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Quality Installation / Quality Maintenance Sensitivity Studies

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Preface
This project was carried out within the Heat Pump Programme, HPP 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 IEA Heat Pump Programme
The Implementing Agreement for a Programme of Research, Development, Demonstration and Promotion of Heat Pumping Technologies (IA) forms the legal basis for the IEA Heat Pump Programme. Signatories of the IA are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the IA 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 IEA Heat Pump Programme is played by the IEA Heat Pump Centre (HPC). Consistent with the overall objective of the IA 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.

For further information about the IEA Heat Pump Programme and for inquiries on heat pump issues in general contact the IEA Heat Pump Centre at the following address:

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Final Report: IEA HPP Annex 36

Quality Installation / Quality Maintenance Sensitivity Studies

(Avoiding Heat Pump Efficiency Degradation Due to Poor Installations and Maintenance)

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Sweden
United Kingdom
United States (Operating Agent)

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Disclaimer

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ABSTRACT

It is widely recognized that residential and commercial heat pump equipment experience significant “in-field” performance loss (i.e., capacity and efficiency) depending on how the components are sized, matched, installed, and subsequently field-maintained. IEA Annex 36 evaluated how deficiencies in these areas cause heat pumps to perform inefficiently and hence waste considerable energy. Some investigations (France and UK) included field tests and assessments. Other efforts included laboratory (France, UK, US) and/or modelling work (France, UK, US), and statistical analyses of large failure databases (Sweden).

The outcome from this Annex activity clearly identifies that poorly installed and/or maintained heat pumps operate inefficiently and waste considerable energy compared to their “as-designed” potential. Additionally, it is clear that small faults for a given field-observed practice are significant, that some attribute deviations (in various equipment applications and geographical locations) have a larger impact than others, and that multiple faults or deviations have a cumulative impact on heat pump performance.

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PART 1: EXECUTIVE SUMMARY

Introduction to Annex 36

A significant opportunity for improving heat pump operating performance (capacity, efficiency, etc.) is related to how well the cooling and/or heating equipment is installed and subsequently maintained in varied building applications. One challenge in promoting proper installation and maintenance practices is that the marketplace incentivizes 'cutting corners' in the collective belief that low first cost is the important driver. This results in poor equipment performance and poor value as received by the end-user. Generally, an underlying cause of this predicament is that the various industry participants (e.g., original equipment manufacturers, equipment distributors, system designers, installers, etc.) – as well as the end users (e.g., building owners, operators, customers) do not appreciate how cumulative small deviations in installation and maintenance impact overall equipment performance.

It is universally recognized that heat pump equipment suffers significant performance loss depending on how well the equipment is designed, installed, and subsequently maintained. Some common deficiencies for heat pumps include:

- oversized equipment,
- improper refrigerant charge,
- incorrect airflow over the coil,
- incorrect water flow through the coil
- leaky air ducts.

However, the heating, ventilation, and air-conditioning (HVAC) industry lacked good quantitative information on the relative value and importance of varied field installation / maintenance practices and the effect that deficiencies may cause. Prior to the undertaking of this Annex activity, it was unclear whether small variances for a given-field-observed practice are significant, whether multiple faults or deficiencies have an additive impact on heat pump performance, and whether some deviations (in various equipment applications and geographical locations) have a larger impact than others. Earlier investigations examined existing equipment in the field and the impact that corrective measures may provide. However, the resulting anecdotal data provided limited quantifiable information on the impact of the various corrective measures on equipment performance.

Annex 36 investigated heat pump faults commonly found in operating equipment, and to assess the relative capacity and efficiency degradations caused by faults. The resultant effort is aimed at positioning stakeholders to better understand how installation and maintenance practices affect heat pump performance.

Objectives

The desired end-product objectives for the Annex 36 work were:

- Develop information for use by key stakeholders in industry (HVACR and construction trades), government (policy makers), and building sector (owners/operators).
- Produce reliable data to position each participating Annex country to evaluate its pertinent industry standards and practices for varied heat pump applications to ensure optimum heat pump performance.
- Reduce usage of energy and emissions of greenhouse gases by encouraging use of quality heat pump installation and maintenance practices.

Target Audience and Benefits

The sectors targeted to benefit from this Annex activity included:

- HVAC practitioners responsible for designing, selecting, installing, and maintaining heat pump systems in varied applications.
- Building owner/operators interested in achieving improved comfort conditioning and efficient performance from their HVACR equipment.

- Entities charged with minimizing energy utilization (i.e., utilities, utility commissions, energy agencies, legislative bodies, etc.) for different types of heat pump applications in varied geographic conditions.

Improved understanding on how quality installation (QI) and quality maintenance (QM) impacts equipment performance will help program managers to focus attention, resources, and effort on the important design, installation, and maintenance parameters that impact their varied programmatic efforts. Application of this improved understanding can help homeowners and building owners / operators appreciate the complete value-to-cost proposition – not merely the "first price" – when making equipment purchasing, installation, and maintenance decisions. Ultimately, discriminating customers will realize enhanced comfort, reduced energy usage, improved occupant productivity, and enhanced occupant safety.

Annex Activities and Time Schedule

Each Annex participant focused their independent efforts on those heat pump types and applications that are in general usage within their residential and commercial building sectors; see Table ES-1. Each country's recognized standards, industry practices, and measurement instrumentation approaches served as the basis for the individual efforts.

Table ES-1: Annex 36 Tasks and Time Schedule

Start Date	End Date	Annex 36 Activities
Nov 2010	Apr 2011	Task 1 – Critical literature survey: Undertake literature review and critical analysis of the results from prior related research to identify QI and QM metrics and to secure quantitative and qualitative impacts of various deviations from the specified levels recommended in industry's relevant QI and QM standards. This may include benchmarking of current QI and QM standards and practices in place in each participating country.
May 2011	Oct 2011	Task 2 – Identify sensitivity parameters: Building on the literature search, and working with pertinent country experts, develop a consensus of the QI and QM elements to be included in the participant's subsequent investigation.
Nov 2011	Jul 2012	Task 3 – Field investigation, Modeling and/or lab-controlled measurements: Field investigations, computer modeling, and/or laboratory-controlled experiments undertaken to verify the results provided by earlier researchers, to fill-in 'holes' in regard to quantifying the impacts of QI and QM vs. non-QI and non-QM applications, and to better understand the singular and additive impacts on equipment effectiveness.
Aug 2012	Apr 2013	Task 4 – Simulations on seasonal impacts: Simulations undertaken to determine the role that seasonal temperature differences have on the varied QI and QM elements and the resultant impact on heat pump performance. This modeling of different geographic temperatures impacts within a country or region (i.e., different outdoor temperature and humidity conditions) help to identify regional effects if any.
May 2013	Nov 2013	Task 5 – Report and information dissemination: Individual country analysis finalized and the individual country reports consolidated into a single Annex 36 volume.

Work Emphasis

Reports from the four participating countries are consolidated in this volume. The specific focus areas and emphasis undertaken by each country are identified in Table ES-2 below.

Table ES-2: Annex 36 Participants' Focus Areas and Work Emphasis

Annex 36 Participants	Focus Area	Work Emphasis
France	EdF – Space heating and water heating applications.	*Field:* Customer feedback survey on heat pump system installations, maintenance, and after-sales service. *Lab*: Water heating performance tests on sensitivity parameters and analysis.
Sweden	SP – Large heat pumps for multi-family and commercial buildings. KTH – Fault detection and diagnoses in heat pump systems.	*Field:* SP – Literature review of operation and maintenance for larger heat pumps. Interviews with real estate companies owning heat pumps. KTH - investigations and statistical analysis of 68,000 heat pump failures. *Modeling/Lab*: Determination of failure modes and analysis of found failures (SP) and failure statistics (KTH).
United Kingdom	DECC – Home heating with ground-to-water, water-to-water, and air-to-water systems.	*Field:* Monitor 83 domestic heat pumps and make modifications to improve performance. *Lab:* Investigate the impact of thermostatic radiator valves on heat pump system performance.
United States (Operating Agent)	NIST – Air-to-air heat pump in residential applications (cooling and heating).	*Lab*: Cooling and heating tests with imposed faults to develop correlations for heat pump performance degradation due to faults. *Modeling*: Seasonal analyses modeling to evaluate the effect of installation faults on heat pump seasonal energy consumption. Include effects of different building types (slab vs. basement foundations, etc.) and climates in the assessment of impact of various faults on heat pump performance.

Participants
ACCA → Air Conditioning Contractors of America
DECC → Department of Energy and Climate Change (UK)
EdF → Electricité de France
KTH → Royal Institute of Technology (Sweden)
NIST → National Institute of Standards and Technology (US)
ORNL → Oak Ridge National Laboratory (US)
SP → Technical Research Institute of Sweden

Country Specific Significant Results and Findings

The individual country reports go into significant details as to the work undertaken, application details, assumptions and constraints used, as well as results and implications of same. However, some significant findings drawn from the country reports are indicated in the balance of this summary.

FRANCE – *FOCUSED PROJECTS ON HEAT PUMP SPACE HEATING AND WATER HEATING*

The country report details work in four independent areas of interest:

1. Customer study aimed at determining the satisfaction level of heat pump end users,
2. Field analysis of an in-situ heat pump to explore how various attributes impact COP,
3. Sensitivity analysis of refrigerant charge on air-to-air heat pump performance,
4. In-situ field trials in ten residential homes of a Japanese thermodynamic ECS (Ecocute program) water heater to ascertain what technology transfer was necessary to adapt them to the European market.

Some of these efforts were commenced starting as early as 2005, others commenced in 2012.

Customer Survey

The customer survey was aimed at ascertaining the satisfaction level of the user of heat pump systems based on: heating system by type of heat pump, ease of maintenance, and the repair process. Feedback was collected from 202 owners of houses equipped with a thermodynamic heating system. Heat pumps were installed between January 2005 and October 2008 in four geographical regions: PACA, Midi-Pyrénées, Bretagne and north-eastern France. The information was collected in June and July 2009 based on telephone interviews. The distribution of heat pump types involved in the survey activity are shown in Table ES-3.

Table ES-3: Distribution of Heat Pump Types in French Literature Survey

Type of Heat pump	Distribution (%)
Air/water (heating only)	26
Air/water (additional heating)	28
Air/air	23
Geothermal	22

Among the air/water heat pumps, approximately two-thirds (2/3) are an outdoor single piece. In most cases, the heat pump manages the heating of the entire accommodation (94%), a little less for the air/air heat pumps (85%). As far as air/water and geothermal heat pumps are concerned, the emitters are mainly hot water radiators (71%). Even so, the heating floors represent one-fourth of the installations. In one-third of the cases, the heat pumps produce domestic hot water in addition to heating. One-third of heat pumps also provide air conditioning, but heat pumps meeting all three uses (heating, air conditioning, and domestic hot water) are rare. The average reported price of a heat pump installation, over the entire sample, is 13,500 Euros (€).

As can be seen in Table ES-4 below, the maintenance frequency for most heat pumps is usually carried out once a year. A guarantee of the proper operation of the installation and peace of mind for the customer are the main motives for signing a maintenance contract. Maintenance contracts come in a wide range of prices ranging from 80 € to 350 € with an average value of 144 €. Yet, there is no significant difference between the types of heat pumps.

Table ES-4: Frequency of Heat Pump Maintenance

Twice a year	5%
Once a year	89%
Once every two years	1%
As and when required	1%
No opinion	3%

One-fourth of field-surveyed heat pump owners had encountered technical incidents or heat pump failures (see Table ES-5 for frequency by failure mode). In most cases, it is the heating function that is affected by the incident. In nearly two-thirds of the cases, the incident resulted in the system failing to operate. However, in more than half the cases, the incident is regarded as "minor" by the customer.

The main conclusions that emerged from this survey were:

- One-third of heat pump owners indicated that a maintenance contract is not useful.
- Only 38% of heat pump owners had a maintenance contract, which was most often contracted at the time of installation. The average value of a maintenance contract was 144 €.
- The comfort provided by the heat pumps was considered to be good.
- The identified weak points were minor; but, equipment noise level was identified as an important issue.
- There were a rather large number of incidents; one-fourth of heat pump owners experienced problems.
- Apart from the investment cost, the consumption and maintenance costs were affordable.
- The main improvements expected by customers concern the reduction in the noise level and comfort control.
- Improvements expected by the customers concern: reduction in the installation cost, reduction in the noise level, comfort control, more explanations from the installers, and improvement of efficiencies for energy saving or for comfort.

Table ES-5: Failure Occurrences by Failure Mode

Incident	% of total occurrences	Incident	% of total occurrences
Water or liquid leak	16%	Filter clogged, obstructed	4%
Installation error, electrical connection error	12%	Circulator failure	2%
Refrigerant leak	8%	Compressor failure	2%
Adjustment problem	8%	Fan failure	2%
Frost, ice on fan	8%	Failure of a heat pump component	2%
Manufacturing defect on the heat pump	6%	Scale deposit	2%
Failure of the electronic board	6%	Difficulty in restarting the heat pump	2%
Shut down of the heat pump	6%	Air bubbles in the system	2%
Electrical problems: Circuit breaker "trips"	6%	Heat pump overheating	2%
Insufficient domestic hot water	4%	Need to redo the drilling	2%

Field Analysis and Modeling of an In-situ Heat Pump

A centralized air-to-air type heat pump monitored for a winter period (October 2011 - February 2012) by EDF R&D shows satisfactory performance levels over the examined period; an average COP of 3.3 is obtained. The selected instrumentation was simpler to set up on this centralized installation than on a multi-split.

A simple air/air heat pump model was created in order to estimate the impact of certain parameters, such as the rated COP or the sizing, on the seasonal COP. This model was developed from the SIMFAST calculation core used at the ENERBAT department and on a simple heat pump model developed in Excel based on data from a heat pump manufacturer. In spite of strong hypotheses (particularly an identical heating season for all regions compared), the conclusions of this preliminary work show that:

- Sizing the heat pump at 100% of requirements does not lead to an energy optimum, all the more so as the base temperature at which the calculations are performed is rarely attained with this model, which is based on a thirty-year weather forecast. The standby consumption significantly

degrades the seasonal COP, particularly when heating requirements are low and spread out over time.
- The machine's load factor, particularly for newly built homes, has an impact on homes with low-load requirements; and thus the importance of proper sizing and control for this equipment type.

The causes for the lowered performance levels were combined in a family, classified according to possibilities of action: design, sizing rules, installation rules, and maintenance. The degradation of the COP is due to objective factors such as: non-optimized regulation for the on-site performance; standby consumption appreciably degrading the seasonal COP; non-optimized sizing. Additionally, there are subjective factors such as the occupant's choice of the temperature setting and in particular the comfort perception.

Specific installation and maintenance malfunction causes were explored in laboratory tests aimed at understanding the influence of low refrigerant charge on equipment performance.

Impact of Low Refrigerant Charge on Air-to-Air Heat Pump Performance

Another part of the EDF work involved the performance of air-to-air heat pumps and particularly, the causes for the lack of performance of this equipment. Evaluation on the performance differences between the field reality, and those measured in the laboratory, was carried out on air-to-air heat pumps starting in 2011 and carried through 2013.

An air-to-air, ductless, variable-speed heat pump, containing R-410A, was tested in the laboratory at the normative points of +7°C, +2°C, -7°C (44.6°F, 35.6°F, 19.4°F) for low refrigerant charge. The main objective of these tests was to demonstrate the influence of a low-refrigerant fault on the COP calculation and on the power output for similar conditions. Since the air-to-air machine has integrated controls, the investigation included comparing varied output parameters such as compressor speed (or frequency), electrical input power, condensation and evaporation temperatures, pressure (suction and discharge), and discharge temperature.

It was found that at low thermal loads the heat pump performance was affected by low refrigerant charge. For a lack of 40% of refrigerant, with respect to the rated load, the COP drops by 18% at a test condition of +7°C/20°C (44.6°F/68°F). At this condition, the impact is not as much on the compression rate or the evaporation pressure as it is on the compressor load (i.e., rotational speed). The performance degradation is weaker for lower operating temperatures (at which the COP is lower); however, the actual field required thermal load of the machine becomes greater. The rotational frequencies adopted by the machine are already very high for a load of -48% and -52% of the rated load; thus, the system could not provide the entire demand.

Prototyping of ECS Thermodynamic Water Heaters

Starting in 2009, EdF undertook in-situ field trials (in ten residential homes) of a Japanese thermodynamic ECS (Ecocute programme) water heater to ascertain what technology transfer was necessary to adapt them to the European market. The objectives of the on-site testing were to:
- adapt the machine as best as possible to "French" use of domestic hot water,
- measure the Coefficient of Performance over one year,
- participate in setting up a commercial offer,
- assess with the selected installer, the possible implementation difficulties for heat pumps,
- work on the product design by including an Eco-design dimension.

Based on the resulting operating data, an extrapolation using a weather forecast file for the Paris region resulted in an average annual COP in the order of 2.5; this value was appreciably lower than that which EDF initially identified as a desired objective. However, the most critical requirements were met, including during the winter period when the outside air temperature was approximately -11°C (12.2°F)!

The analysis of operating parameters identified the items to be improved, such as: the insulation of the tank, temperature stratification, and software management. Modifications were made to attain the ambitious objective of an average effective COP of 3, which subsequently positioned this equipment to favorably compare to other equipment options.

SWEDEN (KTH) – *A COMPREHENSIVE ANALYSIS OF FAULTS IN SWEDISH HEAT PUMP SYSTEMS*

The project 'Smart Fault Detection and Diagnosis (SFDD) for heat pump systems,' was undertaken in the energy department of the Royal Institute of Technology (KTH) in cooperation with Folksam (one of the largest insurance companies in Sweden), and also with some of the main Swedish heat pump manufacturers. The project was aimed at improving the performance of the systems while reducing maintenance times and costs.
The project was organized in several steps:

- *Literature survey*: Aimed at analyzing the available fault detection and diagnosis (FDD) techniques. Attention was placed on those techniques, which have been applied, or had influence in developing fault detection and/or diagnosis tools for heat pump systems.
- *Building a comprehensive database*: Identifying the most common faults occurring in domestic heat pump systems, which became the basis for the developing of the FDD system.
- *Design of fault detection system.*

The project objectives were:

- Build a reference database which includes the most common faults occurring in modern heat pump systems.
- Design a smart fault detection and diagnosis system at three different phases: commissioning, operation, and maintenance.

Review of FDD Techniques

The first part of the project studied the different techniques applied in FDD in general systems, and more in detail about heat pump systems. Fault detection of heat pumps can be performed with different methods. The most applied are indicated in Table ES-6.

Fault detection and diagnosis is a discipline that has been widely applied in different industries, for example in nuclear power plants, though it is still in its infancy for heat pump applications. Fault detection refers to the automatic process of determining whether a fault has occurred in the system. The next step, known as fault diagnosis, aims at determining the exact location, time, and magnitude of the fault. The aim of a fault detection and diagnosis system is to help the technician repair the fault by making the process more efficient and effective; thus, reducing repair and time costs.

Table ES-6: FDD Advantages and Disadvantages

FDD Methods	*Advantages*	*Disadvantages*
Data-driven methods	• They are well suited for problems in which the theoretical knowledge is poor or in systems where analytical models are complex and time consuming to develop. • They do not need a physical understanding of the system. • They are useful in systems where training data are easy to collect. • The computational effort required is generally manageable.	• A large number of data is needed, representing both normal and faulty operation. • The derived models cannot be used to draw conclusions beyond the range of training data. • The models are specific to the system for which they are trained.
Analytical methods	• They are based on physical principles; so, if the model is well formulated, they can provide the most accurate description of the system. • They can model both normal and faulty operation; therefore, fault conditions can be identified without having to run the system.	• They can be complex and time consuming to develop. • They can be computational intensive, therefore, not suitable for online FDD. • Models can require inputs that can be difficult to get.
Knowledge-based methods	• They are simple to develop and apply. • They can be useful when a large number of data and knowledge about failure of the system is available. • They are well suited when a deep knowledge about the system is lacking. • Their way of reasoning is transparent therefore easy to verify.	• They become more and more complicated and difficult to apply for complex system. • They are specific to a system or a process. • Their resolution can be approximate and strongly dependent on the expertise knowledge available.

Faults Database

To create an efficient FDD tool, it is fundamental to have a preliminary knowledge about faults, and about the most common faults that occur in heat pumps systems. Using the data provided by the Swedish insurance company Folksam, and the main Swedish heat pump manufacturers, statistics have been assembled that reveal the most common types of faults and problems occurring in installed Swedish residential heat pumps. The procedure of analysis was as follows:

- Data were categorized according to the different types of heat pumps (i.e., air-to-air, air-to-water, brine-to-water, exhaust air).
- Data where grouped into different categories by fault or a reason of failure.
- The most frequent and costliest faults reported to the insurance company and the heat pump manufacturers were determined.
- Meetings with heat pump manufactures were held in order to review the analysis and to investigate the root cause of the faults reported.

The Folksam database tabulated 13,993 faults and included fault data from 2001 to 2011; due to an error in the data files, data from year 2007 could not be included in the statistics. After removing wrong data, and data not including the model or the type of heat pump, the number of faults included in the analysis shrunk to 8,659. The same methodology was applied to the data coming from the heat pump manufacturers. There were about 37,000 faults field-reported to the heat pump manufacturers from beginning of 2010 to the end of 2012 (three years). These reports include the fault or complaints (e.g. about the noise in the system). The breakdown of faults, by equipment type, from the two sources are shown in Table ES-7:

Table ES-7: Fault Breakout by Equipment Type

Equipment Type	*Folksam Study*	*OEM Study*
Brine-to-water heat pump faults	41%	31%
Air-to-air heat pump faults	31%	8%
Air-to-water heat pump faults	15%	56%
Exhaust air heat pump faults	13%	5%

Table ES-8a is a summary of the results after processing the faults data provided by the insurance company, Folksam and Table ES-8b illustrates similar from the OEM-provided data.

Table ES-8a: Summary of Heat Pump Faults in the Folksam Study

	Type of Heat Pump (HP)			
	Air-to-Air HP	Air-to-Water HP	Brine-to-Water HP	Exhaust Air HP
The most common faults	Compressor (30%)	Compressor (52%)	Shuttle valve (22%)	Compressor (40%)
	Fan (19%)	Evaporator (6%)	Compressor (19%)	Control and Electronics (15%)
	Control and Electronics (13%)	Control and Electronics (4%)	Control and Electronics (14%)	Evaporator (11%)
The costliest faults	Compressor (46%)	Compressor (52%)	Compressor (49%)	Compressor (56%)
	Control and Electronics (12%)	Evaporator (6%)	Shuttle valve (9%)	Evaporator (15%)
	Fan (8%)	Refrigerant Leakage (5%)	Control and Electronics (9%)	Control and Electronics (9%)

Table ES-8b: Summary of Heat Pump Faults in the OEM Study

	Type of Heat Pump (HP)			
	Air-to-Air HP	Air-to-Water HP	Brine-to-Water HP	Exhaust Air HP
The most common faults	Fan (26%)	Pressure switch (44%)	Control and Electronics (31%)	Control and Electronics (32%)
	Control and Electronics (25%)	Control and Electronics (25%)	Shuttle valve (19%)	Shunt valve/motor (19%)
	Temperature Sensors (16%)	Temperature sensors (10%)	Liquid pumps (17%)	Temperature sensors (11%)
The costliest Faults	Control and Electronics (23%)	Pressure switch (25%)	Control and Electronics (28%)	Control and Electronics (24%)
	Refrigerant Leakage (17%)	Control and Electronics (21%)	Liquid pumps (18%)	Refrigerant leakage (17%)
	Fan (15%)	Compressor (19%)	Shuttle valve (12%)	Domestic Hot Water tank (13%)

Findings

When a good system model is available, fault detection approaches based on analytical models outperform the other cited techniques. Although some reported data were incomplete, the resulting inconsistencies were compensated for by a large amount of collected data that leads to more valuable statistical analysis. The categorized heat pump faults – based on the type of system (air-to-air, air-to-water, brine-to-water, exhaust air) and reported failure modes – highlighted the most problematic components.

- Although the results were slightly different between the Folksam and the heat pump manufacturers sources, somewhat similar results were obtained in terms of common fault identification and relative costs to correct the fault or faulty components.
- Based on the database provided by Folksam, the faults related to compressor, control and electronics, fans, and shuttle valves are the most frequent and also the costliest faults which were reported to the insurance company during the ten-year period.

SWEDEN (SP) – *OPERATION AND MAINTENANCE OF HEAT PUMPS IN APARTMENT BUILDINGS OWNED BY SMALLER PROPERTY COMPANIES*

The SP's programme of work was aimed at improving the quality of installation and maintenance of heat pump systems in Sweden. It focused on heat pumps serving apartment buildings in Sweden, where heat pumps are used mainly for space heating and domestic hot water heating. The full SP report describes three studies:

1. That part of the EFFSYS+ project "Operation and Maintenance of Refrigeration and Heat Pump Systems" focusing on heat pumps installed in Swedish apartment buildings and non-residential premises.
2. An English summary of a study focusing on experience from operation and maintenance of larger heat pumps and research in the heat pump field in Sweden during the 1980s.
3. The study "Improved Reliability of Residential Heat Pumps" made on behalf of the research fund of Länsförsäkringsbolagen, focused on heat pumps installed in Swedish single-family houses.

Operating and Maintenance of Refrigeration and Heat Pumps System

A main contribution of SP is the project "Operation and Maintenance of Refrigeration and Heat Pumps System" with the scope to maintain the installed capacity and efficiency of the heat pumps by applying maintenance methods, not to modify the installations.

The principle task was to identify the 3-to-5 most common problems with heat pump installations and suggest maintenance methods to prevent them. The first part of the project analysed conventional operation and maintenance principles and identified concepts that can be applied on heat pumps to prevent common failures. The second part in the project examined common and expensive failures of heat pumps servicing apartment buildings and non-residential premises in Sweden. An indirect expense was related to the use of back-up heating systems when the heat pumps were unable to operate properly because of failures and poor adjustments. The third part was to recommend cost-effective maintenance methods to prevent three of the identified failures.

The method used to find common failures and problems was to interview personnel working with heat pumps at property companies. A reference group of representatives from property companies, interest groups, and designers was also a part of the project. The main result of the interviews was that heat pumps themselves are seen as reliable; that is, in terms of providing space heating and domestic hot water heating. However, the main difficulties were:

- Lack of knowledge by building service personnel of proper purchasing, design and commissioning of heat pump systems; the ability to identify the requirements and to apply a heat pump system that suits the building.
- Lack of adjustment and functional control of the whole building system at heat pump start-up.
- Lack of knowledge on how to survey and control the heat pump system during operation.

It is more difficult for very small property companies (less than ten apartments and/or premises) to gather this knowledge than it is for larger companies with greater resources and more heat pumps installed in their buildings. It was therefore decided to focus on small property companies and to create a manual on operation and maintenance of heat pumps for apartment buildings. The outcome of this task was the creation of a maintenance manual for owners of heat pumps in multi-family houses. The resultant manual addresses problems associated with:

- Procurements competences – knowledge about heat pumps in the owners' purchasing group
- Heat pump system design for an existing building.
- Coordinator at design and commissioning – somebody with a responsibility of the whole building services system.
- Communication between the designer of the heat pump and the software engineer.
- Systems descriptions and manuals for operation and maintenance.
- An effective interface of the control and/or surveillance system.

Large Heat Pump – Research and Development During the 1980's in Sweden

A literature review of operational experiences of large heat pumps and research from the 1980's identified marketplace difficulties and best operating procedures for heat pump systems. But one difficulty was, and still is, to convince the owner that purchasing of operation and maintenance services will lead to economical profits.

To achieve better operational results at existing heat pump plants, five interventions for operation and maintenance were identified (and are valid still today):

1) Reducing the temperature of the supply flow from the heat pump condenser and making better use of its capacity.
2) Better tools for monitoring heat pump performance and related adjustments to achieve maximum performance.
3) Operation and maintenance readiness for maximum availability.
4) Supervisory and alarm equipment.
5) Training, particularly including practical tests, for operating personnel, service companies, consultants etc., concerning the above measures. This could help poorly performing units to achieve more profitable operation. Ideas for improving components, design, and supervisory and control equipment would also benefit from better knowledge.

It was estimated that, if the measures described above were realised on existing heat pumps at the time, an energy saving of 10 % could be achieved. It was opined that the cost for introducing the measures would be recouped by the value of one year's energy savings.

Improved Reliability of Swedish Residential Heat Pumps

Heat pump heating systems are common in Swedish single-family houses. Many owners are pleased with their installation, but statistics show that a certain number of heat pumps break every year, resulting in high costs for both insurance companies and owners. On behalf of the research fund of Länsförsäkringsbolagen, SP studied the cause of the most common failures for residential heat pumps. The objective of the study was to identify measures to reduce the number of failures, (i.e., improving the reliability of heat pumps). The approach used was analysis of (1) publicly-available information on failure and sale statistics, and (2) interviews with heat pump manufacturers, installers, service representatives and insurance assessors. The interviews resulted in categorizing the most common failures based on whether they:

- could have been prevented by better operation and maintenance,
- were caused by a poorly performed installation,
- could have been prevented if certain parameters had been measured, recorded and followed up,
- were due to poor quality of components or systems.

The interviews indicated that failures on residential heat pumps are probably often caused by poor installation, neglected maintenance and monitoring, poor quality of standard components, or components that are used outside their declared operating range. Furthermore, many of the common failures can be caused by several of these categories and different types of measures must be taken to improve system reliability. Current focus is mainly on the energy efficiency of the heat pump and questions regarding lifetime and reliability may be neglected at purchasing. In order to improve the reliability of heat pumps installed in single-family houses, and reduce the costs for the insurance companies, there must be incentives for people to choose a product with good quality while engaging the best installers and service technicians. Many failures and costs are due to lack of knowledge. In addition, the heat pump owner must be informed to maintain the heat pump to ensure it functions as well as possible. In order to create these incentives, communication and cooperation between the stakeholders in the heat pump industry is needed.

UK – *IMPROVEMENTS TO DESIGN AND INSTALLATION STANDARDS OF DOMESTIC HEAT PUMP SYSTEMS*

The UK's programme of work was aimed at improving the quality of installation and maintenance of heat pump systems in the UK. It focused on the residential sector, where heat pumps are used for space and water heating, but are typically not used for cooling. The work comprised five main projects:

1) A literature review of issues relating to installation and maintenance, including, reviews of consumer protection, training courses, insurance, standards and field trials.
2) Analysing the performance of real systems and making interventions to improve performance in field trials.
3) Laboratory experiments to analyse the effect on heat pumps from cycling, water buffer tanks and the efficiency of hot water tanks.
4) Improvements to standards for design and installation.
5) Dissemination of improvements to standards through UK-wide roadshows for heat pump installers and social housing providers.

The comprehensive field trials carried out by the Energy Saving Trust, and analysed by DECC, have resulted in an update in the UK's standards for design and installation of heat pumps. DECC subsidised a series of road shows to inform installers and social housing landlords about these revisions.

Field Trails

In 2007, the Department of Energy and Climate Change (DECC) identified the lack of reliable field trial data as a major problem for the nascent UK heat pump industry. Working with both the public and private sectors, EST, funded by DECC and the UK heat pump industry, set up the first large scale field trial of domestic heat pumps in the UK. Eighty-three (83) heat pumps, located throughout the UK were monitored for one year (April 2009 to March 2010). Out of these, reliable data were obtained for 71 sites. The most common configuration was a ground-source heat pump supplying radiators and domestic hot water.

The overall system efficiencies were low; between 1.2 and 2.2 COPs for the air-source heat pumps, and between 1.8 and 3.4 for the ground-source heat pumps. The mean values were 1.8 ± 0.06 for the air-source heat pumps, and 2.39 ± 0.06 for the ground-source heat pumps. Six additional air-source heat pumps, selected by a manufacturer instead of being selected at random, were also monitored and showed better system efficiencies (between 2.4 and 3.0), demonstrating that good performance can be achieved with heat pumps that are designed, installed and configured correctly. [Note: It should be remembered the "system efficiency", as defined here, is strongly influenced by the householders' use of domestic hot water and by the heat losses from the domestic hot water cylinder.]

For this investigation, detailed analysis was carried out on a site-by-site basis to ascertain the reasons for low performance. The main design and installation issues were:

- Over-use of the electric backup heat. This may be caused by under-sizing of the heat pump or poor control strategies.
- Under-sizing of the ground loop.
- High central heating flow temperatures, leading to poor performance.
- High consumption of electricity by circulation pumps on the central heating side.
- Incorrectly sized domestic hot water tanks, leading to overuse of the electric immersion, or alternatively, wasting energy if more water is heated than the householder can use.
- Excessive use of defrosting (for air-source heat pumps).

As a result of these findings, two steps were taken. First, a number of heat pumps were selected for interventions to improve performance, and monitored for an additional year. Secondly, the Microgeneration Certification Standard MIS 3005 was updated to address these issues.

For the resultant Phase II field trials, 38 heat pumps were selected for interventions to improve performance, plus six OEM heat pumps were added to the sample. The interventions were classified into four categories:

1) Major: requiring the input of a heat pump specialist (e.g. replacing the heat pump, repairing the ground loop, refilling the refrigerant) (12 heat pumps).
2) Medium: requiring the input of a plumber (e.g. refilling the ground loop, adding a buffer tank, replacing a domestic hot water tank, replacing flap valves, replacing radiators, replacing circulation pumps) (9 heat pumps).
3) Minor: altering controls, disabling the auxiliary direct electric heater, switching off unnecessary circulation pumps (11 heat pumps).
4) None (6 heat pumps).

Figure ES-1 below compares system efficiencies that resulted from Phases I and II of the trial. These have been corrected to take account of the difference in degree-day heating requirements for the two phases of monitoring (winter 2009-10 was colder than winter 2011-12).

Nine (9) out of twelve (12) heat pumps with major modifications showed improvements in system efficiency from Phase I to Phase II. Of these, eight showed an improvement of ≥0.3. For two of the three sites for which system efficiency was reduced in Phase II, the reason was increased immersion consumption for domestic hot water.

Eight (8) out of nine (9) heat pumps that received medium interventions showed improved system efficiencies in Phase II. In five cases, the improvement was ≥0.3. The greatest improvement, from 2.31 to 3.29, was at a site where software was altered to reduce the use of the auxiliary heater. The remaining heat pump in this category developed a fault during the second phase.

These substantial improvements in system efficiency indicate that a majority of the problems were rectified.

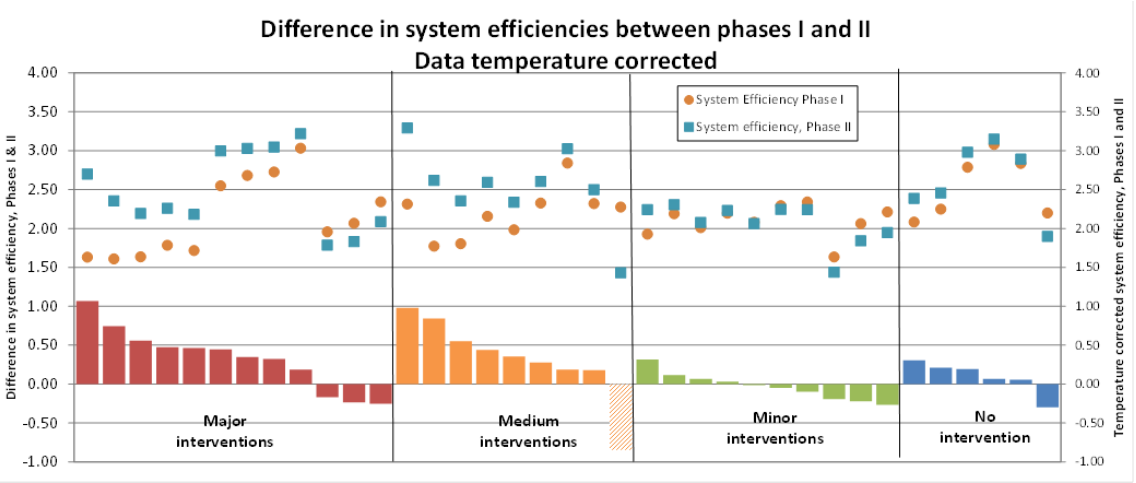


Figure ES-1: Difference in Temperature-Corrected System Efficiencies Between Phases I and II of the Energy Saving Trust Heat Pump Field Trials

Laboratory Experiments – Effect of Cycling on Heat Pump Efficiency

Heat pumps with fixed-speed compressors may cycle rapidly (i.e., short cycle) in warm weather, when the house does not require the full heat output of the heat pump. Heat pumps with variable-speed compressors are less likely to be affected. Short cycling can adversely affect the lifetime of the compressor and other components and may also reduce efficiency. In addition, the spike in current draw-off when the compressor starts up can have adverse effects on the grid. As a result of laboratory experiments, it was found that:

- *For air-sourced heat pumps*: As the operating runtime fraction decreases, the heat pump efficiency goes down dramatically.

- *For ground-source heat pumps*: For short runtimes, ground-source heat pump efficiency is reduced; but, the effect is less marked than for air-source heat pumps. (This is because the ground temperature does not cool as rapidly when the runtime is short.)

One way of reducing cycling is to include a buffer tank in the central heating circuit. This may be a two-pipe system (i.e. situated either in the flow or return pipes), or a four-pipe system (through which both the flow and return pipes flow). There are two, counter-prevailing principles at work when buffer tanks are incorporated into the water circuit:

1) The heat pump is more efficient when the return temperature is lower (i.e., the case with no buffer tanks).
2) Yet, inclusion of a buffer tank results in a larger hysteresis, which means that the heat pump will cycle less.

Experiments were carried out using a large heating volume (7 radiators), a medium heating volume (4 radiators) and a small heating volume (1 radiator). For the 7 and 4 radiator cases, ground-source heat pumps were relatively unaffected by the choice of configuration (2-3% performance change when the return temperature changed from 40 to 41°C (104 - 105°F)). While the performance of air-source heat pumps was improved when a buffer tank (of any configuration) was inserted, or when the hysteresis was increased for the case of no buffer tank. It is concluded that buffer tanks are not necessary in all heat pump circuits. Although they may be useful either when the volume of water heated is insufficient to avoid cycling or, alternatively, when heat pumps are operated on a cheap tariff and heat storage is necessary. If installed, a 2-pipe configuration in the return pipe is the preferred arrangement.

Additional experiments were undertaken to determine the most efficient way to heat domestic hot water with a heat pump, using a range of cylinder sizes, storage temperatures, heating patterns, sterilization patterns and tapping patterns. The study investigated the heat losses from the cylinder as a function of storage temperature, at low, medium, and high consumption levels. In all cases, cylinder losses increased by at least 20 percentage points as the storage temperature increases from 45˚C to 55˚C (113°F to 131°F). This would be compounded by a reduced efficiency of the heat pump when operating at higher temperatures (from the Heat Emitter Guide, the seasonal performance factor – SPF4 – for an air-source heat pump would be reduced from 3.0 at a flow temperature of 45˚C (113°F) to 2.4 at 55˚C (131°F)). For these reasons, the study recommends:

- For optimum efficiency, the storage temperature should be slightly lower than the flow temperature of the heat pump.
- Where usage is low, the cylinder should not be heated continuously.
- Weekly sterilization to 60˚C (140°F) should be carried out during periods of low use (e.g., at night).
- Heat pumps typically heat domestic hot water with a flow temperature of 45-50˚C (113°F to 131°F) and flow rate of 0.4 ls^{-1} (6.3 gpm). In order to transfer the heat to the water, a coil with a large surface area is required.
- Current kW ratings for heating coils in domestic hot water cylinders are calculated according to standard BS EN 12897, which applies to an 80˚C (176°F) flow temperature and a flow rate of 0.25 ls^{-1} (4 gpm). This is not appropriate for heat pumps and therefore a new system of rating is required.

Changes to UK Standards

Following the findings of the first phase of the Energy Saving Trust Field Trials, DECC, the Energy Saving Trust and the UK heat pump industry worked together to improve the design and installation standards for small scale heat pumps (< 45 kWth). As a result, the MCS MIS 3005 Issue 3.1 Standard came into force in March 2012. The principal differences between this standard and its predecessor are:

- *New sizing requirements*: A heat pump must be able to supply the steady-state space heating requirement of the house for 99% of the hours of a typical year without recourse to the electric auxiliary heater. In order to size the heat pump correctly, the installer must carry out a room-by-room calculation of heat loss. Additionally, for air-source heat pumps, the reduction in capacity as the temperature falls must be taken into account.

- *New guidance on heat emitter design*: The Heat Emitter Guide for Domestic Heat Pumps was revised to allow the installer to size and design the heat emitters correctly. It assigns an efficiency rating based on the flow temperature at the design ambient temperature for the location. As part of the handover process, the installer must show the householder this efficiency rating, which is displayed as 1 to 6 stars (6 being the most efficient). For good performance, low flow temperatures are required; guidance is given on the appropriate spacing of pipes in underfloor systems and the appropriate sizing of radiators.
- *New guidance on designing ground loops and boreholes:* The Ground Heat Exchanger Look-up Tables specify the power that can be extracted per meter of ground loop (or borehole) as a function of full time equivalent run hours, mean annual ground temperature, and ground conductivity.

The new standards also provide guidance on the sizing of domestic hot water cylinders and re-iterated the need to insulate all pipes and tanks thoroughly, and to sterilize the domestic hot water cylinder once per week to protect against Legionella and other bacteria.

US – SENSITIVITY ANALYSIS OF INSTALLATION FAULTS ON HEAT PUMP PERFORMANCE

The US investigation explores the impact that installation errors (e.g., improper refrigerant charge, incorrect air flow, oversized equipment, leaky ducts faults, etc.) have on the performance of a single-speed, 8.8 kW (2.5-ton), ducted, split-system heat pump having rated seasonal cooling and heating efficiencies of 3.81 SPFc and 2.26 SPFh, respectively (13 SEER and 7.7 HSPF) in a single-family, single-zone house. The laboratory/modeling project combined building, equipment, and climate effects in a comprehensive evaluation of the impact of installation faults on heat pump annual energy consumption via seasonal simulations of the house/heat pump system. Addressed was the significance of specific operational deviations (faults), whether the deviations (when combined) have an additive effect on equipment performance, and whether some deviations are more greatly impacted by climatic conditions than others. The parameters for the effort (see Table ES-9) were based on the requirements in the ANSI/ACCA 5 QI – 2010 Standard (*HVAC Quality Installation Specification*) with two additional items added as a result of the literature survey: (1) excessive liquid line refrigerant subcooling and (2) undersized thermal expansion valve (TXV).

The results from the critical literature survey are included in the country report in three sections; the identified publications are related to:

- air conditioner and heat pump installation and maintenance issues,
- heat pump oversizing/undersizing and cycling loses,
- heat pump fault detection and diagnostics.

Table ES-9: Parameters included in the investigation

Effects	***Varied parameters***
Building subsystem	– Duct leakage (unconditioned space)
Residential split, air-to-air heat pump equipped with a thermostatic expansion valve (TXV)	– Equipment sizing – Indoor coil airflow – Refrigerant charge – Presence of non-condensable gases – Electrical voltage – TXV undersizing – Excessive liquid line subcooling
U.S. Climates (cooling and heating)	– Hot and humid – Hot and dry – Mixed – Heating dominated – Very cold

Single-family houses (the structures representative for the geographic area)	– House on a slab (air ducts in unconditioned attic) – House with a basement (ducts in conditioned basement)

Laboratory Analysis

The laboratory analysis, using the same heat pump and test apparatus, expanded on prior work undertaken at NIST related to heat pump faults. Specifically added to the new work are: improper electric line voltage, TXV undersizing, and improper liquid line subcooling. The goal of the laboratory work was to develop the correlations that characterize heat pump performance that operate with varied faults. Table ES-10 lists nine (9) studied faults, including their definitions and studied ranges. The first seven (7) were included in the laboratory testing and subsequent modeling, the last two (2) faults were modeled only.

Table ES-10: Studied faults, definitions, and fault ranges

Fault name	Symbol	Definition of fault level	Fault Levels (%)
Improper indoor airflow rate	AF	% above or below correct airflow rate	CM: -36, -15, +7, +28 HM: -36, -15, +7, +28
Refrigerant undercharge	UC	% mass below correct (no-fault) charge	CM: -10, -20, -30 HM: -10, -20, -30
Refrigerant overcharge	OC	% mass above correct (no-fault) charge	CM: +10, +20, +30 HM: +10, +20, +30
Excessive liquid line refrigerant subcooling (indication of improper refrigerant charge)	SC	% above the no-fault subcooling value	CM: +100, +200 HM: none
Presence of non-condensable gases	NC	% of pressure in evacuated indoor section and line set, due to non-condensable gas, with respect to atmospheric pressure	CM: +10, +20 HM: +10, +20
Improper electric line voltage	VOL	% above or below 208 V	CM: -8, +8, +25 HM: -8, +8, +25
TXV undersizing	TXV	% below the nominal cooling capacity	CM: -60, -40, -20 HM: none
Duct leakage	DUCT	% of total equipment airflow that leaks out of the duct distribution system (60% supply leakage, 40% return leakage).	CM: +0, +10, +20, +30, +40, +50 HM: +0, +10, +20, +30, +40, +50
Heat pump sizing	SIZ	% above or below optimum heat pump capacity	CM: -20, +25, +50, +75, +100 HM: -20, +25, +50, +75, +100
Notes: CM = Cooling Mode HM = Heating Mode			

Simulations of Building / Heat Pump Systems with Installation Faults

The full report details simulations of annual energy consumption (combined heating and cooling) of a heat pump operating under various levels of different installation faults. The simulations focused strictly on performance issues of the house/heat pump systems related to heat pump capacity and heat pump energy consumption; heating and cooling energy consumption is delineated. The effect of the installation faults on occupant comfort was not the main focus of the study, and the effort did not seek to quantify any impacts on indoor air quality or noise generation (e.g., airflow noise from air moving through restricted ducts). Additionally, the study did not address the effects that

installation faults have on equipment reliability/robustness (number of starts/stops, etc.), maintainability (e.g., access issues), or costs of initial installation and ongoing maintenance.

Impact of Single Installation Faults on Heat Pump Performance

Table ES-11 shows representative examples of annual energy used by a heat pump installed with different installation faults. The increase in energy consumption, for each fault type, is presented on a relative base as an isolated fault on an otherwise fault-free installed system. The table data was extracted from simulation results presented in the full U.S. report. It is anticipated that the selected levels of individual faults reflect an installation condition which might not be noticed by a poorly-trained, or inattentive, technician.

Table ES-11: Annual energy use by a heat pump resulting from single-fault installation, referenced to a fault-free installation

Fault type	***Fault level***	***Relative energy use (%)*** ***[100 is baseline]***								
		Slab-on-grade house installation (Air ducts located in unconditioned attic)					Basement house installation (Air ducts in conditioned basement)			
	%	HOU	LV	WAS	CHI	MIN	LV	WAS	CHI	MIN
AF	- 36	112	113	114	113	111	111	112	112	110
UC	- 30	121	122	123	120	117	120	121	119	117
OC	+ 30	110	110	114	113	112	111	116	115	113
SC	+ 200	118	116	119	118	116	118	120	120	119
NC	+ 10	102	102	101	101	101	102	101	101	101
VOL	+ 8	102	101	102	102	101	101	102	102	102
TXV	- 40	114	110	107	105	107	109	105	103	102
DUCT	+ 30	118	117	124	126	126	100*	100*	100*	100*
SIZ (2)	+ 50	115	113	105	101	99	114	108	104	102
	U.S. cities included in the study: **HOU** → Houston, TX **LV** → Las Vegas, NV **WAS** → Washington, DC **CHI** → Chicago, IL **MIN** → Minneapolis * It is assumed that duct leakage into basements have no energy impact (inside thermal boundary) (2) Oversize scenario (2) described in Section 5.5.5 of the US report.									

Tables ES-11 indicates that there are no drastic differences in the effect of installation faults on energy use in a slab-on-grade house and a basement house, except for the duct leakage fault (DUCT). For the slab-on-grade house, duct leakage has the potential to result in a higher increase in energy use than any other fault. The impact of this fault is higher for the heating dominated climate (Chicago and Minneapolis, 26 %) than for the cooling dominated climate (Houston, 18 %). Obviously, duct leakage will also result in some increase of energy use for the basement house; however, the modeling approach employed could not discern this increase.

The next most influential faults were refrigerant undercharge (UC), refrigerant overcharge (OC), and improper airflow across the indoor coil (AF). For the 30% undercharge fault (UC) level, the energy use increase is of the order of 20%, irrespective of the climate and building type. Refrigerant overcharge (OC) can also result in a significant increase in energy use, 10 - 16% at the 30% overcharge fault level. Improper indoor airflow (AF) can affect similar performance degradation. [Note: Excessive refrigerant subcooling (SC) correlates to refrigerant overcharge (OC); 100% subcooling is approximately equivalent to 20% refrigerant overcharge.]

Equipping a house with an oversized heat pump (SIZ) has a small effect if the air duct is oversized accordingly (which may be the case with a new construction). However, if the air duct is undersized or otherwise too restrictive, and the nominal indoor airflow is maintained by adjusting the fan speed (scenario 2), 15% increase in energy use for the house in Houston is predicted. The full report notes that there are latent issues associated with varied fault conditions and that the number of hours at or greater than an indoor relative humidity (RH) of 55% is dramatically impacted.

The undersized cooling TXV fault (TXV) also has the potential to significantly increase the energy use. The effect of this fault will be most pronounced in localities with a high number of cooling mode operating hours. The cooling mode TXV, undersized by 40%, results in 14% more energy consumption in Houston as compared to 5% in Chicago.

The impact of the remaining faults – non-condensables (NC) and improper voltage (VOL) – is under 4%. The non-condensables and improper voltage faults, however, represent a substantial risk for durability of equipment and are very important to be diagnosed during a heat pump installation.

Impact of Dual Installation Faults on Heat Pump Performance

The combination of two faults, A and B, were considered in the following four combinations as listed in Table ES-12 below:

Table ES-12: Combinations of studied faults

Fault combination case	Level of fault A	Level of fault B
a	moderate	moderate
b	moderate	worst
c	worst	moderate
d	worst	worst

The 'moderate level' is the value at the middle of the range, while the 'worst level' is the highest (or lowest) probable level of the fault value (see Table ES-10). Table ES-13 presents specific fault sets for the combined annual heating and cooling cases (sets 1 – 11) and cooling only case (sets 12 – 14). The furthest right-hand column in the table shows the effect of the combined faults on energy use: the faults' effect on increased energy consumption may be additive (A+B), less than additive (<A+B), or greater than additive (>A+B).

Table ES-13: Dual fault sets considered in annual simulations (heating and cooling) and their approximate collective effect on annual energy use

Multiple fault set #	Fault A (moderate & worst level) [(a)]	Fault B (moderate & worst level) [(a)]	Effect on Energy Use
1	Duct leakage (20%, 40%)	Oversize[(b)] (25%, 50%)	A + B
2	Duct leakage (20%, 40%)	Indoor coil airflow (-15%, -36%)	< A + B
3	Duct leakage (20%, 40%)	UC (-15%, -30%)	A + B or > A+ B
4	Duct leakage (20%, 40%)	OC (15%, 30%)	A + B
5	Duct leakage (20%, 40%)	NC (10%, 20%)	A + B
6	Oversize [(b)] (25%, 50%)	UC (-15%, -30%)	A + B
7	Oversize[(b)] (25%, 50%)	OC (15%, 30%)	A + B
8	Oversize (25%, 50%)	NC (10%, 20%)	A + B
9	Indoor coil airflow (-15%, -36%)	UC (-15%, -30%)	A + B
10	Indoor coil airflow (-15%, -36%)	OC (15%, 30%)	< A + B
11	Indoor coil airflow (-15%, -36%)	NC (10%, 20%)	< A + B
12	Duct leakage (20%, 40%)	TXV undersizing[(c)] (-20%, -60%)	A + B
13	Oversize (25%, 50%)	TXV undersizing[(c)] (-20%, -60%)	A + B
14	Indoor coil airflow (-15%, -36%)	TXV undersizing[(c)] (-20%, -60%)	< A + B

Notes:
(a) – Moderate = mid-level value, worst = lowest/highest level value.
(b) – Oversize scenario (2) was selected because it covers the prevalent field bias (undersized ducts).
(c) – TXV undersizing only impacts cooling; faults listed as Fault A exist in cooling and heating (applies to Sets 12, 13, 14).

Faults effects may be additive (A+B), less than additive (<A+B), or greater than additive (>A+B).

Simulations were performed for 14 dual fault sets (refer to Table ES-13), with four (4) fault combinations per set, in the 9 house/climate combinations for a total of 504 runs. The relative impact on annual energy use is shown in Figure ES-2 and Figure ES-3.

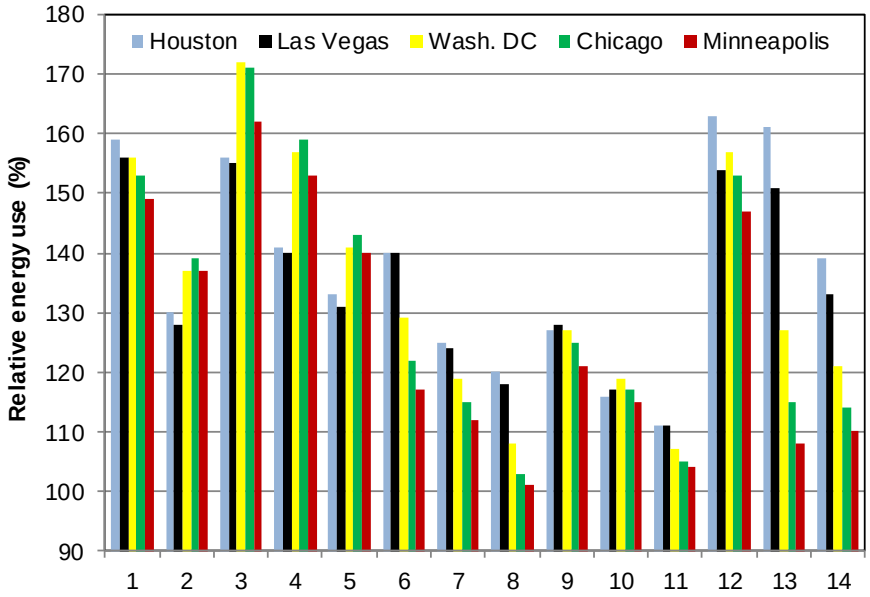


Figure ES-2: Annual energy use for ducts located in unconditioned attics with 14 dual-faults relative to the energy use for the houses with fault-free installations (Faults defined in Table ES-13; Table ES-12 case d, worst level for both faults; 100 is baseline).

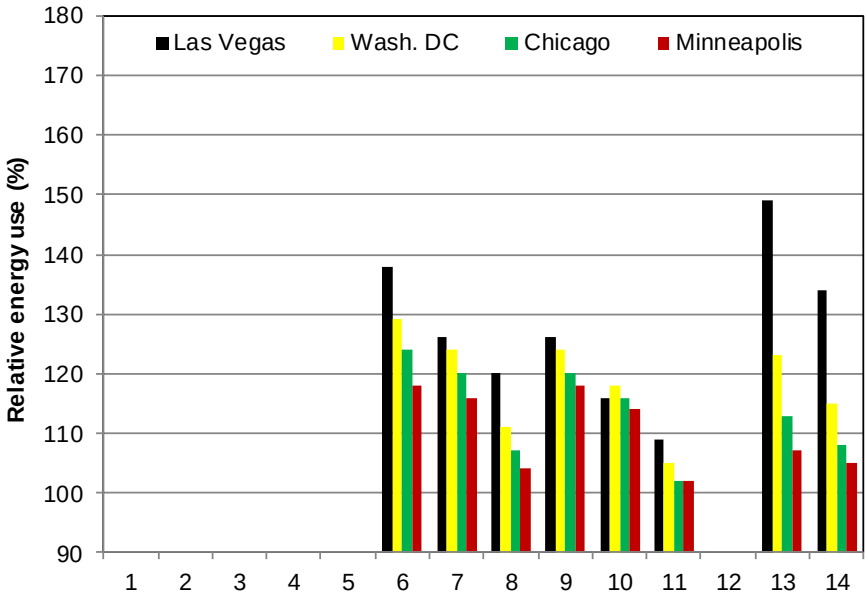


Figure ES-3: Annual energy use for ducts located in conditioned basements with 8 dual-fault installations referenced to the energy use for the houses with fault-free installations (Faults defined in Table ES-13, Table ES-12 case d; worst level for both faults; the omitted dual faults involve duct leakage, which was not considered in houses with basement; 100 is baseline).

Effects of Triple Faults

Triple faults were not simulated in this study. Nevertheless, the occurrence of three simultaneous faults is plausible, particularly for the most common faults such as refrigerant undercharge, improper indoor airflow, or duct leakage. The report notes that it is reasonable to assume that the effect of a triple fault will be as least as high as that of any of the possible three fault pairs considered individually; however, the effect of the third fault can increase the effect of the other two faults in an additive manner.

Findings

Extensive simulations of house/heat pump systems in five U.S. climatic zones lead to the following conclusions:

- The study found that duct leakage, refrigerant undercharge, oversized heat pump with undersized ductwork, low indoor airflow due to undersized ductwork, and refrigerant overcharge have the most potential for causing significant performance degradation and increased annual energy consumption.
- Effect of different installation faults on annual energy use is similar for a slab-on-grade house (ducts located in the unconditioned attic) and a basement house (ducts located in the semi-conditioned basement), except for the duct leakage fault.
- Effect of different installation faults is similar in different climates except for the following cases:
  - Duct leakage: significant increase in the indoor relative humidity for an installation in a hot and humid climate.
  - Heat pump oversizing with undersized air ducts: in heating-dominated climates, heat pump oversizing reduces the use of backup heat, which may compensate for the increased indoor fan energy use associated with overcoming the higher external static pressure.
  - Undersized cooling mode TXV: little effect in heating-dominated climates, while a significant increase of energy use is possible in cooling-dominated climates.
- The report notes that a significant increase in annual energy use can be caused by lowering the thermostat in the cooling mode to improve indoor comfort in cases of excessive indoor humidity levels. As an example, for Houston, TX, lowering the thermostat setting by 1.1°C (2.0°F) increased the annual energy use by 20%, and the energy use increase rate is even higher due to further lowering the setting (the effect is not linear).
- The effect of simultaneous faults can be additive (e.g., duct leakage and non-condensable gases), little changed relative to the single fault condition (e.g., low indoor airflow and refrigerant undercharge), or well-beyond additive (duct leakage and refrigerant undercharge).
- The report authors contend that the laboratory and modeling results from this fault analysis on a 8.8 kW (2.5 ton) heat pump are representative of all unitary equipment, including commercial split-systems and single package units (e.g., roof top units).

Annex 36 Future Research Needs

As can be observed in the above summary, the type of investigations undertaken by the participating countries were very different, and the studied equipment applications were broad. However, there are a number of cross-cutting needs that universally accrue to all and are in need of future exploration:

1) Determine the effect of simultaneous, multiple faults / deficiencies (at different severity levels) through rigorous laboratory measurements.
2) Relatedly, there is need to collect and analyze in field fault data (type, frequency, degree of severity) to quantify the impact that heat pump inefficiency on a national level.
3) Determine the effect of installation / operational faults on energy use of energy-efficient heat pumps installed in energy-efficient buildings; combination of laboratory measurements and building/heat pump simulations.
4) Development of open communication protocols for heat pump commissioning and re-commissioning; entails a common set of error codes as well as a universal access port/method to retrieve the error codes.
5) Effort is needed to improve cold-climate efficiency and reliability of heat pump technologies.
6) Need to quantify the effect that installation and maintenance deficiencies have on occupant comfort / health, equipment reliability / robustness / maintainability, and operational / maintenance costs.
7) Reliable, cost-effective, in-situ means to measure heat pump capacity (delivered kW/hr or BTU/hr) versus operating efficiency (delivered capacity divided by total energy input).
8) Investigate methods for controlling bacteria such as Legionella in domestic hot water applications.
9) Information on installation rules and maintenance procedures needs to be created and provided in a manner that installers and service personnel can easily understand and implement.
10) Approaches to improve communications and cooperation among the various stakeholders (manufacturers, distributors, customers, insurance companies, efficiency programs, etc.) to ensure that incentives, encouragements, and rewards result in quality equipment being purchased with installations and maintenance undertaken by trained, qualified service personal.

PART 2: COUNTRY REPORTS

This final Annex 36 report is a compilation of individual efforts undertaken by the four participating countries. Five individual reports, noted below, are incorporated into this volume.

Annex 36 Participants	Research Entity	Report Title	Author(s)	Focus Area
France	EdF R&D	***Focused Projects on Heat Pump Space Heating and Water Heating***	J. M. Lebreton	Space heating and water heating applications.
	Work Undertaken *Field*: Customer feedback survey on heat pump system installations, maintenance, and after-sales service. *Lab*: Water heating performance tests on sensitivity parameters and analysis.			
Sweden	Royal Institute of Technology (KTH)	***A Comprehensive Analysis of Faults in Swedish Heat Pump Systems***	Erica Roccatello Hatef Madani	Heat pumps for multi-family and commercial buildings. Air-to-air, air-to-water, geothermal and exhaust heat pumps.
	SP Technical Research Institute of Sweden	***Operation and Maintenance of Heat Pumps in Apartment Buildings Owned by Smaller Property Companies***	Lennart Rolfsman Kristin Larsson	
	Work Undertaken *Field*: Literature review of operation and maintenance for larger heat pumps. Investigations and statistical analysis of 22000 heat pump failures. *Modeling/Lab*: Determination of failure modes and analysis of found failures and failure statistics.			
United Kingdom	UK Department of Energy & Climate Change	***Improvements to Design and Installation Standards of Domestic Heat Pump Systems***	Penny Dunbabin	Domestic heat pump systems (air-to-water/ground-to-water) and hot water tanks
	Work Undertaken *Field*: Monitoring of 84 domestic heat pump systems and targeting interventions to improve performance. *Lab*: Investigate the impact of cycling and buffer tanks on heat pump performance and assessing the efficiency of domestic hot water tanks with heat pumps.			
United States (Operating Agent)	National Institute of Standards and Technology	***Sensitivity Analysis of Installation Faults on Heat Pump Performance***	Piotr Domanski Vance Payne	Air-to-air residential heat pumps installed in residential applications (cooling and heating).
	CDH Energy Corporation		Hugh Henderson	
	Work Undertaken *Lab*: Cooling and heating tests with imposed faults to develop correlations for heat pump performance degradation due to faults. *Modeling*: Seasonal analysis modeling to evaluate the effect of installation faults on heat pump seasonal energy consumption. Includes effects of different building types (slab vs. basement foundations, etc.) and climates in the impact assessment of various faults on heat pump performance.			

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COUNTRY REPORT:
FRENCH CONTRIBUTION TO THE IEA HEAT PUMP PROGRAMME ANNEX 36 ON QUALITY OF INSTALLATION AND MAINTENANCE OF HEAT PUMPS

Focused Projects on Heat Pump Space Heating and Water Heating

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Abstract

Électricité de France (EDF) investigated space heating and water heating applications for residential applications; representative of several different operating conditions in France. For the Annex 36 effort, the focus was on:

- In-field performance of air-to-air heat pumps,
- Lab and field investigations of a thermodynamuic water heater,
- Performance testing of an air-to-air heat pump in part-load mode.

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1 INTRODUCTION / SCOPE

In answer to the energy efficiency in buildings and to offer eco-efficient solutions to its customers, the EDF group supports the technical development of efficient and low carbon electrical solutions, in the existing and new buildings. One of the issues is to promote the more efficient systems for space heating and sanitary hot water, using renewable energies.

Literature surveys show that although the comfort provided by the heat pumps is considered to be good, one fourth of heat pump owners had encountered technical incidents or failures on their heat pump.

The possible causes of these malfunctions have to be found and studied and, as a partner of the Annex 36, we chose to report on the following topics:

- Real performance of air to air heat pumps (sizing, installation, maintenance, etc.);
- Feedback of measurement campaigns on thermodynamic water heaters;
- Performance tests of air to air heat pumps in part-load mode.

2 LITERATURE SURVEY

2.1 Context

The residential thermodynamic heating market is developing rapidly and the reliability criterion is a major asset. The facilities concerned by this market are the air, water and geothermal systems.

2.2 Objective

The objective of this survey is to obtain the satisfaction level of the user of these systems based on the following 3 main lines:

- Heating system by heat pump
- Maintenance of the installation
- Repair process

2.3 Methodology

Feedback was collected from 202 owners of houses equipped with a thermodynamic heating system. Heat pumps were installed between January 2005 and October 2008 in four geographical regions: PACA, Midi-Pyrénées, Bretagne and north-eastern France.

The information was collected in June and July 2009 based on telephone interviews.

2.4 Results

2.4.1 Sample Characteristics

Within the sample retained, the distribution of systems is per Table FR-1:

Table FR-1: System Distribution

Type of Heat pump	Distribution (%)
Air/water (heating only)	26
Air/water (additional heating)	28
Air/air	23
Geothermal	22

Among the air/water heat pumps, almost two-thirds (2/3) are outdoor single piece. In most cases the heat pump manages the heating of the entire accommodation (94%), a little less for the air/air heat pumps (85%).

As far as air/water and geothermal heat pumps are concerned, the emitters are mainly hot water radiators (71%). Even so, the heating floors represent one fourth of the installations.

In one third of cases, the heat pumps produce domestic hot water in addition to heating. One-third of heat pumps also provide air conditioning, but heat pumps meeting all 3 uses (heating, air conditioning and ECS) are rare.

The average reported price of a heat pump installation over the entire sample is 13500€.

2.4.2 Maintenance

38% of heat pump owners have a maintenance contract.

Table FR-2: Frequency of Maintenance Inspection

Twice a year	5%
Once a year	89%
Once every two years	1%
As and when required	1%
No opinion	3%

Usually the maintenance of heat pumps is carried out once a year.

A guarantee of the proper operation of the installation and peace of mind for the customer are the main motives for signing a maintenance contract.

Maintenance contracts come in a wide range of prices ranging from 80€ to 350€ with an average value of 144€. On the other hand, there is no significant difference between the types of heat pumps.

2.4.3 Incidents Encountered

One-fourth of heat pump owners have encountered technical incidents or failures on their heat pump. In most cases, it is the heating function that is affected by the incident. In two-thirds of the cases, the incident causes complete interruption of the system. In more than half the cases, the incident is regarded as “minor” by the customer.

Table FR-3: Occurrence of Incidents

Water or liquid leak	16%		Filter clogged, obstructed	4%
Installation error, electrical connection error	12%		Circulator failure	2%
Refrigerant leak	8%		Compressor failure	2%
Adjustment problem	8%		Fan failure	2%
Frost, ice on fan	8%		Failure of a heat pump component	2%
Manufacturing defect on the heat pump	6%		Scale deposit	2%
Failure of the electronic board	6%		Difficulty in restarting the heat pump	2%
Shut down of the heat pump	6%		Air bubbles in the system	2%
Electrical problems: Circuit breaker “trips”	6%		Heat pump overheating	2%
Insufficient ECS	4%		Need to redo the drilling	2%

2.4.4 Satisfaction

The overall satisfaction with regard to heat pumps is very good, as is the comfort provided. The satisfaction with regard to the use of heat pumps is very good; for most, the control, programming and settings do not seem to pose any problem.

On the subject of comfort, the dissatisfaction is significant with regard to the indoor noise level and even more with regard to the outdoor noise level.

The installation cost is unquestionably the subject of recurring criticism to add up to a total of 18% dissatisfied users.

2.4.5 Desired Improvements

The main improvements expected by the customers concern:

- Reduction in the installation cost,
- Reduction in the noise level,
- Comfort control,
- More explanations from the installers,
- Improvement of efficiencies for energy saving or for comfort.

2.5 Conclusions

The main conclusions that emerge from this survey are:

- One-third of heat pump owners feel that a maintenance contract is not useful.
- Finally, only 38% of heat pump owners have a maintenance contract, which is most often contracted at the time of installation. The average value of a maintenance contract is 144€.
- The comfort provided by the heat pumps is considered to be good, the weak points are marginal and the noise level is the most important.
    - There is a rather large number of incidents (one-fourth of heat pump owners experienced problems)
    - Apart from the investment cost, the consumption and maintenance costs are affordable.
    - The main improvements expected by customers concern the reduction in the noise level and comfort control.

3 MODELLING AND / OR LAB EXPERIMENT

3.1 Actual performance levels of air/air heat pumps: Identifying differences between the laboratory performance and the performance perceived by the users.

3.1.1 Context

The market of heat pumps as home heating equipment is expanding in France (and in other European countries). The air/air systems market is very dynamic, the majority of sales concern the mono- or multi-split systems particularly for the renovation of electrically heated homes; only a very small part of mono-splits is centrally installed, i.e. on plenum or ducts and above all in new apartments.

The performance levels of heat pumps are generally provided in the manufacturer’s catalogues. They may also be available in the list of referenced, certified or labeled products. These various certifications make it possible to compare products but do not guarantee an annual performance or minimum monetary savings as the actual performance will particularly depend on the insulation of the house, the climate, the sizing of the equipment and the behavior of users.

Thus, various measurement campaigns were carried out in France. However, although it is quite easy to measure the quantity of heat produced on a water system (energy meter or flow meter and temperature sensors), it is much more difficult to do so on an air system where the air flow measuring equipment is too inaccurate or too bulky to be installed in winter at the home of a private individual. These various measurement campaigns will therefore not concern the air/air systems.

Unfortunately, the rapid growth of the market has led to counter-references on products installed by companies more concerned by the financial windfall than by the permanence of installed products.

The lack of performance assessments concerning air/air heat pumps combined with the disappointment of some users relating to the acoustic comfort of indoor units and the thermal discomfort relating to the movement of air in the home has thus led to severe criticism against air/air heat pumps. This is why the current study proposes to come back to the performance of air/air heat pumps: the measurement, on-site assessment, a sensitivity study as well as an analysis of the comfort and causes of failure are dealt with.

3.1.2 Objectives

The factors influencing the performance of air/air heat pumps in actual operation need to be targeted.

3.1.3 Results

3.1.3.1 Performance Measurement

The performance tests for heat pumps designed for comfort heating are subject to standard EN14511 that imposes in the case of air/air heat pumps only input conditions on the indoor exchanger as the difference on air flows from one product to the other is significant and above all the latter is not variable (or not yet).

In spite of the use of a test standard, the conditions can thus vary from one equipment to the other, mainly because of the uncertainty regarding the air supply outlet temperature.

The monitoring of in-situ performance levels involves heavy instrumentation difficult to implement. In particular, for the air flow measurement, which must be accurate but also “invisible” for the occupant of the apartment, which is not countable with most air flow measuring equipment. The monitoring of the performance of an in-situ air/air heat pump is thus much more expensive and tricky to install than the monitoring of a heat pump on water system. The installation of sensors requires meticulousness and a good knowledge of aeraulic phenomena. The accuracy of sensors used and the impact of their location currently result in a relatively low accuracy of the result.

3.1.3.2 Field Performance

Very little field monitoring has been done to date on air/air heat pumps. Monitoring undertaken by ADEME in 2010-2011 shows that the air/air heat pumps of this test do not present performance levels below other types of heat pumps monitored in the previous season. This average annual COP, of air/air heat pumps, measured in actual conditions is however lower (as for the other tested machines) than the rated average catalogue COP.

A centralised air/air type heat pump monitored for a winter period by EDF R&D shows satisfactory performance levels over the examined periods: An average COP of 3.3 is obtained over the period from 15 October 2011 to 29 February 2012. The instrumentation selected, which was simpler to set up on this centralised installation than on a multi-split, is satisfactory.

3.1.3.3 Thermal Comfort

The perception of comfort results from a perception specific to each individual, depending on the initial condition of the subject, his/her activity, as well as his/her other perceptions (acoustic, visual, etc.). To assess the perception of a ventilation, fan convector unit type system with respect to a convection heater type system (radiator, or convection heater), a sensory analysis seems necessary. The question that may be asked being: “To obtain an equivalent comfort level, between a Joule emitter and a fan convector unit, does the ambient temperature need to be increased?” This then leads us to envisage an in-depth study on the comfort perceived with forced convection emitters and particularly a comparison with electrical convection type heaters or panel heaters may be envisaged.

Discussions between R&D and the Applied Physiology Laboratory at Strasbourg specialised in comfort perception tests on humans, allowed us to envisage a study at fixed ambient temperature, where the sample group of subjects must assess the comfort level perceived with two types of emitters considered separately.

3.1.3.4 Modelling of the Operation of an In-situ Air/Air Heat Pump

A simple air/air heat pump model was created in order to estimate the impact of certain parameters, such as the rated COP or the sizing, on the seasonal COP. From the SIMFAST calculation core used at the ENERBAT department and a simple heat pump model developed in Excel based on data from a heat pump manufacturer.

In spite of strong hypotheses (particularly an identical heating season for all regions compared), the conclusions of this preliminary work show that:

- Sizing the heat pump at 100% of requirements does not lead to an energy optimum, all the more so as the base temperature at which the calculations are performed is rarely attained with this model,

which is based on a thirty-year weather forecast. The standby consumption significantly degrades the seasonal COP particularly when heating requirements are low and spread out over time.
- The impact of the machine's load factor, particularly on newly built homes and thus the importance of the regulation and sizing on these homes of low requirements.

The prospects of this work will be:
- Validation of the larger model, including heat pump profiles from other manufacturers.
- A finer modeling, enabling in particular to set the heating seasons in function of the geography.

3.1.3.5 Analysis of Causes and Effects

A variant of a problem resolution method was created on the loss of efficiency of an in situ air/air heat pump. The causes for the lowering of performance levels were combined in a family, classified according to possibilities of action: design, sizing rules, installation rules, maintenance. Certain causes of malfunction led us to undertake laboratory tests to determine the influence of a lack (or loss) of refrigerant pressure.

3.1.4 Conclusions

The analysis of performance differences between actual and theoretical conditions measured in the laboratory, for air/air heat pumps, will make it possible to target the actions to be implemented, or developments to be undertaken to assess and limit the impact of certain malfunctions.
- A seasonal COP of 3 can be attained in the field but one must be aware that this performance depends on the climate, the supporting structure and the behavior of users;
- A standardised seasonal COP is in the process of being defined (project of prEN14825 standard);
- The measurement of the on-site seasonal COP on an air/air heat pump requires heavy instrumentation.

The degradation of the COP is due to objective factors such as:
- Non-optimised regulation for the on-site performance;
- Standby consumption appreciably degrading the seasonal COP;
- Non-optimised sizing: The recommendation will be for 90% of requirements at Tbase.

Then, subjective factors such as the human factor are also involved (choice of the temperature setting) and in particular the comfort perception. The question of the comfort perception provided by a forced draught heating system with respect to a convective or radiative system may then be asked. A study entrusted to the laboratory (LINC) of the Strasbourg University has been initiated.

Finally, malfunctions relating to installation or maintenance defects should be analysed by laboratory tests.

3.1.5 Prospects

The prospects of this work will be structured as follows:
- Continuation of seasonal COP monitoring on-site: Monitoring of the heat pump presented in this document and analysing follow-ups of the multi-split as part of the SEPEMO European project;
- Need to objectify the thermal comfort: Service for a study to assess the comfort relating to the use of a ventilation system.
- Investment in modelling to refine parametric studies
  - Integration of new machine ranges and different manufacturers;
  - Study of the influence of regulation (load factor);
  - Validation of models by field monitoring;
  - Study of the case of a low-energy house.
- Laboratory tests to determine the impact of installation defects.
- Development of the flexibility of RR heat pumps;
  - Developing a regulation enabling arbitration (COP or control) and sturdiness (adaptiveness, self-diagnostic).
  - Studying the interest of a µ-storage system (ANR project)
- Support to the sector
  - Sizing, installation and maintenance rule (RAGE).

3.2 Feedback from a Measurement Campaign of the Thermodynamic Water Heater Performance

3.2.1 Context

The need for an alternative to the accumulation tank was identified since 2007 by the R&D. A state-of-the-art prototype created in 2009 shows that the thermodynamic tank is nowadays provided extensively with the various technologies present: buried sensors, on exhaust air, on outside air, on the heating return. However, none of these systems are completely independent; all of them include a supplementary heater. The products on outside air aren't many and are limited to outside air temperatures of -5°C. Products on ambient air, monitored on-site since several months by EDF R&D show seasonal COPs in the order of 2. A few years back, we had already identified a considerable advance by Japanese manufacturers in the production of thermodynamic ECS (Ecocute programme). The analysis of these products however showed that a technology transfer was necessary to adapt them to the European market.

This document presents how this technology transfer was possible, by presenting the performance levels of 10 prototypes of ECS thermodynamic tanks installed in the Paris region from December 2008 onwards.

3.2.2 Objectives Pursued

On the technical level, the product will have to meet the following requirements:

- The system must be mono block
- The annual average COP, including the defrosting must be at least 3
- It must be able to operate in outside air temperatures of up to – 10°C
- It must be devoid of a supplementary heater
- Its thermal power must be able to satisfy a demand profile of the type that is encountered in residential areas
- The heat pump is of the outside air/water type

The performance levels of this type of equipment is highly dependent on demand profiles and weather conditions, therefore, it seemed necessary for us to install prototypes in private homes.

The objectives of this on-site test campaign that started at the end of December 2008 are:

- To measure the Coefficient of Performance over one year
- To participate in setting up a commercial offer
- To assess with the selected installer, the possible implementation difficulties for heat pumps
- To work on the product design by including an Eco-design dimension

Work for product improvement is constantly undertaken during this test campaign in order to adapt the machine as best as possible to "French" use of domestic hot water. Ultimately, it should be verified that the requirements of the specifications are properly respected.

3.2.3 **Results**

Various tests in the Climatron Laboratory at the R&D des Renardières site made it possible to develop the initial system to lead to a prototype meeting the specifications. We were then able to carry out normative type tests in the laboratory, and install 10 prototypes at the sites for field test. The normative type tests place the tested prototype very favourably when compared with the requirements of the electricity performance NF make, in particular:

- The measured COP is 3.13 for a requirement of 2.2,
- The preheating time measured is in the order of 3h for a requirement of less than 12h.

The main results from the collection of data from December 2008 to June 2009, are listed in Table FR-4.

Table FR-4: Data Collection Results

Site no.	1	2	3	4	5	6	7	8	9	10
Number of occupants	3 or 5	3	4	2 or 3	5	1 or 2	3	4	5	3 or 5
Consumption (l/d)	126	103	95	86	102	18	34	63	72	78
Average COP	1.86	1.87	2.19	1.82	1.52	0.72	1.44	1.44	1.89	1.96
Table Note	*Site 6 presents an atypical performance with respect to the others: The water consumption is very low, the system operates almost exclusively to compensate for thermal losses of the tank.*									

The following may be noted:

- There is a strong dispersion of results, as far as water consumption is concerned,
- These water consumptions have daily averages that do not take into account periods when users are absent,
- The data analysed therefore include in particular the coldest period of winter, observed in January 2009, but do not include the summer and autumn periods when the outside temperature is milder; finally, the COP value will be higher (this will be weighted by the reduction of ECS consumptions, particularly in summer).

Based on these operating data, an annual extrapolation using a weather forecast file of the Paris region allows us to display an average annual COP in the order of 2.5; this value is appreciably lower than that which we have fixed, requiring improvements to the product. The most restrictive requirements have been met, including during the winter period when the outside air temperature was approximately – 11°C!

Since we can only look back about eight months, it is difficult to give an opinion about the influence of the presence of scales in the gas cooler (equivalent to the condenser), in spite of encouraging results. This risk of malfunction due to the scaling is identified and is the subject of specific monitoring.

The analysis of operating parameters has allowed us to highlight the points to be improved upon, such as the insulation of the tank, stratification and management (software). Modification were made to attain the ambitious objective of an average effective COP of 3, which will position this equipment very favourably with respect to competitors. The estimated gains brought about by every modification are indicated in Table FR-5, the initial COP is 2.52 (after considering the gain brought about by the software improvement).

Table FR-5: Estimated Improvement Gains *(as a result of each modification)*

Improvement made	Estimated COP	Implementation Date
Reduced standby consumption from 6 W to 4 W	2.57	31/10/2009
Improved air circulation	2.78	31/10/2009
Reduced thermal losses		
Safety unit on cold line	2.83	31/10/2009
Improved anti-splash nozzle	2.91	31/10/2009
Compressor improvement	3.00	30/11/2009
Optimised defrosting logic		
Optimised inverter		
Optimised compressor, pump and fan speed		

Among these modifications, the anti-splash nozzle is essential, in fact, without the stratification phenomenon, we cannot count on the effect of modifications on the temperature management. All these modifications are the consequence of observations from our field tests. At the end of November 2009, pre-production versions, including all the previous modifications, were installed on site in order to verify the effectiveness of improvement measures that we took (an observation period of 3 months should be respected).

Beyond the performance aspects, the objective of these tests was also to prepare the sales arguments and various recommendations for installation and use necessary for marketing the product.

The setting-up of prototypes on site allowed us to validate the installation procedure and identify problems that we may face during deployment of a commercial offer, in particular:

- The presence of a condensate lift pump was deemed necessary every second time, considering the location of the waste water discharge into the subsoil,
- The maximum permissible pressure of 2.3 bars originally is significantly insufficient as it interferes with the equilibrium necessary for the proper operation of thermostatic valve heads; the final version will include the necessary modifications and the maximum pressure will be increased to 3.5 bars.

The installation of the equipment did not raise any particular issues, but confirmed that two persons were necessary for installation, as is the case for an electrical tank of 300 l. Finally, the 8 months of operation

allow us to validate the technological choices made during the design of prototypes, which allows us to say that the product is technologically accomplished.

3.2.4 Prospects and Conclusions

The developed prototype is perfectly suitable for installation in individual homes. In order to extend its application range, it is necessary to reflect on the use of interior air, or even a mixture of interior air/outside air in order to facilitate its integration into an existing support structure. Another path to progress is the definition of a "thermodynamic head" which would allow this type of technology to be used in multifamily housing by retaining the existing tank, for example.

The work carried out enabled us to measure the differences between our specifications and the actual product. The average annual COP, in the order of 2.50 (extrapolation over one year based on field results to date), is below the objective of 3.

The equipment developed meets the most restrictive requirements, in terms of daily volumes output, as well as operating in exceptional climatic conditions; for this, the winter period enabled us to verify that the ECS requirement could be met even in the worst outside air temperature conditions (-14°C !).

Finally, the thermodynamic part may be disconnected from the tank, facilitating, for example, its implementation in multifamily housing.

The experience acquired shows that this equipment is not adapted to "small families", the COP being highly dependent on the quantity of water drawn. This product must be prescribed for families of at least 3 persons; it will provide the best performance when replacing a tank of 200 – 300 litres.

This note, prepared after 8 months of operation, obviously does not include the analysis that must be carried out at the end of one year of operation, particularly as regards the influence of the presence of scales on the gas cooler performance, but the results obtained are encouraging. Based on all tests carried out in the laboratory, 8 months of field tests and realistic improvements identified, we believe that this product provides a new technical dimension to the production of thermodynamic ECS.

4 IMPACT OF SENSITIVITY PARAMETERS

4.1 Air/Air Heat Pump Performance Tests in Degraded Mode

4.1.1 Context

Statistical monitoring of total consumption was carried out on homes equipped with different types of heating. This monitoring does not demonstrate the advantage provided by the heat pumps used as heating units. It was decided to more particularly study air/air heat pumps, with their use as heating units growing, particularly in the renovation of homes equipped with Joule convection heaters.

As part of a project undertaken by EDF R&D, the performance of air/air heat pumps and particularly the causes for the lack of performance of this equipment are studied. A work on the performance differences between the reality and those measured in the laboratory is thus carried out on air/air heat pumps since 2011.

Actions were undertaken to measure the performance of these systems in situ, through a thesis on the one hand and by the participation in the SEPEMO European project, on the other hand. Almost non-existent feedback regarding performance of this equipment in in-situ operation (particularly the Seasonal Performance Factor, SPF) penalizes the installation of these systems and the more or less widespread recognition of the renewable energy part that we can potentially attribute to them.

The actions undertaken since 2011 have more particularly shown that it was necessary to carry out tests to quantity the influence of problems relating to the installation and maintenance. Actions in this direction began in the second half of 2012.

4.1.2 Objectives Pursued

An air-to-air heat pump, containing R410A, was tested in the laboratory at the normative points (+7°C, +2°C, -7°C) for low refrigerant charge. The main objective of these tests is to demonstrate the influence of a

low-refrigerant fault on the COP calculation and on the power output for similar conditions. As the air-to-air machine used includes its own regulation, the parameters such as the compressor speed (or the frequency), the electrical input power, the condensation and evaporation temperatures and pressure (suction and discharge) and the discharge temperature are compared.

4.1.3 Results

4.1.3.1 Preliminary Test

The preliminary test, in accordance to the BS EN14511 Standard (*Air Conditioners, Liquid Chilling Packages and Heat Pumps with Electrically Driven Compressors for Space Heating and Cooling Test Conditions*; 2007) was designed to verify the performance of the machine before removing the refrigerant and carrying out partial fluid load tests. The first test was carried out under the conditions of the EN14511 standard and according to the procedure applied in the laboratory for this type of test.

For stabilised operation of machines, and to avoid the cycling of the compressor, the machine set point is fixed at a high temperature. However, as we cannot modify the internal regulation of the machine (to fix the compressor speed at a rated speed), the machine provided the maximum power, with a very high compressor frequency. In fact, the set point remaining fixed and the air inlet temperature for the indoor unit being fixed at 20°C as provided for by the test, the regulation of the system, not detecting any modification of parameters, requires the compressor to rotate at high speed to meet the demand.

The result of this test shows a poor performance, excessive power consumption and it is not possible to fix the compressor frequency, except with the manufacturer's intervention, which is not planned. Even the increase in the outside temperature will not enable one to observe modifications in the behaviour of the machine, and the modification of the internal flow does not result in a sufficient change.

After verifying the proper operation of the sensors and the outdoor unit, the conclusion of the test is that the machine regulates on the inlet-outlet temperature difference (and the variation of this difference) of the indoor exchanger. To summarise, the machine requires significant power to attain its set point, and as soon as a variation is noted, the power variation adapts to the requirements.

It has thus been decided to conduct the tests differently. The mode is experimental in the laboratory. The cell temperature will be maintained at 20°C, but with a base load to be provided so that the compressor adapts its frequency to an “average” value.

4.1.3.2 Validated Initial Test

The initial test is repeated on this principle and the COP, power and input power values are close to the manufacturer’s catalogue data. However, the difference between the manufacturer’s data and the values noted during the test demonstrate the difficulty in controlling the machine and the climatic cell (see Table FR-6).

Table FR-6: Comparison of the Laboratory Test with the Manufacturer’s Reference Test

	Inside T	Outside T	Pcalo (W)	Pcalo (W)	COP
Normative test	20°C	7°C	3260	600	5.33
Lab test	20°C	6.7°C	2036	368	5.6

The power involved as part of our full load test is lower, the compressor’s rotation frequency measured is 42 Hz, this explains an improved performance by 5% in our case with respect to nominal data.

4.1.3.3 Full load Refrigerant Test

The full-load tests present compressor speeds equivalent to the test at +7°C, which thus correspond to a stabilised operating state under climatic conditions that were possible to create in the laboratory. See Table FR-7.

The test at -7°C presents defrosting, the calculated COP includes defrosting cycles. We can note the near-absence of the compressor’s frequency variation for the 3 outside temperatures tested. The required load

inside being low, this can explain maintaining a low compressor frequency by the regulation, even when the outside temperature is lower.

Table FR-7: Full Load Refrigerant Test

	Inside T	Outside T	F(Hz)	Pcalo (W)	COP
Test +7°C/20°C v1	19°C	6.6°C	35	1490	4.3
Test +7°C/20°C v2	19.2°C	6.6°C	38	1762	5.2
Test +7°C/20°C v3	20°C	6.7°C	41.8	2036	5.6
Test +2°C/20°C v3	20.7°C	2.3°C	42.1	1855	3.6
Test -7°C/20°C v3	22°C	-7.2°C	38	1090*	2.3
Table Note	*estimated excluding defrosting				

4.1.3.4 Tests with Lack of Refrigerant Load

The tests performed as part of the loss of performance due to a lack of fluid are carried out at 3 outside temperatures (+7°C, +2°C, -7°C) for the speed no. 3 of the indoor unit (the speed 4 corresponding rather to a boost and exceeds the capacities of the air flow measurement system used in the laboratory). This speed however is representative of most equipment.

The tests were carried out for loads ranging from 48% to 130% of the nominal refrigerant mass that is 1150g for this machine.
Considering that the conditioning of cells was different for these tests from standard cases, and so that the machine behaves more realistically, the test conditions did not enable to view all the phenomena. More particularly, icing conditions do not appear, even at outside air temperatures of +2°C, due to the low thermal load of the machine during the test.

Fig. FR-1 shows the variation of the COP in function of the refrigerant load for three (3) temperature conditions. The COP includes the possible de-icing. These results may not be generalised as several operating conditions could not be maintained sufficiently stable in order to be able to replicate normative conditions. In particular, the humidity sometimes undergoes significant variations and the difficulty in controlling the machine did not allow these tests to be performed freely.

Nevertheless, Fig. FR-1 shows interesting trends. The machine's performance drops with the lack of refrigerant and this reduction seems to accelerate for a lack of load greater than 40%. It is possible that the variable-speed compressor, and above all, the electronic pressure regulator,y compensate for the loss of fluid by regulation. On the other hand, when this compensation is no longer possible, the drop in performance is significant.

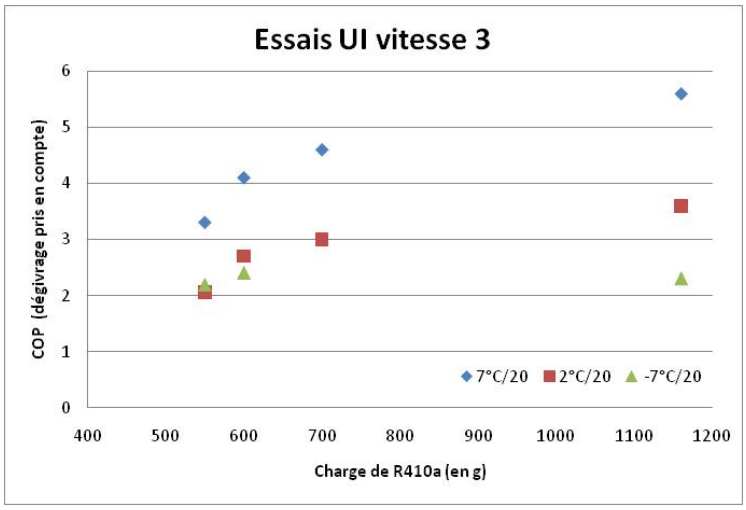


Figure FR-1: Performance of the Air/Air Heat Pump in Function of the Refrigerant Load

Fig. FR-2 shows the frequencies noted in relation to the load in the machine.

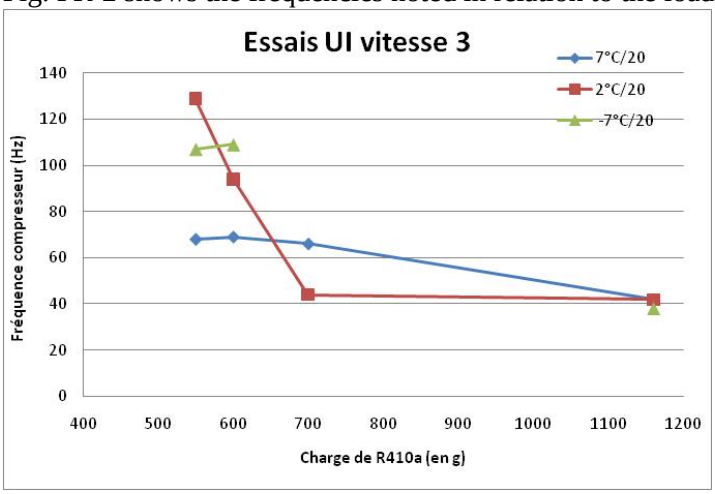


Figure FR-2: Compressor Frequency (adopted by the machine for variable R410A loads for 3 temperature conditions)

The test at +2°C, 700g does not seem to represent the general behaviour observed. The frequency noted is very low. These various conditions may be explained with the load on the indoor unit being quite low.

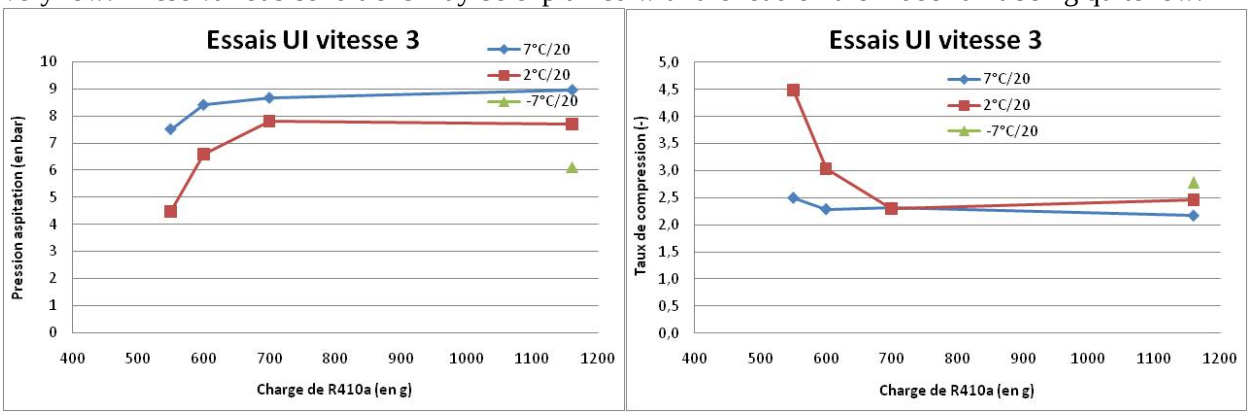


Figure FR-3: Influence of the Lack of Refrigerant Load on the Pressure Conditions in the Installation

The drop in performance discussed above (see Fig. FR-1) may be explained by two phenomena, the increase in the compressor's rotational speed (to provide the quantity of energy required by the heat pump), and the modifications in the machine's internal conditions.

The lack of refrigerant leads to a drop in evaporation pressure and an increase in the compression rate. The volumetric efficiency is affected, along with the isotropic efficiency of the compressor, which must operate on greater discharge/suction pressure differences.
This analysis more markedly shows that in the only performance study of a lack of refrigerant of up to 40% for this machine, the system is able to compensate for the loss of fluid for low thermal loads.

4.1.4 Prospects and Conclusions

The performance levels of the machine are affected by the lack of refrigerant for low thermal loads.

For a lack of 40% of refrigerant with respect to the rated load, the COP drops by 18% for the test at +7°C/20°C. At this stage, the impact is not felt as much on the compression rate or the evaporation pressure, it is the compressor that is loaded (increase in the rotational speed).

The impact is also weaker for lower temperatures at which the COP is lower; however in actual conditions, the required thermal load of the machine will be greater. The rotational frequencies adopted by the machine are already very high for a load of -48% and -52% of the rated load, the system thus would not be able to provide the entire demand.

5 CONCLUSION AND RECOMMENDATIONS

The heat pump market is expanding in France as well as in other European countries. Particularly, the air/air systems market is very dynamic, especially for the thermodynamic water heaters. The majority of sales concern the mono or multi-split air/air systems, particularly for the renovation of electrically heated homes and the thermodynamic water heaters systems for the replacement of electrical boilers.

Unfortunately, the rapid growth of the market has led to counter-references on products installed by companies more concerned by the financial windfall than by the durability of installed products.

The results of this Annex show that the sizing of the systems is an important step to obtain an efficient system. Particularly, we observed that: design, sizing rules, installation rules, and maintenance are the key points which are in relationship with the performance.

In parallel, the French government decided to support a new energy savings program (‘Rules of the Art for the Grenelle of Environment;’ RAGE) dedicated to the new buildings in order to respect the European energy targets. The main purpose of the RAGE program is to establish the rules of installation, maintenance, and commissioning of different systems, such as heat pumps.

Concerning the thermodynamic water heaters, which are part of RAGE program, field tests show that the correctly installed and sized units have good performances. They are higher than those observed by ADEME in previous field tests and who concluded that a bad performance was the consequence of a poor quality installation and sizing.

Finally, these technical rules will be disseminated to installers in several months, in order to improve the quality of the installation process and the customer satisfaction.

COUNTRY REPORT:
SWEDISH (1) CONTRIBUTION TO THE IEA HEAT PUMP PROGRAMME ANNEX 36 ON QUALITY OF INSTALLATION AND MAINTENANCE OF HEAT PUMPS

A Comprehensive Analysis of Faults in Swedish Heat Pump Systems

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ABSTRACT

The present work represents a start of a wider project called 'Smart **F**ault **D**etection and **D**iagnosis (**FDD**) for Heat Pump Systems'. The aim of the project is that of building an efficient fault detection and diagnosis tool that can be applied to heat pump systems during both installation and operation. Fault detection and diagnosis is a discipline, which has been widely applied in different sectors, but it is still in its infancy in the Heat Pump sector.

The aim of a fault detection and diagnosis system will be that of helping the technician while repairing the fault, in order to make the process more efficient and effective, thus reducing material and time costs.

The present work focuses on the developing of a literature survey. First, the FDD process in general is described with the aims, steps and advantages of the method. Subsequently, research aiming at understanding the different types of techniques for fault detection and diagnosis has been carried out, highlighting advantages and disadvantages of each technique.

The next step of the present work was to build a Comprehensive fault database**.** In order to create an efficient FDD tool it is fundamental to have, at least a preliminary knowledge, about the most common faults occurring in heat pumps systems. Other interesting questions are: "Which faults are most costly?" or, "What are the reasons for that?"

What has never been done before, and the aim of the present work, is to build statistics that reveals the most common types of faults in heat pump systems. The databases are provided by some of the main Swedish heat pump manufactures and the insurance company Folksam. Both manufacturers and Folksam have been involved in the project, and given useful contributions not only in terms of the data provided, but also in the form of shared knowledge and experience.

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1. INTRODUCTION

A Heat pump is a device that uses electrical energy in order to transfer low temperature heat into high temperature heat.

It extracts low temperature heat from ground, air or water and converts it into useful heat for space or water heating. It is another way to use solar energy. Energy from the sun heats soil, water, and air - heat which the heat pump then collects.

Figure KTH-1 shows in principle how a heat pump works: heat is transferred from the heat source to the heat sink by means of a refrigerant. The refrigerant evaporates in the evaporator and condenses in the condenser due to the different pressures at which the both processes takes place.

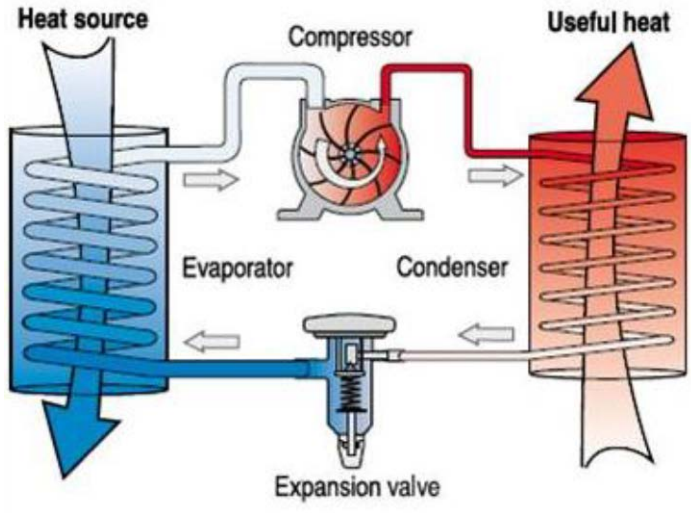


Figure KTH-1: Heat pump Function

Heat pumps can provide heating, cooling and hot water. They can be used in several applications:

- Residential buildings: mainly used in northern and central Europe, they represent an efficient way for heating with low emission levels
- Commercial buildings, where both heating and cooling load is needed, can be provided
- In the industrial field, heat pumps are mainly applied to recover waste heat
- District heating and cooling systems, can typically use sea water, ground water, rock or sewage



Figure KTH-2: The Heat Pump for District Heating in Stockholm *(the largest in the world, and uses sea water as a heat source)*

The most common types of heat pumps installed nowadays in Sweden are small reversible air to air units, which can provide both heating and cooling.

In the residential sector, the heat pump market is mature and currently developing, while in non- residential (commercial) buildings still represents a minor share among currently sold heat pumps.

The spreading of the heat pump technology has several advantages, starting from increasing the use of renewable energy, to the reduction of final and primary energy demand and CO2 emissions.

The application of heat pumps is a growing market in many European countries, encouraged by the European policy. The EU directive on renewable energy 2009/28/CE has set at 20% the share of energy from renewable source to be reached by 2020, and has included air source heat pumps and ground source heat pumps as renewable energy, with the constraint of a certain efficiency of the system.

Regarding greenhouse gases' emissions, heat pumps represent a great potential in reduction of CO2 emission into the atmosphere.

It is been estimated that heat pumps could save 50% of the building sector's CO2 emissions, and 5% in the industrial sector. 'This means 1.8 billion tons of CO2 per year could be saved by heat pumps, corresponding to nearly 8% of total CO2 emissions'.

The graph below highlights the benefit that can be achieved by using a heat pump instead of other types of heating systems, in terms of CO2 reduction.

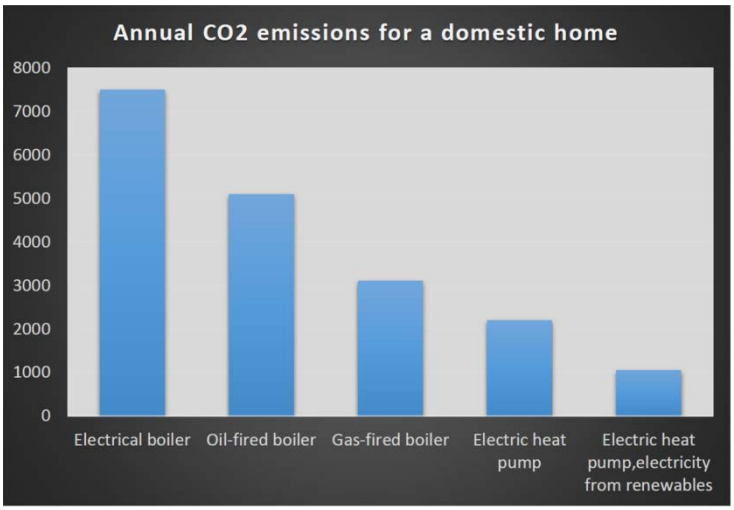


Figure KTH-3: Annual CO_2 Emissions for a Domestic Home *with 15000 kWh Annual Heating Demand (kg CO_2/year)*

Considering the spreading of the heat pump technology in Europe, a significant market share is present only in Sweden, Switzerland and Austria. In other countries still it is not the first choice for a new heating system.

In view of the advantages that heat pumps have also from the environmental point of view, this technology has a wide growing potential. An effort is then needed in order to further optimize the technology and stimulate the heat pump market.

1.1. Project Description

The present master thesis work is part of a wider project called 'Smart Fault Detection and Diagnosis for Heat Pump Systems'.

Fault detection and diagnosis is a discipline, which has been widely applied in different sectors, for example in nuclear power plants, though it is still in its infancy in the Heat Pump sector.

Fault Detection refers to the automatic process of determining whether a fault has occurred in the system. The next step, known as Fault diagnosis, aims at determining the exact location, time and magnitude of the fault. A further explanation of FDD processes will be given in the next chapter.

The aim of a fault detection and diagnosis system will be that of helping the technician while repairing the fault, in order to make the process more efficient and effective, thus reducing repair and time costs.

The project has as partners the main Swedish heat pump manufacturers, which will provide their knowledge about real heat pump systems by sharing their databases about faults occurring in their heat pumps.

Active part in the project is taken by Folksam, which is one of the main insurance companies in Sweden, and will also provide information about faults occurring in heat pump systems installed in Sweden.

The project is organized in several steps:

- Literature survey, analyzing the available FDD techniques, focusing on recent studies and mainly on what has already been done on the topic regarding heat pump systems.
- Building a comprehensive database of the most common faults occurring in domestic heat pump systems, which will be the basis for the developing of the FDD system.
- Design of fault detection system.

The goals of the project can be summarized as follows:

- Building a reference database which includes the most common faults occurring in modern heat pump systems.
- Designing a smart fault detection and diagnosis system at three different phases: commissioning, operation, and maintenance.

1.2. Master Thesis Project

The present master thesis will focus on the first step of the project.

1.2.1. Literature Survey

The first part focuses on the FDD technique: aims, steps and advantages of the application. Subsequently, research aimed at understanding the different types of techniques applied for fault detection and diagnosis purposes has been carried out.

The attention is placed on those techniques, which have been applied or had influence in developing fault detection and/or diagnosis tools for heat pump systems.

1.2.2. Comprehensive Faults' Database

Thanks to the data provided by Folksam, the large insurance company and also by some of the main Swedish heat pump manufacturers, it has been possible to build statistics revealing the most common types of faults and problems occurring in installed heat pumps.

The data provided as well as the whole project has its focus on Swedish residential heat pumps. To make the work more valuable, the contact with the manufactures has been emphasized.

A dynamic and helpful cooperation has thus been pursued, aiming for a better understanding of the analysed databases. Furthermore, meetings with companies resulted in discussions and corrections, thus improving the information.

2. LITERATURE SURVEY

2.1. General Methods of FDD

Fault detection and diagnosis is an area of research that aims at automating the process of detecting faults and diagnosis - their causes in physical systems.

The objective of a fault detection and diagnosis system (FDD) is the early detection of faults, before these lead to relevant loss of performance and further damages within the system.

A FDD system is composed of three key processes:

1. *Fault detection*: is the process of determining that some fault has occurred in the system.
2. *Fault isolation*: aims at determining the kind of fault and the exact location of the fault, as well as the time of detection.
3. *Fault identification*: includes determining the size and time- variant behavior of the fault.

Fault diagnosis commonly refers to the process of fault isolation and fault identification.

Automated FDD can have different applications: during commissioning, which involves checking if the systems are installed and operate properly - or during operation and maintenance. A reduction of costs will be pursued by preventing catastrophic failures and allowing shorter maintenance times.

The features required for an FDD tool can be listed as:

- robustness to data and with a low false alarm rate
- sensitivity to all possible faults in the process, allowing even a missed detection rate
- promptness - short detection delays

Methods for FDD were first developed for critical systems, such as nuclear power plants, for which early detection of malfunctions is of great importance but it was then applied also in other fields to guarantee better system performances.

In this first section, general methods for fault detection and diagnosis used in industrial systems will be briefly presented.

The first and simplest approach to fault detection is limit checking; then the methods for FDDs can be grouped into three main categories:

- data driven
- model based analytical methods
- knowledge based methods

All FDD approaches involve the construction of a model for describing the system. The main difference among the classes of methods lays in how the models are constructed. They can be entirely developed from measured data, as for the data driven methods, or they can be based on physical and mathematical knowledge about the system. Sometimes the system is described based on heuristic knowledge where such models can be used.

2.1.1. Limit Checking

Limit checking consists of rising alarms when measured variables cross predefined thresholds. This method was one of the first approaches used for fault detection. It is simple to develop, but it may also be imprecise, as the measurements are evaluated individually.

<u>Data-driven Methods</u>

The peculiarity of data driven methods is that they are not based on *a priori* knowledge of the system. The FDD process is entirely based on process data, instead of a mathematical knowledge about the system.

Data driven methods can be further classified into statistical and non-statistical methods.

Belonging to the class of statistical methods can be found, for example, polynomial regression techniques, partial least square (PLS), principal components analysis (PCA) and Fischer discriminant analysis (FDA).

Pattern recognition techniques, as artificial neural network, are part of the non-statistical data driven methods.

PCA, PLS and FDA are dimensionality reduction techniques whose aim is to capture the main directions within the measured data during training and represent them in a lower dimensional space. Fault detection is then performed applying multivariate statistics.

Compared to a non- statistical approach, these methods are based on a more rigorous statistical procedure which is able to capture the most important information among the process data; therefore they are more suited to be applied to large scale systems. These techniques require a large number of data to be implemented, thus they are useful when data are abundant and when a detailed knowledge about the system is lacking.

The advantages of these methods are:

- They are well suited for problems in which the theoretical knowledge is poor or in systems where analytical models are complex and time consuming to develop.
- They do not need a physical understanding of the system.
- They are useful in systems where training data are easy to collect.
- The computational effort required is generally manageable.

The disadvantages can be listed as:

- A large number of data is needed, representing both normal and faulty operation.
- The derived models cannot be used to draw conclusions beyond the range of training data.
- The models are specific to the system for which they are trained.

Analytical Methods

Analytical methods can calculate the status of the system using a detailed model based on physical laws. FDD is performed by analyzing the status using actual measurements.

The advantages of applying analytical methods are:

- They are based on physical principles: so if the model is well formulated, they can provide the most accurate description of the system.
- They can model both normal and faulty operation; therefore, faulty conditions can be identified without having to run the system under faulty conditions to obtain data.

On the other hand the disadvantages are the following:

- They can be complex and time consuming to develop.
- They can be computational intensive, therefore not suitable for online FDD.
- Models can require inputs that can be difficult to get.

When a good model is available this kind of approach provides the most precise outputs.

Knowledge Based Methods

Knowledge based methods can be applied when a detailed knowledge about the system is difficult to achieve, for example in complex systems, but when a good heuristic knowledge about the behavior of the system is available.

The base for these methods is that they apply models based on qualitative knowledge to draw conclusions regarding the state of the system. Some of these models accept qualitative data as inputs. Preprocessing is necessary where Fuzzy logic sometimes provides a tool for converting quantitative into qualitative inputs.

Many of the knowledge based approaches are well suited for fault diagnosis because of their ability to incorporate reasoning and simulate the experts' approach in diagnosing.

Examples of knowledge based methods are causal analysis and expert systems.

Causal analysis is primarily used for fault diagnosis. Graphs, for example, are constructed in order to underline the relationship among process variables and faults. The result is a list of possibly occurring faults. Drawbacks of these methods are that they can rarely be applied for large-scale systems, and they lack resolution in the results.

Expert systems methods provide logical rules for the system leading to the identification of the fault.

The approach is that of human reasoning and the knowledge that an expert has. It can be programmed in a series of *if-then* rules - or using Horns clause logic like in the computer language Prolog.

These methods are not developed on the basis of physical or engineering laws governing the system, but just on logic. Therefore knowledge also about faulty behavior is required.

To summarize the pros and cons of this class of methods we can say that:

- They are simple to develop and apply
- They can be useful when a large number of data and knowledge about failure of the system is available
- They are well suited when a deep knowledge about the system is lacking
- Their way of reasoning is transparent therefore easy to verify

On the other hand:

- They become more and more complicated and difficult to apply for complex system
- They are specific to a system or a process
- Their resolution can be approximate and strongly dependent on the expertise knowledge available

2.2. FDD in HVAC Systems

The last two decades and especially the last few years have seen a considerable development of research in the field of fault detection and diagnosis for HVAC systems.

More complex systems provide a better performance in terms of stable indoor conditions, but they are also failing more frequently.

A survey of over 55 000 air conditioning units showed that more than 90% of them were operating with one or more kinds of faults. Poorly maintained, degraded and improperly controlled equipment waste an estimated 15-30% of the energy used in commercial buildings. Much of this waste could be prevented with widespread adoption of automated maintenance.

Regarding heat pumps, a field analysis of Swiss heat pump installations showed that the averaged efficiencies of systems installed in buildings were about 10% lower than the efficiencies measured under standard laboratory conditions. Fault detection and diagnosis' objective is that of reducing and early detected malfunction to maintain good performance and maximize the COP of the system. Which is the most adequate method to achieve just that for the system considered?

Most methods have good performance, but they have only been tested in experimentally; in addition, some of them work well just for a single fault.

The choice of the FDD method should take the following account:

- Type of system for which the FDD is developed and the degree of knowledge available
- Input data available
- Degree of automation pursued
- Cost-to-benefit ratio
- Tolerance to false alarms

The following section will focus on the techniques applied in HVAC systems, and more specifically, in heat pump systems. A more detailed explanation of the methods and examples of application will be provided.

<u>Principal Component Analysis</u>
Among data driven techniques, PCA is a widely used one for monitoring industrial systems. It is a dimensionality reduction technique that captures the variability of data and represents them into a lower dimensional space. The representation of data provided by PCA can simplify and improve the performance of multivariate statistics in detecting faults.

PCA can produce lower dimensionality representation of the data which gives a better generalization of the independency of the training set. It is simpler than grasping the entire dimensionality of the observation space.

The structure abstracted by PCA can be useful in identifying either the variables responsible for the fault, or the ones most affected by the fault.

PCA can thus separate the observation space into a subspace capturing the systematic trends of the process and another subspace containing essentially the random noise.

Given a training set of *n* observations, each of them containing *m* measurements, the first step is to determine a set of orthogonal vectors (loading vectors). A number $a<m$ is the number of loading vectors selected for representing the whole training set. These will be called the principal components.

By appropriately determining the number of principal components *a*, the systematic variations can be decoupled from the random variations.

Dimensionality reduction is then applied to the actual measurements, then multivariate statistics, like Q or T2 statistics are applied in order to perform the fault detection.

This kind of approach was applied by Y. Chen and L. Lan (2009) specifically for fault detection on a heat pump system. The paper provides a list of common faults in heat pumps:

1. poor efficiency of heat transfer on condenser;
2. reduction of water flow through evaporator;
3. fouling of the evaporator on the water side;
4. leakage of gas in the suction or exhaust valve plate of the compressor;
5. refrigerant leakage;
6. refrigerant overcharge;
7. blockage in the high pressure liquid pipeline;
8. excessive valve lift of the expansion valve;
9. air or other non-condensable gas in the system.

The paper provides a description of the PCA method and the way how the reduction order was determined.

Steady state process data are collected for the training and the PCA model is built from this. Using the PCA model, the thresholds for the normal operating conditions are calculated using the SPE statistic (which is equivalent to the Q statistic).

The model was tested both under normal operating conditions and after the introduction of two different faults: 20% and 60% fouling of the condenser.

During the faulty operation conditions the SPE was considerably above the limits. Thus the faults where efficiently detected. PCA is a suitable method for capturing the data variation and can be applied for fault detection in heat pumps.

<u>Regression Models</u>

Regression techniques have often been applied for fault detection purposes. They are applied in order to develop a model for describing the system or correlating a system performance index to input parameters.

The model obtained is a black box model, in the sense that the parameters relating inputs and outputs have no physical meaning.

Wang and Zhou (2010) developed a system-level FDD strategy for HVAC systems based on regression models for evaluating features. The method includes a sensor fault detection scheme as a first step, performed with PCA. The system is then divided into subsystems. For each subsystem some performance indexes are chosen. Using a set of data from normal operation for the training, a regression model is then built for each **P**erformance **I**ndex (PI).

The coefficients for the regression model are determined from data measured during three days of normal operating conditions in summer and winter conditions. Another two days data were used to validate the models. Typical faults were introduced in the system model.

Residuals between predicted values and calculated values for the PIs were compared to thresholds in order to determine if a fault is occurring in one of the subsystems. The sensitivity of the PIs to the faults introduced was analyzed.

The FDD method combined with sensors' FDD method resulted to be effective in diagnosing the system level faults.

Navarro-Esbri et al. (2006) applied a black block model for on line fault detection of refrigerant leakage in a vapor compressor chiller. The model developed is a multiple regression model and fault detection is performed by comparing the actual measured variables to the predicted ones. Residuals have been evaluated through the comparison with dynamic thresholds.

Experimental tests have been performed in order to validate the sensitivity, robustness and promptness of the method, with good results.

Regarding heat pump systems, Choi et al. (2012) developed a refrigerant amount detection algorithm for a ground source heat pump unit. The authors analyzed the influence that the amount of refrigerant has on heat pump performance. It was proved that undercharge or overcharge lowers the COP of heat pump raising the power consumption. The parameters most sensible to refrigerant charge amount were found to be the degrees of subcooling and superheating. The subcooling was selected as a fault detection and diagnosis parameter. A regression model based on process data, correlating the charge amount and the subcooling, was developed. It was proved that this model can be effectively used for detection of faulty amounts of refrigerant.

For heat pumps, Kim et al. (2010) compared and discussed the performance of different methods for the development of a model for a residential heat pump system, which can be then applied for fault detection. A large number of measurements were collected from the system, during cooling mode operation. The data were first filtered by a steady state detector, using a moving window standard deviation technique. Different models for the evaluation features of the heat pump system were developed. A multi-variable polynomial regression (MRP) technique was applied and different order of regression models was built. Then an artificial neural network model was developed. Among the polynomial regression models, the ones with higher order showed better performance, but they also needed an exponentially higher number of parameters. The third order regression model was chosen, being considered a good compromise. The ANN model (below) was also able to predict all features, but with less accuracy than the 3rd order MRP model.

Artificial Neural Networks (ANN)

ANN is a method applied for model generation. It builds a map connecting input and output variables which consists of interconnected neurons arranged in layers. The most common approach is the three-layer configuration. One common way to use ANN for fault detection is to assign process variables as input neurons and fault classes as outputs.

The training of neural networks is entirely based on the availability of data, so it is a good tool for classifying data in systems where training data are abundant. ANNs are most useful when modeling complex systems.

A number of researchers also used black-box models based on ANNs for FDD (Katipamula and Brambley, 2005; Peitsman and Bakker 1996; Lee et al. 1996b; Bailey 1998; Morisot and Marchio 1999; Zmeureanu 2002; Reddy et al. 2003). Habbi et al. (2009) described a procedure for leak detection in heat exchangers. The authors applied a fuzzy model and a statistical chart for the evaluation of residuals. The output parameter of the model is the predicted exit temperature, which is dependent of four measured variables. The input measurements are converted into qualitative input, like small, medium, low. According to the qualitative inputs a certain rule for the evaluation of the predicted exit temperature is applied.

A cumulative sum algorithm (CUSUM) is applied on the residual in order to identify whether the system is in faulty or normal operation. Thresholds to be applied to the CUSUM charts are determined by considering

leak free data samples. A feature of the method is that it applies a dynamic fuzzy model which can be generated entirely from measured data.

Han et al. (2011) applied a machine learning technique (support vector machine, SVM) combined with multi-label approach, in order to perform FDD of multiple-simultaneous faults (MSF). Li and Braun (2007) had approached the problem of MSF by using features generated from a physical model, and then applying decoupling technique. In this work, the authors applied a data-driven method, instead of model based one. They adopted Li and Braun (2007) idea of decoupling for fault diagnosis of multiple faults.

The advantage is that this method allows a diagnosis of multiple faults, by training just with data from individual faults. Multiple faults result as a combination of individual classes. The method had a good performance in terms of correct classified samples, when it was trained with individual faults and normal operation data.

Regarding heat pumps, Zogg et al. *(2006)* developed a tool for fault diagnosis for heat pumps. A method based on the black box model and a clustering technique was proposed. First the model structure was chosen, and the parameters were then identified during the training phase by applying a LS algorithm.

Clustering method was then applied for fault classification. Clusters representing all different faults were built. The properties of these clusters, such as centers and standard deviations, were obtained from parameters identified during the training phase. During the operation a membership grade of the parameters to the fault clusters was calculated, thus the fault occurring in the system could be identified. The method was applied to a commercial heat pump tested on a test bench and it was able to classify gradual faults with a very low classification rate.

Analytical Methods in HVAC Systems

This class of methods applies a model based on physical or mathematical laws in order to predict outputs of the system. Residuals are generated through the comparison between predicted and actual values of features. Methods based on the evaluation of residuals are commonly referred to as analytical redundancy methods. Residuals can be non- zero due to faults, disturbances or noise, and modeling errors, so detection can be performed only by applying appropriate thresholds.

Carling (2002) developed a method for detection of airside fouling of coils using temperature efficiency as performance index. First, the reference value for the temperature efficiency was evaluated, then thresholds representing normal operation, were set on the basis of the variation of the estimated parameter in different time and days of measurements.

Li and Braun (2007) proposed a new methodology for FDD of vapor compression air conditioners: this one is able to deal with multiple simultaneous faults. The method adopted is an improvement of the statistical rule-based technique previously developed by Rossi and Braun (1997). This method can be classified onto the model-based methods, as it applies a steady-state model for feature generation.

Regarding fault detection, a simple fault detection classifier is adopted. It differs from the method presented by Rossi and Braun (1997) for the fact that it just needs statistical information about normal operation, such as mean values and variance of residuals. A boundary is set and if the current operation points are not inside the normal operation region it will be classified as faulty.

Concerning fault diagnosis, a mathematical methodology for decoupling is presented. Then this approach is applied to the case of an air-conditioning system.

First air conditioner common faults and their causes were listed; then a decoupling feature was detected for each of them. The decoupling features are parameters that can be either measured or derived from measurements.

Considering the decoupling features for fault diagnosis, the FDD problem was converted to n (number of features) fault detection problems which can be solved through the methodology previously described. For

each considered feature statistical thresholds are applied and if these are violated, the fault coupled with that specific feature is considered to be occurring.

The method was validated, in terms of sensitivity and robustness, by artificially introducing faults into a roof top air conditioner. The method proved to be very effective during tests.

Within the applications on heat pumps, Zogg (2002) describes the development of two FDD tools for heat pump systems. The software *Heat watch* is based on a steady- state model of a heat pump. It calculates and displays the physical parameters but does not interpret them. Fault diagnosis is performed by checking the trends of the parameters.

The diagnosis systems identify the current parameters and compare them to the reference parameters. The reference parameters are identified in advance, during the initialization step.

For the diagnosis, steady state values are used only. A method for steady state detection is applied. Measured signals are used to evaluate parameters through a set of steady state equations. Moreover, performance parameters and characteristics of the systems are calculated. FDD is performed by checking the trend of the displayed parameters. With this method, a high quality of performance was reached during tests.

Knowledge-based methods in HVAC systems
Expert systems can be based on expert knowledge or on first principle laws.

SHALLOW-KNOWLEDGE expert systems formulate the knowledge that an expert can have by using a set of *if-then* rules.

The method does not depend on a functional understanding or from the physic of the system. Shallow knowledge systems map the observations to conclusions directly. This kind of method can also be applied to areas where the fundamental principles or system description are lacking.

The advantages of that approach are: flexibility and are conclusions easy to verify. The main difficulty is the knowledge acquisition, which is the step of collecting adequate knowledge from domain experts and translating it into computer programs.

DEEP KNOWLEDGE expert systems are based on structural description of the system.

Governing equations provide a set of constraints on the values of process variables. Violations of these constraints are an indication that a fault is occurring. Each constraint is associated with the set of faults which cause the violation of the constraint.

Rossi and Braun (1997) developed a method for fault detection and diagnosis for vapor compression air conditioners. The outcomes of a steady- state physical model are statistically compared to on-line measurements. For operating on real systems, a pre-processing step using a steady state detector would be needed. The method suggested by the authors is moving window steady state detector.

Fault detection is performed based on mean values and covariance of residuals evaluated in normal and current operation. An inequality is formulated and if that is not verified, it indicates the occurrence of a fault.

A rule based method is developed for fault diagnosis. Seven temperatures are considered as features. Each fault causes an increment or a decrease of a specific feature. By analyzing the effect that a certain fault has on the considered features, it is possible to understand which fault is occurring. As before, this step was performed using a statistical approach.

Carling (2002) compared three different approaches previously developed for fault detection in HVAC systems. The methods compared and tested by the author were:

- A qualitative model based method, described by Glass et. al, 1995: this method detects faults by comparing qualitative controller outputs and the corresponding model based predictions, based on

temperature measurements. The method was able to detect 4 out of 18 faults. It presented a low false alarm rate.

- A rule based method, described in House 2001: it uses if-then rules to examine measured temperatures and controller outputs, and by this identifies potential faults. It detected 8 out of 18 faults tested.
- A model based method, modified version of the method described by Ruud (1997): in order to detect faults, the method analyses residuals between steady state model values or set points and measured values. Also standard deviations of residuals are analysed. The method detects a fault when a mean value or a standard deviation of a residual exceeds a threshold. The method was able to detect 6 out of 18 faults. At certain conditions, it raised frequent false alarms.

The APAR method **(P**erformance **A**ssessment **R**ules) applies a series of *if-then –else* rules and controller outputs to identify faults.

First, the mode of operation is defined, then the rules concerning the specific operational mode are applied. For example, during mode 1 the supply air temperature must be larger than the mixed air temperature, plus the emperature rise across the supply fan. An alarm is generated when the exponentially weighted moving average (EWMA) of the measured signals violate the rule at the end of one hour.

The methods have been compared according to the number of faults detected and the false alarm rate. The qualitative method was easy to set up, required no additional sensors, but detected few faults. The APAR method was easily readable, detected more faults but also generated more false alarms. The RAFT method also detected more faults, but also generated more false alarms. The RAFT method is more time consuming to implement and requires a larger number of sensors.

Fuzzy Logic

Fuzzy logic provides a qualitative description of numerical values and it is useful when applied to complex systems. A process variable can be translated to fuzzy concepts via a membership function, so that a number in the interval [0 1] is associated to each variable. Otherwise fuzzy logic allows the representation of variables in linguistic terms.

This technique has been often combined mainly with expert systems. If the number of if-then rules within the expert systems grows significantly, then fuzzy logic could be a good way to reduce the number of rules. The steps are (Chiang et al. 2001):

- Fuzzification: finding the membership function so that the input data belongs to a fuzzy set.
- Rule evaluation: treats the input data and finds the output membership function following the rules.
- Defuzzification: it is the inverse process. It calculates single output numerical values for a fuzzy output variable, on the basis of the membership function for this variable.

Fuzzy logic can be combined also with data driven methods, Neural Networks and analytical methods for fault diagnosis.

Steady-state Detector

The majority of research within the FDD field is concentrated on steady-state methodologies. In order to perform FDD analysis, it is important then to define and identify when the system is in steady-state conditions. Many authors suggest as data pre-treatment, the application of a moving window steady-state detector.

Kim et al. (2008) designed a steady-state detector for fault detection and diagnosis of a residential air conditioner. In this application a method using a moving window is applied in order to select when the system is working in steady-state: thresholds for selected parameters are defined from mean values. The parameters cannot exceed those thresholds over a predefined time period, referred to as the time window, for steady-state to be declared. The time window is moving in time. Thresholds were selected to both minimize the inclusion of non-steady state data and maximize the recognition of steady-state. The tool developed was able to detect steady-state condition under normal operation and under faulty operation (20% refrigerant undercharge) of the tested system.

3. HEAT PUMPS IN RESIDENTIAL BUILDINGS IN SWEDEN

As mentioned before, Sweden has developed into one of the most important markets in Europe regarding residential heat pumps. The sale figures are shown in Fig. KTH-4 below. The data originates from The Swedish Heat Pump Association, founded 1981. It contains sales from year 1981. It can be concluded that the recent installation of heat pumps started to increase 1995, and peaked 2008 (SVEP 2013).

The market share is divided into four main types of heat pumps: air-to-air, air-to-water, brine-to-water and exhaust air. Heat pumps are thus classified on the basis of the type of heat source they use: either outside air, or heat from soil or rock, or water, or exhaust air; and on the sink – air or water.

The market share of heat pumps in Sweden sees the predominance of brine-to-water heat pumps, with a number of sales almost reaching 25000 in 2010, even if that year saw a decreasing of sales, probably due to the economic crisis and market saturation. Today, over one million heat pumps are working in Sweden, mainly in small houses. All heat pumps in Sweden can be estimated to take 20 TWh from the surrounding (earth, water and air) and deliver 30 TWh to buildings while consuming 10 TWh of electricity.

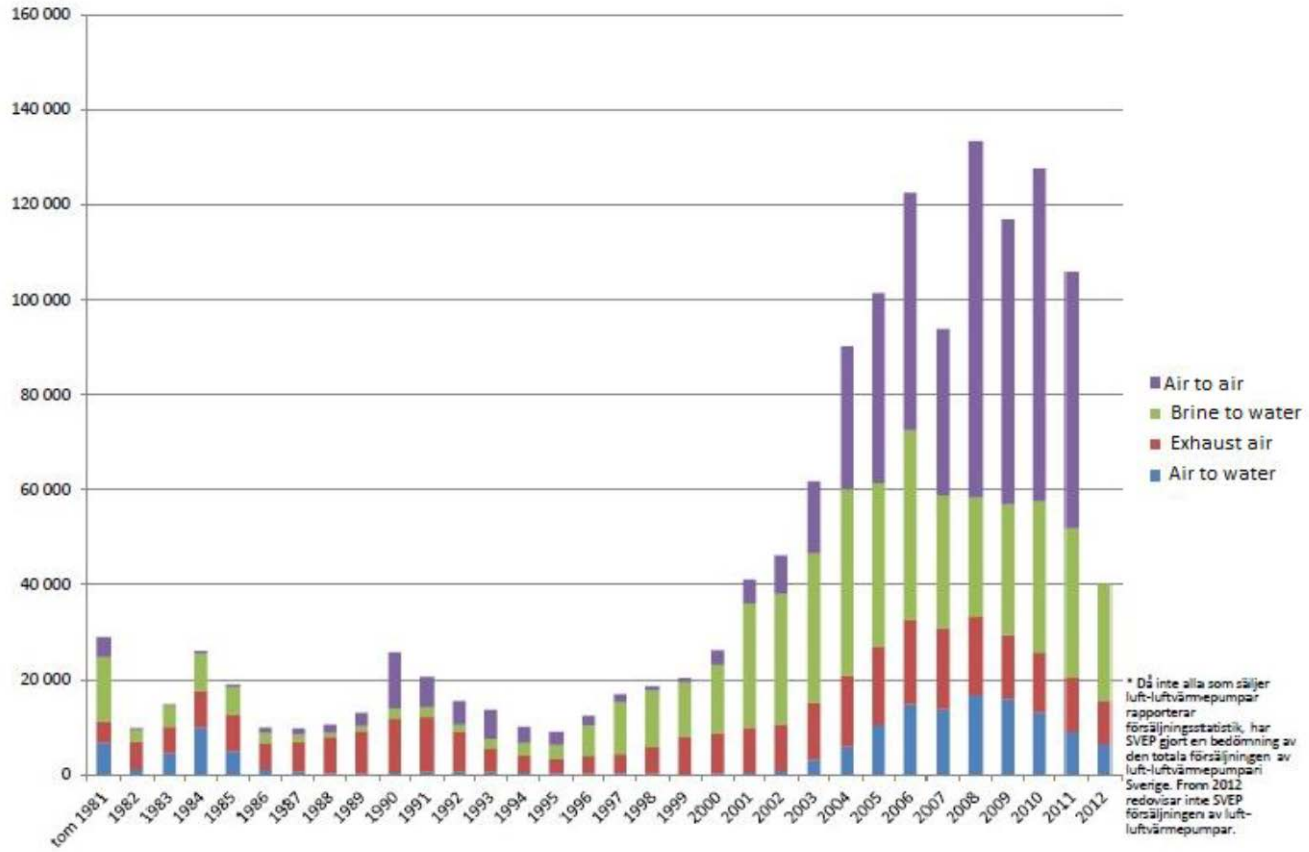


Figure KTH-4: Heatpump Sales 1982-2012 in Sweden (2012 not complete) (http://www.svepinfo.se/aktuellt/statistik/***)***

Fig. KTH-5 below shows the number of units sold in the last three years, divided into the three main types of systems. Both air-to-air and air-to-water heat pumps showed in 2010 a similar market share, counting a number of sold units between 10000 and 15000. However, in the last two years, the sales of air to water heat pumps have dropped more rapidly.

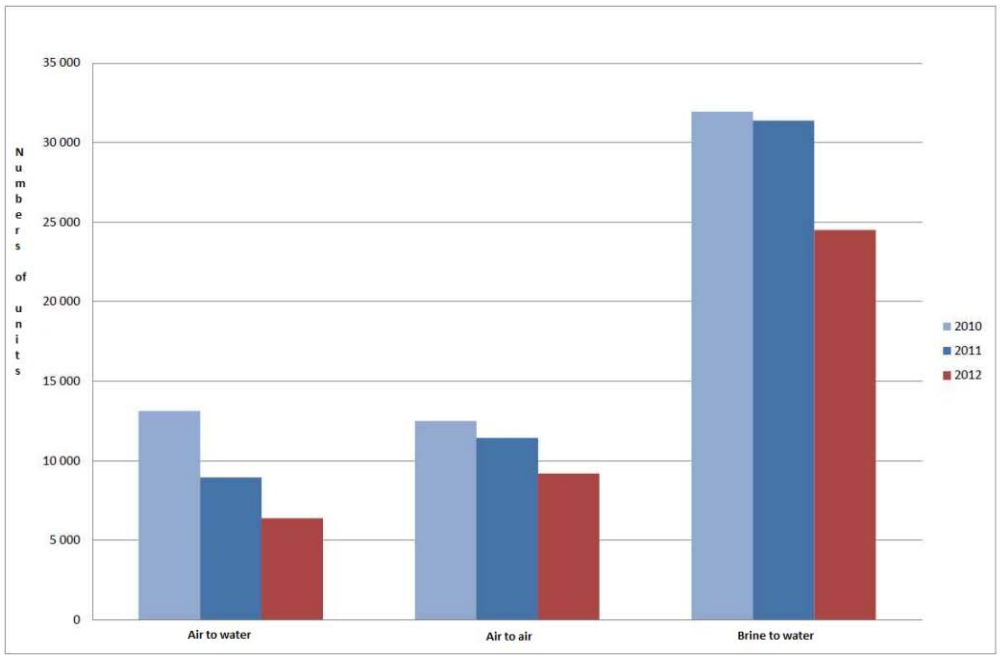


Figure KTH-5: Number of Heat Pumps Sold in Sweden in Years 2010, 2011, and 2012. (http://www.svepinfo.se/)

The following paragraphs will describe the characteristics of each type of heat pump more in detail and will show examples of configurations for the different systems.

3.1. Air-to-Air Heat Pump

Air-to-air heat pump uses outside air as its heat source. Heat is transferred from outside air to inside air by means of a refrigerant. Hot water is not produced, but the cycle can be inverted and the heat pump can be used as an air conditioning system.

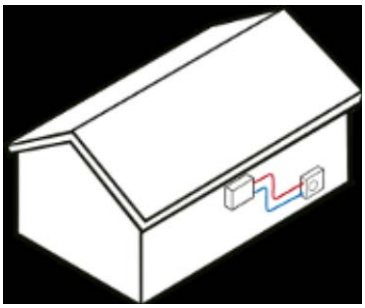


Figure KTH-6: Air-to-Air Heat Pump

Heat is extracted by blowing outside air through a heat absorbing air heat exchanger. A refrigerant is then heated, compressed, and heat plus the electric energy used in the compressor is transferred into the building.

The scheme that follows shows the main components of an air-to-air heat pump.

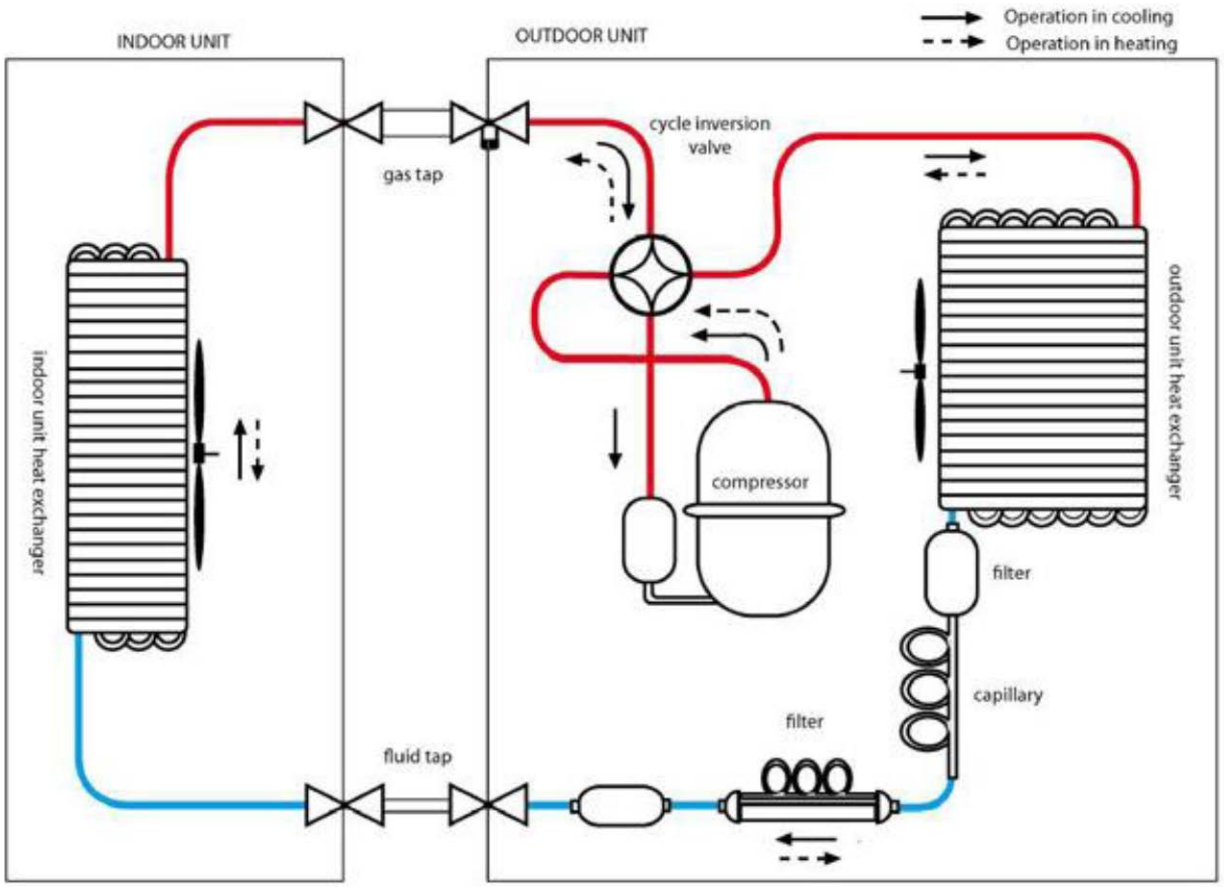


Figure KTH-7: Air-to-Air Heat Pump System Configuration *(http://www.artelgroup.com/)*

OUTDOOR UNIT:

- Fan: draws the outside air through an air heat exchanger, which is the evaporator in the refrigerating cycle.
- Evaporator: air is blown on the evaporator where it exchange heat with the refrigerant.
- 4 way valve: it's a key component in air to air heat pumps. It allows to invert the cycle, from heating mode in winter, to cooling mode in summer, and is also used for defrosting the outdoor coil.
- Compressor: takes the refrigerant vapour from low pressure to a higher pressure (and temperature). Scroll compressors are often applied in refrigerating systems. They are preferred to other types e.g. because of the absence of vibration during operation. The low noise level, is combined with a good efficiency and reliability.
- Expansion valve: the refrigerant going through the expansion device reduces its pressure and temperature to the lower level. The most common type is the thermostatic expansion valve. The amount of superheat can be adjusted with a screw acting on a spring. This kind of device keeps the superheat almost constant during operation. Another type of expansion device that can be used is the electronic expansion valve. The valve consists of a needle valve where the opening is controlled using a motor and control unit. Different superheating values can be set on the control unit.

INDOOR UNIT

- Indoor fan,
- Condenser: it is an air to air heat exchanger, which releases air to the indoor rooms (while running in heating mode).



Figure KTH-8: Copper Fin Heat Exchangers *(coils)*



Figure KTH-9: Thermostatic Expansion Valve

The price of an air-to-air heat pump ranges between 15 000 - 25 000 SEK, including installation. The advantages of this type of heat pump can be summarized as follows:

- Easy to install.
- Reversible, can produce both heating and cooling,
- Can be used to dehumidify the air,
- Low maintenance costs.

On the other hand, air-to-air heat pumps become inefficient when the outside temperature is considerably low. For this reason they are not suitable for Swedish climate in the north, where the temperature wintertime can go below -20 °C. Combination with a supplementary heat source is then compulsory. Another problem has been that defrosting in the Swedish climate is harder than normally.

3.2. Air-to-Water Heat Pumps

An air to-water-heat pump collects heat from the ambient air and transfers it to the water heating system of the house, as well as to the hot water tank.



Figure KTH-10: Air-to-Water Heat Pump *(http://www.nibe.eu)*

In principle, the function is the same as the air-to air-heat pump: a fan blows outside air through a heat exchanger, whereby heat is transferred to the refrigerant in the refrigerating cycle. Fig KTH-11 shows a basic configuration of the system.

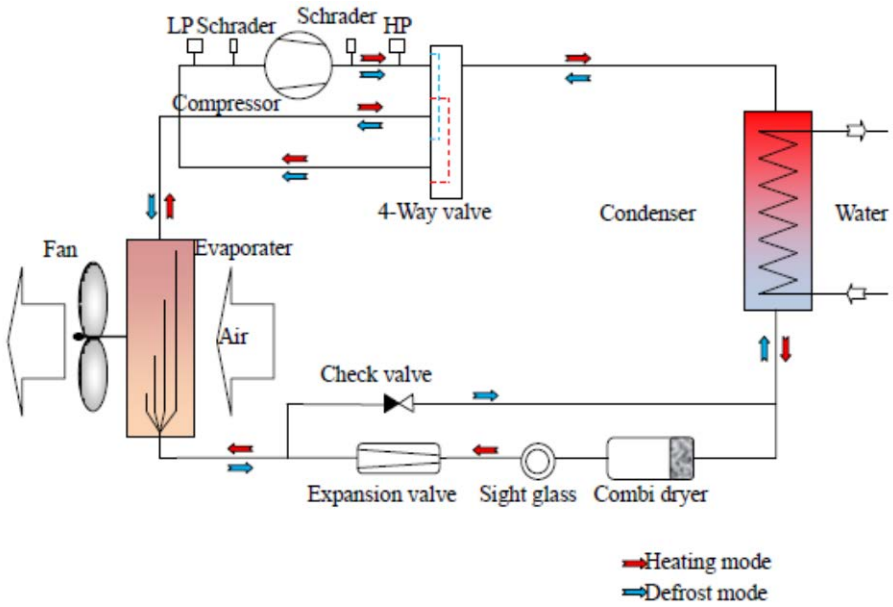
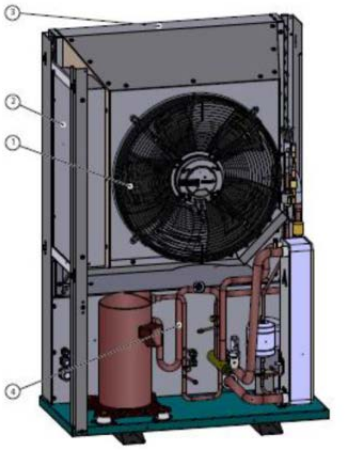


Figure KTH-11: Air-to-Water Heat Pump *(http://www.ctcvarme.se/)*

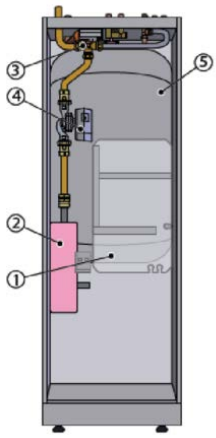
However, air-to-water heat pumps are present in different configurations. A split system has an outdoor unit which consists of a fan, an air heat exchanger, the evaporator, the expansion device, and a mechanism for defrosting. The indoor unit includes compressor, condenser, heat distribution and control. The difference, compared to an indirect system, is that in that case the outside heat exchanger does not have the function as evaporator. That requires an extra heat exchange step, transferring heat from outside air to the inside heat pump unit.

A third configuration consists of a heat pump unit placed outside, including all the components. Fig. KTH-12 and Fig. KTH-13 show an example of the last configuration described: the heat pump is placed outdoors and the control unit is placed indoors.



Pos.	Description
1	Fan
2	Electrical cabinet
3	Air heat exchanger
4	Refrigerant circuit

Figure KTH-12: Outdoor Unit (www.thermia.se)



Pos.	Description
1	Control module
2	Immersion heater
3	Reversing valve
4	Circulation pump
5	Water heater

Figure KTH-13: Indoor Unit with Hot Water Accumulator (5) (www.thermia.se)

Fig. KTH-14 (next page) shows more in detail an example configuration and the different components of an air-to-water heat pump.

In the evaporator, air releases heat to the refrigerant. The heat exchange is driven by a fan, which can be e.g. a variable-speed fan: the air velocity is controlled by the temperatures in the refrigerant circuit.

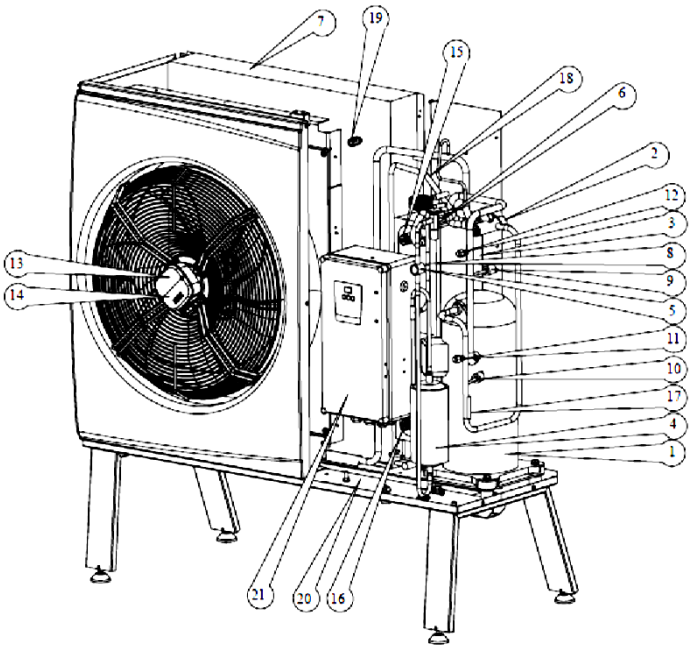
The vapor is then sent to the compressor through which pressure and temperature are increased. Before and after the compressor, two pressure switches are placed in order to keep the pressure levels between predefined ranges.

The fluid then passes through the four-way valve and reaches the condenser, where the heat is transferred to the water heating system. The refrigerant passes through a drying filter and the expansion valve, where the pressure and temperature drop and the process starts again.

Several temperature sensors are placed in the circuit: before and after the condenser, a hot gas sensor, to control the temperature of the hot gas after the evaporator, temperature sensors on the evaporator controlling the defrosting mechanism. Ice is likely to form on the evaporator surface, so defrosting cycles are needed in order to avoid ice formation on the exchanger. This leads to several problems, from the loss of efficiency of the heat exchange, to the rupture of the evaporator or the breaking of the fan (Granryd et al. 2009).

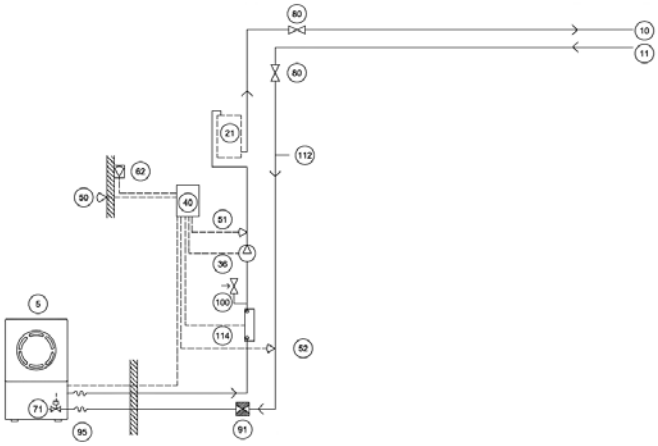
Fig. KTH-15 (next page) shows a further example of system configuration. What can be seen is the heating system side and the different sensors connected to the control unit. The supply line is equipped with an electrical auxiliary heater and, through the circulation pump, feeds the volume tank and the heating systems of the house. Sensors control the temperature both on supply and return line. The outside temperature is measured, as well as the room temperature. All the signals are managed by the control unit. For safety purposes a shut off valve and a safety valve are placed in the circuit.

The advantages are similar to those concerning air to air heat pumps: simple installation and low total cost, compared to e.g. geothermal or ground source heat pumps. On the other hand, the seasonal efficiency is poorer and it needs supplementary heat sources in Sweden. The price for an air-to-water heat pump can vary between 50 000–110 000 SEK, including installation.



1	Compressor		11	Schrader valve HP
2	4 way valve		12	Schrader valve LP
3	Condenser		13	Fan
4	Drying filter		14	Capacitor to fan
5	Inspection glass		15	condenser sensor out
6	Expansion valve		16	condenser sensor in
7	Evaporator		17	hot gas sensor
8	One way valve		18	defrosting sensor before evaporator
9	Low pressure switch		19	Defrosting sensor in the evaporator
10	High pressure switch		21	Connection box/ control panel

Figure KTH-14: Air-to-Water Heat Pump - Components *(*http://www.ctcvarme.se/*)*



Pos	Qty.	Name
5	1	Heat pump unit
10	1	Supply line
11	1	Return line
21	1	Volume tank
36	1	Circ.pump (radiator)
40	1	Control unit
50	1	Outdoor sensor
51	1	Supply line sensor
52	1	Return line sensor
62	1	Room sensor
71	1	Flow guard
80	2	Shut-off valve
91	1	Strainer filter ball
95	2	Flexible hose
100	1	Safety valve (3,0 bar)
112	1	Expansion tank
114	1	Additional heater

Figure KTH-15: Air-to-Water Heat Pump - Systems Configuration, Heating Circuit. The entire refrigerant circuit placed outdoors. (courtesy of Thermia)

3.3. Brine-to-Water Heat pumps

A brine-to-water heat pump is characterized by a brine circuit collecting heat from different sources and transferring it by a heat exchanger to the refrigerant circuit. As air-to-water heat pump, they produce warm water both for the heating system and for sanitary uses.

Brine-to-water heat pumps is a very common type of heat pumps in Sweden. There are several reasons for that. The efficiency of these systems is higher compared to most other types of heat pumps. The advantage is that heat is extracted at a temperature level, which is almost constant throughout the year. This fact has an important influence on the system performance. Even though the system has higher investment costs, it is reliable and has a long life.

This type of heat pump can be subdivided into different types according to the heat source.

Bedrock heat pumps exploit the solar energy which is stored in the rock. A hole is drilled, and the brine circulates into the hole extracting the heat from the soil. The depth of the hole can vary typically from 100 to 250 m for a villa.

Surface soil source heat pumps let the brine circulate through a collector placed at around 1 m depth in the soil. The collector is typically 200 to 600 m long. This solution avoids the drilling, needed for the installation of the geothermal heat pump, but requires a larger surface for the collectors.

Another possible solution can be that of extracting heat from lake or sea water**.** The principle is the same as the one described for ground source or geothermal heat pumps. The solution is often applied for larger systems.

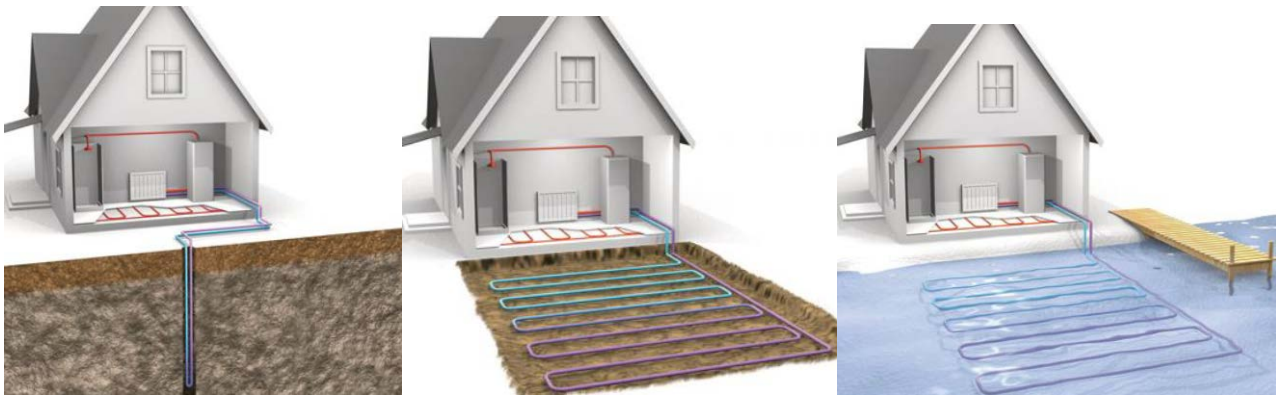


Figure KTH-16: Examples of Varied Heat Sources (website: www.thermia.se)

Fig. KTH-17, Fig. KTH-18, and Fig. KTH-19 show the components and the system configuration of a brine to water heat pump. The system has three different circuits: the brine loop that collects the heat from the source and transfers it to the refrigerant; the refrigerant, which after compression, transfers the heat to the water used in the heating system and in the hot water heater (tank). The function of the shuttle valve is that of switching the flow to the heating system or to the hot water heater. An auxiliary heater provides additional heat at peak loads, when needed.

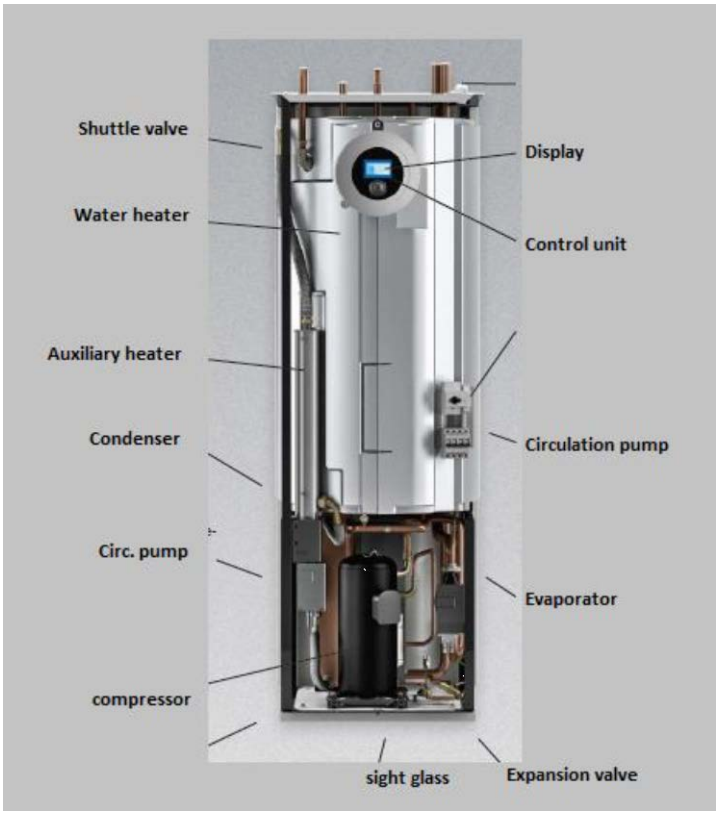


Figure KTH-17: Brine-to-Water Heat Pump System *(www.ivt.se)*

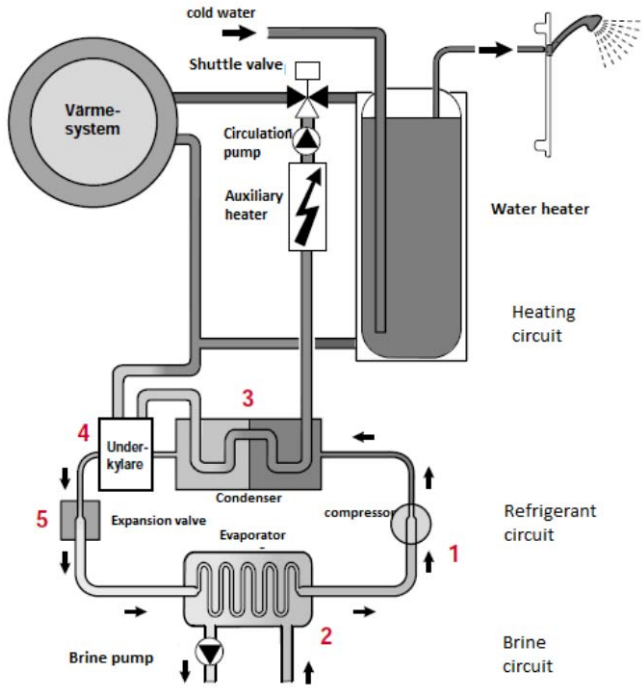
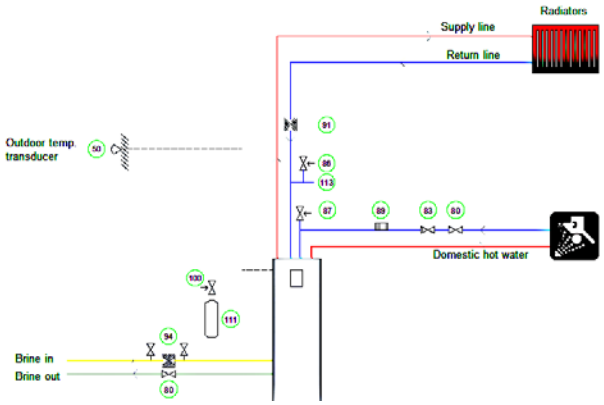


Figure KTH-18: Brine-to-Water Heat Pump Scheme *(underkylare means subcooler, www.thermia.se)*



Pos.	Description
50	Outdoor sensor
86	Safety expansion valve
113	Expansion vessel
87	Safety valve
89	Vacuum valve
83	One way valve
100	Safety valve
111	Expansion vessel
94	filling slot

Figure KTH-19: Brine-to-Water Heat Pump Scheme *(courtesy of Swedish company Thermia)*

The prices for a brine-to-water heat pump range from 100 000 to 200 000 SEK for a villa, depending on the size, system, type and place of installation. Sweden is one of the countries with the largest market for brine-to-water heat pumps. As of late-2013, there were approximately 450 000 installations in villas.

3.4. Exhaust Air Heat Pumps

Exhaust air heat pumps can represent an efficient way for exploiting warm and humid air which is taken out from the building during ventilation. This represents one of the main heat losses in buildings and it is reduced significantly using this heat recovery. This form of heat recovery can also diminish the need for deep boreholes, as the ventilation heat can be dumped there, when not needed.

Obviously, the application of this kind of system is only possible in those buildings with controlled evacuation of ventilation air. As in the case of air to water heat pumps, different configurations of the system can be chosen.

Some solutions use a brine to water heat pump to recover the heat. During winter, especially in cold climates, the heat provided only from the exhaust air heat pump is not enough in order to meet the load of the building and the hot water production. Therefore, it is necessary to have an auxiliary heat source, which could be another heat pump.

The advantage of exhaust air heat pumps, beside the fact that it can recover heat from the ventilation, is the ease of installation, and consequently a relativetotal cost which is significantly lower as compared to geothermal heat pumps. The price of an exhaust air heat pump including installation ranges between 40 000–70 000 kr.



Figure KTH-20: Exhaust Air Heat Pump System *(*http://www.nibe.eu*)*

4. FAULTS' DATABASE: AIMS AND METHODOLOGY

The aim of the FDD project is to use the FDD tool to improve the efficiency of the building when applied to heat pump systems during installation and operation.

When this task is presented, several preliminary questions arise: what are the most common type of faults in heat pumps? Moreover, what are the faults determining the higher cost when repaired? What are the reasons for that?

To create an efficient FDD tool, it is fundamental to have a preliminary knowledge about faults, and about the most common ones occurring in heat pumps systems.

What has never been done before, and the aim of the present work, is to build statistics revealing the most common types of faults in heat pump systems, based on the knowledge that heat pumps' manufacturers have about their products and the databases of faults occurring in their systems.

The databases which have been analyzed will be presented in two different statistics:

- data from the Swedish insurance company Folksam.
- data from Swedish heat pumps' manufacturers.
-

The procedure of analysis was the following:

- Data were categorized according to the different types of heat pumps (air to air, air to water, brine to water, exhaust air).
- Data where grouped into different categories representing a faulty component or a reason of failure.
- Charts were constructed in order to make the most frequently reported faults visible. E.g. the faults imposing.
- high total cost and the most expensive components.
- Whenever present, detailed reasons for faults where analysed and collected in diagrams.
- Meetings with heat pump manufactures were held, for presenting the data.

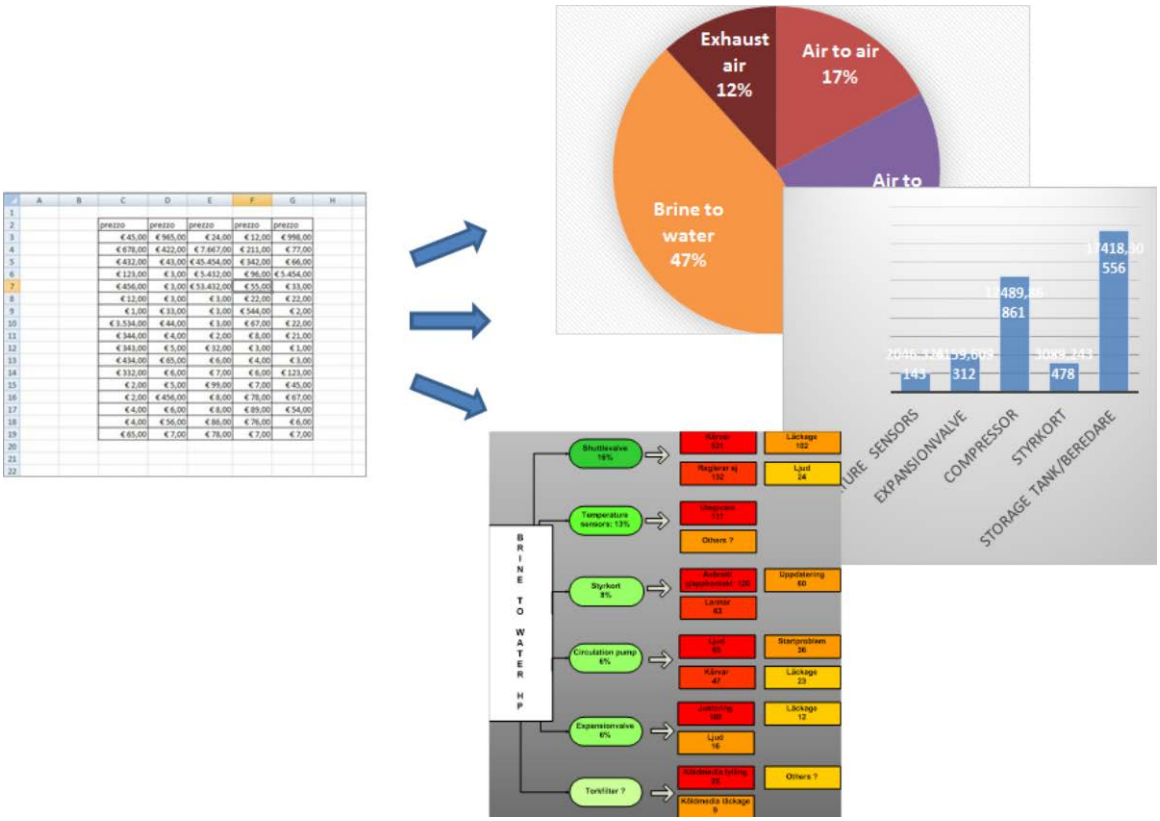


Figure KTH-21: Procedure of Analysis

The Folksam database contained 13 993 faults and included data of faults from year 2001 to year 2008. Due to an error in the data files, data from year 2007 could not be included in the statistic.

A selection was made to make the database more updated. Old systems were not included in the statistics and only machines produced from 2001 where included.

Removing wrong data and data not presenting the model or the type of heat pump, the number of faults included in the statistic shrunk to 8659. The share of faults among the different types of heat pumps is shown in Fig. KTH-22.

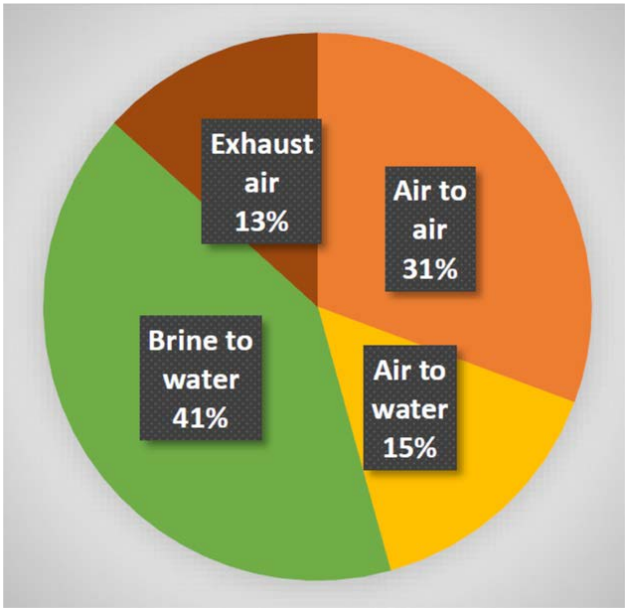


Figure KTH-22: Distribution of Faults into Different Types of Heat Pumps *(Folksam)*

Faults regarding brine to water heat pumps are most common. This agrees with the distribution of heat pumps installed in Sweden, which had a predominance of such systems.

The same methodology has been applied to the data coming from heat pump manufacturers. Table KTH-1 summarizes the number of faults received from them, selected and included in the statistics. Figure KTH-23 represents the data distribution among the types of systems. Since there are some confidential issues regarding the faults provided by the heat pump manufacturers, the faults provided by the heat pump manufacturers are not presented in this study; they will be presented in the later stages of the project.

Table KTH-1: Numbers of Faults from Manufacturers

Totally received	Uncompleted data*	Models not classified	Faults occurring before 2010	Total number of faults in study
67952	8612	4372	17895	37073

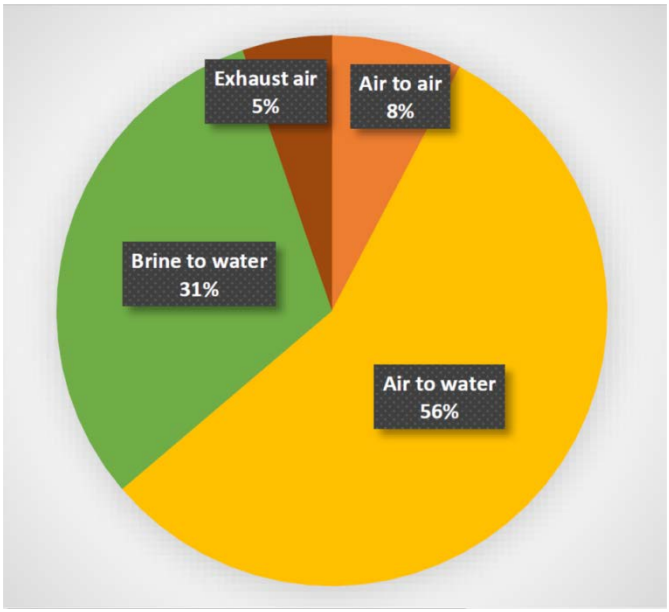


Figure KTH-23: Distribution of Faults into Different Types of Heat Pumps, *statistical data from manufacturers*

5. RESULTS

This chapter will present the results of the statistic provided by Folksam. Folksam is one of the largest insurance companies in Sweden, with about 4 million customers and 600000 insurance claims settled every year. Folksam collected statistics on heat pumps claims since 1999. The present study removed the old data and only used data starting from 2001. The statistics regarding different types of heat pumps is presented separately, with focus on the faults resulting in the highest total cost.

5.1. Air to Air Heat Pumps

Fig. KTH-24 shows the cost distribution of faults of models belonging to the category air to air heat pumps. What is immediately evident is the predominance of compressor faults. There can be several reasons for this. One reason can be that the compressor technology and reliability has improved during the last years. Another reason can be the attention of the manufacturers and repairers in detecting and reporting the right fault. It is known from experience that often compressors were reported to be faulty even if the problem was not due to the compressor itself, but had other causes. Compressors might thus have been “over reported”. Components like electronics and fans also show a high failure rate, but their average cost is less compared to compressors.

Failures in electronics were common in all types of pumps, while fan failures are most typical in air to air heat pumps. Beside the failure of the component itself, there can be several reasons for the breakdown of a fan. A common failure is for example first the formation of a layer of frost on the evaporator: then the blades of the fan hits the ice and breaks. The fan is not the primary cause then, but the malfunctioning defrost system. The indoor unit fan can cause noise problems. That is mainly due to bearing failures.

Other faults as evaporator faults or refrigerant leakages also present a high total cost. The 4-way valve represents almost the 3% of the total cost.

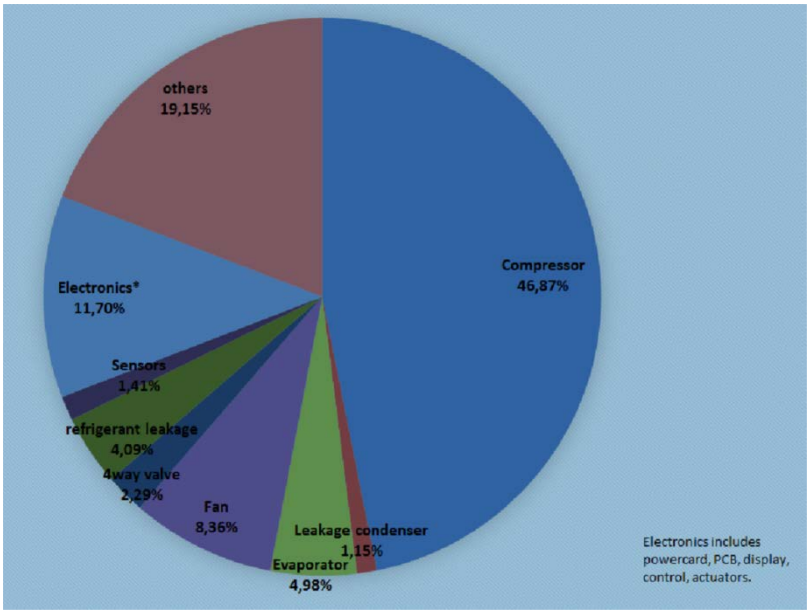


Figure KTH-24: Aggregated Cost Distribution; Folksam Air-to-Air Heat Pump

5.2. Air-to-Water Heat Pumps

Fig. KTH-25 shows the cost distribution of faults of models belonging to the category air-to-water heat pumps. The cost distribution of faults regarding air to water heat pumps shows similar features as air-to-air heat pumps. Compressor failures represent more than 50% of the total cost, followed by faults in the evaporator, refrigerant leakage, electronics, four-way valve and fan.

Other faults like leakage from the hot water tank, frost induced faults, leakage in pipes, leakage in condenser, pressostats or electric failures, represented a smaller percentage of the cost. Some of them however, are expensive to replace, as leakage from the hot water tank. The cause is also here, both depending on the cost of the component and the work needed for the replacement.

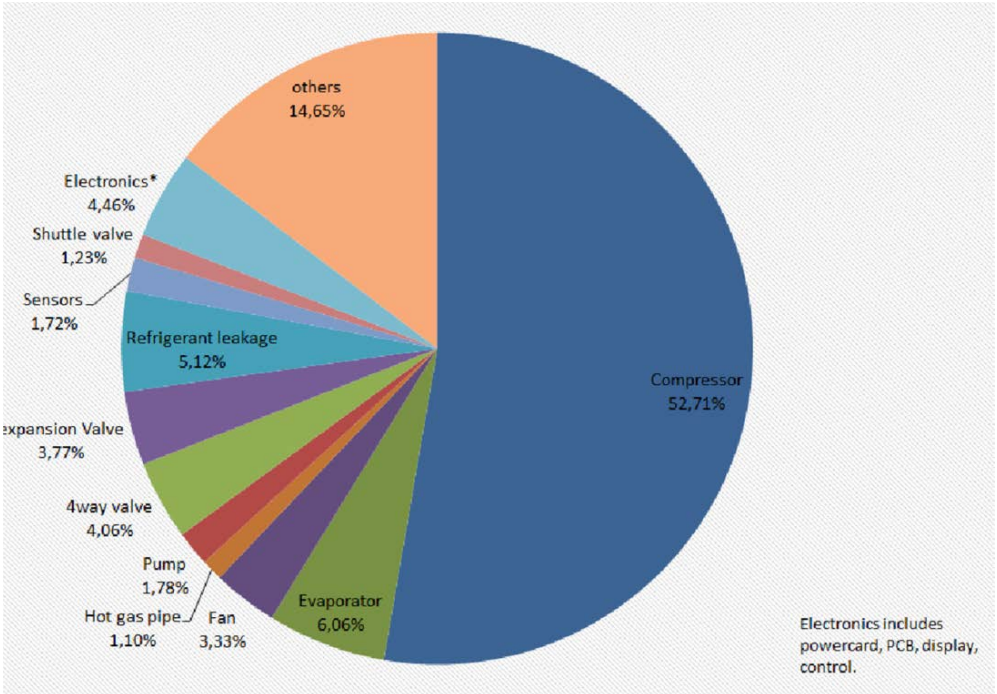


Figure KTH-25: Aggregated Cost Distribution; Folksam Air-to-Water Heat Pump

5.3. Brine-to-Water Heat Pumps

Fig. KTH-26 shows the cost distribution of faults of models belonging to the category brine-to-water heat pumps. Besides the percentage of cost due to compressors faults, which is almost 50%, the shuttle valve is the component representing a highest total cost, almost 10%. The average cost of the component is low but the failure rate is very high. Electronic failures are common in brine-to-water heat pumps, similar to other types of heat pumps. The brine loop represents the 4 % of the total cost; almost half of the failures are noted as leakage by the brine, while the other half are noted as brine pump failure. According to the statistic, the most expensive components to replace are the evaporator, the compressor, and the hot water tank.

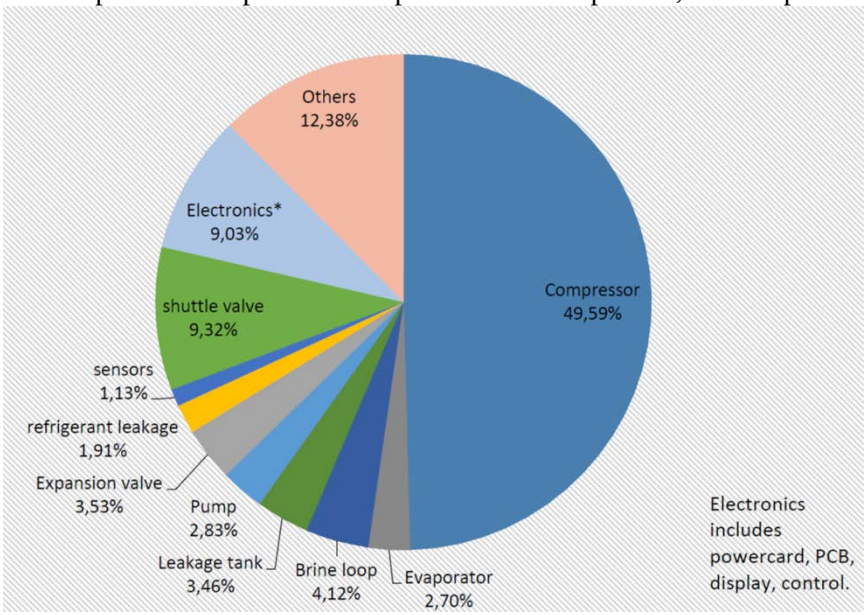


Figure KTH-26: Aggregated Cost Distribution; Folksam Brine-to-Water Heat Pump

5.4. Exhaust Air Heat Pumps

Fig. KTH-27 shows the cost distribution of faults of models belonging to the category of exhaust air heat pumps. The statistics based on data from Folksam for exhaust air heat pump sees compressors faults having the largest share of the cost, more than 50%, and the evaporator, representing almost 15% of the cost. Failures of pumps, expansion valves and refrigerant leakage, represent important parts of the cost as for other types of heat pumps. Tank leakage represents almost 5% of the cost, due to the high cost for replacement of this component.

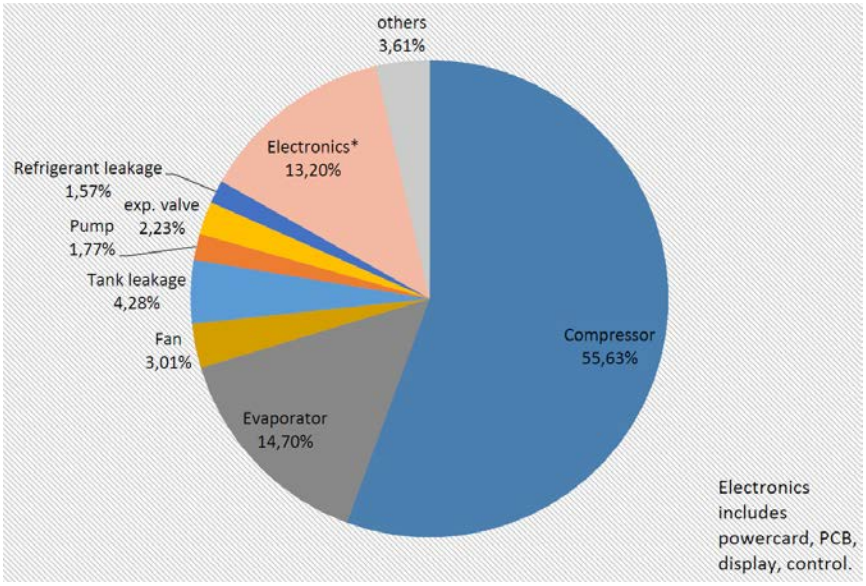


Figure KTH-27: Aggregated Cost Distribution; Folksam Exhaust Air Heat Pump

5.5. Summary of Results

Table KTH-2 shows a summary of the results from processing the faults data provided by the insurance company, Folksam.

Table KTH-2: Folksam Statistical Overview

	Type of Heat Pump (HP)			
	Air-to-Air HP	Air-to-Water HP	Brine-to-Water HP	Exhaust air HP
The most common faults	Compressor (30%)	Compressor (52%)	Shuttle valve (22%)	Compressor (40%)
	Fan (19%)	Evaporator (6%)	Compressor (19%)	Control and Electronics (15%)
	Control and Electronics (13%)	Control and Electronics (4%)	Control and Electronics (14%)	Evaporator (11%)
The costliest faults	Compressor (46%)	Compressor (52%)	Compressor (49%)	Compressor (56%)
	Control and Electronics (12%)	Evaporator (6%)	Shuttle valve (9%)	Evaporator (15%)
	Fan (8%)	Refrigerant Leakage (5%)	Control and Electronics (9%)	Control and Electronics (9%)

Appendix A provides a summary information from the perspective of the equipment manufacturers.

6. CONCLUSIONS

This work falls in the streamline that considers heat pumps as a promising technology in the field of heating systems, in terms of use of renewable sources and CO2 emissions reduction.

Sweden is one of the European countries where the technology is more commercially mature and widespread, mainly in the geothermal heat pumps sector.

In order to increase the use of heat pumps it is very important to put effort in research and improve the efficiency and reliability of the technology.

The project ‘Smart Fault Detection and Diagnosis (SFDD) for heat pump systems’, started in the energy department of KTH in cooperation with Folksam, one of the largest insurance companies in Sweden, and also some of the main Swedish heat pump manufacturers. The project aims at improving the performance of the systems, and reducing maintenance times and costs.

The aim of this work, being the first step of the project, was to study the different techniques applied in fault detection and diagnosis in general systems, and more in detail about heat pump systems. Fault detection of heat pumps can be performed with different methods. The most applied are: data- driven methods, analytical methods, and knowledge based methods.

Advantages and drawbacks of the different techniques and examples of application have been studied. The conclusion that can be drawn is that methods for fault detection based on analytical models outperform the other cited techniques, whenever a good model of the system is available.

The second step of the work was to build a statistic database of faults based on data provided by the insurance company Folksam and Swedish heat pump manufacturers. The faults in heat pumps were categorized on the basis of the type of system (air-to-air, air-to-water, brine-to-water, exhaust air) and different statistics were built in order to highlight the most problematic components in terms of number of faults, total cost per fault, and average cost of the single fault or faulty component.

The results were discussed with each company taking part in the project before building the final statistics. Most of the results were coherent among the different manufacturers as well as with their knowledge and experience about their products. For all types of systems, compressor and electronic problems are prevailing,

while the most expensive components are water tanks, compressors and evaporators. Although some reported data were incomplete, the problem was compensated by a large number of data collected in order to make the statistic more valuable, and the results proved to be reasonable according to experienced experts. Since there are some confidential issues regarding the faults provided by the heat pump manufacturers, the faults provided by the heat pump manufacturers are not presented in this study and they will be presented in the later stages of the project.

7. ACKNOWLEDGEMENTS

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APPENDIX A: EQUIPMENT MANUFACTURER PERSPECTIVE

A Comprehensive Study on the Important Faults in Heat Pump System During the Warranty Period

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ABSTRACT

The heat pump market has become mature in many countries. There are millions of heat pumps installed worldwide. So any improvement in the installation, operation, and maintenance of heat pump systems can save a considerable amount of energy and cost, and reduce Green House Emissions to a large extent. The present study suggests a Smart Fault Detection and Diagnosis (SFDD) mechanism as the heart of the next generation of heat pumps. A SFDD mechanism can minimize the installation and control errors, decrease the performance degradation during operation, avoid unnecessary visual inspections and components replacement, and reduce the maintenance cost and down-time of the system. To develop a SFDD mechanism, the first essential step is to obtain knowledge about the common and expensive faults in the heat pumps. The heat pump manufacturers are one of the best sources to find out the most common and costliest faults occurred recently in the heat pump systems. The present paper describes the results from a comprehensive study done on the most recent faults which were reported to some of the heat pump manufacturers in Sweden. The most common and the costliest faults in the Air-toair, air-towater, brine-to-water, and exhaust air heat pumps are presented. Some of the faults such, as faulty pressure switches or fans, are only related to the heat pump unit, i.e. the thermodynamic cycle which facilitates the heat pumping cycle. Some of the common and expensive faults, such as faulty shuttle or shunt valve, are related to the faulty components in the heating systems. Generally, the results show that faults in Control and Electronics are the most common and costliest faults in all types of heat pumps. Faults in Control and Electronics include any fault related to control unit, electrical faults (such as short circuit, etc.), PCB, display, soft starter, overcurrent and motor protection relay, etc.

1. INTRODUCTION

Heat pump technology provides an efficient and sustainable solution for heating and cooling of the buildings which could be otherwise heated by oil or gas boilers or pure electrical heater. When the heat pump operates in a healthy condition, there is great opportunity to save a large amount of energy and cut the global CO2 emissions. Bettgenhäuser et.al (2013) predicts that the CO2-eq emissions will be reduced by 40% if 50% of all new buildings and 30% of retrofitted buildings are equipped with heat pumps by 2030.

There are millions of heat pumps installed worldwide. In Sweden alone, it is estimated that over one million heat pumps have been installed. The market value of the heat pumps installed in Sweden alone is estimated to be approximately 1 billion euro, excluding the maintenance cost (Forsen, 2012). The market share of heat pumps for the current and newly built single family houses in Sweden is 50% and 90%, respectively (Forsen, 2012). So any improvement in the installation, operation, and maintenance of heat pump system can save a considerable amount of energy and cost, and reduce Green House Emissions to a large extent.

However, it is not uncommon for the heat pump systems to operate with lower system efficiency than the typical efficiencies measured at standard conditions in manufacturers' laboratories. Sometimes, a trivial installation or control error can lead to the performance degradation of the whole system to a large extent.
Furthermore, there have been several heat pump faults reported to the insurance companies or heat pump manufacturers during the recent years. A considerable number of these faults are associated with confusion for the servicemen, long down-time for the system, and unnecessary component replacement. These consequently might end up with some unnecessary costs for the end-customer, heat pump manufacturer, and insurance company which could have been avoided if smart procedures were used.

Therefore, a Smart Fault Detection and Diagnosis procedure (SFDD) is suggested by the author as the solution to the problems mentioned. A SFDD system as the heart of a smart heat pump is able to minimize the installation and control errors, decrease the performance degradation during operation, avoid unnecessary visual inspections and components replacement, and reduce the maintenance cost and down-time of the system.

FDD for heat pumps or vapor compression chillers was investigated by (Zogg 2002; Zogg et. al. 2006; Chen and Lan 2009; Kim et. al. 2008 & 2009; Navarro-Esbri et. al. 2006; Choi et. al. 2012; Rossi and Braun 1997; Li and Braun 2007). For example, Zogg (2002) and Zogg et al. (2006) presented a FDD system for heat pumps based on a gray-box process model whose parameters are identified online. The faults are classified from the parameters using ‘‘vector clustering’’ technique. Chen and Lan (2009) developed a PCA-based (Principal Component Analysis) fault detection method to detect the faults in air-source heat pump. Kim et al. (2009) implemented seven artificial faults on a R410A residential unitary split heat pump operating in the cooling mode. The study monitored several fault detection features and identified the most sensitive features for each fault (Kim et al., 2009).

To develop a SFDD mechanism, the first essential step, which is usually missing in the previous studies, is to obtain knowledge about the most common and costliest faults in the heat pumps. To design an efficient SFDD system, it is very important to know what usually causes the faults in the heat pump system and what kind of faults are expected to be detected and diagnosed by a SFDD system. The heat pump manufacturers are one of the best sources to find out the most common and costliest faults occurred recently in the heat pump systems. The present paper describes the results from a comprehensive study done on the most recent faults which were reported to some of the heat pump manufacturers in Sweden.

2. METHODOLOGY

Approximately 68000 fault reports, sent to several heat pump manufacturers during the warranty period, were collected and processed. Some of these reports were real faults in the system and some of them were complaints, for example about the noise or leakage in the system. About 8600 fault reports are not clear enough to be analyzed; about 4300 fault reports do not mention the type of the heat pump which makes it hard to interpret; so these unclear and uncompleted fault reports are excluded from the study. It is very important to know the faults in the most recent models of heat pumps. The old fault reports usually refer to the problems which are already known and solved during the recent years; so approximately 18000 faults which were reported before 2010 are also excluded from this study. Therefore, about 37000 faults reported to the Heat Pump Manufacturers, called OEMs, from the beginning of 2010 to the end of 2012 were analyzed and the results are presented in the following section (see Table KTH-3).

Table KTH-3: Number of Faults Analyzed in the Present Study

Total number of fault reports received	Number of fault reports with missing key info	Number of fault occurred before 2010	Total number of faults analyzed in this study
67952	12984	17895	37037

While the number of brine-to-water heat pumps i.e. Ground Source Heat Pumps (GSHPs) which were sold in Sweden between 2010 and 2012 are considerably higher than the air-to-air or air-to-water heat pumps, the number of fault reports in the present study is the highest for the air-to-water heat pumps. It should be mentioned that the number of faults reports studied in the present paper cannot give any indication about the quality of different types of heat pumps. Due to the fact that the present study does not cover all the faults reported to the heat pump manufacturers in Sweden, it is not possible to say in general which type of heat pump has the highest or lowest number of faults.

3. RESULTS

3.1. Air-to-Air Heat Pump System

In Sweden, over one hundred thousand air-to-air heat pumps were sold during 2010-2011 (two years). This type of heat pump is also very common in North America due to the large number of buildings which are renovated where the old gas furnace is replaced by an air-to-air heat pump. Since air-to-/air HPs just use the outside air as the heat source and do not need any heating water distribution system, they are cheaper and easy to install, and popular, particularly in the mild climates for both heating and cooling applications.

Figure KTH-28: shows the number distribution of different faults that occurred in air-to-air heat pump systems. The most common faults reported to HP manufacturers are faulty fan (26 %), control/electronics (25 %), and faulty temperature sensors (16 %). Fans were very frequently reported as the faulty component in OEMs fault database though the cost for repair or replacement was relatively low (362€ on average).

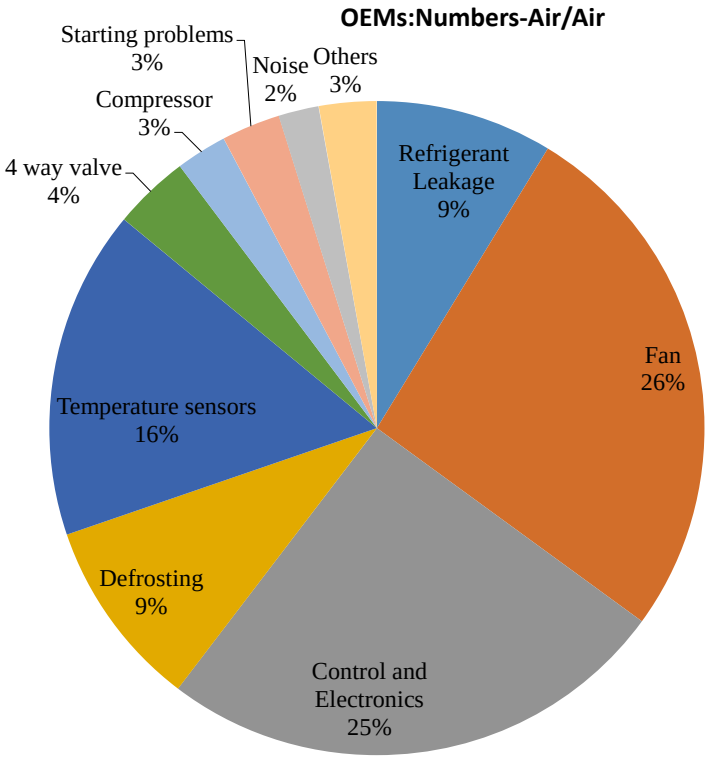


Fig. KTH-28: Air-to-Air Heat Pumps – the Share of Different Faults from the Total Number of Faults Reported to Heat Pump Manufacturers

Usually, noise is one of the main issues regarding the indoor fan. Regarding the outdoor fan, usually ice formed on the evaporator leads to the fan damage; a malfunctioned control unit or sensor may lead to inappropriate defrosting and the ice formed on the evaporator can break the outdoor fan. The control/electronics faults mainly correspond to faulty PCB (Printed Circuit Board) and control card.

The temperature sensors are usually used to measure indoor, outdoor, or air temperature for defrosting control. Usually, the temperature sensors which are exposed to the outside air are more likely to become faulty. The air moisture can enter and freeze in the temperature sensor and result in the sensor damage. The rapid temperature change during defrosting can be another reason for the temperature sensor's failure. Furthermore, bad wiring can also be another reason for some of these faulty temperature sensors.

Fig. KTH-29 shows the cost distribution of faults reported to OEMs. Control/electronics, refrigerant leakage and faulty fan imposed the highest total costs among the different faults reported to OEMs. Surprisingly, as shown in Fig. KTH-29, according to OEMs fault database, temperature sensors imposed higher costs than the compressors during the last three years (2010-2012).

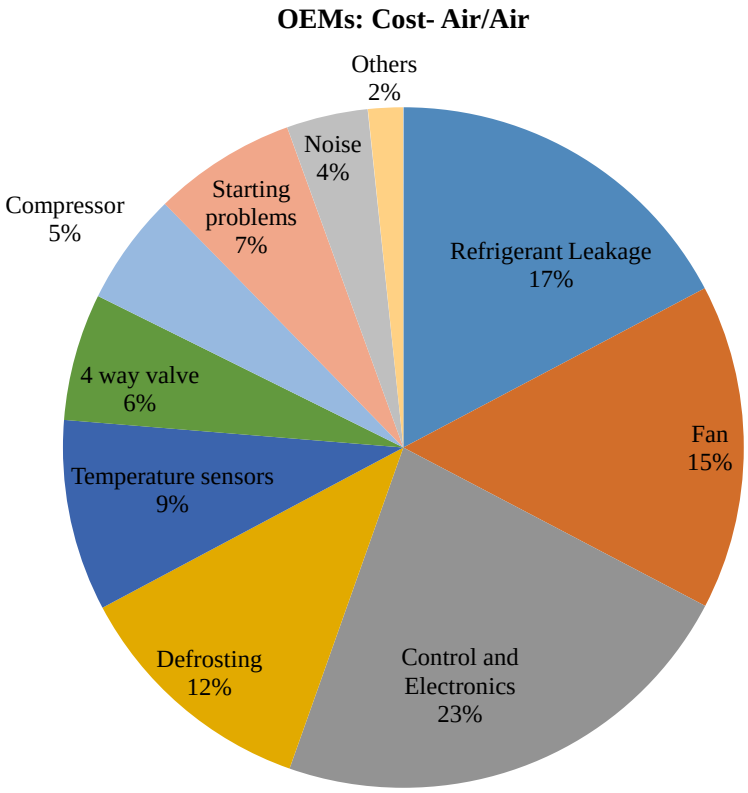


Figure KTH-29: Air-to-Air Heat Pumps–- the Share of Different Faults from the Total Cost of Faults Reported to Heat Pump Manufacturers

Defrosting, starting problem and noise are symptoms which were wrongly reported by servicemen as the cause of heat pump fault/failure. A defrosting problem can be caused by a faulty temperature sensor, four-way valves, or control unit. A starting problem is a general symptom meaning that the heat pump system could not be started after installation. This type of problem which leads to replacement of the HP unit by a new one should be obviously investigated more in detail to find what the causes were. The fault reported as "noise" can be caused by a compressor which is mounted improperly, by an air filter which is blocked, by a fan which hits the ice formed on the evaporator, by a whistling valve which cannot be closed completely, or by wear in the motor. Sometimes, the normal sound from the unit, for example in defrosting periods, is reported as fault by the people who are very sensitive to the noise.

It is worth mentioning again that the faults shown in Fig. KTH-28 and Fig. KTH-29 are the faults reported to the OEMs, not essentially the root cause or the real fault occurred in the machine. As mentioned before, some of them such as defrosting problem, starting problem, or noise are only symptoms; the root cause is located somewhere else. Some components which were introduced by servicemen as the fault such as fan or compressor can be either the root cause or only a victim of another fault in the system. Some of the faults reported by the servicemen such as defrosting problem, faulty four-way valve, temperature sensor, or control/electronics are highly inter-connected and they cannot be completely separated or distinguished from each other.

3.2. Air-to-Water Heat Pump System

In Sweden, over 20000 air-to-water heat pumps were sold during 2010-2012 (three years). Fig. KTH-30 and Fig. KTH-31 present the number and cost distribution of faults, which occurred in air-to-water heat pump systems respectively. As shown in Fig. KTH-30 and Fig. KTH-31, pressure switches (pressostats), control / electronics, temperature sensors, and compressors are the most common and costliest faults in air-to-water HPs reported to OEMs between 2010 - 2012.

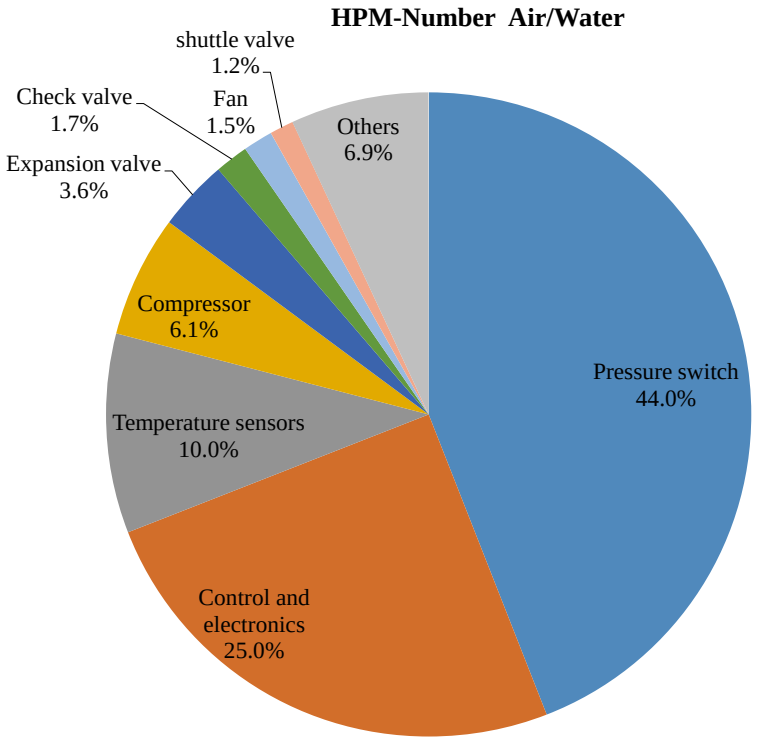


Figure KTH-30: Air-to-Water Heat Pumps: The Share of Different Faults from the Total Number of Faults Reported to Heat Pump Manufacturers

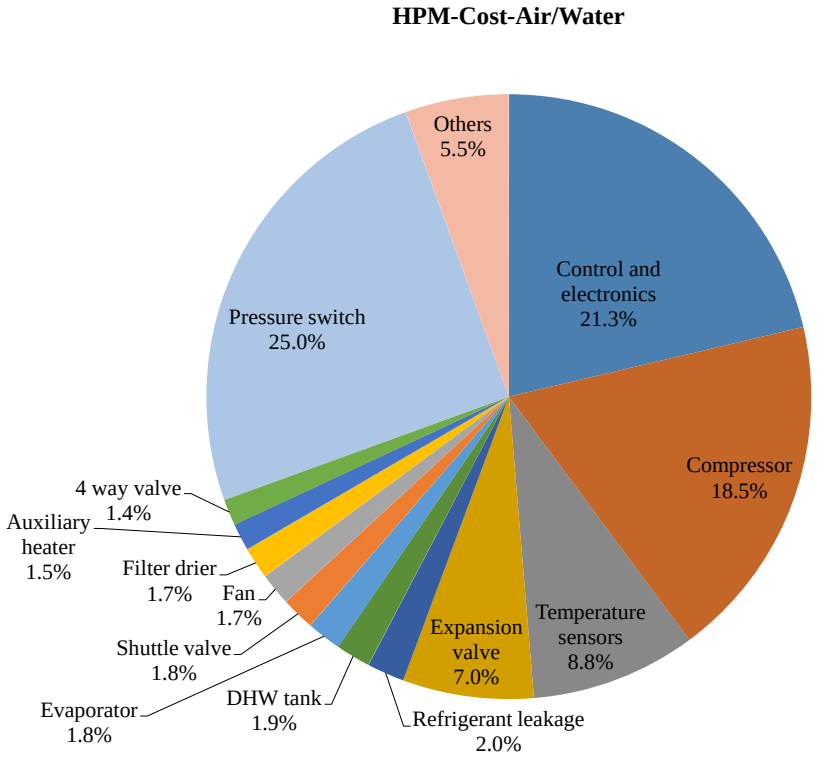


Figure KTH-31: Air-to-Water Heat Pumps: the Share of Different Faults from the Total Cost of Faults Reported to Heat Pump Manufacturers

As shown in Fig. KTH-30 and Fig. KTH-31, 44% of the faults number and 25 % of the faults cost is devoted to pressostats (pressure switches). There are two pressure switches, one at low and one at high pressure sides of a HP unit, located before and after the compressor. Some of them are the cheap low-quality switches which are vulnerable or very sensitive to the dusts in the air; so they can alarm for example about too-low pressure, whereas too-low pressure does not exist in the reality. An improper defrosting can lead to too-low

evaporation pressure and consequently the pressure switch at the low pressure side can react to too-low pressure. Furthermore, some other faults such as faulty compressor can result in too-high pressure and this involves the pressure switch at high pressure side. Moreover, high vibration at the discharge line right after the compressor may also lead to the fault in high pressure switch. So in many cases, pressure switches are not the cause of fault report; but they only react to too-high or too-low pressures caused by other faults in the system.

Control/electronics faults include any fault related to the control unit, PCB, display, soft starter, overcurrent and motor protection relay, contactor, short circuits, etc. The faults in temperature sensors were already discussed in 3.1. In addition to the sensors mentioned before, in the air/water HP systems, there is a temperature sensor measuring the supply or return temperature from the heating distribution system (radiator or floor heating) which can become faulty. Regarding faults in the expansion valve, generally when a thermostatic expansion valve is used in the HP unit, the thermostatic expansion valve can be wrongly adjusted which can lead to too low superheat temperature and "hunting" can occur in the HP unit and make the HP unit unstable.

3.3. Brine-to-Water

Brine-towater heat pumps i.e. ground source heat pumps are the most common type of heat pump installed in Sweden. There are more than 400000 brine-to-water heat pumps installed in Sweden. During 2010-2012, more than 80000 brine-to-water heat pumps were sold in the country. Fig. KTH-32 and Figure KTH-33 present the number and cost distribution of faults occurred in brine-to-water heat pump systems, i.e. GSHP systems respectively. As shown in Fig. KTH-32 and Fig. KTH-33, Control/Electronics, liquid pumps, and shuttle valves are the most common and the most expensive faults occurred in brine-to-water HPs during the last three years (2010-2012).

Control/Electronics include any fault related to a control unit, electrical faults (such as short circuit, etc.), PCB, display, soft starter, overcurrent and motor protection relay, mobile application, online services, etc. The shuttle valve, which is the three-way valve switching between space heating and DHW lines, is the second most common fault reported to OEMs. Dirt and particles in the heating water, particularly in the old systems, are very common root cause of the faults in shuttle valves.

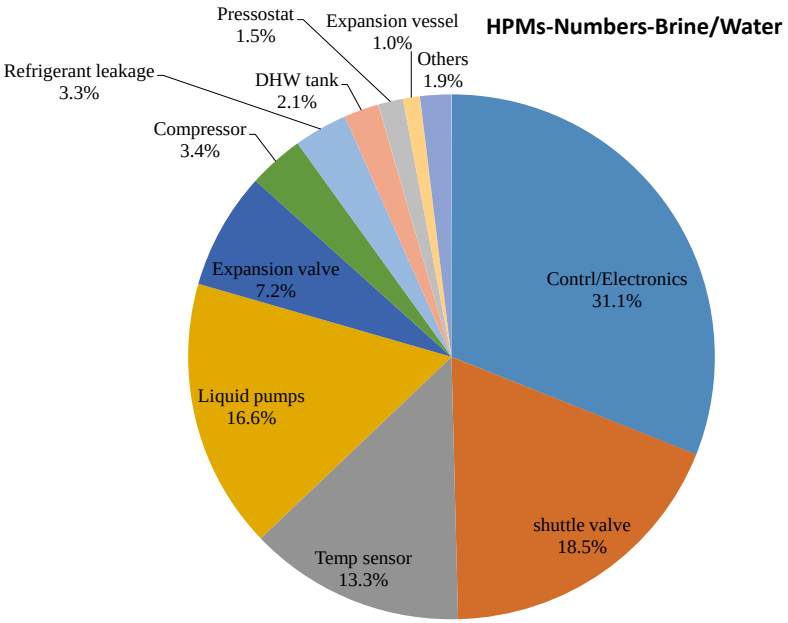


Figure KTH-32: Brine-to-Water Heat Pumps: the Share of Different Faults from the Total Number of Faults Reported to Heat Pump Manufacturers

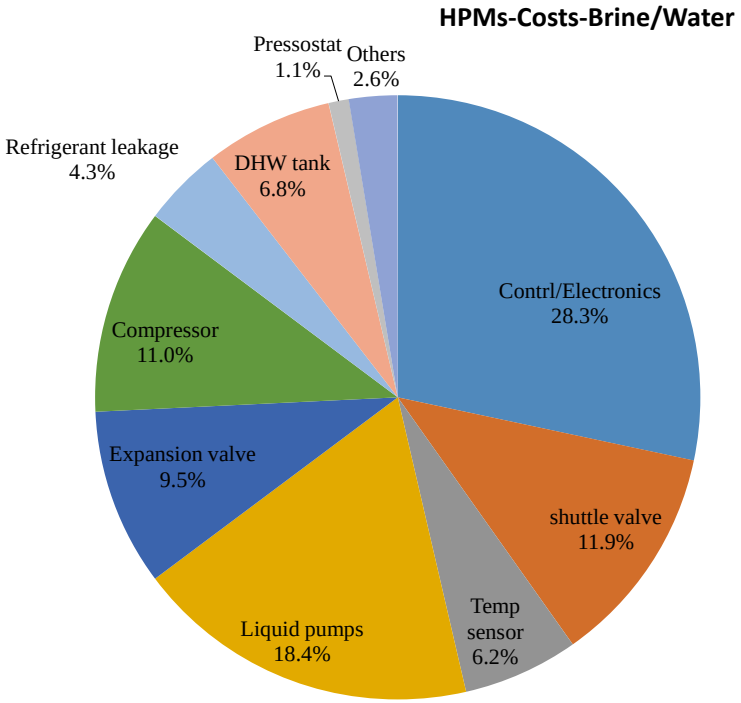


Figure KTH-33: Brine-to-Water Heat Pumps: the Share of Different Faults from the Total Cost of Faults Reported to Heat Pump Manufacturers

In the brine-to-water HP systems, the brine pumps beside the pumps in the heating distribution system can cause system failure. This makes the liquid pump as the second costliest faults, with about 18 % share of total costs reported to OEMs. Some of the brine pumps have failed to operate due to the frost and/or leakage in the brine loop; some of the brine pumps had problem at startup and some of them alarmed about fault whereas there was not any real fault in the pump.

Furthermore, expansion valves and temperature sensors are among the most common and costliest faults reported to OEMs. The problems in expansion valves can be due to wrong adjustment or noise. Beside the other common temperature sensors in a heat pump, in the brine-to-water heat pump systems, there are usually two temperature sensors before and after the ground heat exchanger, close to the evaporator. The moisture entering the sensors can damage them and cause faults in the systems.

3.4. Exhaust Air-to-Water Heat Pump

During 2010-2012 approximately 30000 Exhaust air-to-water heat pump systems were sold in Sweden. Fig. KTH-34 and Fig. KTH-35 present the number and cost distribution of faults occurred in exhaust air heat pump systems, respectively. As shown in Fig. KTH-34, Control/Electronics, Shunt valve/motor, and temperature sensors are the most frequent faults reported to the OEMs. Furthermore, Control/Electronics, refrigerant leakage, and DHW tanks are the costliest faults reported to the OEMs during the last three years (2010-2012).

The shunt valve, which is both a frequent and expensive fault in the exhaust air heat pump, is a motorized valve that continuously regulates the heat supply between the heat pump and the auxiliary heater. As shown in Fig. KTH-34, refrigerant leakage is not a very frequent fault (3.6 % of the faults number); but the cost imposed by the fault is so high that makes the leakage the second most expensive fault reported to the OEMs. Similarly, leakage in DHW tank which is very expensive fault occurs so rarely that it cannot be seen in the figure of faults number (Fig. KTH-34). The faults called “pipes or couplings”, shown in Fig. KTH-34 and Fig. KTH-35, are usually due to the leakage in the pipes and couplings which can be in the heating distribution system or between the tank and heat pump unit. Control/Electronics include any fault related to

control unit, electrical faults (such as short circuit, etc.), PCB, display, motor protection relay, etc.; however, it excludes the faults in the shunt motor/valve.

Compressors faults have a relatively large share in both number and cost of the faults. Noise is one of the most common causes of the fault reports in exhaust air heat pumps. Sometimes, blockage in the air filter leads to too low air flow in the evaporator and consequently too low evaporation temperature and pressure. This can put the operating point out of operating envelope of compressor and consequently can damage the compressor if the compressor does not have any proper protection. Similar to outside air source heat pumps, a faulty fan is common in exhaust air heat pumps. This can stem from noise or improper control of fans which are usually variable-speed ones.

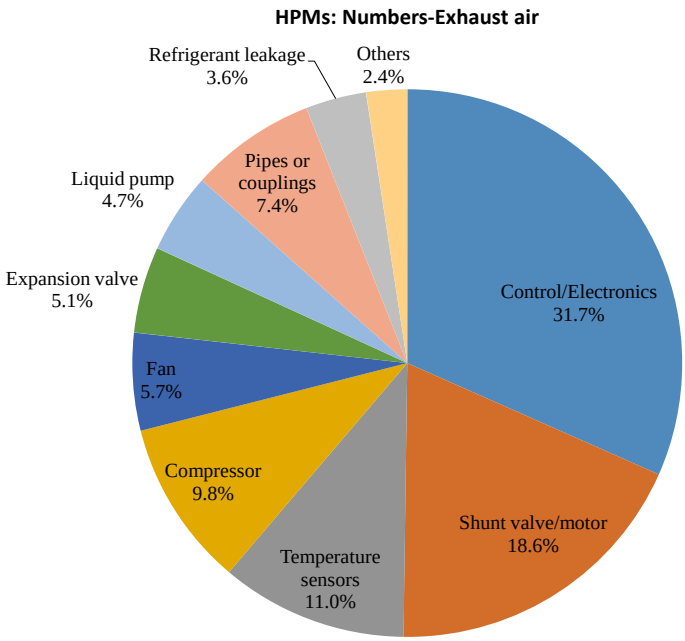


Figure KTH-34: Exhaust Air-to-Water Heat Pumps: the Share of Different Faults from the Total Number of Faults Reported to Heat Pump Manufacturers

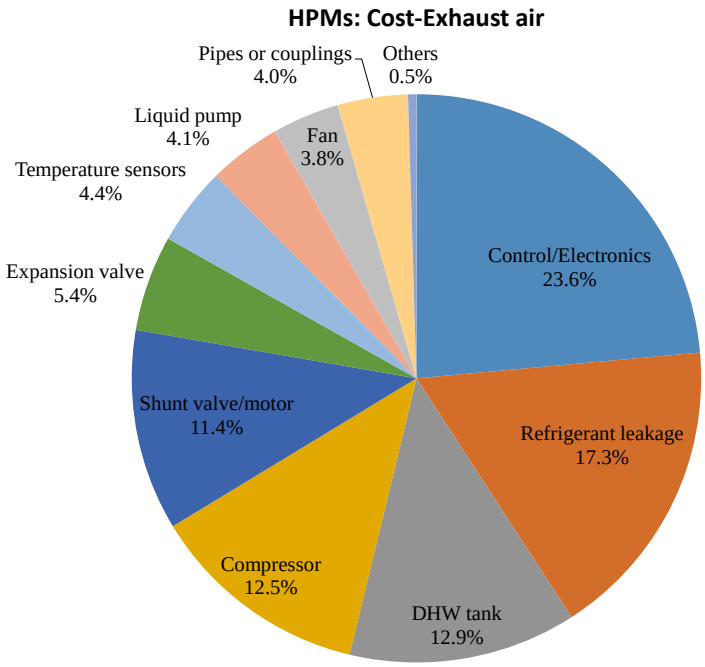


Figure KTH-35: Exhaust Air-to-Water Heat Pumps: the Share of Different Faults from the Total Cost of Faults Reported to Heat Pump Manufacturers

4. CONCLUSION

Table KTH-4 presents a summary of the most common faults and the faults with the highest total costs which were reported to the heat pump manufacturers from 2010 to the end of 2012 (three years).

Table KTH-4: A Summary of the Most Common and Costliest Faults in Different Types of Heat Pump Systems - According to the Reports to Heat Pump Manufacturers During 2010-2012

	Type of Heat Pump (HP)			
	Air-to-Air HP	Air-to-Water HP	Brine-to-Water HP	Exhaust air HP
The most common faults	Fan (26%)	Pressure switch (44%)	Control and Electronics (31%)	Control and Electronics (32%)
	Control and Electronics (25%)	Control and Electronics (25%)	Shuttle valve (19%)	Shunt valve/motor (19%)
	Temperature Sensors (16%)	Temperature sensors (10%)	Liquid pumps (17%)	Temperature sensors (11%)
The costliest faults	Control and Electronics (23%)	Pressure switch (25%)	Control and Electronics (28%)	Control and Electronics (24%)
	Refrigerant Leakage (17%)	Control and Electronics (21%)	Liquid pumps (18%)	Refrigerant leakage (17%)
	Fan (15%)	Compressor (19%)	Shuttle valve (12%)	Domestic Hot Water tank (13%)

The results presented in this study show that faults in Control and Electronics are almost the most common and costliest faults in all types of heat pumps. Faults in Control and Electronics include any fault related to control unit, electrical faults (such as short circuit, etc.), PCB, display, soft starter, overcurrent and motor protection relay, etc. To conclude, in order to reduce the number and cost of the faults in the heat pump systems, it is essential to pay special attention to the control and electronics in heat pump systems.

As shown in Fig. KTH-36, some of the faults such as faulty pressure switches or fans, are only related to the heat pump unit (i.e., the thermodynamic cycle which facilitates the heat pumping cycle). A lot of the frequent and expensive faults are related to the faulty components in the heating systems. For example, about one-fifth of the fault numbers in GSHPs are related to the faulty shuttle valve, a simple three way valve which switches between the space heating and domestic hot water application. Some other faults, such as faults in control and electronics or temperature sensors, can be related either to the heat pump unit or the heating system. For example, if the faulty temperature sensor is the one which measures the ambient temperature in order to control the supply temperature of the heating water, the system control will not operate properly, and a system failure or significant performance degradation can occur.

The faults reported to the heat pump manufacturers are usually the faults occurred within the warranty period; most occur when the heat pump age is less than a few years (usually, less than 3 or 5 years). So the characteristics of these faults can be different from the faults reported when the equipment is out of warranty, such as the faults related to the components ageing, or the faults stemming from the performance degradation over a long time.

Some of these fault reports are not real faults in the system but only complaints from the end-users; for example, about the noise in the fan. On the other hand, there can be some serious performance degradations in the heating systems which were not reported to the HP manufacturers. Usually, the performance degradation in the system is not found until it leads to system failure.

As a final conclusion, the number and cost of faults in heat pumps is not high when it is compared to the market value of the heat pumps. Furthermore, heat pump technology provides an efficient and sustainable solution for heating and cooling of the buildings. When the heat pump operates in a healthy condition, there is great opportunity to save a large amount of energy and cut the global CO2 emissions. However, there is still a huge potential for improvement of heat pump efficiency and for reduction in the number and cost of the faults which occur in the system. A lot of faults that occurred in the system can be avoided by doing some minor changes in the whole system. For example, using higher quality pressure switches, shuttle valves, or temperature sensors does not add so much to the cost of the system but it reduces the number of

faults considerably. Furthermore, a Smart Fault Detection and Diagnosis (SFDD) system can be designed to detect and diagnose the possible faults in installation and operation phases and to reduce the maintenance cost and down time of the system.

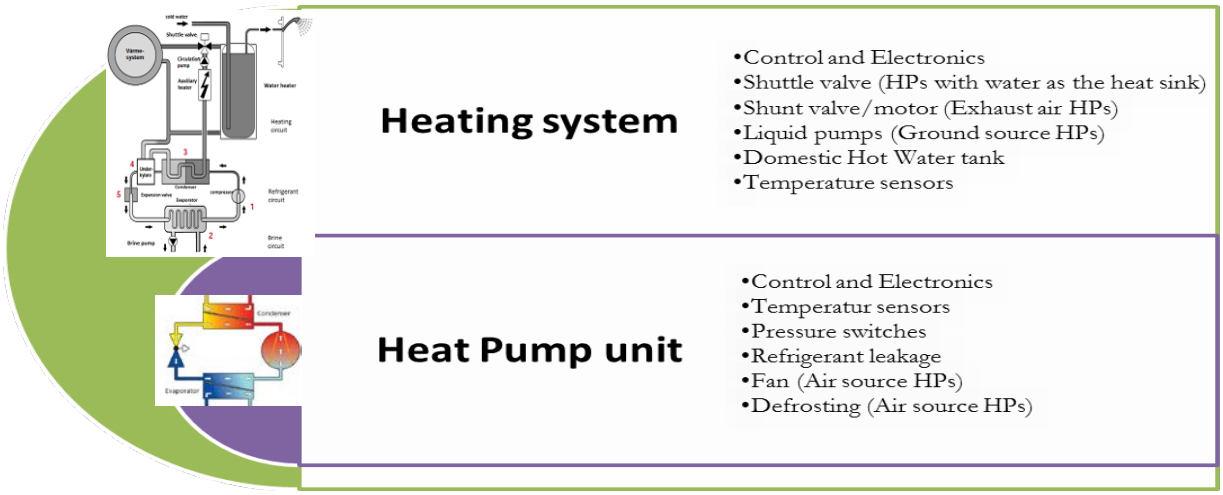


Figure KTH-36: Important Faults Reported to OEMs at Two Different Boundary Levels: *Heat Pump Unit and Heating System Levels*

5. ACKNOWLEDGEMENTS TO APPENDIX A

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COUNTRY REPORT:
SWEDISH (2) CONTRIBUTION TO THE IEA HEAT PUMP PROGRAMME ANNEX 36 ON QUALITY OF INSTALLATION AND MAINTENANCE OF HEAT PUMPS

Operation and Maintenance of Heat Pumps in Apartment Buildings Owned by Smaller Property Companies

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ABSTRACT

This paper presents the results of SP's programme of work to improve the quality of installation and maintenance of heat pump systems in Sweden. It focuses on heat pumps serving apartment buildings in Sweden, and in which heat pumps are used mainly for space heating and domestic hot water heating. The paper describes three studies:

1. That part of the EFFSYS+ project "Operation and Maintenance of Refrigeration and Heat Pump Systems" focusing on heat pumps installed in Swedish apartment buildings and non-residential premises.
2. An English summary of a study focusing on experience from operation and maintenance of larger heat pumps and research in the heat pump field in Sweden during the 1980s.
3. The study "Improved Reliability of Residential Heat Pumps" made on behalf of the research fund of Länsförsäkringsbolagen, focusing on heat pumps installed in Swedish single-family houses.

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1. INTRODUCTION

Development of larger heat pumps was skyrocketing in Sweden in the late 1970s and beginning of the 1980s. These were machines with thermal capacities of 100 kW up to a few MW, and many were installed in district heating systems, in larger buildings, and in industrial processes. The number of new installations stagnated in the second half of the 1980s, but operational experiences from this expansive period in Sweden has been gathered and recorded. In parallel with this, at the beginning of the 1980s, the Swedish market for heat pumps for single-family houses was growing rapidly. Today, this market segment is mature, with the millionth heat pump being installed in 2010 in Sweden, in a total single-family housing stock of about 1.7 million (SVEP, 2013).

Now the market for heat pumps for apartment buildings and commercial premises is expanding. SP is operating an ongoing project investigating operation and maintenance of these types of heat pumps (EFFSYS+, 2013), the main goals of which are to:

- Increase heat pump owners' knowledge and awareness, to improve operational and maintenance performance and standards.
- Identify and analyse the three most common failures for the heat pumps.
- Suggest profitable preventive maintenance methods to protect against each of these failures.

The objective of the work is to maintain the installed capacity and efficiency of heat pumps through appropriate maintenance methods, not to modify the installations. The project will be concluded in June 2014.

2. SCOPE AND METHODS

The scope of SP's contribution to IEA HPP Annex 36 consisted mainly of the results of the "Operation and Maintenance of Refrigeration and Heat Pump Systems" project, which focused on heat pumps serving multi-family houses and commercial and other premises (EFFSYS+, 2013). Two other studies have been presented within the framework of IEA HPP Annex 36 concerning issues regarding operation and maintenance of heat pumps. The first study summarises experience and research in the heat pump field in Sweden during the 1980s. The second is a study from 2012 that examined common failures of heat pumps in Swedish single-family houses and how to reduce the number of failures.

2.1. Operation and Maintenance of Refrigeration and Heat Pumps Systems

The objective of the "Operation and Maintenance of Refrigeration and Heat Pump Systems" project was to identify the three to five most common problems with heat pump installations and suggest maintenance methods to prevent at least three of them.

The first part of the project was to analyse conventional operation and maintenance principles and to identify concepts that can be applied to heat pumps to prevent common failures. The second part of the project was to examine common and expensive failures of heat pumps servicing multi-family houses and commercial premises in Sweden. One indirect cost of problems with heat pumps is that for the use of a back-up heating system when the heat pumps are unable to operate properly because of failures or poor adjustment. The third part of the project was to recommend cost-effective maintenance methods to prevent three of the identified failures. The method used to identify common failures and problems was to interview personnel working with heat pumps in property companies. A reference group of representatives from property companies, interest groups and designers was also a part of the project. Outcomes of the projects, ideas, problems and experiences were discussed in meetings with the reference group.

2.2. Large Heat Pumps – Research and Development during the 1980s in Sweden

During the first years of the 1980s, about 130 000 residential (i.e. small) heat pumps were installed in Sweden (VAST, 1987). Installations of large heat pumps, with outputs of 1 MW or more, were numerically few in comparison but more impressive in their thermal capacity. About 100 plants and 130 units were installed, creating about 1300 MW of thermal capacity; far more than the total capacity of the small heat pumps installed in the residential sector. This was also a period of major research and development in the heat pump field in Sweden. A literature review of operational experience of large heat pumps and research from the 1980s was performed in order to summarise and disseminate some of the knowledge acquired. Some of this information was available only in Swedish, but an English summary report of the literature review was written.

2.3. Improved Reliability of Swedish Residential Heat Pumps

Today, heat pump heating systems are common in Swedish single-family houses. Most owners are pleased with their installation, but statistics show that a certain number of heat pumps fail every year, resulting in high costs for both insurance companies and owners. On behalf of Länsförsäkringsbolagens Forskningsfond (Länsförsäkringsbolagens Research Fund), SP Energy Technology has studied the cause of the most common failures for residential heat pumps. The objective of the study was to suggest what measures should be taken to reduce the number of failures, i.e. to improve the reliability of heat pumps. The methods used were analysis of public failure and sales statistics, and interviews with heat pump manufacturers, installers, service representatives and insurance assessors.

Based on interviews, the most common failures were categorised by whether they:

- could have been prevented by better operation and maintenance;
- were caused by a poor quality of installation workmanship;
- could have been prevented if certain parameters had been measured, recorded and followed up;
- were due to poor quality of components or systems.

The study was described in a report written in Swedish (Hauglund Stignore and et. al, 2012). A summary of the study has been presented in English within the IEA HPP Annex 36.

3. INTERVIEWS WITH PROPERTY COMPANIES OWNING HEAT PUMPS

Five property companies that own heat pumps were visited and interviewed during the autumn of 2011 and the spring of 2012. Tenants of the companies were apartment buildings, schools, commercial premises and offices. The heat sources of the heat pumps were mainly geothermal (boreholes) or exhaust air, but a few used outdoor air as a heat source. The thermal capacities of the heat pumps ranged from 10 kW to 1.8 MW. All of the heat pumps were used for both space heating and domestic hot water production. Some of the heat pumps were integrated in systems incorporating air conditioning and food storage refrigeration. The major parts of the heat pumps had been installed from 2000 and in existing buildings, with installation based on economic or environmental policy grounds. In general, operation and maintenance services were outsourced.

The properties owned by the companies used district heating, electric heating, or oil-fired boilers as secondary heat sources for peak loads and back-up. The degree of detail of the electricity and heat measurements varied from one site to another. Some measured only the total electricity supply to the buildings, while some were as detailed as measuring the electricity supply to the compressors of the heat pumps. The control and supervisory systems were also diverse, with some having IT systems and others having only the heat pump displays for control and supervisory purposes.

3.1. Problems Experienced by the Interviewed Companies and the Reference Group

The persons interviewed and the reference group in the project did not feel that costs for service and/or failed components were the main problems with operating heat pumps. Instead, the difficulties were seen to be how to operate the heat pump systems and efficiency issues:

- to operate energy-efficiently; what is the efficiency of the system, and how should the efficiency of the heat pump be indicated?
- to maintain the installed performance; knowing the performance when the system was installed and commissioned, and how to maintain it;
- operational priorities: for example, when the heat pump is integrated in a system delivering space heating, domestic hot water heating and air conditioning;
- control and supervisory systems: what to monitor, and alarm limits.

The main issue is to operate the system energy-efficiently and to maintain good performance of the heat pump. The heat pumps were considered as reliable, but sensitive to operating conditions and requiring a completely new basis of knowledge in terms of design and operation compared to, for example, oil-fired boilers. Many of the operational troubles and lack of performance can be traced back to the design and commissioning stages, according to the interviews. The main issues were described in the interviews as lack of:

- *Procurement competence – Knowledge of heat pumps in the owners' purchasing group*
  The heat pump owners found that they have to rely on suppliers' knowledge, information and goodwill when purchasing heat pump systems. The owners have difficulty in expressing their requirements in technical specifications due to a lack of knowledge and experience. They need to be able to determine the required performance and quality and describe it in the contract in order to reduce the risk of ending up with a poor performance system.
- *Designing a heat pump system for an existing building*
  It is hard to convert the heating demand of a building into an equivalent necessary installed thermal capacity of a heat pump. The companies interviewed reported both oversized and undersized heat pumps. Heat pumps need to be an integrated part of the building services system, which means that the heat pump designer must be aware of other technical fields, such as the effects of ventilation and radiator heat exchange areas for examples.
- *Coordinator of design and commissioning – Somebody with responsibility for the whole system*
  The different technical systems are supplied by different companies with different areas of specialisation, such as piping, ventilation, heat pumps and control systems for example. It is not unusual for there to be a gap in communication between the different system designers, with the result that these systems operate poorly together. The likelihood of over-dimensioned or under-dimensioned systems or parts of systems is reduced if somebody has overall responsibility for the whole system. Performance and operation of the entire building services system must be checked when the heat pump is taken into operation, and the parameter values must be documented.
- *Communication between the heat pump designer and the software engineer*
  To be able to create a well-functioning control and surveillance system, the heat pump designer must provide the software engineer with a full and exact description of how the heat pump is supposed to operate. The software engineer must have information on operating conditions and operating limits in order to determine the parameters that will control operation. The settings of these parameters must be decided by not later than the commissioning stage.
- *Systems descriptions and manuals for operation and maintenance*
  Basic function descriptions are often poor, while detailed operating and maintenance manuals may be missing entirely. Some of the owners of complex systems asked for operating prioritisations, when they have installed a system with functions such as space heating, domestic water production and air conditioning. There is also a need to know when the heat pump is operating well and efficiently, and when it is not. Performance verifications for a couple of operating conditions are required as references and for monitoring operating performance trends.
- *An effective interface of the control and/or supervisory system*
  The interface must be functional and easy to view for operators and owners. The parties interviewed also emphasised the importance of follow-ups and analysis of operation and performance of the heat pump systems. The control and/or supervisory system should have a follow-up function. It is important to define good key operational indicators and performance analysis.

Finally, there is a request for a forum for research, development and experience of heat pump design and operation. There is a need to develop design principles for heat pump systems and improve heat pump owners' knowledge of, and familiarity with, the systems. Better knowledge will make it easier to write

requirement specifications for heat pumps and for service agreements, to ensure good quality heat pump installations and operation.

3.2. Outcome of the Interviews

According to the interviews, the three main problems with the operation of heat pumps are:

1. Lack of design and performance details of building service system when purchasing, designing and commissioning heat pumps, as needed in order to be able to identify and specify the heat pump system requirements, and to design a heat pump system that suits the building.
2. Lack of adjustment and functional control of the whole building service system when starting up the heat pump.
3. Lack of knowledge of how to monitor and measure the performance of the heat pump system, and of how to control it during its operation.

The main result of the interviews was that heat pumps themselves are seen as reliable: that is, in terms of providing space heating and domestic hot water production. The issues are how to supervise and control the systems, and to make the entire building services system, of which the heat pump is an integral part, operate as needed and energy-efficiently. Knowledge of how to define requirements when purchasing heat pump systems and service agreements is needed in order to simplify future operation and maintenance. It is harder for very small property companies to gather this knowledge than it is for larger companies with greater resources and more heat pumps installed in their buildings. It was therefore decided to focus on small property companies and to write a manual on operation and maintenance of heat pumps serving apartment buildings. The next step was therefore to meet this category of heat pump owners to discuss their experience of heat pump systems' operation and maintenance. The main objective was to identify what type of information they need in order to assist their daily work with heat pumps. The method used was to invite heat pump owners to a workshop and discuss operation and maintenance of heat pumps.

4. WORKSHOP

A workshop was held for owners of apartment buildings already having heat pumps, as well as for those thinking of installing heat pumps in the future. Service companies, installers and designers of heat pumps were also invited. The workshop was held in Borås, Sweden, in October 2012, bringing together six apartment building owners and one representative of a trade organisation. The decision to focus on owners of small property companies was made because about 80 % of Swedish property owners own less than ten apartments and need support for operation and maintenance of heat pumps (Fastighetsägarna, 2012).

4.1. Discussion of Competence in Small Property Companies

In small property companies, it is common for the same person to be the owner of the building and the janitor. It can therefore be difficult for such companies to possess the necessary knowledge of, and the resources for, operation and maintenance of the physical building services system. The janitor needs at least to be able to identify that the heat pump is malfunctioning and to call the service company for support. He/she also needs to be able to determine whether the purchased heat pump system and its service support meet the requirements specified in the purchase contract. According to the workshop participants, an owner of a small property company must think twice when purchasing new equipment. The janitor and the service company need to be able to use and control the equipment. Training may be needed, and janitors' and service personnel's attitudes towards the system and its supervisory equipment are critical for the operation. The necessary depth of knowledge and interest of the personnel are essential for effective operation and maintenance. The owner of the building must also understand that maintenance is necessary, and have the resources to plan and perform or arrange for maintenance. In general, according to those attending the workshop, property companies have no understanding that preventive maintenance can reduce overall costs. In their view, once a heat pump has been installed, it should just operate without problems, with maintenance being regarded as a needless cost. In order to be able to promote and support preventive maintenance, it is necessary to record and analyse details of all failures, maintenance measures and costs, considering both technical and economic aspects.

4.2. Outcome of the Workshop

In general, the workshop participants were of the opinion that there are few problems with the heat pumps themselves. The main problem is to ensure that all the physical building services systems in the building deliver the required performance and operate energy-efficiently together. There is a need for means of controlling, monitoring and investigating the operation. There is also a need for templates for requirement specifications for system procurement and service agreements.

On conclusion of the workshop, the participants were asked to list what they needed to support them in their work with operation and maintenance of heat pumps. The following items were listed:

- In a heat pump procurement action, what to include in the requirement specifications regarding future maintenance. It is hard, for an organisation performing its first heat pump procurement, to define technical requirements and what to ask for.
- A protocol for the settings of the heat pump system to be used for commissioning. Documentation of settings is often poor, or not performed at all, during installation and commissioning. Each adjustable parameter and setting must be included in the protocol. In future operational problems, fault tracing will be easier if the original installation settings are known.
- A checklist for the property market of parameters to survey during operation.
- Create an operating log that can accompany the heat pump for documentation and communication. It may sound simple, but there is a need for clarification and instructions of who should write in the log, what must be documented and how, and finally, who shall perform follow-ups and analyse operational and maintenance results.
- Provide information on how to spot poor operating conditions that require the service organisation to be called for support. As an example, the electricity demand each month can be recorded and statistics may show when the electricity demand differs from the normal.
- Develop arguments to explain that maintenance is necessary, and that it can be profitable to replace some of the corrective maintenance with preventive maintenance. For example, the arguments can point out the costs that arise if the heat pump stops working and a back-up heating system has to be put into operation. This applies particularly if the back-up system has not been used for a long time.

The most basic and important request put forward by the participants was for information and knowledge of how to make the entire heating system in the building operate well. The first step to improve the system is to ensure that the whole system is working well: the second step is to optimise energy efficiency. The conclusion from the workshop was that there is a need for a manual for heat pump owners and janitors in smaller apartment buildings. The issues presented above must be addressed in the manuals, and conventional maintenance principles must be described.

5. MANUAL FOR HEAT PUMP OWNERS

The outcome of the EFFSYS+ “Operation and Maintenance of Refrigeration and Heat Pump Systems” project will be a maintenance manual for owners of heat pumps in apartment buildings, due for publication in June 2014. According to the interviews and the participants in the workshop and in the reference group, heat pumps are seen as reliable in terms of delivering space heating and domestic hot water production. Performance efficiency is not generally measured when the system is installed, and not monitored during normal operation, which means that there is no indication of how efficient and reliable the heat pumps are. One recommendation will be to monitor the performance of installed heat pumps.

5.1. Maintenance Work

As part of the work of the project, maintenance engineers working at a chemical plant, a paper mill and a steelworks were interviewed to discuss their organisation and maintenance work. Although their maintenance organisation structures differed, they were all concerned with preventing unplanned production line stops from occurring. Key indicators were used in order to evaluate both the reliability of parts of the production line and the maintenance departments’ own maintenance activities. These indicator factors covered both time efficiency and cost efficiency of work. Each of the maintenance organisations emphasised

that details of experience and knowledge of the systems and components should be documented, both for the production of statistics and for dissemination of results and information.

Maintenance work must be firmly founded on determination of the organisational structure, planning, documentation and follow-ups, all supported by methodical working. Maintenance objectives need to be decided within each organisation, based on the operational quality that is required and on the resources available. Heat pump owners will be recommended to consider what their maintenance objectives are. They must also consider, and try to quantify, the costs and risks resulting from a lack of maintenance.

5.1.1. Documentation and Maintenance Statistics

A central feature of the maintenance organisations that were interviewed was a tool and structure for planning, documentation and performance monitoring. The tool included the following elements:

- An inventory of equipment,
- Drawings,
- Object descriptions,
- Planning and preparation of activities,
- Documentation and results of activities,
- Spare parts lists,
- Documentation and analysis of costs for failures and maintenance activities,
- Operational follow-ups, quantified by key indicator factors.

A maintenance tool for smaller multi-family houses does not need to be as sophisticated as this. However, drawings, descriptions, service work performed, and parameter and setting values in the systems must be updated, documented and organised in order to assist day-to-day work and larger service measures. By documenting and analysing the cost for different failures and remedial activities, the owner can identify the most costly failures and focus on solving them. Documentation of the work performed will also serve as communication between the owner and the service personnel, as well as between different service technicians. Keeping drawings and system descriptions (hardware and software) are important not just be performance monitoring, but also for safety aspects. The time and cost for planning, preparing and performing service activities will also be reduced with up-to-date drawings and descriptions.

5.1.2. Risk Assessment

The maintenance engineers interviewed were actively concerned with risk assessment. Maintenance plans for the different parts and equipment of the process were based on how important they were for production. The overall objective of the maintenance was to prevent unplanned operational stops. Possible failures were therefore rated in respect of their seriousness, i.e. their potential for stopping production, using a system known as ABC-classification. Equipment or items that would stop production if they failed were classified as Class A. Class B objects were those that, in the event of failure, could be replaced by another object to prevent a stop in production. Class C objects were those that would not stop production if they failed. Safety and environmental aspects were also considered when deciding classifications. Failures of objects that could cause injuries or pollution must be prevented. Class A objects will receive mainly preventive maintenance, while Class C objects will receive mainly corrective maintenance. Corrective maintenance is maintenance that is performed when a failure occurs, while preventive maintenance is performed in order to prevent a failure from occurring.

For a smaller apartment building property-owning company, it may not be possible or necessary to perform a complete risk assessment and ABC-classification of all the components in the heating system. Nevertheless, there must be a risk assessment of the consequence of heat pump breakdown. Questions to ask are:

1. What is the cost of a heat pump system breakdown?
    a. Does a back-up heating system exist, and/or it is necessary to have a back-up heating system?
    b. What is the operating cost and the standby cost of a back-up heating system?
2. What is the maximum time for correcting a breakdown?
    a. How quickly must a service technician start to solve the problem after a breakdown has occurred?
    b. How much is it worth to reduce the reaction time in response to a breakdown?

3. Is it worth having a monitoring system, which can detect performance fall-off and possible breakdowns?
    a. What level of monitoring is needed?
    b. What is the maximum cost for such system?

5.1.3. Regular Inspections

Overhaul, i.e. regular inspections, is a preventive maintenance activity. Typically, the building janitor walks through the building and inspects the systems. This not only detects faults and performance fall-offs, but can also be used to perform planned and scheduled maintenance activities. Inspection improves knowledge of the building and its installed systems, and makes it easier to become aware of abnormal operation. The proposed maintenance manual will include a basic checklist of what the janitor can inspect in this way, and why.

5.2. Procurement

The reason for buying a heat pump system for heating rather than other heating systems is energy efficiency. A heat pump system is normally more expensive than other heating systems, but its higher efficiency should offset this extra investment cost. The required efficiency performance of the heat pump must be specified in the purchase contract, checked at commissioning, and be possible to monitor and maintain during the lifetime of the installation.

A heat pump is dependent on the operating modes of the HVAC system (heat, ventilation and air condition system). Seasonal variations over the year, variations over the week, and even between day and night, create demands that the system must be able to cope with while delivering the required overall annual efficiency. In particular, variations in heat transfer media flows and temperatures must be known and specified to the supplier as part of the procurement process.

The buyer must demand performance data from the seller, in the form of COPs for at least two full-load operating points, and an annual energy-based COP, i.e. a seasonal performance factor. Remember that the annual heat profile for the building may not be exactly known, even if the energy demand when based on another heat source is known. Also needed are maximum heat power demands and system temperatures in the existing heat distribution system. The control system must be coordinated with the controls of the other subsystems in the HVAC system, with someone being appointed responsible for this coordination during design, commissioning and subsequent operating stages. Remember that it is not normally possible to successfully commission a system based on only one set of load conditions: Commissioning results must be confirmed by testing at high and low loads.

Documentation must at least describe how the overall HVAC system is designed and operates under all working conditions. All control system settings must be listed, starting with values filled in from the commissioning stage, with spaces for the result of subsequent measuring and testing to be recorded. Values must be clearly noted, supported by the date, the signature of the person entering the data and details of any changes, whether of the system itself or of operational duties and values.

The annual heat load may be met with a heat pump system with or without a back-up heating system using another heat source. In each individual case, the design will depend on economic and technical constraints. However, a risk analysis of how to handle operational problems and breakdowns must be performed for every heating system. A heat pump system may be designed both for total energy coverage of heat loads and an ability to cope with component breakdown.

Another factor to be considered when installing any heating system is that of operational security. If the purchaser does not possess all the necessary knowledge in-house, the information must be available locally. The service response timeframe needed is determined by the effect of the most serious component breakdown on heat production. The service organisation must be familiar with the design and construction of the system, and carry all necessary spare parts. It is often worthwhile specifying that the installer must be responsible for service and operation for a warranty period of at least two years. The cost of checking stipulated operational performance at certain warranty points may be very expensive: charges must be in relation to the size of the system and possible losses due to failure to reach performance targets.

5.3. Commissioning

A heat pump is part of the total HVAC system in the building. It is important that the control system is commissioned as a total system. Someone must be responsible for coordinating the different subsystems such as heating, ventilation and the control system for example. The suppliers of subsystems must participate in this coordinated commissioning, and this must be stated in their contracts.

Operating modes differ, depending on ambient and indoor conditions. All those modes must be identified, with details given to the suppliers of subsystems. It is normal that not all stages of commissioning can be performed at one time. System performance must be measured to ensure that it can deliver the required functions and performance for both summer and winter operation. The documentation produced by commissioning must include building drawings, operating manuals and lists of set values for all manual settings in the control system.

5.3.1. Coordination Responsibility

Commissioning may involve different suppliers, who must work together. Without a coordinator with overall responsibility for all parties' work, there is a risk of sub-optimisation. If one supplier, for example, concentrates exclusively on its own systems, the effect may be to reduce the performance of the system as a whole. The purchasing and commissioning manual must emphasise the importance not only of checking the quality of all delivered components and systems, but also of concentrating on the functions and performance of the HVAC system as a whole. The role of the coordinator is to oversee the planning of the various stages of commissioning.

5.3.2. Function and Performance Testing

Function and performance of the entire HVAC system must be checked as the concluding stage of commissioning. Testing of individual electrical cables, pipes, valves and ensuring that all connections are as shown on drawings is the same as for all other commissioning work. Control systems must react correctly in response to input signals, and also generate the correct output signals and responses.

As far as possible, all variations in flows and temperatures at the evaporator and condenser must be tested with the heat pump system in operation. Values at low loads, high loads, and in response to load changes must be measured. Unscheduled stopping of components such as pumps, fans or compressors that might be accidentally done during operation by personnel must generate fault modes to protect the overall status of other components. Emergency functions in particular must be both fail-safe and in accordance with the EU Machine Safety Directive.

It must be possible to measure the heating capacity and efficiency, i.e. heating capacity divided by electrical drive power, with reasonable tolerance. This tolerance may depend on the capacity of the heat pump. For heat pumps up to about 100 kW thermal capacity, a tolerance of 10-15 % is acceptable. For large units, the uncertainty of measurement of measured data must not exceed 10 % for a 95 % confidence interval. Note that the conditions when making these measurements may differ from those specified and used for warranty condition testing. There are some standard guidelines for field measurements of heat pumps: one such standard may be found at Nordtest, Methods NT VVS 115 and NT VVS 116 (Nordtest Standards, 2013). Measuring the annual efficiency is a very demanding issue, but is important as it shows why the system was chosen rather than other ways of heating. At the very least, all electrical energy must be measured. Much of the heat load depends on ambient conditions. A good way to get an overall impression of the working performance of the system is to plot a 24-hour curve of electrical energy demand for the compressors against a corresponding 24-hour curve of average ambient temperature. When new 24-hour electrical energy data diverges from earlier plots, the reason should be explored.

5.3.3. Checklist for Starting up the System

It should be decide who must take part in commissioning the entire installation system. Include the operating teams and service personal who will be operating and servicing the system after it has been commissioned, if they are not the same as the start-up personnel.

1. Make sure that drawings have been prepared, showing all aspects of the system and including any late modifications:
    a. wiring diagrams;

   b. piping and instruments;
   c. control system algorithms.
2. Check all wiring and piping according to the drawings:
   a. external cabling and cabinets, including the direction of rotation of all machinery, fans, pumps and compressors;
   b. that all valves operate as intended;
   c. that all emergency stop switches, pushbuttons, devices etc. operate as intended, and that the emergency stop function is again immediately available when the function has been reset;
   d. All alarm signals.
3. Pressure-test and leak-test piping.
4. If thermal insulation has not been applied, do so.
5. Charge the systems with liquid and refrigrants.
6. Start and test-run every subsystem, if possible by itself:
   a. observe function, vibration, noise and abnormal operation.
7. Start up the whole system.
8. Try to operate in all possible operational modes, and decide when other test operation must be performed.
9. Measure capacity and COP for at least one possible operating condition.
10. Bring all documentation up to date, i.e. showing any late modifications:
    a. all diagrams;
    b. list of settings and alarm levels;
    c. first entries in the operational log book.
11. Make sure that operation is understood by the operating team, and decide what to look for over a longer time scale.

5.4. Heat Pump Maintenance

The manual will include maintenance activities that the janitor can perform. The owner of the properties must clearly identify the internal and external operation and maintenance organisations. Responsibilities must be defined, as must activities to be performed, together with details of what internal resources and personnel that are available, and what activities that must be outsourced. Activities that should be included are:

- schedule of maintenance activities,
- planning and preparation of maintenance activities,
- checking of alarms,
- documentations and drawings,
- regular inspections of the building and its technical installations,
- inspection/testing of refrigeration circuit , in accordance with Swedish regulations,
- performance/operation monitoring,
- follow-ups of maintenance activities.

5.4.1. Supervisory System and Performance Measurements

The supervisory system must be purchased together with the heat pump system and tested as part of commissioning. A simple monitoring check can be performed by measuring the total electrical energy demand, of the system and of the compressors separately, on a daily basis. Determine the average ambient temperature per 24 hours and plot an energy profile for the system. This can done either manually or (preferably) by computer software that compares each 24-hour period with the existing profile for the building. If any change in the pattern of consumption occurs, the reason for the change should be investigated. Other variables that can be plotted as part of the energy profile include ambient temperature, as well as local variations such as wind and non-climate-related loads. Those local variations must be determined by the operating and/or service personnel and noted in the system records. With such factors accounted for, it becomes easier to identify abnormal factors and/or effects.

At least once a year, service work that has been performed during the year should be listed, with details of the work and the costs of every item. Whenever any item of service work is performed, ensure that the service personnel always investigate why it had to be done. In order to build up a long-term record of system

behavioural and operating and maintenance costs, it is important to know what types of faults or abnormal operating conditions that are the most common or the most costly.

5.4.2. Daily Manual Inspections

It is not practical to try to computerise all aspects of supervision. A problem that might, for example, start with loose screws, leading to vibration or leaks that would be indicated by leaking lubricants, would not necessarily be detected by a computerised supervisory system. Issues to be considered by physical inspections include:

- Always keep the system clean in order to be able to see changes. Demand the same from service personnel.
- Become familiar with what normal operation looks, sounds and feels like. Use normal senses such as sight, smell and hearing. Many problems can be caught very early if changes are noticed.
- Go into machine rooms and look at the equipment every day.
- Note every change and service in an operating log kept in the machine room or close to the equipment:
  - notes from the daily inspection round and any resulting service work,
  - service items that include changed parts; note what part and where it was bought, in the system documentation,
  - amount of charged refrigerants and type, by whom and when,
  - amount of charged lubricants and type.
- Look at the energy profile, if possible every day,

5.4.3. Operating Journal

All setting adjustments must be documented, and the documentation must be available in the machinery room. Time, settings, the reason for the adjustment, and by whom it was made must be noted for every manual adjustment. If there is no performance monitoring system, at least each month's electricity use must be noted and monitored. A rising electricity demand can be a sign of a fall-off in system performance. If there is a trend of rising electricity demand, despite unchanged load conditions, a service technician should be consulted. Service performed by external companies must also be documented and described.

6. OTHER STUDIES

Two other studies have been presented within the framework of IEA HPP Annex 36, concerning issues regarding operation and maintenance of heat pumps: "Large Heat Pumps – Research and Development during the 1980s in Sweden" (Rolfsman and et.al., 2012), and "Improved Reliability of Residential Heat Pumps" (Haglund Stignor, 2012). Some of the recommendations to improve installation and operational quality given in these reports are described in this chapter.

6.1. Recommendations for Operation and Maintenance Stated in Sweden in the 1980s

In 1986, Byggforskningsrådets (BFR; the former Swedish Council for Building Research) Heat Pump Group stated that interest in, and knowledge of operation and maintenance of heat pumps were seen as deficient (BFR Värmepumpsgrupp, 1986). Instructions were developed by BFR to help building owners and service personnel to understand and monitor operation of heat pumps. Owners often lacked suitably trained personnel, whether due to cost or to unawareness. It was thought that establishment of service companies would be profitable, but one difficulty was, and still is, to convince owners that purchasing of operation and maintenance services would be economically profitable.

Five operational and maintenance measures were given in order to achieve improved operational results from existing heat pump plants:

- Reducing the temperature of the supply flow from the heat pump condenser and making better use of its capacity.
- Tools for monitoring heat pump performance and activities in order to achieve maximum performance.
- Operation and maintenance readiness for maximum availability.
- Supervisory and alarm equipment.

- Training, particularly including practical tests, for operating personnel, service companies, consultants etc., concerning the above measures. This could help poorly performing units to achieve more profitable operation. Ideas for improving components, design, and supervisory and control equipment would also benefit from better knowledge.

It was estimated that, if the measures described above were realised on existing heat pumps at the time, an energy saving of 10 % could be achieved. The cost for introducing the measures would probably be recouped by the value of one year's energy saving. The measures proposed by the BFR Heat Pump Group in the 1980s are still valid, as can be seen from the interviews described in Section 3 and the workshop described in Section 4.

6.2. Failures of Heat Pumps in Swedish Single-Family Houses

The study of common failures of heat pumps installed in Swedish single-family houses had the objective to suggest what measures to be taken to reduce the number of failures (Haglund Stignor, 2012). Based on failures statistics and interviews with stakeholders in the heat pump industry, common failures were identified and an attempt was made to categorize the failures. The interviewees generally experienced that that the owners were pleased with their heat pumps and there are not many troubles with heat pumps compared to the number of installed heat pumps in Swedish single-family houses. The interviewees told that heat pump failures often are caused by:

- poor installation,
- neglected maintenance and surveillance,
- poor quality of standard components,
- components are used outside their declared operating range.

The results shows that many of the common failures can be caused by several of these categories, and different types of responses are needed in order to improve reliability. The quality of the installations could be improved by improving installers' knowledge of heat pumps and by requesting them to fill in an installation record. Requesting completed installation records reduces the risk of steps in the commissioning process being overlooked. It is also important that the owner of the heat pump performs the preventive maintenance recommended by the manufacturer, and that he/she monitors parameters indicating that satisfactory operating conditions are being maintained. Another example of how to improve reliability is that a certified heat pump service technician should regularly check the heat pump system and its operating performance. Poor quality components can be such as temperature sensors, valves and circulation pumps. Compressors often fail because the heat pump is being required to operate too far outside, or close to, its operating limits, and is not necessarily due to poor quality of the compressor itself. By monitoring and investigating deviant operating parameters, as for examples temperatures of heat transfer medium and the electricity use, insufficient operation can be detected for avoiding a compressor failure.

In order to further improve the reliability of heat pumps installed in single-family houses and to reduce costs for the insurance companies, there must be incentives for purchasers not only to choose a high quality product but also to employ the best installers and service technicians, as some failures and costs are due to lack of training. Insurance companies can, for example, offer lower premiums for customers who use the services of a certified heat pump installer and/or choose a heat pump with a quality label that indicates the use of components with a certain quality, and use an efficient service organisation. Today's focus is mainly concerned with the energy efficiency of heat pumps, with the result that less attention may be paid to aspects regarding lifetime and reliability when purchasing. In addition, the heat pump owner must realise that it is necessary to look after the heat pump to ensure that it operates as well as possible. In order to achieve these incentives, communication and cooperation between the stakeholders in the heat pump industry is needed.

7. ACKNOWLEDGEMENTS

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COUNTRY REPORT:
UK CONTRIBUTION TO THE IEA HEAT PUMP PROGRAMME ANNEX 36 ON QUALITY OF INSTALLATION AND MAINTENANCE OF HEAT PUMPS

Improvements to Design and Installation Standards of Domestic Heat Pump Systems

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ABSTRACT

The UK's programme of work was aimed at improving the quality of installation and maintenance of heat pump systems in the UK. It focused on the residential sector, where heat pumps are used for space and water heating but are typically not used for cooling. The work comprised five main projects:

a) A literature review of issues relating to installation in maintenance including, reviews of consumer protection, training courses, insurance, standards and field trials.
b) Analysing the performance of real systems and making interventions to improve performance in field trials.
c) Laboratory experiments to analyse the effect on heat pump from cycling, buffer tanks and the efficiency of hot water tanks.
d) Improvements to standards for design and installation.
e) Dissemination of improvements to standards through UK wide roadshows for heat pump installers and social housing providers.

The comprehensive field trials carried out by the Energy Saving Trust and analysed by the UK Department of Energy & Climate Change (DECC) have resulted in an update in the UK's standards for design and installation of heat pumps. DECC subsidised a series of road shows to inform installers and social housing landlords about these revisions

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1. INTRODUCTION

This report presents the UK Department of Energy & Climate Change's (DECC's) program of work to improve the quality of installation and maintenance of heat pump systems in the UK. It focuses on the residential sector, where heat pumps are used for space and water heating but are not used for cooling. The work covers five main issues:

1) Documented problems with heat pumps and the results of field trials,
2) Laboratory experiments,
3) A review of standards for design and installation,
4) Consumer rights,
5) Recommendations for further improvements.

The comprehensive field trials carried out by the Energy Saving Trust and analysed by DECC have resulted in an update in the UK's standards for design and installation of heat pumps. DECC subsidised a series of roadshows to inform installers and social housing landlords about these revisions.

2. THE UK MARKET FOR HEAT PUMPS

Heat pumps are a relatively new technology in the UK. Table UK-1, supplied by BSRIA, shows the cumulative sales of heat pumps for all sectors of the market from 2006-2011.

Table UK-1: Cumulative UK Sales Data of Heat Pumps 2006-11 *(Supplied by BSRIA)*

Heat pump type	Cumulative UK sales 2006-2011
Air-water monobloc	78,000
Air-water split	92,000
Exhaust air-water	25,000
Ground-source	116,000

Annual sales are shown below (Figures from BSRIA; Karpathy, 2012a).

This study focusses on the residential sector. In the UK, the default residential heating system is a gas boiler (85% of houses), with the remainder being electric heating (around 9%), oil (4%) and solid fuel or LPG (2%) (Palmer and Cooper, 2012).

Karpathy (2012b) indicates that around 59% of heat pumps installed in the residential sector are in new build properties. In general, these would be expected to be replacing gas. Data from DECC's Renewable Heat Premium Payment Scheme indicates that the majority of heat pumps retrofitted in the residential sector replace oil, electric storage heating, solid fuel or LPG. Table UK-2 shows the annual UK sales of heat pumps in 2011 (Karpathy, 2012b]).

Table UK-2: Annual UK Sales Data of Heat Pumps 2011 *(Supplied by BSRIA)*

Annual sales 2011	Proportion	Type
12000	50%	Air/water monoblock
3840	16%	Exhaust air/water
4800	20%	Air to water split systems
3360	14%	Ground to water

3. A SUMMARY OF DIFFERENT APPROACHES TO CALCULATING HEAT PUMP EFFICIENCY

Before describing the various field trials that have been carried out, it is helpful to present the various different methods of calculating heat pump efficiency and the parameters which need to be measured or inferred in each case. Table UK-3 shows the terminology which will be used throughout the report.

Table UK-3: Terminology Used to Describe Electricity and Heat Parameters for the Measurement of Performance of Heat Pumps

Heat	Q_{H_hp}	Space heating provided by the heat pump
	Q_{W_hp}	Water heating provided by the heat pump to the domestic hot water cylinder
	Q_{H_aux}	Space heating, provided by the auxiliary electric heater
	Q_{W_aux}	Water heating, provided by the electric immersion to the domestic hot water cylinder (=$E_{Immersion}$)
	Q_{W_out}	Heat in the water supplied to the hot taps
Electricity	$E_{S_fan/pump}$	Electricity used by the source pump (for ground-source) or fan (for air-source)
	E_{HP}	Electricity used by the heat pump (excluding the ground loop/air inlet and auxiliary heating/immersion)
	$E_{Immersion}$	Electricity used to supplement domestic hot water production
	E_{Aux}	Electricity used to supplement space heating
	E_{bt_pump}	Electricity used by the buffer tank pump (if present)
	$E_{B_fan/pump}$	Electricity used by the fan or pump of the central heating system

The performance of a heat pump system may be calculated in many different ways. Nordman and Nowak (2012) describe the SEPEMO definitions, which will be used here. Also used is the system efficiency, described further in section 4.2.2.

Table UK-4 shows how these parameters are used to calculate different SEPEMO Seasonal Performance Factors and Seasonal Efficiency.

Table UK-4: Parameters to be Monitored for Calculation of Different Seasonal Performance Factors

	Component		SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}
Heat *	Heat supplied by heat pump	Q_{hp}	x	X	X	x
	Heat supplied to central heating circuit	Q_{H_hp}				
	Heat supplied to domestic hot water cylinder	Q_{W_hp}				
	Heat extracted from domestic hot water cylinder	Q_{W_OUT}				
Electrical components	Compressor	E_{hp}	x	X	x	x
	Heat source fan/ pump	$E_{S_fan/pump}$		x	x	x
	Auxiliary electricity for space heating	E_{Aux}			x	x
	Domestic hot water immersion	$E_{Immersion}$			x	x
	Buffer tank pump	E_{bt_pump}				x
	Space heating and domestic hot water fans/pumps	$E_{B_fan/pump}$				x

4. REVIEW OF FIELD TRIALS AND DOCUMENTED PROBLEMS WITH RESIDENTIAL HEAT PUMPS

4.1. Published Field Trials Prior to 2008

A number of small scale field trials of domestic heat pumps were carried out prior to 2010. The largest sample size was 10. Performance ranged from very good (SPF4=3.62) to very poor. Several reports noted that performance could be improved by more careful control of the electric immersion (which may be a control problem, or may be that the heat pump was under-sized for the demand), and also by reducing the electricity used by circulation pumps on the central heating side. Table UK-5 summarises the trials. Further details are given in Gaunt, Matthews & Willson (2013).

Table UK-5: Summary of Findings of UK Heat Pump Field Trials Prior to the Energy Saving Trust Field Trials

Location	Number and type of heat pump(s)	Performance	Comments
York	1 air to air	Poor	Most of the heat provided by electric immersion
Salisbury	1 ground to water	SPF4=3.16	Good performance; could be improved by reducing energy used by building circulation pump
Nottingham	1 ground to water	SPF4=3.62	
West Lothian	10 air to water	SPF4=2.7	
Harrogate	10 ground to water	Not quoted	Low temperature heating
Leeds	1 ground to water	SPF2=2.7 SPF4=2.15	Circulation pumps replaced to improve system efficiency

4.2. The Energy Saving Trust Field Trials (Phase I)

In 2007, the Energy Saving Trust (EST) identified the lack of reliable field trial data as a major problem for the nascent UK heat pump industry. Working with both the public and private sectors, EST set up the first large scale field trial of domestic heat pumps in the UK. Eighty-three (83) heat pumps, located throughout the UK, were monitored for one year, from April 2009 to March 2010. Out of these, reliable data were obtained for 71 sites, the configurations of which are shown in Table UK-6. The most common configuration was a ground-source heat pump supplying radiators and domestic hot water.

Table UK-6: Heating Configurations Examined in the First Phase of the Energy Saving Trust Field Trials

	Heat pump type			
Heat Sink	**Air-to-water**	**Exhaust-air**	**Ground-to-water**	**Total**
Radiators + domestic hot water	11		26	37
Radiators only	6		1	7
Underfloor heating + domestic hot water	1	1	11	13
Underfloor heating + radiators	3		3	6
Underfloor heating + radiators + domestic hot water			5	5
Underfloor heating only			3	3
Total	21	1	49	71

Full details of the first phase of the Energy Saving Trust's Trials are described by Dunbabin & Wickins (2006).

4.2.1. Monitoring Specification

The monitoring specification was designed to be compatible with the Energy Saving Trust's Gas Condensing Boiler Field Trial (Orr, Lelyvveld, & Burton, 2009).

The heat output was measured as the heat supplied to the central heating circuit (Q_{H_HP}) plus the heat in the domestic hot water **transmitted to the taps**, (Q_{W_OUT}). Note that this is **not** the heat **input** to the domestic hot water tank, but the heat of the hot water transmitted to the taps.

The electricity input (E) included the electricity required to run the compressor and ground loop pump or air inlet fan ($E_{S_fan/pump}$), the auxiliary electricity for space heating (E_{Aux}) and domestic hot water immersion and the electricity used to run circulation pumps on the central heating side ($E_{B_fan/pump}$).

Fig. UK-1 shows the monitoring schematic for the first phase of the trials. Note that some of the heat pumps had integral domestic hot water tanks, while some had separate tanks.

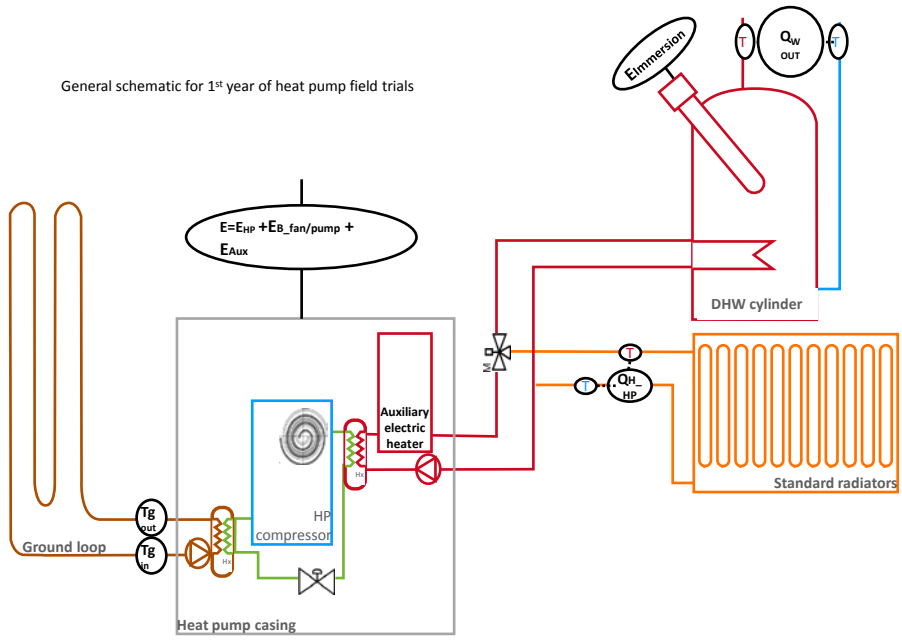


Figure UK-1: General Schematic for the First Phase of the Energy Saving Trust Heat Pump Field Trials

The **"system efficiency"** was calculated as follows:

$$SEFF = \frac{(Q_{H_{hp}} + Q_{W_OUT})}{(E_{hp} + E_{S_{fan-pump}} + E_{Aux} + E_{Immersion} + E_{B_{pump}})} \quad (1)$$

This configuration gives the best indication of **system efficiency** to the householder and is well suited to heat pumps with integral domestic hot water tanks. However, it does not fit into the SEPEMO categorization of heat pump efficiency (Nordman. & Nowak, 2012), or allow comparison with estimated seasonal coefficients of performance quoted by manufacturers. For this reason, more detailed monitoring was carried out in Phase II of the trials, as described in section 4.3.1.

4.2.2. System Efficiencies

The overall system efficiencies are shown in Figure UK-2. It can be seen that overall system efficiencies were low, between 1.2 and 2.2 for the air-source heat pumps, and between 1.8 and 3.4 for the ground-source heat pumps. The mean values were 1.8 ± 0.06 for the air-source heat pumps and 2.39 ± 0.06 for the ground-source heat pumps. Six additional air-source heat pumps, selected by a manufacturer instead of being selected at random, were also monitored and showed better system efficiencies (between 2.4 and 3.0), demonstrating that good performance can be achieved with heat pumps that are designed, installed and configured correctly.

It should be remembered the "system efficiency", as defined here, is strongly influenced by the householders' use of domestic hot water and by the heat losses from the domestic hot water cylinder.

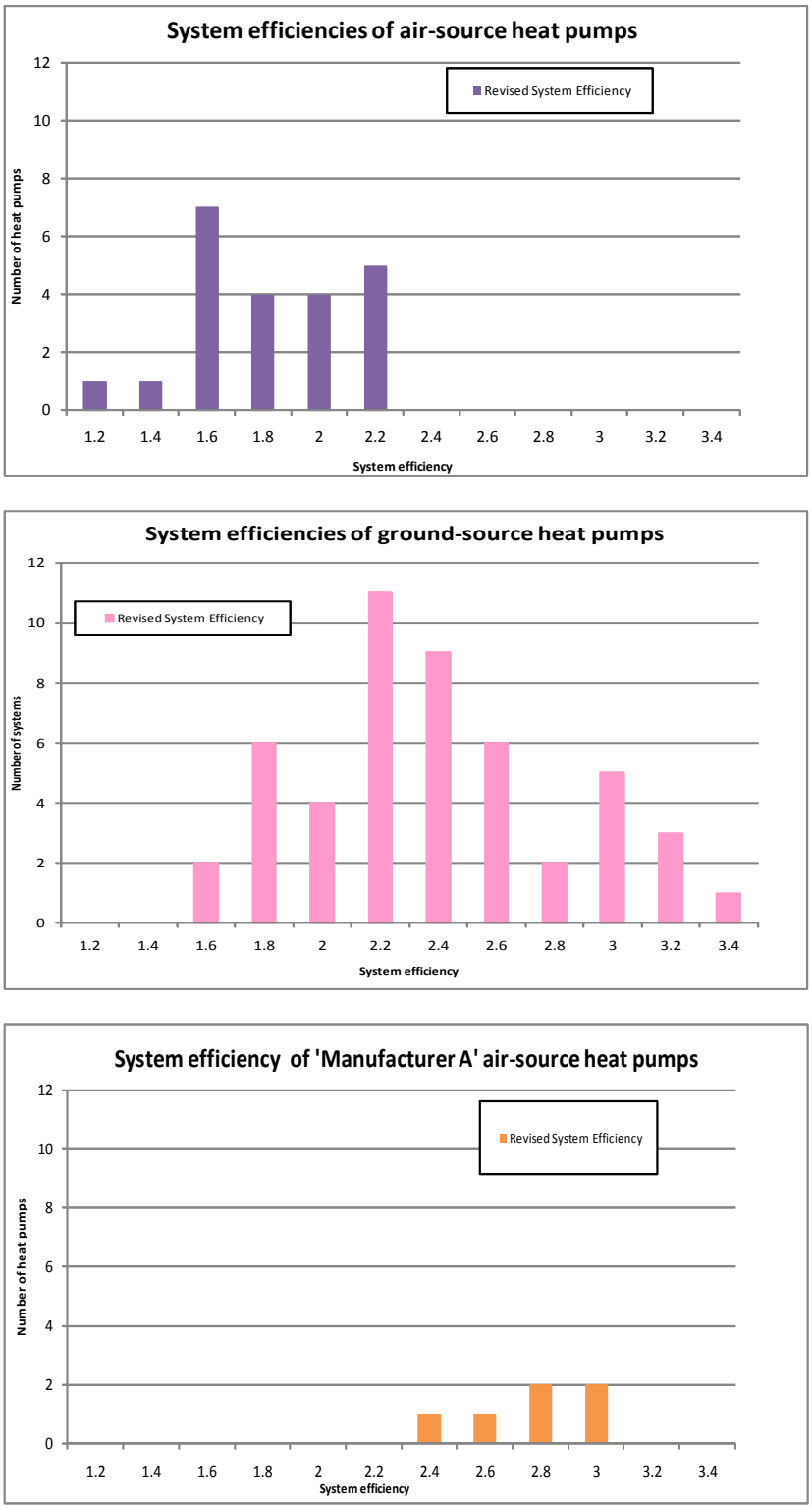


Figure UK-2: System Efficiencies for Air-Source Heat Pumps *(Top Chart),* ***Ground-Source Heat Pumps*** *(Middle Chart), and* ***non-Randomly Selected Air-Source Heat Pumps Monitored by a Manufacturer*** *(Bottom Chart),* Results From The First Phase Of The Energy Saving Trust Heat Pump Field Trials

4.2.3. Principal Design and Installation Issues Highlighted by the Study

Detailed analysis was carried out on a site-by-site basis as reported by Dunbabin & Wickins (2012). The main design and installation issues highlighted by the study were:

- Over-use of the electric backup. This may be caused by under-sizing of the heat pump or poor control strategies.
- Under-sizing of the ground loop.
- High central heating flow temperatures, leading to poor performance.
- High consumption of electricity by circulation pumps on the central heating side.

- Incorrectly sized domestic hot water tanks, leading to over-use of the electric immersion, or alternatively, to wasting energy if more water is heated than the householder can use.
- Excessive use of defrosting (for air-source heat pumps).

As a result of these findings, two steps were taken. Firstly, a number of heat pumps were selected for interventions to improve performance and monitored for a further year, as discussed in section 4.3. Secondly, the Microgeneration Certification Standard MIS 3005 was updated to address these issues. This is discussed further in Section 6.

4.3. *The Energy Saving Trust Field Trials (Phase II)*

The first phase of the Energy Saving Trust Field Trials was invaluable in highlighting faults in design and installation. Thirty-eight (38) heat pumps were selected for interventions to improve performance and a further six (6) heat pumps were added to the sample.

The interventions were classified into four categories:

- Major: requiring the input of a heat pump specialist (e.g. replacing the heat pump, repairing the ground loop, refilling the refrigerant) (12 heat pumps).
- Medium: requiring the input of a plumber (e.g. refilling the ground loop, adding a buffer tank, replacing a domestic hot water tank, replacing flap valves, replacing radiators, replacing circulation pumps) (9 heat pumps).
- Minor: altering controls, disabling the auxiliary direct electric heater, switching off unnecessary circulation pumps (11 heat pumps).
- None (6 heat pumps).

4.3.1. Monitoring Specification

The monitoring specification was enhanced for Phase II of the trials, in order to allow calculation of SEPEMO seasonal performance factors SPF_{H1-H4} as described in Nordman & Nowak (2012). A typical schematic is shown in Fig. UK-3.

It was not possible to determine all four SEPEMO SPF's in all cases, but for the majority of cases, at least SPF_{H2} and SPF_{H4} were calculated. Calculation of SPF_{H2} allows comparison with the EU Renewable Energy Sources Directive, which sets a minimum SPF_{H2} of 2.5 for a heat pump to be considered "renewable".

In addition, the overall "system efficiencies", from Phases I and II were corrected for their variation with external temperature and compared, in order to establish whether performance had improved.
For further details, see Dunbabin, Charlick & Green (2013).

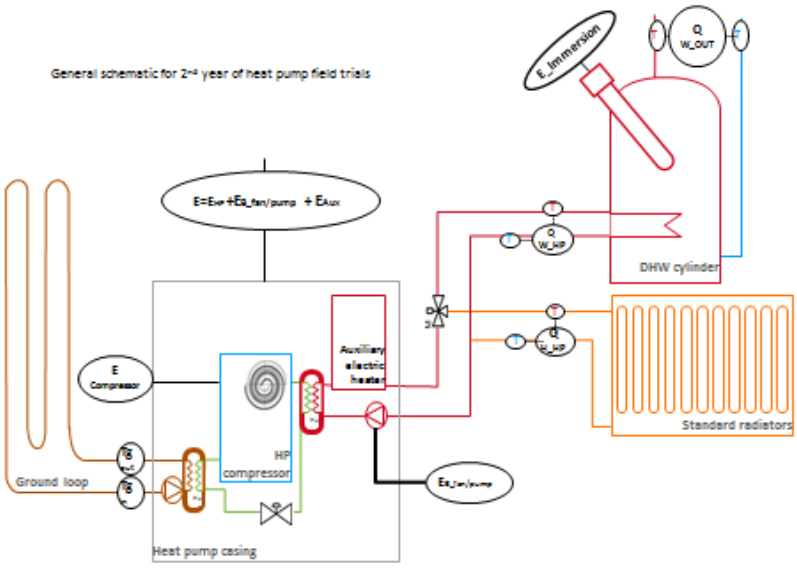


Figure UK-3: General Schematic for the Second Phase of the Energy Saving Trust Heat Pump Field Trials

4.3.2. Results – Comparison of System Efficiencies from Phases I and II of the Trials

Fig. UK-4 shows the comparison of system efficiencies from Phases I and II of the trial. These have been corrected to take account of the difference in degree day heating requirements for the two phases of monitoring (winter 2009-10 was colder than winter 2011-12).

Nine out of 12 heat pumps with major modifications showed improvements in system efficiency from Phase I to Phase II. Of these, eight showed an improvement of ≥ 0.3. For two of the three sites for which system efficiency was reduced in Phase II, the reason was increased immersion consumption for domestic hot water.

Eight out of nine heat pumps that received medium interventions showed improved system efficiencies in Phase II. In five cases, the improvement was ≥ 0.3. The greatest improvement, from 2.31 to 3.29, was at a site where software was altered to reduce the use of the auxiliary heater. The remaining heat pump in this category developed a fault during the second phase.

These substantial improvements in system efficiency indicate that problems have been rectified.

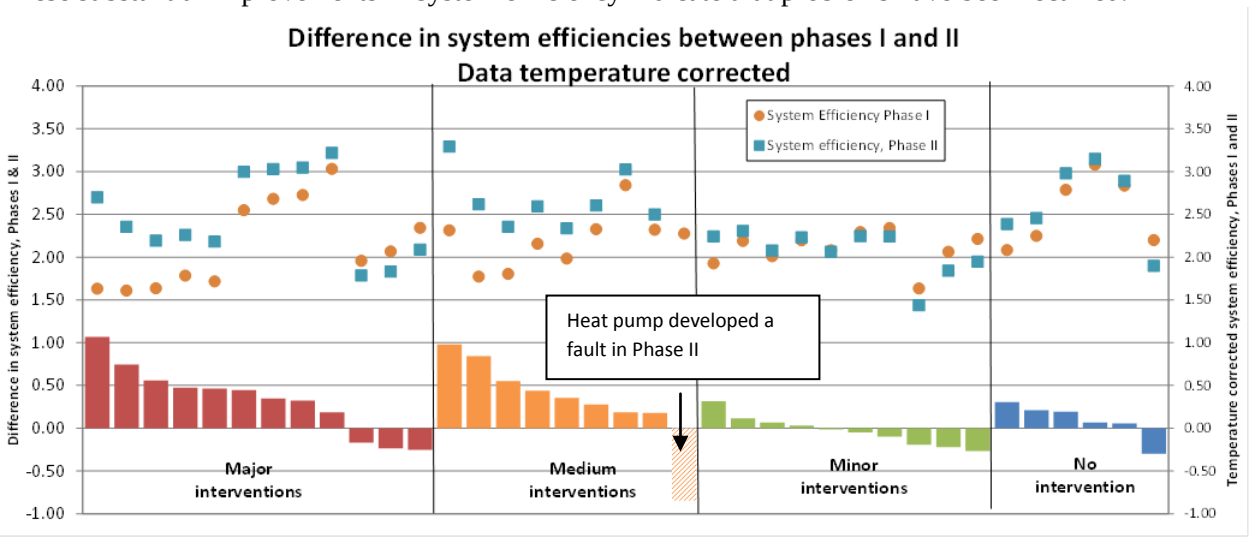


Figure UK-4: Difference in Temperature-Corrected System Efficiencies between Phases I and II of the Energy Saving Trust Heat Pump Field Trials

4.3.3. Results – SPF_{H2}

SPF_{H2} corresponds to SCOP in the EN 14511. It is defined as follows:

$$SPF_{H2} = \frac{(QH_{hp} + QW_{hp})}{(E_{hp} + ES_{fan/pump})} \qquad (2)$$

Note that the auxiliary electricity usage and domestic hot water immersion are not included.

It was possible to calculate SPF_{H2} for 15 air-to-water and 21 ground-to-water heat pumps. Fig. UK-5 shows the results. 20 out of 21 ground-source heat pumps met the EU Renewable Energy Sources Directive criterion to be considered "renewable", i.e. SPF_{H2} >= 2.5. Nine (9) of the fifteen (15) air-to-water heat pumps in the sample met this criterion. Six of the air-source heat pumps investigated were new heat pumps, sized, designed and installed to the new MCS standards. Of these, five met the EU's criterion to be considered "renewable".

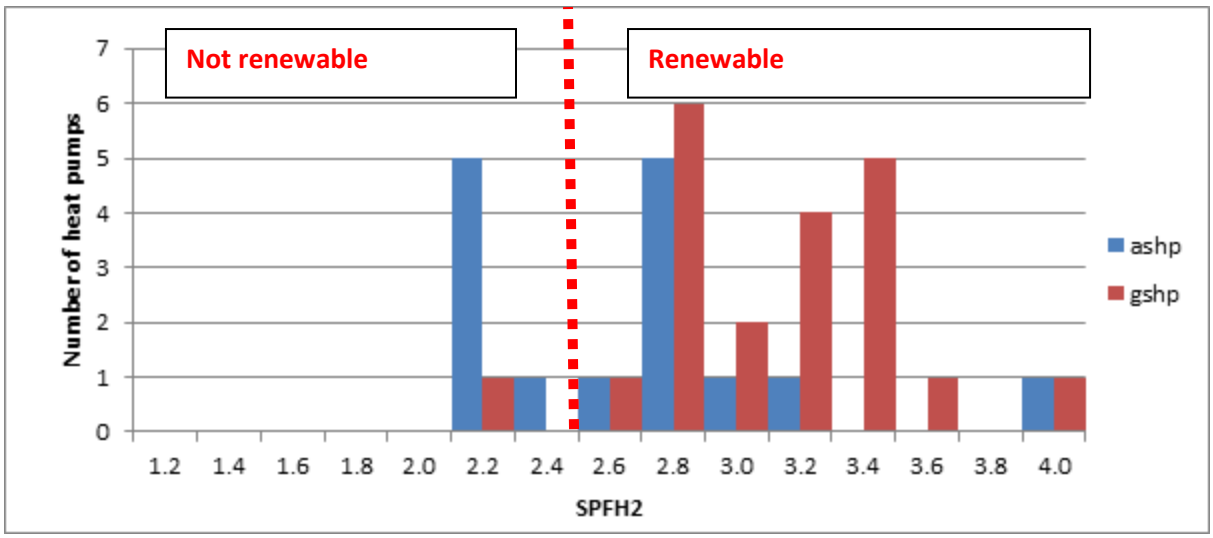


Figure UK-5: Histogram of SPF_{H2} Results from the Second Phase of the Energy Saving Trust Heat Pump Field Trials

4.3.4. Results – SPF_{H4}

SPFH4 is defined as follows:

$$SPF_{H4} = \frac{(QH_{hp} + QW_{hp} + E_{Immersion})}{(E_{hp} + ES_{fan/pump} + E_{Aux} + E_{Immersion} + E_{B_pump})} \quad (3)$$

Fig. UK-6 shows SPF_{H4} for 22 ground-source heat pumps (one of which developed a fault in phase II) and 15 air-source heat pumps.

The range is wider; from 2.0 to 3.6 for the air-source heat pumps and from 2.0 to 3.8 for the ground-source heat pumps. Table UK-7 summarises the results.

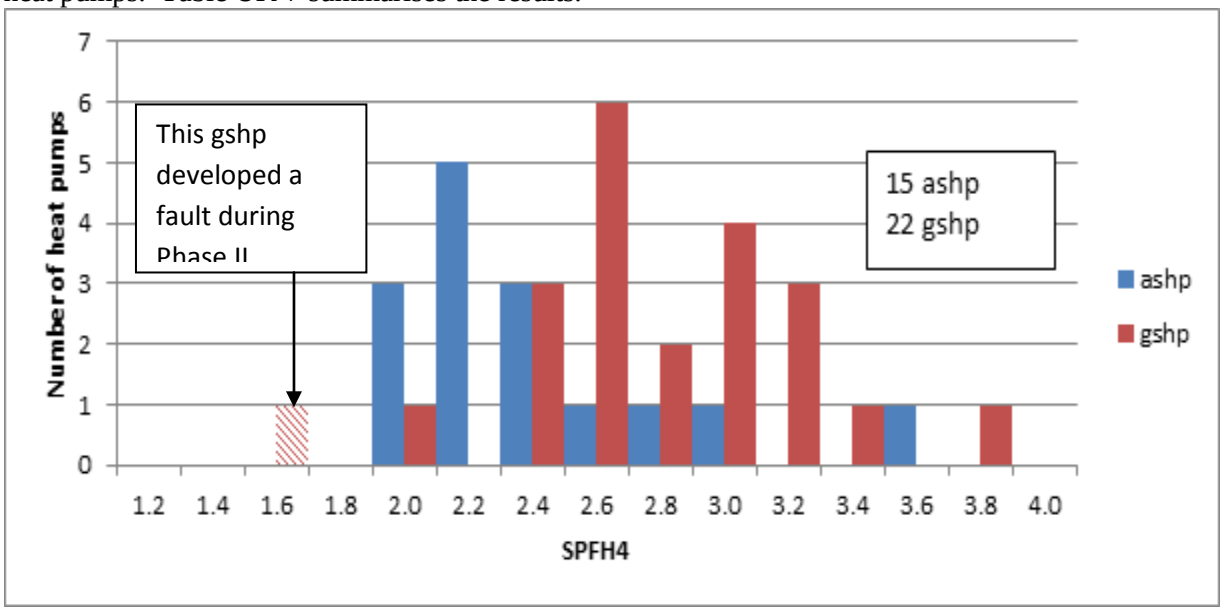


Figure UK-6: Histogram of SPF_{H4} Results from the Second Phase of the Energy Saving Trust Heat Pump Field Trials

Table UK-7: Seasonal Performance Factors (SPF_{H2} and SPF_{H4}) and System Efficiencies from the Second Phase of the Energy Saving Trust Field Trials

		SPF_{H2}	***SPF_{H4}***	***System efficiency***	***Proportion of systems considered renewable***
Air-source	Average	2.72	2.45	2.16	9/15 – note that 5 out of the 9 heat pumps considered renewable were completely new installations and were MCS 3005 compliant.
	Standard deviation	0.45	0.44	0.44	
	Range	2.2 - 3.9	2.0 - 3.7	1.7 - 2.7[a]	
	Number of systems	15	15	16	
Ground-source	Average	3.08	2.82[b]	2.54	20/21
	Standard deviation	0.40	0.42	0.47	
	Range	2.2 - 3.9	2.0 - 3.9	1.5 - 3.3	
	Number of systems	21	21	26	
Table notes	a). One air-source heat pump had a particularly low system efficiency, partly due to relatively high domestic hot water use (around 90%) and due to long pipes and high heat losses from the tank. This point has been omitted from the table. b). One ground-source heat pump developed a fault and had high auxiliary electricity use. This is shown as a shaded point in the middle and bottom charts of 0, but is excluded, the SPF_{H4} and system efficiency columns of Table UK-7. SPF_{H2} is not affected by use of the auxiliary electricity and so results from this heat pump are included in the SPF_{H2} column of Table UK-7.				

4.3.5. Case Studies

Many case studies were examined, as reported in Green et. al. Here, we show three cases for which performance was improved.

Site 444: Air-source heat pump

This air-source heat pump supplies heat to radiators only. During Phase I, it was used intermittently, from 2 to 7 hours per day. The 6 kW heat pump was replaced with an 8 kW. Various changes were made to the controls including changing the heating pattern to continuous during the winter months, configuring weather compensation and re-locating the external temperature sensor in an attempt to reduce the amount of defrost. Overall, the system efficiency improved, as shown in Table UK-8.

Table UK-8: Seasonal Performance Factors (SPF_{H1-H4}), System Efficiency for Air-Source Heat Pump 444; Phases I and II

	Dates	***SPF_{H1}***	***SPF_{H2}***	***SPF_{H3}***	***SPF_{H4}***	***System efficiency***	***Temperature corrected system efficiency***
Phase I	01/04/2009-31/03/2010	N/A	N/A	N/A	1.63	1.63	1.63[a]
Phase II	01/05/2011-31/03/2012	(3.20)	(2.44)	(2.44)	2.24	2.24	2.19
Table note	a). There has been a slight re-adjustment in the reported system efficiency since the first year results were published, due to removal of a small amount of faulty data. Note that temperature corrections may vary for different sites, depending on i) the amount of data recorded and ii) the local degree days. For this site, the temperature correction was zero in Phase I.						

Fig. UK-7 shows the effect of the changes on the overall seasonal performance factor, SPF_{H4}, for this site, as a function of daily average ambient temperature. The seasonal performance factor is improved at all ambient temperatures, but particularly at higher temperatures (5-15°C), reflecting the effect of the change in weather compensation for this site.

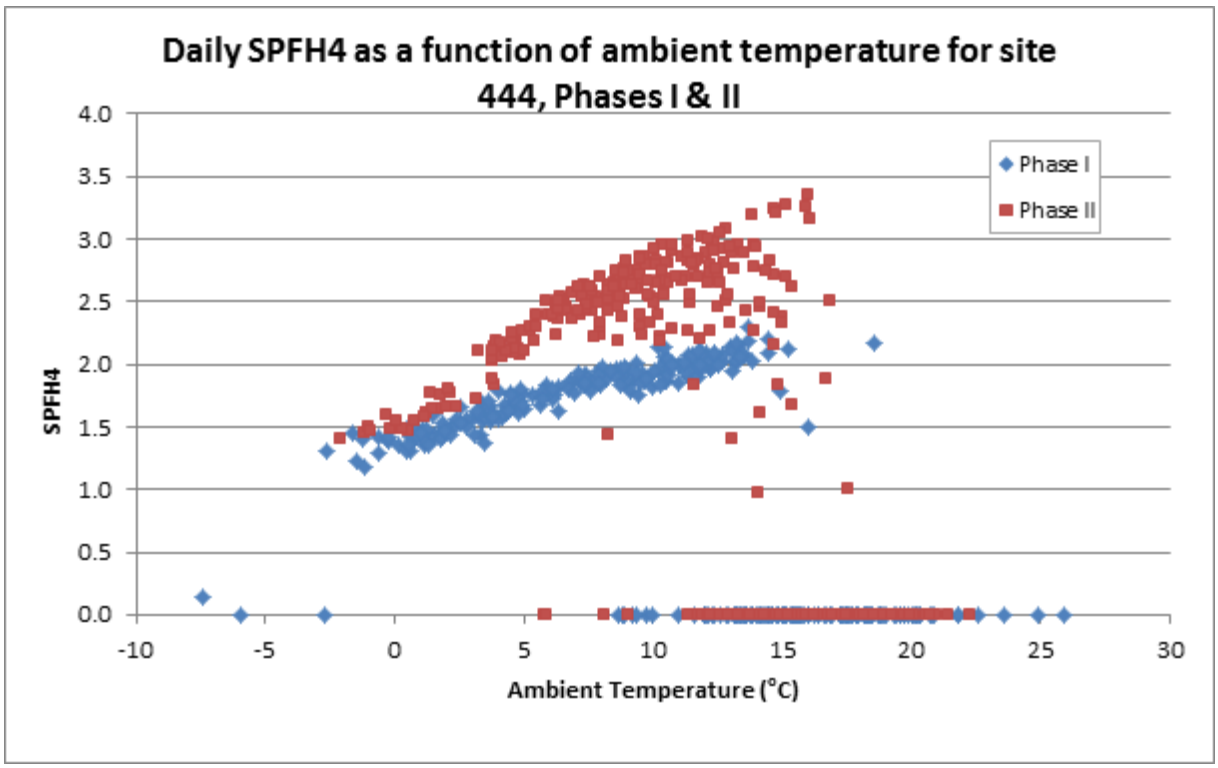


Figure UK-7: Daily SPF_{H4} as a Function of Daily Average Ambient Temperature for Site 444, Phases I & II

In the first phase of the trial, the householder used the heat pump intermittently, for between 2 and 7 hours per day, while in Phase II, the heat pump was set to continuous operation. As a result, daily average internal temperatures increased. During Phase I, the average internal temperature in the living room was between 15 and 20°C during the heating season, while in Phase II, it was much more constant, between 18 and 20°C. The number of degree days for the Phase II heating season was 25% lower than that for Phase I (partly because of the exclusion of faulty heat meter data from April 2011). However, the change to continuous heating meant that the heat used in Phase II was only 7% lower than that in Phase I, while improved performance meant that the electricity used was 32% lower than in Phase I, see Table UK-9.

Table UK-9: Degree Day Heating Requirements and Actual Heat Supplied, Air-source Heat Pump 444, Phases I & II

	Heat Output (kWh)	Electricity input (kWh)	Degree days	Heat supplied / degree day
Apr-09 to Mar-10	4206	2574	2057	2.04
May-11 to Feb-12 [(a)]	3925	1755	1541	2.54
Note: (a) Faulty heat meter data in April 2011. so period runs from 01/05/2011 to 28/02/2012				

Site 416 Ground-source heat pump

This ground source heat pump extracts heat from a 300m ground loop and supplies underfloor heating and domestic hot water for a highly efficient 4 bedroomed detached house built in 2008. Fig. UK-8 shows the schematic.

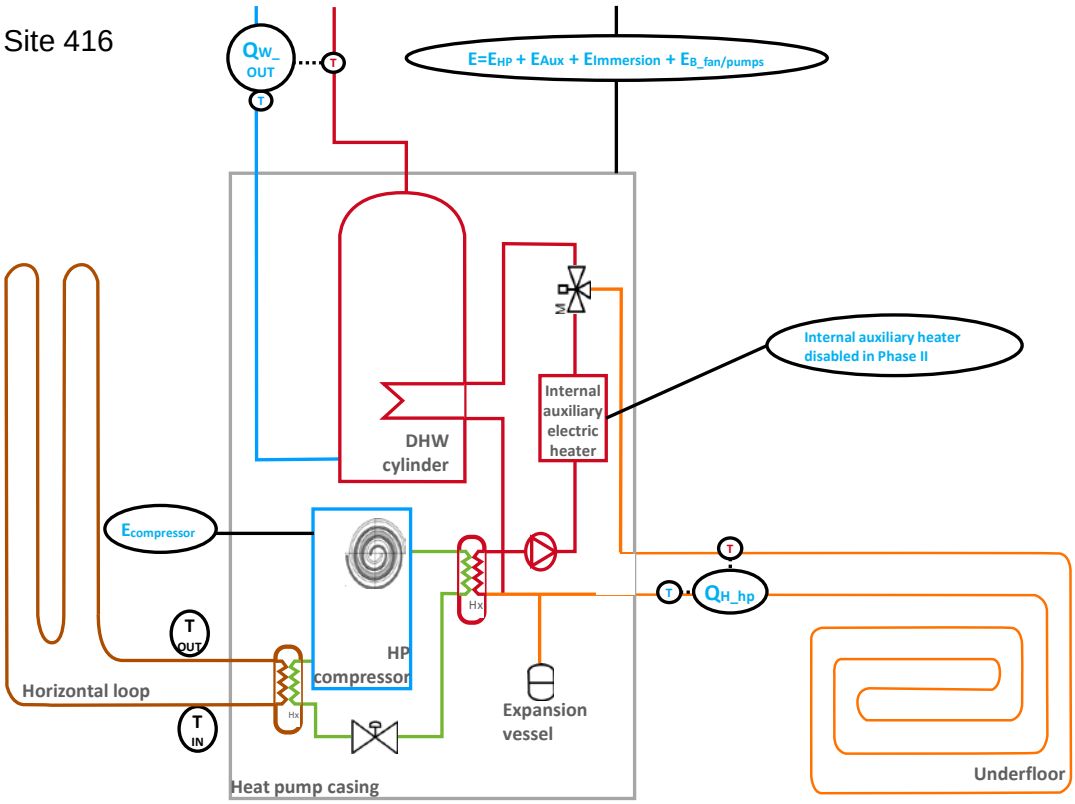


Figure UK-8: Schematic for Ground-source Heat Pump 416, Phase II

During Phase I, excessive use of the auxiliary heating resulted in poor overall efficiency. For Phase II, only two modifications were made at this site; **the auxiliary heating was disabled** and the circulation pump was replaced by a variable speed DC pump. Table UK-10 shows the system efficiencies for Phases I and II and SPF_{H1-H4} for Phase II.

Table UK-10: Seasonal Performance Factors and System Efficiency for Ground-source Heat Pump 416, Phases I & II

	SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}	System efficiency	Temperature corrected system efficiency
Phase I	N/A	N/A	N/A	N/A	2.35	2.31
Phase II	4.41	(3.94)	(3.94)	3.89	3.28	3.29

This example demonstrates very clearly the importance of controlling the use of auxiliary electric heating. An important corollary of this finding is that controls must clearly indicate to the householders whether auxiliary electric heating is enabled or not; the controls should not simply be labelled “summer setting” or “winter setting”.

Site 432: Ground-source Heat Pump

This house is a one bedroomed, semi-detached bungalow. A 4 kW ground source heat pump extracts heat from a vertical panel and supplies heat to radiators and domestic hot water. The vertical panel can be recharged by solar thermal panels in the summer. Fig. UK-9 shows the schematic.

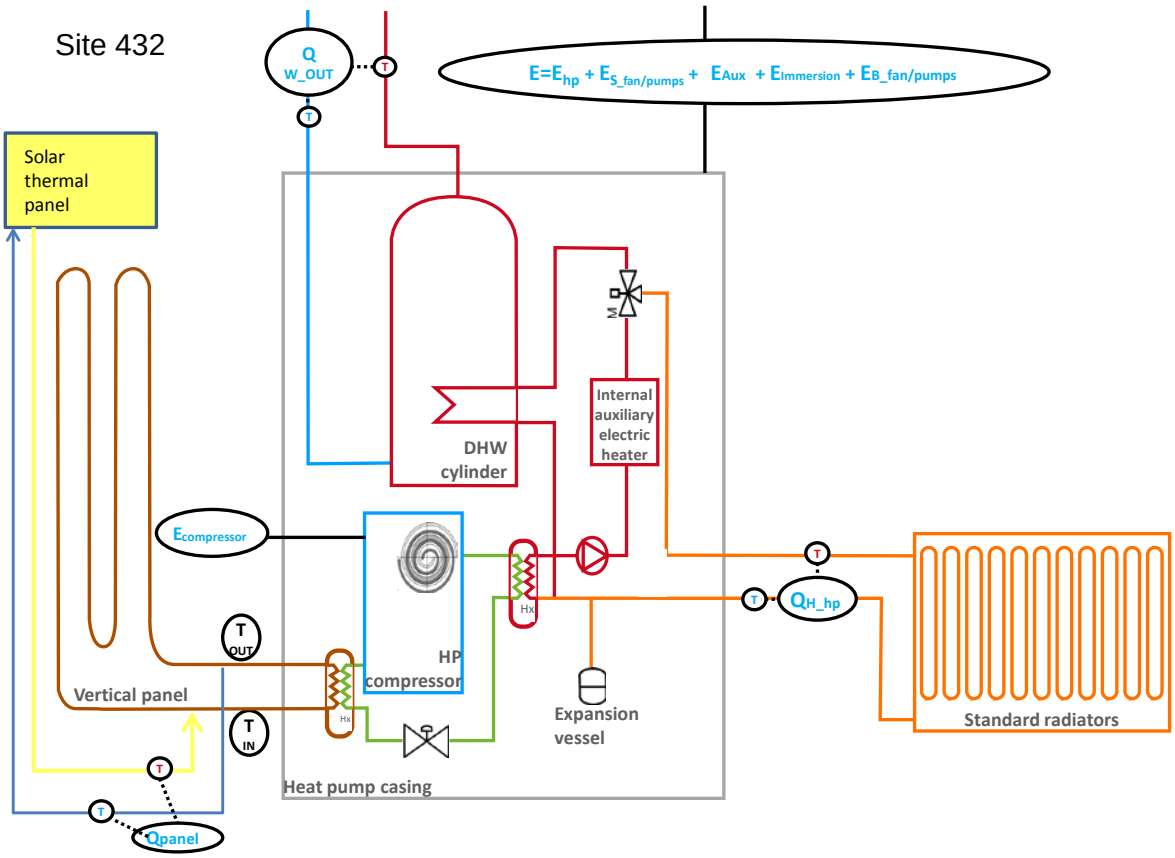


Figure UK-9: Schematic for Ground-source Heat Pump 432, Phase II

A range of interventions were carried out at this site including:

- disabling one of the circulation pumps on the central heating circuit;
- reducing the temperature of the domestic hot water to 46°C with a weekly sterilisation pattern;
- making comprehensive changes in control parameters, notably an increase in the delay before the auxiliary heaters are engaged (now 90 minutes);
- altering the parameters for weather compensation.
-

Table UK-11 shows the electricity consumption by component. The electricity consumed by the compressor has been measured separately at this site; auxiliary heating has been estimated by inspection of the time series.

Table UK-11: Electricity Consumption by Different Components for Ground-source Heat Pump 432, Phase II

	Compressor	Ground loop pump	Auxiliary heating	Circulation pump on central heating side
Electricity consumption during Phase II (kWh)	1896	(421)	(163)	(181)
Table Note	Estimated figures are in brackets.			

Table UK-12 shows a clear improvement in both SPF_{H4} and system efficiency during Phase II as compared to Phase I. This is due to lower use of the auxiliary heating. The circulation pump on the central heating side is internal to the heat pump; further improvements in SPF could probably be achieved by using lower energy pumps.

Table UK-12: Seasonal Performance Factors and System Efficiency for Ground-source Heat Pump 432, Phases I & II

	SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}	System efficiency	Temperature corrected system efficiency
Phase I	N/A	N/A	N/A	(2.26)	1.82	1.80
Phase II	3.59	(2.94)	(2.81)	(2.64)	2.28	2.35
Table Note	Estimated figures are in brackets.					

For this heat pump, a single auxiliary electric heater can provide supplementary heat for either space heating or domestic hot water. During the first phase of the trial, the low SPF_{H4} was attributed to over-use of the auxiliary electric heater. The control parameters were changed to ensure that the supplementary space heating did not come on at the expense of the heat pump.

Fig. UK-10 shows a time series of total electricity demand minus the compressor electricity during the second phase of the trials. The weekly sterilisation pattern is apparent and accounts for almost all of the consumption of the auxiliary heater.

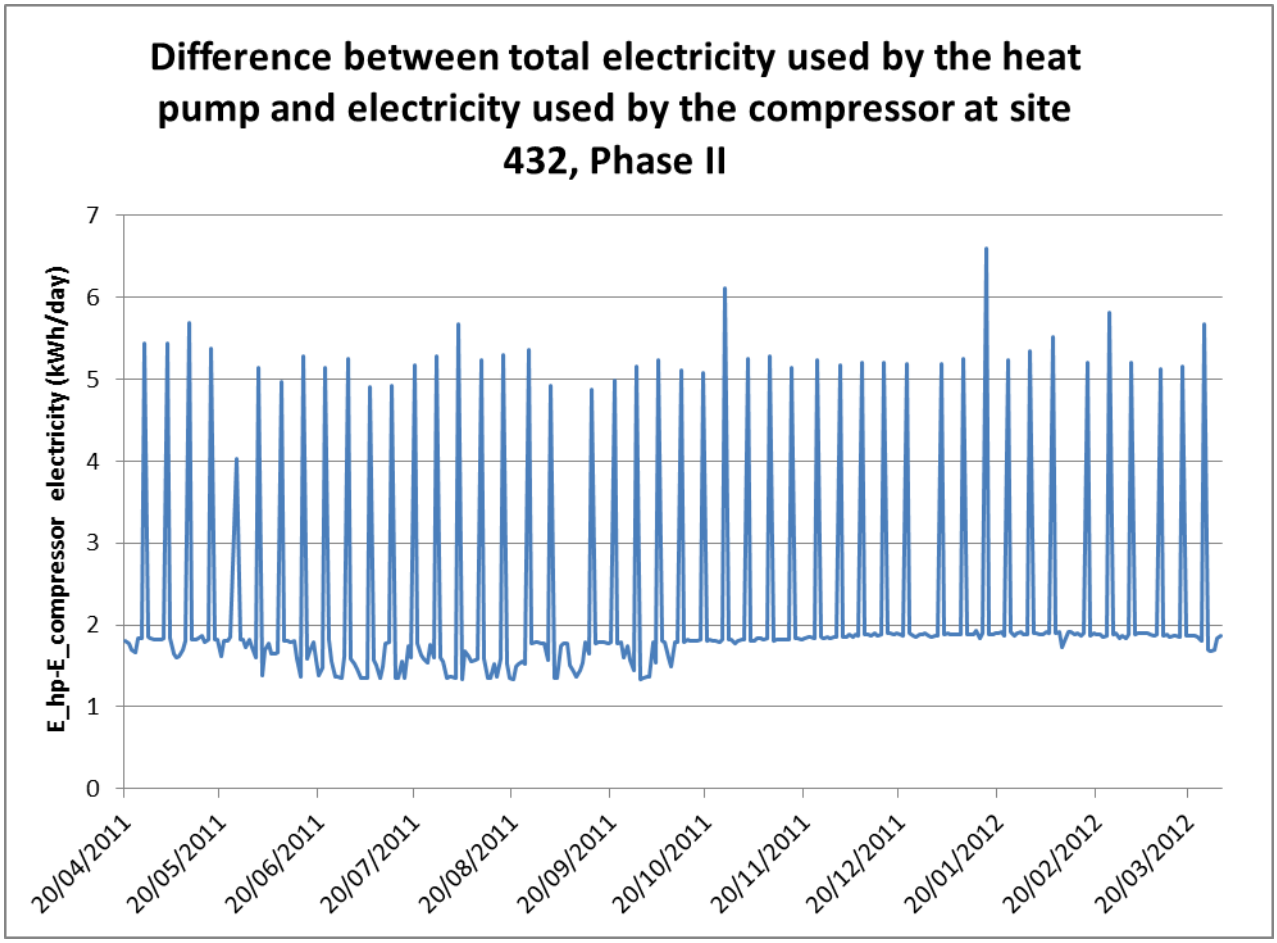


Figure UK-10: Difference Between Total Electricity Used by the Heat Pump and Electricity Used by the Compressor at Site 432, Phase II

Fig. UK-11 shows a scatterplot of space heating efficiency and water heating efficiency with days on which the sterilisation cycle is undertaken shown separately.

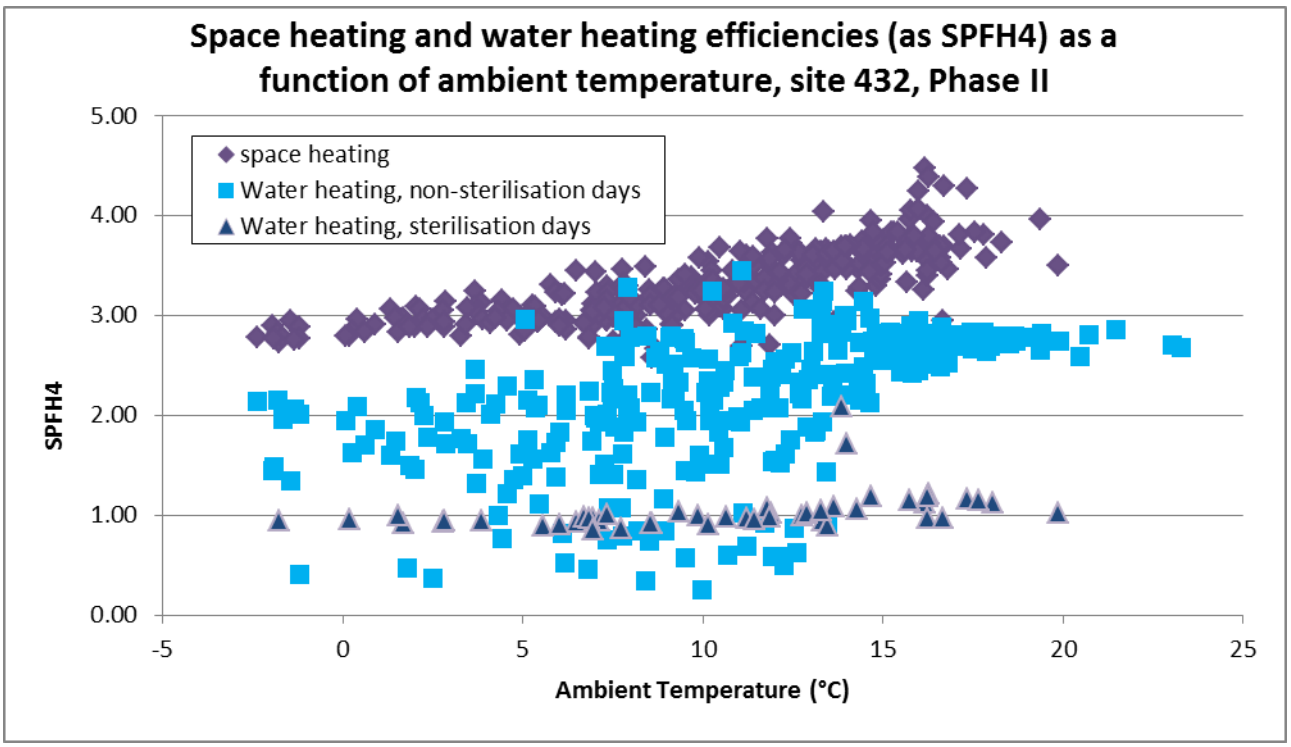


***Figure UK-11: Space and Water Heating Efficiencies (as SPF_{H4}) as a Function of Ambient Temperature**, showing effect of sterilisation, site 432, Phase II*

The figure shows that space heating efficiency ranges from 2.8 to 4.5, while the water heating efficiency (on non-sterilisation days) is between 0.5 and 3.0, with the greatest concentration of points being at around 2.8. This indicates good efficiency. On sterilisation days, the water heating efficiency is 1.0, as would be expected. This example shows the importance of configuring water sterilisation cycles properly. There could be merit in investigating alternative means of control of Legionella and other bacteria.

4.3.6. Conclusions and Recommendations

It is essential to define which system boundaries are being used when referring to seasonal performance factors. DECC considers that SPF_{H4} gives the best overall indication of the combined quality of a product and its installation and would therefore recommend that SPF_{H4} is used whenever possible. The MCS MIS Heat Emitter Guide (MSC 022, 2012b) currently requires the installer to provide the householder with an estimate of SPF_{H4} for space heating.

DECC would strongly recommend that the electricity used by auxiliary heater and domestic hot water immersion is measured and that appropriate diagnostics are given to the householder. Some manufacturers already monitor the electricity consumption of each component of the heat pump and DECC would encourage others to do so. DECC also considers that controls should be absolutely clear to the average householder. It is not sufficient to have a "winter" setting which activates the auxiliary electric heating.

Software controlling the sterilisation pattern should be carefully configured so that the immersion is not used excessively. There is a case for research projects to investigate alternative methods of controlling bacteria such as Legionella.

For air-source heat pumps, software controlling defrost should be carefully configured so that defrost is used only when required. Appropriate diagnostics should be provided to the householder.

Specifically for the UK market, DECC proposes that MCS MIS 3005 should be amended so that the installer shows the householder a weighted average of SPF_{H4} for space and water heating (if domestic hot water is supplied by the heat pump).

5. LABORATORY EXPERIMENTS

Some installation issues are best investigated by means of laboratory experiments. DECC commissioned experiments to investigate the effect of cycling on heat pump efficiency, and to provide recommendations on the best way to integrate buffer tanks within the heat pump circuit and the best way to heat domestic hot water with a heat pump.

5.1. Effect of Cycling on Heat Pump Efficiency

Heat pumps with fixed speed compressors may cycle rapidly in warm weather, when the house does not require the full heat output of the heat pump. Heat pumps with variable-speed compressors are less likely to be affected. Rapid cycling can adversely affect the lifetime of the compressor and other components and may also reduce efficiency. In addition, the spike in current draw-off when the compressor starts up can have adverse effects on the grid.

Green et al. investigated the effect of cycling on the performance of both air and ground source heat pumps with fixed-speed compressors. The experiments were carried out at EA Technology's test house facility. Supplementary modelling was carried out by Curtis (2013), who also describes the effect of cycling on the lifetime of components.

In the experiments, cycling was induced by using thermostatic radiator valves to decrease the volume of water being supplied to the radiators. Fig. UK-12 and Fig. UK-13 show the results for two experiments conducted in mild conditions (11-12 ˚C ambient temperature) and constant return temperature (40˚C); the green triangles indicate the COP of the cycle. For a run time of 9.3 minutes, the COP is 2.89; when the heating volume is decreased such that the run time falls to 2.4 minutes, the COP is reduced to only 1.9.

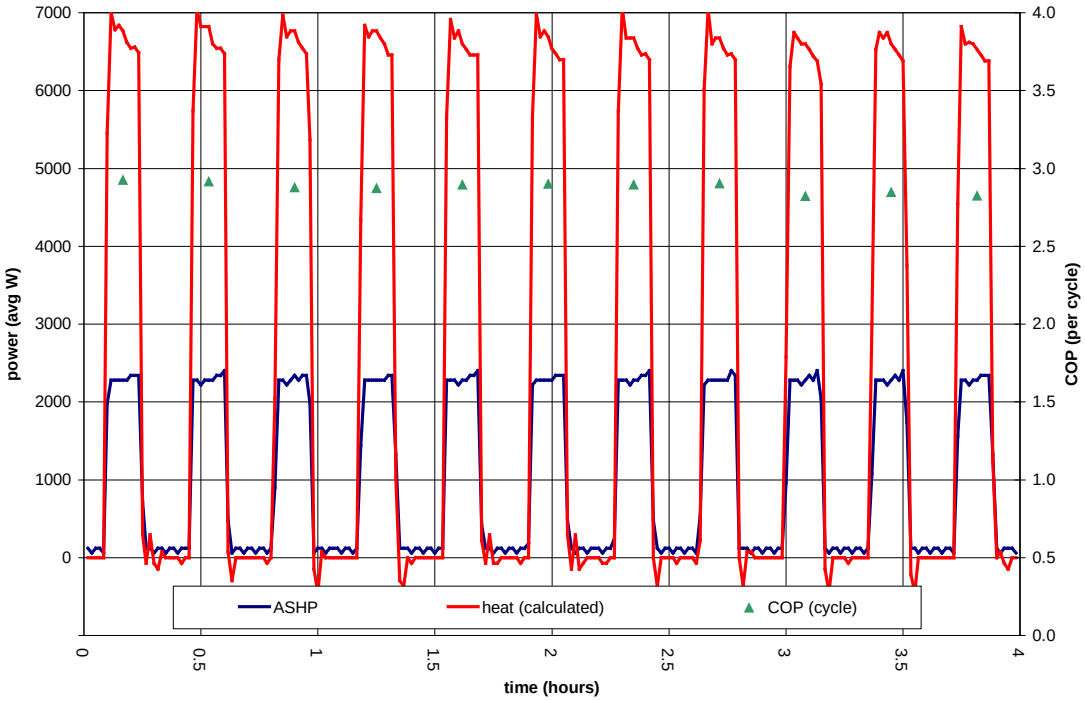


Figure UK-12: Test-House Measurements of Air-Source Heat Pump COP (Run-time 9.3 Minutes, COP=2.89)

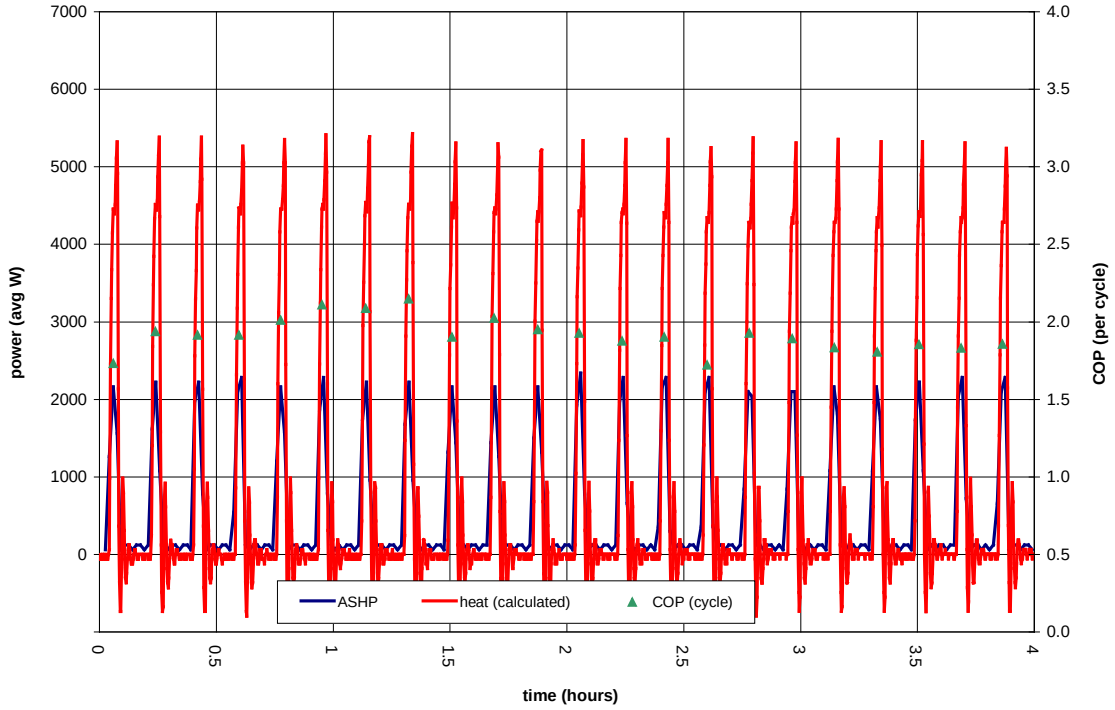


Figure UK-13: Test-House Measurements of Air-Source Heat Pump COP **(Run-time 2.4 Minutes, COP~1.9)**

Fig. UK-14 summarises the results for air-source heat pumps. The figures marked on the diagram indicate the run times in minutes. It can be seen that reducing the volume of water in the heating circuit has an adverse effect on the COP.

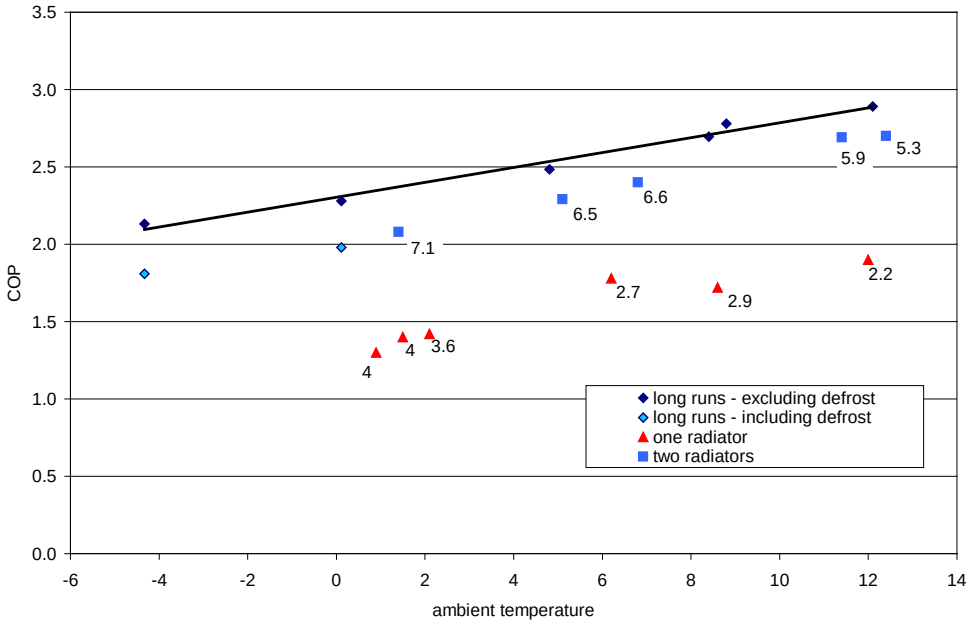


Figure UK-14: Test House Measurements of Air-Source Heat Pump COP as a Function of Ambient Temperature (Figures Represent Run-Times in Minutes)

It can be seen that run times of around 5-6 minutes appear to give good performance but that this falls off dramatically as the run time is reduced to 4 minutes or less. Similar tests were carried out for ground-source heat pumps, as shown in Fig. UK-15 below.

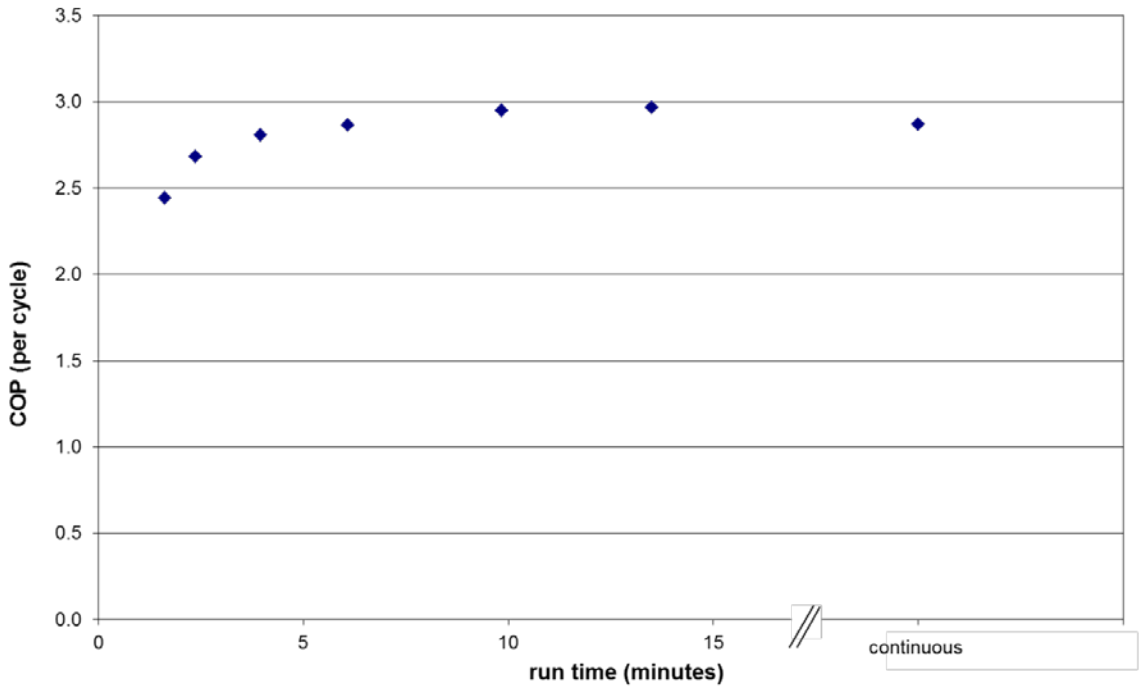


Figure UK-15: Test-House Measurements of Ground-Source Heat Pump (COP as a Function of Run-Time in Minutes)

For ground-source heat pumps, the COP is also reduced for short run times, although the effect is much less marked than for air-source. This is because the ground temperature does not cool as rapidly when the run time is short and this partially offsets the reduction in COP due to cycling of the compressor. Fig. UK-16 summarises this effect. Light blue triangles show the manufacturer's estimate of steady state COP; dark blue triangles show the measured COP as a function of run time, and pink squares show the ground return temperature, which falls as the run time increases.

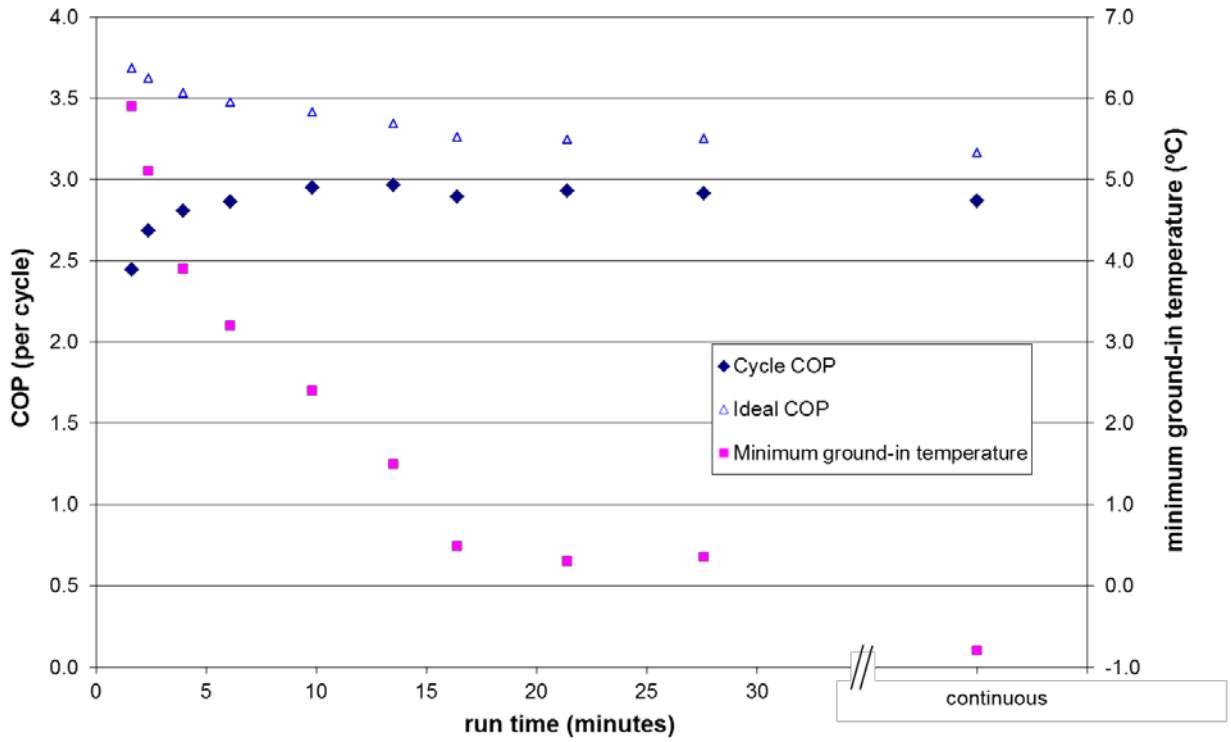


Figure UK-16: Ground-Source Heat Pump COP as a Function of Run-Time in Minutes, Test-House Measurements and Manufacturer's Data and Minimum Ground Return Temperatures

The study recommended that, for fixed speed compressors, the heating system should be designed so that the cycling period is always at least 6 minutes. This may be achieved by various means:

- Ensuring that the volume of water in the heating circuit is sufficient.
- Inserting a buffer tank.
- Increasing the hysteresis (most domestic heat pumps in the UK are controlled on return temperature, with a weather compensation algorithm. In this case, increasing the hysteresis means that the return temperature can drop further before the heat pump switches back on, thereby reducing cycling). Hysteresis ranges in typical heat pump circuits are 2-10˚C.

5.2. Integration of Buffer Tanks into Heat Pump Circuits

One way of reducing cycling is to include a buffer tank in the central heating circuit. This may be a "two pipe" system (i.e. situated either in the flow or return pipes), or a four pipe system (through which both the flow and return pipes flow).

Charlick (2013) carried out laboratory tests to examine the effect on efficiency of including buffer tanks in these three configurations. The study used a single flow temperature of 40˚C, a range of hysteresis temperatures from 2 to 10˚C and two sizes of buffer tank. The heat loss from the buffer tanks was assessed by calculation and thermal imaging. The study also measured return temperatures for the different buffer tank configurations. As would be expected, the lowest temperatures were found for the configuration with no buffer tank and the highest for the 4 pipe system. The size of the buffer tank had little effect.

There are two principles at work:

1) The heat pump is more efficient when the return temperature is lower.
2) A larger hysteresis means that the heat pump will cycle less. Run times of < 6 minutes adversely affect the performance, particularly for air-source heat pumps.

Experiments were carried out using a large heating volume (7 radiators), a medium heating volume (4 radiators) and a small heating volume (1 radiator). For the 7 and 4 radiator cases, ground-source heat pumps were relatively unaffected by the choice of configuration (2-3% when the return temperature changed from 40 to 41°C), while the performance of air-source heat pumps was improved when a buffer tank (of any configuration) was inserted, or when the hysteresis was increased for the case of no buffer tank.

The results for the one radiator case were most marked for the air-source heat pump. Fig. UK-17 shows the COP for various configurations of buffer tank, with 2˚C, 5˚C and 10˚C of hysteresis.

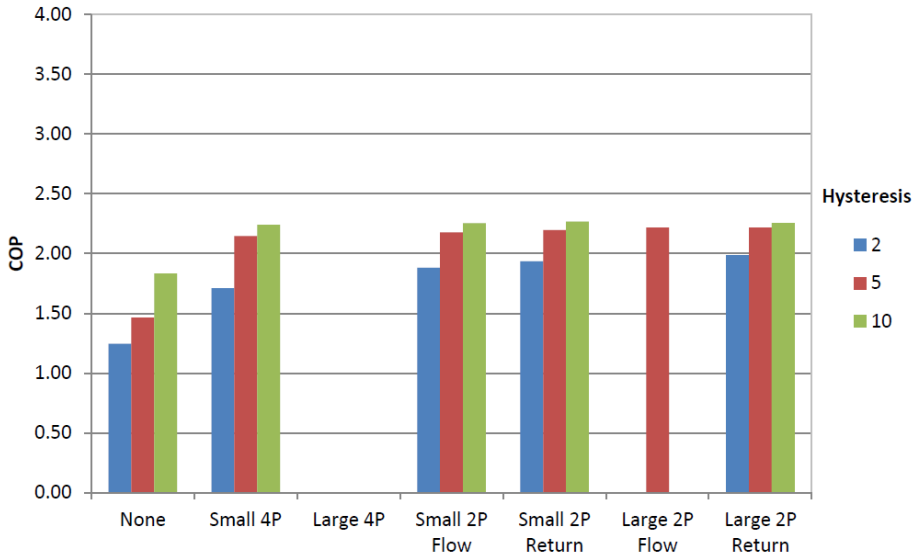


Figure UK-17: Air-Source Heat Pump COP as a Return Temperature Hysteresis and Buffer Tank Configuration, single radiator

Charlick (2013) concluded that buffer tanks are not necessary in all heat pump circuits, although they may be useful either when the volume of water heated is insufficient to avoid cycling or, alternatively, when heat pumps are operated on a cheap tariff and heat storage is necessary. If installed, a 2 pipe configuration in the return pipe is preferred.

5.3. Configuring Heat Pumps to Heat Domestic Hot Water Efficiently

Charlick (2013) undertook laboratory experiments to determine the most efficient way to heat domestic hot water with a heat pump, using a range of cylinder sizes, storage temperatures, heating patterns, sterilisation patterns and tapping patterns.

The first part of the study investigated the heat losses from the cylinder as a function of storage temperature, at low, medium and high consumption levels. Fig. UK-18 shows the results for a 180 litre cylinder.

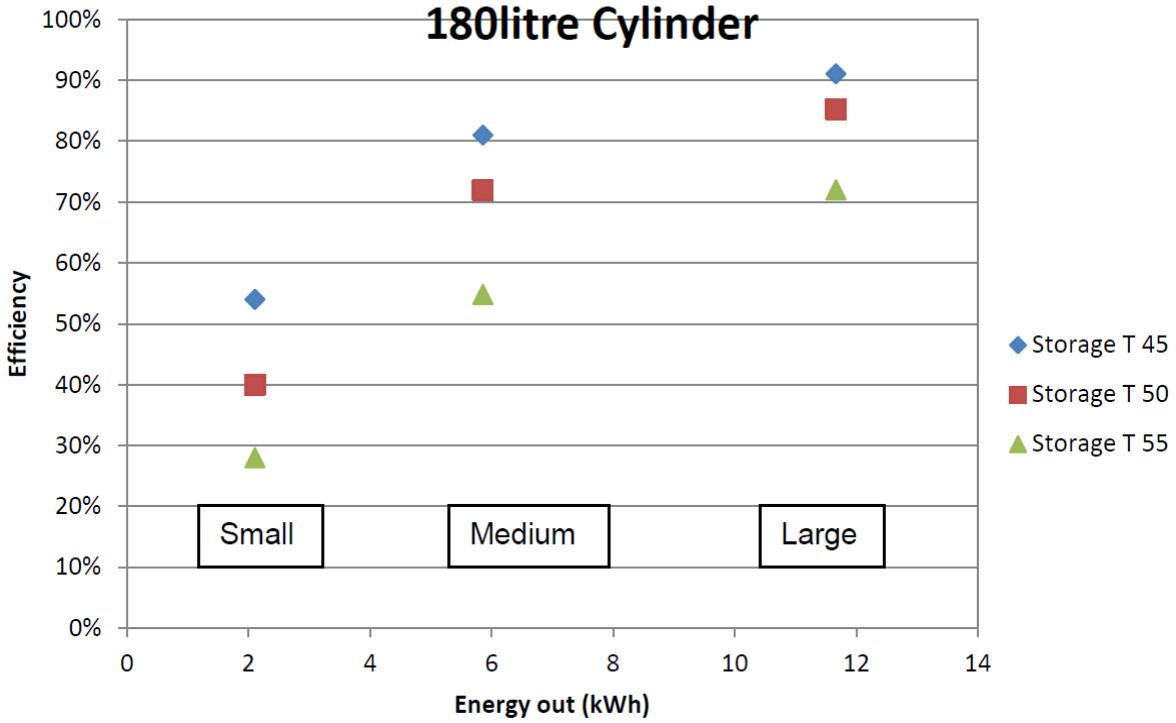


Figure UK-18: Storage Efficiency of Domestic Hot Water Cylinders as a Function of Storage Temperature and Householder Consumption

In all cases, cylinder losses increased by at least 20 percentage points as the storage temperature increases from 45˚C to 55˚C degrees. This would be compounded by a reduced efficiency of the heat pump when operating at higher temperatures (from the Heat Emitter Guide, the SPF4 for an air source heat pump would reduce from 3.0 at a flow temperature of 45˚C to 2.4 at 55˚C). For these reasons, the study recommends:

- For optimum efficiency, the storage temperature should be slightly lower than the flow temperature of the heat pump.
- Where usage is low, the cylinder should not be heated continuously.
- Weekly sterilisation to 60˚C should be carried out during periods of low use (eg at night).
- Heat pumps typically heat domestic hot water with a flow temperature of 45-50˚C and flow rate of 0.4 ls^{-1}. In order to transfer the heat to the water, a coil with a large surface area is required.
- Current kW ratings for heating coils in domestic hot water cylinders are calculated according to standard BS EN 12897 (2006), which applies to an 80˚C flow temperature and a 0.25 ls^{-1} flow rate. This is not appropriate for heat pumps and therefore a new system of rating is required.

6. REVISIONS TO INSTALLATION STANDARDS

6.1. Revisions to the MIS 3005 Standards

Following the findings of the first phase of the Energy Saving Trust Field Trials, DECC, the Energy Saving Trust and the UK heat pump industry worked together to improve the design and installation standards for small scale heat pumps (< 45 kWth). As a result, the MCS MIS 3005 Issue 3.1 (2012) standard came into force in March 2012. The principal differences between this standard and its predecessor are:

New sizing requirements; a heat pump must be able to supply the steady state space heating requirement of the house for 99% of the hours of a typical year without recourse to the electric auxiliary heater. In order to size the heat pump correctly, the installer must carry out a room by room calculation of heat loss in accordance with BS EN 12831 (2003). For air-source heat pumps, the reduction in capacity as the temperature falls must be taken into account.

New guidance on heat emitter design; the Heat Emitter Guide (2012a) allows the installer to size and design the heat emitters correctly. It assigns an efficiency rating based on the flow temperature at the design ambient temperature for the location; this is the temperature exceeded for 99% of the hours in a typical year and is location dependent. As part of the handover process, the installer must show the householder this efficiency rating, which is displayed as 1 to 6 stars (6 being the most efficient). For good performance, low flow temperatures are required; guidance is given on the appropriate spacing of pipes in underfloor systems and the appropriate sizing of radiators.

New guidance on designing ground loops and boreholes; the Ground Heat Exchanger Look-up Tables (MCS 022, 2012b) specify the power than can be extracted per meter of ground loop (or borehole) as a function of full time equivalent run hours, mean annual ground temperature and ground conductivity.

The new standards also provide guidance on the sizing of domestic hot water cylinders and re-iterate the need to insulate all pipes and tanks thoroughly and to sterilise the domestic hot water cylinder once per week to protect against Legionella and other bacteria.

6.2. Roadshows to Inform Installers about the New MIS 3005 Standards

In order to prepare installers for the new MIS 3005, DECC held ten roadshows throughout the UK, attended by 174 installers and manufacturers. Further details can be found in reference of the course can be found on the webinar reference for Microgeneration Certification Scheme (2012).

Delegates judged the course content as being well adapted to their needs. Table UK-13 indicates the degree of satisfaction with the training on six key issues:

Table UK-13: Delegates' Satisfaction Scores for the Installer Roadshows

	% Satisfied with course content
How to perform a power and energy heat loss calculation and estimated cost of various systems?	94
Knowledge of heat emitters and the link to a heat pump system?	94
Domestic HW cylinder selection based on BS 6700 (or EN 806)?	94
Noise and recirculation issues for ASHP?	95
Design for Ground Heat Exchangers, including Reynolds Number > 2500 & pumping power < 2.5%?	84
Pressure testing to EN 805, Commissioning, Handover and documentation	89

Sixty-none percent (69%) of delegates stated that they would be confident or very confident about being able to follow the standards correctly, while a further 23% stated that they would be "OK". 8% felt that they would require more training in order to feel able to implement the procedures.

Following these roadshows, a number of manufacturers and installation companies have started their own in house training on the new standards.

6.3. Roadshows to Inform Social Housing Providers about the New MIS 3005 Standards

Compared to other European countries, a relatively small proportion of households in the UK live in social housing (18%), as opposed to 13% in privately rented accommodation and 69% of owner-occupiers (Office of National Statistics, 2009).

However, social housing providers have been proactive in installing heat pumps and certain Government programs have been targeted at increasing the number of renewable heat measures installed in the social housing sector. It is therefore important to ensure that social housing providers have the information to act as intelligent customers when specifying heat pump systems. To this end, DECC commissioned ten one-day roadshows, at locations throughout the UK, to explain the key issues to social housing providers. The workshops were attended by 227 delegates and a webinar was published, see http://www.greenb.org.uk/heatpumps/.

6.4. Proposals for Further Revisions to Standards

As a result of the second phase of the Energy Saving Trust Field Trial, various proposals have been made to further improve standards. These include:

- Configuring controls carefully, so that the auxiliary electric heater, the domestic hot water immersion and (for air-source heat pumps) and defrost schedules are not engaged unnecessarily.
- Use of low energy circulation pumps on the central heating side (this will be mandatory when the EU Eco-design Directive comes into force).
- Configuring the heating system volume such that the cycling run-time is always greater than 6 minutes, even in mild conditions.
- Setting the domestic hot water cylinder storage temperature to just below the heat pump flow temperature and sterilising once a week at night time.
- Fitting the domestic hot water tank with large coils to improve heat transfer.

DECC is very keen to encourage detailed metering of heat pump performance as standard; in particular, monitoring of the auxiliary electric heater would alert the householder to faults. The Renewable Heat Incentive policy includes a number of proposals for monitoring of heat pump efficiency.

7. CONSUMER RIGHTS

Dunbabin et.al (2012) describes the various measures in place for consumer protection in the UK. A summary is presented here.

If a customer has an issue or complaint about a heat pump, it is likely to fall within at least one of the following categories (Energy Saving Trust, 2013):

- Marketing / Sales;
- Technical, relating to the product;
- Technical, relating to the installation (for example, performance of the installed system, including poor performance, inability to hear the house in cold weather, high running costs, insufficient domestic hot water supplied, design problems);
- Non-technical or contractual relating to the service and/or customer care, including warranty or maintenance issues, repair of damage, materials, planning permission, misinformation about or difficulty in registering for Government support schemes such as the Renewable Heat Premium Payment or the Renewable Heat Incentive, inadequate handover pack on completion of installation, company not MCS certified.

Protection is provided for customers in these four areas.

Sales / Marketing

The Trading Standards Institute operates codes of practice concerning such issues as restriction of cold calling, consumer protection from unfair trading and deposits. It takes disciplinary action against companies found to be in breach of the agreed codes of practice.

Products

The European Heat Pump Association (EHPA) Quality Label sets minimum coefficients of performance, as measured under laboratory conditions, for a range of temperatures. Tests must be carried out according to EN 14511 by approved laboratories and sound power levels must be supplied.

The Microgeneration Certification Scheme MCS certifies products based on:

a) Evidence of compliance with BS EN14511:2007 Parts 1 – 4 of "Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling". (Independent third party tests in an accredited test laboratory).
b) Verification of the establishment and maintenance of the manufacturing company's quality management system in accordance with the Factory Production Control requirements (FPC).
c) Review of the technical documentation relating to the material or product.

Note that the temperatures at which the tests are made are not exactly the same as those for the EHPA Quality Label.

Table UK-14 compares the two sets of requirements.

Table UK-14: Comparison of COP Requirements for MCS and for the EHPA Quality Label

HP type	Collector temperature °C	Emitter temperature °C	COP for MCS requirements	COP for EHPA requirements
Ground to Water	0	35	≥3.5	≥ 4.30
Ground to Air	0	35	≥3.2	not covered
Water to Water	10	35	≥3.8	≥ 5.10
Water to Air	10	35	≥3.5	Not covered
Air to Water	2	35	≥3.2	≥ 3.10
Air to Air	2	35	≥3.0	Not covered
Exhaust Air	2	35	≥2.5	Not covered
Direct exchange ground coupled to water	4	35	Not covered (would require Environment Agency permission in the UK)	≥ 4.30

Design and Installation

In order to receive any Government subsidies, householders must ensure that their installation is carried out by an installer accredited by the Microgeneration Certification Scheme (2012). The procedures for design and installation are set out in MSCS 022 (2013a), MCS 022 (2012b), Microgeneration Certification Scheme (2012), and Office of National Statistics (2009). As part of the procedure, the installer must show the householder the energy efficiency star rating of the installation, ranging from 1 (poor) to 6 (excellent), as described in section 6.1.

Contractual Issues

In addition to statutory law, consumer protection is covered by trade bodies and insurance backed guarantees.

The Renewable Energy Consumer Code (RECC)

The RECC has been approved by DECC for the Microgeneration Certification Scheme (2012). It covers the following:

- Marketing,
- Pre-contractual information,
- Quotations,
- Deposits,
- Contracts,
- Guarantees,
- After-sales service.

The RECC analyse complaints received and provided a free conciliation service; 250 serious complaints have been handled between 2007 and 2011, with 100 of the serious complaints resolved in the year 2010/2011. The Code is continuously being improved to address issues arising such as tightening the application processes and providing training on consumer protection legislation.

The RECC has also recently introduced a Deposit and Advance Payment Insurance Scheme for the insurance of all consumer funds paid in advance, which was set up in response to risks facing small businesses (i.e. in the event the company goes into receivership before the installation has taken place) and therefore their customers. This also includes insurance of workmanship warranties to cover consumers if the company goes out of business before the end of the insurance period.

The Renewable Energy Installation Guarantee Agency (REIGA)
Other insurance-backed guarantees exist, such as the REIGA scheme, which provides an independent 10 year guarantee for the installation of renewable energy devices, such that customers with these devices will not be reliant on the installers to guarantee their installations. If a customer has a problem with an installer who is a member of RIEGA, REIGA will approach the installer on behalf of the customer to establish the details and appoint a surveyor to report on the issues. REIGA will then take the necessary action to ensure the issue is resolved, which may involve appointing another installer to remedy any problem.

It is important to note that none of these insurance-backed guarantees currently guarantees a minimum seasonal performance factor in the customer's home. Some manufacturers are starting to monitor performance of domestic installations. DECC strongly encourages this approach, which should result in better customer protection.

8. FURTHER RECOMMENDATIONS

It is essential to define which system boundaries are being used when referring to seasonal performance factors. DECC considers that SPF_{H4} gives the best overall indication of the combined quality of a product and its installation and would therefore recommend that SPF_{H4} is used whenever possible. The MCS MIS Heat Emitter Guide (MCS 022, 2012b) currently requires the installer to provide the householder with an estimate of SPF_{H4} for space heating.

DECC would strongly recommend that the electricity used by auxiliary heater and domestic hot water immersion is measured and that appropriate diagnostics are given to the householder. Some manufacturers already monitor the electricity consumption of each component of the heat pump and DECC would encourage others to do so. DECC also considers that controls should be absolutely clear to the average householder. It is not sufficient to have a "winter" setting which activates the auxiliary electric heating.

Software controlling the sterilisation pattern should be carefully configured so that the immersion is not used excessively. There is a case for research projects to investigate alternative methods of controlling bacteria such as Legionella.

For air-source heat pumps, software controlling defrost should be carefully configured so that defrost is used only when required. Appropriate diagnostics should be provided to the householder.

DECC recommends that the volume of the heating system should be sufficient to avoid rapid cycling (with a run time of < 6 minutes) in mild temperatures (10-15˚C) since this has an adverse effect on seasonal performance factor.

Specifically for the UK market, DECC proposes that MCS MIS 3005 should be amended to recommend that the storage temperature of the domestic hot water tank should be kept just below the temperature at which the heat pump can supply heat (with the exception of the weekly sterilisation cycle). This should reduce unnecessary use of the electric immersion.

DECC proposes that MCS MIS 3005 should be amended so that the installer shows the householder a weighted average of SPF_{H4} for space and water heating (if domestic hot water is supplied by the heat pump).

In Sweden, the insurers' group Folksam collects data on heat pump faults. This could be a useful strategy for all microgeneration measures.

9. ACKNOWLEDGEMENTS

With special thanks to Jaryn Bradford, Tom Byrne and Jack Melling of the Energy Saving Trust for setting up and managing the field trials and to Helen Charlick and Iain Summerfield of Kiwa, Rob Green and Tony Knowles of EA Technology for many hours of patient and detailed analysis. Many thanks also to Colin Gleeson (C.Eng) from the University of Westminster and Roger Nordman of SP, Sweden for peer review, to Sarah Gaunt and David Matthews of the National Energy Foundation and GSHPA for their input on the literature survey and to Zoltan Karpathy of BSRIA for supplying sales data. Particular thanks is due also to the Microgeneration Certification Scheme members for their hard work on updating the MCS standards and the continued dialog with DECC.

10. REFERENCES

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COUNTRY REPORT:
US CONTRIBUTION TO THE IEA HEAT PUMP PROGRAMME ANNEX 36 ON QUALITY OF INSTALLATION AND MAINTENANCE OF HEAT PUMPS

Sensitivity Analysis of Installation Faults on Heat Pump Performance

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ABSTRACT

Numerous studies and surveys indicate that typically-installed HVAC equipment operate inefficiently and waste considerable energy due to different installation errors (faults) such as improper refrigerant charge, incorrect airflow, oversized equipment, leaky ducts. This study seeks to develop an understanding of the impact of different faults on heat pump performance installed in a single-family residential house. It combines building effects, equipment effects, and climate effects in a comprehensive evaluation of the impact of installation faults on a heat pump's seasonal energy consumption through simulations of the house/heat pump system.

The study found that duct leakage, refrigerant undercharge, oversized heat pump with nominal ductwork, low indoor airflow due to undersized ductwork, and refrigerant overcharge have the most potential for causing significant performance degradation and increased annual energy consumption. The effect of simultaneous faults was found to be additive (e.g., duct leakage and non-condensable gases), little changed relative to the single fault condition (e.g., low indoor airflow and refrigerant undercharge), or well-beyond additive (duct leakage and refrigerant undercharge). A significant increase in annual energy use can be caused by lowering the thermostat in the cooling mode to improve indoor comfort in cases of excessive indoor humidity levels due to installation faults.

The goal of this study was to assess the impacts that HVAC system installation faults had on equipment electricity consumption. The effect of the installation faults on occupant comfort was not the main focus of the study, and this research did not seek to quantify any impacts on indoor air quality or noise generation (e.g., airflow noise from air moving through restricted ducts). Additionally, the study does not address the effects that installation faults have on equipment reliability/robustness (number of starts/stops, etc.), maintainability (e.g., access issues), or costs of initial installation and ongoing maintenance.

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1. INTRODUCTION

Space cooling is responsible for the largest share (at 22.3 %) of the electrical energy consumption in the U.S. residential sector (DOE, 2011). Space heating, for which a significant portion is provided by heat pumps, accounts for an additional 8.7 % electricity use. Consequently, there are increasing requirements that space-conditioning equipment be highly efficient to improve building energy efficiency as well as address environmental concerns. To this end, state and municipal governments and utility partners have implemented various initiatives that promote sales of high-efficiency air conditioners (ACs) and heat pumps (HPs). However, there is a growing recognition that merely increasing equipment's laboratory-measured efficiency without ensuring that the equipment is installed and operated correctly in the field is ineffective. A key component for maximizing field equipment performance is to ensure that such equipment is sized, selected, and installed following industry recognized procedures. Consistent with this goal, the Air Conditioning Contractors of America (ACCA) released in 2007 a quality installation (QI) standard for heating, ventilating and air-conditioning (HVAC) equipment, which has been updated since then and achieved widespread recognition by various entities in the U.S. concerned with reducing energy consumption by buildings (ACCA, 2010). A companion standard (ACCA, 2011b) defines the verification protocols to ensure that HVAC systems have been installed according to the QI Standard. A related ACCA standard (ACCA, 2013) addresses residential maintenance issues.

Numerous studies and surveys indicate that typically-installed HVAC equipment operate inefficiently and waste considerable energy due to different installation errors (faults) such as improper refrigerant charge, incorrect airflow, oversized equipment, leaky ducts. However, it is unclear whether the effects of such faults are additive, whether small variances within a given fault type are significant, and which faults (in various applications and geographical locations) have a larger impact than others. If this information is known, better attention, resources, and effort can be focused on the most important design, installation, and maintenance parameters.

This project seeks to develop an understanding of the impact of different commissioning parameters on heat pump performance for a single-family residential house application. It combines building effects, equipment effects, and climate effects in a comprehensive evaluation of the impact of installation faults on seasonal energy consumption of a heat pump through simulations of the house/heat pump system. The evaluated commissioning parameters include:

- Building subsystem:
  - Duct leakage (unconditioned space).
- Residential split, air-to-air heat pump equipped with a thermostatic expansion valve (TXV):
  - Equipment sizing,
  - Indoor coil airflow,
  - Refrigerant charge,
  - Presence of non-condensable gases,
  - Electrical voltage,
  - TXV undersizing.
- Climates (cooling and heating):
  - Hot and humid,
  - Hot and dry,
  - Mixed,
  - Heating dominated,
  - Cold.
- Single-family houses (the structures representative for the climate):
  - House on a slab,
  - House with a basement.

The goal of this study is to assess the impacts that HVAC system installation faults have on equipment electricity consumption. The effect of the installation faults on occupant comfort is not the main focus of the study, and this research did not seek to quantify any impacts on indoor air quality or noise generation (e.g., airflow noise from air moving through restricted ducts). Additionally, the study does not address the effects that installation faults have on equipment reliability/robustness (number of starts/stops, etc.), maintainability (e.g., access issues), or costs of initial installation and ongoing maintenance.

2. LITERATURE SURVEY

The literature survey is presented in three sections. Section 2.1 presents selected publications related to air conditioner and heat pump installation and maintenance issues; Section 2.2 focuses on heat pump oversizing/undersizing and cycling losses; and Section 2.3 presents relevant studies on heat pump fault detection and diagnostics (FDD).

2.1. Field Surveys, Installation and Maintenance Issues

Numerous field studies have documented degraded performance and increased energy usage for typical air conditioners and heat pumps installed in the United States. Commonly, system efficiency, peak electrical demand, and comfort are compromised. This degraded performance has been linked to several problems, which include:

- improperly designed, insulated, or balanced air distribution systems in the house;
- improperly selected heat pump, either by the fact of overall performance characteristics due to mix-matched components, or improper capacity (too large or too small) in relation to the building load;
- heat pump operating with a fault.

The first two problem categories are a result of negligent or incompetent work prior to the heat pump installation. The third problem category, a heat pump operating with a fault, can be a result of improper installation or improper maintenance. Field study reports describing observations and measurements on new installations are less common than publications on existing installations. For this reason, in this literature review, we also include reports on maintenance practices, in particular those covering large numbers of systems.

While discussing heat pump performance measurements taken in the field, we have to recognize that these field measurements offer significant challenges and are burdened by a substantial measurement uncertainty, much greater than the uncertainty of measurements in environmental chambers, which are in the order of 5 % at the 95 % confidence level. Typically, field study reports do not estimate the measurement uncertainty of the reported values, but the number of installations covered by some of these studies provides an informative picture about the scope and extent of field installation problems. We may also note that most of the articles on field surveys are not published in indexed journals. Consequently, they are not searchable by publication search engines, and many of them are not readily available. In this literature review, we gave a preference to citing publications which can be readily obtained by a reader if desired.

In a study of new installations, Proctor (1997) performed measurements on a sample of 28 air conditioners installed in 22 residential homes in a hot and dry climate (Phoenix, AR, USA). Indoor heat exchanger airflow averaged 14 % below specifications, and only 18 % of the systems had a correct amount of refrigerant. The supply duct leakage averaged 9 % of the air handler airflow, and the return leakage amounted to 5 %. The author cites several prior publications, which reported similar problems.

Davis and Robison (2008) monitored seven new high efficiency residential heat pumps. They diagnosed several installation errors, which included a malfunctioning TXV, non-heat pump thermostat installed, incorrect indoor unit installed, and incorrect control wiring preventing proper system staging. The authors reported that once the problems were repaired, the systems performed at the expected levels.

Parker et al. (1997) investigated the impact of indoor airflow on residential air conditioners in 27 installations in Florida. They measured airflows ranging from 62.8 $m^3 \cdot h^{-1} \cdot kW^{-1}$ to 246.4 $m^3 \cdot h^{-1} \cdot kW^{-1}$ (130 cfm/ton to 510 cfm/ton) while a typical manufacturer's recommendation calls for 193.2 $m^3 \cdot h^{-1} \cdot kW^{-1}$ (400 cfm/ton). Undersized return ducts and grills, improper fan speed settings, and fouled filters were the causes of improper airflow along with duct runs that were long, circuitous, pinched or constricted. Additional flow resistance can result from the homeowner tendency to increase air filtration via higher efficiency filters during replacement; the measurements showed that substitution of high-efficiency filters typically reduces the airflow by 5 %. Low airflow has system energy-efficiency implications; reduction of airflow by 25 % from 193.2 $m^3 \cdot h^{-1} \cdot kW^{-1}$ to 144.9 $m^3 \cdot h^{-1} \cdot kW^{-1}$ (400 cfm/ton to 300 cfm/ton) can reduce the efficiency of the air conditioner by 4 %. The authors commented that airflows below 169.1 $m^3 \cdot h^{-1} \cdot kW^{-1}$ (350 cfm/ton) render

invalid most field methods for determining refrigerant charge and can lead to improper charging by a service technician who often does not check the evaporator airflow.

Downey and Proctor (2002) reported on the field survey of 13,000 air conditioners installed on residential and commercial buildings. The measurements were collected during routine installation, repair, and maintenance visits. Of the 8,873 residential systems tested, 5,776 (65 %) required repairs, and of the 4,384 light commercial systems tested, 3,100 (71 %) required repairs. Improper refrigerant charge was found in 57 % of all systems. The authors noted that the simple temperature split method for identifying units with low airflow is flawed because it does not account for the system operating condition.

Proctor (2004) presented results from a survey study involving 55000 units. He reported that 60 % of commercial air conditioners and 62 % of residential air conditioners had incorrect refrigerant charge. In all, 95 % of residential units failed the diagnostic test because of duct leakages, poor duct insulation or excessive airflow restriction, improper refrigerant charge, low evaporator airflow, non-condensables in the refrigerant or an improperly sized unit.

Rossi (2004) presented measured performance data and statistics on unitary air conditioners. The data were gathered using commercially available portable data acquisition systems during normal maintenance and service visits. Out of 1,468 systems considered in this study, 67 % needed service. Of those 15 % required major repairs (e.g., compressor or expansion device replacement), and 85 % required a tune-up type service (e.g., coil cleaning or refrigerant charge adjustment). Approximately 50 % of all units operated with efficiencies of 80 % or less, and 20 % of all units had efficiencies of 70 % or less of their design efficiency.

Mowris et al. (2004) reported on field measurements of refrigerant charge and airflow, commonly referred to as RCA. Over a three-year period, 4,168 new and existing split, package, and heat pumps were tested. The measurements showed that 72 % of the tested units had improper refrigerant charge, and 44 % had improper airflow. Approximately a 20 % efficiency gain was measured after refrigerant charge and airflow were corrected.

Neme et al. (1999) considered four installation issues − equipment sizing, refrigerant charging, adequate airflow, and sealing ducts − and assessed the potential benefits from improved installation practices. The authors relied on an extensive list of publications to determine the range of intensity of the four installation faults and the probable air conditioner efficiency gain resulting from a corrective action. The cited literature indicated the maximum efficiency improvement of 12 % for corrected airflow, 21 % for corrected refrigerant charge, and 26 % for eliminated duct leakage. The authors concluded that improved HVAC installation practices could save an average of 25 % of energy in existing homes and 35 % in new construction. They also pointed out that air conditioner oversizing has the potential of masking a number of other installation problems, particularly improper refrigerant charge and significant duct leakage, while a correctly sized air conditioner makes other installation problems more apparent, particularly at severe operating conditions.

Neal (1998) presented a methodology for calculating a field-adjusted seasonal energy efficiency ratio, which he referred to as SEERFA, with the goal to account for four installation errors and better represent the seasonal performance of the air conditioner installed in the field as compared to the seasonal energy efficiency ratio (SEER) derived from tests in environmental chambers. He used correcting factors of value 1 or smaller, one for each installation fault, which act as multipliers on the SEER. He provided an example indicating that, on average, a homeowner's cooling cost is approximately 70 % higher than it could be with quality air conditioner installation. It should be noted that the proposed algorithm assumes no interaction between different faults, which seems to be an improper assumption.

While the scope and specific findings presented in the above publications may differ, they uniformly document the prevalence of air conditioner and heat pump faults in the field and a significant performance degradation of this equipment.

2.2. Heat Pump Oversizing and Undersizing and Part-load Losses

It is generally accepted that equipment oversizing will lead to significant part load losses due to cycling. Unit cycling increases energy use due to efficiency losses (Parken et al., 1985) and also can degrade the

moisture removal capacity of the unit which leads to higher space humidity levels (Shirey et al., 2006). For nearly 50 years, proper sizing for residential air conditioners and heat pumps has typically been defined using the ACCA Manual J (ACCA, 2011a).

The energy efficiency of a cycling system is governed by how quickly after startup the capacity and efficiency of the air conditioning unit reaches steady-state conditions. Parken et al. (1977) defined the "Cyclic Degradation" parameter (C_D) as a simplified metric to predict part-load losses. This parameter was integrated into the calculation procedure to determine the SEER for air conditioners and heat pumps. That procedure has been incorporated into federal energy efficiency standards (Federal Register, 1979) and into AHRI Standard 210/240 (AHRI, 2008). The default value for C_D in these calculation procedures is 0.25.

Many researchers have demonstrated the sensible and latent capacity of the air conditioner at startup is a complicated process (Henderson, 1990; O'Neal and Katipamula, 1991). The response includes the delays associated with pumping refrigerant from the low-side to the high-side of the system to establish the steady-state operating pressures as well as the first order delays due to heat exchanger capacitance. Several models have been proposed that represent the overall response as some combination of first order (time-constant) response, delay times, and other non-linear effects. Henderson (1992) compared all these and showed they generally could be represented as an equivalent time constant.

As part of developing a model for latent degradation, Henderson and Rengarajan (1996) showed that the parameter C_D can be directly related to equivalent time constant for capacity at startup while assuming a thermostat cycling rate parameter (Nmax) of 3.1 cycles per hour. O'Neal and Katipamula (1991) and Parken et al. (1977) also indirectly showed a similar relationship. The default value of 0.25 for C_D is equivalent to an overall time constant of 1.27 minutes.

Over the years, since the SEER test and rating procedure has been developed, manufacturers have had a strong incentive to improve the cyclic performance of their systems. Dougherty (2003) demonstrated that the typical value of C_D is now in the range 0.05 to 0.10 for most systems. So cyclic degradation and the part-load efficiency losses may be of less consequence than was previously thought.

Henderson and Rengarajan (1996) developed a similar part load model to consider the degradation of air conditioner latent or moisture removal capacity at cyclic conditions. This model focused on situations when the fan operated continuously, but the compressor cycled. A more comprehensive study was completed by Shirey et al. (2006), and a more detailed model was developed with physically-based model parameters. The resulting model and the more comprehensive understanding of parametric conditions for a wide variety of systems and conditions allowed them to develop a refined model for latent degradation that could also consider the case when the fan cycles on and off with the compressor (Auto Fan Mode) – the practice most commonly used with residential systems.

Field testing and simulation analysis have been used to assess the impact of oversizing on energy use and space humidity levels. Sonne et al. (2006) changed out oversized air conditioner units in four Florida houses and replaced them with units sized according to ACCA Manual J (ACCA, 2011a). Detailed performance data was collected both before and after the right-sized unit was installed. Their study found mixed results in terms of seasonal energy use and space humidity levels. In some houses, energy use was higher, in some it was lower, and in others the results were inconclusive. Similarly, relative humidity (RH) appears to be either slightly higher and or unchanged after the right-sized unit was installed. They also speculated that duct leakage impacts were greater for the right-sized unit since longer periods of system operation were required to meet the same load. More duct leakage increases the thermal losses to the attic (supply ducts are colder for longer 'on' periods) and brings in more fresh air into the system. Both these effects increase the sensible and latent loads imposed on the system.

A simulation study by Henderson et al., (2007) also confirmed the modest and somewhat unexpected impact of oversizing. They found that, when 20% of duct leakage was factored into the simulations, both energy use and space humidity levels were only slightly affected, even when both latent degradation effects and part load cyclic efficiency losses were considered. For example, oversizing by 30 % in Miami for the HERS Reference house increased energy use by only 2 % and actually resulted in slightly lower space humidity levels.

2.3. Laboratory Studies of Performance Degradation of Heat Pumps Due to Faults

Several studies on degradation of the air conditioner and heat pump performance due to different faults are documented in the literature. While in most cases the main interest of these studies was the fault detection and diagnosis (FDD), some of the findings can be used in the analysis of effects of faulty installation. Reports of major studies on FDD for HVAC systems started to appear in the literature in the nineties, and the number of publications noticeably increased in the last fifteen years.

Table US-1 lists a few examples of studies published since 2001. The reports by Kim et al. (2006) and Payne et al. (2009) present detailed literature reviews up to the dates these reports were published and include laboratory data for the cooling and heating mode, respectively. These laboratory data are used in our report; however, they had to be extended through tests in environmental chambers to provide complete coverage of the whole range of installation faults of interest in this study (see chapter 3 of this report).

Table US-1. Selected studies on faults detection and diagnosis

Investigators	System Type	Study Focus
Comstock and Braun (2001)	Centrifugal chiller	Experiment, eight single faults
Kim et al. (2006, 2009)	Split residential heat pump	Experiment for cooling mode, single-faults
Chen and Braun (2001)	Rooftop air conditioner	Simplified rule-based chart method
Navarro-Esbri et al. (2007)	General vapor compression system	Dynamic model based FDD for real-time application
Payne et al. (2009)	Single-speed, split residential heat pump	Experiment for heating model, single-faults
Wang et al. (2010)	HVAC system for new commercial buildings	System-level FDD involving sensor faults
Cho et al. (2005)	Air-handling unit for buildings	Multiple faults
Li and Braun (2007)	Direct expansion vapor compression system	Multiple faults
Du and Jin (2008)	Air handling unit	Multiple faults
Southern California Edison Design and Engineering Services (SCE 2012)	Single-speed, split residential air conditioner	Single faults, dual faults, and triple faults

A large number of laboratory cooling mode tests were performed by Southern California Edison (SCE 2012) to determine the effects of common faults on air conditioner performance. These faults included indoor airflow, outdoor airflow, refrigerant charge, non-condensables, and liquid line restrictions.

SCE single-fault tests at a low refrigerant charge showed similar degradations in cooling capacity and total power as Kim et al. (2006); SCE reported -3 % and 0 % change in cooling capacity and total power, respectively, at 13 % undercharge while Kim et al. (2006) reported -5 % and -2 % change at 10 % refrigerant undercharge. However, at higher fault levels, SCE measured much higher performance degradation than Kim et al.; cooling capacity and total power changed by -54 % and -5 %, respectively, at 27 % undercharge (SCE) compared to -17 % and -3 % at 30 % undercharge (Kim et al., 2006). These large differences in cooling capacity change for a similar fault level exemplify differences in the effect a given fault may have on different systems. In the case of refrigerant undercharge fault, it is possible that different internal volumes were a factor in the different system responses.

SCE also performed several tests with dual and triple faults, which included reduction of the outdoor airflow by imposing different levels of airflow restriction. For the highest level of outdoor airflow blockage, 40 % refrigerant undercharge, and 56 % reduction in indoor airflow, the cooling capacity decreased by almost 70 %. The conducted multiple fault tests show the range of possible performance degradation, however, more tests are required to allow modeling of these faults within annual simulations of the house/heat pump system.

3. HEAT PUMP PERFORMANCE DEGRADATION DUE TO FAULTS

A significant number of laboratory tests were taken by Kim et al. (2006) and Payne et al. (2009) to characterize heat pump performance degradation due to faults. For the purpose of this study, we conducted additional tests using the same heat pump and test apparatus to expand the ranges of previously studied faults and to include faults that were not covered earlier, specifically, improper electric line voltage and improper liquid line subcooling. The goal of this experimental effort was to enable the development of correlations that characterize the heat pump performance operating with these faults. These correlations are presented in a non-dimensional format with performance parameters expressed as a function of operating conditions and fault level.

3.1. Laboratory Measurements

3.1.1. Experimental Apparatus and Test Conditions

The studied system was a single-speed, split heat pump with an 8.8 kW (2.5 ton) rated cooling capacity. The heat pump was equipped with a thermostatic expansion valve (TXV). Figure US-1 shows a schematic diagram of the experimental setup with the locations of the main measurements. The air-side measurements included indoor dry-bulb and dew-point temperatures, outdoor dry-bulb temperature, barometric pressure, and pressure drop across the air tunnel (not shown on the schematic). Twenty-five node, T-type thermocouple grids and thermopiles measured air temperatures and temperature change, respectively. On the refrigerant side, pressure transducers and T-type thermocouple probes measured the inlet and exit parameters at every component of the system.

Table US-2 presents the cooling and heating test conditions (indoor dry bulb, indoor dew point, and outdoor dry bulb temperatures), and Table US-3 presents the measurement uncertainties. For the uncertainty analysis and detailed description of the experimental setup the reader should refer to Kim et al. (2006).

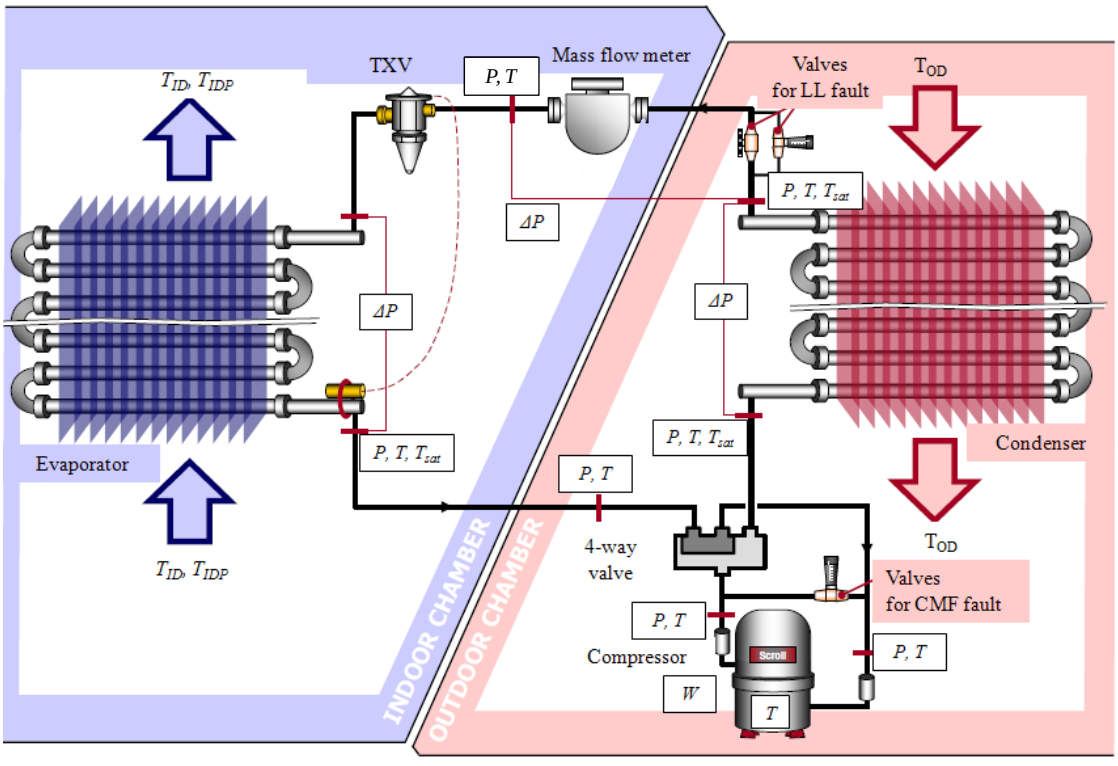


Figure US-1. Schematic diagram of experimental apparatus (Kim et al. (2006))

Table US-2. Cooling and heating test temperatures

Cooling			Heating		
T_{ID} °C (°F)	T_{IDP} °C (°F)	T_{OD} °C (°F)	T_{ID} °C (°F)	T_{IDP} °C (°F)	T_{OD} °C (°F)
21.1 (70)	10.3 (50.5)	27.8 (82)	18.3 (65)	dry	-8.3 (17)
21.1 (70)	10.3 (50.5)	37.8 (100)	21.1 (70)	dry	-8.3 (17)
26.7 (80)	15.8 (60.4)	27.8 (82)	21.1 (70)	dry	1.7 (35)
26.7 (80)	15.8 (60.4)	35.0 (95)	21.1 (70)	dry	8.3 (47)
26.7 (80)	15.8 (60.4)	37.8 (100)			

Note: The dew-point temperature in the cooling mode corresponds to a relative humidity of 50 %

Table US-3. Measurement uncertainties

Measurement	Measurement Range	Uncertainty at the 95% confidence level
Air dry-bulb temperature	(-9 ~ 38) °C ((15 ~ 100) °F))	±0.4 °C (±0.7 °F)
Air dew-point temperature	(0 ~ 38) °C (32 ~ 100) °F)	±0.4 °C (±0.7 °F)
Air temperature difference	(0 ~ 28) °C ((0 ~ 50) °F)	±0.3 °C (±0.5 °F)
Air nozzle pressure	(0 ~ 1245) Pa ((0 ~ 5) in H_2O)	±1.0 Pa (0.004 in H_2O)
Refrigerant temperature	(-12 ~ 49) °C ((10 ~ 120) °F)	±0.3 °C (±0.5 °F)
Refrigerant mass flow rate	(0 ~ 272) kg/h ((0 ~ 600) lb/h)	±1.0 %
Cooling capacity	(3 ~ 11) kW ((3 ~ 11) kW)	±4.0 %
Power	(25 ~ 6000) W ((25 ~ 6000) W)	±2.0 %
COP	2.5 ~ 6.0	±5.5 %

3.1.2. Studied Faults and Their Implementation

Table US-4 lists seven studied faults, including their definition and range. The first six faults were studied experimentally. The impact of the last listed fault, cooling-mode TXV undersizing, was determined based on a detailed analysis; the inherent variable-opening capability masks the TXV undersizing, and the performance penalty occurs only after the outdoor temperature is below a certain threshold temperature, referred to by us as the "departure temperature", which is related to the level of this fault. We did not include the TXV mismatched fault in the heating mode because it is very unlikely to occur as the heating TXV is installed in the outdoor section at the factory at time of assembly.

Table US-4. Definition and range of studied faults

Fault name	Symbol	Definition of fault level	Fault range (%)
Improper indoor airflow rate	AF	% above or below correct airflow rate	-50 ~ 20
Refrigerant undercharge	UC	% mass below correct (no-fault) charge	-30 ~ 0
Refrigerant overcharge	OC	% mass above correct (no-fault) charge	0 ~ 30
Improper liquid line refrigerant subcooling (indication of improper refrigerant charge)	SC	% above the no-fault subcooling value	0 ~ 200
Presence of non-condensable gases	NC	% of pressure in evacuated indoor section and line set, due to non-condensable gas, with respect to atmospheric pressure	0 ~ 20
Improper electric line voltage	VOL	% above or below 208 V	-8.7 ~ 25
TXV undersizing, cooling	TX	% below the nominal cooling capacity	-60 ~ -20

The indoor airflow fault was implemented by lowering the speed of the nozzle chamber booster fan to increase the external static pressure across the indoor air handler. The fault level was calculated as a ratio of the fault-imposed air mass flow rate to the no-fault air mass flow rate, with the -100 % fault level indicating a complete loss of airflow.

The no-fault refrigerant charge was set in the cooling mode at the AHRI 210/240 Standard A-Test condition (AHRI, 2008). The refrigerant undercharge and overcharge faults were implemented by adding or removing the refrigerant from a correctly charged system. The fault level was defined as the ratio of the refrigerant mass by which the system was overcharged or undercharged to the no-fault refrigerant charge, with 0 % indicating the correct, no-fault charge, -100 % indicating no refrigerant charge, and 100 % indicating doubled charge.

The amount of refrigerant in a TXV-equipped system can also be estimated by examining the refrigerant subcooling in the liquid line; this method is commonly used by field technicians installing or servicing a heat pump. Therefore, we also characterized the effect of refrigerant overcharge by noting the liquid line subcooling at increased charge levels. The ratio of fault-imposed subcooling to the no-fault subcooling indicated the fault level with the 0 % fault corresponding to the proper subcooling, and the 100 % fault indicating a doubled subcooling level.

The non-condensable gas fault is caused by incomplete evacuation of the system during installation or after a repair that required opening the system to the atmosphere. When a new heat pump is installed, the outdoor unit is typically pre-charged, and the installer needs to evacuate the indoor section and the connecting tubing before charging it with refrigerant. Industry practice (ACCA, 2010) is to evacuate the system to a vacuum of 500 μPa (29.9 in Hg vacuum). The non-condensable gas fault was implemented by adding dry nitrogen to the evacuated system before the charging process. This fault level is defined by the ratio of pressure in the evacuated indoor section due to non-condensable to the atmospheric pressure. The 0 % fault level occurs when the refrigerant charging process starts with a vacuum, and the 100 % fault level would occur when the nitrogen filled refrigerant lines are at atmospheric pressure before the refrigerant is charged.

The electrical line voltage fault was implemented by varying the supply voltage to the system from the nominal, no-fault value of 208 VAC. The fault level was defined by the percentage by which the line voltage was above or below the nominal level, with a positive fault indicating a voltage above 208 VAC.

TXV mismatch results in the TXV being unable to adjust its opening to match the refrigerant mass flow rate pumped by the compressor. This fault level is defined as the ratio of the difference in the nominal system capacity and the TXV capacity with respect to the nominal system capacity. With this definition, it is assumed TXVs are rated at the midpoint of their opening range of ±40 %.

3.2. Fault Effects on Cooling Mode Performance

3.2.1. Cooling Mode Normalized Performance Parameters and Correlations

The cooling mode tests considered the effect of faults on six performance parameters: total cooling capacity ($Q_{tot,}$, capacity includes the indoor fan heat), refrigerant-side cooling capacity ($Q_{R,}$, capacity does not include the indoor fan heat), coefficient of performance (COP), sensible heat ratio (SHR), outdoor unit power ($W_{ODU,}$ includes the compressor, outdoor fan, and controls powers), and total power (W_{tot}, includes W_{ODU} and indoor fan power). These parameters are presented in a dimensionless, normalized format obtained by dividing the values as obtained for the heat pump operating under a selected fault, to their value obtained for the heat pump operating fault free. We used Eq. (3.1) to correlate the dimensionless parameters as a function of the indoor dry-bulb temperature (T_{ID}), outdoor dry-bulb temperature (T_{OD}), and fault level (F).

$$Y=\frac{X_{\text{fault}}}{X_{\text{no-fault}}}=1+(a_1+a_2T_{ID}+a_3T_{OD}+a_4F)F \tag{3.1}$$

where a_1, a_2, a_3, and a_4 are correlation coefficients, X_{fault} and $X_{\text{no-fault}}$ are performance parameters for a faulty and fault-free heat pump, and Y is a dimensionless parameter representing the ratio of the faulty performance from that of the fault-free heat pump.

Table US-5 shows coefficients for a correlation using three input variables, T_{ID}, T_{OD}, and F. The coefficients were determined by means of a multivariate, polynomial regression method using the normalized values of performance parameters determined from heat pump test data. If the heat pump is fault free, values of all normalized parameters equal unity. The fit standard error of the normalized correlation dependent variable, Y, was a maximum of 3 % over the range of operating conditions listed in Table US-2. Table US-6 shows an example of propagation of uncertainty for the faulty COP and cooling capacity obtained from calculations using the measurement uncertainties of the corresponding fault-free values and the 3 % uncertainty in the dimensionless parameter Y.

Table US-5. Correlations for non-dimensional performance parameters in the cooling mode

Fault	Performance parameter Y	$Y=1+(a_1+a_2T_{ID}+a_3T_{OD}+a_4F)\cdot F^*$				FSE**
		a_1	a_2	a_3	a_4	
Improper indoor airflow rate (AF)	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				1.65E-02
	Q_{tot}	1.85E-01	1.77E-03	-6.40E-04	-2.77E-01	1.53E-02
	Q_R	2.95E-01	-1.17E-03	-1.57E-03	6.92E-02	5.39E-03
	SHR***	5.93E-02	5.16E-03	1.81E-03	-2.89E-01	9.82E-03
	W_{ODU}	-1.03E-01	4.12E-03	2.38E-03	2.10E-01	6.91E-03
	W_{tot}	1.35E-02	2.95E-03	-3.66E-04	-5.88E-02	5.68E-03
Refrigerant undercharge (UC))	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				1.17E-02
	Q_{tot}	-5.45E-01	4.94E-02	-6.98E-03	-1.78E-01	1.02E-02
	Q_R	-9.46E-01	4.93E-02	-1.18E-03	-1.15E+00	1.44E-02
	SHR***	4.19E-01	-2.12E-02	1.26E-03	1.39E-01	8.56E-03
	W_{ODU}	-3.13E-01	1.15E-02	2.66E-03	-1.16E-01	5.14E-03
	W_{tot}	-2.54E-01	1.12E-02	2.06E-03	5.74E-03	5.29E-03
Refrigerant overcharge (OC)	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				2.00E-02
	Q_{tot}	4.72E-02	-1.41E-02	7.93E-03	3.47E-01	1.96E-02
	Q_R	-1.63E-01	1.14E-02	-2.10E-04	-1.40E-01	5.67E-03
	SHR***	-7.75E-02	7.09E-03	-1.93E-04	-2.76E-01	7.34E-03
	W_{ODU}	2.19E-01	-5.01E-03	9.89E-04	2.84E-01	5.17E-03
	W_{tot}	1.46E-01	-4.56E-03	9.17E-04	3.37E-01	5.43E-03
Improper liquid line refrigerant subcooling (SC)	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				2.26E-02
	Q_{tot}	6.77E-02	0.00E+00	-1.22E-03	-1.91E-02	2.18E-02
	Q_R	4.16E-02	0.00E+00	-3.51E-04	-1.55E-02	1.39E-03
	SHR***	-9.04E-02	0.00E+00	2.13E-03	1.60E-02	3.06E-02
	W_{ODU}	2.11E-02	0.00E+00	-4.18E-04	4.25E-02	4.34E-03
	W_{tot}	1.06E-02	0.00E+00	-2.93E-04	3.88E-02	4.84E-03
Non-condensable gas (NC)	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				1.71E-02
	Q_{tot}	2.77E-01	-1.75E-02	1.78E-02	-1.96E+00	1.63E-02
	Q_R	-1.78E+00	4.04E-02	1.78E-02	9.98E-01	9.59E-03
	SHR***	-4.67E-01	1.69E-02	9.89E-04	2.90E-01	5.59E-03
	W_{ODU}	-6.92E-01	2.01E-02	1.20E-02	6.62E-01	6.13E-03
	W_{tot}	-5.37E-01	1.52E-02	1.09E-02	4.36E-01	6.20E-03
Improper line voltage (VOL)	COP	$Y_{COP} = Y_{Qtot}/ Y_{Wtot}$				1.98E-02
	Q_{tot}	5.84E-01	-1.21E-02	-8.57E-03	-3.35E-01	1.80E-02
	Q_R	1.03E-01	-6.10E-03	3.64E-03	-1.04E-01	6.41E-03
	SHR***	-6.65E-02	5.21E-03	-2.10E-03	4.23E-02	2.95E-02
	W_{ODU}	7.66E-01	-3.85E-03	-1.83E-02	1.14E+00	4.39E-03
	W_{tot}	9.06E-01	-6.37E-03	-1.75E-02	1.10E+00	7.39E-03
TXV undesizing, cooling (TXV)	Refer to Eqs. (3.6, 3.7) and Table US-7					

* All temperatures are in Celsius
** FSE (fit standard error) equals the square root of the sum of the squared errors divided by the degrees of freedom
*** The applicable range of SHR for wet coil predictions: 0.7 to 0.85

The following is an explanation of the procedure used to calculate the dimensionless capacity and COP due to undersizing of the cooling mode TXV. This fault occurs if the expansion valve's equivalent orifice area is too small to control refrigerant superheat during periods of low ambient temperature conditions at reduced

condenser pressures. A properly sized TXV will regulate refrigerant flow rate and maintain proper superheat over a wide range of indoor and outdoor air temperatures. However, if the indoor TXV is undersized for the particular outdoor unit, the system performance is degraded due to a restricted mass flow of refrigerant at certain evaporator and condenser pressure differentials. The rated TXV capacity and nominal system capacity are used to determine the TXV undersizing fault level. For example, if a 7.0 kW (2 ton) TXV is installed in a system with the nominal capacity of 8.8 kW (2.5 ton), the fault level is 20 %. (F = 1-7.0/8.8=0.20).

Since the pressure difference between upstream and downstream becomes smaller with decreasing outdoor temperature, the TXV opens to increase refrigerant mass flow rate at low outdoor temperatures. The outdoor temperature at which the TXV reaches its maximum orifice size, referred to as the "departure temperature", is determined from calculations and empirical fits to previous data. The resulting departure temperature below which the TXV cannot supply adequate mass flow rate is given by Eq. (3.2).

$$T^{dep}[°C]=80.326\cdot F+11.682 \tag{3.2}$$

Table US-6. Example uncertainty propagation due to normalized correlation (Y) uncertainty of 3 % for faulty COP and cooling capacity at AHRI Standards 210/240 B-test condition (AHRI, 2008)

Fault	Parameter	Parameter Value	Uncertainty (%) (95 % confidence level)
10 % reduced indoor airflow	COP	3.67	± 6.4
	Cooling capacity	9.4 kW	± 5.0

The cooling capacity and the gross COP of the undersized TXV-equipped system can be expressed as functions of outdoor temperature and fault level. To develop equations for the normalized capacity and COP, non-dimensional variables for outdoor temperature, cooling capacity, and gross COP are defined by Eqs. (3.3, 3.4, 3.5), respectively, where T_{OD} has Celsius units.

$$T_r=\frac{T_{OD}}{35\ °C} \tag{3.3}$$

$$Y_Q=\frac{Q_{undersized}}{Q_{no-fault}} \tag{3.4}$$

$$Y_{COP}=\frac{COP_{undersized}}{COP_{no\text{-}fault}} \tag{3.5}$$

The correlations for determining normalized cooling capacity and normalized gross COP are given by Eqs. (3.6) and (3.7) and are presented in a graphical form in Figure US-2. The coefficients are listed in Table US-7.

$$Y_Q=a_1+a_2T_r+a_3F+a_4T_r^2+a_5T_rF+a_6F^2 \text{ if } T_{OD}\le T^{dep} \text{ or } Y_Q=1 \text{ if } T_{OD}>T^{dep} \tag{3.6}$$

$$Y_{COP}=b_1+b_2T_r+b_3F+b_4T_r^2+b_5T_rF+b_6F^2 \text{ if } T_{OD}\le T^{dep} \text{ or } Y_Q=1 \text{ if } T_{OD}>T^{dep} \tag{3.7}$$

Table US-7. Normalized capacity and COP correlation coefficients for a TXV undersizing fault

Coefficients for Y_Q		Coefficients for Y_{COP}	
a_1	9.1440E-01	b_1	8.4978E-01
a_2	2.0903E-01	b_2	4.0050 E-01
a_3	-5.4122E-01	b_3	-8.4120E-01
a_4	1.2194E-01	b_4	7.5740E-02
a_5	-2.9428E-01	b_5	-3.3105E-01
a_6	-3.0833E-02	b_6	2.0290E-01

A complete and detailed discussion of the TXV undersizing fault correlation development is beyond the scope of this report and is presented by Payne and Kwon (2014).

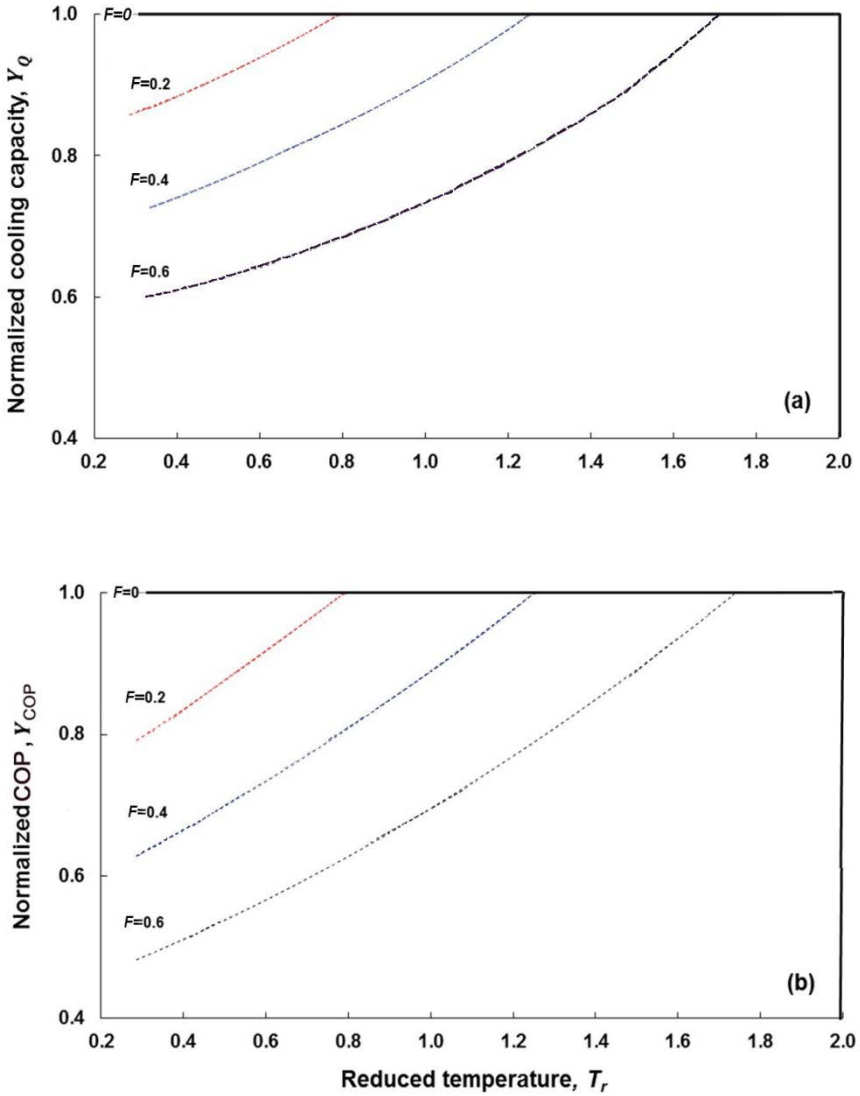


Figure US-2. Normalized performance parameters for the cooling mode TXV undersizing fault: (a) capacity, (b) COP

3.2.2. Cooling Mode Charts with Normalized Performance Parameters

Figures US-3 through US-8 show variations of the normalized performance parameters with respect to fault levels at five operating conditions. The figures present the measured data points and correlations developed for COP, capacity, SHR, total power and, for some faults, the outdoor unit power. The outdoor unit power is included for improper indoor airflow (AF) and improper liquid line refrigerant subcooling (SC) faults where the trends of the total power and the outdoor unit power were not similar. In some of the figures there is a significant difference between the correlation fits and the actual data points. The correlations were developed for all indoor and outdoor test conditions, and thus the fit sum of squared deviations was minimized. In addition, the normalized value for the heat pump operating with no fault was calculated from the fault-free correlation as presented by Kim et al. (2010); therefore, no-fault tests may actually have normalized values somewhat different from unity due to the inability of the no-fault correlation to predict the no-fault parameter exactly. Scatter of normalized no-fault data around unity indicates measurement uncertainty, correlation uncertainty, and uncertainty caused by different system installations. The data for Figures US-6 and US-8 were collected after the system was removed and re-installed in the test chambers; therefore, one would expect more scatter in the normalized no-fault correlations due to this installation repeatability uncertainty. This installation repeatability uncertainty is also indicative of what could be seen in field installations when applying the same no-fault correlations from system to system.

Figure US-3 shows the normalized parameters at a reduced and increased indoor airflow. For the studied airflow range from -50 % to +20 % of the nominal value, the change in outdoor unit power ranged from -3 % to 0 %, respectively, with small variations between different operating conditions. Total power varied from -5 % to 2 % within the same range of airflow rate, which indicates the varied power of the indoor fan at this fault. COP and capacity were markedly degraded at a decreased airflow and somewhat improved at the increased airflow above the nominal level; however, these increases in COP and capacity were associated with a significant increase in SHR, which may not be a desirable change from the homeowner's comfort point of view. The difference between total power and outdoor unit power is due to the power of the indoor blower, which was nominally 430 W. Outdoor unit power was relatively constant under this fault. As a result, COP slightly increased at the max fault level by the increased indoor airflow.

Figures US-4 and US-5 show the variation of the normalized values for refrigerant charge faults. The changes in COP and total capacity for refrigerant undercharge are larger than those for refrigerant overcharge. A 30 % undercharge reduced capacity by almost 15 % on average reducing COP by 12 % while a 30 % overcharge produced little reductions or small increases in capacity with 6 % greater total power and 3 % reduced COP on average because of the increased discharge pressure. In case of different outdoor temperature conditions, COP and capacity increased as the outdoor temperature increased for the undercharged condition. Farzad et al. (1990) also showed that higher refrigerant flow rate is one reason for the higher capacity at higher outdoor temperatures for the conditions of undercharge.

In this study, a subcooling temperature of 4.4 ^{o}C (8.0 ^{o}F) was regarded as the no-fault condition under the considered test conditions. Figure US-6 shows the effects of increased subcooling at the TXV inlet. The departure of the normalized values of COP and cooling capacity from the correlations in the figure are mostly due to the TXV attempting to correct mass flow rate (reduce effective orifice size) as subcooling increases. If more data were available with subcooling being varied randomly from high to low values, hysteresis effects and TXV hunting effects would be better captured. COP and capacity normalized correlations for higher levels of subcooling still represent the general trends in system performance. Increased subcooling is a symptom of excessive refrigerant charge, and it has the same effect; higher subcooling leads to reduced condensing area and increased condensing pressure. In the studied heat pump, refrigerant overcharging by 30 % corresponded to approximately doubling of refrigerant subcooling. For this level of fault the COP degradation was within 4 %. For the highest subcooling fault of 181 % of the nominal value, the impact on the capacity was minor but the outdoor unit power increased by 15 %, which resulted in a similar decrease in the COP.

Figure US-7 shows the variation of the normalized values for chosen performance parameters versus non-condensable gas (NC) fault level. Non-condensable gases increase the condensing pressure above that corresponding to the saturation pressure of the refrigerant at the same temperature due to the partial pressure of the NC components. As a result, increased total power consumption and decreased COP can be seen in the Figure US-7. Maximum degradation of COP at the 20 % fault level was about 5 % for the condition of T_{ID}=26.7 ^{o}C (80.0 ^{o}F) and T_{OD}=27.8 ^{o}C (82.0 ^{o}F).

Figure US-8 shows the variation of the normalized values for chosen performance parameters for the line voltage variation fault conditions. A line voltage of 208 V was set as the no-fault condition. Total external static pressure (ESP) for the indoor air handler was set at 125 Pa (0.5 in H_2O) at the no-fault line voltage, which produced a nominal indoor fan power demand of 430 W. As voltage increased, fan speed and static pressure increased thus producing increased fan power. Total power consumption increased almost linearly as the fault level increased. The fan power increased more than the compressor power when the voltage was increased. An average increase of 27 % for the fan power and 9 % for the compressor power occurred at the max fault level. At fault levels over 20%, the degradation of COP is greater than 10 %.

The presented measurements for the cooling mode indicate that the refrigerant undercharge fault has the highest potential for degrading air conditioner efficiency. For 30 % percent undercharge – a fault level commonly observed during field surveys – the system efficiency is decreased between 7 % and 15 %, depending on operating conditions.

A reduction of the airflow rate by 30 % (also a commonly observed fault) can reduce the efficiency by 6 %, and this level of degradation persists independently of operating conditions. Refrigerant overcharging by

30 % resulted in COP degradation on the order of 4 %. COP degradation within 3 % was measured for improper electric voltage and non-condensable gas faults. The non-condensable gas fault can be misdiagnosed in the field as refrigerant overcharge, which may prompt a serviceman to remove some of the refrigerant from the system, thus triggering an undercharge fault.

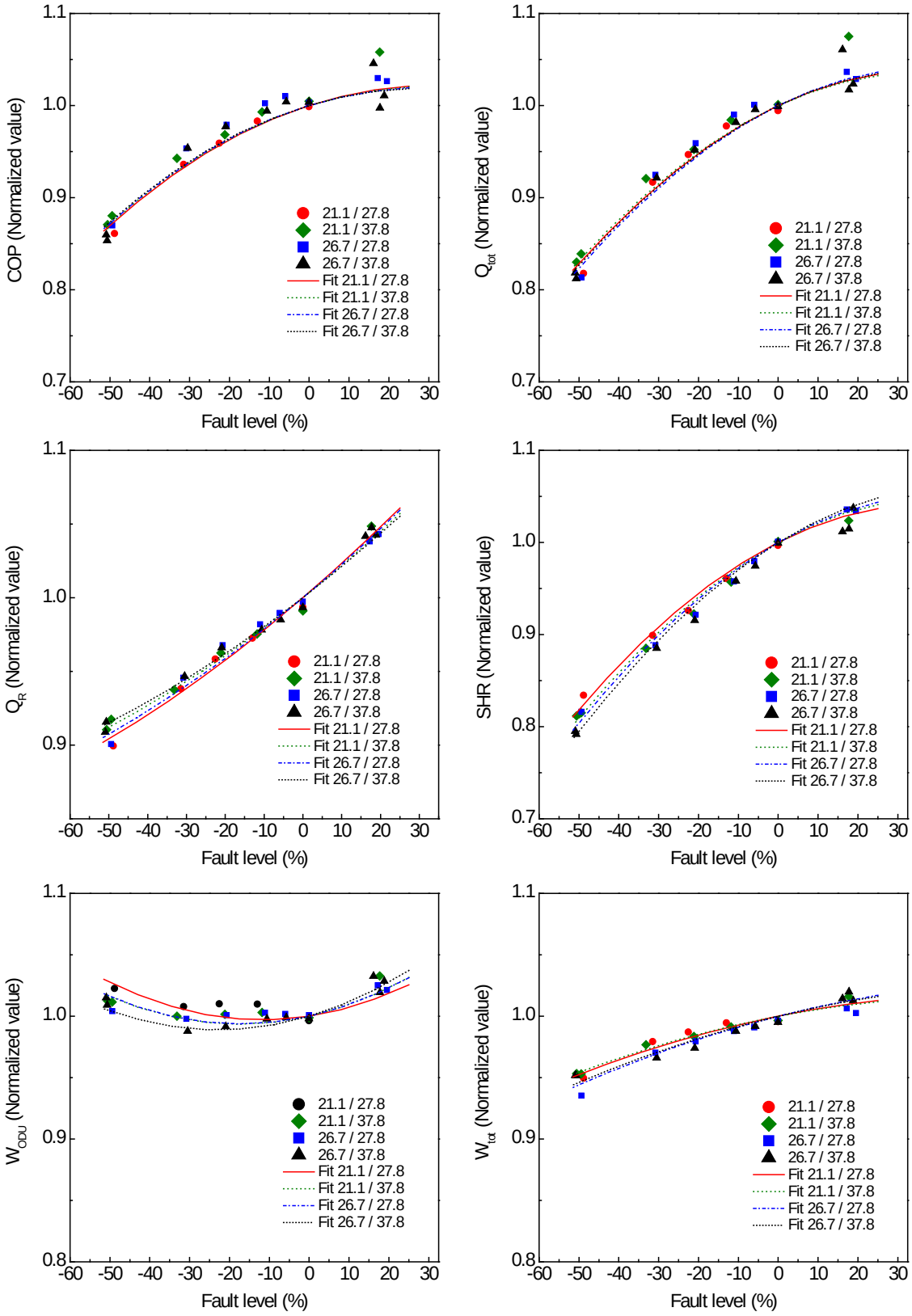


Figure US-3. Normalized cooling performance parameters for improper indoor airflow (The numbers in the legend denote test conditions, T_{ID} (^oC)/ T_{OD} (^oC))

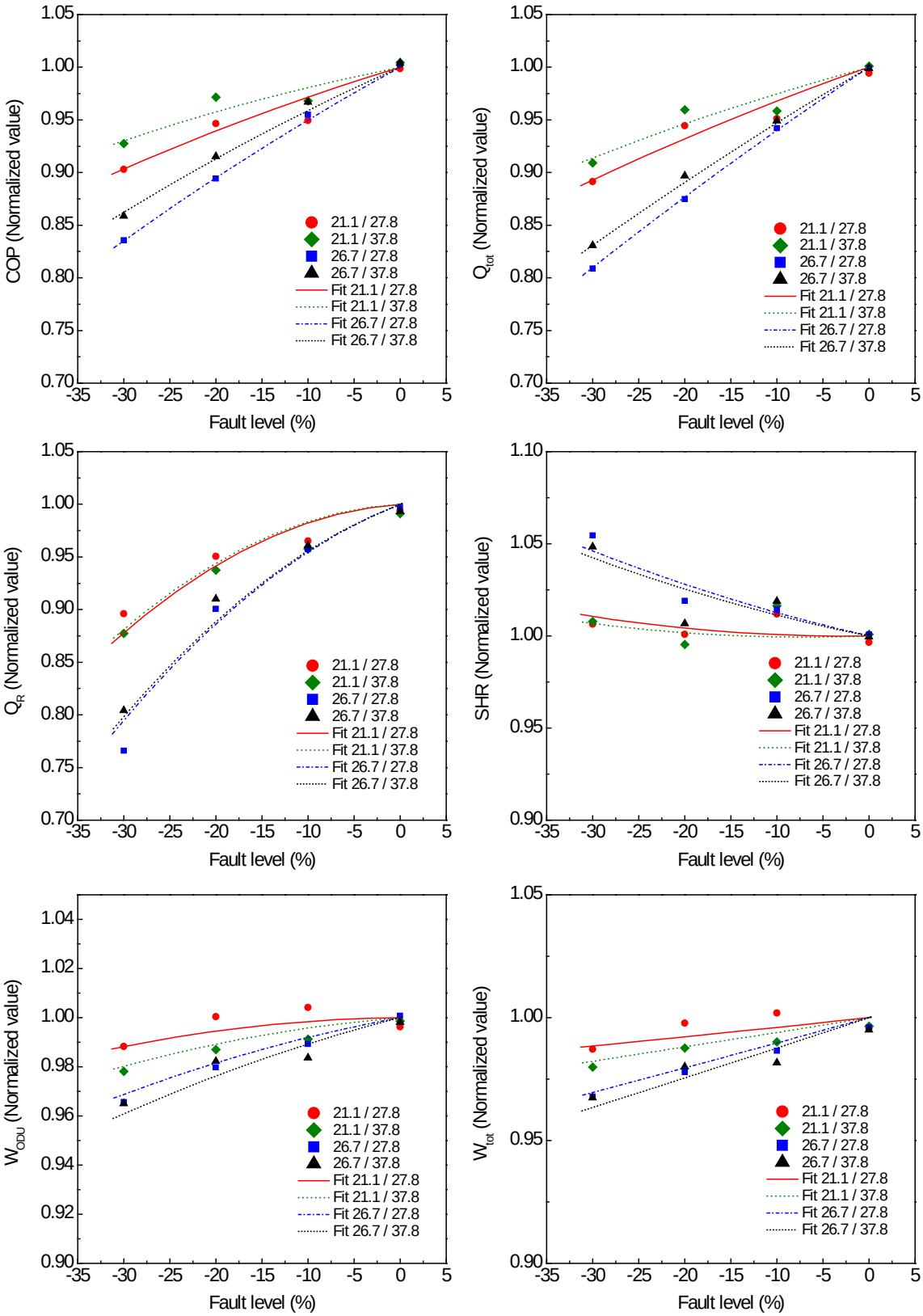


Figure US-4. Normalized cooling performance parameters for refrigerant undercharge (The numbers in the legend denote test conditions, T_{ID} (°C)/ T_{OD} (°C))

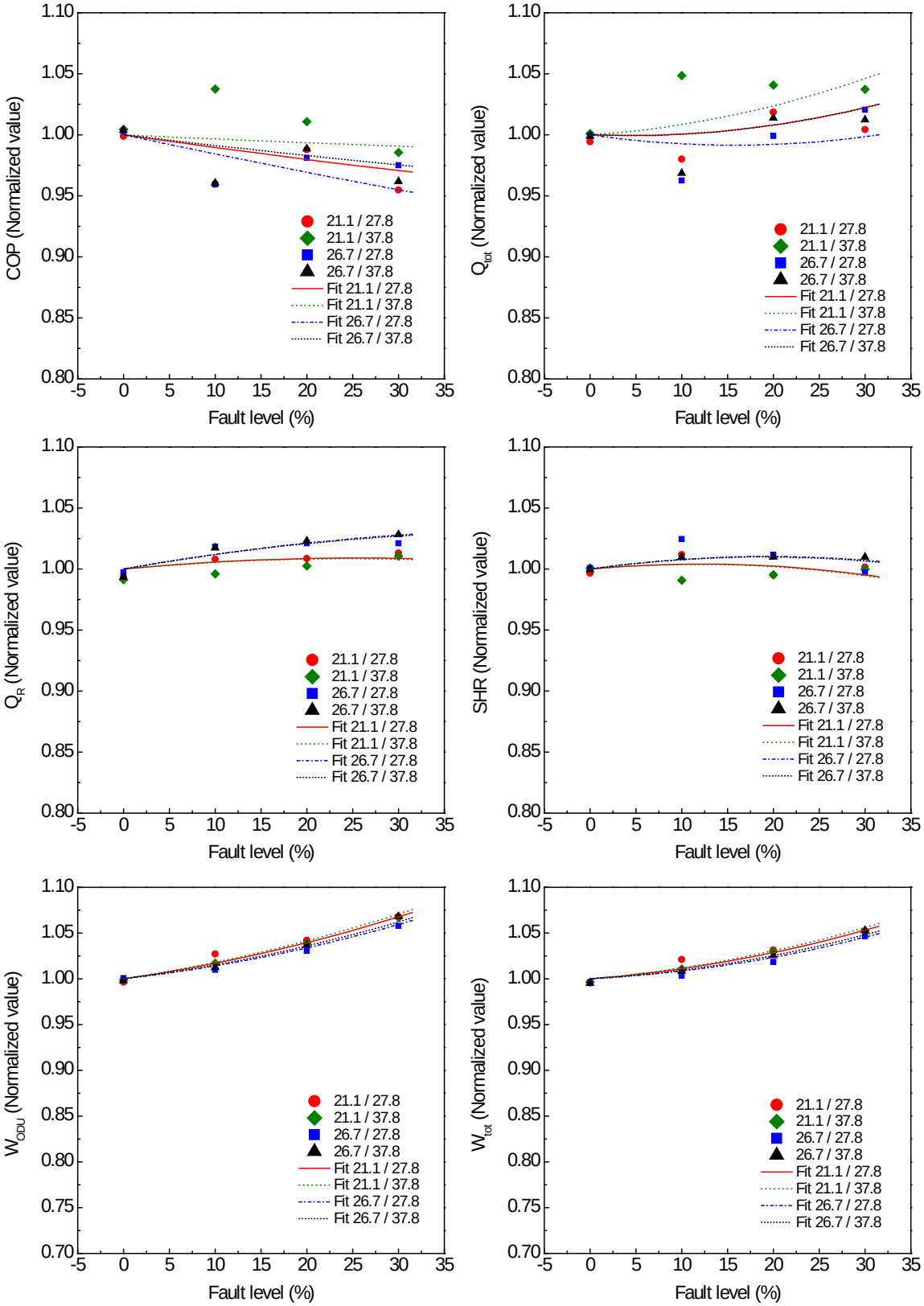


Figure US-5. Normalized cooling performance parameters for refrigerant overcharge (The numbers in the legend denote test conditions, T_{ID} (°C)/ T_{OD} (°C))

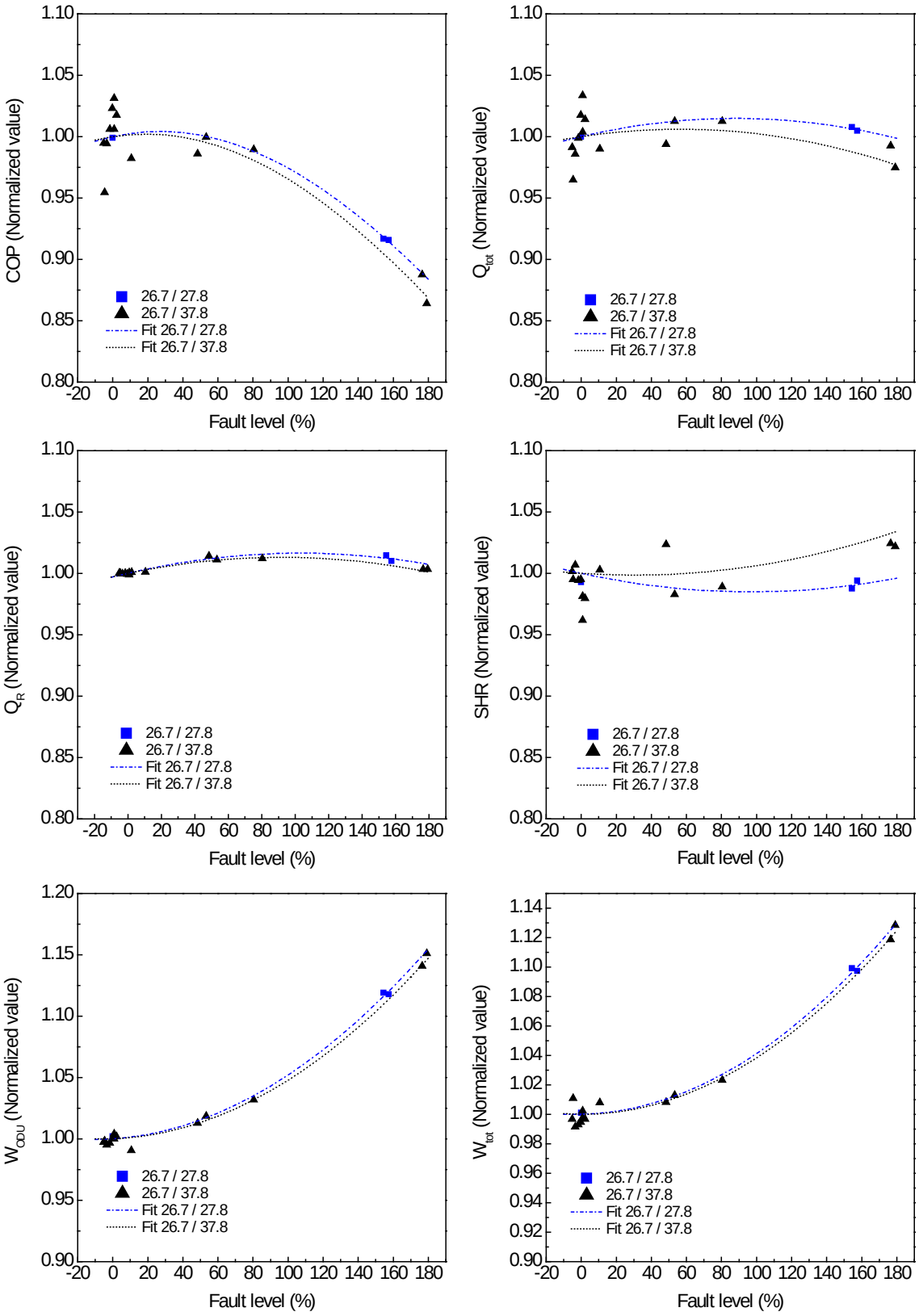


Figure US-6. Normalized cooling performance parameters for improper liquid line refrigerant subcooling (The numbers in the legend denote test conditions, T_{ID} (°C)/ T_{OD} (°C))

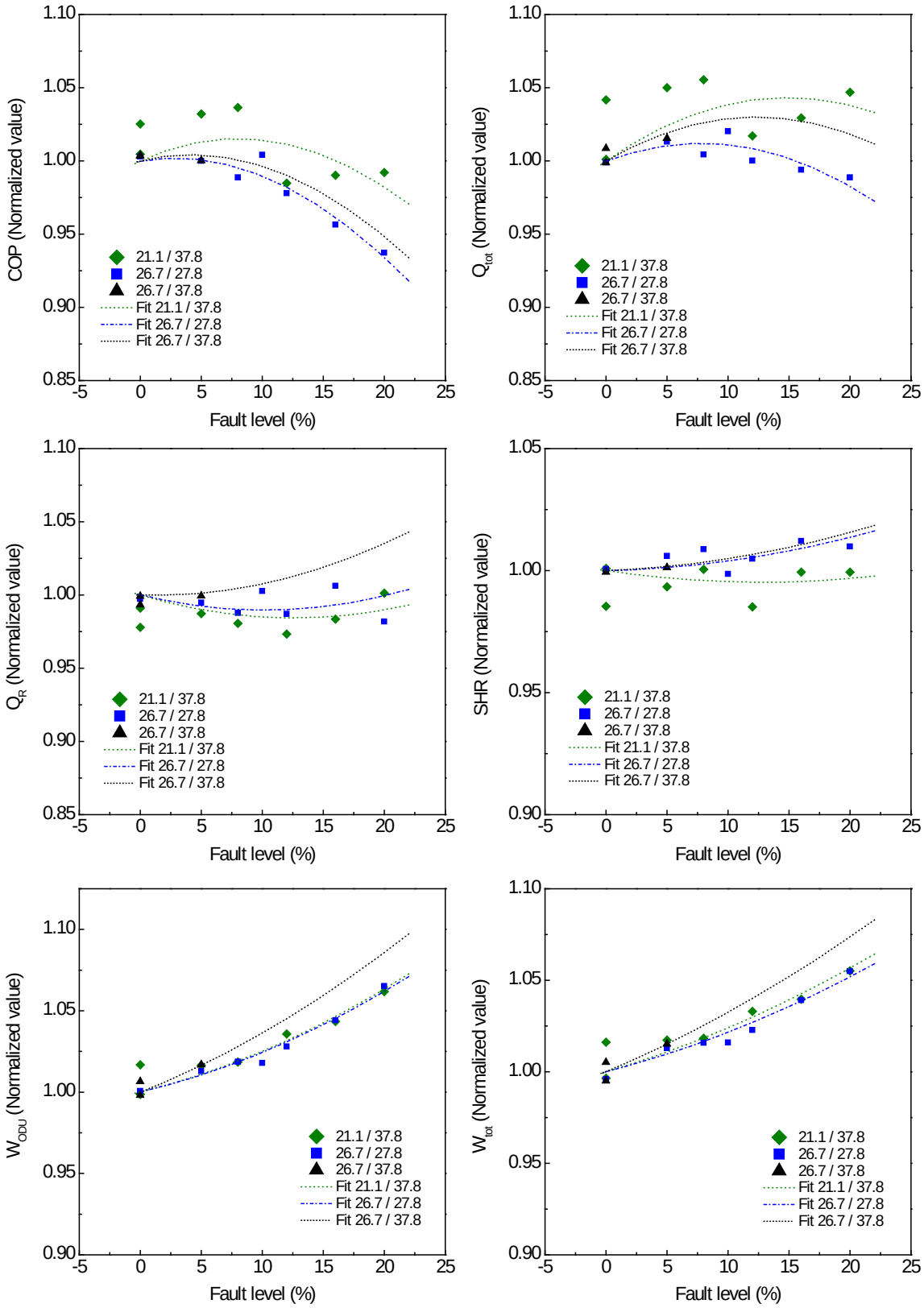


Figure US-7. Normalized cooling performance parameters for the presence of non-condensable gas (The numbers in the legend denote test conditions, T_{ID} (°C)/ T_{OD} (°C))

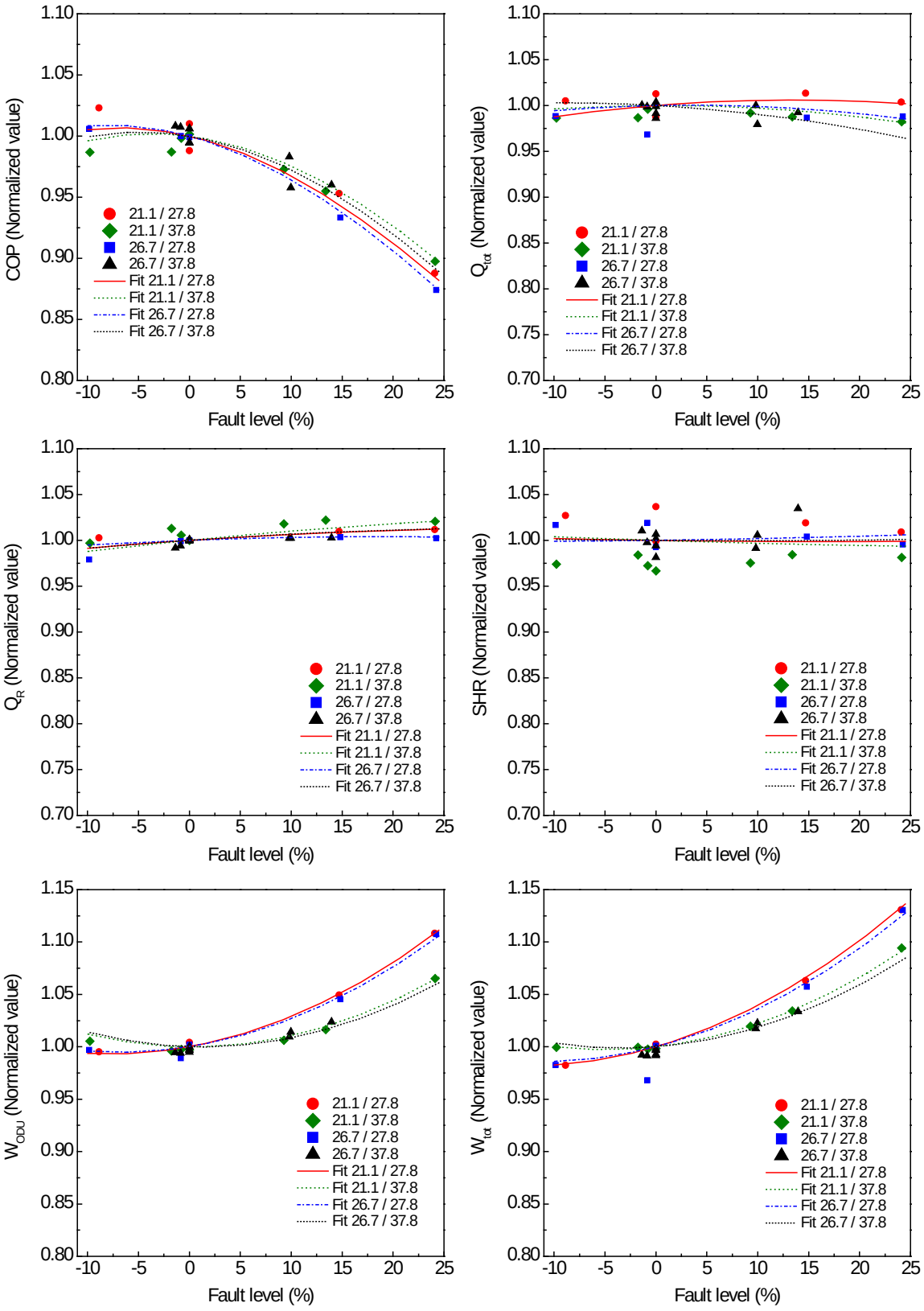


Figure US-8. Normalized cooling performance parameters for improper electric line voltage (The numbers in the legend denote test conditions, T_{ID} (°C)/ T_{OD} (°C))

3.3. Fault Effects on Heating Mode Performance

3.3.1. Heating Mode Normalized Performance Parameters and Correlation

The heating mode tests considered the effect of faults on five performance parameters: coefficient of performance (COP), total heating capacity ($Q_{tot,}$ includes the indoor fan heat), refrigerant-side heating capacity ($Q_{R,}$ does not include the indoor fan heat), outdoor unit power ($W_{ODU,}$ includes the compressor, outdoor fan, and controls powers), and total power (W_{tot}, includes W_{ODU} and indoor fan power). These parameters are presented in a dimensionless, normalized format obtained by dividing these parameter values as obtained for the heat pump operating under a selected fault by the no-fault value. The normalized parameters were correlated as a function of outdoor dry-bulb temperature (T_{OD}) and fault level (F). These two parameters were the only values varied for the heating mode tests; indoor dry-bulb temperature did not vary enough to use in the heating mode correlations.

$$Y=\frac{X_{fault}}{X_{no\text{-}fault}}=1+(a_1+a_2T_{OD}+a_3F)\cdot F \tag{3.8}$$

where a_1, a_2, and a_3 are correlation coefficients, X_{fault} and $X_{no\text{-}fault}$ are performance parameters for a faulty and fault-free heat pump, and Y is a dimensionless parameter representing the ratio of the faulty performance from that of the fault-free heat pump.

Table US-8 shows the correlation coefficients. They were determined by means of a multivariate, polynomial regression method using the normalized values of performance parameters determined from heat pump test data. If the heat pump is fault free, values of all normalized parameters equal unity.

Table US-8. Correlations for non-dimensional performance parameters in the heating mode

Fault	Performance Parameter Y	$Y=1+(a_1+a_2T_{OD}+a_3F)\cdot F^*$			FSE**
		a_1	a_2	a_3	
Improper indoor airflow rate (AF)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			3.27E-02
	Q_{tot}	0.1545961	0.0078768	-0.1746421	2.72E-02
	Q_R	0.0009404	0.0065171	-0.3464391	1.82E-02
	W_{ODU}	-0.177359	-0.0125111	0.4784914	1.21E-02
	W_{tot}	0.0311053	-0.009332	0.7942998	2.87E-02
Refrigerant undercharge (UC)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			2.68E-02
	Q_{tot}	-0.104922	0.0156348	-1.3702726	8.02E-03
	Q_R	-0.0338595	0.0202827	-2.6226343	2.55E-02
	W_{ODU}	0.0615649	0.0044554	-0.2598507	8.79E-03
	W_{tot}	0.0537015	0.004334	-0.2272758	7.85E-03
Refrigerant overcharge (OC)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			6.08E-03
	Q_{tot}	-0.1198701	-0.0004505	0.5052803	5.20E-03
	Q_R	-0.0029514	0.0007379	-0.0064112	3.14E-03
	W_{ODU}	-0.0594134	0.0159205	1.8872153	9.19E-03
	W_{tot}	-0.053594	0.0140041	1.6948771	8.43E-03
Improper liquid line refrigerant subcooling (SC)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			1.77E-02
	Q_{tot}	-0.0369891	0.0014081	0.0113751	1.06E-02
	Q_R	-0.0389621	0.0019259	0.0079344	1.41E-02
	W_{ODU}	0.1353483	-0.001264	0.008241	8.45E-03
	W_{tot}	0.1023326	-0.0007392	0.0128456	6.11E-03
Noncondensable gas (NC)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			9.69E-03
	Q_{tot}	0.0852956	0.0058473	-0.9522349	9.37E-03
	Q_R	-0.2081656	0.0058006	0.6035798	2.48E-03
	W_{ODU}	0.181571	0.0008425	0.6093669	3.95E-03
	W_{tot}	0.1840392	-0.0001309	0.3935121	3.92E-03
Improper line voltage (VOL)	COP	$Y_{COP}=Y_{Qtot}/Y_{Wtot}$			1.13E-02
	Q_{tot}	0.1107829	-0.0040167	-0.1347848	9.87E-03
	Q_R	0.0912687	-0.0006155	-0.2343559	5.60E-03
	W_{ODU}	0.1604092	0.0011052	0.9262117	1.80E-03
	W_{tot}	0.283868	0.0009125	0.7759193	3.61E-03

* All temperatures are in Celsius

** FSE (fit standard error) equals the square root of the sum of the squared errors divided by the degrees of freedom

3.3.2. Heating Mode Charts with Normalized Performance Parameters

Figure US-9 shows the effects of reduced airflow over the indoor coil during heating mode operations. The airflow rate through the indoor heat exchanger was controlled by changing the speed of the nozzle chamber booster fan. As shown in the graphs, effects of this fault condition for COP and power are noticeable. Especially, for the higher outdoor temperature condition (T_{OD}=8.3 °C (47 °F)) with a 50 % reduced airflow rate, COP was degraded by over 30 % and total power increased by more than 20 %.

Figure US-10 shows the effects of refrigerant undercharge. At the maximum fault level of 30 %, COP decreased by more than 8 % for the higher outdoor temperature condition (8.3 °C (47 °F)). The decrease was greater for the lower temperature lift case due to the lower pressure ratio and resulting lower mass flow rate potential (pressure drop) across the expansion valve as compared to the -8.3 °C (17 °F) case. Mass flow rate is proportional to the square root of the pressure drop. Therefore, the reduction in mass flow rate due to removing refrigerant and lowering liquid line subcooling (lowering liquid line pressure) will have a greater effect upon mass flow rate at higher condenser pressure (higher outdoor temperatures). Capacity reduction had a greater effect upon COP than compressor power demand due to undercharge; refrigerant-side capacity decreased by an average of 22 % while outdoor unit power demand decreased an average of only 5 % for this maximum fault level and 8.3 °C (47 °F) test condition.

Figure US-11 shows the effects of refrigerant overcharge. The control effect of the TXV is seen in the refrigerant-side capacity; capacity remains nearly constant (±1 %) while compressor power demand increases to approximately 15 % at 30 % fault level. The TXV maintains outdoor coil exit superheat by increasing pressure drop and limiting mass flow. Compressor power demand increases, being more pronounced at the lower temperature lift (lower pressure ratio), highest outdoor temperature. At the lower pressure ratio case, system capacity and refrigerant mass flow are already greater than the higher pressure ratio case, and the addition of refrigerant produces a greater change in power demand for a given fault level.

Refrigerant overcharge demonstrates itself in increased refrigerant subcooling in the liquid line. When subcooling was doubled from its nominal value (a fault level of 100 %), compressor power demand increased by approximately 15 % with little change in capacity (Figure US-12). This resulted in an almost 12 % decrease in COP. Increased subcooling (increased refrigerant charge) affects compressor power demand more than capacity due to the TXV control of evaporator exit superheat.

Figure US-13 shows the effects of non-condensable gas. The non-condensable gas will accumulate in the condenser (indoor coil) and thus reduce the heat transfer area available, and raise the condenser pressure in direct proportion to the volume of the non-condensable gas. At the highest fault level of approximately 20 %, the COP decreases by approximately 8 % at the lowest outdoor test temperature. The non-condensable gas appears to have equal effect upon compressor power demand at all fault levels and outdoor temperatures, while capacity is more affected at the higher pressure ratio produced at the lowest outdoor temperature.

Figure US-14 shows the effects of varying the system working voltage above and below the nominal value of 208 VAC. The changes in compressor power demand are a result of increased evaporator refrigerant saturation temperature at the higher indoor airflow rates. Changing the supply voltage changes all of the electric motors' rotational speeds; therefore, lowering the voltage is equivalent to reducing compressor pumping capacity while leaving heat transfer area constant. At higher voltages, the higher compressor speed produces more of an effect on power demand than the indoor airflow rate produces on capacity. The TXV regulates refrigerant flow to maintain superheat at the higher indoor airflow rates. Capacity increases less than 2 % at the highest voltage while compressor power demand increases by more than 10 % resulting in an almost 10 % decrease in COP.

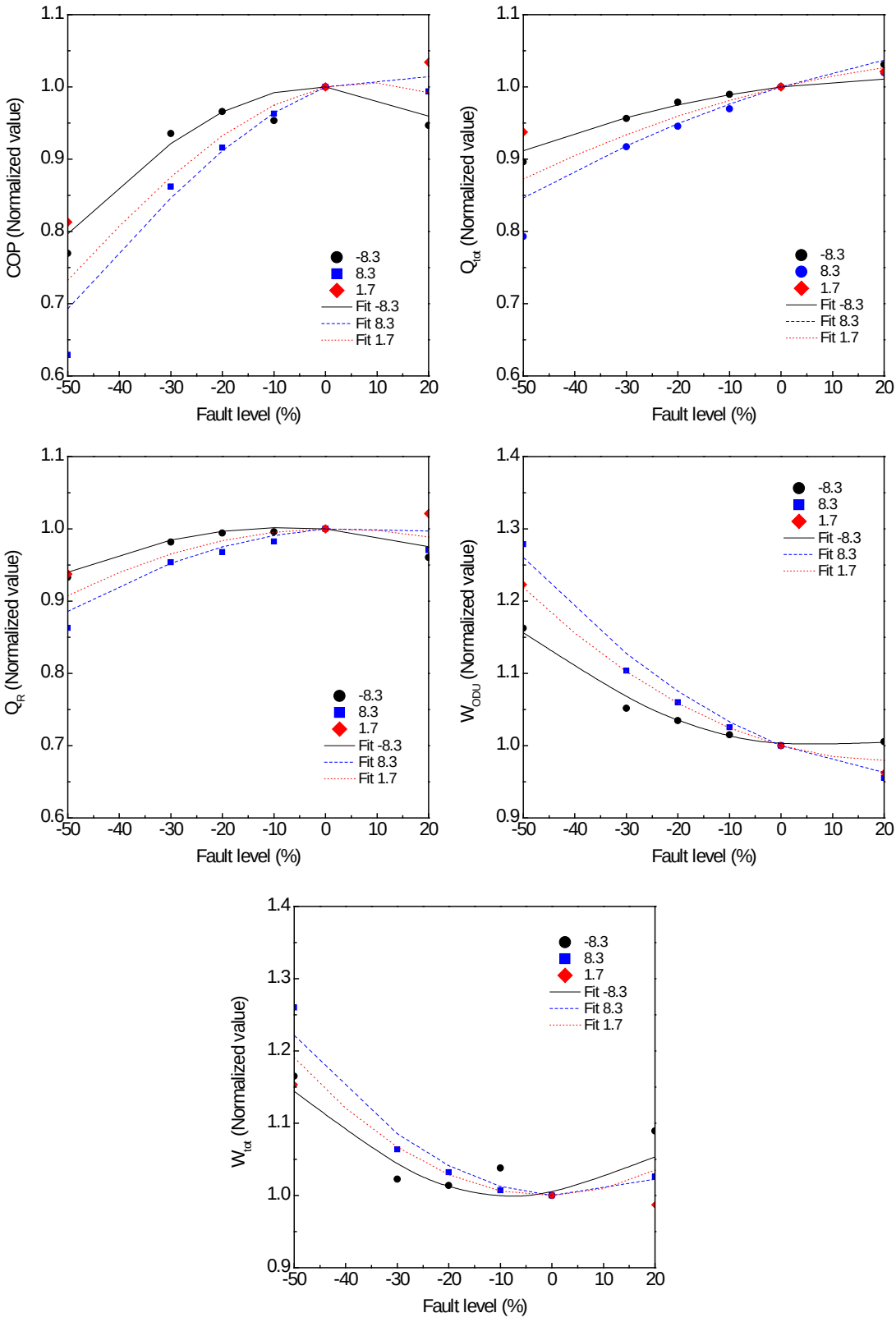


Figure US-9. Normalized heating performance parameters for improper indoor airflow (The number in the legend denotes T_{OD} (^oC))

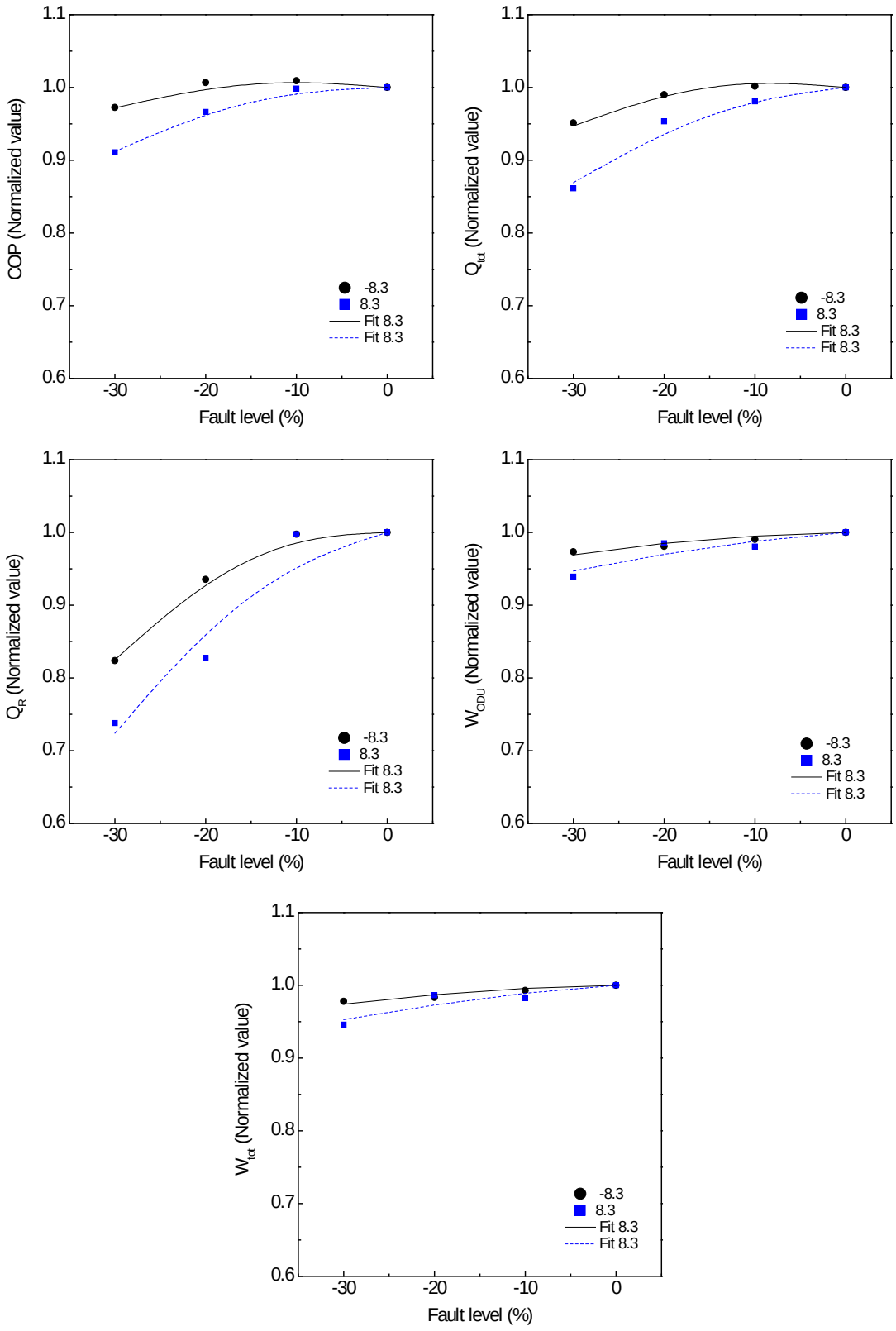


Figure US-10. Normalized heating performance parameters for refrigerant undercharge (The number in the legend denotes T_{OD} (^oC))

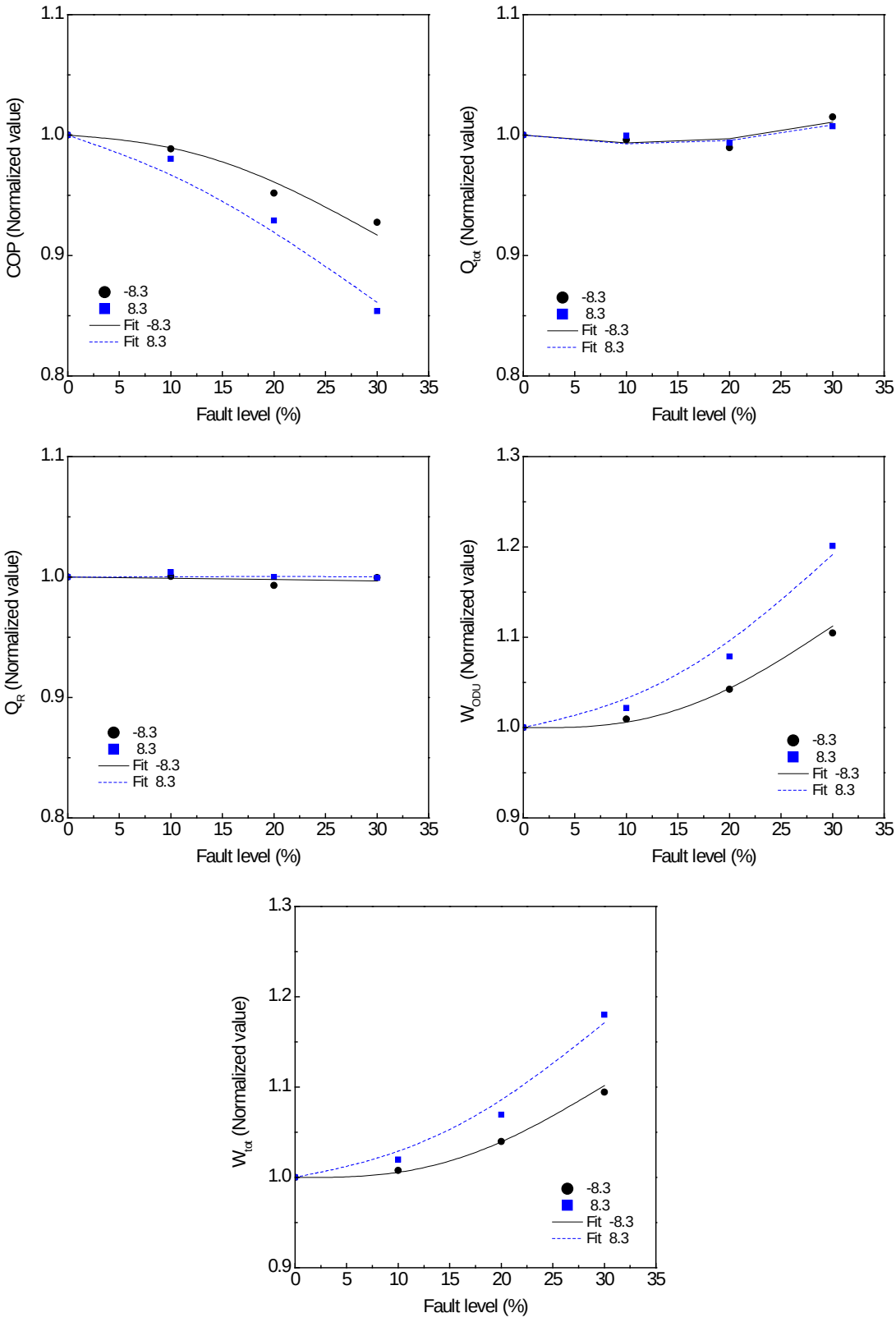


Figure US-11. Normalized heating performance parameters for refrigerant overcharge (The number in the legend denotes T_{OD} (^oC))

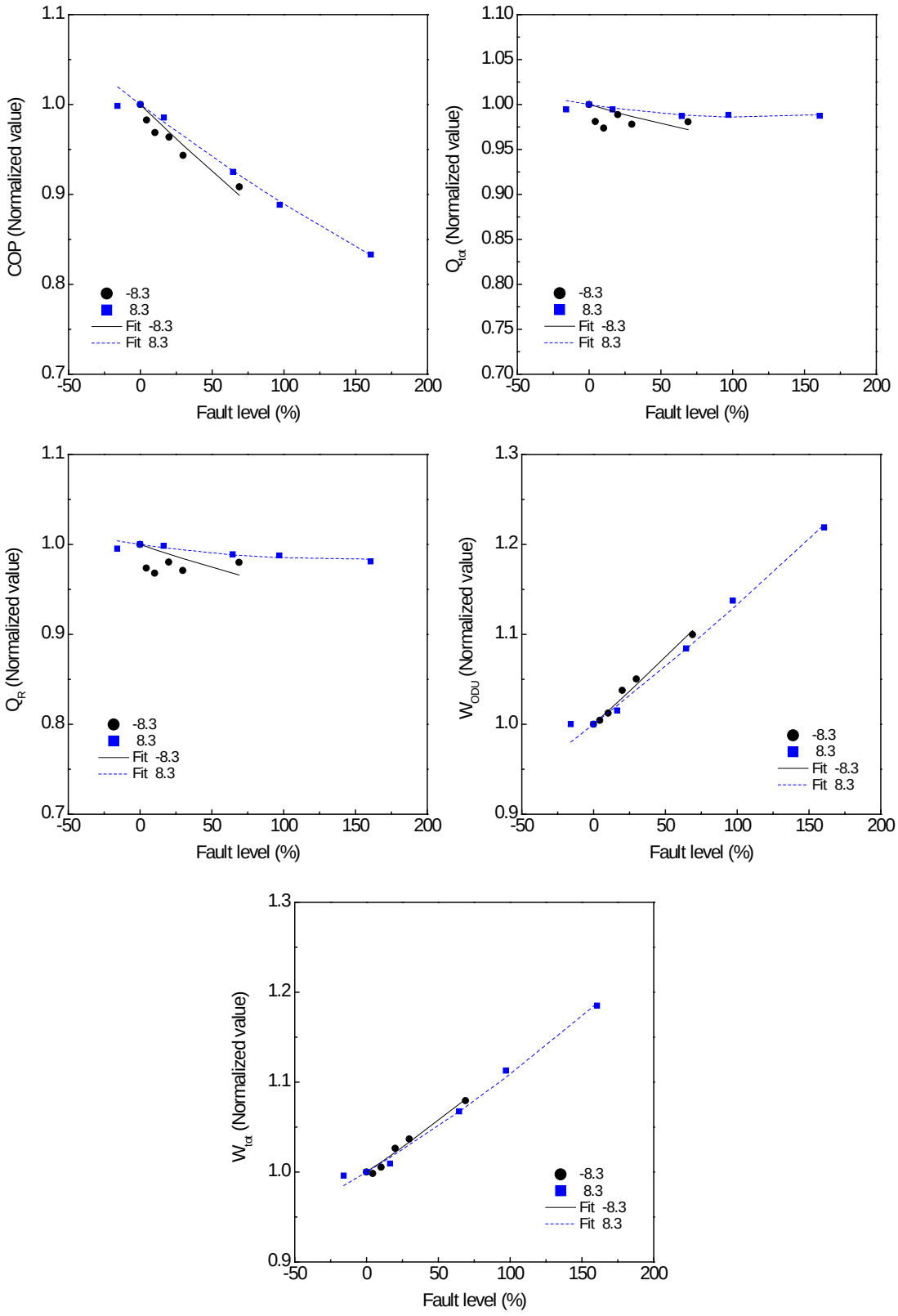


Figure US-12. Normalized heating performance parameters for improper refrigerant subcooling (The number in the legend denotes T_{OD} (^oC))

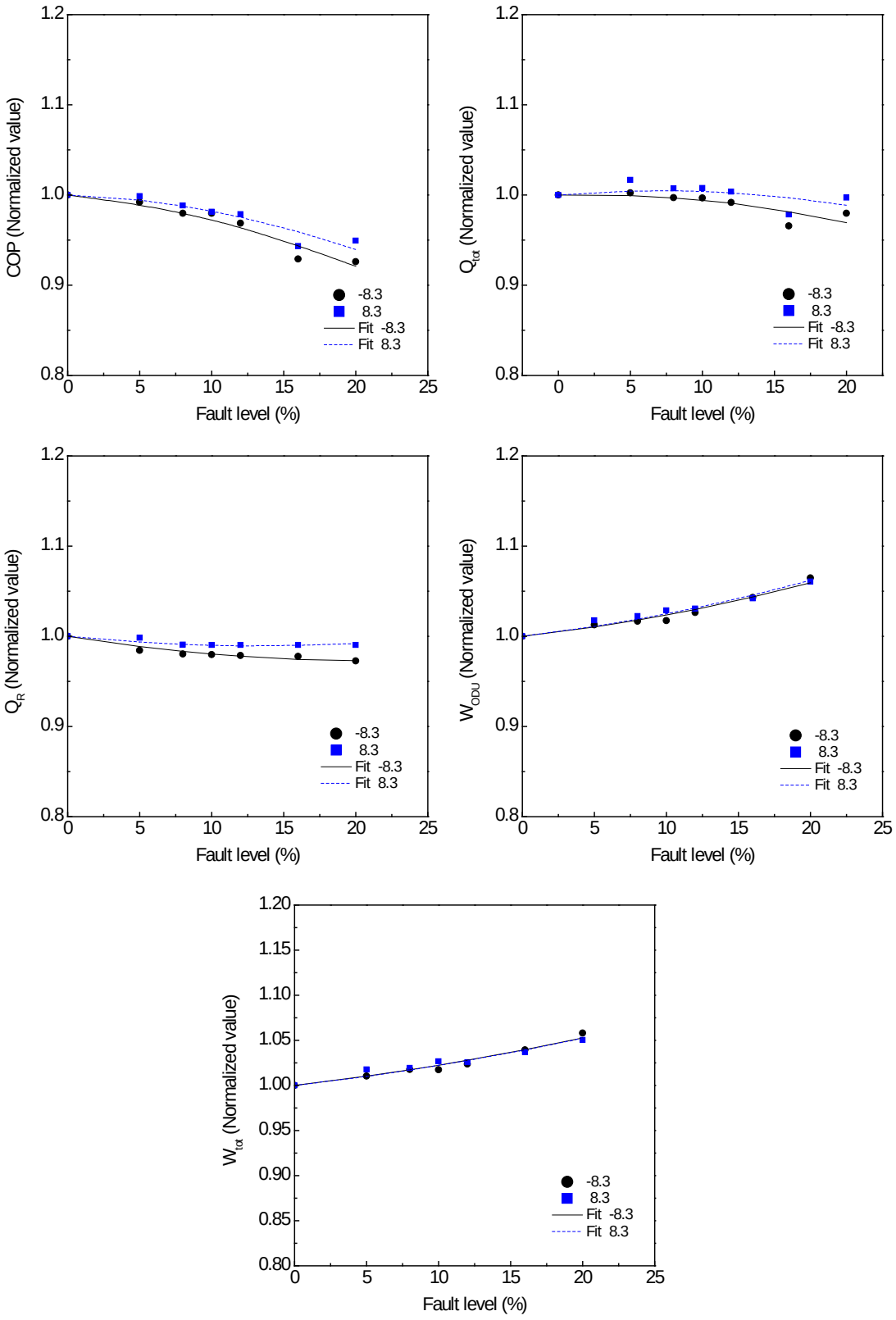


Figure US-13. Normalized heating performance parameters for the presence of non-condensable gas (The number in the legend denotes T_{OD} (^{o}C))

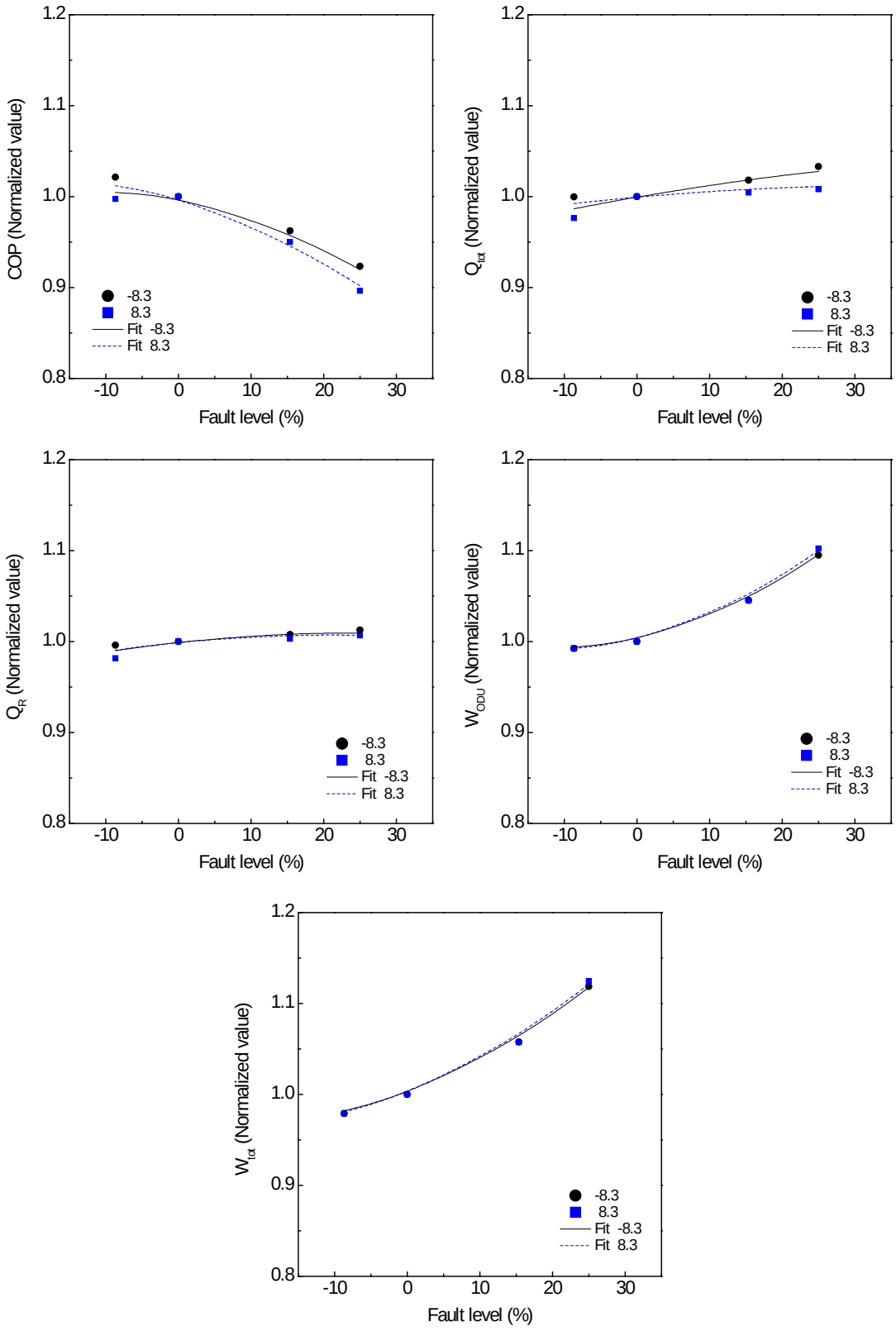


Figure US-14. Normalized heating performance parameters for improper line voltage (The number in the legend denotes T_{OD} (^oC))

4. BUILDING/HEAT PUMP MODELING APPROACH

4.1. Building/Heat Pump Systems Simulation Models

Several building simulation models are available for modeling residential buildings. Many include well-developed user interfaces aimed at specific audiences – such as residential home energy raters in the United States who seek to determine the Home Energy Rating System score (HERS) (RESNET, 2006). Table US-9 summarizes the features of these mainstream software tools. Energy Gauge USA, RemRate, and TREAT all have hundreds of users and are widely known in the residential energy efficiency community. However, while these tools include models for commonly-used systems and equipment operating at or near their nominal performance ratings, they do not have the flexibility to consider degraded, abnormal, or off-design performance.

Table US-9. Comparison of residential building simulation software tools

Energy Gauge USA	Fully developed hour-by-hour building simulation model (based on DOE-2.1e). Tool is commonly used by energy raters to develop a Home Energy Rating System (HERS) score. www.energygauge.com Advantages: good well-documented, building model with sound equipment components Disadvantages: no flexibility to add extra correlations or components
RemRATE	Building simulation model (using temperature bin calculations). Tool is commonly used by energy raters to develop a Home Energy Rating System (HERS) score. www.archenergy.com/products/remrate Advantages: good well-documented, building model with models for common equipment components Disadvantages: no flexibility to add extra correlations or components
TREAT	Hourly building simulation model aimed at residential energy analysis, for both single-family and multi-family homes. Used widely in the multi-family energy efficiency sector. www.psdconsulting.com/software/treat Advantages: robust, well-documented, building model focused on multi-family issues Disadvantages: no flexibility to consider alternate technologies

DOE-2 is the original, U.S. federally-funded building simulation model or calculation engine developed in the 1970s that is still used as the basis of some of the mainstream residential software tools (i.e., Energy Gauge USA). The DOE-2 software offers some flexibility but is no longer maintained or supported.

EnergyPlus is a state-of-the-art, very flexible, building simulation tool used for research evaluations and mainstream energy analysis and design assistance. Its development is supported by the U.S. Department of Energy (DOE). This detailed calculation engine works at sub-hourly time steps and can consider both residential and commercial buildings.

TRNSYS is a highly flexible, transient simulation tool that focuses on thermal systems, primarily aimed at building and HVAC applications (Klein et al., 2007). TRNSYS was originally developed at the University of Wisconsin to simulate the transient performance of solar thermal systems (http://sel.me.wisc.edu/trnsys/). TRNSYS is a modular tool where multiple components can be combined to build up a complex thermal system. TRNSYS includes several components necessary to simulate the transient performance of a building, including building envelope components, HVAC equipment, and utilities to read hourly weather data from TMY files. Because of its flexibility, this tool is uniquely able to consider new concepts and technologies – such as the research evaluation of this project. The core of the TRNSYS simulation model is the building envelope model based on the Type 56 multi-zone building model. The inputs to Type 56 are defined using the TRNBuild software tool (see Figure US-15) and then saved in a .BUI file. Type 56 then reads this file at runtime to generate the detailed building description. The building model includes all the basic characteristics of a residential building:

- Heat loss and gains through building walls, roof and floor
- Solar gains through windows
- Interactions between multiple zones (house, attic, rooms)
- Scheduled internal sensible and moisture loads for people, equipment, etc.

- Interactions with the heating, ventilation, and air conditioning equipment
- Scheduled set points for temperature and humidity

Table US-10 summarizes the advantages and disadvantages for each of these software tools. Because of its flexibility, we selected the building model developed in TRNSYS to study the integrated performance of a heat pump in residential application.

Table US-10. Comparison of general building calculation models

DOE-2	An hour-by-hour building simulation model developed by the national laboratories in the US in the mid-1970s to predict energy use in commercial and residential buildings (http://gundog.lbl.gov/). DOE-2.1e is no longer under active maintenance, but is still the underlying calculation engine for several software packages, including Energy Gauge. A private software developer (JJ Hirsh and Associates) owns and maintains the newest version of the DOE-2.2 calculation engine and the widely used interface program (eQuest). http://www.doe2.com/ Advantages: well understood, flexible simulation program. Disadvantages: no longer updated or supported.
EnergyPlus	Flexible building simulation model for commercial and residential buildings. Public domain calculation engine developed by the US Department of Energy (DOE). www.energyplus.gov Advantages: state of the art building model with robust, well-developed equipment components. Disadvantages: limited flexibility to add correlations to degrade performance.
TRNSYS	Highly flexible, research grade package for analyzing transient thermal systems. Includes pre-developed models for building envelope and other HVAC components. www.trnsys.com Advantages: highly flexible, can consider any user-defined equation or component models. Disadvantages: difficult to use, and cumbersome to define building envelope details.

In this study, we used a building model developed in TRNSYS to simulate the integrated performance of heat pumps in residential applications (CDH Energy Corp., 2010). This model was originally applied to simulate an integrated desiccant system's performance (Henderson and Sand, 2003), and it was later refined to consider several issues germane to this residential study, including duct leakage and the part load performance of air conditioners (Henderson et al., 2007) and refrigerant charge impacts (Sachs et al., 2009). The model is driven by typical meteorological year weather data sets TMY3 (Wilcox and Marion, 2008) on a small time-step (e.g., 1.2 minutes). A detailed thermostat model turns the mechanical systems "on" and "off" at the end of each time step depending on the calculated space conditions.

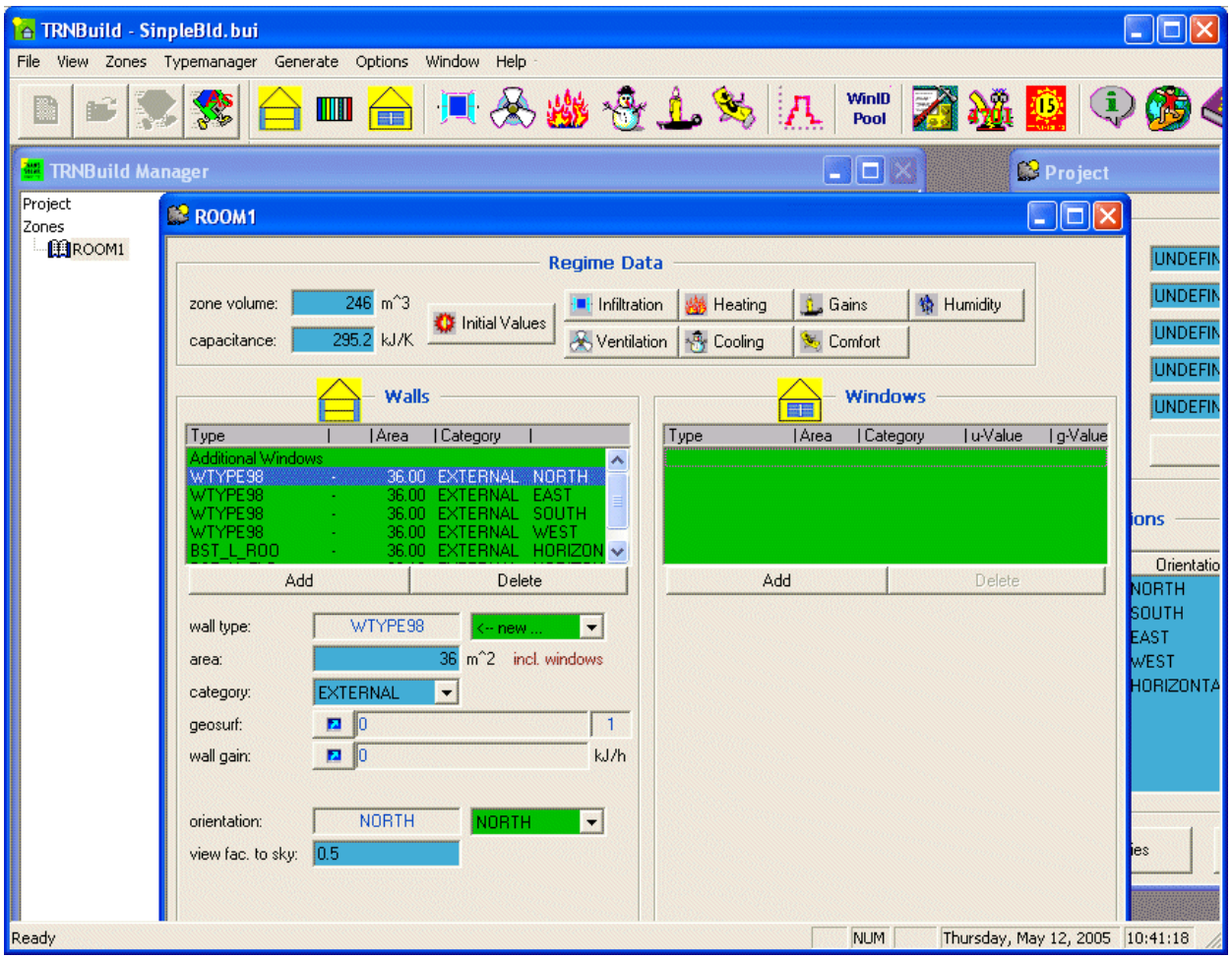


Figure US-15. Screen shot of TRNBuild used to define the building envelope details

4.2. Building and Weather City Definitions

Table US-11 lists the climates with representative locations and house structures considered in this study. Two houses were modeled: a slab-on-grade house, and a house with a basement. The simulated residential buildings corresponded to a code-compliant house, with a HERS score of approximately 100, with appropriate levels of insulation and other features corresponding to each climate. The slab-on-grade houses were modeled with ducts located in the attic. The houses with basements were modeled with ducts located in a semi-conditioned space. For Houston, TX, only a slab-on-grade house was studied because houses with basements are rarely built in this location.

The selected cities represent each of the International Energy Conservations Code (IECC) climate zones 2 through 6 shown in Figure US-16 from a hot and humid climate to a heating dominated climate. This selection enabled prediction on how different faults will affect air conditioner and heat pump performance in the most prevalent climates in the U.S. TMY3 weather data were used for each location.

Table US-11. Climates, locations and structures considered

Zone	Climate	Location	Slab-on-grade house	House with basement
2	Hot and humid	Houston, TX	Yes	No
3	Hot and dry climate	Las Vegas, NV	Yes	Yes
4	Mixed climate	Washington, DC	Yes	Yes
5	Heating dominated	Chicago, IL	Yes	Yes
6	Cold	Minneapolis, MN	Yes	Yes

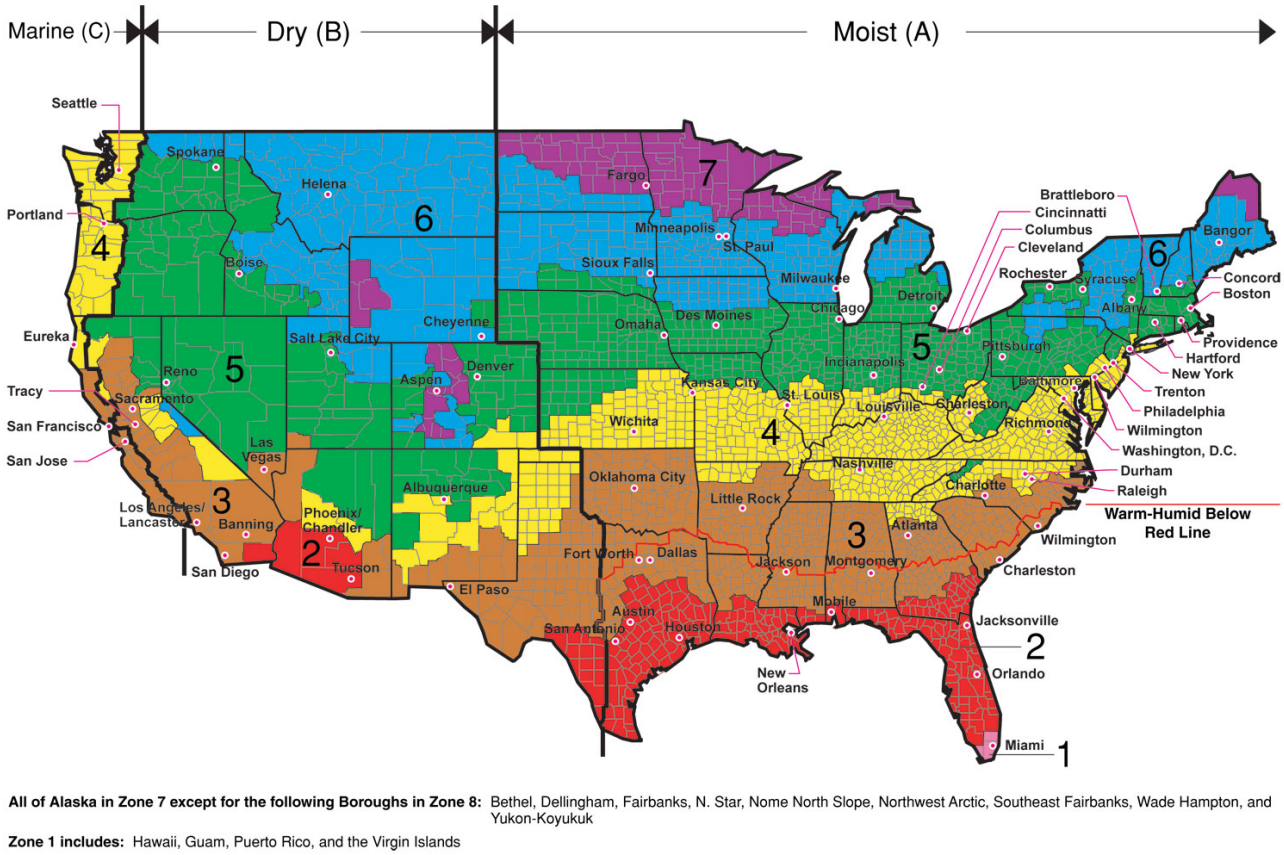


Figure US-16. IECC climate zone map

4.3. Building and Enclosure Thermal Details

A 185.8 m^2 (2,000 ft^2) three-bedroom house was modeled as a slab-on-grade with a separate attic zone – or a 2-zone model – in TRNSYS Type 56. This house is similar to that simulated by Rudd et al. (2013) for a recently completed ASHRAE research project (RP-1449). Also, a 3-zone model was developed for the house with a basement zone. The basement was not directly conditioned but coupled to the main zone via zone-to-zone air exchange. The characteristics of the buildings are listed in Table US-12 for each climate.

Table US-12. Specifications for simulated houses (HERS Index ≈100)

a) I-P units

Parameter	Houston, TX (Climate Zone 2)	Las Vegas, NV (Climate Zone 3)	Washington, DC (Climate Zone 4)	Chicago, IL (Climate Zone 5)*
Wall insulation R-value (nominal)	13	13	13	19
Cavity	13	13	13	19
Sheathing	0	0	0	0
framing factor	0.23	0.23	0.23	0.23
Ceiling insulation R-value	30	30	38	38
Slab insulation R-value (2' down)	0	0	0	0
Basement Walls	na	na	na	na
Window U-value ($Btu \cdot h^{-1} ft^{-2} F^{-1}$)	0.75	0.65	0.40	0.35
Window SHGC	0.40	0.40	0.40	0.40
Building enclosure air leakage (ACH50)	7	7	7	7
Enclosure ELA (in^2)	98.1	98.1	98.1	98.1
Duct air leakage to outside (%)	6% sup, 4% ret	6% sup, 4% ret	6% sup, 4% ret	6% sup, 4% ret

Supply duct area in attic (ft^2)	544	544	544	544
Return duct area in attic (ft^2)	100	100	100	100
Duct R-value	6	6	6	6
SEER, EER	13, 9.6	13, 9.6	13, 9.6	13, 9.6
HSPF, COP	7.7, 2.3	7.7, 2.3	7.7, 2.3	7.7, 2.3
Internal heat gain (lumped)** (people+lighting+appliances)	72.70 kBtu/day	72.70 kBtu/day	72.70 kBtu/day	72.70 kBtu/day
Internal moisture generation	12 lb/day	12 lb/day	12 lb/day	12 lb/day
HERS	106	108	108	107

*This house was also used in simulations for Minneapolis, MN (Climate Zone 6)
**DOE Building America benchmark (Hendron, 2008)

b) SI units

Parameter	Houston, TX (Climate Zone 2)	Las Vegas, NV (Climate Zone 3)	Washington, DC (Climate Zone 4)	Chicago, IL (Climate Zone 5)*
Wall insulation R(SI)-value (nominal)	2.29	2.29	2.29	3.35
Cavity	2.29	2.29	2.29	3.35
Sheathing	0	0	0	0
framing factor	0.23	0.23	0.23	0.23
Ceiling insulation R(SI)-value	5.38	5.38	6.69	6.69
Slab insulation R(SI)-value (2' down)	0	0	0	0
Basement Walls	na	na	na	na
Window U-value ($W \cdot m^{-2} \cdot K^{-1}$)	4.3	3.7	2.3	2.0
Window SHGC	0.40	0.40	0.40	0.40
Building enclosure air leakage (ACH50)	7	7	7	7
Enclosure ELA (m^2)	0.063	0.063	0.063	0.063
Duct air leakage to outside (%)	6% sup, 4% ret	6% sup, 4% ret	6% sup, 4% ret	6% sup, 4% ret
Supply duct area in attic (m^2)	50.5	50.5	50.5	50.5
Return duct area in attic (m^2)	9.3	9.3	9.3	9.3
Duct R(SI)-value	1.1	1.1	1.1	1.1
SEER (I-P), COP	13, 9.6	13, 9.6	13, 9.6	13, 9.6
HSPF (I-P), COP	7.7, 2.3	7.7, 2.3	7.7, 2.3	7.7, 2.3
Internal heat gain (lumped)** (people+lighting+appliances)	76.70 MJ/day	76.70 MJ/day	76.70 MJ/day	76.70 MJ/day
Internal moisture generation	5.4 kg/day	5.4 kg/day	5.4 kg/day	5.4 kg/day

* This house was also used in simulations for Minneapolis, MN (Climate Zone 6)
**DOE Building America benchmark (Hendron, 2008)

The slab-on-grade house only has perimeter slab insulation in climate zones 4 and 5 (Figure US-17). For the house with a basement (Figure US-18), the basement is connected to the main house by openings that are assumed to allow zone-to-zone air exchange of heat and moisture equivalent to 236 $L \cdot s^{-1}$ (500 cfm). The basement walls are modeled as 102 mm (4 inch) thick concrete, with R (SI)-1.76 (R-10) exterior foam insulation in climate zones 3, 4 and 5.

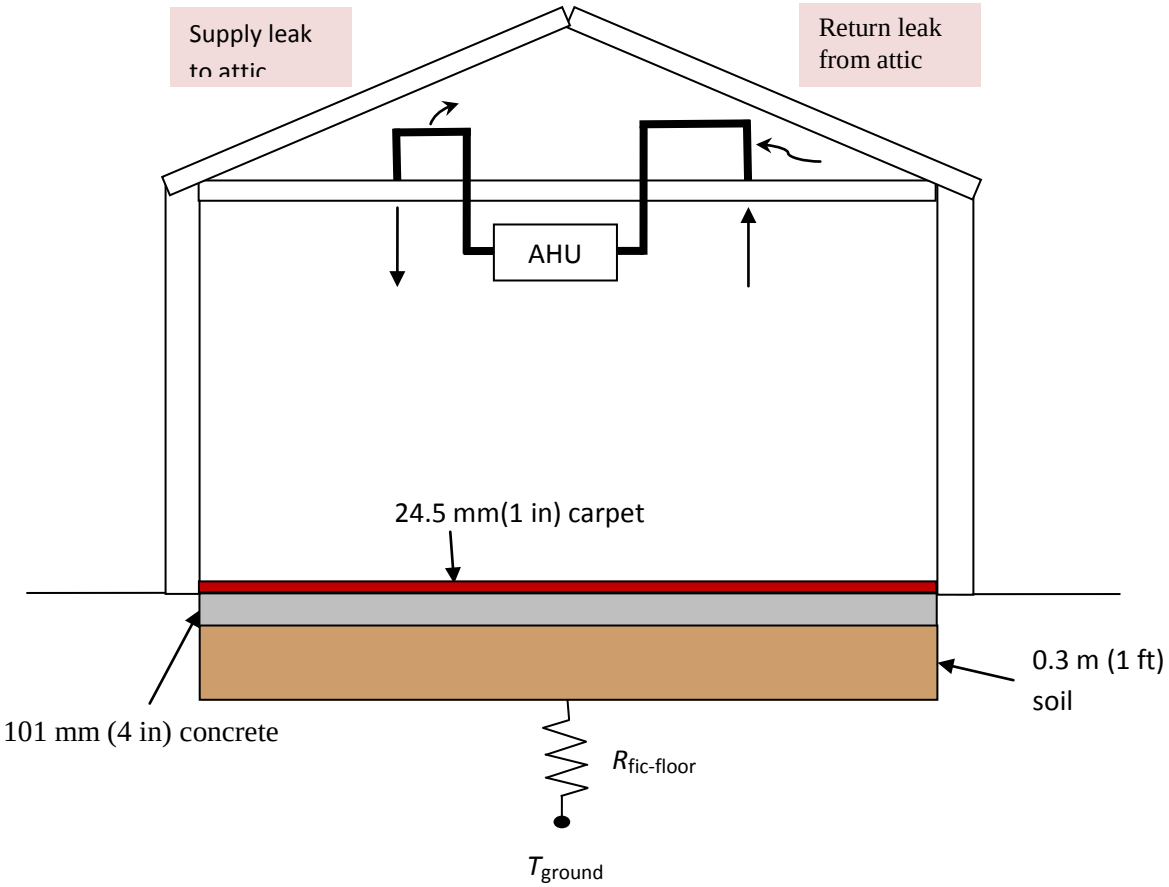


Figure US-17. Schematic of a slab-on-grade house (ducts located in the unconditioned attic)

Both the slab-on-grade and basement homes are modeled by adding a "fictitious layer" into the resistance between the zone and ground temperature. This fictitious R-value is added to provide the amount of heat loss through the surfaces determined by the F-factor method ($R_{effective}$) as recommended by Winkelmann (1998). A schematic of this model is shown in Figures US-17 and US-18. Tables US-13 and US-14 summarize the calculations to determine the necessary R-value for the fictitious layer.

The above-ground portions of the slab-on-grade and basement houses are identical for each climate zone. Each model has exterior walls with layers of drywall, insulation (R (SI)-2.3 (R-13) or R(SI)-3.3 (R-19), depending on the climate zone), and stucco as the outside surface. Windows take up approximately 22 % of all of the exterior walls; 10.2 m^2 (109.6 ft^2) on the north and south facing walls, and 6.5 m^2 (70.4 ft^2) on east and west facing walls.

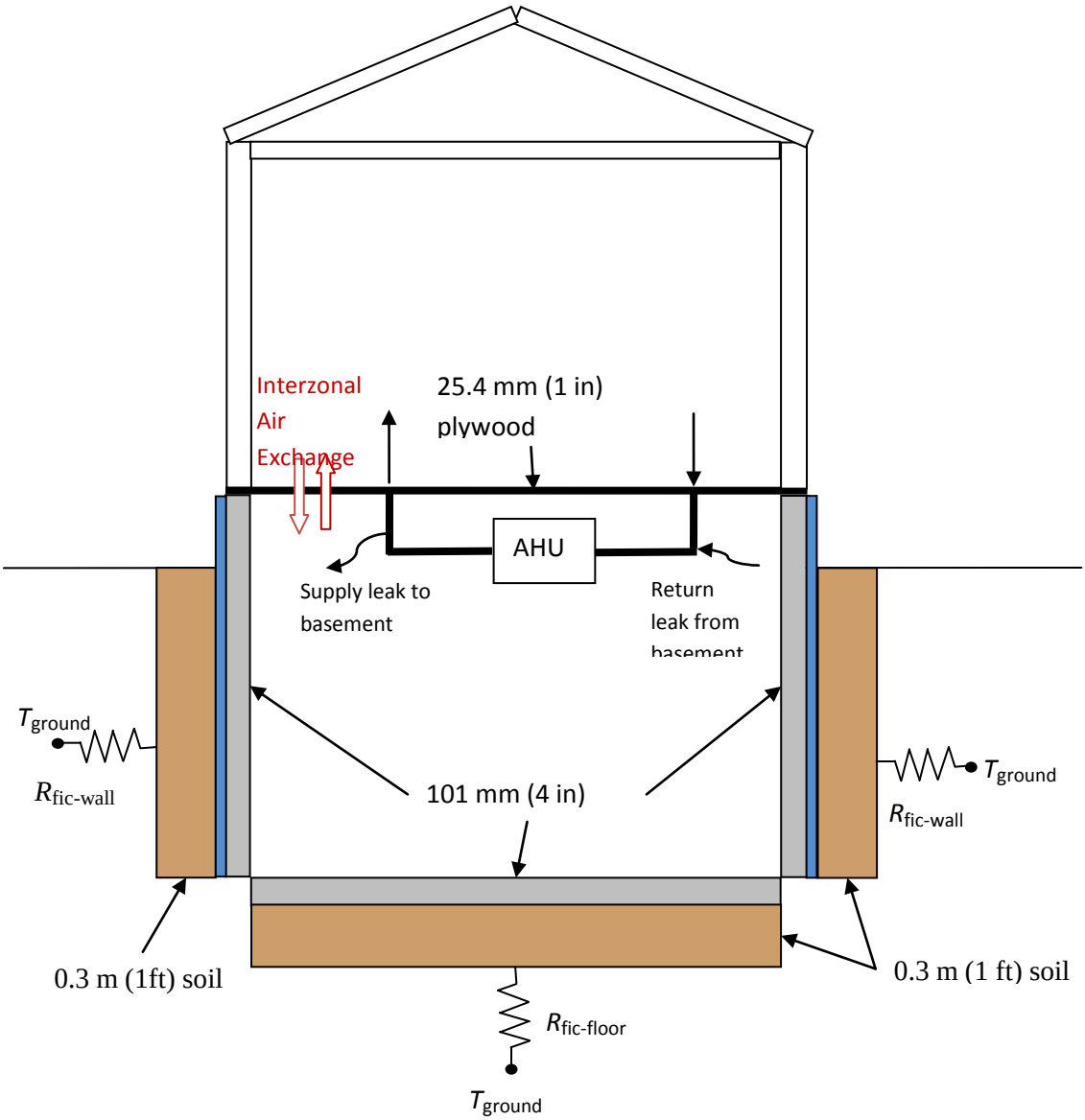


Figure US-18. Schematic of a house with basement (ducts located in the semi-conditioned basement)

The ceiling (i.e. boundary between main zone and attic) is made up of a layer of drywall, framing and insulation (R(SI)-5.3 (R-30) or R(SI)-6.7 (R-38), depending on climate zone. The attic has gable walls on the east and west sides, and roof surface on the north and south sides. The roof is sheathed in plywood and then covered with asphalt shingles. The east and west surfaces (gables) are made up of plywood on the inside surface with stucco on the outside surface.

Table US-13. Calculation of R-values for basement walls and floor

Basement Wall						
Material	Thickness		Resistance		Total R-Value	
					R(SI)	R
	m	ft	K·m·W^{-1}	h·ft·°F·Btu^{-1}	K·m² W^{-1}	h·ft²·°F·Btu^{-1}
Concrete	0.10	0.33	0.775	1.33	0.0775	0.44
Soil	0.30	1.00	1.18	2.01	0.354	2.00
Foam	0.035	0.115	25.1	43.5	0.881	5.00
R_{fic}	Massless				0.111	0.63
			$R_{effective}$		1.42	8.08
Basement Floor						
Material	Thickness		Resistance		Total R-Value	
					R(SI)	R
	m	ft	K·m·W^{-1}	h·ft·°F·Btu^{-1}	K·m² W^{-1}	h·ft²·°F·Btu^{-1}
Concrete	0.10	0.33	0.775	1.33	0.0775	0.44
Soil	0.30	1.00	1.18	2.01	0.354	2.00
R_{fic}	Massless				3.26	18.5
			$R_{effective}$		3.69	20.95

Table US-14. Calculation of R-values for slab-on-grade floor

Slab Resistance – Climate Zones 2 and 3						
Material	Thickness		Resistance		Total R-Value	
					R(SI)	R
	m	ft	K·m·W^{-1}	h·ft·°F·Btu^{-1}	K·m² W^{-1}	h·ft²·°F·Btu^{-1}
Carpet	0.025	0.083	14.52	25.13	0.363	2.06
Concrete	0.10	0.33	0.775	1.33	0.0775	0.44
Soil	0.30	1.00	1.18	2.01	0.354	2.01
R_{fic}	Massless				0.958*	5.44*
			$R_{effective}$		1.75	9.95
Slab Resistance – Climate Zones 4 and 5						
Material	Thickness		Resistance		Total R-Value	
					R(SI)	R
	M	ft	K·m·W^{-1}	h·ft·°F·Btu^{-1}	K·m² W^{-1}	h·ft²·°F·Btu^{-1}
Carpet	0.025	0.083	14.52	25.13	0.363	2.06
Concrete	0.10	0.33	0.775	1.33	0.0775	0.44
Soil	0.30	1.00	1.18	2.01	0.354	2.01
R_{fic}	Massless				2.19*	12.42*
			$R_{effective}$		2.98	16.93

*The difference in R_{fic} between climate zones 2/3 and 4/5 is due to the perimeter insulation of the slab in climate zones 4 and 5.

4.3.1. Building Enclosure Air Leakage

The AIM-2 infiltration model (Walker and Wilson, 1998; ASHRAE, 2009a) relates infiltration to wind and indoor-outdoor temperature difference for each time step. All simulations in this study used coefficients representing shelter from buildings located across the street. An equivalent leakage area (ELA) of 0.0633 m^2 (98.1 in^2) was chosen to provide the desired seven air changes per hour (ACH) at 50 pascal pressure differential (ACH50 for the main zone in each building model).

The attic used the same AIM-2 equations to determine leakage as a function of wind and temperature difference. The attic ELA was set to be 0.366 m^2 (567 in^2) for each of the climate zones, or about 5 times the leakage rate for the HERS 100 house (Fugler, 1999). In houses with basements, that zone was assumed to have no leakage to outdoors.

4.3.2. Duct Leakage and Thermal Losses

For the slab-on-grade houses, the ducts were modeled to be in the attic space and all the air leakage and thermal losses/gains go into that zone. The details of the duct model are given in Appendix A. For houses with basements, there is no duct leakage to the attic (all leaks are assumed to be into the conditioned space, so they are ignored). Duct leakage was assumed to be 10 % of flow, or 6 % on the supply side and 4 % on the return side. Duct insulation was assumed to be R(SI)-1.1 (R-6) with a supply duct area of 50.5 m^2 (544 ft^2) and a return duct area of 9.3 m^2 (100 ft^2) for a 10.6 kW (3-ton) unit. The duct areas were increased and decreased proportionally based on the size (or nominal tonnage) of the heat pump unit.

4.3.3. Moisture and Thermal Gains

The scheduling or profile of internal heat and moisture generation was taken from the Building America Benchmark Definition (Hendron, 2008). Sensible gains from all sources were assumed to be 76.7 MJ/day (72.7 kBtu/day).

Internal moisture generation from all sources was specified as 5.4 kg/day (12 lb/day), or less than half of the ASHRAE Standard 160 moisture generation rate of 14.2 kg/day (31.2 lb/day) for a three-bedroom house (ASHRAE, 2009b). The ASHRAE 160 value is meant to be a "worst case" design condition and therefore would not be expected to correspond to average conditions.

4.3.4. Moisture and Thermal Capacitance

Moisture storage in the building materials and furnishings and the rate of mass transfer into storage are important hygrothermal parameters affecting the diurnal swings in indoor humidity. Building material moisture storage was modeled with a simple lumped parameter method with mass factor added to the air node in the zone model:

$$C\ \frac{\mathrm{d}w_\mathrm{i}}{\mathrm{d}t} = m \cdot (w_\mathrm{i} - w_\mathrm{o}) + Q_\mathrm{internal} - Q_\mathrm{AC,latent} \qquad (4.1)$$

The moisture capacitance term is usually set to a multiple of the air mass inside the house. The Florida Solar Energy Center used more detailed moisture models including Effective Moisture Penetration Depth (EMPD) to show that reasonable factors for the air mass multiplier are 20 to 30 times the air mass (EPA, 2001).

As a result of the calibration efforts (Appendix C in Rudd et al., 2013), a 30x multiplier for moisture capacitance was used for the main zone and the basement. The attic used a moisture capacitance factor of 15x.

Thermal capacitance was simulated by adding internal walls to the model with 371.6 m^2 (4000 ft^2) of exposed wall surface area. The thermal mass of the air node was also increased by a factor of 20 to 12,331 $kJ \cdot K^{-1}$ (6494 $Btu \cdot F^{-1}$) to reflect the impact of furniture and other material in the space. The attic was assumed to have a thermal capacitance of 1x and the basement (where applicable) was assumed have a thermal capacitance multiplier of 10x.

4.3.5. Window Performance

The window model in Type 56 uses the window parameters generated by LBNLs WINDOW5 software, which is considerably more detailed than the NFRC rating values generally used in residential practice and building codes. The LBNL WINDOW5 inputs for this project were determined following the methodology developed by Arasteh et al. (2009) for use in EnergyPlus.

4.4. Mechanical Ventilation

The only mechanical ventilation option considered in this study is an exhaust fan. The fan operated continuously to provide sufficient ventilation to the house. Figure US-19 shows the airflow configuration used in this study. The fans provided an average rate 98.5 $m^3 \cdot h^{-1}$ (58 cfm) required by ASHRAE Standard 62.2 (ASHRAE, 2013) for the 185.8 m^2 (2000 ft^2) three-bedroom house. The exhaust fan power was assumed to be 0.85 $kJ \cdot m^{-3}$ (0.4 $W \cdot cfm^{-1}$).

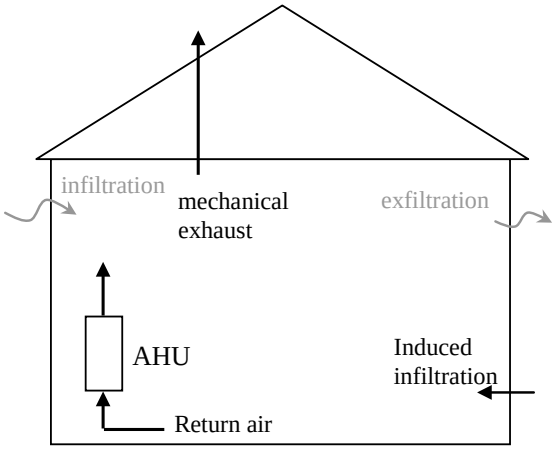


Figure US-19. Schematic of a mechanical exhaust system

The combined impact of infiltration, ventilation, and duct leakage were considered by using the equations below. The duct leakage was always a net out, so that additional net flow was an exhaust.

V_{in} = incoming ventilation flow
V_{out} = sum of all exhaust flows (exhaust fan, net duct leakage, etc.)
$V_{balanced}$ = MIN (V_{in}, V_{out})
$V_{unbalanced}$ = MAX (V_{in}, V_{out}) – $V_{balanced}$
V_{inf} = infiltration flow calculated for building for the timestep
$V_{combined}$ = MAX ($V_{unbalanced}$, V_{inf} + 0.5·$V_{unbalanced}$) + $V_{balanced}$

The net mechanical inlet flows were subtracted from $V_{combined}$ to determine the remaining non-mechanical ventilation (or infiltration) rate acting on the building envelope. A mass balance tracked CO_2 levels in the space and confirmed the net impact of ventilation to be similar between the cases.

4.5. Air Flow Imbalance

Duct leakage is often exacerbated by interactions with building envelope leakage, depressurization caused by exhaust fan operation, and supply and return imbalances caused by closing interior doors (for central return systems). Cummings and Tooley (1989) and Modera (1989) both showed that the pressures induced by air handlers were much greater than the naturally-induced pressures from wind and stack effects in cooling dominated climates. Pressure mapping by Cummings and Tooley (1989) also showed that the supply/return imbalances caused by closing interior doors were also substantial.

One option for considering these interactions is to use a multi-zone, flow-pressurization model such as CONTAM 3.0 (Walton and Dols, 2010). A model can be developed to evaluate the interactions of building envelope leakage paths, duct leakage, and zone pressurization with the supply air (when doors are closed). These models can track airflow, but cannot consider the thermal performance of the building envelope, nor the energy use of the space-conditioning systems.

In a small, time-step thermal building simulation model, it is possible to properly account for the combined effects of "unbalanced" duct leakage, unbalanced ventilation, and infiltration using a simpler approach. The following procedure accounts for the interactions of unbalanced ventilation and duct leakage with infiltration. The calculation is based on the approach summarized in Barnaby and Spitler (2004) as well as the ASHRAE Handbook of Fundamentals, Chapter 17 (ASHRAE 2009a):

V_{in} = incoming ventilation airflow
V_{out} = sum of all exhaust airflows (exhaust fan, supply duct leaks, etc.)
$V_{balanced}$ = MIN (V_{in}, V_{out})
$V_{unbalanced}$ = MAX (V_{in}, V_{out}) − $V_{balanced}$
V_{inf} = infiltration flow calculated for building for the timestep
$V_{combined}$ = MAX ($V_{unbalanced}$, V_{inf} + 0.5· $V_{unbalanced}$) + $V_{balanced}$

4.6. Heat Pump Specifications and Modeling

A conventional heat pump unit with a 13 SEER and 7.7 HSPF rating was used in the simulations. The cyclic degradation coefficient, C_D, of the heat pump was 0.15 in both cooling and heating. The required size of the unit was determined for each climate using ACCA Manual J (ACCA 2011a). Houses in Houston and Las Vegas had a heat pump with cooling capacity of 10.6 kW (3 ton) and 12.3 kW (3.5 ton), respectively. The Washington D.C., Chicago, and Minneapolis houses had 8.8 kW (2.5-ton) units.

The detailed heat pump model required separate inputs for the gross COP at nominal conditions, sensible heat ratio (SHR), and indoor fan power. Table US-15 lists the rated parameters and corresponding inputs to the heat pump model. The fan power assumed for rated conditions and used to calculate SEER is listed along with the actual fan power assumed for operation. The fan power at rated conditions was assumed to be 0.53 kJ·m^{-3} (0.25 W/cfm), while the actual fan power was 1.06 kJ·m^{-3} (0.5 W/cfm).

Table US-15. Heat pump cooling characteristics

Unit Description	Rated Performance			Input Parameters		
SEER 13 unit; Single-speed PSC fan motor	Rated SEER Btu·W^{-1}·h^{-1}	Rated COP	Rated Fan Power kJ·m^{-3}	Gross COP	Actual Fan Power kJ·m^{-3}	SHR
	13	2.81	0.53	4.05	1.06	0.77

Note: Gross COP is a ratio of gross cooling capacity (refrigerant-side capacity) and outdoor unit power (includes compressor, outdoor fan, and controls powers) at the nominal rating point: 35 °C (95 °F) outdoor dry-bulb temperature, 26.7 °C/19.4 °C (80 °F/67 °F) indoor dry-bulb/wet-bulb temperature, and 217.4 m^3·h^{-1}kW^{-1} (450 cfm/ton) supply airflow.

The airflow in the cooling and heating mode was assumed to be 181.1 m^3·h^{-1}·kW^{-1} (375 cfm/ton). Data from the laboratory testing at NIST was used to correct the normalized fan power from the nominal value of 1.06 kJ·m^{-3} (0.5 W·cfm^{-1}) as the airflow changes from the nominal value of 181.1 m^3·h^{-1}·kW^{-1} (375 cfm/ton). The data showed a linear trend. The best fit to the measure data (Eq. 4.2) was used to predict the variation in fan power as the airflow varies.

$$\left(\frac{W_{fan}}{V}\right) = 1.06 - \left[\left(\frac{V}{Q}\right) - 181.1\right] \cdot 0.00316 \qquad (4.2)$$

where $\left(\frac{W_{fan}}{V}\right)$ = normalized fan power from the nominal value, kJ·m^{-3}

$\left(\frac{V}{Q}\right)$ = airflow to system capacity ratio, m^3·h^{-1}·kW^{-1}

The heating performance for the heat pump used the generic performance curves developed for EnergyGauge (Parker et al.,1999). The generic model is based on catalog data from a series of single-speed heat pump products (ranging from 10 to 14.5 SEER) and was shown to be appropriate over a range of heat pump efficiency levels. The generic model predicts the variation in heating capacity and power input as a function of outdoor dry-bulb temperature, indoor entering temperature, and the airflow ratio (actual airflow divided by nominal airflow).

The impact of defrost operation was determined by the defrost degradation function shown in Figure US-20, which is also used in EnergyGauge. This simple function[1] predicts the degradation as a function of outdoor dry-bulb temperature for a time-initiated, temperature-terminated defrost controller. The impact of defrost starts at temperatures below 8.3 °C (47 °F), peaks at 11 % by 2.8 °C (37 °F), and tapers off to 7.5 % at lower ambient temperatures. For comparison, the graph also includes the degradation rate implied by ASHRAE Standard 116 (ASHRAE, 2010).

[1] Actually, defrost is a function of both temperature and ambient humidity. While more sophisticated defrost models are available in EnergyPlus (see the 2012 Engineering Reference Manual), these approaches were found to have flaws and could not be successfully implemented here for this study.

The heat pump gross COP at 8.3 °C (47 °F) was 2.7. The nominal gross heating capacity, also at 8.3 °C (47 °F), was 10 % greater than the nominal gross cooling capacity. A supplemental 10 kW electric heater was activated if the space temperature dropped 0.28 °C (0.5 °F) below the heating set point or to 20.3 °C (68.5 °F) in Chicago, Washington, DC, and Minneapolis. The degraded performance of a heat pump due to faults was modeled by applying the heat pump normalized performance parameters described in Section 3.

Table US-16 lists thermostat set points for heating and cooling. The 21.1 °C (70 °F) heating set point was selected as appropriate for temperate climates, while the 22.2 °C (72 °F) set point was deemed as more appropriate for the warmer climates. The cooling set point of 25.6 °C (78 °F) was selected as most consistent with homeowner preferences in warm climates. In colder climates 24.4 °C (76 °F) was used.

The impact of thermostat deadband and anticipator were explicitly considered in this short time-step model in the cooling mode as per Henderson (1992). The deadband was ±0.56 °C (1.0 °F) around the desired temperature point. The anticipator temperature gain was 1.4 °C (2.5 °F), and the time constant of the anticipator was 90 seconds. The sensing element of the thermostat had a time constant of 300 seconds. The result was the temperature "droop" with runtime fraction of about 1.1 °C (2.0 °F). In the heating mode a simple deadband of ±0.6 °C (1.0 °F) around the set point was used without an anticipator or sensing element time constant.

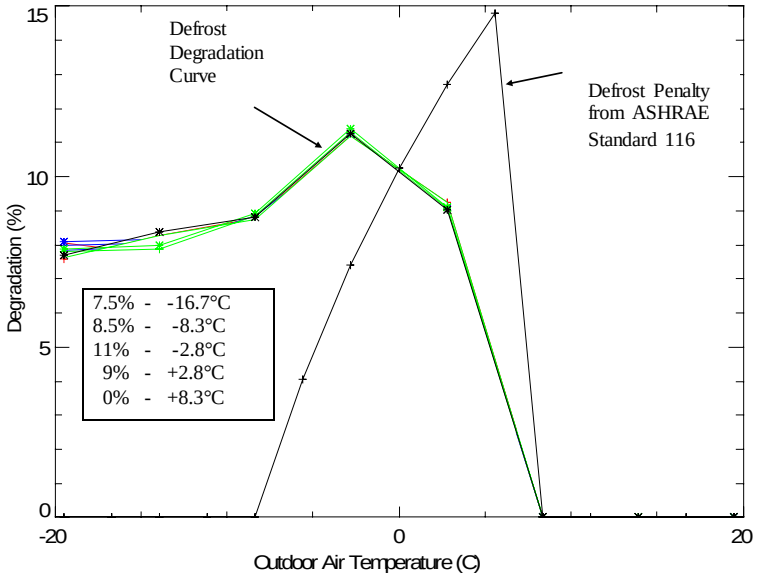


Figure US-20. Capacity degradation due to defrost as a function of outdoor temperature (The different color lines on the plot show the defrost degradation from catalog data. The table of values summarizes the average values used in the simulations.)

Table US-16. Thermostat cooling and heating set points

Zone	Location		Cooling Set Point °C (°F)	Heating Set Point °C (°F)
2		Houston, TX	25.6 (78)	22.2 (72)
3		Las Vegas, NV		
4		Washington, DC	24.4 (76)	21.1 (70)
5		Chicago, IL		
6		Minneapolis, MN		

4.7. Cost of Electricity

Total heat pump operating costs were determined using the electric rates listed in Table US-17.

Table US-17. Cost of electricity

Zone	Location	Electric Utility	Cost	
			$/MJ	$/kWh
2	Houston, TX	Entergy	0.306	0.085
3	Las Vegas, NV	NV Energy	0.454	0.126
4	Washington, DC	Pepco	0.508	0.141
5	Chicago, IL	ComEd	0.461	0.128
6	Minneapolis, MN	Northern States Power	0.389	0.108

Note: Electric costs are from Form 826 data for local utility in 2010 for residential sector (EIA, 2012)

5. SIMULATIONS OF BUILDING/HEAT PUMP SYSTEMS WITH INSTALLATION FAULTS

Section 4.2 discussed the IECC climate zones and baseline houses considered in this study. The selected house options include a slab-on-grade house and a house with a basement for Las Vegas, Washington, DC, Chicago, and Minneapolis, and a slab-on-grade house only for Houston.

The following sections present results of annual simulations of energy consumption for a heat pump operating under different levels of different installation faults. These annual simulations focused on performance issues of the house/heat pump systems related to heat pump capacity and energy consumption while maintaining the target indoor dry-bulb temperature (shown in Table US-16) within the temperature band imposed by the thermostat. For a few faults, we performed additional annual simulations with a lowered thermostat set-point temperature to mimic this common response to elevated indoor humidity levels caused by installation faults.

Results of annual simulations of energy consumptions are presented in the format consistent with Table US-18. The threshold 55 % relative humidity value used in the third column was selected as the level above which humidity might start to be a concern. This threshold is slightly lower than the limit of 60 % relative humidity, which has historically been identified as the space condition where mold growth can occur in the building envelope (Sterling et al., 1985). The ‘Space Temp Max’ column contains the highest indoor temperature reached during the cooling season. The column ‘AC Energy’ contains the energy used by the compressor and outdoor fan to provide cooling; the column ‘Htg Energy’ contains the energy used by the compressor, outdoor fan, and backup heat to provide heating; and the column ‘AHU Fan Energy’ contains the energy used by the indoor fan during the whole year. The column ‘TOTAL ENERGY’ contains the total energy used by the heat pump throughout the entire year, which consists of the energy use listed in the three previous columns, and the energy used by the home exhaust fan.

5.1. Annual Energy Consumption in Baseline Houses

Table US-18 presents simulation results of the annual heat pump operating energy consumption, energy cost, and relative energy cost referenced to that of the slab-on-grade house for each locality. The energy use of the basement house is from 17 % to 19 % lower than that for the slab-on-grade house in most climates. Most of this difference is due to duct leakage: the basement house has the ducts in the basement (with no losses) instead of ducts in the attic for the slab-on-grade house with the assumed typical leakage of 10 % (Section 4.3.2).

Table US-18 also includes results from additional runs for the slab-on-grade house without duct leakage (i.e., treated as ducts in the conditioned space) denoted in the table as “Slab, Ducts Inside”. When the duct leakage and duct thermal losses are eliminated, the slab-on-grade and basement houses perform within 3 % for Las Vegas and within 9 % for the cold climates. The basement house does have higher energy use in the colder climates.

Table US-18. Annual energy consumption and cost in baseline houses

		Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	Slab-on-Grade	1,512	26.6	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	Basement														
	Slab, Ducts inside	1,715	25.2	1,555	588	0.3	2,142.9	4.3	0.789	13,007	6,623	4,339	24,700	$583	79%
Las Vegas	Slab-on-Grade	-	27.0	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	Basement	-	25.3	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	81%
	Slab, Ducts inside	-	25.3	1,536	668	0.3	2,204.5	3.7	1.000	15,941	8,763	5,207	30,642	$1,072	78%
Washington, DC	Slab-on-Grade	253	25.1	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	Basement	654	24.0	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	82%
	Slab, Ducts inside	280	24.1	944	1,532	12.9	2,476.3	4.4	0.801	6,301	15,111	4,179	26,322	$1,031	73%
Chicago	Slab-on-Grade	189	25.0	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	Basement	289	24.0	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	83%
	Slab, Ducts inside	203	24.0	815	2,288	70.0	3,103.7	4.5	0.819	5,369	24,753	5,238	36,092	$1,283	71%
Minneapolis	Slab-on-Grade	13	25.2	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	Basement	61	24.0	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	83%
	Slab, Ducts inside	15	24.1	711	2,902	216.9	3,613.5	4.5	0.838	4,670	36,410	6,098	47,909	$1,437	69%

5.2. Simulation with Single Faults

5.2.1. Studied Faults

Table US-19 summarizes the studied faults and their level values used in simulations.

Table US-19. Studied faults in the cooling and heating mode

Fault Type	Fault Levels (%)	
	Cooling mode	Heating mode
Heat Pump Sizing (SIZ)	-20, 25, 50, 75, 100	-20, 25, 50, 75, 100
Duct Leakage (DUCT)	0, 10, 20, 30, 40, 50	0, 10, 20, 30, 40, 50
Indoor Coil Airflow (AF)	-36, -15, 7, 28	-36, -15, 7, 28
Refrigerant Undercharge (UC)	-10, -20, -30	-10, -20, -30
Refrigerant Overcharge (OC)	10, 20, 30	10, 20, 30
Excessive Refrigerant Subcooling (SC)	100, 200	-
Non-Condensable Gases (NC)	10, 20	10, 20
Electric Voltage (VOL)	-8, 8, 25	-8, 8, 25
TXV Undersizing (TXV)	-60, -40, -20	-

5.2.2. Effect of Heat Pump Sizing

Changing the size of the heat pump for a given house – either undersizing or oversizing – impacts the heat pump performance in several ways:

- Cycling losses increase as the unit gets larger; the unit runs for shorter periods and the degraded performance at startup has more impact (parameters used in simulations are: time constant = 45 seconds, or $C_D \sim 0.15$).
- In the cooling mode, the shorter run periods impact the moisture removal capability (i.e., ability to control indoor humidity levels) because operational steady-state conditions are an even smaller portion of the runtime fraction.
- In the cooling mode, continuous fan operation with compressor cycling greatly increases moisture evaporation from the cooling coil. However, this impact is minimal with auto fan control (indoor fan time 'on' and 'off' the same as that of the compressor), since only a small amount of evaporation occurs with the assumed 4 % airflow during the off-cycle with the indoor fan off. If the air conditioner controls include an off-cycle fan delay – that keeps the fan on for 30-90 seconds after the compressor stops – then the impact of off-cycle evaporation is in between these two extremes (Shirey et al., 2006). The results in this study assumed auto fan operation with no fan delay.
- In the heating mode, the backup heater runtime is lower for the oversized unit since the larger heat pump meets more of the winter heating needs.

Heat pump sizing also affects the level of duct losses. This study considered two heat pump sizing scenarios with regard to the sizing of the air duct. In scenario (1), the heat pump and air duct are proportionally

undersized or oversized, i.e., the duct flow area increases proportionally to the increase of heat pump capacity. As a result, the air mass flux through the duct remains unchanged, and the duct surface area increases with the square root of capacity ratio (unit capacity/design building load). The duct losses to the attic (thermal and air leak losses) tend to increase with the unit size since the surface area of the duct and the amount of airflow increases; however, the lower indoor fan runtime associated with an oversized heat pump has the opposing influence (reduces duct losses to the attic) since in the model the losses only occur when the fan is "on". Not included in this analysis is the impact that oversizing has on moisture control, especially at part load (see Sonne et al. (2006) for an in-depth review on this topic).

In scenario (2), the duct has been sized for a heat pump of nominal capacity and remains unchanged for different size heat pumps. When the heat pump is oversized, the fan speed is increased but the airflow does not reach the target flow rate because the unit is not capable of overcoming the increased external static pressure. Since the indoor fan works against increased static pressure, the fan power changes per the fan curve, i.e., fan power increases with an increasing unit size. The increased pressure in the duct increases the duct leakage. Table US-20 shows the realized airflow per unit capacity, external static pressure, and duct leakage for scenario (1) and scenario (2).

Table US-21 compares the effect of 100 % oversizing on the cooling and the heating performance for the slab-on-grade house for the five studied cities and two oversizing scenarios. For scenario (1) - duct size changes - oversizing degrades the cooling COP only modestly (about 2 %). The thermostat has "droop" that causes the average space temperature to drop by (1.1 ~ 1.7) °C ((2 ~ 3) °F) with lower runtime fractions. In addition, the larger ducts have more losses to the uninsulated attic, but the shorter indoor runtime has the opposing effect. The net effect is that the energy use in the cooling mode increases by (2 ~ 3) %. In the heating mode, the larger heat pump meets more of the space heating load, so less operation of the inefficient auxiliary resistance heater is required. As a result, the heating energy decreases by (3 ~ 4) % in the cooling-dominated climates and almost 9 % in the heating-dominated climates. Overall, the total annual energy use is barely affected in the cooling-dominated climates and decreases in the heating dominated climates by about 4 %. Note, that the simulations in this section use a duct leakage rate of 10 %, which is assumed to be a "no fault" installation condition. For scenario (2) - no change in duct size -, the increased fan power (while working against increased static pressure) and fan heat added to the load are the main factors contributing to the significant increase in energy used in cooling-dominated climates (Houston, Las Vegas, Washington, DC).

Tables US-20. Indoor airflow information for heat pump sizing scenario (1) and scenario (2)

a) SI units	Heat Pump Sizing (%)	Fan Speed (%)	Normalized Airflow ($m^3 \cdot h^{-1} \cdot kW^{-1}$)	Normalized Fan Power ($kJ \cdot m^{-3}$)	Static Pressure (Pa)	Duct Leakage (%)	
						Supply	Return
Scenario (1): Duct size changes proportionally with HP size	80	100	181.1	1.06	167	6	4
	100	100	181.1	1.06	167	6	4
	125	100	181.1	1.06	167	6	4
	150	100	181.1	1.06	167	6	4
	175	100	181.1	1.06	167	6	4
	200	100	181.1	1.06	167	6	4
Scenario (2): Duct size stays the same as HP size changes	80	90	202.4	0.80	137	5.4	3.6
	100	100	181.1	1.06	167	6.0	4.0
	125	115	168.1	1.45	224	7.0	4.6
	150	120	145.5	1.68	249	7.3	4.9
	175	125	130.9	1.90	274	7.7	5.1
	200	130	120.8	2.11	299	8.0	5.4

b) I-P units	Heat Pump Sizing (%)	Fan Speed (%)	Normalized Airflow (cfm/ton)	Normalized Fan Power ($W \cdot cfm^{-1}$)	Static Pressure (inch)	Duct Leakage (%)	
						Supply	Return
Scenario (1): Duct size changes proportionally with HP size	80	100	375	0.50	0.76	6	4
	100	100	375	0.50	0.76	6	4
	125	100	375	0.50	0.76	6	4
	150	100	375	0.50	0.76	6	4
	175	100	375	0.50	0.76	6	4
	200	100	375	0.50	0.76	6	4
Scenario (2): Duct size stays the same as HP size changes	80	90	419	0.38	0.55	5.4	3.6
	100	100	375	0.50	0.67	6.0	4.0
	125	115	348	0.68	0.90	7.0	4.6
	150	120	301	0.79	1.00	7.3	4.9
	175	125	271	0.89	1.10	7.7	5.1
	200	130	250	0.99	1.20	8.0	5.4

Table US-22. Effect of 100 % unit oversizing on annual energy use for a slab-on-grade house for scenario (1) and scenario (2)

Scenario (1): Duct size changes proportionally with HP size	Cooling COP (%)	Cooling Load (%)	Cooling Energy (%)	Heating Energy (%)	Total Energy (%)
Houston	-2.0	1.2	3.3	-4.1	0.9
Las Vegas	-2.5	-0.6	1.9	-3.3	0.1
Washington	-1.9	0.3	2.2	-7.9	-3.6
Chicago	-1.8	0.0	1.8	-8.9	-4.6
Minneapolis	-1.7	0.2	2.0	-8.6	-4.3
Scenario (2): Duct size stays the same as HP size changes	Cooling COP (%)	Cooling Load (%)	Cooling Energy (%)	Heating Energy (%)	Total Energy (%)
Houston	-10.3	9.6	22.2	-0.6	24.2
Las Vegas	-11.9	5.6	19.8	2.2	21.7
Washington	-10.3	9.6	22.1	-10.9	8.0
Chicago	-10.2	10.2	22.7	-13..5	2.1
Minneapolis	-10.2	10.8	23.4	-14.2	-0.9

Tables US-22 and US-23 show in detail the effect of heat pump sizing on the total energy performance for scenario (1). The impact of oversizing is modest for the house with the basement (Table US-23) since the ducts are in the conditioned space. In this case, oversizing increases cooling energy because of efficiency losses from cyclic degradation, therefore overall energy use in cooling-dominated locations such as Houston and Las Vegas increases. In the heating-dominated climates, such as Chicago, the heating energy is affected by cyclic degradation as well; however, the larger heat pump meets more of the heating load, which reduces the need for backup heating. The net effect is a slight decrease in overall energy use.

For the slab-on-grade house (Table US-22) the impact of duct leakage further complicates the situation. In addition to the factors discussed for the house with the basement, oversized heat pumps have reduced runtimes, which reduce duct losses and result in a less energy being used than by the baseline system. Combining all effects, the net impact on energy use in Houston and Las Vegas is neutral. In Chicago, oversizing actually reduces energy use by as much as 5 % for the slab-on-grade house.

Tables US-24 and US-28 show in detail the effect of sizing on the total performance for scenario (2), and Figures US-21 and US-22 show relative energy input for the slab-on-grade house and house with a basement, respectively. The indoor fan power changes associated with heat pump sizing have proportionally bigger impact in the basement house then the slab-on-grade house since the cooling loads are smaller. In heating, the added fan power from oversizing in the basement house attenuates the drop in heating energy. The houses located in cooling dominated climates use less energy when the heat pump is undersized because the heat pump does not handle all the cooling load (the indoor temperature increases on hot days). For the heating dominated climates, the energy use is increased because of the significantly increased use of the resistant heater.

Table US-22. Effect of heat pump sizing on annual energy use for a slab-on-grade house with duct sized to match heat pump size (scenario (1))

Houston	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	1,521	27.7	2,401	918	15.1	3,319.0	4.4	0.784	16,078	8,710	5,377	30,897	$730	98%
Normal	1,512	26.6	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
Oversized 25 %	1,527	25.6	1,606	597	1.0	2,202.5	4.3	0.785	16,901	8,369	5,575	31,577	$746	100%
Oversized 50 %	1,544	25.3	1,347	493	0.3	1,840.0	4.3	0.784	17,012	8,283	5,589	31,616	$746	101%
Oversized 75 %	1,561	25.1	1,162	420	0.2	1,581.6	4.3	0.784	17,119	8,232	5,605	31,687	$748	101%
Oversized 100 %	1,587	25.1	1,022	365	0.2	1,387.3	4.3	0.785	17,213	8,191	5,618	31,754	$750	101%

Las Vegas	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	-	28.5	2,376	1,095	2.4	3,470.2	3.8	0.999	19,716	11,448	6,559	38,455	$1,346	98%
Normal	-	27.0	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
Oversized 25 %	-	25.6	1,587	680	0.3	2,267.3	3.7	0.999	20,758	11,070	6,696	39,256	$1,374	100%
Oversized 50 %	-	25.2	1,326	562	0.3	1,887.8	3.7	0.999	20,806	10,983	6,690	39,210	$1,372	100%
Oversized 75 %	-	25.1	1,140	479	0.2	1,618.7	3.7	0.999	20,863	10,927	6,692	39,215	$1,373	100%
Oversized 100 %	-	25.1	1,000	417	0.2	1,417.4	3.6	1.000	20,926	10,877	6,697	39,232	$1,373	100%

Washington, DC	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	238	25.9	1,480	2,330	147.7	3,810.1	4.5	0.809	7,909	22,789	5,144	36,573	$1,432	102%
Normal	253	25.1	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
Oversized 25 %	276	24.3	974	1,633	46.6	2,606.4	4.4	0.808	8,178	20,958	5,498	35,365	$1,385	98%
Oversized 50 %	280	24.0	815	1,385	22.3	2,199.9	4.4	0.809	8,216	20,487	5,568	35,004	$1,371	97%
Oversized 75 %	287	24.0	701	1,197	9.0	1,898.1	4.4	0.809	8,251	20,223	5,605	34,811	$1,363	97%
Oversized 100 %	303	23.9	616	1,049	1.8	1,664.9	4.4	0.809	8,280	20,044	5,619	34,674	$1,358	96%

Chicago	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	189	26.0	1,269	3,178	421.6	4,446.0	4.6	0.827	6,690	39,279	6,002	52,703	$1,874	103%
Normal	189	25.0	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
Oversized 25 %	193	24.2	830	2,430	178.3	3,259.1	4.5	0.827	6,863	35,494	6,875	49,963	$1,776	98%
Oversized 50 %	193	24.0	694	2,120	112.0	2,813.7	4.5	0.827	6,892	34,615	7,122	49,361	$1,755	96%
Oversized 75 %	190	24.0	597	1,860	76.2	2,456.5	4.4	0.827	6,916	34,162	7,254	49,065	$1,745	96%
Oversized 100 %	197	23.9	524	1,648	53.4	2,171.7	4.4	0.827	6,941	33,826	7,329	48,828	$1,736	95%

Minneapolis	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	12	26.1	1,107	3,727	827.2	4,833.6	4.5	0.847	5,819	58,359	6,525	71,436	$2,143	103%
Normal	13	25.2	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
Oversized 25 %	15	24.3	722	3,066	432.9	3,788.6	4.5	0.846	5,958	52,707	7,992	67,388	$2,022	98%
Oversized 50 %	15	24.1	604	2,735	322.0	3,339.1	4.5	0.846	5,981	51,531	8,452	66,696	$2,001	97%
Oversized 75 %	15	24.0	521	2,440	253.2	2,960.7	4.4	0.847	6,012	50,862	8,743	66,349	$1,990	96%
Oversized 100 %	16	23.9	457	2,195	203.5	2,651.8	4.4	0.848	6,028	50,388	8,950	66,098	$1,983	96%

Table US-23. Effect of heat pump sizing on annual energy use for a house with basement with duct sized to match heat pump size (scenario (1))

Las Vegas	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	-	26.0	1,908	898	0.3	2,806.8	3.7	1.000	15,839	9,411	5,305	31,287	$1,095	99%
Normal	-	25.3	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
Oversized 25 %	-	25.2	1,260	574	0.2	1,834.1	3.6	1.000	16,354	9,413	5,416	31,915	$1,117	101%
Oversized 50 %	-	25.1	1,064	479	0.2	1,542.6	3.6	1.000	16,568	9,430	5,467	32,196	$1,127	102%
Oversized 75 %	-	25.0	920	411	0.2	1,331.1	3.6	1.000	16,735	9,434	5,503	32,404	$1,134	103%
Oversized 100 %	-	24.9	812	360	0.2	1,171.9	3.6	1.000	16,871	9,452	5,537	32,592	$1,141	103%

Washington, DC	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	647	24.2	914	2,294	62.1	3,208.5	4.4	0.773	4,930	19,645	4,331	29,638	$1,161	101%
Normal	654	24.0	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
Oversized 25 %	666	23.9	603	1,562	6.5	2,164.2	4.4	0.776	5,085	18,867	4,565	29,249	$1,146	100%
Oversized 50 %	669	23.8	507	1,313	0.5	1,820.1	4.3	0.778	5,142	18,855	4,607	29,336	$1,149	100%
Oversized 75 %	677	23.8	439	1,128	0.1	1,566.8	4.3	0.779	5,196	18,899	4,627	29,454	$1,154	100%
Oversized 100 %	694	23.7	387	986	0.1	1,373.4	4.3	0.780	5,236	18,897	4,635	29,500	$1,155	101%

Chicago	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	276	24.2	777	3,199	240.2	3,976.0	4.5	0.795	4,131	33,239	5,368	43,470	$1,546	103%
Normal	289	24.0	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
Oversized 25 %	287	23.9	512	2,351	60.6	2,862.8	4.4	0.799	4,259	30,692	6,039	41,721	$1,483	99%
Oversized 50 %	285	23.8	431	2,010	31.1	2,441.1	4.4	0.800	4,308	30,474	6,179	41,693	$1,482	99%
Oversized 75 %	285	23.8	373	1,745	15.4	2,117.8	4.4	0.801	4,344	30,363	6,254	41,693	$1,482	99%
Oversized 100 %	292	23.7	328	1,541	6.5	1,868.8	4.4	0.803	4,373	30,362	6,307	41,774	$1,485	99%

Minneapolis	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	61	24.3	635	3,793	551.4	4,428.1	4.4	0.809	3,375	49,371	5,978	59,456	$1,784	104%
Normal	61	24.0	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
Oversized 25 %	64	24.0	418	2,984	219.1	3,401.9	4.4	0.812	3,480	44,581	7,176	55,969	$1,679	98%
Oversized 50 %	66	23.9	352	2,602	147.8	2,954.8	4.4	0.814	3,520	43,888	7,479	55,619	$1,669	97%
Oversized 75 %	70	23.8	304	2,299	99.1	2,603.2	4.4	0.816	3,548	43,438	7,688	55,405	$1,662	97%
Oversized 100 %	70	23.7	268	2,057	65.8	2,324.7	4.4	0.817	3,570	43,260	7,846	55,408	$1,662	97%

Table US-24. Effect of heat pump sizing on annual energy use for a slab-on-grade house with fixed duct size (scenario (2)

Houston	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	1,692	27.6	2,317	906	14.4	3,223.4	4.5	0.790	15,621	8,583	4,292	29,227	$690	93%
Normal	1,512	26.6	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
Oversized 25 %	1,443	25.8	1,687	605	1.0	2,292.1	4.3	0.780	17,709	8,481	7,330	34,252	$809	109%
Oversized 50 %	1,320	25.3	1,492	505	0.3	1,996.8	4.1	0.774	18,713	8,470	8,147	36,062	$851	115%
Oversized 75 %	1,244	25.2	1,343	433	0.2	1,775.8	4.0	0.769	19,587	8,482	8,832	37,634	$889	120%
Oversized 100 %	1,205	25.1	1,224	379	0.2	1,602.7	3.9	0.766	20,351	8,483	9,513	39,078	$923	124%

Las Vegas	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	-	28.3	2,303	1,109	3.0	3,411.4	3.9	1.000	19,197	11,620	5,060	36,608	$1,281	93%
Normal	-	27.0	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
Oversized 25 %	-	25.9	1,658	708	0.3	2,366.3	3.6	0.999	21,651	11,509	8,447	42,339	$1,482	108%
Oversized 50 %	-	25.2	1,461	588	0.3	2,048.9	3.5	0.997	22,773	11,483	9,346	44,335	$1,552	113%
Oversized 75 %	-	25.2	1,311	505	0.2	1,815.7	3.4	0.995	23,742	11,495	10,108	46,077	$1,613	118%
Oversized 100 %	-	25.2	1,194	442	0.2	1,635.9	3.3	0.993	24,605	11,504	10,878	47,719	$1,670	122%

Washington, DC	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	326	25.7	1,424	2,266	138.8	3,689.5	4.6	0.816	7,658	22,274	4,795	35,459	$1,389	99%
Normal	253	25.1	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
Oversized 25 %	216	24.4	1,025	1,604	42.9	2,628.8	4.4	0.804	8,581	20,448	7,423	37,184	$1,456	103%
Oversized 50 %	134	24.1	907	1,363	17.4	2,269.7	4.2	0.796	9,077	19,830	8,047	37,685	$1,476	105%
Oversized 75 %	89	24.1	817	1,181	5.6	1,998.2	4.1	0.790	9,511	19,567	8,464	38,273	$1,499	106%
Oversized 100 %	59	24.0	745	1,036	0.7	1,781.8	4.0	0.785	9,892	19,396	8,822	38,841	$1,521	108%

Chicago	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	187	25.8	1,215	3,123	399.8	4,338.1	4.6	0.836	6,451	38,633	5,807	51,622	$1,835	101%
Normal	189	25.0	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
Oversized 25 %	192	24.3	874	2,398	164.7	3,272.0	4.4	0.821	7,209	34,553	9,278	51,772	$1,841	101%
Oversized 50 %	183	24.1	777	2,086	97.3	2,862.8	4.3	0.811	7,657	33,246	10,145	51,779	$1,841	101%
Oversized 75 %	183	24.0	700	1,829	63.6	2,529.5	4.1	0.803	8,033	32,604	10,640	52,008	$1,849	102%
Oversized 100 %	173	24.0	640	1,623	40.8	2,262.7	4.1	0.798	8,365	32,100	11,041	52,238	$1,857	102%

Minneapolis	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	16	26.0	1,059	3,673	797.0	4,732.5	4.6	0.857	5,604	57,677	6,432	70,445	$2,113	102%
Normal	13	25.2	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
Oversized 25 %	13	24.5	761	3,033	408.9	3,793.4	4.4	0.840	6,255	51,321	10,781	69,089	$2,073	100%
Oversized 50 %	11	24.1	677	2,694	293.2	3,371.5	4.3	0.828	6,659	49,336	11,945	68,672	$2,060	99%
Oversized 75 %	8	24.0	612	2,403	224.3	3,014.5	4.1	0.820	7,000	48,228	12,630	68,589	$2,058	99%
Oversized 100 %	5	24.0	560	2,161	171.5	2,720.2	4.0	0.814	7,295	47,255	13,164	68,445	$2,053	99%

Table US-25. Effect of heat pump sizing on annual energy use for a house with basement with fixed duct size (scenario (2)

Las Vegas	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	-	25.8	1,835	904	0.3	2,739.1	3.8	1.000	15,274	9,464	4,062	29,531	$1,034	93%
Normal	-	25.3	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
Oversized 25 %	-	25.2	1,313	591	0.3	1,903.5	3.6	1.000	17,007	9,695	6,773	34,207	$1,197	108%
Oversized 50 %	-	25.1	1,167	497	0.2	1,664.2	3.4	1.000	18,072	9,796	7,564	36,163	$1,266	114%
Oversized 75 %	-	25.1	1,054	429	0.2	1,483.1	3.3	0.999	18,965	9,870	8,223	37,790	$1,323	120%
Oversized 100 %	-	25.0	963	377	0.2	1,340.7	3.2	0.997	19,736	9,918	8,875	39,260	$1,374	124%

Washington, DC	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	863	24.0	873	2,212	53.8	3,085.1	4.5	0.779	4,734	18,958	4,126	28,550	$1,118	97%
Normal	654	24.0	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
Oversized 25 %	514	24.0	630	1,535	4.6	2,165.2	4.3	0.775	5,301	18,466	6,135	30,634	$1,200	104%
Oversized 50 %	359	23.9	563	1,302	0.5	1,865.1	4.1	0.768	5,663	18,592	6,610	31,596	$1,238	108%
Oversized 75 %	263	23.9	509	1,128	0.1	1,637.6	4.0	0.764	5,965	18,725	6,900	32,323	$1,266	110%
Oversized 100 %	216	23.8	467	995	0.1	1,461.3	3.9	0.761	6,231	18,818	7,159	32,940	$1,290	112%

Chicago	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	318	24.1	741	3,119	213.1	3,860.3	4.5	0.804	3,963	32,140	5,292	42,126	$1,498	100%
Normal	289	24.0	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
Oversized 25 %	263	24.0	536	2,313	54.7	2,848.6	4.3	0.795	4,446	29,982	8,099	43,258	$1,538	102%
Oversized 50 %	237	23.9	480	1,991	28.8	2,470.1	4.2	0.787	4,758	29,840	8,751	44,080	$1,567	104%
Oversized 75 %	224	23.8	435	1,745	13.8	2,179.5	4.1	0.780	5,019	29,858	9,130	44,738	$1,591	106%
Oversized 100 %	213	23.8	398	1,550	6.1	1,948.1	4.0	0.776	5,243	29,944	9,426	45,345	$1,612	107%

Minneapolis	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Undersized 20 %	72	24.2	604	3,724	505.2	4,327.8	4.5	0.818	3,230	47,917	6,016	57,894	$1,737	101%
Normal	61	24.0	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
Oversized 25 %	58	24.0	438	2,943	204.3	3,381.6	4.3	0.808	3,636	43,494	9,633	57,495	$1,725	101%
Oversized 50 %	54	24.0	392	2,580	139.4	2,972.0	4.2	0.798	3,894	42,806	10,527	57,959	$1,739	102%
Oversized 75 %	50	23.9	356	2,297	93.9	2,653.1	4.1	0.791	4,110	42,474	11,075	58,390	$1,752	102%
Oversized 100 %	49	23.8	327	2,067	62.8	2,393.8	4.0	0.786	4,300	42,343	11,497	58,871	$1,766	103%

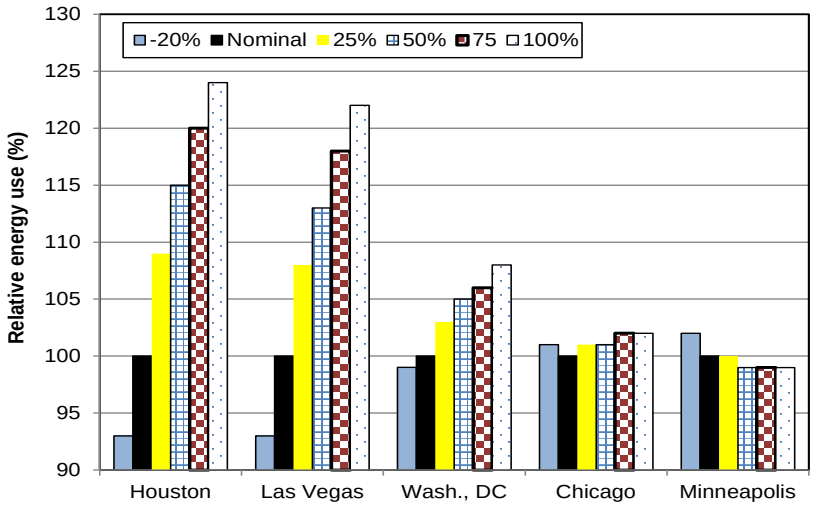


Figure US-21. Annual energy use for slab-on-grade houses for different heat pump sizings, scenario (2)

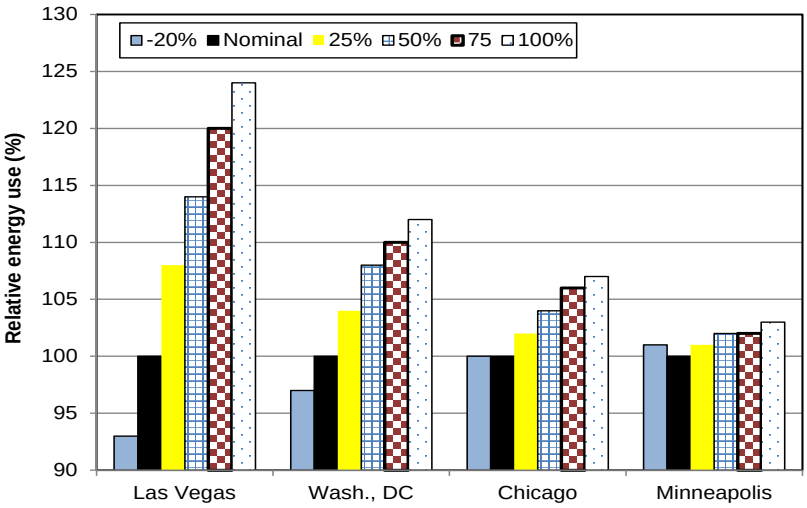


Figure US-22. Annual energy use for houses with basement for different heat pump sizings, scenario (2)

5.2.3. Effect of Duct Leakage

Per the earlier discussion in Section 4.3.2, the effect of duct leakage has been evaluated only for slab-on-grade houses where ducts were installed in the attic (i.e., in the unconditioned space). The baseline houses include ducts in the attic with a leakage rate of 10 % (leakage distributed 60 % on the supply side and 40 % on the return side) as well as thermal losses through the duct wall. Table US-26 compares this base case to other levels of duct leakage with the thermostat set at the default set point temperature (Table US-16). The entry "0 % & No thermal" in the left most column denotes an idealistic installation with zero air leakage and no thermal loss (i.e., an insulation with an infinite R). For all other simulation cases, the duct insulation is assumed to be R(SI)-1.1 (R-6).

As expected, the baseline duct losses increase energy use in the baseline houses; our simulations showed a 20 % and 30 % increase for the cooling climates and heating climates, respectively, compared to the 0 % leak case. As the duct leakage increases, energy use increases by at least 8 % for the cooling climates and by 12 % for the heating climates for each 10 % increment in the duct leakage fault. A slight improvement of the cooling COP shown with the increasing fault level is caused by a somewhat higher refrigerant saturation temperature (and pressure) in the evaporator when the air returning to the indoor section is at higher temperature due to duct losses. This COP improvement, however, can't compensate for the significant increase in the cooling load, which is the cause of the increased energy use.

Table US-27 shows the effect of duct leakage on annual energy use for the slab-on-grade house from lowering the cooling set point by 1.1 °C (2.0 °F). For completeness, the table includes all studied locations, although houses in Houston and Washington, DC are most likely to be operated at a lowered set point temperature to improve the indoor comfort. Table US-28 shows simulation results for the indoor set point temperature lowered by an additional 1.1 °C (2.0 °F), i.e., by 2.2 °C (4.0 °F) below the default value for the house in Houston.

Reducing the set point results in a lower number of hours with relative humidity above 55 % for small levels of duct leaks only (Tables US-27 and US-28). For large levels of duct leakage, the number of hours with relative humidity above 55 % actually increases. This result is caused by the fact that lowering the set point requires longer operational runtimes (with correspondingly higher energy consumption and duct leakage) and, depending on the ratio of sensible to latent capacities, lowering the indoor temperature may actually increase the relative humidity, although the indoor comfort might improve due to a lower dry-bulb temperature.

Table US-26. Effect of duct leakage on annual energy use for a slab-on-grade house at default cooling set point

Houston	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	1,715	1,555	588	0.3	2,142.9	4.3	0.789	13,007	6,623	4,339	24,700	$583	79%
0 % Leak	1,537	1,794	685	2.1	2,479.0	4.3	0.812	15,046	7,761	5,020	28,559	$674	91%
10 % Leak	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
20 % Leak	1,632	2,160	815	9.4	2,975.1	4.4	0.767	18,179	9,383	6,025	34,317	$810	109%
30 % Leak	1,922	2,327	883	17.5	3,209.7	4.5	0.753	19,574	10,393	6,500	37,198	$878	118%
40 % Leak	2,738	2,489	953	35.5	3,441.7	4.5	0.743	20,922	11,773	6,970	40,397	$954	128%
50 % Leak	3,364	2,649	1,032	61.8	3,681.0	4.6	0.734	22,231	13,578	7,454	43,995	$1,039	140%

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	-	1,536	668	0.3	2,204.5	3.7	1.000	15,941	8,763	5,207	30,642	$1,072	78%
0 % Leak	-	1,817	786	0.3	2,602.5	3.7	1.000	18,952	10,273	6,147	36,104	$1,264	92%
10 % Leak	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
20 % Leak	-	2,114	951	1.2	3,065.4	3.8	0.998	22,081	12,339	7,241	42,393	$1,484	108%
30 % Leak	-	2,261	1,054	3.7	3,315.3	3.8	0.998	23,580	13,718	7,831	45,861	$1,605	117%
40 % Leak	-	2,405	1,170	8.6	3,575.4	3.9	0.997	25,028	15,353	8,445	49,558	$1,735	126%
50 % Leak	-	2,549	1,290	22.7	3,838.7	3.9	0.996	26,444	17,362	9,067	53,605	$1,876	137%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	280	944	1,532	12.9	2,476.3	4.4	0.801	6,301	15,111	4,179	26,322	$1,031	73%
0 % Leak	175	1,100	1,803	54.5	2,902.7	4.4	0.823	7,361	19,093	4,898	32,084	$1,257	89%
10 % Leak	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
20 % Leak	368	1,314	2,133	134.8	3,446.8	4.5	0.799	8,825	24,760	5,817	40,133	$1,572	112%
30 % Leak	523	1,419	2,294	192.5	3,712.5	4.6	0.791	9,528	28,180	6,265	44,704	$1,751	124%
40 % Leak	814	1,523	2,457	270.0	3,979.2	4.6	0.786	10,216	32,335	6,715	49,997	$1,958	139%
50 % Leak	1,165	1,625	2,595	382.3	4,219.9	4.7	0.781	10,884	37,541	7,121	56,278	$2,204	157%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	203	815	2,288	70.0	3,103.7	4.5	0.819	5,369	24,753	5,238	36,092	$1,283	71%
0 % Leak	190	943	2,639	187.4	3,582.0	4.5	0.839	6,217	32,197	6,045	45,190	$1,607	88%
10 % Leak	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
20 % Leak	192	1,119	3,007	394.4	4,125.5	4.6	0.818	7,410	42,561	6,962	57,664	$2,050	113%
30 % Leak	220	1,208	3,150	532.6	4,358.0	4.6	0.812	8,003	48,636	7,354	64,725	$2,301	126%
40 % Leak	310	1,296	3,285	697.0	4,581.3	4.7	0.806	8,591	55,589	7,731	72,642	$2,583	142%
50 % Leak	427	1,386	3,408	900.9	4,793.8	4.7	0.801	9,174	63,893	8,090	81,888	$2,912	160%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	15	711	2,902	216.9	3,613.5	4.5	0.838	4,670	36,410	6,098	47,909	$1,437	69%
0 % Leak	13	822	3,258	443.5	4,079.8	4.4	0.856	5,407	47,766	6,885	60,789	$1,824	88%
10 % Leak	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
20 % Leak	15	973	3,577	802.2	4,550.0	4.5	0.839	6,421	62,936	7,678	77,767	$2,333	113%
30 % Leak	27	1,050	3,698	1,009.5	4,748.5	4.6	0.833	6,937	71,179	8,013	86,861	$2,606	126%
40 % Leak	48	1,127	3,816	1,234.7	4,942.6	4.6	0.829	7,444	80,060	8,341	96,576	$2,897	140%
50 % Leak	89	1,207	3,946	1,483.7	5,152.5	4.7	0.825	7,964	89,955	8,695	107,345	$3,220	155%

Note: All simulation cases account for thermal losses along with leakage losses except the case denoted "0 % & No thermal".

Table US-27. Effect of duct leakage on annual energy use for a slab-on-grade house at lowered cooling set point by 1.1 °C (2.0 °F)

Houston	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	1,186	1,929	610	0.3	2,539.6	4.2	0.801	15,943	6,870	5,143	28,687	$677	79%
0 % Leak	988	2,220	710	2.1	2,930.4	4.2	0.822	18,386	8,042	5,934	33,093	$781	91%
10 % Leak	1,035	2,451	777	5.1	3,227.6	4.3	0.792	20,333	8,844	6,536	36,445	$861	100%
20 % Leak	1,213	2,663	845	9.5	3,508.7	4.4	0.772	22,105	9,724	7,105	39,666	$937	109%
30 % Leak	1,867	2,858	915	18.0	3,773.2	4.5	0.757	23,717	10,759	7,641	42,848	$1,012	118%
40 % Leak	2,851	3,051	989	36.0	4,040.3	4.5	0.746	25,288	12,191	8,182	46,392	$1,095	127%
50 % Leak	3,336	3,237	1,069	63.5	4,306.1	4.6	0.736	26,785	14,046	8,720	50,283	$1,187	138%

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	-	1,788	684	0.3	2,472.5	3.7	1.000	18,346	8,965	5,840	33,883	$1,186	78%
0 % Leak	-	2,114	805	0.3	2,918.3	3.7	1.000	21,779	10,517	6,893	39,920	$1,397	92%
10 % Leak	-	2,280	884	0.3	3,164.2	3.7	0.999	23,494	11,496	7,474	43,196	$1,512	100%
20 % Leak	-	2,444	973	1.2	3,416.7	3.8	0.998	25,155	12,625	8,070	46,581	$1,630	108%
30 % Leak	-	2,603	1,079	3.7	3,681.6	3.8	0.997	26,742	14,031	8,696	50,201	$1,757	116%
40 % Leak	-	2,760	1,198	8.8	3,957.3	3.9	0.996	28,275	15,712	9,347	54,067	$1,892	125%
50 % Leak	-	2,917	1,323	22.6	4,239.9	3.9	0.995	29,786	17,787	10,015	58,319	$2,041	135%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	157	1,171	1,554	13.0	2,725.8	4.4	0.813	7,717	15,317	4,600	28,365	$1,111	74%
0 % Leak	65	1,364	1,831	54.2	3,195.0	4.4	0.835	9,008	19,345	5,392	34,477	$1,350	89%
10 % Leak	158	1,499	2,001	89.0	3,500.4	4.5	0.818	9,918	22,035	5,907	38,592	$1,512	100%
20 % Leak	301	1,632	2,170	134.5	3,802.2	4.5	0.806	10,802	25,092	6,416	43,042	$1,686	112%
30 % Leak	563	1,758	2,331	192.6	4,089.5	4.6	0.797	11,632	28,528	6,901	47,793	$1,872	124%
40 % Leak	1,015	1,883	2,500	270.1	4,383.0	4.6	0.791	12,442	32,734	7,396	53,304	$2,088	138%
50 % Leak	1,311	2,008	2,647	382.6	4,654.5	4.7	0.785	13,246	38,021	7,854	59,853	$2,344	155%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	182	1,002	2,303	69.9	3,304.9	4.4	0.828	6,521	24,884	5,577	37,714	$1,341	71%
0 % Leak	173	1,159	2,657	187.4	3,815.3	4.4	0.847	7,554	32,354	6,438	47,078	$1,674	88%
10 % Leak	176	1,267	2,849	281.2	4,115.8	4.5	0.833	8,277	37,266	6,945	53,220	$1,892	100%
20 % Leak	175	1,375	3,024	394.4	4,398.2	4.5	0.823	8,994	42,715	7,422	59,863	$2,128	112%
30 % Leak	246	1,483	3,169	533.5	4,651.8	4.6	0.815	9,705	48,830	7,850	67,117	$2,386	126%
40 % Leak	365	1,591	3,311	697.0	4,901.8	4.6	0.809	10,407	55,823	8,272	75,233	$2,675	141%
50 % Leak	498	1,699	3,438	901.3	5,136.4	4.7	0.803	11,098	64,171	8,668	84,668	$3,010	159%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	8	884	2,919	216.9	3,802.3	4.4	0.845	5,730	36,561	6,416	49,439	$1,483	70%
0 % Leak	4	1,021	3,276	443.2	4,297.0	4.4	0.864	6,627	47,932	7,251	62,541	$1,876	88%
10 % Leak	6	1,114	3,449	612.5	4,563.3	4.5	0.852	7,251	55,263	7,701	70,946	$2,128	100%
20 % Leak	7	1,209	3,598	802.2	4,807.8	4.5	0.843	7,883	63,128	8,113	79,855	$2,396	113%
30 % Leak	8	1,304	3,724	1,009.6	5,028.1	4.6	0.836	8,501	71,417	8,485	89,134	$2,674	126%
40 % Leak	48	1,399	3,845	1,234.7	5,244.2	4.6	0.831	9,116	80,330	8,850	99,027	$2,971	140%
50 % Leak	129	1,497	3,979	1,484.0	5,476.5	4.7	0.826	9,745	90,267	9,242	109,985	$3,300	155%

Figures US-23 and US-24 present the number of hours above 55 % relative humidity and relative energy use, respectively, for a slab-on-grade house in Houston with different duct leak rates at the three studied thermostat set point temperatures. The energy use is related to that of a house with 10 % leak rate (assumed as a representative no-fault duct installation) at the default thermostat set point (Table US-26). At a leak rate greater than 20 %, the heat pump was unable to lower the number of hours above 55 % relative humidity although the amount of moisture in the air was lowered and a lower indoor air temperature improved indoor thermal comfort to some degree. For the house with a 40 % duct leakage, the energy use is predicted to be 47 % and 97 % higher than for the reference house if the set point temperature is lowered by 1.1 °C and 2.2 °C, respectively (Figure US-24).

The results contained in Table US-29 (derived from Tables US-27 and US-28) present a change in the annual energy use for the baseline houses due to lowering the cooling set point. For Las Vegas, Washington, Chicago, and Minneapolis, the change in energy use is the same for the slab-on-grade house and the house with a basement. The use of energy increased by the same percentage for a slab-on-grade house and a house with a basement located in the same climate. As expected, the effect of lowering the set point temperature was small on the total energy use in houses located in heating dominated climates.

Table US-5.28. Effect of duct leakage on annual energy use for a slab-on-grade house in Houston at lowered cooling set point by 2.2 °C (4.0 °F)

Houston	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
0 % & No thermal	536	2,759	995	0.3	3,511.5	4.3	0.841	22,498	11,221	7,602	42,053	$993	82%
0 % Leak	389	3,106	1,116	2.3	3,948.8	4.3	0.855	25,383	12,592	8,550	47,256	$1,116	93%
10 % Leak	435	3,376	1,193	5.3	4,276.4	4.4	0.825	27,602	13,468	9,253	51,054	$1,205	100%
20 % Leak	737	3,627	1,280	9.7	4,601.1	4.4	0.803	29,645	14,516	9,936	54,829	$1,295	107%
30 % Leak	1,889	3,842	1,352	18.0	4,895.4	4.5	0.786	31,384	15,547	10,519	58,182	$1,374	114%
40 % Leak	2,685	4,054	1,438	35.3	5,194.0	4.5	0.772	33,084	17,064	11,122	62,002	$1,464	121%
50 % Leak	3,197	4,250	1,517	61.9	5,475.9	4.6	0.760	34,637	18,858	11,677	65,904	$1,556	129%

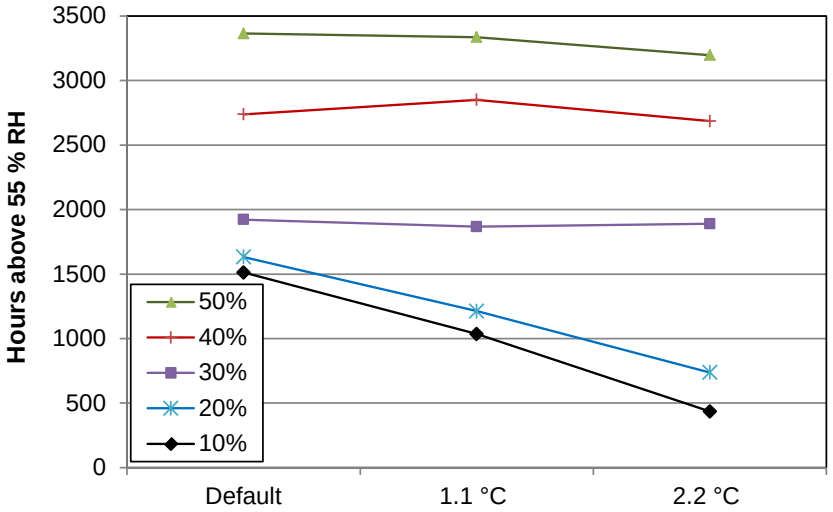


Figure US-23. Number of hours above 55 % relative humidity for a slab-on-grade house in Houston with duct leak rates from 10 % to 50 % at three thermostat set point temperatures

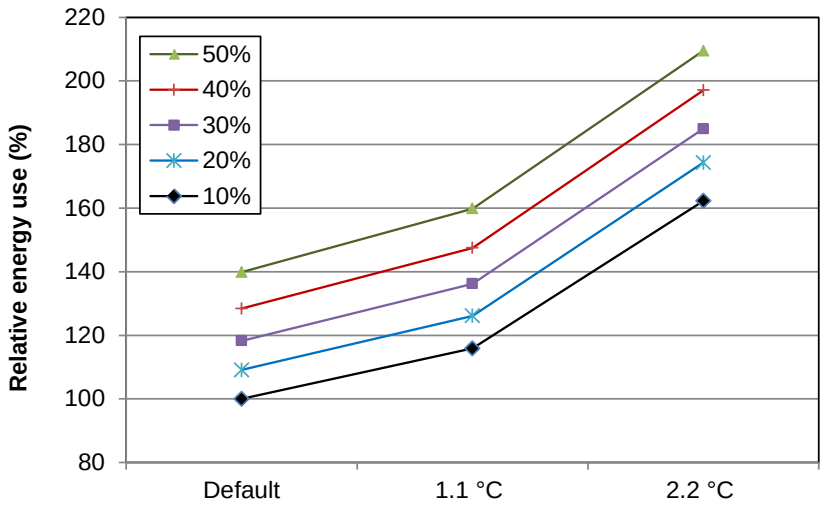


Figure US-24. Energy use for a slab-on-grade house in Houston with duct leak rates from 10 % to 50 % at three thermostat set point temperatures related to energy use for the house at the default set point and 10 % leak rate (Shown in Table US-26)

Table US-29. Effect of lowering cooling set point by 1.1 °C (2.0 °F) on annual energy use of a baseline slab-on-grade house and a house with basement

Slab-on-grade house		Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy	Cooling Energy
Houston	Normal Set Pt	1,512	26.6	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%	100%
	Lower Set Pt	1,035	26.1	2,451	777	5.1	3,227.6	4.3	0.792	20,333	8,844	6,536	36,445	$861	116%	122%
Las Vegas	Normal Set Pt	-	27.0	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%	100%
	Lower Set Pt	-	26.6	2,280	884	0.3	3,164.2	3.7	0.999	23,494	11,496	7,474	43,196	$1,512	110%	114%
Washington, DC	Normal Set Pt	253	25.1	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%	100%
	Lower Set Pt	158	24.4	1,499	2,001	89.0	3,500.4	4.5	0.818	9,918	22,035	5,907	38,592	$1,512	107%	122%
Chicago	Normal Set Pt	189	25.0	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%	100%
	lower Set Pt	176	24.4	1,267	2,849	281.2	4,115.8	4.5	0.833	8,277	37,266	6,945	53,220	$1,892	104%	121%
Minneapolis	Normal Set Pt	13	25.2	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%	100%
	lower Set Pt	6	24.5	1,114	3,449	612.5	4,563.3	4.5	0.852	7,251	55,263	7,701	70,946	$2,128	103%	123%

House with basement		Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy	Cooling Energy
Las Vegas	Normal Set Pt	-	25.3	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%	100%
	Lower Set Pt	-	24.2	1,809	731	0.3	2,539.6	3.7	1.000	18,571	9,573	5,999	34,874	$1,221	110%	115%
Washington, DC	Normal Set Pt	654	24.0	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%	100%
	Lower Set Pt	306	23.0	976	1,925	27.0	2,900.5	4.4	0.793	6,484	19,284	4,895	31,394	$1,230	107%	129%
Chicago	Normal Set Pt	289	24.0	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%	100%
	lower Set Pt	160	23.0	826	2,796	129.7	3,621.3	4.4	0.810	5,419	31,661	6,111	43,923	$1,562	104%	129%
Minneapolis	Normal Set Pt	61	24.0	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%	100%
	lower Set Pt	43	23.0	689	3,434	354.2	4,123.5	4.4	0.824	4,520	46,328	6,958	58,538	$1,756	103%	132%

5.2.4. Effect of Indoor Coil Airflow

This fault covers the case where a heat pump properly sized for the building load operates with improperly sized ductwork. As a result, the indoor coil airflow is not nominal. The effect of improper airflow in the cooling mode was determined using the baseline performance maps for the air conditioner used in a past study because they were shown to be very close to the correlations derived from NIST lab testing (Section 3.2.1). The impact of indoor airflow on heat pump performance in the heating mode was not considered in the heat pump baseline performance maps, therefore, the NIST correlations were used to determine this impact. The simulated indoor airflows, ranging from -36 % to +28 % of the nominal flow corresponded to external static pressures of (177, 171, 168, 165, and 149) Pa ((0.71, 0.69, 0.67, 0.66, 0.60) inch H_2O), respectively.

Reduced airflow results in an increase in energy consumption, and this effect is similar for all houses in all climates studied (Tables US-30 and US-31). Figure US-25, generated for slab-on-grade houses, also provides a good representation of simulation results for houses with a basement. For the lowest airflow, 36 % below the nominal value, the energy use increased from 11 % to 14 %.

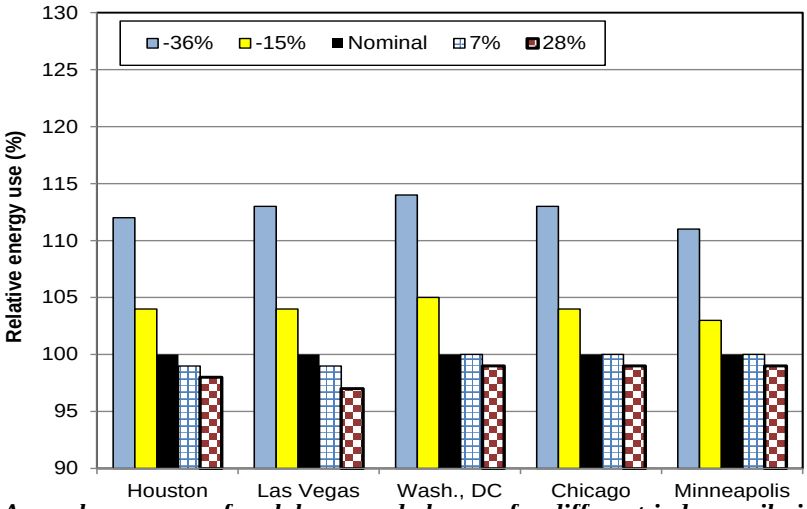


Figure US-25. Annual energy use for slab-on-grade houses for different indoor coil airflows relative to energy use for the house in the same location with nominal airflow rate

In the cooling mode, reducing the airflow below the nominal value of 50.3 $L \cdot kJ^{-1}$ (375 cfm/ton) causes a decrease in the indoor coil temperature and provides better humidity control, but results in higher energy use because the sensible capacity is reduced and running time increased. Conversely, providing more airflow hurts humidity control in the house but decreases energy use. The efficiency of the system goes up, and more importantly, the latent removal decreases so energy use decreases. To account for a possible scenario where the homeowner lowers the temperature setting on the thermostat in an effort to make the indoor environment more comfortable, Tables US-32 and US-33 provide simulation results for both houses for cases where the thermostat set point is reduced 1.1 °C (2.0 °F) below the "default" values shown in Table US-16.

Tables US-30 and US-32 show the energy usage penalties associated with lowering the airflow and reducing the thermostat set point to aid in humidity control. In Table US-30 for Houston, a hot and humid climate, the slab-on-grade house spends 1,183 hours above 55 % RH even with the airflow reduced by 36 %, resulting in a 12 % increase in annual energy usage (the total energy draw was 35,334 MJ). Keeping the airflow at the nominal value but lowering the thermostat set point by 1.1 °C (2.0 °F), as shown in Table US-32, reduces the number of hours above 55 % RH to a comparable number of hours of 1,035 while increasing the energy use by 16 % (36,445 MJ compared to 31457MJ). Reduced airflow or lowered cooling set point in other climates - in which the number of hours above 55 % was small - resulted in significant energy use penalties and a small reduction of high RH hours.

Table US-30. Effect of indoor coil airflow on annual energy use for a slab-on-grade house when operating at the default cooling set point

Houston	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	1,183	2,272	853	9.7	3,125.5	3.9	0.770	18,783	10,982	4,838	35,334	$834	112%
-15 % flow	1,364	2,074	785	6.6	2,858.7	4.2	0.780	17,332	9,405	5,331	32,800	$774	104%
nominal flow	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
7 % flow	1,617	1,951	743	4.9	2,693.9	4.4	0.787	16,455	8,465	5,609	31,262	$738	99%
28 % flow	2,026	1,878	726	4.7	2,603.3	4.5	0.793	16,080	8,259	5,727	30,798	$727	98%

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	-	2,268	1,000	1.3	3,268.2	3.3	0.992	23,192	14,547	5,902	44,373	$1,553	113%
-15 % flow	-	2,057	910	0.6	2,966.6	3.6	0.998	21,369	12,396	6,454	40,951	$1,433	104%
nominal flow	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
7 % flow	-	1,933	856	0.4	2,789.0	3.8	1.000	20,232	11,126	6,775	38,865	$1,360	99%
28 % flow	-	1,866	837	0.3	2,702.4	3.9	1.000	19,667	10,875	6,936	38,211	$1,337	97%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	26	1,409	2,175	119.1	3,583.9	4.0	0.786	9,295	26,391	4,623	41,041	$1,607	114%
-15 % flow	153	1,271	2,042	98.0	3,312.6	4.3	0.801	8,476	23,334	5,148	37,689	$1,476	105%
nominal flow	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
7 % flow	305	1,184	1,959	87.0	3,143.5	4.5	0.812	7,974	21,618	5,455	35,778	$1,401	100%
28 % flow	520	1,132	1,931	83.7	3,063.8	4.6	0.821	7,738	21,376	5,617	35,463	$1,389	99%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	160	1,217	3,046	350.6	4,263.5	4.1	0.798	7,920	43,548	5,500	57,699	$2,052	113%
-15 % flow	183	1,089	2,909	301.3	3,997.4	4.4	0.816	7,159	39,221	6,212	53,323	$1,896	104%
nominal flow	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
7 % flow	190	1,009	2,817	277.3	3,826.5	4.6	0.831	6,695	36,913	6,640	50,980	$1,813	100%
28 % flow	216	960	2,781	270.0	3,740.4	4.6	0.844	6,462	36,577	6,858	50,628	$1,800	99%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	3	1,066	3,613	727.2	4,679.7	4.1	0.813	6,917	62,805	6,037	76,491	$2,295	111%
-15 % flow	9	950	3,496	646.3	4,446.1	4.4	0.834	6,225	57,542	6,909	71,408	$2,142	103%
nominal flow	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
7 % flow	15	878	3,418	606.3	4,295.6	4.6	0.851	5,803	54,874	7,454	68,863	$2,066	100%
28 % flow	27	832	3,387	592.6	4,218.8	4.6	0.866	5,581	54,499	7,735	68,546	$2,056	99%

Table US-31. Effect of indoor coil airflow on annual energy use for a house with basement when operating at the default cooling set point

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	-	1,765	805	0.3	2,569.9	3.3	0.994	18,011	11,795	4,641	35,178	$1,231	111%
-15 % flow	-	1,616	750	0.3	2,366.2	3.5	1.000	16,707	10,302	5,148	32,889	$1,151	104%
nominal flow	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
7 % flow	-	1,529	713	0.3	2,241.9	3.7	1.000	15,890	9,341	5,446	31,409	$1,099	99%
28 % flow	-	1,478	698	0.3	2,175.8	3.8	1.000	15,443	9,145	5,585	30,905	$1,082	98%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	245	851	2,091	39.7	2,941.7	4.0	0.746	5,661	22,995	3,795	33,183	$1,300	113%
-15 % flow	463	777	1,974	30.8	2,751.8	4.3	0.765	5,217	20,502	4,276	30,727	$1,203	105%
nominal flow	653	742	1,907	27.0	2,649.5	4.4	0.775	5,007	19,120	4,471	29,330	$1,149	100%
7 % flow	743	729	1,895	25.7	2,624.5	4.4	0.779	4,933	18,999	4,554	29,218	$1,144	100%
28 % flow	1,030	699	1,861	22.7	2,560.1	4.5	0.791	4,790	18,680	4,694	28,895	$1,132	99%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	218	729	2,992	170.2	3,721.0	4.0	0.761	4,784	37,000	4,800	47,316	$1,682	112%
-15 % flow	250	663	2,862	142.6	3,524.9	4.3	0.784	4,388	33,460	5,478	44,058	$1,567	104%
nominal flow	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
7 % flow	299	620	2,768	126.9	3,387.0	4.5	0.802	4,131	31,369	5,877	42,108	$1,497	100%
28 % flow	377	590	2,727	118.4	3,317.3	4.6	0.818	3,992	30,921	6,082	41,726	$1,484	99%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	49	597	3,615	428.4	4,211.9	4.0	0.770	3,921	52,744	5,433	62,829	$1,885	110%
-15 % flow	55	542	3,497	377.7	4,039.3	4.3	0.796	3,589	48,444	6,277	59,042	$1,771	103%
nominal flow	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
7 % flow	68	506	3,410	348.1	3,915.8	4.5	0.816	3,372	46,004	6,795	56,902	$1,707	100%
28 % flow	78	482	3,373	330.3	3,854.6	4.5	0.834	3,255	45,391	7,067	56,445	$1,693	99%

Table US-32. Effect of indoor coil airflow on annual energy use for a slab-on-grade house when operating at a cooling set point that is 1.1 °C (2.0 °F) lower than the default value

Houston	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	572	2,789	884	9.8	3,672.5	3.9	0.779	22,733	11,373	5,685	40,522	$957	111%
-15 % flow	846	2,556	813	6.6	3,369.4	4.2	0.788	21,082	9,739	6,283	37,836	$893	104%
nominal flow	1,035	2,451	777	5.1	3,227.6	4.3	0.792	20,333	8,844	6,536	36,445	$861	100%
7 % flow	1,139	2,413	770	4.9	3,183.6	4.4	0.794	20,083	8,766	6,629	36,209	$855	99%
28 % flow	1,628	2,326	752	4.7	3,078.5	4.5	0.799	19,631	8,556	6,773	35,692	$843	98%

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	-	2,613	1,022	1.3	3,635.2	3.4	0.991	26,339	14,874	6,565	48,509	$1,698	112%
-15 % flow	-	2,382	931	0.6	3,312.5	3.6	0.998	24,409	12,684	7,207	45,031	$1,576	104%
nominal flow	-	2,280	884	0.3	3,164.2	3.7	0.999	23,494	11,496	7,474	43,196	$1,512	100%
7 % flow	-	2,242	874	0.4	3,116.5	3.8	0.999	23,156	11,360	7,571	42,818	$1,499	99%
28 % flow	-	2,166	855	0.3	3,021.1	3.9	1.000	22,516	11,112	7,754	42,114	$1,474	97%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	4	1,742	2,212	119.1	3,953.8	4.0	0.798	11,319	26,773	5,100	43,923	$1,720	114%
-15 % flow	58	1,576	2,075	97.8	3,650.9	4.3	0.811	10,365	23,644	5,674	40,415	$1,583	105%
nominal flow	158	1,499	2,001	89.0	3,500.4	4.5	0.818	9,918	22,035	5,907	38,592	$1,512	100%
7 % flow	203	1,473	1,989	87.4	3,461.8	4.5	0.820	9,777	21,902	6,007	38,418	$1,505	100%
28 % flow	461	1,410	1,960	83.6	3,369.8	4.6	0.828	9,487	21,640	6,178	38,036	$1,490	99%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	131	1,490	3,067	350.6	4,556.5	4.1	0.808	9,571	43,759	5,878	59,939	$2,131	113%
-15 % flow	160	1,336	2,927	301.3	4,263.4	4.3	0.824	8,683	39,398	6,625	55,439	$1,971	104%
nominal flow	176	1,267	2,849	281.2	4,115.8	4.5	0.833	8,277	37,266	6,945	53,220	$1,892	100%
7 % flow	176	1,240	2,833	277.4	4,073.6	4.5	0.837	8,131	37,061	7,068	52,992	$1,884	100%
28 % flow	199	1,183	2,799	270.0	3,981.5	4.6	0.848	7,859	36,740	7,300	52,631	$1,871	99%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	-	1,318	3,636	727.0	4,954.4	4.0	0.822	8,438	63,034	6,391	78,595	$2,358	111%
-15 % flow	1	1,178	3,515	646.2	4,693.4	4.3	0.841	7,628	57,722	7,294	73,375	$2,201	103%
nominal flow	6	1,114	3,449	612.5	4,563.3	4.5	0.852	7,251	55,263	7,701	70,946	$2,128	100%
7 % flow	6	1,091	3,436	606.3	4,526.0	4.5	0.856	7,118	55,039	7,854	70,742	$2,122	100%
28 % flow	13	1,036	3,404	592.6	4,440.5	4.6	0.869	6,853	54,661	8,141	70,387	$2,112	99%

Note: Although the relative energy use shown in this table is equal or less than the values shown in Table US-30 (baseline), the total energy use for cases presented in Table US-32 is higher than those presented in Table US-30.

Table US-33. Effect of indoor coil airflow on annual energy use for a house with basement when operating at cooling set point that is 1.1 °C (2.0 °F) lower than the default value

Las Vegas	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	-	1,765	805	0.3	2,569.9	3.3	0.994	18,011	11,795	4,641	35,178	$1,231	111%
-15 % flow	-	1,616	750	0.3	2,366.2	3.5	1.000	16,707	10,302	5,148	32,889	$1,151	104%
nominal flow	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
7 % flow	-	1,529	713	0.3	2,241.9	3.7	1.000	15,890	9,341	5,446	31,409	$1,099	99%
28 % flow	-	1,478	698	0.3	2,175.8	3.8	1.000	15,443	9,145	5,585	30,905	$1,082	98%

Washington, DC	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	67	1,118	2,111	39.9	3,229.1	3.9	0.767	7,332	23,207	4,165	35,435	$1,388	113%
-15 % flow	184	1,021	1,993	30.8	3,013.7	4.2	0.784	6,755	20,680	4,683	32,850	$1,287	105%
nominal flow	306	976	1,925	27.0	2,900.5	4.4	0.793	6,484	19,284	4,895	31,394	$1,230	100%
7 % flow	378	959	1,913	25.7	2,871.9	4.4	0.797	6,387	19,164	4,983	31,266	$1,225	100%
28 % flow	666	918	1,878	22.7	2,796.0	4.5	0.808	6,191	18,831	5,126	30,880	$1,209	98%

Chicago	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	111	952	3,004	170.2	3,956.5	4.0	0.777	6,166	37,130	5,104	49,132	$1,747	112%
-15 % flow	142	867	2,872	142.6	3,738.2	4.3	0.798	5,663	33,556	5,809	45,759	$1,627	104%
nominal flow	160	826	2,796	129.7	3,621.3	4.4	0.810	5,419	31,661	6,111	43,923	$1,562	100%
7 % flow	165	811	2,777	127.2	3,588.5	4.5	0.815	5,337	31,467	6,227	43,762	$1,556	100%
28 % flow	193	774	2,738	118.5	3,511.6	4.5	0.829	5,153	31,024	6,438	43,346	$1,541	99%

Minneapolis	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
-36 % flow	17	799	3,627	428.5	4,425.4	4.0	0.788	5,165	52,870	5,709	64,475	$1,934	110%
-15 % flow	31	726	3,508	377.7	4,233.4	4.2	0.811	4,735	48,549	6,579	60,595	$1,818	104%
nominal flow	43	689	3,434	354.2	4,123.5	4.4	0.824	4,520	46,328	6,958	58,538	$1,756	100%
7 % flow	46	677	3,421	347.9	4,097.9	4.4	0.829	4,448	46,096	7,111	58,386	$1,752	100%
28 % flow	54	644	3,383	330.3	4,027.2	4.5	0.845	4,285	45,481	7,383	57,881	$1,736	99%

Note: Although the relative energy use shown in this table is equal or less than the values shown in Table US-31 (baseline), the total energy use for cases presented in Table US-33 is higher than those presented in Table US-31.

5.2.5. Effect of Refrigerant Undercharge

When the amount of refrigerant charge in the TXV-controlled system is below the nominal value, the performance of the unit is degraded. Tables US-34 and US-35 show the results for the slab-on-grade house and the basement house, respectively. Figure US-26 shows the relative energy use for the slab-on-grade house, which provides a good representation of the energy use in the house with a basement as well. The figure indicates that the energy use increases exponentially with increasing refrigerant undercharge. For the 30 % refrigerant undercharge level the energy use increases by as much as (17 ~ 23) %. The moisture removal capacity of the unit is also degraded when the unit is undercharged.

Table US-34. Effect of refrigerant undercharge on annual energy use for a slab-on-grade house

	Under Charge	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	-10 %	1,581	2,052	778	5.8	2,830.4	4.2	0.787	17,098	8,787	5,731	32,348	$764	103%
	-20 %	1,676	2,176	855	8.5	3,031.2	4.0	0.789	17,901	9,562	6,138	34,333	$811	109%
	-30 %	1,811	2,366	1,000	20.2	3,366.3	3.8	0.792	19,131	11,284	6,817	37,963	$896	121%
Las Vegas	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	-10 %	-	2,044	900	0.4	2,944.4	3.6	1.000	21,109	11,573	6,955	40,369	$1,413	103%
	-20 %	-	2,177	1,000	1.0	3,176.8	3.5	1.000	22,133	12,652	7,504	43,021	$1,506	110%
	-30 %	-	2,379	1,199	3.7	3,578.0	3.2	1.000	23,671	14,919	8,451	47,773	$1,672	122%
Washington, DC	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	-10 %	281	1,246	2,020	91.5	3,266.2	4.4	0.811	8,304	22,133	5,512	36,680	$1,437	102%
	-20 %	312	1,317	2,168	109.9	3,485.6	4.2	0.815	8,690	23,868	5,882	39,172	$1,534	109%
	-30 %	382	1,433	2,450	154.3	3,882.8	3.9	0.819	9,319	27,533	6,552	44,135	$1,729	123%
Chicago	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	-10 %	189	1,063	2,886	286.2	3,948.6	4.4	0.830	6,984	37,586	6,663	51,964	$1,848	102%
	-20 %	193	1,123	3,035	327.6	4,158.4	4.2	0.834	7,311	40,065	7,017	55,125	$1,960	108%
	-30 %	188	1,221	3,281	433.9	4,502.1	3.9	0.841	7,842	45,504	7,597	61,674	$2,193	120%
Minneapolis	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	-10 %	15	925	3,475	621.3	4,400.0	4.4	0.850	6,056	55,608	7,425	69,821	$2,095	101%
	-20 %	15	977	3,604	687.0	4,581.0	4.2	0.855	6,342	58,734	7,730	73,538	$2,206	106%
	-30 %	15	1,062	3,804	839.1	4,866.2	3.9	0.862	6,802	65,356	8,212	81,101	$2,433	117%

Table US-35. Effect of refrigerant undercharge on annual energy use for a house with basement

	Under Charge	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	-10 %	-	1,606	745	0.3	2,350.5	3.6	1.000	16,512	9,651	5,552	32,448	$1,136	103%
	-20 %	-	1,703	817	0.3	2,520.0	3.4	1.000	17,302	10,405	5,952	34,391	$1,204	109%
	-30 %	-	1,861	966	0.3	2,827.1	3.2	1.000	18,620	12,013	6,678	38,042	$1,331	120%
Washington, DC	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	-10 %	694	762	1,954	27.5	2,715.9	4.3	0.777	5,111	19,412	4,583	29,838	$1,169	102%
	-20 %	755	800	2,099	35.1	2,898.9	4.1	0.780	5,321	20,749	4,892	31,693	$1,241	108%
	-30 %	851	863	2,384	55.2	3,246.7	3.8	0.785	5,673	23,624	5,479	35,508	$1,391	121%
Chicago	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	-10 %	294	647	2,838	131.2	3,485.1	4.3	0.800	4,283	31,904	5,881	42,800	$1,522	101%
	-20 %	295	679	2,995	156.3	3,673.9	4.2	0.804	4,453	33,893	6,200	45,277	$1,610	107%
	-30 %	304	732	3,282	221.2	4,014.3	3.9	0.810	4,749	38,240	6,774	50,495	$1,795	119%
Minneapolis	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	-10 %	65	529	3,472	357.3	4,001.0	4.3	0.813	3,497	46,578	6,752	57,559	$1,727	101%
	-20 %	68	554	3,616	403.2	4,169.8	4.1	0.818	3,636	49,132	7,036	60,536	$1,816	106%
	-30 %	69	597	3,861	518.1	4,457.6	3.9	0.825	3,871	54,824	7,522	66,950	$2,008	117%

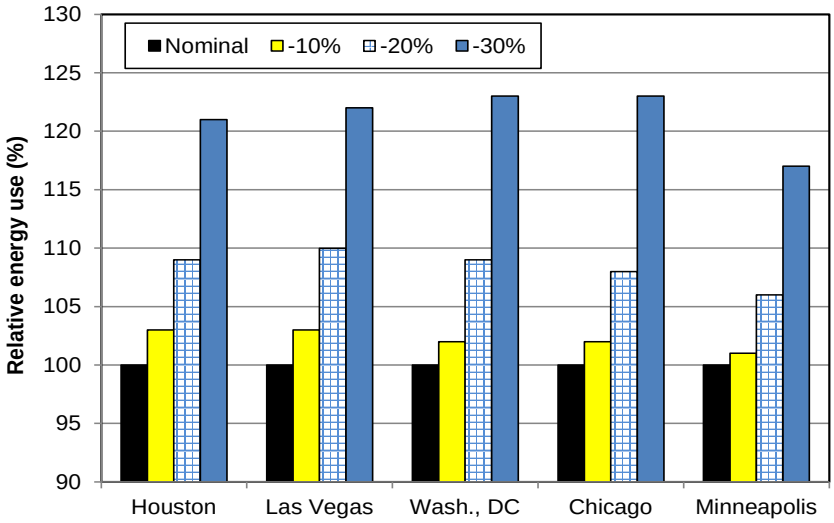


Figure US-26. Annual energy use for slab-on-grade houses at different levels of refrigerant undercharge relative to the annual energy use for the house in the same location when the heat pump operates with the nominal refrigerant charge

5.2.6. Effect of Refrigerant Overcharge

When the amount of refrigerant charge in the system is above the correct (nominal) value, the performance of the unit is degraded. Tables US-36 and US-37 show the results for the slab-on-grade house and for the basement house, respectively. The heat pump uses (10 ~ 16) % more energy when overcharged by 30 %, with somewhat higher increases in energy use occurring in localities with a significant heating season (i.e.,

Chicago, Washington DC, and Minneapolis). Figure US-27 shows the relative energy use for the slab-on-grade house, which provides a good representation of the energy use in the house with a basement as well. The moisture removal capability of the unit is not affected by the overcharge fault.

Table US-36. Effect of refrigerant overcharge on annual energy use for a slab-on-grade house

	Over Charge	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	10 %	1,553	1,955	764	5.8	2,718.5	4.3	0.786	16,743	8,912	5,505	31,891	$753	101%
	20 %	1,572	1,937	778	6.5	2,714.9	4.2	0.787	17,006	9,616	5,498	32,851	$776	104%
	30 %	1,547	1,932	796	7.3	2,728.4	4.1	0.786	17,486	10,736	5,525	34,478	$814	110%
Las Vegas	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	10 %	-	1,945	884	0.4	2,828.3	3.7	0.999	20,689	11,742	6,681	39,843	$1,394	102%
	20 %	-	1,929	904	0.6	2,833.4	3.6	0.999	21,042	12,711	6,693	41,178	$1,441	105%
	30 %	-	1,919	925	0.7	2,843.8	3.5	0.999	21,577	14,180	6,717	43,206	$1,512	110%
Washington, DC	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	10 %	277	1,191	2,004	94.3	3,194.7	4.4	0.810	8,144	22,476	5,391	36,744	$1,439	102%
	20 %	281	1,183	2,037	100.6	3,220.6	4.3	0.811	8,296	23,977	5,435	38,439	$1,506	107%
	30 %	264	1,181	2,074	106.4	3,255.3	4.2	0.809	8,544	26,260	5,493	41,029	$1,607	114%
Chicago	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	10 %	190	1,016	2,871	293.4	3,887.7	4.5	0.828	6,849	38,136	6,561	52,277	$1,859	102%
	20 %	191	1,009	2,904	308.1	3,913.7	4.4	0.829	6,976	40,241	6,604	54,552	$1,940	107%
	30 %	189	1,008	2,945	321.4	3,953.0	4.3	0.827	7,188	43,466	6,671	58,056	$2,064	113%
Minneapolis	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	10 %	15	885	3,465	633.6	4,350.4	4.5	0.848	5,945	56,322	7,341	70,339	$2,110	102%
	20 %	15	879	3,494	656.0	4,372.9	4.4	0.848	6,053	58,820	7,379	72,984	$2,190	106%
	30 %	13	878	3,528	678.6	4,405.8	4.3	0.847	6,236	62,694	7,435	77,096	$2,313	112%

Table US-37. Effect of refrigerant overcharge on annual energy use for a house with basement

	Over Charge	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	10 %	-	1,538	733	0.3	2,270.4	3.6	1.000	16,261	9,814	5,363	32,170	$1,126	102%
	20 %	-	1,527	747	0.3	2,274.2	3.6	1.000	16,565	10,587	5,372	33,256	$1,164	105%
	30 %	-	1,521	763	0.3	2,283.8	3.5	1.000	17,021	11,782	5,394	34,929	$1,223	111%
Washington, DC	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	10 %	695	734	1,940	28.9	2,674.8	4.3	0.776	5,050	19,735	4,514	30,031	$1,176	102%
	20 %	695	730	1,972	32.5	2,702.7	4.3	0.776	5,154	21,110	4,561	31,557	$1,236	108%
	30 %	658	730	2,009	35.0	2,739.6	4.1	0.775	5,317	23,264	4,623	33,936	$1,329	116%
Chicago	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	10 %	295	624	2,823	138.0	3,447.7	4.4	0.799	4,234	32,451	5,818	43,234	$1,537	102%
	20 %	294	621	2,862	146.8	3,482.6	4.3	0.798	4,319	34,408	5,877	45,335	$1,612	107%
	30 %	285	621	2,902	156.0	3,523.1	4.2	0.797	4,457	37,502	5,945	48,636	$1,729	115%
Minneapolis	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	10 %	65	510	3,462	369.4	3,971.6	4.4	0.812	3,456	47,300	6,702	58,190	$1,746	102%
	20 %	65	507	3,497	385.6	4,004.3	4.3	0.812	3,528	49,673	6,757	60,690	$1,821	106%
	30 %	62	507	3,534	402.7	4,040.4	4.2	0.810	3,640	53,431	6,818	64,621	$1,939	113%

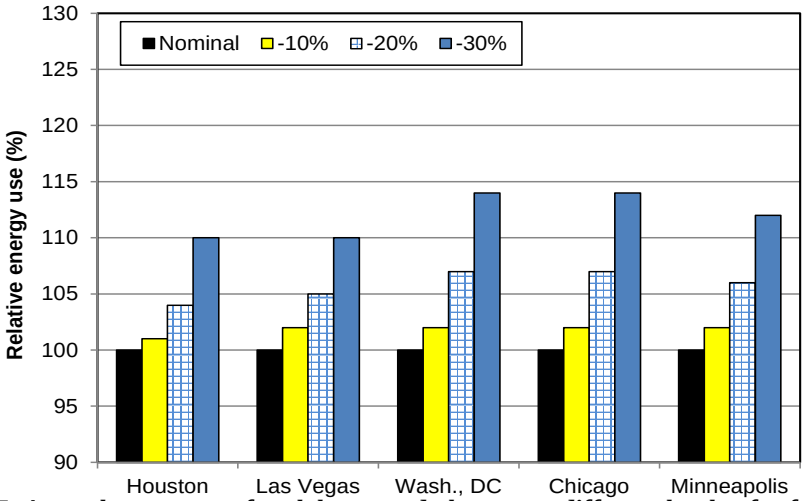


Figure US-27. Annual energy use for slab-on-grade houses at different levels of refrigerant overcharge relative to the annual energy use for the house in the same location when the heat pump operates with the nominal refrigerant charge

5.2.7. Effect of Excessive Refrigerant Subcooling

The level of this fault was determined by an increase of refrigerant subcooling at the TXV inlet at the operating condition defined by the AHRI Standard 210/240 test-A (AHRI, 2008). Refrigerant subcooling is indicative of refrigerant charge in a TXV-equipped system, and excessive subcooling is equivalent to the fault of refrigerant overcharge. When the amount of subcooling at the TXV inlet is increased the cooling system performance is degraded. Table US-38 shows the results for the slab-on-grade house, and Table US-39 shows the results for the basement house. Figure US-28 shows the relative energy use for the slab-on-grade house, which provides a good representation of the energy use in the house with a basement as well. In general, increasing subcooling increases the capacity of the unit but degrades its efficiency. Both the cooling and heating energy use increased by about 20 % at the maximum fault level (200 %, i.e., an increase of subcooling from 4.4 °C (8.0 °F) to 13.2 °C (24.0 °F)). We may note that a 100 % increase in subcooling corresponds approximately to the 20 % overcharge fault discussed in Section 5.2.6.

Table US-38. Effect of excessive refrigerant subcooling on annual energy use for a slab-on-grade house

	Excessive Sub-Cooling	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	100 %	1,432	1,964	735	4.9	2,699.3	4.1	0.782	17,560	9,496	5,466	33,253	$785	106%
	200 %	1,483	1,976	710	4.5	2,686.0	3.5	0.786	20,480	10,377	5,439	37,028	$874	118%
Las Vegas	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	100 %	-	1,950	848	0.4	2,797.3	3.6	0.998	21,599	12,522	6,607	41,460	$1,451	106%
	200 %	-	1,971	818	0.3	2,789.5	3.0	1.000	25,241	13,716	6,589	46,277	$1,620	118%
Washington, DC	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	100 %	194	1,199	1,954	90.1	3,153.1	4.3	0.803	8,565	24,299	5,321	38,916	$1,524	108%
	200 %	223	1,208	1,912	88.8	3,119.5	3.6	0.807	10,023	26,696	5,264	42,714	$1,673	119%
Chicago	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	100 %	180	1,024	2,812	286.4	3,836.3	4.3	0.820	7,217	41,052	6,474	55,475	$1,972	108%
	200 %	183	1,031	2,769	282.3	3,799.9	3.7	0.824	8,446	44,775	6,412	60,365	$2,146	118%
Minneapolis	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	100 %	11	892	3,416	620.7	4,308.1	4.3	0.839	6,266	60,185	7,270	74,453	$2,234	108%
	200 %	12	898	3,375	614.9	4,272.7	3.6	0.843	7,332	64,959	7,210	80,233	$2,407	116%

Note: Subcooling of 4.4 °C (8.0 °F) was used as a no-fault condition.

Table US-39. Effect of excessive refrigerant subcooling on annual energy use for a house with basement

	Excessive Sub-Cooling	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	100 %	-	1,533	705	0.3	2,237.5	3.5	0.999	16,878	10,496	5,285	33,391	$1,169	106%
	200 %	-	1,554	682	0.3	2,236.6	3.0	1.000	19,787	11,530	5,283	37,332	$1,307	118%
Washington, DC	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	100 %	532	737	1,891	28.0	2,628.0	4.2	0.770	5,293	21,594	4,435	32,054	$1,255	109%
	200 %	620	741	1,850	27.6	2,591.2	3.6	0.774	6,182	23,960	4,373	35,246	$1,380	120%
Chicago	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	100 %	260	628	2,767	134.2	3,395.2	4.2	0.791	4,450	35,482	5,729	46,393	$1,650	110%
	200 %	278	631	2,721	133.7	3,352.8	3.6	0.794	5,197	39,277	5,658	50,863	$1,808	120%
Minneapolis	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	100 %	57	513	3,410	362.3	3,922.9	4.2	0.803	3,633	51,396	6,620	62,380	$1,871	109%
	200 %	60	516	3,372	359.6	3,887.3	3.6	0.808	4,244	56,387	6,560	67,922	$2,038	119%

Note: Subcooling of 4.4 °C (8.0 °F) was used as a no-fault condition.

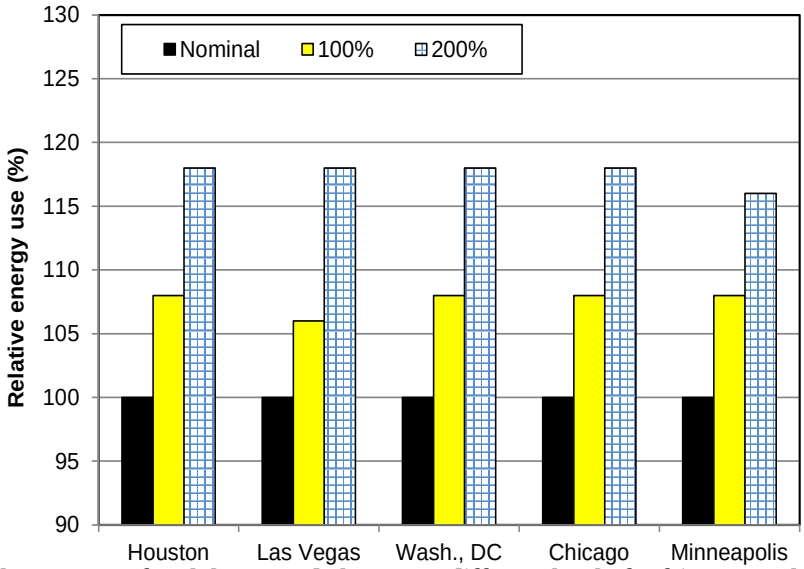


Figure US-28. Annual energy use for slab-on-grade houses at different level of refrigerant subcooling relative to the annual energy use for the house in the same location with the heat pump operating with the nominal refrigerant charge and subcooling

5.2.8. Effect of Non-Condensable Gases

If the refrigerant system gets non-condensable gases (e.g., air) mixed in with the refrigerant, the performance of the unit is degraded. Table US-40 shows the results for the slab-on-grade house, and Table US-41 shows the results for the basement house. The overall results show a (1 ~ 2) % energy use increase in climates with a significant heating season and a 4 % increase in the warmer climates. The moisture removal capability of the unit is only minimally affected by the non-condensable gases in the system.

Table US-40. Effect of non-condensable gases on annual energy use for a slab-on-grade house

	Non Condensibles	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	10 %	1,527	2,006	735	4.9	2,740.9	4.2	0.785	17,359	8,579	5,550	32,220	$761	102%
	20 %	1,579	1,985	713	4.3	2,697.7	4.0	0.787	17,947	8,598	5,463	32,739	$773	104%
Las Vegas	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	10 %	-	1,976	848	0.3	2,823.9	3.6	0.999	21,368	11,295	6,670	40,065	$1,402	102%
	20 %	-	1,949	821	0.3	2,769.7	3.5	1.000	22,127	11,328	6,542	40,730	$1,426	104%
Washington, DC	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	10 %	255	1,234	1,947	86.0	3,180.8	4.3	0.809	8,468	21,875	5,368	36,442	$1,427	101%
	20 %	277	1,233	1,901	81.1	3,133.9	4.1	0.810	8,793	21,906	5,289	36,719	$1,438	102%
Chicago	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	10 %	186	1,055	2,802	275.8	3,856.9	4.3	0.827	7,126	37,276	6,508	51,642	$1,836	101%
	20 %	188	1,055	2,754	264.2	3,808.7	4.2	0.829	7,395	37,352	6,427	51,905	$1,846	101%
Minneapolis	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	10 %	13	918	3,406	603.0	4,324.7	4.3	0.847	6,182	55,304	7,298	69,515	$2,085	101%
	20 %	14	919	3,366	582.3	4,284.4	4.1	0.848	6,416	55,348	7,230	69,726	$2,092	101%

Table US-41. Effect of non-condensable gases on annual energy use for a house with basement

	Non Condensibles	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	10 %	-	1,565	705	0.3	2,270.4	3.5	1.000	16,771	9,474	5,363	32,339	$1,132	102%
	20 %	-	1,550	685	0.3	2,235.4	3.4	1.000	17,390	9,540	5,280	32,941	$1,153	104%
Washington, DC	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	10 %	649	760	1,882	26.0	2,641.7	4.2	0.775	5,236	19,279	4,458	29,704	$1,163	101%
	20 %	677	761	1,841	23.2	2,602.0	4.0	0.776	5,438	19,434	4,391	29,995	$1,175	102%
Chicago	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	10 %	288	647	2,753	127.3	3,400.0	4.2	0.797	4,387	31,817	5,737	42,674	$1,517	101%
	20 %	287	649	2,706	119.9	3,354.6	4.1	0.799	4,560	32,046	5,661	42,999	$1,529	102%
Minneapolis	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	10 %	61	528	3,399	347.5	3,927.5	4.2	0.810	3,584	46,560	6,628	57,503	$1,725	101%
	20 %	64	530	3,354	333.6	3,884.3	4.1	0.812	3,726	46,815	6,555	57,828	$1,735	101%

5.2.9. Effect of Voltage

When input voltage to the unit is changed from the nominal value, the performance of the unit is degraded. Tables US-42 and US-43 show the results for the slab-on-grade house and the basement house, respectively. The condition of 25 % overvoltage results in a (9 ~10) % increase in annual energy consumption. This effect on the energy use does not include an adjustment for indoor fan power change with voltage. The undervoltage of 8 % resulted in an insignificant (within 1 %) change in the energy use. Higher levels of undervoltage were not studied because of a possible heat pump catastrophic failure.

Table US-42. Effect of voltage on annual energy use for a slab-on-grade house

	Voltage	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	-8 %	1,508	1,992	748	5.0	2,740.1	4.3	0.785	16,677	8,464	5,549	31,421	$742	100%
	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	8 %	1,519	1,974	752	5.3	2,725.8	4.3	0.785	16,970	8,733	5,520	31,954	$754	102%
	25 %	1,547	1,966	767	5.9	2,733.2	3.9	0.786	18,676	9,616	5,535	34,559	$816	110%
Las Vegas	-8 %	-	1,977	863	0.3	2,840.0	3.7	0.999	20,715	11,143	6,708	39,299	$1,375	100%
	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	8 %	-	1,958	870	0.3	2,827.2	3.7	0.999	20,741	11,523	6,678	39,674	$1,389	101%
	25 %	-	1,947	888	0.4	2,835.3	3.4	0.999	22,465	12,694	6,697	42,587	$1,491	109%
Washington, DC	-8 %	252	1,213	1,969	88.6	3,181.3	4.5	0.809	8,062	21,594	5,368	35,756	$1,400	99%
	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	8 %	256	1,202	1,979	89.7	3,181.8	4.4	0.809	8,289	22,211	5,369	36,601	$1,434	102%
	25 %	274	1,197	2,010	94.8	3,207.7	3.9	0.810	9,211	24,175	5,413	39,530	$1,548	110%
Chicago	-8 %	188	1,035	2,830	280.3	3,865.2	4.5	0.827	6,770	36,879	6,522	50,904	$1,810	99%
	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	8 %	189	1,027	2,842	283.5	3,868.8	4.4	0.827	6,988	37,781	6,529	52,030	$1,850	102%
	25 %	189	1,022	2,879	294.2	3,901.4	3.9	0.828	7,786	40,678	6,584	55,779	$1,983	109%
Minneapolis	-8 %	13	901	3,430	611.0	4,331.0	4.5	0.846	5,871	54,824	7,309	68,736	$2,062	100%
	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	8 %	14	894	3,440	616.2	4,334.1	4.4	0.846	6,064	55,920	7,314	70,029	$2,101	101%
	25 %	14	890	3,470	635.5	4,360.3	3.9	0.848	6,764	59,502	7,358	74,356	$2,231	108%

Table US-43. Effect of voltage on annual energy use for a house with basement

	Voltage	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	-8 %	-	1,561	716	0.3	2,277.2	3.6	1.000	16,243	9,319	5,379	31,672	$1,109	100%
	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	8 %	-	1,545	721	0.3	2,266.0	3.6	1.000	16,275	9,635	5,352	31,994	$1,120	101%
	25 %	-	1,536	736	0.3	2,272.1	3.4	1.000	17,649	10,604	5,367	34,351	$1,202	109%
Washington, DC	-8 %	656	746	1,905	26.8	2,650.9	4.4	0.775	4,987	18,969	4,473	29,161	$1,142	99%
	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	8 %	657	740	1,916	27.1	2,655.6	4.3	0.775	5,125	19,545	4,481	29,883	$1,170	102%
	25 %	674	736	1,945	29.6	2,681.6	3.9	0.775	5,687	21,380	4,525	32,324	$1,266	110%
Chicago	-8 %	286	634	2,782	129.7	3,415.8	4.5	0.797	4,175	31,353	5,764	42,023	$1,494	99%
	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	8 %	289	629	2,793	131.8	3,422.0	4.3	0.797	4,302	32,212	5,775	43,020	$1,530	102%
	25 %	295	627	2,829	138.4	3,455.6	3.9	0.798	4,792	34,940	5,831	46,295	$1,646	110%
Minneapolis	-8 %	61	518	3,421	353.6	3,938.8	4.4	0.810	3,408	45,973	6,647	56,759	$1,703	99%
	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	8 %	61	514	3,432	357.0	3,946.0	4.3	0.810	3,514	47,027	6,659	57,931	$1,738	102%
	25 %	63	511	3,469	369.9	3,980.7	3.9	0.811	3,911	50,497	6,717	61,857	$1,856	108%

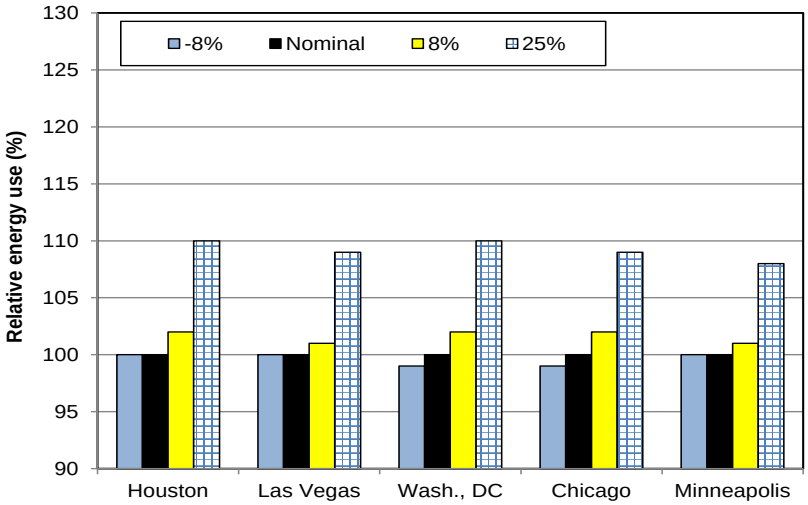


Figure US-29. Annual energy use for slab-on-grade houses at different levels of input voltages relative to the energy use for the house in the same location when the heat pump operates with nominal voltage

5.2.10. Effect of TXV Sizing

Only undersizing of the TXV in the cooling mode is considered in this study. When the size of the TXV does not match the compressor size, the performance of the system is degraded. Table US-44 shows the results for the slab-on-grade houses, and Table 45 shows the results for the basement houses. Generally, the impact is modest at 20 % undersizing in any climate and remains relatively small for Minneapolis at even higher fault levels. However, the impact becomes significant at 40 % undersizing, particularly in hot climates where the energy use increases by (10 ~ 14) %. Moisture removal is only modestly affected.

Table US-44. Effect of TXV sizing on annual energy use for a slab-on-grade house

	Undersized TXV	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Houston	0 %	1,512	1,981	749	5.1	2,730.5	4.3	0.785	16,660	8,537	5,529	31,457	$743	100%
	20 %	1,516	2,000	749	5.1	2,748.6	4.3	0.785	16,855	8,536	5,566	31,688	$748	101%
	40 %	1,534	2,312	749	5.1	3,060.6	3.6	0.784	20,357	8,533	6,198	35,819	$846	114%
	60 %	1,575	2,767	749	5.1	3,515.9	2.8	0.780	25,508	8,531	7,120	41,890	$989	133%
Las Vegas	0 %	-	1,966	865	0.3	2,831.1	3.7	0.999	20,531	11,251	6,687	39,200	$1,372	100%
	20 %	-	1,973	865	0.3	2,837.4	3.7	0.999	20,623	11,242	6,702	39,298	$1,375	100%
	40 %	-	2,210	865	0.3	3,074.8	3.3	1.000	23,723	11,242	7,263	42,959	$1,504	110%
	60 %	-	2,647	864	0.3	3,511.2	2.6	1.000	29,509	11,235	8,294	49,770	$1,742	127%
Washington, DC	0 %	253	1,207	1,971	89.0	3,178.0	4.5	0.809	8,098	21,759	5,363	35,952	$1,408	100%
	20 %	257	1,234	1,971	89.0	3,204.0	4.3	0.809	8,341	21,754	5,407	36,233	$1,419	101%
	40 %	260	1,449	1,971	89.0	3,420.1	3.6	0.810	10,317	21,758	5,771	38,577	$1,511	107%
	60 %	258	1,751	1,970	88.9	3,720.8	2.8	0.810	13,097	21,748	6,279	41,855	$1,639	116%
Chicago	0 %	189	1,031	2,833	281.2	3,863.9	4.5	0.827	6,816	37,118	6,520	51,186	$1,820	100%
	20 %	188	1,058	2,833	281.2	3,890.7	4.4	0.827	7,064	37,117	6,566	51,478	$1,830	101%
	40 %	188	1,246	2,833	281.2	4,079.1	3.6	0.830	8,792	37,116	6,884	53,523	$1,903	105%
	60 %	182	1,512	2,833	281.2	4,344.2	2.8	0.834	11,229	37,113	7,331	56,405	$2,006	110%
Minneapolis	0 %	13	897	3,432	612.5	4,328.9	4.5	0.846	5,912	55,105	7,305	69,053	$2,072	100%
	20 %	13	922	3,432	612.5	4,354.1	4.3	0.847	6,139	55,106	7,348	69,324	$2,080	100%
	40 %	13	1,087	3,431	612.5	4,518.4	3.5	0.851	7,649	55,099	7,625	71,104	$2,133	103%
	60 %	11	1,321	3,431	612.5	4,751.9	2.8	0.856	9,787	55,097	8,019	73,634	$2,209	107%

Table US-45. Effect of TXV sizing on annual energy use for a house with basement

	Undersized TXV	Hours Above 55 % RH	AC Runtime (h)	Htg Runtime (h)	Backup Heat Runtime (h)	AHU Fan Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Htg Energy (MJ)	AHU Fan Energy (MJ)	TOTAL ENERGY (MJ)	Total Costs	Relative Energy
Las Vegas	0 %	-	1,552	718	0.3	2,269.9	3.7	1.000	16,107	9,407	5,362	31,607	$1,106	100%
	20 %	-	1,558	718	0.3	2,276.2	3.7	1.000	16,187	9,408	5,376	31,704	$1,110	100%
	40 %	-	1,738	718	0.3	2,455.9	3.2	1.000	18,575	9,408	5,801	34,516	$1,208	109%
	60 %	-	2,117	717	0.3	2,834.1	2.5	1.000	23,631	9,403	6,694	40,460	$1,416	128%
Washington, DC	0 %	654	742	1,907	27.0	2,649.5	4.4	0.775	5,008	19,120	4,471	29,330	$1,149	100%
	20 %	653	756	1,907	27.0	2,663.0	4.3	0.775	5,132	19,120	4,494	29,477	$1,155	101%
	40 %	649	877	1,907	27.0	2,784.1	3.5	0.778	6,269	19,121	4,698	30,819	$1,207	105%
	60 %	635	1,066	1,907	27.0	2,972.8	2.8	0.782	8,022	19,120	5,017	32,890	$1,288	112%
Chicago	0 %	289	631	2,785	129.8	3,416.1	4.4	0.797	4,198	31,565	5,765	42,259	$1,503	100%
	20 %	283	645	2,785	129.8	3,430.6	4.3	0.797	4,327	31,569	5,789	42,418	$1,508	100%
	40 %	284	750	2,785	129.8	3,534.9	3.5	0.801	5,312	31,564	5,965	43,573	$1,549	103%
	60 %	282	908	2,785	129.7	3,692.9	2.7	0.808	6,779	31,559	6,232	45,302	$1,611	107%
Minneapolis	0 %	61	515	3,424	354.2	3,939.8	4.4	0.810	3,428	46,239	6,648	57,048	$1,711	100%
	20 %	61	527	3,424	354.2	3,950.5	4.3	0.811	3,531	46,236	6,667	57,165	$1,715	100%
	40 %	59	611	3,424	354.2	4,035.1	3.5	0.815	4,326	46,235	6,809	58,102	$1,743	102%
	60 %	56	739	3,424	354.2	4,163.3	2.7	0.822	5,516	46,235	7,026	59,507	$1,785	104%

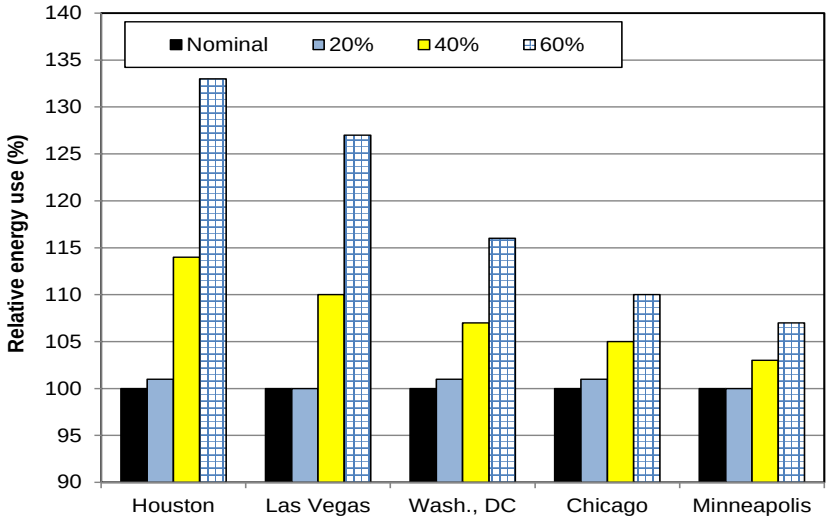


Figure US-30. Annual energy use for slab-on-grade houses at different levels of TXV undersizing relative to the annual energy use for the house when the heat pump operates with a properly sized TXV

5.2.11. Discussion of the Effects of Single Faults

Figure US-31 shows examples of annual energy used by a heat pump installed with different installation faults in a slab-on-grade house. The levels of individual faults were selected to reflect, to some degree, the installation condition which might not be noticed by a poorly trained technician. (The authors recognize the speculative aspect of this selection.)

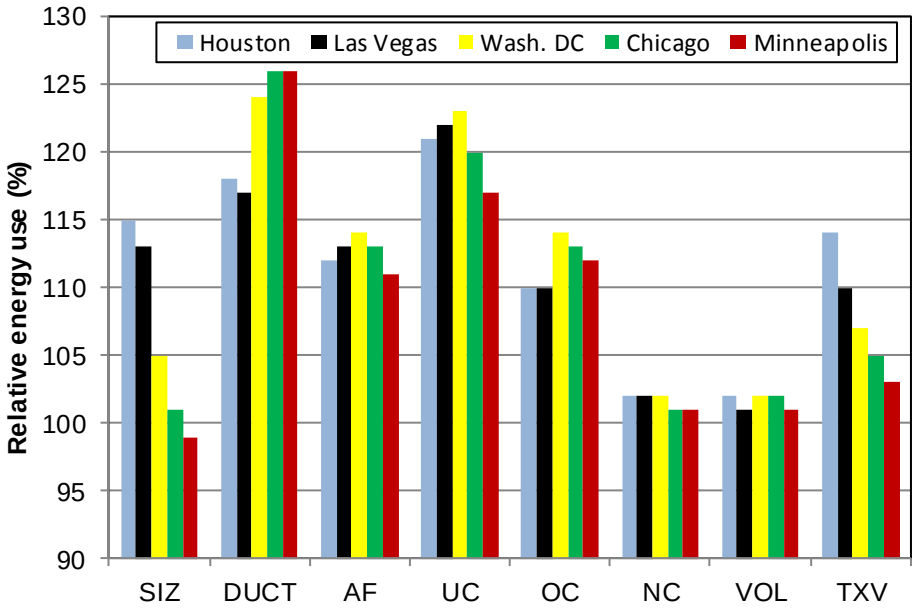


Figure US-31. Annual energy use by a heat pump in a slab-on-grade house resulting from a single-fault installation, referenced to a fault-free installation. (Table 46 shows the selected fault levels)

Table US-46 Levels of individual faults used in Figure US-31

Fault Type	Fault Level (%)
Heat Pump Sizing (SIZ)[(a)]	+ 50
Duct Leakage (DUCT)	+ 30
Indoor Coil Airflow (AF)	- 36
Refrigerant Undercharge (UC)	- 30
Refrigerant Overcharge (OC)	+ 30
Non-Condensable Gases (NC)	+ 10
Electric Voltage (VOL)	+ 8
TXV Undersizing (TXV)	- 40

(a) Oversize scenario (2) described in Section 5.2.2.

Simulation results show no drastic differences in the effect of installation faults on energy use in a slab-on-grade house and a basement house, except for the duct leakage fault. For the slab-on-grade house, this fault has the potential to result in a higher increase in energy use that any other fault. The impact of this fault is higher for the heating dominated climate (Chicago and Minneapolis, 26 %) than for the cooling dominated climate (Houston, 18 %). Obviously, duct leakage will also result in some increase of energy use for the basement house; however, the model we used would not discern this effect.

The second most influential fault is refrigerant undercharge. For the 30 % undercharge fault level, the energy use increase is of the order of 20 % irrespective of the climate and building type. Refrigerant overcharge can also result in a significant increase in energy use, (10 ~16) % at the 30 % overcharge fault level. Improper indoor airflow can affect similar performance degradation.

Equipping a house with an oversized heat pump has a small effect if the air duct is oversized accordingly (which may be the case with a new construction). However, if the air duct is too restrictive and the nominal indoor airflow is maintained by adjusting the fan speed (scenario (2)), a 15 % increase in energy use for the house in Houston is predicted.

The cooling TXV undersized fault has also the potential to significantly increase the energy use. The effect of this fault will be most pronounced in localities with a high number of cooling mode operating hours. The cooling mode TXV undersized by 40 % results in (9 ~ 14) % more energy used in Houston as compared to a (3 ~ 5) % in Chicago.

The impact of the remaining faults – non-condensables and improper voltage – is under 4 %. The non-condensables and improper voltage faults, however, represent a substantial risk for durability of equipment and are very important to be diagnosed during a heat pump installation.

5.3. Simulations with Dual Faults

5.3.1. Studied Fault Combinations

The analysis in this section considers the combination of two faults, A and B. Each set of faults was considered in four combinations (Table US-47).

Table US-47. Combinations of studied faults

Fault combination case	Level of fault A	Level of fault B
a	moderate	moderate
b	moderate	worst
c	Worst	moderate
d	Worst	worst

The moderate level will be the value at the middle of the range, while the worst level will be the highest (or lowest) probable level of the fault value. Table US-48 defines the set or combinations of dual faults simulated for cases where heating and cooling were considered together. Table US-49 defines the sets of faults that apply for the cooling-only case. The most right-hand column, in both tables, shows an approximate effect of the studied fault sets on the energy use: the faults effects may be additive (A+B), less than additive (<A+B), or greater the additive (>A+B).

Table US-48. Dual fault sets considered in simulations (heating and cooling) and their approximate collective effect on annual energy use

Fault set #	Fault A (moderate & worst level)[(a)]	Fault B (moderate & worst level)	Effect on energy use
1	Duct leakage (20 %, 40 %)	Oversize[(b)] (25 %, 50 %)	A+B
2	Duct leakage (20 %, 40 %)	Indoor coil airflow (-15 %, -36 %)	< A+B
3	Duct leakage (20 %, 40 %)	Refrigerant undercharge (-15 %, -30 %)	A+B or > A+B
4	Duct leakage (20 %, 40 %)	Refrigerant overcharge (15 %, 30 %)	A+B
5	Duct leakage (20 %, 40 %)	Non-condensables (10 %, 20 %)	A+B
6	Oversize[(b)] (25 %, 50 %)	Refrigerant undercharge (-15 %, -30 %)	A+B
7	Oversize[(b)] (25 %, 50 %)	Refrigerant overcharge (15 %, 30 %)	A+B
8	Oversize[(b)] (25 %, 50 %)	Non-condensables (10 %, 20 %)	A+B
9	Indoor coil airflow (-15 %, -36 %)	Refrigerant undercharge (-15 %, -30 %)	< A+B
10	Indoor coil airflow (-15 %, -36 %)	Refrigerant overcharge (15 %, 30 %)	< A+B
11	Indoor coil airflow (-15 %, -36 %)	Non-condensables (10 %, 20 %)	< A+B

(a) moderate = mid-level value, worst = lowest/highest level value
(b) Oversize scenario (2) was select because it covers the prevalent field bias (undersized ducts)

Table US-49. Dual fault sets considered in annual simulations (heating and cooling) and their approximate collective effect on annul energy use; TXV fault existing in cooling only

Fault set #	*Fault A (moderate & worst level)*[(a, b)]	*Fault B (moderate & worst level)*	*Effect on energy use*
12	Duct leakage [(a)] (20 %, 40 %)	TXV undersizing (-20 %, -60 %)	A+B
13	Oversize[(b)] (25 %, 50 %)	TXV undersizing (-20 %, -60 %)	A+B
14	Indoor coil airflow (-15 %, -36 %)	TXV undersizing (-20 %, -60 %)	< A+B

(a) Faults listed as Faults A exist in cooling and heating
(b) moderate = mid-level value, worst = lowest/highest level value
(c) Oversize scenario (2) was select because it covers the prevalent field bias (undersized ducts)

5.3.2. Effects of Dual Faults

Simulations were performed for 14 dual fault sets, with 4 runs per set, in the 9 house/climate combinations for a total of 504 runs. Because of similarity between the obtained results, the tables below are limited to representative cases, which include the slab-on-grade house for Houston, Washington, DC, and Minneapolis, and the house with a basement for Washington, DC. For the Houston house, Table US-50 shows results for dual fault sets 1 through 5, which represent all studied dual faults involving duct leakage; Table US-51 shows results for dual fault sets 6 through 8, which represent all studied dual faults involving the oversized heat pump, except the case with duct leakage presented in Table US-50; and Table US-52 presents the remaining three studied cases with dual faults present in both cooling and heating. Table US-53 presents the effect on annual energy use of the undersized cooling TXV with either duct leakage, oversized heat pump, or low airflow rate faults, which occur in both cooling and heating mode. Tables US-54 through US-64 present simulation results for the remaining cases. For nine out of fourteen sets studied, the effect of dual faults was approximately additive (Table US-49). For the remaining five sets – all involving indoor coil airflow – the effect was less than additive. A few results that are not immediately intuitive are discussed at the end of this section.

Table US-50. Relative energy use for dual fault sets 1 to 5 for the slab-on-grade house in Houston

Duct leakage with oversized heat pump, low airflow rate, undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 1		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
25% Oversized	109%	121%	148%
50% Oversized	115%	128%	159%

Dual Fault Set: 2		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
-15% Airflow	104%	110%	129%
-36% Airflow	112%	114%	130%

Dual Fault Set: 3		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
15% Undercharged	105%	115%	136%
30% Undercharged	121%	132%	156%

Dual Fault Set: 4		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
15% Overcharged	103%	112%	132%
30% Overcharged	110%	119%	141%

Dual Fault Set: 5		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
10% Non-Condensibles	102%	112%	131%
20% Non-Condensibles	104%	113%	133%

Table US-51. Relative energy use for dual fault sets 6 to 8 for the slab-on-grade house in Houston

Oversized heat pump with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 6		25% Oversized	50% Oversized
	100%	109%	115%
15% Undercharged	105%	115%	121%
30% Undercharged	121%	133%	140%

Dual Fault Set: 7		25% Oversized	50% Oversized
	100%	109%	115%
15% Overcharged	103%	112%	118%
30% Overcharged	110%	119%	125%

Dual Fault Set: 8		25% Oversized	50% Oversized
	100%	109%	115%
10% Non-Condensibles	102%	112%	118%
20% Non-Condensibles	104%	113%	120%

Table US-52. Relative energy use for dual fault sets 9 to 11 for the slab-on-grade house in Houston

Low airflow rate with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 9		-15% Airflow	-36% Airflow
	100%	104%	112%
15% Undercharged	105%	107%	111%
30% Undercharged	121%	123%	127%

Dual Fault Set: 10		-15% Airflow	-36% Airflow
	100%	104%	112%
15% Overcharged	103%	105%	109%
30% Overcharged	110%	112%	116%

Dual Fault Set: 11		-15% Airflow	-36% Airflow
	100%	104%	112%
10% Non-Condensibles	102%	104%	109%
20% Non-Condensibles	104%	106%	111%

Table US-53. Relative energy use for dual fault sets 12 to 14 involving cooling mode TXV for the slab-on-grade house in Houston

Undersized TXV with duct leakage, oversized heat pump, and low airflow rate faults

Dual Fault Set: 12		20% Duct Leakage	40% Duct Leakage
	100%	109%	128%
20% TXV Undersizing	101%	110%	129%
60% TXV Undersizing	133%	143%	163%

Dual Fault Set: 13		25% Oversized	50% Oversized
	100%	109%	115%
20% TXV Undersizing	101%	110%	116%
60% TXV Undersizing	133%	150%	161%

Dual Fault Set: 14		-15% Airflow	-36% Airflow
	100%	104%	112%
20% TXV Undersizing	101%	103%	107%
60% TXV Undersizing	133%	135%	139%

Table US-54. Relative energy use for dual fault sets 1 to 5 for the slab-on-grade house in Washington, DC

Duct leakage with oversized heat pump , low airflow rate, undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 1		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
25% Oversized	103%	117%	152%
50% Oversized	105%	119%	156%

Dual Fault Set: 2		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
-15% Airflow	105%	112%	137%
-36% Airflow	114%	114%	137%

Dual Fault Set: 3		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
15% Undercharged	105%	117%	146%
30% Undercharged	123%	137%	172%

Dual Fault Set: 4		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
15% Overcharged	104%	116%	145%
30% Overcharged	114%	127%	157%

Dual Fault Set: 5		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
10% Non-Condensibles	101%	113%	140%
20% Non-Condensibles	102%	114%	141%

Table US-55. Relative energy use for dual fault sets 6 to 8 for the slab-on-grade house in Washington, DC

Oversized heat pump with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 6		25% Oversized	50% Oversized
	100%	103%	105%
15% Undercharged	105%	108%	110%
30% Undercharged	123%	127%	129%

Dual Fault Set: 7		25% Oversized	50% Oversized
	100%	103%	105%
15% Overcharged	104%	107%	109%
30% Overcharged	114%	118%	119%

Dual Fault Set: 8		25% Oversized	50% Oversized
	100%	103%	105%
10% Non-Condensibles	101%	105%	107%
20% Non-Condensibles	102%	106%	108%

Table US-56. Relative energy use for dual fault sets 9 to 11 for the slab-on-grade house in Washington, DC

Low airflow rate with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 9		-15% Airflow	-36% Airflow
	100%	105%	114%
15% Undercharged	105%	106%	109%
30% Undercharged	123%	124%	127%

Dual Fault Set: 10		-15% Airflow	-36% Airflow
	100%	105%	114%
15% Overcharged	104%	106%	109%
30% Overcharged	114%	116%	119%

Dual Fault Set: 11 Mult Fault Set: 12		-15% Airflow	-36% Airflow
	100%	105%	114%
10% Non-Condensibles	101%	103%	106%
20% Non-Condensibles	102%	104%	107%

Table US-57. Relative energy use for dual fault sets 12 to 14 involving the cooling mode TXV for the slab-on-grade house in Washington, DC

Undersized TXV with duct leakage, oversized heat pump, and low airflow rate faults

Dual Fault Set: 12		20% Duct Leakage	40% Duct Leakage
	100%	112%	139%
20% TXV Undersizing	101%	113%	140%
60% TXV Undersizing	116%	129%	157%

Dual Fault Set: 13		25% Oversized	50% Oversized
	100%	103%	105%
20% TXV Undersizing	101%	104%	106%
60% TXV Undersizing	116%	123%	127%

Dual Fault Set: 14		-15% Airflow	-36% Airflow
	100%	105%	114%
20% TXV Undersizing	101%	102%	105%
60% TXV Undersizing	116%	118%	121%

Table US-58. Relative energy use for dual fault sets 1 to 5 for the slab-on-grade house in Minneapolis

Duct leakage with oversized heat pump , low airflow rate, undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 1		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
25% Oversized	100%	114%	148%
50% Oversized	99%	114%	149%

Dual Fault Set: 2		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
-15% Airflow	103%	113%	138%
-36% Airflow	111%	113%	137%

Dual Fault Set: 3		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
15% Undercharged	103%	116%	144%
30% Undercharged	117%	132%	162%

Dual Fault Set: 4		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
15% Overcharged	104%	116%	144%
30% Overcharged	112%	125%	153%

Dual Fault Set: 5		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
10% Non-Condensibles	101%	113%	140%
20% Non-Condensibles	101%	113%	140%

Table US-59. Relative energy use for dual fault sets 6 to 8 for the slab-on-grade house in Minneapolis

Oversized heat pump with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 6		25% Oversized	50% Oversized
	100%	100%	99%
15% Undercharged	103%	103%	103%
30% Undercharged	117%	118%	117%

Dual Fault Set: 7		25% Oversized	50% Oversized
	100%	100%	99%
15% Overcharged	104%	103%	103%
30% Overcharged	112%	112%	112%

Dual Fault Set: 8		25% Oversized	50% Oversized
	100%	100%	99%
10% Non-Condensibles	101%	101%	101%
20% Non-Condensibles	101%	101%	101%

Table US-60. Relative energy use for dual fault sets 9 to 11 for the slab-on-grade house in Minneapolis

Low airflow rate with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 9		-15% Airflow	-36% Airflow
	100%	103%	111%
15% Undercharged	103%	104%	106%
30% Undercharged	117%	118%	121%

Dual Fault Set: 10		-15% Airflow	-36% Airflow
	100%	103%	111%
15% Overcharged	104%	104%	107%
30% Overcharged	112%	112%	115%

Dual Fault Set: 11		-15% Airflow	-36% Airflow
	100%	103%	111%
10% Non-Condensibles	101%	102%	104%
20% Non-Condensibles	101%	102%	104%

Table US-61. Relative energy use for dual fault sets 12 to 14 involving the cooling mode TXV for the slab-on-grade house in Minneapolis

Undersized TXV with duct leakage, oversized heat pump, and low airflow rate faults

Dual Fault Set: 12		20% Duct Leakage	40% Duct Leakage
	100%	113%	140%
20% TXV Undersizing	100%	113%	140%
60% TXV Undersizing	107%	120%	147%

Dual Fault Set: 13		25% Oversized	50% Oversized
	100%	100%	99%
20% TXV Undersizing	100%	100%	100%
60% TXV Undersizing	107%	108%	108%

Dual Fault Set: 14		-15% Airflow	-36% Airflow
	100%	103%	111%
20% TXV Undersizing	100%	101%	103%
60% TXV Undersizing	107%	108%	110%

Table US-62. Relative energy use for dual fault sets 6 to 8 for the basement house in Washington, DC

Oversized heat pump with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 6		25% Oversized	50% Oversized
	100%	104%	108%
15% Undercharged	104%	109%	112%
30% Undercharged	121%	126%	129%

Dual Fault Set: 7		25% Oversized	50% Oversized
	100%	104%	108%
15% Overcharged	105%	109%	112%
30% Overcharged	116%	120%	124%

Dual Fault Set: 8		25% Oversized	50% Oversized
	100%	104%	108%
10% Non-Condensibles	101%	106%	109%
20% Non-Condensibles	102%	107%	111%

Table US-63. Relative energy use for dual fault sets 9 to 11 for the basement house in Washington, DC

Low airflow rate with undercharged, overcharged, and non-condensable gases faults

Dual Fault Set: 9		-15% Airflow	-36% Airflow
	100%	105%	113%
15% Undercharged	104%	105%	106%
30% Undercharged	121%	122%	124%

Dual Fault Set: 10		-15% Airflow	-36% Airflow
	100%	105%	113%
15% Overcharged	105%	106%	107%
30% Overcharged	116%	117%	118%

Dual Fault Set: 11		-15% Airflow	-36% Airflow
	100%	105%	113%
10% Non-Condensibles	101%	102%	103%
20% Non-Condensibles	102%	103%	105%

Table US-64. Relative energy use for dual fault sets 13 to 14 involving the cooling mode TXV for the basement house in Washington, DC

Undersized TXV with duct leakage, oversized heat pump, and low airflow rate faults

Dual Fault Set: 13		25% Oversized	50% Oversized
	100%	104%	108%
20% TXV Undersizing	101%	105%	108%
60% TXV Undersizing	112%	118%	123%

Dual Fault Set: 14		-15% Airflow	-36% Airflow
	100%	105%	113%
20% TXV Undersizing	101%	101%	103%
60% TXV Undersizing	112%	113%	115%

While reviewing the above results, a reader may be surprised to see that in a few cases the energy use with two simultaneous faults is as at a similar level as that for the more influential single fault. The most confounding are perhaps the results obtained for the dual fault set # 2 involving air duct leakage and reduced indoor coil airflow (Table US-54). In this case, for the 40 % duct leakage existing alone, the energy use increases by 39 %, and for the 36 % reduction in the airflow, the energy use increases by 14 %; however, when these two faults exist simultaneously, the combined effect is an increase of energy use by 37 %, which is less than that when the duct leakage fault exists alone. This result can be explained by the fact that at a lowered airflow the heat pump satisfies the load using less air (it is a larger temperature spread between the return and supply air). Hence, in absolute numbers, the amount of energy lost due to duct leakage is smaller because the leaked air is a percentage of the total airflow. Simply, duct leakage is a dominating fault, and a reduction of the effect of this fault more than compensates for the losses associated with the reduced airflow (decreased air-side heat transfer coefficient and increased compressor power due to increased temperature lift).

Also, interesting results for the low indoor airflow combined with either the refrigerant overcharge (dual set fault # 10), or non-condensable gases (dual set fault # 11), can be reviewed in Table US-56. If the low airflow fault exists alone, the energy use increases by 14 % for the 36 % airflow reduction. This fault

demonstrates itself in a lower temperature of the evaporator, which results in a somewhat lower sensible capacity and increased latent capacity of the air conditioner. Since in performed simulations the air conditioner had to satisfy the thermostat (i.e., the same sensitive load) and the rate of moisture removal increased, the energy use increased. Now, refrigerant overcharge fault, or non-condensables fault, causes the condenser pressure to increase. This pulls up the pressure (and temperature) of the evaporator, which reduces the latent load that the air conditioner handles. At moderate levels of the overcharge and non-condensables faults, the energetic benefit of the lowered latent load is greater than that of a modest COP penalty associate with these faults. Consequently, moderate levels of refrigerant overcharge and non-condensables faults caused a reduction of energy used by the unit with 36 % reduced air flow. Greater levels of these faults reverse this energy use trend. (Note that the above explanation discusses the first order effects of a rather complicated reaction of the systems to these faults, e.g., a lower air-side heat transfer coefficient, lower indoor fan power, and the effect on performance in the heating mode.)

Relatively less perplexing is the interaction between the low airflow fault and undersized TXV fault (Table US-57, dual fault set # 14). In this case, a 20 % undersized cooling-mode TXV improved the performance of the system operating with a reduced indoor coil airflow. Since a reduced airflow reduces the system capacity, a TVX that was 20 % undersized for the rated capacity showed to be a better match for the 'reduced capacity' system than the TXV properly sized for the rated capacity.

It should be noted that airflow reduction lowers equipment capacity and may compromise occupant's comfort when approaching design conditions. Additionally, in extreme conditions, or in combination with other faults, it may lead to indoor coil frosting during cooling operation and equipment tripping or failure.

5.3.3. Discussion of the Effects of Dual Faults

As expected, the collective impact of two simultaneous faults on the energy consumption varies and depends on the faults considered. In most cases, the collective effect can be described as being additive; however, the effect can exceed, or be markedly below this additive value, including being approximately equal to the individual effect of one of the faults involved, as noted in Tables US-48 and US-49. The above characterization applies to all house/climate combinations. The relative impact on energy use also is similar for all cases studied (Figures US-32, and US-33).

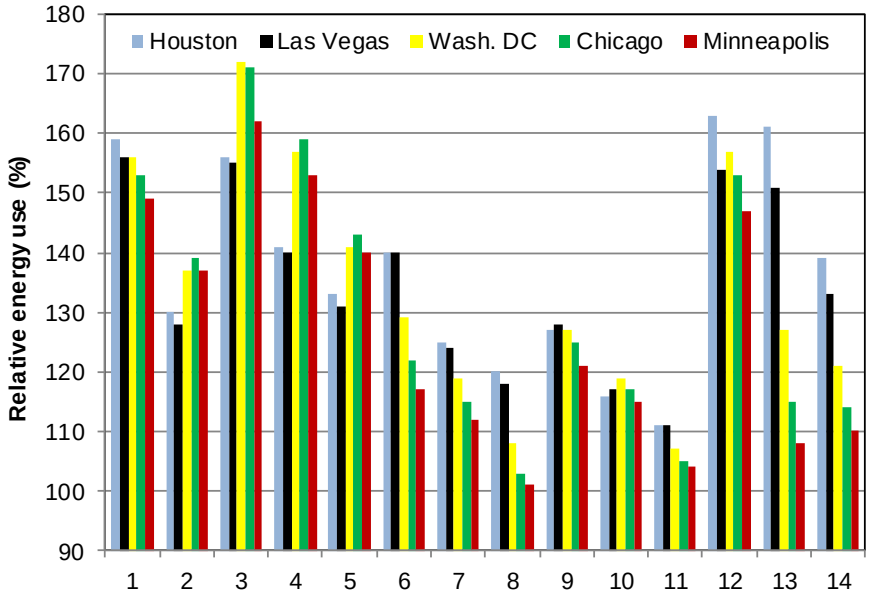


Figure US-32. Annual energy use for slab-on-grade houses with 14 dual-faults relative to the energy use for the houses with fault-free installations (Faults defined in Tables US-48 and US-49; Table US-47 case d, worst level for both faults)

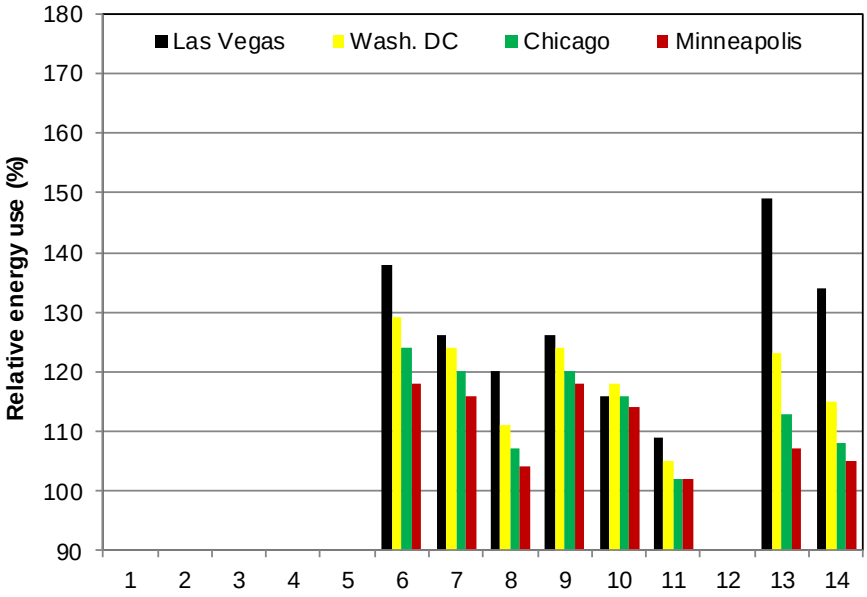


Figure US-33. Annual energy use for houses with basement with 8 dual-fault installations referenced to the energy use for the houses with fault-free installations (Faults defined in Tables US-48 and US-49, Table US-47 case d; worst level for both faults; the omitted dual faults involve duct leakage, which was not considered in houses with basement)

5.4. Effects of Triple Faults

Triple faults were not simulated in this study because the open literature does not provide sufficient data on effects of multiple faults to allow for their characterization and use in annual simulations of building/heat pump systems. Nevertheless, the occurrence of three simultaneous faults is plausible, particularly for the most common faults such as refrigerant undercharge, improper indoor airflow, or duct leakage. It is reasonable to assume that the effect of a triple fault will be as least as high as that of any of the possible three fault pairs considered individually; however, the effect of the third fault can increase the effect of the other two faults in an additive manner. As an example of a triple fault, SCE (2012) reported almost 70 % degradation in capacity for a split air conditioner operating under highly restricted airflow of the condenser, 40 % refrigerant undercharge, and 56 % reduction in the indoor airflow.

6. CONCLUDING REMARKS

Extensive simulations of house/heat pump systems in five climatic zones lead to the following conclusions:

- Effect of different installation faults on annual energy use is similar for a slab-on-grade house (ducts located in the unconditioned attic) and a basement house (ducts located in the semi-conditioned basement), except the duct leakage fault.
- Effect of different installation faults is similar in different climates except for the following cases:
  - Duct leakage: significant increase in the indoor RH for an installation in a hot & humid climate.
  - Heat pump oversizing with undersized air ducts: in heating-dominated climates, heat pump oversizing reduces the use of backup heat, which compensates for the increased indoor fan energy use associated with overcoming the higher external static pressure.
- Undersized cooling mode TXV: little effect in heating-dominated climates, while a significant increase of energy use is possible in cooling-dominated climates.

The effect of simultaneous faults can be additive (e.g., duct leakage and non-condensable gases), little changed relative to the single fault condition (e.g., low indoor airflow and refrigerant undercharge), or well-beyond additive (duct leakage and refrigerant undercharge).

The study found duct leakage, refrigerant undercharge, oversized heat pump with non-oversized ductwork, low indoor airflow due to undersized ductwork, and refrigerant overcharge to have the most potential for causing significant performance degradation and increased annual energy consumption. Increases of energy use by 30 % due to improper installation practices seem to be plausible. A well-designed and documented survey of heat pump installations would be helpful in establishing the prevalence of different installation faults and effective practices for their elimination.

A significant increase in annual energy use can be caused by lowering the thermostat in the cooling mode to improve indoor comfort in cases of excessive indoor humidity levels. For Houston, TX, lowering the thermostat setting by 1.1°C (2.0°F) increased the annual energy use by 20 %, and the energy use increase rate is even higher due to further lowering the setting (the effect is not linear).

The authors contend that the laboratory and modeling results from this analysis using a 2.5 ton heat pump are representative of all unitary equipment including commercial split-systems and single package units (e.g., roof top units).

The goal of this study was to assess the impacts that HVAC system installation faults had on equipment electricity consumption. The effect of the installation faults on occupant comfort was not the main focus of the study, and this research did not seek to quantify any impacts on indoor air quality, or noise generation (e.g., airflow noise from air moving through restricted ducts). Additionally, the study does not address the effects that installation faults have on equipment reliability/robustness (number of starts/stops, etc.), maintainability (e.g., access issues), or costs of initial installation and ongoing maintenance.

7. NOMENCLATURE

A = area [m^2, (ft^2)]
ACH50 = air changes per hour at 50 pascal pressure differential
AF = improper indoor airflow rate fault
AHU = air handling unit
a = coefficient of multivariate polynomial
C = capacitance term: air mass in space multiplied by a multiplication factor in Eq. (4.1)
C_D = heat pump cyclic degradation coefficient
CF = improper outdoor airflow rate (condenser fouling) fault
COP = coefficient of performance
c_p = specific heat of air [$J \cdot g^{-1} \cdot °C^{-1}$, ($Btu \cdot lb^{-1} \cdot °F^{-1}$)]
cfm = volumetric flow rate of air in I-P units ($ft^3 \cdot min^{-1}$)
DUCT = duct leakage fault
EER = energy efficiency ratio [$Btu \cdot h^{-1} \cdot W^{-1}$]
FDD = fault detection and diagnosis
ELA = equivalent leakage area [m^2, (ft^2)]
FSE = fit standard error; equal to the square root of the sum of the squared errors divided by the degrees of freedom
F = fault level [% or dimensionless (fraction)]
F_R = fraction of total return airflow (m_R) from zone 2
F_S = fraction of total supply airflow (m_S) into zone 2
Gross capacity = total capacity (sensible and latent for evaporator) provided by the coil (does not include indoor fan heat)
Gross COP = gross coil capacity divided by outdoor unit power. Outdoor unit power does not include indoor fan power
HP = heat pump
HSPF = heating seasonal performance factor
HVAC = heating, ventilating, air conditioning
Htg = heating
h_i = convective coefficient for exterior of duct [$W \cdot m^{-2} \cdot °C^{-1}$, ($Btu \cdot h^{-1} \cdot ft^{-2} \cdot °F^{-1}$)]
Latent capacity = portion of the cooling capacity that removes moisture (latent) energy (reduces the moisture content (humidity ratio) of the air stream)

LL = liquid line restriction fault
m = number of coefficients or mass flow rate [$kg \cdot s^{-1}$, ($lb \cdot s^{-1}$); or $kg \cdot h^{-1}$, ($lb \cdot h^{-1}$)]
m_R = return airflow to AHU [$kg \cdot s^{-1}$, ($lb \cdot s^{-1}$)]
m'_R = airflow into return duct after accounting for leakage [$kg \cdot s^{-1}$, ($lb \cdot s^{-1}$)], i.e., $m'_R = m_R \cdot (1-F_R)$
m_S = supply airflow from air-handling unit [$kg \cdot s^{-1}$, ($lb \cdot s^{-1}$)]
N = number of data points
NC = presence of non-condensable gases fault
OC = refrigerant overcharge fault, % (or fraction) departure from the correct value
P = pressure [Pa, (mm H_20)]
Q = capacity or heat loss or heat gain [W, ($Btu \cdot h^{-1}$)]
$Q_{internal}$ = internal moisture gains [W, ($Btu \cdot h^{-1}$)]
$Q_{AC,latent}$ = moisture removal by air conditioner [W, ($Btu \cdot h^{-1}$)]
R = thermal resistance in I-P system of units [($h \cdot ft^2 \cdot °F \cdot Btu^{-1}$)]
R(SI) = thermal resistance in SI system of units [$K \cdot m^2 \cdot W^{-1}$]
RH = relative humidity [%]
SC = refrigerant subcooling at the liquid line service valve [°C, (°F)] or excessive refrigerant subcooling fault, % (or fraction) departure from the correct value
SEER = seasonal energy efficiency ratio [($Btu \cdot W^{-1} \cdot h^{-1}$)]
Sensible capacity = portion of cooling capacity that removes sensible energy (decreases the temperature of the air stream)
SHGC = solar heat gain coefficient
SHR = sensible heat ratio (sensible capacity divided by total capacity)
SIZ = heat pump sizing fault, % (or fraction) above or below the correct capacity
T = temperature [°C, (°F)]
T_{ID} = indoor dry-bulb temperature [°C, (°F)]
T_{IDP} = indoor dew-point temperature [°C, (°F)]
T_{OD} = outdoor dry-bulb temperature [°C, (°F)]
TMY3 = data set 3 with typical meteorological year weather data
TXV = thermostatic expansion valve or TXV undersizing fault in cooling
T_{ACout} = average temperature of air leaving AHU [°C]
T^{dep} = outdoor temperature at which a cooling mode TXV opens fully, as calculated by Eq. (3.2) [(°C)]
t = time [s, (s)]
U = overall heat transfer coefficient [$m^3 \cdot h^{-1}$, ($ft^3 \cdot min^{-1}$)]
UC = refrigerant undercharge fault, % (or fraction) departure from the correct value
V = volumetric flow rate [$L \cdot s^{-1}$, ($ft^3 \cdot min^{-1}$)]
VOL = electric line voltage fault
W = power [W, (W)]
W_{ODU} = power of outdoor unit; includes compressor, outdoor fan and control powers [W, (W)]
W_{tot} = total power; includes W_{ODU} and indoor fan power [W, (W)]
w = humidity ratio [$g \cdot g^{-1}$, ($lb \cdot lb^{-1}$)]
w_{ACout} = average humidity ratio of air leaving AHU [$g \cdot g^{-1}$, ($lb \cdot lb^{-1}$)]
X = measured performance parameter
Y = normalized performance parameter

Greek Symbol
Δ = difference

Subscripts
AR = air in the return duct
AS = air in the supply duct
i = indoor or or feature index
in = incoming or inside
inf = infiltration
o = outdoor

out	= outcoming or outside
R	= return duct or refrigerant
r	= reduced
S	= supply duct
sat	= saturation
tot	= total
z1	= zone 1
z2	= zone 2

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APPENDIX A: DUCT LOSSES

Duct losses − leakage and thermal − have been widely evaluated and studied in the field (Cummings and Tooley, 1989; Modera, 1989; Andrews, 1997; Siegel et al., 2003). The impacts of duct leakage and losses are especially significant in homes in the southern and western U.S. where ductwork is often installed outside the conditioned space (e.g., in the attic). Duct losses are complex phenomena where heat is lost to an unconditioned zone (typically the attic), and then in some cases "regained" by reduced heat transfer between the conditioned and unconditioned zones (i.e., heat lost from attic ducts in the winter, tends to warm the attic and reduce heat loss through the ceiling). ASHRAE Standard 152 (ASHRAE 2004) has been developed to characterize the overall impact of thermal conduction and leak losses by determining the overall distribution efficiency (DE) for a system.

We used the leakage model developed for a prior TRNSYS-based simulation study of dehumidification systems (Henderson et al., 2007), as well as a study to evaluate the efficacy of a robust or "fault tolerant" AC unit (Sachs et al., 2009). The model assumes all air leakage and conductions losses are from the ductwork to Zone 2 (the attic), as shown schematically in Figure US-A1. The following is the calculation scheme for the return duct and supply duct.

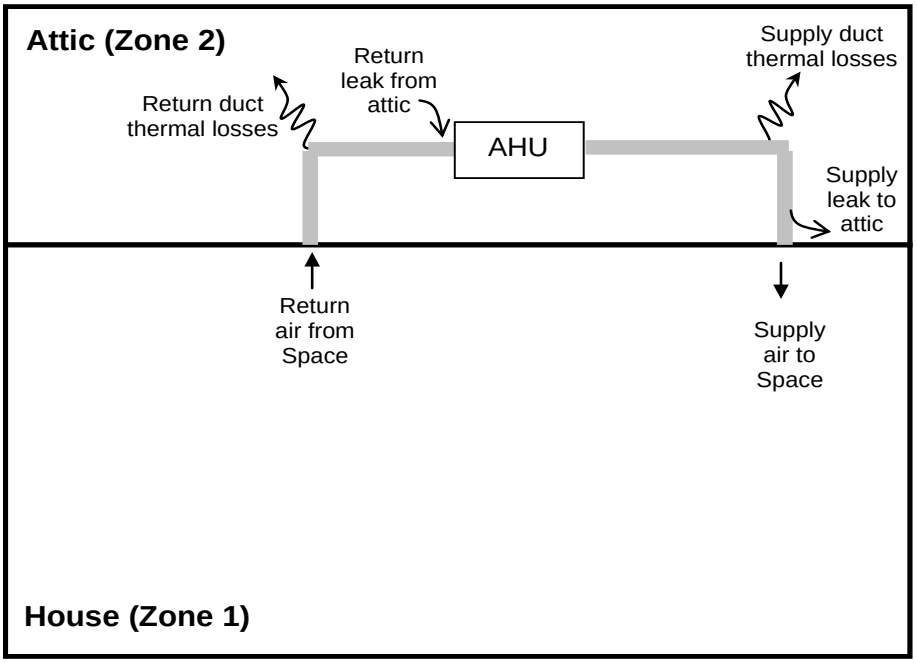


Figure US-A1. Schematic representation of duct leakage in a home with attic ducts

Return Duct

Air from the house zone (Zone 1) enters the return duct. According to evaluations of ASHRAE Standard 152 by Francisco and Palmiter (2000), the temperature change of air in a duct that passes through an unconditioned space at a uniform temperate (T_o) is defined as:

$$\frac{(T_{out} - T_o)}{(T_{in} - T_o)} = e^{-UA/m \cdot c_p} \quad \text{(A1)}$$

Applying Eq. (A1) to our case, the parameters of air arriving at the air handing unit (AHU) are given by:

$$T_{AR} = T_{z2} + (T_{z1} - T_{z2}) e^{-A_R / \left[m'_R \cdot Cp \cdot (R_{duct} + 1/h_i) \right]} \quad \text{(A2)}$$

$$w_{AR} = w_{z1} \quad \text{(A3)}$$

Then the air parameters at the end of the return duct after the thermal losses are:

$$T_{AR}' = T_{AR} \cdot (1 - F_R) + T_{z2} \cdot F_R \quad \text{(A4)}$$
$$w_{AR}' = w_{AR} \cdot (1 - F_R) + w_{z2} \cdot F_R \quad \text{(A5)}$$

The heat gain to Zone 2 from thermal conduction is the same as the heat loss of the return air as it travels through the duct, which is defined as:

$$Q_R = m_R \cdot (1 - F_R) \cdot c_p \cdot (T_{z1} - T_{AR}) \quad \text{(A6)}$$

Supply Duct

Supply air from the AHU unit (i.e., the average for the time step) enters the supply duct. The impact of thermal conduction losses are given by:

$$T_{AS} = T_{z2} + (T_{ACout} - T_{z2}) e^{-A_S / m_S \cdot C_p \cdot R_{duct}} \quad \text{(A7)}$$
$$w_{AS} = w_{ACout} \quad \text{(A6)}$$

A portion of the supply airflow goes to the space (Zone 1), while the balance goes into the attic (Zone 2):

To Space (Zone 1): $m_{S\text{-}space} = m_S \cdot (1 - F_S)$ (A8)
To Zone 2: $m_{S\text{-}z2} = m_S \cdot F_S$ (A9)

The heat gain to Zone 2 from thermal conduction is the same as the heat loss of the supply air as it travel through the duct, which is defined as:

$$Q_S = m_S \cdot c_p \cdot (T_{ACout} - T_{AS}) \quad \text{(A10)}$$

Zone 2 has two impacts from the duct losses:

1) supply air (airflow of $m_{S\text{-}z2}$ at T_{AS} and w_{AS}) enters the zone to condition it,
2) conduction losses from the return duct (Q_R) and the supply duct (Q_S) are added to the zone as a thermal gain.

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Part 3: Appendices for the Overall Annex 36 Effort

Addendum A	Annex 36 Participants
Addendum B	Annex 36 Meeting Dates and Venues
Addendum C	Legal Text for Annex 36

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Addendum A: Annex 36 Participants

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Addendum B: Annex 36 Meeting Dates and Venues

Year	Date	Main Focus	Location
0	30 June 2010	Initial meeting of perspective interested parties	Albuquerque, NM, USA
1	14 Jun 2011	Meeting 1 of the Participants; progress update	Stockholm, Sweden; SP was the host
2	24-25 September 2012	Meeting 2 of the Participants; progress update	Gaithersburg, MD, USA; NIST was the host
3	10-11 October 2013	Meeting 3 of the Participants; report draft review	Fountainebleu, France; EdF was the host
4	12 May 2014	Workshop at IEA Heat Pump conference	Montreal, Canada
Interim web conferences and emails Web meeting 20 March 2012 Web meeting 19 March 2013 Web meeting 11 February 2014			

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Addendum C: Legal Text for Annex 36
Quality Installation / Quality Maintenance Sensitivity Studies
(Avoiding Efficiency Degradation due to Poor Installations and Maintenance)

1. Background

A significant opportunity for improving heat pump operating performance (capacity, efficiency, etc.) is related to how well the equipment is actually installed and subsequently maintained in the field. A difficulty in promoting sound installation and maintenance practices is that the marketplace often incentivizes 'cutting corners' so as to keep first-cost pricing low, resulting in poorly performing equipment and poor value received by the customer. Generally, an underlying cause of this predicament is that the various industry participants (e.g., OEMs, distributors, designers, installers, etc.) – as well as the ultimate building owners/operators/customers – do not appreciate how small deviations in installation and maintenance impact equipment performance.

It is generally recognized that heat pump equipment suffers significant performance loss depending on how the equipment is installed and subsequently maintained. Some commonly noted problems caused by incorrect installation and maintenance practices for both air-to-air and water-to-air heat pumps in the US are:

- improper refrigerant charges (up to 15% off),
- incorrect airflow over the coil (up to 50% off design),
- oversized equipment (routinely, 50% oversized … and often considerably more),
- leaky ducts (up to 50% of airflow),
- etc.

This causes heat pumps to operate inefficiently and waste considerable energy. However, it is unclear whether small variances within a given field-observed practice are significant, whether the deviations have an additive effect on heat pump performance, and whether the attribute deviations (in various equipment applications and geographical locations) have a larger impact than others. If this information is known, better attention, resources, and effort can be focused on those parameters that are most important in the installation, and maintenance of heat pump equipment.

Additionally, improved understanding will help position consumers and building owners / operators to consider the complete value-to-cost equation, not merely the "first price," when making HVAC equipment purchasing, installation, and maintenance decisions. Customers who select high performance HVAC equipment – along with quality installation (QI) and quality maintenance (QM) practices – enjoy enhanced comfort, reduced energy usage, improved occupant productivity, and enhanced occupant safety.

2. Description of Technical Sector

The HVAC industry does not have a good set of quantitative information on the relative value and importance of one field installation (or maintenance) aspect over another. There have been isolated studies that looked at existing equipment in the field and the impact that corrective fixes provide. However, these result in anecdotal data that provided limited real information on the trade-off impacts associated with specific design, installation, and maintenance practices.

By evaluating heat pump faults commonly found in operating equipment, and assessing the relative capacity and efficiency degradations, stakeholders will be better positioned to understand how installation and maintenance practices affect heat pump performance.

3. Objectives and Scope

The desired end-product objectives are:

- Develop information for use by key stakeholders in industry (HVACR and construction trades), government (policy makers), and building sector (owners/operators).
- Produce reliable data to position each participating Annex country to evaluate its pertinent industry standards and practices for the heat pump system types of primary interest to ensure optimum heat pump performance.
- Reduce usage of energy and emissions of greenhouse gases by encouraging use of quality heat pump installation and maintenance practices.

These objectives have parallel interest under the IEA Framework of Implementing Agreements as the intent is to identify, quantify, and deliver information to key industry stakeholders and policy makers, and to provide pertinent resources to building owners and operators. The objectives will be achieved by independent studies and investigations performed by the country-specific participants. The main output of this Annex is information sharing on heat pump installation and maintenance aspects so that better cross-country understanding is achieved to reduce energy consumption (and the related CO_2-emissions) while satisfying the needs of building owners and operators.

4. Means

Each participating Annex partner will focus on ***those heat pump types and applications that are in general usage within their residential and commercial building sectors.*** It is anticipated that each country's recognized standards, industry practices, and measurement instrumentation approaches, will serve as the basis for the individual Annex partner efforts.

In undertaking the sensitivity analyses, each individual Annex partner will be seeking to quantify impacts on system performance (e.g., capacity, energy utilization, etc.) related to varying QI and QM practices /attributes for their varied equipment applications of interest.

Tasks to characterize the impacts and tradeoffs associated with varying levels of QI and QM that each country is to observe are:

Task 1: Critical literature survey
Undertake literature review and critical analysis of the results from prior related research to identify QI & QM metrics, and to secure quantitative and qualitative impacts of various deviations from the specified levels recommended in industry's relevent QI & QM standards. This may include benchmarking of current QI/QM standards and practices in place in each participating country.

Task 2: Identify sensitivity parameters
Building on the literature search, and working with pertinent country experts, develop a consensus of the QI & QM elements to be included in the participant's sensitivity investigation. The table below provides examples of possible elements, metric ranges, and increments, based on current plans for the US QI/QM investigation (focused on the prevailing central air distribution approach used in US systems). Other participants may modify this listing of parameters based upon the heat pump and distribution systems of most interest to their industry and end users.

Possible Example for the QI and QM Sensitivity Analysis		
Attribute	**Possible Reporting Metric range**	**Possible Reporting Increments**
Equipment-related		
Equipment sizing	Nominal heating or cooling capacity (kW or Btu/hr) for building: 90 – 200% of load calcs	10 – 25% intervals
Airflow across the coil	CFM: 50 – 125% of design	10 – 25% intervals
Refrigerant charge	TXV: -6 to 6ºC of subcooling	1-2 ºC intervals

	Orifice: -6 to 6ºC of superheat	
Electrical	Voltage: 80 – 120% of rated voltage	10% intervals
Equipment matching	Mismatched indoor heat exchanger with mismatched refrigerant metering device.	Two scenarios: ~30% too large and ~30% too small
Air distribution system		
Duct leakage (unconditioned space)	Leakage %: 0 – 50% of supply air flow	10% intervals
Airflow balance	Flow imbalance to room (based on design)	Two scenarios: ~30% too low and ~30% too large
Additional examples or heat pump operating faults could include: – compressor and/or reversing valve leakage – condenser air blockage – evaporator air blockage – liquid line restriction – refrigerant undercharge/overcharge – non-condensable gas in system – failure to clean coils, – dirty ducts, – filter loading/changeout – viability and impact of varied measurement approaches (instrumentation, procedures, etc.)		

Task 3: Modeling and/or laboratory-controlled measurements
Computer modeling and/or laboratory-controlled experiments to be undertaken to verify the results provided by earlier researchers, to fill-in 'holes' in regard to quantifying the impacts of QI & QM vs. non-QI & QM applications, and to better understand the singular and additive impacts on equipment effectiveness.

Task 4: Simulations on seasonality impacts
Simulations are to be undertaken to determine the role that seasonal temperature differences have on the varied QI & QM elements and the resultant impact on heat pump performance. This modeling of different geographic temperatures impacts within a country or region (i.e., different outdoor temperature and humidity conditions) help to identify regional effects if any.

Task 5: Report and information dissimenation
Each participant's final report will address the results of the tasks above. As appropriate, the report(s) should also address the viability and impact of varied measurement methods (instrumentation, procedures, etc.) to cost-effectively undertake various QI & QM measurements (e.g., refrigerant charge, airflow, duct leakage, etc.) in the field.

5. Target Audience and Benefits

The sectors targeted for this Annex include:
- HVAC practitioners responsible for designing, selecting, installing, and maintaining heat pump systems in varied applications.
- Building owner/operators interested in achieving improved comfort conditioning and efficiency performance from their HVACR equipment.
- Entities charged with minimizing energy utilization (i.e., utilities, utility commissions, energy agencies, legislative bodies, etc.) in varied heat pump applications and geographic conditions.

Improved heat pump installation and maintenance will result from increased awareness of HVAC contractors/installers, building owners/operators, and consumers about the benefits provided by following recognized quality installation and maintenance practices.

6. Time Schedule

The Annex commenced 1 November 2010, and will remain in force until 30 November 2013. Within the limits of the terms of the Agreement, this Annex may be extended by two or more of the participating countries (Participants), acting in the Executive Committee and taking into account any recommendation of the IEA's Committee on Energy Research and Technology concerning the terms of this Annex 36. Extensions shall thereafter apply only to those Participants.

The following is a tentative work schedule for the different tasks – exact timing will depend upon the actual start date of the Annex.

Start Date	End Date	Activity
November 2010	April 2011	Task 1 – Critical literature survey
May 2011	October 2011	Task 2 – Identify sensitivity parameters
November 2011	July 2012	Task 3 – Modelling and/or lab-controlled measurements
August 2012	April 2013	Task 4 – Simulations on seasonal impacts
May 2013	November 2013	Task 5 – Report and information dissemination

7. Deliverables

The deliverables of the Annex are:

a) An Internet website (linked to the HPC-site), that will serve as the participants' portal for sharing their country efforts and viewing the work done by other countries.
b) Progress reports to the HPC four times annually for publication in the Newsletter, and semi-annually to the HPP Executive Committee (May and November).
c) An annual summary report for inclusion in the HPP Annual report describing the work carried out under the Annex.
d) Final report consolidating the salient findings of the individual country inputs. This report will restate the objectives of the Annex, its key findings, a description of the results sensitivity analysis undertaken by the Annex Participants, and recommendations for further study. **NOTE** – The Annex Final Report will remain confidential (embargoed) among the Participants for a period of eighteen (18) months after submission to the HPP Executive Committee. However, the Executive Summary of the final report shall be made available for general distribution to all IEA HPP countries immediately after approval of the final report by the Executive Committee.

8. Funding

a) Participant's Financial Obligations: Each Participant (country) will bear the costs of its own participation in the Annex, including any necessary reporting, travel costs to undertake their portion of the effort, and costs to organize and hold working meetings hosted by the country. Approximately one face-to-face working meeting per year is anticipated during the course of the Annex (schedule to be determined by the Participants). It is envisioned that many of the Annex activities can be undertaken via web-conferencing and e-mail communications.
b) Publications: The cost/effort to prepare the draft Final Report and summary assessments described in paragraph 7 above ('Deliverables') shall be equally shared by all the Participants.
c) Each Participant shall make a direct financial contribution to the Operating Agent to cover coordination, report submission/review/consolidation, and other Annex-related costs.

The table below shows the fees per Participant, based upon varying numbers of Participants. Each Participant's fee shall be paid in 3 annual instalments.

Number of participants	Participants' fees		
	2010	2011	2012
2	$ 10000	$ 10000	$ 10000
3	$ 8500	$ 8500	$ 8500
4	$ 7500	$ 7500	$ 7500
5	$ 6000	$ 6000	$ 6000
>5	$ 5000	$ 5000	$ 5000

Each Participating country's fee shall be paid in US$, as shown in the above Table.

9. Specific Obligations and Responsibilities of the Participants

a) Each Participant shall nominate a lead representative to participate in the work under this Annex and act as the point of contact (POC) with the Operating Agent. At his/her discretion, the POC may appoint other individuals to lead the Participant's work in each of the Annex Tasks as defined in Section 4.
b) Each Participant shall carry out the equivalent of at least 6-12 person months of task-sharing work during the Annex period, unless otherwise agreed by the Participants.
c) Each Participant shall contribute to the working meetings and workshop(s) on the results achieved through the activities conducted under this Annex, including the identification of speakers and participants.
d) Each Participant shall make a direct financial contribution to the Operating Agent to cover co-ordination and report preparation expenses and other Annex related costs.
e) Each Participant shall provide a Country Report as identified in Section 4 (Task 5) and shall contribute to the deliverables identified in Section 7.

10. Specific Obligations and Responsibilities of the Operating Agent

The Operating Agent shall:

a) Develop, in cooperation with the Participants, a detailed work schedule, a framework for the Country Reports, and a budget for all the activities carried out under this Annex, including methodology and time schedule.
b) Provide the Executive Committee with periodic reports describing the progress of the work being accomplished under the Annex.
c) Deliver the results as described in Section 7 ('Deliverables').
d) Provide to the Executive Committee, within six months after completion of all the Annex work, a draft of the Final Report for its approval and transmittal to IEA headquarters.
e) In co-ordination with the Participants, use its best efforts to avoid duplication with activities of other related programs and projects implemented by or under the auspices of the Agency or by other competent bodies.
f) Provide the Participants with necessary guidelines for the work they carry out, assuring minimum duplication of effort.
g) Co-ordinate the efforts of all Participants and ensure the flow of information within the Annex.
h) Provide general administration.

The IEA Heat Pump Centre will assist in the establishment of the Annex, in the organisation of the workshop(s), the publication of the proceedings, and the publication of the Final Report.

11. Information and Intellectual Property

a) *Executive Committee's Powers.* The publication, distribution, handling, protection and ownership of information and intellectual property arising from this Annex, shall be determined by the Executive Committee, acting by unanimity, in conformity with this Annex.
b) *Right to Publish.* The Participants shall have the right to publish information provided to, or arising, from their contribution to the Annex under Section 9b, except for proprietary information, as defined in paragraph (c) below.

c) *Proprietary Information.* For the purposes of this Annex, proprietary information shall mean information of a confidential nature such as trade secrets and know-how (for example, computer programmes, design procedures and techniques, chemical compositions of materials, or manufacturing methods, processes or treatments) which is appropriately marked provided that such information:
   1) Is not generally known or publicly available from other sources;
   2) Has not previously been made available by its owner(s) to others without obligation concerning its confidentiality; and
   3) Is not already in the possession of the recipient Participant(s) without obligation concerning its confidentiality.

   It shall be the responsibility of each Participant supplying proprietary information, and of the Operating Agent, to identify such information as proprietary and to ensure that it is appropriately marked. The Participants and the Operating Agent shall take all necessary measures in accordance with this paragraph, the laws of their respective countries, and international law, to protect the proprietary information provided to, or arising from this Annex.

d) *Production of Relevant Information by Governments.* The Operating Agent should encourage the governments of all Agency Participating Countries to make available, or identify to the Operating Agent, all published or otherwise freely available information known to them that is relevant to the Annex.

e) *Production of Relevant Information by Participants.* Each participant agrees to provide to the Operating Agent all previously existing information, and information developed independently of the Annex, which can assist, or is needed by the Operating Agent, to carry out its functions in this Annex, which is freely at the disposal of the Participants, and the transmission of which is not subject to any contractual and/or legal limitations, under the following conditions:
   1) The Participant will make such information available, at its own costs, provided that such costs are not substantial;
   2) If substantial costs are necessary for the Participant to make such information available, the Operating Agent and all Participants will determine the charge of the costs for each participant, upon approval of the Executive Committee.

f) *Use of Confidential Information.* If a Participant has access to confidential information which would be useful to the Operating Agent in carrying out the studies, assessments, analysis or evaluations described in this Annex, such information may be communicated to the Operating Agent, but shall not become part of any report or other form of documentation issued as part of this Annex, nor shall it be communicated to the other Participants, except as may be agreed between the Operating Agent and the Participant who supplies such information. This information has to be marked clearly as "confidential".

g) *Acquisition of Information for the Annex*. Each Participant shall inform the Operating Agent of the existence of information that can be of value to the Annex, but which is not freely available; and each Participant shall endeavour to make such information available to the Annex under reasonable conditions, in which event the Executive Committee may, acting unanimity, decide to acquire each information.

h) Reports on Work Performed under the Annex. The Operating Agent shall, in collaboration with the other Participants, prepare reports on all work performed under the Annex and the result thereof, including studies, assessments, analysis, evaluations and other documentation, but excluding proprietary information, in accordance with paragraph 11(c) above. **NOTE** – The Annex Final Report will remain confidential (embargoed) among the Participants for a period of eighteen (18) months after submission to the HPP Executive Committee. However, the Executive Summary of the final report shall be made available for general distribution to all IEA HPP countries immediately after approval of the final report by the Executive Committee.

i) *Copyright.* The Operating Agent, or each Participant for its own results, may take appropriate measures necessary to protect copyrightable material generated under this Annex. Copyright obtained shall be the property of the Operating Agent, for the benefit of the Participants provided; however, that Participants may reproduce and distribute such material, but shall not publish it with a view to profit, except as otherwise provided by the Executive Committee. The Contracting Parties understand and agree that the name, acronym and emblem of the IEA has been notified to the World Intellectual Property Organisation (WIPO) Secretariat according to Article 6 of the Paris Convention for the Protection of Industrial Property, as amended on 28

September 1979. The Contracting Parties further understand and agree that the OECD/IEA shall retain the copyright to all IEA deliverables, materials or publications published, or to be published by the IEA, or jointly by the IEA and a third party to this Annex. Should the Contracting Parties use any such deliverables, materials or publications they shall give full acknowledgement to the OECD/IEA as being the source of the material with a copyright notice in the following form: © OECD/IEA, (year of publication).

j) Authors. Each Participant shall, without prejudice to any rights of authors under its national laws, take necessary steps to provide the cooperation from its authors required to carry out the provisions in this paragraph. Each Participant shall assume the responsibility to pay awards or compensation required to be paid to its employees according to the laws of its country.

12. Operating Agent

The Air Conditioning Contractors of America (ACCA), and the National Institute of Standards and Technology (NIST), in collaboration with Oak Ridge National Laboratory (ORNL) and the U.S. Department of Energy (DOE), are designated as Co-Operating Agents.

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13. Participants in this Annex

The Contracting Parties which are Participants in this Annex are the following:

Organization	**Country**
Ademe	France
The Swedish National Energy Administration	Sweden
Department of Energy and Climate Change	UK
Department of Energy	USA

14. Research organizations participating in this Annex

The contracting parties are the ones signing the contract, but others may carry out the research work, states those parties in the table below.

Organization	Contact person	Country	Email	Website	Annex NT leader (Y/N)
NIST	Piotr Domanski	USA	piotr.domanski@nist.gov		
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SP	Roger Nordman	Sweden	roger.nordman@sp.se		
KTH	Jan-Erik Nowacki, Björn Palm	Sweden	nowacki@algonet.se Bjorn.Palm@energy.kth.se		
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