table at page 24 in LOT 10 Annex II. The heat pump is tested according to EN 14511 and CEN/TS 14825 for part load conditions.

The heat pump is a variable capacity heat pump, but since the heat pump is not tested at exactly the required heat effect (within ±3%), the calculations of COP has to be performed in accordance with a staged capacity unit.

At -15°C and -7°C the delivered capacity from the heat pump is lower than the house requires; capacity and COP data are taken from operation in full load at these outdoor temperatures. An exception from the standard is made, since the standard proposes a recalculation of the COP at those points. The recalculation does not seem to make sense and is therefore ignored.

At +2°C and +7°C the heat delivered from the heat pump exceeds the required heat from the house and is therefore operated in part load. COP for part load operation is interpolated by using the equation for staged capacity units at page 26 in the standard. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 57% and 44% of the heat pump capacity.

The heat pump is not tested at +12°C. Full load operation at +12°C is extrapolated from test data at +7°C and +2°C. 50% part load is extrapolated from 50% part load operation at +7°C and +2°C. COP for the required effect is extrapolated by using this data. Each extrapolated COP value is corrected with a degradation factor of 0.975.

Default values are used for the degradation factor (Cd=0.1), turndown ratio heating (=25%), thermostat off mode (50W), crankcase heater mode (=10W) and off mode (=10W). The bin limit is set to -20°C.

8.6 Analysis of the results

The results from the SPF calculations of the different heat pump installations in field is compared with the results obtained from the laboratory data used in calculation models.

Ground source heat pumps
Most of the heat pumps installed in field operates both in floor heating mode and produces domestic hot water. The measurements include both kind of operations and the results are presented in Table 14and Figure 7 below. SPF for domestic hot water production is always lower compared to operation in heating mode. The energy balances is not 100% complete for the field measurement, which is quite common in field measurements, since heat losses are present, but cannot be measured directly as they can be in the laboratory.

Table 14 The table shows two different SPF from the field measurements in two different levels. SPF for heating and DHW (domestic hot water) is lower than SPF for heating only. This is because COP for domestic hot water production is lower than COP for heating

Results field measurements				
	Heating and DHW	Heating and DHW	Heating only	Heating only
	SPF 1	**SPF 3**	**SPF 1**	**SPF 3**
site 3	3,70	3,46	4,66	
site 6			3,86	3,43
site 8	4,13	3,02	4,71	
site 9	3,97	3,64	4,53	
site 11	3,62	3,32	4,71	4,56
site 13	2,71	2,55	3,99	3,83
site 14	4,14	3,55	5,43	5,16

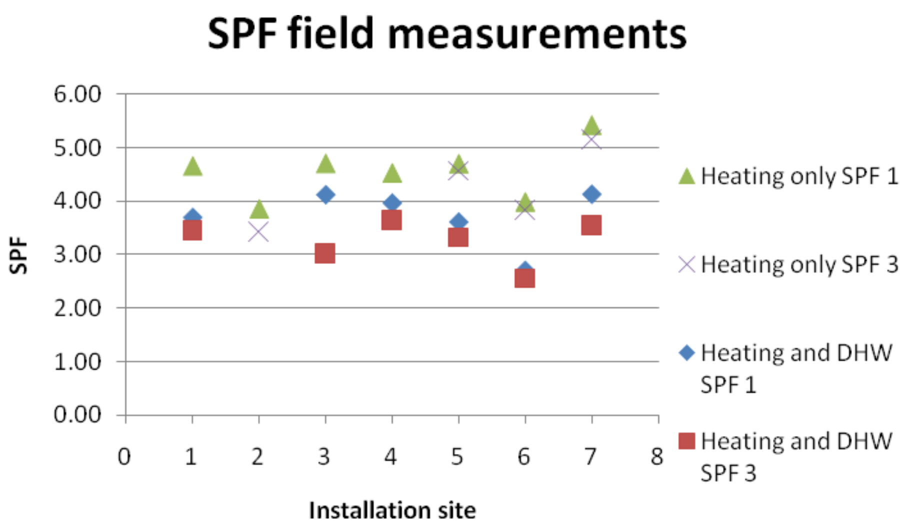


Figure 7 The figure show SPF results from two different SPF, “heat only” and “heat and DHW” (domestic hot water heating) at two different levels, “SPF 1” and “SPF3”, from field testing.

The conditions for measurements in a laboratory and in field differ with respect to various factors e.g. the boundary conditions. SPF_1 in field measurements includes the electrical energy from the heat source brine pump, while “average COP” and “$SCOP_{net}$” only includes the head losses. This could make the electrical energy use a little larger for the field measurements, but on the other hand “average COP” and “$SCOP_{net}$” also contain head losses for the heat sink side which SPF_1 does not. The electrical energy from the heat sink pump for SPF_1 is included in SPF_3.

Table 15. The table shows the results from using Lot 1. Average COP is comparable with SPF 1 from the field measurements. Pdesign shows the maximum capacity needed for the house

Results Lot 1			
	avg COP	etas	Pdesign
site 3	3,57	1,05	7,7
site 6	3,49	1,03	7,6
site 8	3,49	1,02	5,9
site 9	3,49	1,03	7,6
site 11	3,83	1,12	7,2
site 13	3,88	1,12	5,8
site 14	3,88	1,12	5,8

Table 16. The table shows results from using prEN14825. SCOPnet is comparable with SPF 1 from the field measurements. Pdesign shows the maximum capacity needed for the house.

Results prEN14825			
	SCOPon	SCOPnet	Pdesign
site 3	3,66	3,66	8,81
site 6	3,58	3,58	8,7
site 8	3,6	3,6	7,17
site 9	3,58	3,58	8,7
site 11	3,96	3,96	9,64
site 13	4,02	4,02	8,01
site 14	4,02	4,02	8,01

Since the ground source heat pumps in this study is considered monovalent, the comparison of the results are mainly done for SPF_1 from the field measurements and SPF that corresponds to SPF_1 from the calculation models, “average COP” from Lot 1 and “SCOPnet” from prEN14825. The results are presented in Figure 8 below.

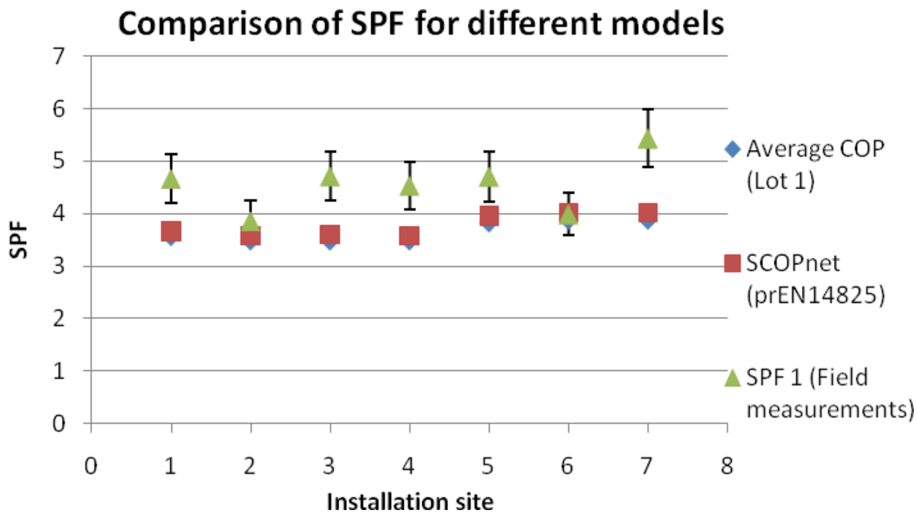


Figure 8 The figure shows a trend that SPF_1 is higher compared to “average COP” and “SCOPnet”. Field measurements imply a higher uncertainty compared to measurements in a laboratory. The bars of error show an error of ±10% to cover the margins of error.

There are two main differences between “average COP” and “$SCOP_{net}$“:

- There are differences in degradation for part load operation
- Lot 1 does not make an capacity balance of the heating demand of the house at each outdoor temperature.

The last factor results in that the design capacity, P_{design}, turns out to be larger for the house when using $SCOP_{net}$ compared to “average COP”. The result show that P_{design} for “average COP” is approximately 13-28% lower compared to “average COP” and SPF is approximately 3-4% lower. The degradation of COP is a little bit tougher when using Lot 1 compared to using prEN14825. The comparison is illustrated in Figure 9 below.

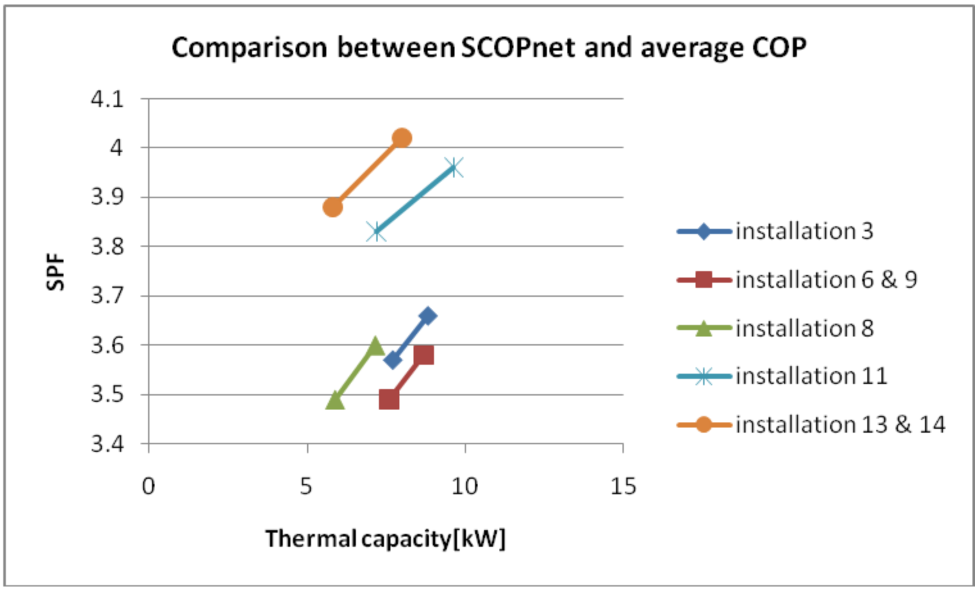


Figure 9. The figure illustrates the differences in design capacity when using EuP Lot 1 and prEN14825. The lower value corresponds to Lot 1 and the higher value corresponds to prEN14825.

Air to air heat pump
Laboratory test data was available for one of the air to air heat pumps that were studied in field. SPF from the field study and SPF from the calculation models are presented in Table 10 below. SCOPnet is the SPF for the heat pump that corresponds to SPF_1. SCOPon is SPF for the heat pump with the backup heater included and corresponds to SPF_2. SCOP is SPF for the heat pump with both backup heater and parasitic losses included. There are some problems by comparing the laboratory test data and the data from field testing, since the field tests do not include the defrosting periods. Therefore SPF from field testing might turn out a little higher than in reality.

Table 17. The table shows a comparison of result for the air to air heat pump.

Results field measurements			
	SCOPnet	**SCOPon**	**SCOP**
prEN 14825	2,52	1,96	
Field measurement	2,4	2,1	
Lot 10		2,15	2,12

8.7 Conclusions from comparisons

Some of the field installations show different SPF_1 despite that the same heat pump model is installed. This can be an indication of how important the sizing of the heat pump is. An oversized heat pump results in for example more part load operation and causes standby losses.

The calculation models show that there can be benefits when installing a heat pump where the bivalent temperature is higher than the lowest operation temperature of the year, even though backup heating is necessary.

9 Requirements for a new calculation model to evaluate SPF from lab measurements

The requirements on a new calculation model differs defpending on the aim of the model. In general, three different uses can be identified:
Based on lab data understand the consequences of technology choice in comparison with competing heating technologies
To understand the consequences of correct sizing of the heat pump
To make a correct dimensioning of the heat pump in a specific house

It should also be possible to study three modes of operation, DHW production, heating or combined DHW production and heating.

Based on the models, it should be possible to make comparisons of e.g. LCC and environmental performance of different systems

What should be included/ not included in the model?

- It should be possible to decide the energy demand of the house in the model, either by given reference loads, or by choosing a specific energy demand of the house. This should be separated into space heating and domestic hot water. When the model itself calculates the losses of the house it can be misleading and not sufficient for the actual house. This can be one boundary of the project. Alternatively, typical houses are used in typical climates, both preset in the model.

- To take into account for the climate at the installation, generally accepted spot climate data, for example Meteonorm data [9], Should be a part of the model.

- The dynamics of the house can be a part of the model. The perceived temperature of the house is not fully consistent with the actual outdoor temperature. At colder temperature dips of for example -15°C, the house will not experience the real outdoor temperature, but experiences a temperature of -12°C instead (due to internal heat gains). Even the irradiance of the sun differs between the seasons (and different spots). The energy demand of the house is affected from those variances over the year, why it might be an idea to calculate the SPF over monthly periods. Also the use of a fictive outdoor temperature would be an alternative. The climate data can be adjusted (flattened out) depending on a number of inputs, but a temperature dip is still needed in order to make a proper effect dimensioning (this is dimensioning the entire system such as deep wells etc.).
  In a serious effort to evaluate dynamics, other factors have to e incorporated in a model, such as form factor, impact of building weight, window area compared to wall area, placement of windows, etc., which make such a model very complex.

- For ground source heat pumps, the temperature of the ground is varying during the year. The model should include a correction for this. This could be expressed as a function where the ground source temperature is a function of the outdoor temperature over the year.

- The model should contain a radiator heat curve where requisite supply temperature is calculated, an example of this can be found in the thesis of Fredrik

Karlsson [8]. At a colder outdoor temperature, the supply temperature should peak; this makes the test scheme tables in EN 14511 deficient. Also other heat distribution systems, such as under floor heating, and mixed systems should be included in the model.

- Part load performance of the heat pump must be properly taken into account, and be based on relevant testing standards.

Night set back is a choice in some calculation models; this is not relevant for heat pumps and should not be a part in a new calculation model.

- Back up heaters is sometimes necessary to complete the energy demand of the house. Back up heaters should be included in the calculation model. Supplementary heating should be possible to choose between different sources of supplementary heat, e.g. electricity, solar or biomass heating.

- The possibility to include the production of domestic hot water to the SPF calculations is also a necessity in future calculation models. It should also be described how this shall be measured in tests alternatively, how the amount of produced domestic hot water shall be estimated. Today there are two main ways how to do the measurements, including the losses or not (one can measure the amount of energy that is obtained by tappings or the amount of tap water the heat pump is producing). A lot of work has already been done in this respect in the IEA HPP Annex 28 [13]. Also, there is a CEN standard on the way on how to treat DHW production. This standard however does not take into consideration combined heating and DHW production.

- Accumulators should be possible to include in the model.

- A model must contain clear system boundaries for what is to be included in the calculations and how measurements are performed. As a basis, the system boundaries presented in the SEPEMO project [12] is recommended.

- The model must be transparent so it is possible to follow and understand the calculations. The studied models all contain parts that are more or less transparent. For example how the estimation of the number of equivalent heating hours is performed is not shown in any method.

An outcome of the results should be to see that a properly sized heat pump is the best alternative to install. An oversized heat pump will result in unnecessary on/off cycling losses and an undersized heat pump will result in unnecessary high back-up heating.

For the calculation, either BIN methods or hour by hour calculations could be used. The existing calculation models based on heat pump performance testing according to standards are all using BIN models. Therefore, to keep a clear connection to existing test standards, it is the easiest to base a new model on BIN models. A hybrid model using chronological BIN’s could also be an interesting option to look into.

The drawback with this approach might be that dynamic effects, especially in cases with large or well stratified accumulators are not treated in a way that the full potential of these units are revealed.
In the proposed IEA Annex, a thorough investigation of the positive and negative effects of these approaches should be performed.

10 Conclusions

For a new calculation method to better represent real SPF values there is a need to rely on consistent sets of performance data acquired from lab testing. These lab testings guarantee consistency, repeatability and reliability.

If the objective is to give better values for individual houses, more details on the building envelope, climate data etc. must be provided for the specific setup.

If the objective is to give reliable values for typical conditions, type houses in type climates should be used, but with better details than currently used in existing models.

A new model should include combined DHW and heating to the full extent.

Other key numbers, such as energy performance, energy savings, environmental performance and life cycle cost should be developed in a harmonized way. These key numbers act as a complement to SPF values.

System boundaries should be transparent and comparable with other heating technologies. The use of more than one system boundary allows analyzing parasitic losses from pumps, fans and piping work. The use of different system boundaries also allows to communicate what parts of a heat pump system that working properly or not satisfying in the final installation.

It is important to not only act as a national project in the case of SPF, since much of the activities are on an European or even global level, so the results from this project will be very valuable input to the international work within IEA.

11 Further work

The results from this project will be fed into the IEA Annex on SPF, and further development of a calculation method can be proposed to relevant stakeholders from that Annex.

12 Publications from this project

Within this project, SP have presented results in the form of an article to the Swdish magazine KYLA., "Jämförelse av metoder och fältmätningar för utvärdering av årsvärmefaktor (SPF) ". The article was planned to be published in issue 3, 2010.
An abstract has been submitted to the forthcoming IEA heat pump conference in 2011.

13 References

1. Meeting notes SPF Annex Chicago 2009-01-27
2. Meeting notes Paris: HPP SPF Annex 2010-03-05
3. A common method for testing and rating of residential HP and AC seasonal performance, draft legal text IEA HPP Annex
4. Ekonomisk och driftserfarenhetsmässig utvärdering av bergvärmepumpar, Utarbetad av Per Levin, Projektengagemang Energi och klimatanalys Danderyd, februari, 2008
5. Two large field-teste of new heat pumps in Germany, Miara, M. 9th IEA heat pump conference, Zyrich, 2008
6. Årsmätning på fem bergvärmeanläggningar i Sjuhärad, Energimyndigheten, 2005
7. Fältmätning på Luft/Luft värmepumpar i svenska enfamiljshus, SP, 2009, to be published
8. Capacity Control of Residential Heat Pump Heating Systems, Fredrik Karlsson, Building Services Engineering, Department of Energy and Environment, Chalmers University of Technology, Göteborg, Sweden 2007
9. Meteonorm, Global Meteorological Database for Engineers, Planners and Education, http://www.meteonorm.com/pages/en/meteonorm.php
10. Climacheck website, accessed 2010-04-21, http://www.climacheck.com/
11. Prestandaprovning av luft/luft Värmepumpar i fält, SP metod nr 1721, Fahlén, P., Kjellgren, C., SP AP 1994:51, Energiteknik, Borås 1994
12. Seasonal Performance factor and Monitoring for heat pump systems in the building sector (SEPEMO-Build), IEE/08/776/SI2.529222, http://www.sepemo.eu/
13. Final Report IEA HPP Annex 28, Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating, HPP-AN28-1, ISBN: 91-85533-30-0

Appendix 1. References for field measurements, presented in RIS-format.

TY - BOOK
A1 - Abrahamsson, Thore
T1 - Skövdebadet - solenergiuppvärmning? : alternativstudie avseende uppvärmning genom värmepump, solfångare och värmelager i utebassäng
AU - Jonson, Sten
AU - Norin, Fredrik
Y1 - 1979
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningen, 0346-5616 ; 1979:71
SN - 91-540-3035-8
ER -

TY - BOOK
A1 - Andersson, Per-Åke
T1 - SIMSYS : simuleringsprogram för värmecentraler med ny energiteknik
AU - Askling, Åke
AU - Dalenbäck, Jan-Olof
Y1 - 1986
CY - Göteborg
T3 - Intern skrift / Chalmers tekniska högskola, Avdelningen för installationsteknik, 99-0354505-3 ; 24
T3 - Rapport / Chalmers tekniska högskola, Avdelningen för installationsteknik, 0280-2899 ; 1986:2
ER -

TY - BOOK
A1 - Backman, Anders
T1 - Värmeåtervinning ur avloppsvatten med värmepump för 400 lägenheter i Falun : projektering
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:103
SN - 91-540-3991-6
ER -

TY - BOOK
A1 - Björk, Erik
T1 - Praktisk provning av vattenburet värmesystem med värmepump och konvektorer/radiatorer
AU - Wiklund, Sören
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:131
SN - 91-540-3356-X
ER -

TY - BOOK
A1 - Bokalders, Varis

T1 - Energisnåla hus : [30 hus med energisnåla lösningar: solfångare, värmeåtervinning, växthus, braskamin, värmepump, passiva solhus, värmelager]
Y1 - 1981
KW - Uppvärmning (byggnader)
KW - Energieffektiva byggnader
CY - Västerås
PB - Ica bokförl.
SN - 91-534-0568-4
ER -

TY - BOOK
A1 - Boklund, Tord
T1 - Värmepump för luft-vattensystem med mekanisk vinds- och krypgrundsventilation som värmekälla : enplansbyggnad
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:53
SN - 91-540-3940-1
ER -

TY - BOOK
A1 - Buresten, Rune
T1 - Solvärme med värmepump som komplement till en oljeeldad värmecentral i Göteborg
AU - Gunnarsson, Ingemar
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:26
SN - 91-540-3891-X
ER -

TY - BOOK
A1 - Claesson, Staffan
T1 - Lågenergihus i Högås : hydrologiska förutsättningar för värmepump med en mindre tjärn som värmemagasin
AU - Sandén, Kent
Y1 - 1976
N1 - Examensarbete
T3 - Examensarbete / Institutionen för vattenbyggnad, Chalmers tekniska högskola, 99-0211330-3 ; 1976:4
ER -

TY - BOOK
T1 - Energi ur grundvatten : inventering av stora grundvattenmagasin för energiutvinning med värmepump : slutrapport mars 1980
Y1 - 1980
ER -

TY - BOOK
A1 - Eriksson, Sven-Olof
T1 - Stockholmsprojektet : energibehov i flerbostadshus med solvägg, tung stomme och värmepump : kv Konsolen
Y1 - 1993
KW - Flerbostadshus

KW - Energieffektiva byggnader
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - R / Byggforskningsrådet, 1103-6346 ; 1993:44
SN - 91-540-5590-3
ER -

TY - BOOK
A1 - Eriksson, Sven-Olof
T1 - Stockholmsprojektet, kv Konsolen : energiteknisk utvärdering av flerbostadshus med solvägg, tung stomme och värmepump
Y1 - 1993
N1 - Lic.-avh. Stockholm : Tekn. högsk.
CY - [Stockholm]
T3 - Arbetsrapport / Institutionen för byggnadsteknik, Kungliga Tekniska högskolan, 0349-6562 ; 1993:1
ER -

TY - BOOK
A1 - Faarinen, Jukka
T1 - PoD anläggning av öppet absorptionssystem.
Y1 - 1993
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, Ingenjörslinjen 80 poäng, 1102-4070 ; 1993:22I
ER -

TY - BOOK
A1 - Hammarsten, Patrik
T1 - Experimentell studie av R22/R142b som arbetsmedium i en värmepump.
Y1 - 1992
N1 - Examensarbete
CY - Göteborg
ER -

TY - BOOK
A1 - Hill, Anders
T1 - Återvinning av värme med värmepump från lokaler där spillvärme alstras
AU - Matsson, Lars Olof
AU - Ryberg, Anders
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:145
SN - 91-540-3628-3
ER -

TY - BOOK
A1 - Holmgren, Mats
T1 - Användning av värmepump för återvinning av värme vid godssvalning
AU - Stigmarker, Håkan
Y1 - 1984
CY - Jönköping

PB - Svenska gjuteriföreningen
ER -

TY - BOOK
A1 - Hult, Johan
T1 - Jämförelse mellan konventionell gaspanna och värmepump med inbyggd gaspanna
AU - Jonasson, Jens
Y1 - 2000
CY - Malmö
PB - SGC
T3 - Rapport SGC, 1102-7371 ; 113
ER -

TY - BOOK
A1 - Jeal, David
T1 - Erfarenheter av luft-vatten värmepump med djurvärme som energikälla : uppvärmning av personalutrymme vid Ökna lantbruksskola
AU - Norrfors, Mats
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:137
SN - 91-540-3612-7
ER -

TY - BOOK
A1 - Jelbring, Hans
T1 - Prov med värmepump och sjöförlagd värmeväxlare vid Kungliga slottet
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:43
SN - 91-540-3498-1
ER -

TY - BOOK
A1 - Jensen, Lars
T1 - Uteluft och mark som värmekälla för värmepump i kv Bobinen i Malmö : utvärdering
AU - Wetterstad, Lennart
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:89
SN - 91-540-4430-8
ER -

TY - BOOK
A1 - Johansson, Lars
T1 - Energibehov vid torkning : jämförelse mellan mekanisk värmepump och öppet absorptionssystem för torkskåp inom hushåll : CFD-analys avabsorbator
Y1 - 1998
CY - Luleå

T3 - Examensarbete / Luleå tekniska universitet, Civilingenjörsprogrammet, 1402-1617 ; 1998:020
ER -

TY - BOOK
A1 - Johansson, Mats
T1 - Datorprogram för studium av temperatur- och frysningsförlopp i mark kring kylrör till värmepump
AU - Westman, Kristina
Y1 - 1979
CY - Lund
T3 - Rapport BKL, 0281-6318 ; 1979:9
ER -

TY - BOOK
A1 - Kahsay, Samson
T1 - Processintegration av en kompressions-/absorptions-värmepump (KAVP) vid Värö bruk
AU - Wikensten, Bo
Y1 - 1999
N1 - Examensarbete
T3 - Examensarbete / Chalmers tekniska högskola, Institutionen för värmeteknik och maskinlära, 99-1473418-9 ; 99:4
ER -

TY - BOOK
A1 - Karlsson, Hanna
T1 - Utredning om ökade fjärrleveranser från Shell Raffinaderi AB genom installation av värmepump
Y1 - 2002
CY - Göteborg
PB - Chalmers tekniska högsk.
T3 - Examensarbete (Institutionen för kemiteknik och miljövetenskap/Värmeteknik och maskinlära, Chalmers tekniska högskola); 02:12
ER -

TY - BOOK
A1 - Lagerkvist, Knut-Olof
T1 - Mätning och utvärdering av värmepump med avloppsvatten som värmekälla i Falun
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:42
SN - 91-540-4726-9
ER -

TY - BOOK
A1 - Landé, Johan
T1 - Praktiska prov med olika köldmedier [i] brine-vatten värmepump
Y1 - 1992
CY - Stockholm
PB - Institutionen för mekanisk värmeteori och kylteknik, KTH
T3 - Trita-REFR, 1102-0245 ; 92:8
ER -

TY - BOOK
A1 - Ljungqvist, Jarl
T1 - Värmepump med frånluft som värmekälla för tappvatten i befintligt flerbostadshus i Göteborg
AU - Ohlsson, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:126
SN - 91-540-3818-9
ER -

TY - BOOK
A1 - Mogensen, Palne
T1 - Fullskaleförsök med berg som värmekälla för värmepump i Järfälla : mätning och utvärdering
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:123
SN - 91-540-4470-7
ER -

TY - BOOK
A1 - Mårtensson, Alf-Göran Mårtensson
T1 - Redovisning av försök beträffande värmeåtervinning från djurstall med värmepump placerad i bostadshuset samt nedsmutsningens inverkan på lamellbatteriers funktion.
Y1 - 1983
CY - Lund
T3 - Intern stencil / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT) och konsulentavdelningen/teknik, 99-0412863-4 ; 6
ER -

TY - BOOK
A1 - Nilsson, Christer
T1 - Torkning av spannmål med hjälp av värmepump och balktork
Y1 - 1990
CY - Lund
PB - LBT
T3 - Sammanställt / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 1101-5845 ; 3
ER -

TY - BOOK
A1 - Nordström, Hans
T1 - Mätning, modellbygge, simulering och reglering tillämpat på en anläggning med värmepump
AU - Svensson, Lennart
Y1 - 1977
CY - Lund
ER -

TY - BOOK
A1 - Nordström, Hans

T1 - Mätning, modellbygge, simulering och reglering tillämpat på en anläggning med värmepump
AU - Svensson, Lennart
Y1 - 1977
CY - Lund
T3 - Rapport BKL, 0281-6318 ; 1977:9
ER -

TY - BOOK
A1 - Norén, Bjarne
T1 - Studie av kombinationen gasturbin med värmeväxlare och värmepump
Y1 - 1981
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5106
ER -

TY - CHAP
A1 - Norgren, Mårten
T1 - Värmepump i Bodens reningsverk betalar sig på tre år
Y1 - 1982
JF - Kommun-aktuellt
VL - 1982: nr 8, s. 15
PB - Kommun-aktuellt
SN - 0347-5484
KW - Uppvärmning (byggnader)-- Sverige -- Norrbotten
KW - Värmepumpar-- Sverige -- Norrbotten
KW - Boden
ER -

TY - BOOK
A1 - Nyman, Bertil
T1 - Värmebalans i småhus med kombinerat uppvärmningssystem - solfångare, värmepump, värmeväxlare samt tidsstyrd värme och ventilation : [licentiatavhandling]
Y1 - 1982
CY - Stockholm
PB - KTH
ER -

TY - BOOK
T1 - Oskadliggörande av läckande CFC-köldmedier från värmepump- och kylanläggningar : förstudie
AU - Jacobson, Sten Olle
AU - Keck, Karl-Erik
AU - Jacobson, Lars
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:71
SN - 91-540-4932-6
ER -

TY - BOOK
A1 - Persson, Sven-Erik
T1 - Värmepump med ismaskin för 43 lägenheter i Sälen : förstudie

Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:49
SN - 91-540-3479-5
ER -

TY - BOOK
A1 - Sjöbris, Karin
T1 - Värmeåtervinnings- och rökgasreningsanläggning : kondensation av rökgaser i kombination med värmepump : förstudie
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:118
SN - 91-540-4836-2
ER -

TY - BOOK
A1 - Stock, Christina
T1 - Kriterier för dimensionering av bergborrande brunnar för energiuttag med värmepump.
Y1 - 1984
T3 - Examensarbete / Institutionen för vattenbyggnad, Kungl. Tekniska högskolan, 99-0535463-8 ; 291
ER -

TY - BOOK
T1 - Stora värmepumpsboken : nyttig kunskap, vägledning och inspiration när du funderar på att köpa värmepump
Y1 - 2009
CY - Arvika
PB - Thermia Värme AB
SN - 91-631-7643-2
ER -

TY - BOOK
A1 - Svensson, Torbjörn
T1 - Värmeåtervinning ur avloppsvatten med värmepump för fjärrvärmeanslutning i Enköping : förstudie
AU - Wahlberg, Herje
AU - Wahlberg, Olof
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:58
SN - 91-540-3913-4
ER -

TY - BOOK
A1 - Svärd, Ann-Christine
T1 - Bestämning av värmefaktorn för en värmepump vid Östra Sjukhuset, Göteborg
Y1 - 1979
N1 - Examensarbete
CY - Göteborg

T3 - Examensarbete / Chalmers tekniska högskola, Avdelningen för installationsteknik, 99-0354504-5 ; 4
ER -

TY - BOOK
A1 - Tengblad, Niklas
T1 - Resultat från prov med liten luft-luft värmepump med propan/gasol som köldmedium
Y1 - 1993
CY - Stockholm
PB - Avdelningen för tillämpad termodynamik och kylteknik, Institutionen för energiteknik, Kungliga Tekniska högskolan
T3 - Trita-REFR, 1102-0245 ; 93:12
ER -

TY - BOOK
A1 - Jagenäs, Arne
T1 - Mätning av ljudnivå från värmepump.
Y1 - 1981
T3 - Rapport / Chalmers tekniska högskola, Avdelningen för byggnadsakustik. S, 0282-3446 ; 81:03
ER -

TY - BOOK
A1 - Jansson, Lennart
T1 - Värmepump med uteluft som värmekälla : projektering av värmepumpsystem för befintligt flerbostadshus i Söderköping
AU - Strindehag, Ove
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:124
SN - 91-540-3814-6
ER -

TY - BOOK
A1 - Jilar, Torbjörn
T1 - Fjärås centrum - solvärmesystem med värmepump : utvärderingsresultat
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:11
SN - 91-540-4692-0
ER -

TY - BOOK
A1 - Jilar, Torbjörn
T1 - Solvärmeteknik i stor skala : Ingelstad - en värmecentral utan värmepump
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:103
SN - 91-540-4171-6
ER -

TY - BOOK

A1 - Johansson, Jan
T1 - Naturvärmesystem för biltestningsanläggning i Slagnäs : ground heat storage sytem for the car test plant at Slagnäs
AU - Josephsson, Börje
Y1 - 1995
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, 0349-6023 ; 1995:139E
ER -

TY - BOOK
A1 - Jonsson, Eric
T1 - Fullskaleprov med solfångare i kombination med ytjordvärmesystem för uppvärmning av småhus : [ett energisnålt alternativ för uppvärmning av småhus] : [beskrivning av ett kombinationssystem] : [utvärdering och rapport]
Y1 - 1983
CY - [Eskilstuna]
PB - [Eskilstuna kommun]
ER -

TY - BOOK
A1 - Jönsson, Sven-Erik
T1 - Värmepump för spillvärme till fjärrvärmenät : teknisk och ekonomisk utredning gällande spillvärme från Domsjö sulfitfabrik till Örnsköldsviks fjärrvärmenät : förstudie
AU - Sandberg, Edvard
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:72
SN - 91-540-3278-4
ER -

TY - BOOK
A1 - Kangro, Andrus
T1 - Värmeåtervinning med värmepump i djurstallar : mätning och utvärdering av anläggning i Klovarp
Y1 - 1982
CY - Lund
T3 - Specialmeddelande / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0593 ; 115
SN - 91-576-1167-X
ER -

TY - BOOK
A1 - Larsen, Johan
T1 - Uppvärmningsalternativ med värmepump för industri
AU - Usinovski, Dime
Y1 - 2003
CY - Göteborg
PB - Chalmers Lindholmen högsk.
ER -

TY - BOOK
A1 - Lemmeke, Leif

T1 - Renvatten som värmekälla till värmepump : en studie i Åkarp
AU - Persson, Ulf
AU - Luterkort, Staffan
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:75
SN - 91-540-4125-2
ER -

TY - BOOK
A1 - Lindahl, Anders
T1 - Värmepump för utvinning av havsvattenvärme : förstudie beträffande havsvattenvärmeväxlare
AU - Stenström, Börje
AU - Öst, Staffan
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:123
SN - 91-540-3337-3
ER -

TY - BOOK
A1 - Nilsson, Emma
T1 - Småhus med solceller och värmepump : utvärdering av Ekosolkonceptet
Y1 - 2008
CY - Göteborg
PB - Chalmers tekniska högskola
ER -

TY - BOOK
A1 - Stråhlin, Fredrik
T1 - Optimering av värmeoproduktion : en studie vid Birka Energi AB
Y1 - 1999
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Luleå tekniska universitet, Civilingenjörsprogrammet, 1402-1617 ; 1999:281
UR - http://epubl.luth.se/1402-1617/1999/281
ER -

TY - BOOK
A1 - Westerlund, Lars
T1 - Öppna absorptionssystem : absorbatorers egenskaper
Y1 - 1989
N1 - Examensarbete
CY - Luleå
T3 - Teknisk rapport / Tekniska högskolan i Luleå, 0349-3571 ; 1989:17T
ER -

TY - BOOK
T1 - Årsmötet 1975. D. 2, Krångede och Gammelänge kraftstationer - ökning av turbineffekten

Y1 - 1975
CY - Stockholm
PB - Svenska kraftverksfören.
ER -

TY - BOOK
A1 - Collin, Mats
T1 - Ångmotordriven värmepumpsprocess för bostadsbebyggelse : förstudie och jämförelse med dieselmotordriven värmepump
AU - Palmgren, Mikael
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:133
SN - 91-540-3604-6
ER -

TY - BOOK
A1 - Dahlberg, Marcus
T1 - Marknadspotential för en kemisk kyl/värmepump
Y1 - 2002
CY - Stockholm
T3 - Trita-IEO. EX, 1403-7777 ; 2002:26
ER -

TY - BOOK
A1 - Franck, Per-Åke
T1 - Värmepump med vertikalt jordvärmesystem och vindkonvektorer
AU - Modin, Björn
Y1 - 1981
CY - Göteborg
T3 - Rapport / Jordvärmegruppen, Chalmers tekniska högskola, 0349-9723 ; 2
ER -

TY - BOOK
A1 - Luterkort, Staffan
T1 - Gasmotordriven värmepump för fjärrvärmenätet i Oxie : förstudie
AU - Lemmeke, Leif
AU - Torisson, Tord
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:128
SN - 91-540-4237-2
ER -

TY - BOOK
A1 - Mårtensson, Alf-Göran
T1 - Energibesparing med värmepump - stalluft som värmekälla
Y1 - 1978
KW - Värmepumpar
KW - Stallbyggnader
KW - Energisparande
CY - Uppsala
PB - Sveriges lantbruksuniv.

SN - 91-7088-931-7
ER -

TY - BOOK
A1 - Saue, Olav Jarle S.
T1 - Solinfångande bassäng, pulserande värmelager och värmepump : diskussion av driftstrategier
Y1 - 1982
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5118
ER -

TY - BOOK
T1 - Skarpnäcks trädgårdsstadsförening 80 år : från kakelugn till värmepump : 1927-2007
AU - Barkman, Clas
KW - Skarpnäcks trädgårdsstadsförening
KW - Stockholm-- historia -- Skarpnäck -- 1900-talet -- sekelskiftet 2000
CY - Johannneshov
PB - Skarpnäcks trädgårdsstadsförening
ER -

TY - BOOK
A1 - Sönsteröd, Gunnar
T1 - Värmepump och oljepanna i bivalent drift : fältmätningar på typfall
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:3
SN - 91-540-4846-X
ER -

TY - BOOK
A1 - Agerstrand, Torgny
T1 - Projekt Bergvärme : energiuttag med värmepump ur bergborrade hål : förstudie
AU - Ericsson, Lars O.
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:172
SN - 91-540-3418-3
ER -

TY - BOOK
A1 - Backman, Lennart
T1 - Spillvärme från mejeri till fjärrvärmenät via värmepump : alternativprojektering i Östersund
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:46
SN - 91-540-3228-8
ER -

TY - BOOK
A1 - Berntsson, John
T1 - Konstruktion av slavenhet för styrning av värmepump
AU - Bäckhage, Eric
Y1 - 2006
N1 - Examensarbete
CY - Göteborg
PB - Chalmers tekniska högskola
ER -

TY - BOOK
A1 - Bärring, Mats
T1 - Återvinning av värmeförluster i fjärrvärmekulvertar med värmepump : kulverttyper och systemoptimering
AU - Rosenberg, Mats
Y1 - 1986
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1986:53
SN - 91-540-4571-1
ER -

TY - BOOK
A1 - Cedheim, Lars
T1 - Nedbrytning av glykoler i värmepump och solvärmesystem : förstudie
Y1 - 1986
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1986:4
SN - 91-540-4506-1
ER -

TY - BOOK
A1 - Cedheim, Lars
T1 - Nedbrytning av glykoler i värmepump och solvärmesystem : laboratorieundersökning
AU - Blume, Ulla
AU - Lundgren, Björn
Y1 - 1988
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:107
SN - 91-540-4980-6
ER -

TY - BOOK
A1 - Elmroth, Arne
T1 - SPARSAM : fem energisnåla småhus med glasveranda och värmepump
AU - Granberg, Gunnar
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:47
SN - 91-540-4736-6
ER -

TY - BOOK
A1 - Grafström, Hans
T1 - Värmepump vid friluftsbad med havsvatten som värmekälla : projektering, uppföljning och utvärdering
AU - Lagergren, Staffan
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:45
SN - 91-540-3692-5
ER -

TY - BOOK
A1 - Hammerbo, Erik
T1 - En femte värmepump vid Rya värmepumpverk, Göteborg
AU - Johansson, Björn
Y1 - 1994
N1 - Examensarbete
T3 - Examensarbete / Chalmers tekniska högskola, Institutionen för installationsteknik, 1400-9552 ; 104
ER -

TY - BOOK
A1 - Hultmark, Göran
T1 - Grundvattenvärme med värmepump vid strandbadet i Karlskoga : utvärdering
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:141
SN - 91-540-4479-0
ER -

TY - BOOK
A1 - Jacobsson, Carl
T1 - Värmeutvinning ur havsvik med värmepump för villauppvärmning
AU - Jönsson, Sven
Y1 - 1981
N1 - Examensarbete
CY - Göteborg
T3 - Examensarbete / Institutionen för vattenbyggnad, Chalmers tekniska högskola, 99-0211330-3 ; 1981:2
ER -

TY - BOOK
A1 - Johansson, Anders
T1 - Förvärmning av uteluft till värmepump med markkollektor : datormodell
Y1 - 1990
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1990:68
SN - 91-540-5246-7
ER -

TY - BOOK
A1 - Lindvall, Anna

T1 - Strategier vid val av värmepump till småhus : en energijämförelse mellan varvtalsreglerade och on-/offreglerade värmepumpar
Y1 - 2007
CY - Helsingborg
PB - LTH Ingenjörshögskolan vid Campus Helsingborg, Lunds universitet
T3 - Examensarbete (LTH Ingenjörshögskolan vid Campus Helsingborg), 1651-2197 ; THID-07/5069
ER -

TY - BOOK
A1 - Margen, Peter H.
T1 - Optimering av solvärmecentraler med respektive utan värmepump : VVS 78
Y1 - 1978
CY - Nyköping
PB - Studsvik energiteknik
ER -

TY - BOOK
A1 - Olsson, Ulf
T1 - Dieselmotordriven värmepump för gruppbebyggelse och större fastigheter : förstudie i Fisksätra
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:45
SN - 91-540-3226-1
ER -

TY - BOOK
A1 - Rosengren, Bengt
T1 - Förvärmning med värmepump av dricksvatten till mjölkkor : Preheating of drinking water to dairy cows by means of heat pump
Y1 - 1991
CY - Lund
T3 - Specialmeddelande / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0593 ; 179
ER -

TY - BOOK
A1 - Schroeder, Kjell
T1 - Sjövärmeprojekt Motala Väster : utvärdering och analys av värmekälla och värmepump
AU - Svensson, Torbjörn
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:58
SN - 91-540-5019-7
ER -

TY - BOOK
T1 - Solenergi och värmepump värmer kontor och industrilokaler.

Y1 - 1983
T3 - SV-rapport, 99-0308735-7 ; 1983:1
ER -

TY - BOOK
A1 - Stillesjö, Staffan
T1 - Värmepump eller direktel för lågenergihus i grupp : förstudie
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:187
SN - 91-540-4281-X
ER -

TY - BOOK
T1 - Termokemisk lagring i kombination med dieseldriven värmepump : fullskaleprov i Televerkets arbetslokal i Jakobsberg
AU - Bakken, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:102
SN - 91-540-3782-4
ER -

TY - BOOK
T1 - Värmeåtervinning med värmepump från Scaniarinken i Södertälje
AU - Hagstedt, Bertil
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:123
SN - 91-540-3812-X
ER -

TY - BOOK
A1 - Wilén, Peter
T1 - Grundvatten som värmekälla för husuppvärmning med värmepump : litteraturstudie, system och ekonomi
Y1 - 1981
CY - Göteborg
T3 - Rapport / Jordvärmegruppen, Chalmers tekniska högskola, 0349-9723 ; 1
ER -

TY - BOOK
T1 - Energi ur grundvatten : inventering av stora grundvattenmagasin för energiutvinning med värmepump. Kartbilaga till slutrapport : länsvis redovisning av tätorternas och tätortsregionernas energiresurser i stora grundvattenmagasin
Y1 - 1980
ER -

TY - BOOK

A1 - Jacobson, Lars
T1 - Jordvärmesystem med värmepump i befintlig och ny bebyggelse : förprojektering av sju objekt i Västsverige
Y1 - 1982
KW - Geotermisk energi
KW - Värmepumpar
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:112
SN - 91-540-3793-X
ER -

TY - BOOK
A1 - Larsson, Lars I.
T1 - Kraftvärmeverk och värmepump för utnyttjning av rökgasens kondensationsvärme
Y1 - 1980
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5101
ER -

TY - BOOK
A1 - Löving, Ronald
T1 - Elmotordriven värmepump för luft/vatten i Backatorp, Göteborg
AU - Bäckström, Bernt
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:87
SN - 91-540-3988-6
ER -

TY - BOOK
A1 - Möller, Kjell
T1 - Gastät lagring och torkning av spannmål med värmepump : Airtight storage and drying of grain using a heat pump
AU - Nilsson, Christer
Y1 - 1988
CY - Lund
PB - LBT
SN - 91-576-3577-3
ER -

TY - BOOK
A1 - Andersson, Frank
T1 - Fjärås centrum : projektering av solvärmesystem med värmepump
AU - Länsberg, Mats
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:8
SN - 91-540-3170-2
ER -

TY - BOOK

A1 - Areskoug, Mats
T1 - Miljöfysik
Y1 - 1997
N1 - Prel.uppl.
KW - Miljöfysik
CY - Malmö
PB - Lärarhögskolan
ER -

TY - BOOK
A1 - Areskoug, Mats
T1 - Miljöfysik
Y1 - 1998
CY - Malmö
PB - Malmö högskola, Lärarutbildningen
ER -

TY - BOOK
A1 - Axby, Fredrik
T1 - Rökgaskondensering med värmepump = Flue gas condensing with heat pump
AU - Pettersson, Camilla
Y1 - 2004
KW - Fjärrvärme
KW - Värmeteknik
CY - Stockholm
PB - Värmeforsk
T3 - Värmeforsk, 0282-3772 ; 891
ER -

TY - BOOK
A1 - Backman, Anders
T1 - Värmeåtervinning ur avloppsvatten via värmepump : förstudie i Gävle
AU - Hallenberg, Jonas
AU - Bustad, Tohmmy
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:44
SN - 91-540-3922-3
ER -

TY - BOOK
A1 - Berendson, Jaak
T1 - Värmepump Motala Väster : korrosionsuppföljning
Y1 - 1991
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1991:50
SN - 91-540-5374-9
ER -

TY - BOOK
A1 - Bergström, Ulf
T1 - Direktverkande värmepump för småhus : utvärdering av funktion och energibesparing genom fältmätningar
AU - Fehrm, Mats

AU - Mattsson, Per Olof
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:18
SN - 91-540-3641-0
ER -

TY - BOOK
A1 - Bovin, Jan
T1 - Stirlingmotordriven värmepump : förstudie avseende planerat experimentbygge i Kågeröd
Y1 - 1990
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1990:25
SN - 91-540-5180-0
ER -

TY - BOOK
A1 - Bäckström, Bernt
T1 - Värmepump med energistapel : förprojektering av befintligt mindre flerbostadshus i Göteborg
AU - Hallén, Tomas
AU - Samuelsson, Torbjörn
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:140
SN - 91-540-3618-6
ER -

TY - BOOK
A1 - Carlsson, Jan
T1 - Värmepump typ Stirling : examensarbete
AU - Åkersten, Lars-Erik
Y1 - 1981
CY - Göteborg
T3 - Rapport / Institutionen för energiteknik, Ångteknik med reaktorteknologi, Chalmers tekniska högskola. T, 0280-9249 ; 93
ER -

TY - BOOK
A1 - Edman, Jan
T1 - Isbana som värmekälla för värmepump : förstudie i Hörby
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:157
SN - 91-540-3389-6
ER -

TY - BOOK
T1 - Energiutvinning ur ytvatten via värmepump : förprojektering i Borlänge
AU - Backman, Anders
Y1 - 1980

CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:109
SN - 91-540-3326-8
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Louddens värmepump : utvärdering av avloppsvatten/fjärrvärme-värmepump
AU - Solin, Lars
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:16
SN - 91-540-4679-3
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Uppsala värmepump : utvärdering av avloppsvatten/fjärrvärme, värmepump
AU - Solin, Lars
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:97
SN - 91-540-4960-1
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Visby värmepump : utvärdering
AU - Nilsson, Madelaine
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:29
SN - 91-540-5020-0
ER -

TY - BOOK
A1 - Fransson, Kenneth
T1 - Energiåtervinning i animalieproduktionen med värmepump : värmekälla mjölk och stalluft
Y1 - 1982
N1 - Examensarbete
CY - Uppsala
T3 - Rapport / Sveriges lantbruksuniversitet, Institutionen för arbetsmetodik och teknik, 0347-9773 ; 77
SN - 91-576-1314-1
ER -

TY - BOOK
T1 - Från kakelugn till värmepump : en presentation av värmepumpinstallationerna i Forsmarks bruk
Y1 - 1987
CY - Vällingby

PB - Vattenfall
ER -

TY - BOOK
T1 - Från S- till F-system med värmepump : ombyggnad av självdragssystem till mekanisk frånluftsventilation med värmepump : ett praktikfall
AU - Erikson, Bengt E.
Y1 - 1985
CY - Gävle
PB - Statens inst. för byggnadsforskning
T3 - Meddelande / Statens institut för byggnadsforskning, 0347-4348 ; 85:1
SN - 91-540-9219-1
ER -

TY - BOOK
A1 - Grafström, Hans
T1 - Värmepump för uppvärmning av friluftsbad : förstudie i Sölvesborg
AU - Lagergren, Staffan
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:44
SN - 91-540-3224-5
ER -

TY - BOOK
A1 - Gustafsson, Gösta
T1 - Avfuktning med värmepump i växthus : Dehumidification with heat pump in greenhouses
AU - Nimmermark, Sven
Y1 - 1991
CY - Lund
T3 - Rapport / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0259 ; 74
ER -

TY - BOOK
T1 - Gårdsvärme : ved, halm, torv, värmepump
AU - Liedholm, Hans
AU - Leuchovius, Torbjörn
Y1 - 1983
CY - Stockholm
PB - LT
SN - 91-36-02108-3
ER -

TY - BOOK
A1 - Hallén, Tomas
T1 - Västra klinikerna i Jönköping : grundvatten som värmekälla för värmepump
AU - Edberg, Bo
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:88

SN - 91-540-4428-6
ER -

TY - BOOK
A1 - Hallenberg, Jonas
T1 - Avloppsvatten som värmekälla till värmepump : utvärdering av installation i Boden
AU - Wahlberg, Herje
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:125
SN - 91-540-3816-2
ER -

TY - BOOK
A1 - Hallenberg, Jonas
T1 - Sjövatten som värmekälla till värmepump : erfarenheter från uppvärmning av två småhus i Falun
AU - Norbäck, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:46
SN - 91-540-3694-1
ER -

TY - BOOK
A1 - Henningsson, Lars
T1 - Värmepump : installationsförslag och marknadsundersökning
Y1 - 1982
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, 0349-6023 ; 1982:067E
ER -

TY - BOOK
A1 - Henriksson, Jan
T1 - Bygg din egen värmepump
AU - Nilsson, Hans
Y1 - 1983
CY - Ljungby
PB - Colego
ER -

TY - BOOK
A1 - Hill, Anders
T1 - Ammoniak som köldmedie i värmepump
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:103
SN - 91-540-5120-7
ER -

TY - BOOK
A1 - Hirschfeldt, Claes-Henrik

T1 - Utvärdering av en öppen kemisk värmepump
AU - Olofsson, Richard
Y1 - 1995
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå,
Ingenjörslinjen 80 poäng, 1102-4070 ; 1995:44I
ER -

TY - BOOK
A1 - Härröd, Peter
T1 - Energibesparing med värmepump i banddiskmaskin
AU - Jorlöv, Andreas
Y1 - 2000
N1 - Examensarbete
CY - Göteborg
PB - Chalmers Lindholmen högsk., Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Högström, Carina
T1 - Värmepump med propan som köldmedium
Y1 - 1999
N1 - Lic.-avh. Stockholm : Tekn. högsk.
CY - Stockholm
T3 - Trita-REFR, 1102-0245 ; 99:26
ER -

TY - BOOK
A1 - Karlsson, Erik
T1 - Värmepump 1, Findus i Bjuv
AU - Taberman, Marie
Y1 - 2003
CY - Göteborg
PB - Chalmers Lindholmen högsk., Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Lemmeke, Leif
T1 - Storskalig värmeförsörjning med värmepump :
principförslag med kombinerat utnyttjande av yt- och
grundvatten som värmekälla
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:126
SN - 91-540-3597-X
ER -

TY - BOOK
A1 - Lindelöf, Jan
T1 - Effektiv effektivare värmepump : idéhandbok om
värmepumpar
AU - Sandström, Bengt
AU - Sandquist, Hans
Y1 - 1995
CY - Stockholm
PB - Stockholm energi
SN - 91-630-3394-1

ER -

TY - BOOK
A1 - Lundén, Roger
T1 - Värmepump med effektutjämningssystem : en studie av ett system där värmepump kombineras med värmeackumulering i jord
Y1 - 1977
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningen, 0346-5616 ; 1977:85
SN - 91-540-2784-5
ER -

TY - BOOK
A1 - Lundqvist, Per G.
T1 - Stirling värmepump : konstruktion, byggande och prestandaprovning : slutrapport
Y1 - 1996
CY - Stockholm
PB - KTH, Inst för energiteknik
T3 - Trita-REFR, 1102-0245 ; 96:20
ER -

TY - BOOK
A1 - Mårtensson, Hans
T1 - Värmepump i villan
Y1 - 2007
KW - Värmepumpar
CY - Västerås
PB - Ica
SN - 978-91-534-2873-2 (inb.)
ER -

TY - BOOK
T1 - Oljebrännare vs värmepump
AU - Andersson, Ola
Y1 - 1997
N1 - Examensarbete
CY - Göteborg
PB - Ingenjörs- och sjöbefälsskolan vid Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Persson, Lars
T1 - Pelletseldning eller värmepump? : historik, nutid, framtid
Y1 - 2005
CY - Delsbo
PB - GDE-Net
ER -

TY - BOOK
T1 - Processmodell för värmepump
AU - Abrahamsson, Lars
Y1 - 1986
CY - Luleå

T3 - Teknisk rapport / Tekniska högskolan i Luleå, 0349-3571 ; 1986:012T
ER -

TY - BOOK
A1 - Ransmark, Sven-Erik
T1 - Värmepump med Lorenzprocess
Y1 - 1985
T3 - Institutionen för värme- och kraftteknik, Tekniska högskolan i Lund, 0282-1990 ; 3128
ER -

TY - BOOK
A1 - Renntun, Mats
T1 - Värmepump och värmelagring
Y1 - 1978
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5078
ER -

TY - BOOK
A1 - Rönnelid, Mats
T1 - Solfångare och värmepump : utvärdering av ett värmesystem i Uppsala
AU - Tepe, Rainer
Y1 - 2004
KW - Solfångare
KW - Solenergi
KW - Geotermisk energi
KW - Uppvärmning (byggnader)
CY - Borlänge
PB - EKOS, Högskolan i Dalarna
ER -

TY - BOOK
A1 - Saros, Georg
T1 - Att välja värmepump : handledning för att välja värmepump och att räkna på ekonomin
AU - Pettersson, Anders
Y1 - 1998
CY - Stockholm
PB - Konsumentverket :
SN - 91-7398-636-4
ER -

TY - BOOK
A1 - Tepe, Rainer
T1 - Solfångare och värmepump : marknadsöversikt och resultat av en preliminär simuleringsstudie av värmesystem med solfångare och bergvärmepump
AU - Rönnelid, Mats
Y1 - 2002
KW - Solfångare
KW - Solenergi
KW - Solenergi
CY - Borlänge
PB - SERC, Högskolan i Dalarna

T3 - Centrum för solenergiforskning, Högskolan Dalarna, 1401-7555 ; 75
ER -

TY - BOOK
A1 - Tepe, Rainer
T1 - Solfångare och värmepump : marknadsöversikt och resultat av en preliminär simuleringsstudie av värmesystem med solfångare och bergvärmepump
AU - Rönnelid, Mats
Y1 - 2002
KW - Solfångare
KW - Solenergi
KW - Geotermisk energi
KW - Uppvärmning (byggnader)
CY - Borlänge
PB - EKOS, Högskolan i Dalarna
T3 - EKOS publikation, 1650-1497 ; 2002:2
ER -

TY - BOOK
T1 - Värmepump : ett tekniskt system i utveckling
Y1 - 1988
CY - Stockholm
PB - Styr. för teknisk utveckling
T3 - STU-information, 0347-8645 ; 671
SN - 91-7850-243-8
ER -

TY - BOOK
T1 - Värmepump för avloppsvatten i Sala.
Y1 - 1982
CY - Vällingby
T3 - SV-rapport, 99-0308735-7 ; 1982:3
ER -

TY - BOOK
T1 - Värmepump för avloppsvatten i Sundsvall.
Y1 - 1982
CY - Vällingby
T3 - SV-rapport, 99-0308735-7 ; 1982:4
ER -

TY - CHAP
A1 - Waldenstad, Bengt
T1 - Testas i Härnösand : Stirlingmotor driver värmepump
AU - Österberg, Roine
Y1 - 1985
JF - VVS & energi
VL - 1985:5, s. 35-36
PB - VVS & energi
SN - 0280-9524
KW - Stirlingmotorer-- Sverige -- Ångermanland
KW - Värmepumpar-- Sverige -- Ångermanland
KW - Härnösand
ER -

TY - BOOK

A1 - Wirén, Mikael
T1 - Avancerad värmepump för enbostadshus :
marknadsförutsättningar i Sverige, Norge, USA och Kanada
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:48
SN - 91-540-4891-5
ER -

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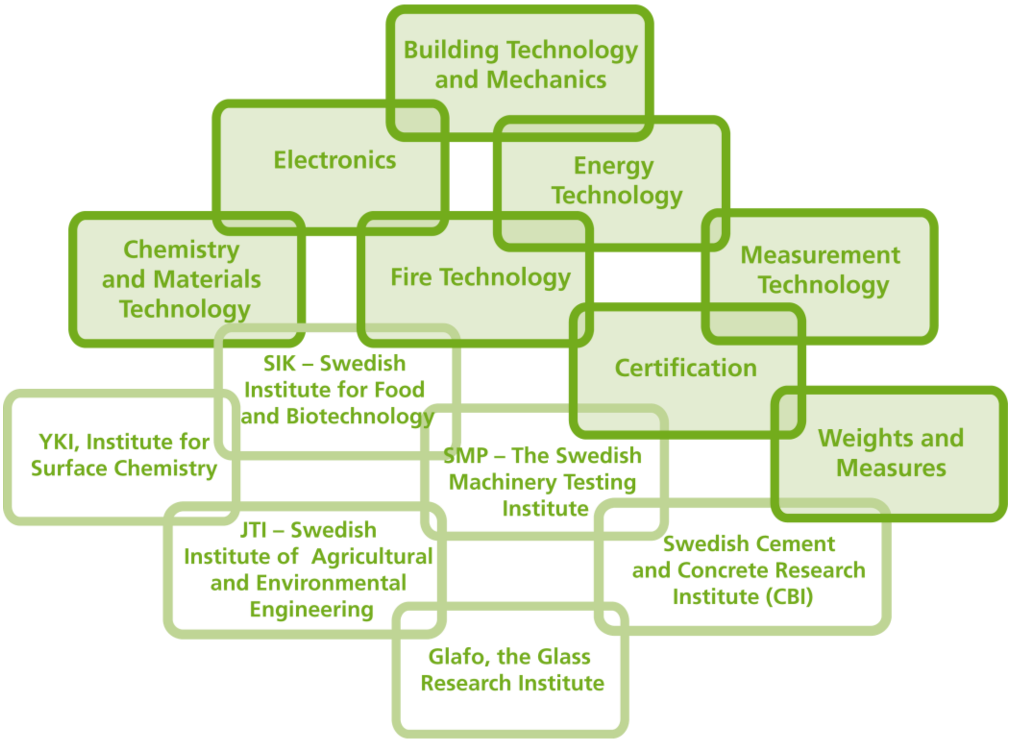

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www.sp.se

Energy Technology
SP Report 2010:49
ISBN 978-91-86319-86-1
ISSN 0284-5172

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10.7 Appendix VIII – National report Netherlands

Appendix E

Method for determination of the efficiency of air to water heat pumps for heating purposes

E.1 Principle

This appendix describes a method for determination of the efficiency of air to water heat pumps. The efficiency and energy fraction of hybrid systems, heat pumps with external supplementary heating, is also described.

In E.2 the amount of energy delivered by preferential heat generator is determined as well as the efficiency of the non-preferential generator. E.3 gives a method for determining the efficiency of an air to water heat pump. E.4 describes the measurements needed for the use of this appendix.

The efficiency and the energy fraction of air to water heat pumps depends on several building specific variables. Therefore this method can be applied in two ways:

1. For specific buildings the efficiency and the energy fraction can be determined
2. For a range of building specific variables the efficiency and the energy fraction can be determined. In this case these variables should be classified according to the rules in appendix J.

ANNOTATION: Examples of building specific variables are the used area(E.19) and heating demand(E.1).

E.2 Determination of the energy fraction for heating

In case only one generator is used in heating system *si,* the value of the energy fraction is:

$$F_{H;gen;si,gpref} = 1$$

For an air to water heat pump the non-integrated supplementary heating for the fraction of the air to water heat pump of the total heating supply is determined as follows:

$$F_{H;gen;si,gpref} = \frac{Q_{H;hp;pr;an}}{\sum_{mi} Q_{H;dis;nren;si,mi}} \qquad \text{(E.1)}$$

In which:

F;H;gen;*si,gpref* the fraction of energy delivered the preferential generator or generators(*gpref*);

Q;H;hp;pr;an the total amount of energy delivered by the air to water heat pump per year determined according to E.3.2, in MJ;

Q;H;dis;nren;*si,mi* the amount of energy used for heating, in month *mi*, by the non-preferential generators(nren) delivered to the distribution part of the system *si*, determined according to 14.1.2, in MJ.

E.3 Determination of the generation efficiency for heating

E.3.1 Yearly value for the generation efficiency

Determine the year round efficiency for heating system *si* according:

$$\eta_{H;gen;hp;si} = \frac{Q_{H;hp;pr;an}}{Q_{H;hp;us;an;el}} \quad \text{(E.2)}$$

In which:

η;H;gen;hp;*si* the dimensionless efficiency of the air to water heat pump for heating system *si*;

Q;H;hp;pr;an the total amount of energy delivered by the air to water heat pump per year determined according to E.3.2, in MJ;

Q;H;hp;us;an;el the total amount of electricity, including electronics and fan, used by the air to water heat pump per year determined according to E.3.3, in MJ;

E.3.2 Yearly value for the heat delivered by the air to water heat pump

Determine the total amount of heat delivered by the air to water heat pump per year, Q;H;hp;pr;an, according:

$$Q_{H;hp;pr;an} = 3{,}6 \times \sum_{\theta_i}\left(t_{H;hp;an;\theta i} \times P_{H;hp;pr;\theta i}\right) \quad \text{(E.3)}$$

In which:

θi is the outdoor temperature θoutdoor according table E.1 in °C;

t;H;hp;an;θi is the time the outdoor temperature is equal to θ outdoor determined according table E.1, per year, in hours;

P;H;hp;pr;θi is the power delivered by the air to water heat pump at temperature θ outdoor according to E.3.4 in kW.

E.3.3 Yearly value for the amount of electricity used by the air to water heat pump

Determine the amount of electric energy consumed by heating by the air to water heat pump, Q;H;hp;us;an;el, according:

$$Q_{H;hp;us;an;el} = 3{,}6 \times \sum_{\theta\mathrm{i}} t_{H;hp;an;\theta\mathrm{i}} \times \frac{P_{H;hp;pr;\theta\mathrm{i}}}{COP_{H;hp;\theta}} \quad \text{(E.4)}$$

In which:

θi is the outdoor temperature θ outdoor according table E.1 in °C;

COP;H;hp;θi is the COP of the air to water heat pump for heating use at temperature θ outdoor according E.3.7.

E.3.4 Values for power delivered by the air to water heat pump as a function of the outdoor temperature

Determine the amount of power delivered by the air to water heat pump, $P_{H;hp;pr;\theta i}$, according:

$$P_{H;hp;pr;\theta\mathrm{i}} = f_{H;hp;pr;\theta\mathrm{i}} \times Min\left(P_{H;hp;max;\theta\mathrm{i}}; P_{H;\theta\mathrm{i}}\right) \qquad \text{(E.5)}$$

In which:

$f_{H;hp;off;\theta i}$ is the dimensionless factor for switching off the air to water heat pump according E.3.6;

$P_{H;hp;max;\theta i}$ is the maximum thermal power delivered by the air to water heat pump at a given outdoor temperature according E.3.15 in kW;

Min(..;..) is the smallest of the two values.

E.3.5 Net heating power for the zones attached to the air to water heat pump

Determine the net heating power of the relevant zones for the air to water heat pump, $P_{H;\theta i}$, according:

$$P_{H;\theta\mathrm{i}} = \frac{\sum_{mi} Q_{H;dis;nren;si,mi} \times f_{Q;H;\theta\mathrm{i}}}{1000} \qquad \text{(E.6)}$$

In which:

$Q_{H;dis;nren;si,mi}$ is the amount of energy used for heating, in month *mi*, delivered by non-durable generators(nren) for the distribution part of the system *si*, according 14.1.2 in MJ;

$f_{Q;H;\theta i}$ is the dimensionless factor for the translation of the yearly average energy consumption to power per condition, according table E.1.

Table E.1 – Time and heat factor as a function of the outdoor temperature

θ_{buiten}	$t_{H;hp;an;\theta i}$	$f_{Q;H;\theta i}$			
°C	h	WN	WB	UN	UB
16	189	0,0000	0,0000	0,0000	0,0000
15	257	0,0000	0,0053	0,0000	0,0000
14	286	0,0000	0,0106	0,0000	0,0000
13	291	0,0000	0,0160	0,0000	0,0068
12	391	0,0079	0,0213	0,0000	0,0137
11	430	0,0158	0,0266	0,0000	0,0205
10	434	0,0237	0,0319	0,0000	0,0274
9	478	0,0316	0,0372	0,0132	0,0342
8	497	0,0395	0,0425	0,0264	0,0411
7	460	0,0474	0,0479	0,0396	0,0479
6	397	0,0553	0,0532	0,0529	0,0548
5	454	0,0632	0,0585	0,0661	0,0616
4	321	0,0711	0,0638	0,0793	0,0685
3	302	0,0790	0,0691	0,0925	0,0753
2	199	0,0869	0,0745	0,1057	0,0821
1	216	0,0948	0,0798	0,1189	0,0890
0	145	0,1027	0,0851	0,1321	0,0958
–1	109	0,1106	0,0904	0,1453	0,1027
–2	96	0,1185	0,0957	0,1586	0,1095
–3	27	0,1264	0,1011	0,1718	0,1164
–4	48	0,1343	0,1064	0,1850	0,1232
–5	40	0,1422	0,1117	0,1982	0,1301
–6	20	0,1501	0,1170	0,2114	0,1369
–7	25	0,1580	0,1223	0,2246	0,1438
–8	10	0,1659	0,1276	0,2378	0,1506
–9	12	0,1738	0,1330	0,2510	0,1574
–10	8	0,1817	0,1383	0,2643	0,1643

ANNOTATION The heat factor depends on the type of building. New versus old and houses versus utility buildings.

E.3.6 Switching off the air to water heat pump

For air to water heat pumps using only ventilation air as a source:

$$f_{H;hp;off;\theta i} = 1$$

For all other air to water heat pumps the factor for switching off the heat pump depends of the mechanism that is used. The switched off factor f;H;hp;off;θi corresponding to the powering off criterion of the heat pump must be used.

Table E.2 – Power off criteria

Powering off criterion	**f;H;hp;off;θi**
No criterion	f;H;hp;off;θi = 1
Below entry temperature evaporator θ;H;hp;off	f;H;hp;off;θi = 1, θ;evap;in;θi >= θ;H;hp;off f;H;hp;off;θi = 0, θ;evap;in;θi < θ;H;hp;off
Below entry temperature evaporator θ;H;hp;off(combi-device)	f;H;hp;off;θi = 1, θ;evap;in;θi >= θ;H;hp;off f;H;hp;off;θi = 0,5, θ;evap;in;θi = θ;H;hp;off f;H;hp;off;θi = 0, θ;evap;in;θi < θ;H;hp;off
Above a certain supply temperature(when using a HT-delivery system	f;H;hp;off;θi = 1, θ;evap;in;θi >= θ;H;hp;supply;off f;H;hp;off;θi = 0, θ;evap;in;θi < θ;H;hp;supply;off
In which: f;H;hp;off;θi is the dimensionless factor for switching off the air to water heat pump; θ;eval;in; θi is the entry temperature for the evaporator according E.3.10 in °C; θ;H;hp;off is the temperature at which, if the entry temperature of the evaporator comes below this value, the air to water heat pump disables in °C; θ;cond;out;max;θi is the outlet temperature of the condenser at maximum power according E.3.13 in °C; θ;H;hp;supply;off is the temperature at which, if the supply temperature (outlet temperature of the condenser) comes above this value, the air to water heat pump disables , in °C.	

E.3.7 COP of power switching air to water heat pumps

For an on and off switching air to water heat pump with outdoor air of outdoor air and ventilation air as a source the COP is determined according:

$$COP_{H;hp;\theta i} = c_{max;1} \times \theta_{evap;max;\theta i} + c_{max;2} \times \theta_{cond;max;\theta i} + c_{max;3} \quad \text{(E.7)}$$

For an on and off switching air to water heat pump with ventilation air only the COP is determined according:

$$COP_{H;hp;\theta i} = c_{max;2} \times \theta_{cond;max;\theta i} + c_{max;3} \quad \text{(E.8)}$$

In which:

C;max;i is a dimensionless constant, determined at maximum power, according E.3.18;

θ;evap;max; θi is the average temperature of the air over the evaporator at a temperature θ outdoor, at maximum power, according E.3.9;

θ;cond;max;θi is the average temperature of the water over the condenser at a temperature θ outdoor, at maximum power, according E.3.12 in °C.

E.3.8 COP of modulating air to water heat pumps

De determination of the COP of an air to water heat pump depends on de amount of power that needs to be delivered. This is shown in table E.3.

Table E.3 – Forwarding schedule

Delivered power heat pump P;H:hp;pr;θi	**Determine according**
P;H;hp;nom;θi<=P;H;hp;pr;θi<=P;H;hp;max;θi	E.3.8.1
P;hp;min<=P;H;hp;pr;θi<P;H;hp;nom;θi	E.3.8.2
P;H;hp<=θi<P;hp;min	E.3.8.3
In which: P;H;hp;pr;θi is the amount of thermal energy delivered by the air to water heat pump at outdoor temperature θ outdoor, determined according E.3.4, in kW; P;H;hp;nom;θi is the thermal energy delivered by the air to water heat pump at nominal power at outdoor temperature θ outdoor, determined according E.3.14, in kW; P;H;hp;max;θi is the thermal energy delivered by the air to water heat pump at maximum power at outdoor temperature θ outdoor, determined according E.3.14, in kW; P;hp;min is the minimum amount of thermal energy delivered by the heat pump, as stated by the manufacturer, or if measured according to E.4.3, in kW.	

ANNOTATION The minimum power is the lowest power at the heat pump will modulate to, or the lower limit of the modulation range. If an air to water heat pump is set to switch on or off at a certain power, this power counts as the lowest minimum power.

E.3.8.1 Determination of the COP between maximum and nominal power

The COP of a modulating air to water heat pump at thermal powers smaller than or equal to the maximum power and or greater than the nominal power is calculated by interpolating the COP between maximum and nominal power:

$$COP_{H;hp;\theta i} = COP_{H;hp;nom;\theta i} + \frac{P_{H;hp;pr;\theta i} - P_{H;hp;nom;\theta i}}{P_{H;hp;max;\theta i} - P_{H;hp;nom;\theta i}} \times \left(COP_{H;hp;max;\theta i} - COP_{H;hp;nom;\theta i}\right) \quad \text{(E.9)}$$

In which:

P;H;hp;pr;θi is the amount of thermal energy delivered by the air to water heat pump, determined according E.3.4, in kW;

P;H;hp;nom;θi is the thermal energy delivered by the air to water heat pump at nominal power, determined according E.3.14, in kW;

P;H;hp;max;θi is the thermal energy delivered by the air to water heat pump at maximum power, determined according E.3.14, in kW;

COP;H;hp;max;θi is the COP of the air to water heat pump at maximum thermal power, according E.3.8.5.

E.3.8.2 Determination of the COP between nominal and minimum power

The COP of a modulating air to water heat pump at thermal powers lower than nominal power and greater than or equal to minimum power is determined by:

$$COP_{H;hp;θi} = COP_{H;hp;nom;θi} \times \frac{\frac{P_{H;hp;pr;θi}}{P_{H;hp;nom;θi}}}{C_c \times \frac{P_{H;hp;pr;θi}}{P_{H;hp;nom;θi}} + (1 - C_c)} \text{ (E.10)}$$

In which:

COP;H;hp;nom;θi is the COP of the air to water heat pump at nominal thermal power, according to E.3.8.4;

P;H;hp;pr;θi is the amount of thermal energy delivered by the air to water heat pump, determined according E.3.4, in kW;

P;H;hp;nom;θi is the thermal energy delivered by the air to water heat pump at nominal power, determined according E.3.14, in kW;

Cc is the dimensionless degradation factor, according to E.3.22.

E.3.8.3 Determination of the COP under minimum power

The COP of a modulating air to water heat pump at thermal powers lower than nominal power and greater than or equal to minimum power is determined by:

$$COP_{H;hp;θi} = COP_{H;hp;nom;θi} \times \frac{\frac{P_{H;hp;min}}{P_{H;hp;nom;θi}}}{C_c \times \frac{P_{H;hp;min}}{P_{H;hp;nom;θi}} + (1 - C_c)} \text{ (E.11)}$$

In which:

COP;H;hp;nom;θi is the COP of the air to water heat pump at nominal thermal power, according to E.3.8.4;

P;hp;min is the minimum amount of thermal energy delivered by the heat pump, as stated by the manufacturer, or if measured according to E.4.3, in kW;

P;H;hp;nom;θi is the thermal energy delivered by the air to water heat pump at nominal power, determined according E.3.14, in kW;

Cc is the dimensionless degradation factor, according to E.3.22.

E.3.8.4 COP nominal power

For a modulating air to water heat pump with outdoor air or outdoor air and ventilation air as a source the COP at nominal power is determined as:

$COP_{H;hp;\theta i} = c_{nom;1} \times \theta_{evap;nom;\theta i} + c_{nom;2} \times \theta_{cond;nom;\theta i} + c_{nom,3}$ (E.12)

For a modulating air to water heat pump using only ventilation air a source the COP at nominal power is determined by:

$COP_{H;hp;\theta i} = c_{nom;2} \times \theta_{cond;nom;\theta i} + c_{nom,3}$ (E.13)

In which:

C;nom;i is a dimensionless constant, at nominal power, according E.3.18;

θ;evap;nom;θi is the average temperature of the air over the evaporator at a temperature θ outdoor, at nominal power, according E.3.9, in °C;

θ;cond;nom;θi is the average temperature of the air over the condenser at a temperature θ outdoor, at nominal power, according E.3.12, in °C.

E.3.8.5 COP maximum power

For a modulating air to water heat pump with outdoor air or outdoor air and ventilation air as a source the COP at maximum power is determined as:

$COP_{H;hp;\theta i} = c_{\max;1} \times \theta_{evap;\max;\theta i} + c_{\max;2} \times \theta_{cond;\max;\theta i} + c_{\max,3}$ (E.14)

For a modulating air to water heat pump using only ventilation air a source the COP at maximum power is determined by:

$COP_{H;hp;\theta i} = c_{\max;2} \times \theta_{cond;\max;\theta i} + c_{\max,3}$ (E.15)

In which:

C;max;i is a dimensionless constant, at maximum power, according E.3.18;

θ;evap;nom;θi is the average temperature of the air over the evaporator at a temperature outdoor at maximum power, according E.3.9, in °C;

θ;cond;nom;θi is the average temperature of the air over the condenser at a temperature θ outdoor, at maximum power, according E.3.12, in °C.

E.3.9 Temperature change over the evaporator

Determine the temperature over the evaporator, θevap;pi,θi, according:

$\theta_{evap;pi,\theta i} = \left(\theta_{evap;in;\theta i} - \frac{dT_{hp;evap;pi}}{2}\right)$ (E.16)

In which:

θ;evap;in;θi is the temperature of the air entering the evaporator, according E.3.10, in °C;

dT;hp;evap;pi is the average temperature drop of the air flowing through the evaporator, at nominal or at maximum power, according E.3.11, in K.

ANNOTATION The index pi represent the power equal to nominal(pi=nom) or maximum(pi=max) power.

E.3.10 Temperature of the air entering the evaporator

For an air to water heat pump with ventilation air as a source the temperature of the air at the entry of the evaporator is:

$$\theta_{evap;in;\theta i} = 20\ ^{\circ}\mathrm{C}$$

With an air to water heat pump with outdoor air as a source, and with a heat pump that used outdoor air as well as ventilation air as a source but where the ventilation air is led through a heat recovery unit (of which the recovered heat enters the building as preheated air), the temperature at the entry of the evaporator is determined by:

$\theta_{evap;in;\theta i} = \theta_{outside;i}$ (E.17)

With an air to water heat pump that uses a mixture of outdoor and ventilation air, the temperature of the air entering the evaporator is determined by:

$\theta_{evap;in;\theta i} = f_{outside;air} \times \theta_{outside;i} + \left(1 - f_{outside;air}\right) \times 20$ (E.18)

In which:

f;outdoor;air is the dimensionless factor for the contribution of outdoor air to the total amount of air entering the evaporator, according 5.5.1.2 of NEN 8088-1.

E.3.11 Average temperature drop over the evaporator

Determine the average temperature drop the evaporator, dT;hp;evap;pi, according:

$dT_{evap;pi} = \left(\frac{\sum_i \theta_{evap;in;pi,i} - \theta_{evap;out;pi,i}}{N}\right)$ (E.20)

In which:

θ;evap;in;pi,i is the temperature of the air entering the evaporator at a measurement in condition i, according E.4.1 for maximum power and E.4.2 for nominal power, in °C.

θ;evap;out;pi,i is the temperature of the air leaving the evaporator at a measurement in condition I, according E.4.1 for maximum power and E.4.2 for nominal power, in °C.

N is the dimensionless amount of measured conditions according NEN-EN 14511-2.

ANNOTATION 1 N refers to the amount of measured conditions according NEN-EN 14511-2, as described in E.4. For heat pumps with ventilation air as a source only, this amount is two. For heat pumps with outdoor air or outdoor air and ventilation air the amount of measurements is three.

ANNOTATION 2 The index pi represent the power equal to nominal(pi=nom) or maximum(pi=max) power.

E.3.12 Average temperature of the condenser

Determine the average temperature in the condenser, θcond;pi,θi, by:

$$\theta_{cond;pi,\theta i} = \frac{2\times\theta_{cond;in;pi,\theta i}+\theta_{cond;out;pi;\theta i}}{3} \text{ (E.21)}$$

In which:

θ;cond;in;pi,θi is the temperature of the water entering the condenser at nominal or maximum power, determined through table E.4 in ˚C;

θ;cond;out;pi,θi is the temperature of the water leaving the condenser at nominal or maximum power, determined through table E.3.13 in ˚C.

ANNOTATION The index pi represent the power equal to nominal(pi=nom) or maximum(pi=max) power.

E.3.13 Temperature of the condenser outlet

Determine the temperature of the water leaving the condenser, θcond;out;pi,θi, by:

$$\theta_{cond;out;pi,\theta i} = \theta_{cond;in;pi,\theta i} + \frac{P_{H;hp;pi,\theta i}}{F_{water;nom}\times 4182} \text{ (E.22)}$$

In which:

θ;cond;in;pi,θi is the temperature of the water entering the condenser at nominal or maximum power, determined through table E.4 in ˚C;

P;H;hp;pi,θi is the power of the air to water heat pump at a given outdoor temperature, determined by E.3.15 for maximum power and E.3.14 for nominal power, in kW;

f;water;nom is the nominal flow of water through the condenser, as stated by the manufacturer, in m^3/s.

ANNOTATION 1 The factor 4182 is the value for the density of water times its specific heat.

ANNOTATION 2 The index pi represent the power equal to nominal(pi=nom) or maximum(pi=max) power.

Table E.4 Condenser outlet temperature as a function of the outdoor temperature

θoutdoor ˚C	θcond;in			
	θsup≤35˚C ˚C	35˚C<θsup≤45˚C ˚C	45˚C<θsup≤60˚C ˚C	θsup≤60˚C ˚C
8	20,0	20,0	24,0	32,3
7	20,0	20,8	25,2	33,8
6	20,0	21,7	26,3	35,4
5	20,0	22,5	27,5	36,9
4	20,0	23,3	28,7	38,5
3	20,0	24,2	29,8	40,0
2	20,0	25,0	31,0	41,5
1	20,0	25,8	32,2	43,1
0	20,0	26,7	33,3	44,6
-1	20,5	27,5	34,5	46,2
-2	21,0	28,3	35,7	47,7
-3	21,5	29,2	36,8	49,2
-4	22,0	30,0	38,0	50,8
-5	22,5	30,8	39,2	52,3
-6	23,0	31,7	40,3	53,8
-7	23,5	32,5	41,5	55,4
-8	24,0	33,3	42,7	56,9
-9	24,5	34,2	43,8	58,5
-10	25,0	35,0	45,0	60,0
ANNOTATION θ;sup is the design supply temperature of the water used for room heating by the heating system				

E.3.14 Nominal power of the air to water heat pump

For an air to water heat pump with outdoor air or outdoor air and ventilation air as a source the nominal power, P;H;hp;nom;θi, is determined by:

$P_{H;hp;nom;\theta i} = \rho_{nom;1} \times \theta_{evap;nom;\theta i} + \rho_{nom;2} \times \theta_{cond;est;nom;\theta i} + \rho_{nom;3}$ (E.23)

In which:

P;nom;i is a dimensionless constant, determined for nominal power, according E.3.19;

θ;evap;nom;θi is the average temperature of the air over the evaporator at an outdoor temperature θoutdoor, at nominal power, according E.3.9, in ˚C;

θ;cond;est;nom;θi is the estimated average temperature of water over the condenser at a temperature θoutdoor, at nominal power, according E.3.16, in ˚C.

E.3.15 Power at full load

For air to water heat pumps with outdoor air or outdoor air and ventilation air as a source the maximum power *P;*H;hp;max;θ*i*, is determined according to:

$P_{H;hp;max\theta i} = \rho_{max;1} \times \theta_{evap;max;\theta i} + \rho_{max;2} \times \theta_{cond;est;max;\theta i} + \rho_{max;3}$ (E.25)

In which:

$p;max;i$ is a dimensionless constant, defined for maximum power, determined according to E.3.19;

θ;evap;max;θ*i* is the average temperature of the air over the evaporator at a temperature θoutdoor, at maximum power, determined according E.3.9, in °C;

θ;cond;est;max;θ*i* is the estimated average temperature of the water over the condenser at a temperature θoutdoor, at maximum power, determined according E.3.16, in °C.

For heat pumps which only use ventilation air as source, the power is determined according:

$$P_{H;hp;max\theta i} = \rho_{max;2} \times \theta_{cond;est;max;\theta i} + \rho_{max;3} \text{ (E.26)}$$

E.3.16 Approximating the temperature over the condenser

Determine the approximate temperature of water over the condenser, θ;cond;est;*pi*,θ*i*, according:

$$\theta_{cond;est;pi,\theta i} = \left(\theta_{cond;in;\theta i} + \frac{1}{3} \times dT_{hp;cond;pi}\right) \text{ (E.27)}$$

In which:

θ;cond;est;*pi*,θ*i* is the estimated average temperature of water over the condenser, at nominal or maximum power, in °C;

θ;cond;in;θ*i* is the temperature of the water entering the condenser, determined according table E.4, in °C;

*dT;*hp;cond;*pi* is the average rise in temperature of the water due to the condenser, upon measurement at nominal or maximum power, determined according E.3.17, in K.

ANNOTATION The index *pi* represents the power that is equal to the nominal power (*pi* = nom) of the maximum power (*pi* = max).

E.3.17 Average temperature increase of the condenser

Determine the average rise in temperature of the condenser, *dT;*hp;cond;*pi*, according:

$$dT_{hp;cond;pi} = \frac{\sum_i \theta_{cond;out;pi,i} - \theta_{cond;in;pi,i}}{N} \text{ (E.28)}$$

In which:

θ;cond;in;pi,i is the temperature of the water entering the condenser upon measurement as per condition i, determined according E.4.1 for maximum power and E.4.2 for nominal power, in °C;

θ;cond;out;pi,i is the temperature of the water leaving the condenser upon measurement as per condition i, determined according E.4.1 for maximum power and E.4.2 for nominal power, in °C;

N is the dimensionless number of measuring conditions upon measurement according to NEN-EN 14511-2.

ANNOTATION 1 *N* refers to the number of measuring conditions upon measurement according to NEN-EN 14511-2, as described in E.4.1. For air to water heat pumps with only ventilation air as source this number is two, for outdoor air or for outdoor air and ventilation air as source this number is three.

ANNOTATION 2 The index pi represents the power that is equal to the nominal power (pi=nom) or maximum power (pi=max).

E.3.18 Determination of the COP constants

The constants for the calculation of the *COP* at maximum power, $c_{max;i}$, with air to water heat pumps with outdoor air or outdoor air and ventilation air as source, are determined according:

$$c_{max;1} = \frac{COP_{H;hp;1} \times (\theta_{cond;2} - \theta_{cond;3}) + COP_{H;hp;2} \times (\theta_{cond;3} - \theta_{cond;1}) + COP_{H;hp;3} \times (\theta_{cond;1} - \theta_{cond;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad (E.29)$$

$$c_{max;2} = \frac{\theta_{evap;1} \times (COP_{H;hp;2} - COP_{H;hp;3}) + \theta_{evap;2} \times (COP_{H;hp;3} - COP_{H;hp;1}) + \theta_{evap;3} \times (COP_{H;hp;1} - COP_{H;hp;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad (E.30)$$

$$c_{max;3} = \frac{\theta_{evap;1} \times (COP_{H;hp;3} \times \theta_{cond;2} - COP_{H;hp;2} \times \theta_{cond;3}) + \theta_{evap;2} \times (COP_{H;hp;1} \times \theta_{cond;3} - COP_{H;hp;3} \times \theta_{cond;1})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})}$$
$$+ \frac{\theta_{evap;3} \times (COP_{H;hp;2} \times \theta_{cond;1} - COP_{H;hp;1} \times \theta_{cond;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad (E.31)$$

The constants for the calculation of the *COP* at maximum power, $c_{max;i}$, with air to water heat pumps with only ventilation air as source, are determined according:

$$c_{max;2} = \frac{COP_{H;hp;1} - COP_{H;hp;2}}{\theta_{cond;1} - \theta_{cond;2}} \quad (E.32)$$

$$c_{max;3} = \frac{\theta_{cond;1} \times COP_{H;hp;1} - \theta_{cond;2} \times COP_{H;hp;2}}{\theta_{cond;1} - \theta_{cond;2}} \quad (E.33)$$

E.3.19 Determination of the power constants

The constants for the calculation of the maximum power as function of the evaporator- and condenser temperature, *p;*max;*i*, with air to water heat pumps with outdoor air of outdoor air and ventilation air as source, are determined according:

$$p_{max;1} = \frac{P_{H;hp;1} \times (\theta_{cond;2} - \theta_{cond;3}) + P_{H;hp;2} \times (\theta_{cond;3} - \theta_{cond;1}) + P_{H;hp;3} \times (\theta_{cond;1} - \theta_{cond;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad \text{(E.34)}$$

$$p_{max;2} = \frac{\theta_{evap;1} \times (P_{H;hp;2} - P_{H;hp;3}) + \theta_{evap;2} \times (P_{H;hp;3} - P_{H;hp;1}) + \theta_{evap;3} \times (P_{H;hp;1} - P_{H;hp;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad \text{(E.35)}$$

$$p_{max;3} = \frac{\theta_{evap;1} \times (P_{H;hp;3} \times \theta_{cond;2} - P_{H;hp;2} \times \theta_{cond;3}) + \theta_{evap;2} \times (P_{H;hp;1} \times \theta_{cond;3} - P_{H;hp;3} \times \theta_{cond;1})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} + \frac{\theta_{evap;3} \times (P_{H;hp;2} \times \theta_{cond;1} - P_{H;hp;1} \times \theta_{cond;2})}{\theta_{evap;1} \times (\theta_{cond;2} - \theta_{cond;3}) + \theta_{evap;2} \times (\theta_{cond;3} - \theta_{cond;1}) + \theta_{evap;3} \times (\theta_{cond;1} - \theta_{cond;2})} \quad \text{(E.36)}$$

With heat pumps with only ventilation air as source the constants, *p*max;*i*, are determined according:

$$\rho_{max;2} = \frac{P_{H;hp;1} - P_{H;hp;2}}{\theta_{cond;1} - \theta_{cond;2}} \quad \text{(E.37)}$$

$$\rho_{max;2} = \frac{\theta_{cond;1} \times P_{H;hp;1} - \theta_{cond;2} \times P_{H;hp;2}}{\theta_{cond;1} - \theta_{cond;2}} \quad \text{(E.38)}$$

In which:

*P;*H;hp;*i* is the power of the air to water heat pumps at condition *i*, at maximum power, determined according E.4.1, in kW;

θ;evap;*i* is the average temperature of the air over the evaporator, at condition *i*, at maximum power, determined according E.3.20, in °C;

θ;cond;*i* is the average temperature of the liquid stream over the condenser, at condition *i*, at maximum power, determined according E.3.20, in °C.

The constants for the calculation of the nominal power of modulating air to water heat pumps, *p;*nom;*i* are determined in the same manner as described above, but with:

*P;*H;hp;*i* is the power of air to water heat pumps at condition *i*, at nominal power, *P;*H;hp;nom;*i*, determined according E.4.2, in kW;

θ;evap;*i* is the average temperature of the air over the evaporator, at condition *i,* at nominal power, θ;evap;nom;*i*, determined according E.3.21, in °C;

θ;cond;*i* is the average temperature of the liquid stream over the condenser, at condition *i*, at nominal power, θ;cond;nom;*i*, determined according E.3.21, in °C.

E.3.20 Average temperature evaporator/condenser at maximum power

Determine the average evaporator- and condenser temperature at maximum power according:

$$\theta_{evap;i} = \frac{\theta_{evap;in;i}+\theta_{evap;out;i}}{2} \text{ (E.39)}$$

And:

$$\theta_{cond;i} = \frac{2\times\theta_{cond;in;i}+\theta_{cond;out;i}}{3} \text{ (E.40)}$$

In which:

θ;evap;in;*i* is the temperature of the entering air stream over the evaporator, at condition *i* at maximum power, determined according E.4.1, in °C;

θ;evap;out;*i* is the temperature of the leaving air stream over the evaporator, at condition *i*, at maximum power, determined according E.4.1, in °C;

θ;cond;in;*i* is the temperature of the entering liquid stream over the condenser, at condition *i*, at maximum power, determined according E.4.1, in °C;

θ;cond;out;*i* is the temperature of the leaving liquid stream over the condenser, at condition *i*, at maximum power, determined according E.4.1, in °C.

E.3.21 Average temperature evaporator/condenser at nominal power

Determine the average evaporator- and condenser temperature at nominal power according:

$$\theta_{evap;nom;i} = \frac{\theta_{evap;in;nom;i}+\theta_{evap;out;nom;i}}{2} \text{ (E.39)}$$

And:

$$\theta_{cond;nom;i} = \frac{2\times\theta_{cond;in;nom;i}+\theta_{cond;out;nom;i}}{3} \text{ (E.40)}$$

In which:

θ;evap;in;nom;*i* is the temperature of the entering air stream over the evaporator, at condition *i* at nominal power, determined according E.4.2, in °C;

θ;evap;out;nom;*i* is the temperature of the leaving air stream over the evaporator, at condition i, at nominal power, determined according E.4.2, in °C;

θ;cond;in;nom;*i* is the temperature of the entering liquid stream over the condenser, at condition *i*, at nominal power, determined according E.4.2, in °C;

θ;cond;out;nom;*i* is the temperature of the leaving liquid stream over the condenser, at condition *i*, at nominal power, determined according E.4.2, in °C.

E.3.22 Determination of the degradation factor

The degradation factor *Cc* is determined with:

$$C_C = \frac{\left(\frac{COP_{hp;pl;2}}{COP_{hp;min}} \times \frac{P_{hp;min}}{P_{hp;pl;2}}\right) - 1}{\frac{P_{hp;min}}{P_{hp;pl;2}} - 1} \quad \text{(E.43)}$$

In which:

*COP;*hp;pl;2 is the *COP* in partial load in condition 2 determined according E.4.2;

*COP;*hp;min is the *COP* at minimum power, determined according E.4.3;

*P;*hp;min is the minimal thermal power, determined according E.4.3, in kW;

*P;*hp;pl;2 is the thermal power in partial load, at condition 2, determined according E.4.2, in kW.

In case no measurement at minimal power has been executed, the degradation factor is provided by:

Cc = 0,8

E.4 Measurement of the energy performance of the air to water heat pump

E.4.1 Maximum power

The performance of an air to water heat pump is to be measured according NEN-EN 14511-2. During this measurement, depending on the application, measurements must be done at the points stated in table E.5.

Table E.5 – Test conditions at maximum power

		Condition	**Evaporator**		**Condenser**	
			Inlet dry bulb temperature °C	**Inlet wet bulb temperature °C**	**Inlet temperature °C**	**Outlet temperature °C**
Outdoor air	Outdoor air	1	7	6	40 (a)	45 (50)
	Outdoor air (floor heating and such)	2	7	6	30	35
	Outdoor air (floor heating and such)	3	-7	-8	a	35
Outdoor air(if -7°C is outside the heat pumps range)	Outdoor air	1	7	6	40 (a)	45 (50)
	Outdoor air (floor heating and such)	2	7	6	30	35
	Outdoor air (floor heating and such)	3	2	1	a	35
Ventilation air	Ventilation air	1	20	12	40 (a)	45 (50)
	Ventilation air (floor heating and such)	2	20	12	a	35

a) The test is done at the flow acquired during the test corresponding the standard testing condition(NEN-EN 14511-2).

ANNOTATION 1 If the air to water heat pump is also applied in heating systems with high temperatures, the values between parenthesis need to be used as well.

When measuring according NEN-EN-14511-2, the outlet temperature of the evaporator is to be measured also. The COP then follows from the various relevant conditions in table E.5. After these tests, the data as shown in table E.6 is known.

During these measurements, the COP is to be determined without the influence of the circulation pump. If measurements include the circulation pump, the absorbed power has to be reduced by the electrical power absorbed by the circulation pump.

Air to water heat pumps using outdoor as well as ventilation air as a source should, when measuring, use, as the air inlet condition, a combination of three relevant outdoor conditions and corresponding ventilation air conditions(with the same supply conditions).

For air to water heat pumps with a variable air flow, per condition as described above, several air flows can be tested. When doing this, different measuring conditions may not be further apart than 100 m³/h.
For the determination of the energy performance of a specific building, the altered test data that is to be used to determine the COP, the nominal power and the energy fraction, is to be determined by interpolating between the values in table E.6 at the relevant air flow.

ANNOTATION 2 An example of an air to water heat pump with variable air flow is an air to water heat pump with ventilation air as a heat source in which the air flow is altered depending on the floor surface area.

Table E.6 – Test data at maximum power

	Evaporator		**Condenser**			
	Inlet temperature ˚C	**Outlet temperature ˚C**	**Inlet temperature ˚C**	**Outlet temperature ˚C**	**COP -**	**Power kW**
Condition 1	θevap;in;1	θevap;out;1	θcond;in;1	Θcond;out;1	COPhp;1	Php;1
Condition 2	θevap;in;2	θevap;out;2	θcond;in;2	θcond;out;2	COPhp;2	Php;2
(Condition 3)	θevap;in;3	θevap;out;3	θcond;in;3	θcond;out;3	COPhp;3	Php;3
ANNOTATION 3 Condition 3 is of no importance(disappears) when only ventilation air is used						

E.4.2 Nominal power

Table E.7 – Test conditions at nominal power

		Condition	**Evaporator**		**Condenser**	
			Inlet dry bulb temperature ˚C	**Inlet wet bulb temperature ˚C**	**Inlet temperature ˚C**	**Outlet temperature ˚C**
Outdoor air	Outdoor air	1	7	6	a	45 (50)
	Outdoor air (floor heating and such)	2	7	6	a	35
	Outdoor air (floor heating	3	-7	-8	a	35

	and such)					
Outdoor air(if -7˚C is outside the heat pumps range)	Outdoor air	1	7	6	a	45 (50)
	Outdoor air (floor heating and such)	2	7	6	a	35
	Outdoor air (floor heating and such)	3	2	1	a	35
Ventilation air	Ventilation air	1	20	12	a	45 (50)
	Ventilation air (floor heating and such)	2	20	12	a	35
a) The test is done at the flow acquired during the test corresponding the standard testing condition(NEN-EN 14511-2).						

For the energy performance determination of a modulating air to water heat pump, beside the full load test described in the previous paragraph, partial load test need to be done. The partial load tests should take place according E.4.1, but at the conditions described in table E.7 and at nominal power. The thermal power the air to water heat pump needs to deliver at nominal power is:

$P_{hp;pl;nom,1,2,\dots} = P_{hp,1,2\dots} \times CR_{nom}$ (E.44)

In which:

P;hp;pl;nom,1,2,… is the thermal power the air to water heat pump needs to deliver at nominal power, at the same conditions as the tests at maximum power(E.4.1) in kW;

P;hp,1,2,… is the thermal full load power, at condition I, according E.4.1, in kW;

CR;nom is capacity ratio between nominal and maximum power, as stated by the manufacturer or supplier, in %.

The power delivered in the partial load tests, Php;pl,1,2,…, should be within a margin 5% of the nominal partial load, Php;pl;nom,1,2,…. The capacity ratio is a fixed value for all conditions, the nominal thermal partial load power is to be determined per condition. The nominal capacity ratio is the ratio of the thermal power at nominal and maximum operation determined by the manufacturer or supplier. This ratio can be chosen to be any value, but the actual performance is approached best when the capacity ratio is chosen so that partial load measurements are executed at thermal powers where the air to water heat pump achieves the highest COP.

ANNOTATION 1 If, for example, a capacity ratio of 60% is used, at the nominal power measurement a thermal power of 60% of the thermal power of the full load test at the same temperature condition needs to be delivered.

Per test, the following data should at least be available:

Table E.8 – Test data at partial load tests

	Evaporator		**Condenser**			
	Inlet temperature ˚C	**Outlet temperature ˚C**	**Inlet temperature ˚C**	**Outlet temperature ˚C**	**COP -**	**Power kW**
Condition 1	θevap;in;1	θevap;out;1	θcond;in;1	Θcond;out;1	COPhp;1	Php;1
Condition 2	θevap;in;2	θevap;out;2	θcond;in;2	θcond;out;2	COPhp;2	Php;2
(Condition 3)	θevap;in;3	θevap;out;3	θcond;in;3	θcond;out;3	COPhp;3	Php;3
ANNOTATION 2 Condition 3 is of no importance(disappears) when only ventilation air is used						

E.4.3 Minimum power

Table E.9 – Test conditions at minimum power

	Evaporator		Condenser	
	Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet temperature °C	Outlet temperature °C
Outdoor air	7	6	a	35
Ventilation air	20	12	a	35
a) The test is done at the flow acquired during the test corresponding the standard testing condition(NEN-EN 14511-2).				

Table E.10 – Test data at minimum load

	Evaporator		Condenser			
	Inlet temperature °C	Outlet temperature °C	Inlet temperature °C	Outlet temperature °C	COP -	Power kW
Minimum load	θevap;in;min	θevap;out;min	θcond;in;min	Θcond;out;min	COPhp;min	Php;min

E.5 Time fraction and flow of ventilation air in the heat pump

For use with NEN 8088-1, for air to water heat pumps, a monthly time fraction is used for the amount of time the device is in operation. The flow of air for ventilation is also calculated.

E.5.1 Monthly total time fraction

Determine the time fraction for operation per month by:

$$f_{t;hp-on;mi} = Min\left[f_{W;t;hp-on;mi} + f_{H;t;hp-on;mi}; 1\right] \text{ (E.45)}$$

In which:

f;t;hp-on;mi is the time fraction the device is in operation, in month mi;

f;W;t;hp-on;mi is the time fraction the device is in operation for domestic water, in month mi;

f;H;t;hp-on;mi is the time fraction the device is in operation for space heating, in month mi;

Min [...] is the smallest of the two values.

E.5.1.1 Time fraction for domestic hot water

For heat pumps that are used only for space heating, the time fraction for domestic hot water is determined by:

$$f_{W;t;hp-on;mi} = 0 \text{ (E.46)}$$

For all other types of heat pumps:

$$f_{W;t;hp-on;mi} = \frac{Q_{W;dis;nren;si,mi} \times F_{W;gensi,gpref,mi}}{t_{mi} \times P_{W;hpmax} \times 1000} \text{ (E.47)}$$

In which:

f;W;t;hp-on;mi is the time fraction the device is in operation for domestic water, in month mi;

Q;W;dis;nren;si,mi is the total amount of energy needed for domestic hot water, in month mi, by the non-sustainable heat generators(nren) delivered to the distribution part of system si, according 19.1.2, in MJ;

F;W;gen;si,gpref,mi is the dimensionless energy fraction for domestic hot water, in month mi, the preferential generator gpref delivers to the system si, according 19.7.4;

t;mi is the length of month mi, according 21.2 in Ms;

P;W;hp;max is the maximum thermal power of the heat pump for the function of domestic hot water, as stated by the manufacturer, in kW.

ANNOTATION The factor 1000 is used to convert the power from kW to W.

E.5.1.2 Time fraction for space heating

For heat pumps that are used only for domestic hot water, the time fraction for space heating is determined by:

$f_{H;t;hp-on;mi} = 0$ (E.48)

For all other types of heat pumps:

$f_{H;t;hp-on;mi} = \sum_{\theta i} f_{H;t;hp-on;\theta i} \times f_{t;\theta i,mi}$ (E.49)

In which:

f;H;t;hp-on;mi is the time fraction the device is in operation for space heating, in month mi;

f;H;t;hp-on;θi is the time fraction the heat pump is in operation for space heating, at outdoor temperature θi, according E.5.1.2.1 for switching heat pumps and according E.5.1.2.2 for modulating heat pumps.

f;t;θi,mi is the time fraction that the outdoor temperature θi appears, in month mi, according table E.11.

Table E.11 – Time fraction of outdoor temperatures per month

θ_i	$f_{t;\theta i;mi}$											
°C	Jan	Feb	Mrt	Apr	Mei	Jun	Jul	Aug	Sept	Okt	Nov	Dec
16	0,0000	0,0164	0,0013	0,0153	0,0578	0,1028	0,1237	0,0712	0,1208	0,0605	0,0000	0,0000
15	0,0000	0,0134	0,0040	0,0250	0,0968	0,0972	0,1035	0,0632	0,1361	0,0753	0,0000	0,0000
14	0,0000	0,0223	0,0108	0,0264	0,0847	0,0917	0,1035	0,0672	0,1278	0,1129	0,0042	0,0000
13	0,0000	0,0119	0,0134	0,0528	0,1102	0,0694	0,0632	0,0605	0,1069	0,1035	0,0181	0,0013
12	0,0000	0,0432	0,0175	0,0833	0,1277	0,0972	0,0349	0,0390	0,0917	0,1129	0,0403	0,0255
11	0,0081	0,0625	0,0349	0,0958	0,0954	0,0431	0,0202	0,0390	0,0667	0,1062	0,1028	0,0188
10	0,0296	0,0357	0,0887	0,0875	0,0860	0,0194	0,0081	0,0242	0,0472	0,0874	0,1153	0,0228
9	0,0538	0,0223	0,1465	0,1083	0,0591	0,0194	0,0067	0,0121	0,0167	0,0739	0,1181	0,0497
8	0,0712	0,0595	0,1008	0,1569	0,0296	0,0208	0,0027	0,0081	0,0069	0,0699	0,1250	0,0699
7	0,1290	0,0744	0,1102	0,0597	0,0215	0,0139	0,0013	0,0027	0,0139	0,0591	0,0861	0,0766
6	0,0726	0,0655	0,1331	0,0431	0,0134	0,0153	0,0000	0,0000	0,0000	0,0309	0,0778	0,0887
5	0,0672	0,0833	0,1492	0,0361	0,0134	0,0014	0,0000	0,0000	0,0028	0,0296	0,1014	0,1398
4	0,0941	0,0744	0,0833	0,0333	0,0040	0,0000	0,0000	0,0000	0,0000	0,0148	0,0611	0,0914
3	0,0538	0,1116	0,0484	0,0361	0,0013	0,0000	0,0000	0,0000	0,0000	0,0108	0,0597	0,0739
2	0,0444	0,0551	0,0336	0,0278	0,0027	0,0000	0,0000	0,0000	0,0000	0,0000	0,0222	0,0887
1	0,0444	0,0744	0,0188	0,0278	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0139	0,1022
0	0,0511	0,0640	0,0027	0,0069	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0222	0,0363
-1	0,0578	0,0432	0,0027	0,0028	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0125	0,0323
-2	0,0457	0,0476	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0111	0,0296
-3	0,0175	0,0030	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0042	0,0121
-4	0,0296	0,0060	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0042	0,0255
-5	0,0444	0,0045	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0054
-6	0,0215	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0054
-7	0,0242	0,0060	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0040
-8	0,0134	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000
-9	0,0161	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000
-10	0,0108	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000

E.5.1.2.1 Time fraction for switching heat pump per outdoor temperature

Determine the time fraction per outdoor temperature for a switching heat pump by:

$$f_{H;t;hp-on;\theta i} = Min\left[\frac{P_{H;\theta i}}{P_{H;hp;max;\theta i}};1\right] \quad \text{(E.50)}$$

In which:

f;H;t;hp-on;θi is the time fraction the heat pump is in operation for space heating, at outdoor temperature θi;

P;H;θi is the net total amount of heat required per space, at an outdoor temperature θi, according E.3.5, in kW;

P;H:hp;max;θi is the maximum thermal energy the heat pump can deliver at outdoor temperature θi, according E.3.15, in kW;

Min […] is the smallest of the two values.

E.5.1.2.2 Time fraction for modulating heat pump per outdoor temperature

Determine the time fraction per outdoor temperature for a modulating heat pump by:

$$f_{H;t;hp-on;\theta i} = Min\left[\frac{P_{H;\theta i}}{P_{hp;min}};1\right] \quad \text{(E.51)}$$

In which:

f;H;t;hp-on;θi is the time fraction the heat pump is in operation for space heating, at outdoor temperature θi;

P;H;θi is the net total amount of heat required per space, at an outdoor temperature θi, according E.3.5, in kW;

P;hp;min is the minimum power the heat pump can deliver, as stated by the manufacturer, or if measured, according E.4.3, in kW;

Min […] is the smallest of the two values.

E.5.2 Ventilation air flow

For heat pumps using ventilation as a source only, the flow of air, q;ve;mech;hp, is determined by:

$q_{ve;mech;hp} = q_{a;total}$ (E.52)

For heat pumps using outdoor air as well as ventilation air, the flow of air, q;ve;mech;hp, is determined by:

$q_{ve;mech;hp} = q_{a;total} \times \left(1 - f_{air;outdoor}\right)$ (E.53)

In which:

q;ve;mech;hp is the amount of air used by the heat pump, in dm^3/s;

q;a;total is total flow of air over the evaporator, as stated by the manufacturer, in dm^3/s;

f;air;outdoor is a factor for the contribution of outdoor air to the total flow of air over the evaporator.

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Report no. HPT-AN39-1