DYNAMIC SIMULATION METHODS OF HEAT PUMP SYSTEMS AS A PART OF DYNAMIC ENERGY SIMULATION OF BUILDINGS

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ABSTRACT

The paper presents simulation modelling and validation of heat pump systems as a part of dynamic energy simulation of buildings. The study focuses on modelling of heat pump systems with IDA Indoor Climate and Energy (IDA ICE) simulation software with field-testing for simulation model validation. The studied heat pump systems included a ground source heat pump, an air-to-water heat pump and an exhaust air heat pump. The results of the study indicated that the studied dynamic simulation tool is an accurate and reliable method to simulate the energy performance of the studied heat pump systems. Simulation results of different case studies indicated that the installation method of the heat pump system has a relatively significant impact on the operation and energy performance of the system. According to the study, modern heat pump systems on the market can be simulated accurately in different operating conditions by calibrating the default heat pump models of the Early Stage Building Optimization (ESBO) Plant model.

INTRODUCTION

The performance and versatility requirements of simulation tools are rapidly increasing as more features are required to accurately and reliably simulate modern HVAC and building services systems. Since the adoption and implementation of the recast EPBD-directive (2010/31/EU), on-site renewable energy production systems and their utilization in the energy production of buildings to reduce delivered primary energy consumption must be studied more closely. Simulation tools have to cope with these new demands and requirements or a suitable method must be selected to take the on-site renewable energy production systems into account.

According to recent studies carried out by Häkämies et al. 2015 and Niemelä 2015, the more simplifications are used in the simulation of different energy systems, the more inaccurate and unreliable the results of the simulation will be, when the energy performance requirements of the EPBD-directive (2010/31/EU) and the simulation of on-site renewable energy production systems are discussed. A general problem with the simulation tools used at the moment is that the energy performance of the heat pump systems and other renewable energy production systems is difficult to be simulated accurately. The main reason for this is that the simulation tools are typically only used to simulate the net energy demand of buildings and to study and verify the indoor climate conditions. More detailed and versatile simulation tools are required in the future to simulate the more complex energy production systems in addition to the net energy demands. There are several dynamic simulation programs on the market, e.g. TRNSYS, EnergyPlus and IDA ICE, which can be used to simulate various energy production systems in addition to the dynamic simulation of net energy demand of buildings. However, the general conclusion at the moment is that the performance, accuracy and reliability of these programs as a simulation method of renewable energy production systems requires further testing and validation to correspond to the modern rapidly evolving energy systems.

The objective of this study was to test and validate a simulation method used to simulate the energy performance of different heat pump systems as a part of dynamic energy simulation of buildings. The studied energy simulation tool was IDA ICE (IDA Indoor Climate and Energy, versions 4.6.1, 4.6.2 and 4.7) and the recently implemented Early Stage Building Optimization (ESBO) Plant model of IDA ICE, which is used to simulate the on-site renewable energy production systems. The studied heat pump systems included a ground source heat pump (GSHP), an air-to-water heat pump (A2WHP) and an exhaust air heat pump (EAHP). The GSHP and A2WHP systems are included in the ESBO Plant model by default. In this study, the EAHP system was developed during the study, as the default ESBO Plant model does not include an EAHP system and as a result, the tested EAHP system was implemented in IDA ICE ESBO Plant by EQUA Simulation.

The main target of the study was to test the functionality, usability and reliability of dynamic simulation of different heat pump systems and to compare the performance of the simulated systems to real systems. To validate the simulation results, extensive testing and calibration was carried out with the simulation models of different systems. Furthermore, the main simulation principles of the heat pump models in ESBO Plant were closely studied. The first step of the study was to collect the energy performance data of real heat pump systems from several manufacturers to compare the simulation results to real measured data. The second step was the detailed calibration process of the simulated heat pump models, which required both energy simulations by using the IDA ICE ESBO Plant and extensive multi-objective optimization by using the new Multi-Objective Building Optimization (MOBO) tool. The primary optimization method used in the study to carry out the main heat pump calibrations was MOBO. The main features and applications of MOBO are described in detail in a study carried out by Palonen et al. 2013.

The study presents the basic theory regarding the simulation of heat pump systems in IDA ICE (version 4.7) and a method to perform a detailed calibration of the GSHP and A2WHP models according to measured product-specific performance data provided by heat pump manufacturers to correspond to real heat pump systems. Furthermore, an additional calibration method is also presented to carry out a detailed calibration of heat pump models in specific case study buildings for improved accuracy and reliability. Additional validation of the simulated heat pump models to correspond to real measured field-testing data is also presented for the studied GSHP and A2WHP types.

SIMULATION AND EXPERIMENT METHODS

Basic theory of modelling heat pumps in IDA ICE

IDA ICE simulation software allows detailed modelling of multi-zone buildings including HVAC systems, solar and internal loads, outdoor climate conditions and performs simultaneous dynamic simulation of both heat transfer and mass flows. The performance and features of IDA ICE has been tested extensively over the years in numerous studies carried out by Sahlin 1996, Björsell et al. 1999, Travesi et al. 2001, Achermann et al. 2003 and Loutzenhiser et al. 2007 for example. A detailed analysis of a parameter estimation study of a water-to-water heat pump model implemented in earlier versions of IDA ICE to simulate the performance of modern GSHP systems has also been carried out by Salvalai 2012.

The revised implementation of the ESBO Plant is integrated in IDA ICE to model different heat pump systems according to real specifications of the systems. EQUA Simulation AB has modelled the operation of heat pump systems in IDA ICE by using Equations (1-2).

$$Q_{cond} = P_{comp} \cdot COP \tag{1}$$

$$Q_{evap} = P_{comp} \cdot EER \tag{2}$$

The operation of heat pump systems is divided into full and partial load properties by using specific partial load methods. The performance of heat pump systems at full load properties are calculated by using constant factors A, B, C, D, E and F, which are calculated from rating conditions and from the type of heat pump or chiller system. The partial load properties are determined according to the full load properties. The full load properties are calculated by using Equations (3-5).

$$P_{compF} = D \cdot exp(E \cdot T_{evap} + F \cdot T_{cond}) \tag{3}$$

$$Q_{evapF} = A \cdot exp(B \cdot T_{evap} + C \cdot T_{cond}) \tag{4}$$

$$P_{comp} = c_{ctr} \cdot P_{compF} \tag{5}$$

The partial load performance can be determined in three different ways depending on the selected type of heat pump system. Equations (6-8) are used in the calculation and calibration method of partial load performance in this study.

$$PLF = 1 + G \cdot \ln(PLR) = \frac{EER}{EER_F} \tag{6}$$

$$PLR = \frac{Q_{evap}}{Q_{evapF}} \tag{7}$$

$$COP = EER + 1 \tag{8}$$

In addition, logarithmic temperature differences in evaporator and condenser at rating conditions are needed. The temperatures on the condenser (hot) and evaporator (cold) sides on partial loads are assumed to obey predefined differential equations. The condensation and evaporation temperatures are calculated by using Equations (9-10).

$$T_{cond} = \frac{T_{cIn} - T_{cOut} \cdot exp\left(\frac{T_{cOut} - T_{cIn}}{\Delta T_c}\right)}{1 - exp\left(\frac{T_{cOut} - T_{cIn}}{\Delta T_c}\right)} \tag{9}$$

$$T_{evap} = \frac{T_{eIn} - T_{eOut} \cdot exp\left(\frac{T_{eOut} - T_{eIn}}{\Delta T_e}\right)}{1 - exp\left(\frac{T_{eOut} - T_{eIn}}{\Delta T_e}\right)} \tag{10}$$

The evaporation and condensation temperatures at rated conditions ($T_{condRat}$, $T_{evapRat}$) can be calculated by using Equations (9-10). The parameters A and D used in the calibration process are defined according to Equations (11-13).

$$D = P_{compRat} \cdot exp(-E \cdot T_{evapRat} - F \cdot T_{condRat}) \tag{11}$$

$$A = P_{compRat} \cdot exp(-B \cdot T_{evapRat} - C \cdot T_{condRat}) \cdot EER_{Rat} \tag{12}$$

$$P_{compRat} = \begin{cases} \frac{Q_{condRat}}{COP_{Rat}} \\ \frac{Q_{evapRat}}{EER_{Rat}} \end{cases} \tag{13}$$

The formula presented above in Equation (13) is used with heat pump systems and the formula presented below (Equation (13)) is used with chiller systems. The desired part load performance mode can be selected manually in IDA ICE (version 4.7) by the user by adjusting the following heat pump model parameters:

- $P_{min} \geq 0$: Partial load model with parasitic sleep power *PPar* or factor *abs(Cc)* is used,
- *PPar* > 0: Parasitic sleep power *PPar* is used, P_{min} used as min value of *COP*,
- PPar < 0: Degradation factor *abs(Cc)* is used

- $Cc > 0$: Heat pump case
- $Cc < 0$: Chiller case, P_{min} used as min value of *COP* (HP) or *EER* (Chiller),

- $P_{min} < 0$: Partial load model with part load exponent *G* is used.

The fourth simulation mode, where the partial load exponent *G* is used, was used in the partial load simulations of heat pump systems in this study. Other partial load modes can also be used and in this case the optimal values for the partial load simulation parameters have to be determined according to the selected simulation mode. Equation (6) has been derived by Jan-Erik Nowacki and implemented in IDA ICE by EQUA Simulation Ab to simulate the partial load performance of different heat pump systems. Eq. (6) is based on real measured field data of different compressor types, which is used to form an approximated method (Eq. (6)) to calculate the partial load performance of heat pump units equipped with specific types of compressors. According to extensive testing and validation of heat pump simulations using the ESBO Plant model, the partial load model with the part load exponent G is recommended to be used for simulation of modern on/off or inverter-type -controlled heat pump systems.

It is highly recommended to use the ideal heat storage tank model in the ESBO Plant model, when heat pump systems are simulated. A single ideal heat storage tank model corresponds accurately to a real modern heat pump system with two separate heat storage tanks and an exchange valve: one high-temperature tank for DHW heating and one low-temperature (temperature controlled according to ambient air temperature) tank for heating of room spaces and ventilation systems. The operation of the ESBO Plant model, including the operation of the ideal heat storage tank model, has been defined in detail in a recent study carried out by Niemelä 2015.

Calibration of a GSHP system according to measured manufacturer product data

The performance of a real GSHP system was simulated with the default ESBO Plant GSHP system model and with a calibrated GSHP system model. The results of the simulations were compared to the real measured performance data provided by system manufacturer. The selected heat pump system used in the simulations was Carrier 61WG-090 brine-to-water heat pump system with scroll-type compressor. The nominal power output of the selected system is 87.4 kW at 0/35 °C (brine inlet temperature 0 °C / water outlet temperature 35 °C) and 100.9 kW at 5/35 °C operating conditions. The heat pump system was simulated in multiple operating conditions to determine the overall performance of the simulated heat pump model. The selected operating conditions and the performance of the studied heat pump system are presented in Table 1. A total of eight operating conditions were selected in the energy performance simulations.

Table 1
The operating conditions and the performance of the studied GSHP system

Operating point (inlet brine temperature / outlet water temperature to the heating system), according to EN14511	COP according to EN14511, measured by manufacturer
0/35 °C	4.20
0/45 °C	3.31
0/55 °C	2.60
0/60 °C	2.31
5/35 °C	4.62
5/45 °C	3.70
5/55 °C	2.94
5/60 °C	2.62
SPF-value of all operating points, 12.5 % of total time is operated at each point	3.2875

The studied operating conditions were selected to correspond to a typical heat pump system, which is used as a hybrid system including boreholes and exhaust air heat recovery in the brine system. Brine temperature measurements in real implementations conducted by Senera Oy indicate that the average inlet brine temperature is 0...5 °C in aforementioned hybrid installations.

Only the performance data of a single operating point can be fed in IDA ICE. The performance of other operating conditions are calculated from the input data using Equations (1-13). The performance data of different operating conditions can be adjusted by altering the calibration parameters *A*-*F* (Equations (3-4) and (11-12)) to correspond to performance data of a specific heat pump system.

The calibration of the overall performance of a heat pump system using multiple operating points can be carried out by using the Seasonal Performance Factor (SPF), which takes the operation of a heat pump system in different operating conditions into account. The SPF is defined as the ratio of total heat energy produced and total electrical energy used by the heat pump system in a longer period of time, typically in one year. The SPF-value of a heat pump system is calculated by Equation (14).

$$SPF = \frac{\sum Q_{cond}}{\sum P_{comp}} \quad (14)$$

The calibration setup and the calibration parameters of the GSHP simulation model is presented in Figure 1. After the studied simulation model has been configured in IDA ICE, calibration parameters *A*-*F* should be determined. The easiest and most efficient way to determine the parameters is to use a modern optimization tool with efficient optimization algorithms. Equations (1-14) can be solved in Excel, e.g. by using Excel Solver, but a far more recommendable option is to determine the calibration parameters directly in IDA ICE by running simulation-based optimization (SBO) analysis, where the optimized design variables are the calibration parameters *A-F* and the optimized objective function is the SPF-value of the studied heat pump system over a specific period of time. MOBO was used as the optimization engine of the SBO analysis in this study. After the IDA ICE model and the MOBO setup have been configured, the optimal values for calibration parameters can be determined by running a SBO analysis using a genetic algorithm (NSGA-II) with approximately 700 - 1 000 simulations. It takes approximately 10-15 minutes to carry out the analysis, depending on the computer performance.

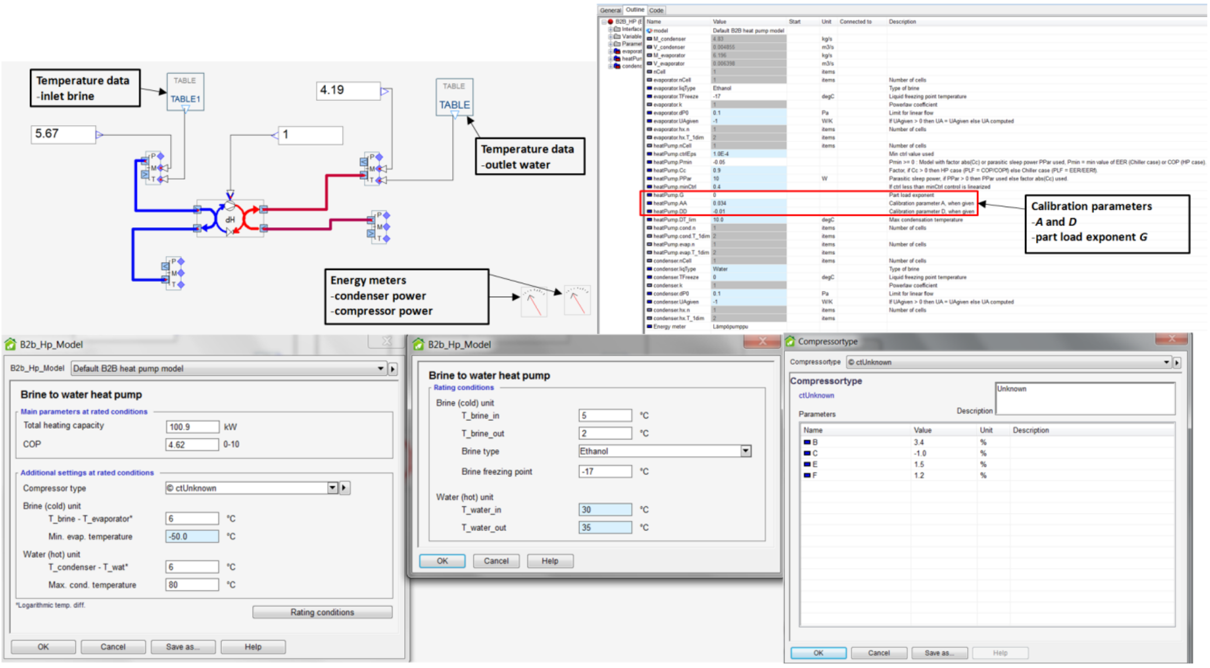

Figure 1 The calibration setup of the GSHP system model (top left), calibration parameters A, D and G (top right), calibration parameters B, C, E and F (bottom right), main parameters and settings at rated conditions (bottom left) and input data of rating conditions (bottom center)

<u>RESULTS AND DISCUSSION</u>

GSHP system calibration

The studied GSHP system is an on/off -model with two individual scroll compressor units. This means that the heat pump unit is always either off or operating at full capacity. In this case, the part load exponent G can be set to 0 and it doesn't have to be determined. When the calibration is carried out according to the EN14511-specific manufacturer product data, the time period of the simulation is determined so that each operating point is run equally long, e.g. 10 or 100 hours. In this scenario, the calibration parameters are determined so that each operating point is equally weighted.

Figure 2 presents the performance of the default GSHP model, where only the main parameters of the model, including logarithmic mean temperature differences of heat exchangers, heat pump capacity, COP and rating conditions, are set according to the studied heat pump system. All of the heat pump calibration parameters (*A-F*) are default values of the ESBO Plant that are meant to be used with a typical heat pump system.

In addition, the performance of the calibrated heat pump model, where the calibration parameters are determined from the SBO analysis, is also presented in Figure 2. The performance of the heat pump system is defined according to the 5/35 °C operating point in IDA ICE in both cases (default and calibrated).

According to the results of the simulations, the performance of the default heat pump model corresponds accurately to the performance of the real heat pump system only at the operating point, which was used to feed the input data of the simulated system (5/35 °C). As a conclusion, the default heat pump model generally gives a little too good energy performance compared to a real system. However, it is important to notice that no default parameters should be accepted in a simulation of a real heat pump system without a critical review, as the default heat pump parameters are determined for just one of many plausible heat pumps.

The calibrated heat pump model corresponds to the measured data fairly accurately. The optimal values of the calibration parameters *A-F* depend heavily on the simulated case. Furthermore, the same performance can be achieved with a little different combination of values.

The simulations also indicated that the operating point used to feed the input data in IDA ICE has a little effect on the overall performance of the simulated heat pump model. This is presented in

Figure 3, where the calibration of parameters *A-F* was carried out according to the operating point 5/45 °C, which was used to feed the input data in IDA ICE.

According to the simulations, a little better overall simulation model accuracy can be achieved by carrying out the calibration using operating point to feed the input data, which represents average operating conditions of several operating points, e.g. 5/45 °C instead of 5/35 °C or 5/55 °C.

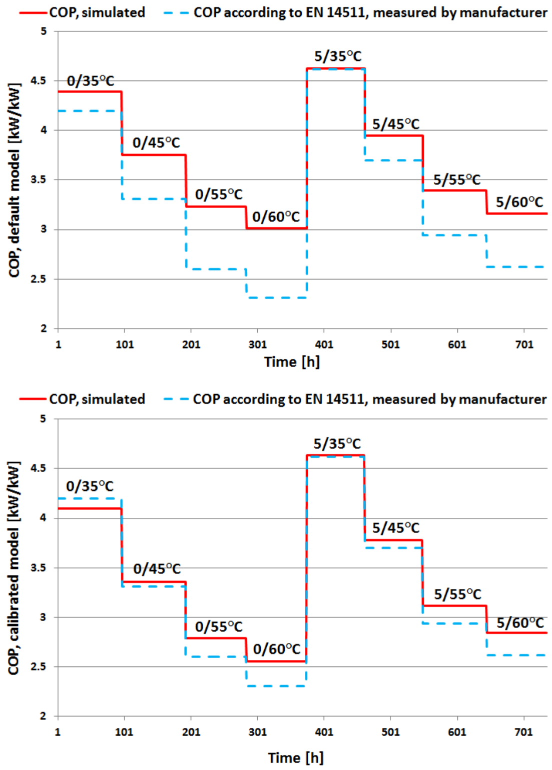

Figure 2 The performance of the simulated GSHP system model, the default heat pump model (second from the bottom) and the calibrated heat pump model (bottom) according to 5/35 °C operating point presented

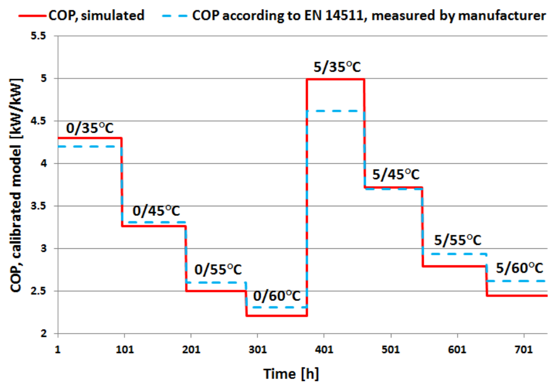

Figure 3 The energy performance of the simulated GSHP system model, calibrated heat pump model according to 5/45 °C operating point

The calibration parameters *A-F* and the SPF-values for the default and calibrated heat pump models (5/35°C and 5/45°C), determined from the SBO analysis, are for the Carrier 61WG-090 heat pump unit:

- ***A***: default 0.034, calibrated (5/35°C) -1.520, calibrated (5/45°C) 2.5855;
- ***B***: default 3.4, calibrated (5/35°C) 2.0479, calibrated (5/45°C) -1.3343;
- ***C***: default -1.0, calibrated (5/35°C) -1.1584, calibrated (5/45°C) -2.8495;
- ***D***: default -0.01, calibrated (5/35°C) 1.8719, calibrated (5/45°C) 0.8162;
- ***E***: default 1.5, calibrated (5/35°C) -1.7546, calibrated (5/45°C) -4.2571;
- ***F***: default 1.2, calibrated (5/35°C) 1.84262, calibrated (5/45°C) 2.1359;
- ***SPF***: default 3.627, calibrated (5/35°C) 3.287, calibrated (5/45°C) 3.287.

Calibration of an A2WHP system according to measured manufacturer product data

The selected heat pump system used in the simulations was Mitsubishi Electric CAHV-P500YA-HPB air-to-water heat pump system with scroll-type compressor and inverter-type control. The nominal power output of the selected system is 63.2 kW at 7/45 °C (intake air temperature 7°C / water outlet temperature 45°C) operating conditions. However, due to the inverter-type control, the power output of the system can be controlled between 10-63 kW. The selected operating conditions and the performance of the studied heat pump system are presented in Table 2. A total of eight operating conditions were selected in the energy performance simulations to correspond to the cold climate heating season conditions.

Table 2
The operating conditions and the performance of the studied A2WHP system

Operating point (intake air temperature / outlet water temperature to the heating system), according to EN14511	COP according to EN14511, measured by manufacturer
-15/45 °C	4.20
-15/55 °C	3.31
-7/45 °C	2.60
-7/55 °C	2.31
0/45 °C	4.62
0/55 °C	3.70
7/45 °C	2.94
7/55 °C	2.62
SPF-value of all operating points, 12.5 % of total time is operated at each point	2.3155

According to the manufacturer's data, the energy performance of the heat pump system doesn't decrease when partial load operation is applied due to the inverter-type control, which means that the part load exponent G can be set to 0 again (definition according to Equation (6)) and it doesn't have to be determined in this case either. The calibration setup of the studied A2WHP simulation model is presented in Figure 4.

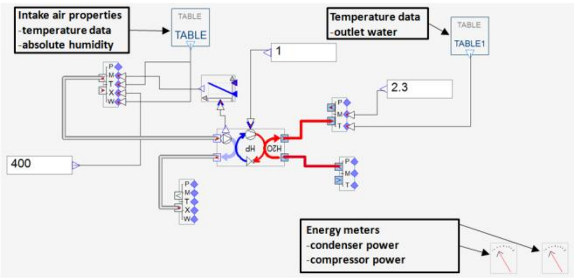

Figure 4 The calibration setup of the A2WHP system model

Figure 5 presents the performance of the default A2WHP model and the performance of the calibrated heat pump model, where the calibration parameters are determined from the SBO analysis.

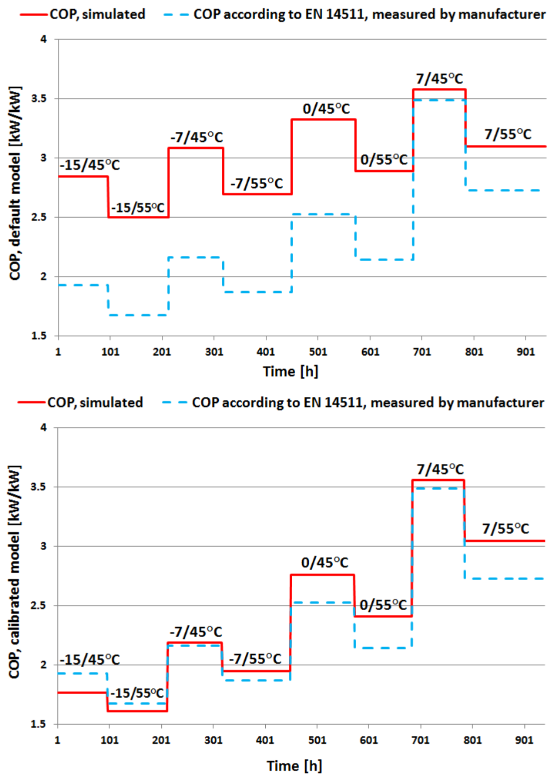

Figure 5 The performance of the simulated A2WHP system model, the default heat pump model (second from the bottom) and the calibrated heat pump model (bottom) according to 7/45 °C operating point presented

For the default heat pump model, all of the heat pump calibration parameters (*A-F*) are again default values of the ESBO Plant that are meant to be used with a typical heat pump system. The performance of the heat pump system is defined according to the 7/45 °C operating point in IDA ICE in both cases (default and calibrated).

As with the GSHP system model, the performance of the default heat pump model corresponds accurately to the performance of the real heat pump system only at the operating point, which was used to feed the input data of the simulated system (7/45 °C). As a conclusion, the default heat pump model gives too good energy performance, especially in cold climate weather conditions. However, the calibrated heat pump model corresponds to the measured data fairly accurately in all selected operating conditions. The calibration parameters *A-F* and the SPF-values for the default and calibrated heat pump models (7/45°C), determined from the SBO analysis, are for the Mitsubishi Electric CAHV-P500YA-HPB heat pump unit running in efficiency priority mode:

- ***A***: default 0.034, calibrated (7/45°C) -4.150;
- ***B***: default 3.4, calibrated (7/45°C) 4.7165;
- ***C***: default -1.0, calibrated (7/45°C) -0.1320;
- ***D***: default -0.01, calibrated (7/45°C) 3.8954;
- ***E***: default 1.5, calibrated (7/45°C) -2.1065;
- ***F***: default 1.2, calibrated (7/45°C) 2.1065;
- ***SPF***: default 3.016, calibrated (7/45°C) 2.315.

Detailed calibration of heat pump models according to specific case conditions

The calibration can also be weighted more to specific and more detailed case conditions, where the operation of the heat pump system is not equally balanced between different operating conditions (e.g. 12.5 % at each operating point, when manufacturer product data selected according to EN14511 is used). In this approach, the calibration is carried out by weighting the operation times of different operating conditions according to the specifications of the case building. This way the calibration parameters determined from the analysis will be weighted more accurately to the real climate conditions and temperature levels of the heating systems in a specific case study building.

This is demonstrated in Figure 6, where the outlet water temperatures and ambient air temperatures are logged from a specific case study simulation. The operation times in each operating condition are used to determine the weighted calibration parameters for the case study. The red lines in Figure 6 present the average operating condition during the specific operating time periods. The weighted operating conditions used in the calibration of the heat pump model in the case study are for the A2WHP system (Figure 6):

- **1*:** heat pump system is operating at -15/60°C operating point for 210 h → *3 % of* the total time during the heating season;
- **2*:** heat pump system is operating at -7/55°C operating point for 820 h → *12 % of* the total time during the heating season;
- **3*:** heat pump system is operating at 0/45°C operating point for 2800 h → *41 % of* the total time during the heating season;
- **4*:** heat pump system is operating at 7/35°C operating point for 2620 h → *38 % of* the total time during the heating season;
- **5*:** heat pump system is operating at 15/30°C operating point for 820 h → *6 % of* the total time during the heating season.

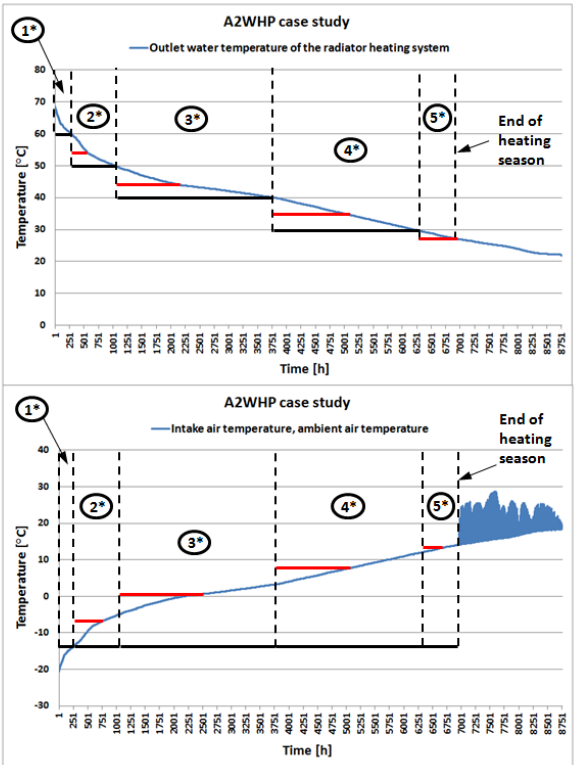

Figure 6 The temperature data of the heating system in the studied building and the ambient conditions for the detailed calibration of an A2WHP system

The ambient air temperature data presented in Figure 6 is the annual ambient temperature data of the 2012 Helsinki-Vantaa test reference year (TRY2012) defined by Kalamees et al. 2012. The studied heat pump model can now be calibrated by weighting the operating times of the calibration model at each operating point according to the simulated case conditions presented in Figure 6. The weighted average SPF-value for the Mitsubishi Electric CAHV-P500YA would be in this specific case approximately 3.21 for the radiator heating system. The DHW heating system must also be taken into account in addition to the radiator heating system. The weighted calibration of a GSHP system can be carried out in the same way. Typically, the annual average inlet brine temperature is more constant than the ambient air temperature, so the more detailed calibration of a GSHP is faster and easier to perform.

Results and conclusions from case studies

According to simulations in multiple case study buildings, where the performance of real dynamic heat pump systems is measured, the calibrated heat pump models perform very accurately in dynamic simulations of buildings. The performance of the simulated heat pump models (SPF, energy produced and electricity used) was within a 3-5 % error margin in real case studies, where the detailed calibration of the heat pump models was used. The case studies were conducted for the A2WHP, GSHP and EAHP systems, where the performance of the real installed heat pump systems was compared to the performance of the calibrated heat pump models. Case studies also indicated that the installation method of the real heat pump system has a relatively significant impact on the energy performance of the system. The operation of the ESBO Plant model corresponds to the variable condensing system, which makes it an effective method to simulate the modern energy efficient heat pump systems.

CONCLUSIONS

The study focused on testing and validation of simulation models of different modern heat pump systems as a part of dynamic energy simulation of buildings using IDA ICE simulation software. The studied heat pump systems were an A2WHP system, a GSHP system and an EAHP system. The results of the simulations indicate that IDA ICE can accurately simulate different heat pump systems by calibrating the default heat pump models. The energy performance of the calibrated models corresponded accurately to the measured performance of real systems in simulations, where multiple operating conditions were used. According to the results, the default simulation model of the A2WHP delivered too good performance in cold climate conditions during heating season. The same conclusion can be made with the GSHP system as well, but the difference between the studied real system and the simulated system is not as significant. However, it is important to notice that no default parameters should be accepted in a simulation of a real heat pump system without a critical review.

The performance of the calibrated models were tested in several case studies and compared to the performance of real installed systems. The results of the case studies showed good accurary between the simulated and real installed heat pump systems.

However, the calibration method using solely manufacturer product data is limited to some extent, when fully dynamic operation of heat pump systems is discussed. According to the case studies, the detailed calibration method of the simulated heat pump models to correspond to specific case conditions provides relatively accurate and reliable results, but further research is still required to validate the simulation of fully dynamic heat pump models operating in varying conditions.

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NOMENCLATURE

$A-F$,	constants calculated from rating conditions and from the type of heat pump/chiller, -;
COP,	coefficient of performance, -;
COP_{Rat},	COP at rated conditions, EER_{Rat}+1;
c_c,	degradation factor, -;
c_{ctr},	input control signal [0:1], -;
DHW,	domestic hot water, -;
EER,	energy efficiency ratio (= COP-1), -;
EER_F,	energy efficiency ratio at full load, -;
EER_{Rat},	COP_{Rat} -1 = given EER at rated conditions;
G,	part load exponent, -;
P_{comp},	compressor power, W;
P_{compF},	full load compressor power, W;
$P_{compRat}$,	compressor power at rated conditions, W;
P_{min},	minimum COP value in partial load operation, -;
PLF,	part load fraction, -;
PLR,	part load ratio-;
$PPar$,	parasitic sleep power, W;
Q_{cond},	condenser power, W;
Q_{condF},	full load condenser power, W;
$Q_{condRat}$,	heating capacity at rated conditions, W;
Q_{evap},	evaporator power, W;
Q_{evapF},	full load evaporator power, W;
$Q_{evapRat}$,	cooling capacity at rated conditions, W;
SPF,	seasonal performance factor, -;
T_{cIn},	condenser side inlet temperature, °C;
T_{cond},	condensation temperature, °C;
$T_{condRat}$,	calculated condensation temperature at rated conditions, °C;
T_{cOut},	condenser side outlet temperature, °C;
T_{eIn},	evaporator side inlet temperature, °C;
T_{eOut},	evaporator side outlet temperature, °C;
T_{evap},	evaporation temperature, °C;
$T_{evapRat}$,	calculated evaporation temperature at rated conditions, °C;
ΔT_c,	condenser side logarithmic (mean) temperature difference, °C;
ΔT_e,	evaporator side logarithmic (mean) temperature difference, °C;

REFERENCES

Achermann, M. and Zweifel, G. 2003. RADTEST – Radiant heating and cooling test cases, Subtask C, A Report of IEA Task 22, Building Energy Analysis Tools, 2003.

Björsell, N., Bring, A., Eriksson, L., Grozman, P., Lindgren, M., Sahlin, P., Shapovalov, A., Vuolle, M. 1999. IDA indoor climate and energy, Proceedings of the IBPSA Building Simulation '99 conference, Kyoto, Japan.

EPBD Recast: Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the Energy Performance of Buildings (recast).

Häkämies, S., Hirvonen, J., Jokisalo, J., Knuuti, A., Kosonen, R., Niemelä, T., Paiho, S., Pulakka, S. 2015. Heat pumps in energy and cost efficient nearly zero energy buildings in Finland, project HP4NZEB, VTT Technical Research Centre of Finland Ltd, Espoo, Finland, 2015.

Kalamees, T., Jylhä, K., Tietäväinen, H., Jokisalo, J., Ilomets, S., Hyvönen, R., Saku, S. 2012. Development of weighting factors for climate variables for selecting the energy reference year according to the EN ISO 15927-4 standard, Energy and Buildings 47 (2012), pp. 53–60.

Loutzenhiser, P., Manz, H., Maxwell, G. 2007. Empirical Validations of Shading / Daylighting / load Interactions in building energy simulation tools, A Report for the International energy Agency SHC Task 34, ECBCS Annex 43 Project C, 2007.

Niemelä, T. 2015. Cost Optimal Renovation Solutions in the 1960s Apartment Buildings, Master's Thesis, Aalto University, School of Engineering, Department of Energy Technology, HVAC 2015, Espoo, Finland.

Palonen, M., Hamdy, M., Hasan, A. 2013. MOBO a new software for multi-objective building performance optimization, Proceedings of BS2013: 13th Conference of International Building Performance Simulation Association, Chambéry, France, August 26–28, 2013.

Sahlin, P. 1996. Modelling and Simulation Methods for Modular Continuous Systems in Buildings. Doctoral Dissertation, Department of Building Sciences, Division of Building Services Engineering, Royal Institute of Technology 1996, Stockholm, Sweden. ISSN 0284-141X.

Salvalai, G. 2012. Implementation and validation of simplified heat pump model in IDA-ICE energy simulation environment, Energy and Buildings 49 (2012), pp. 132–141.

Travesi, J., Maxwell, G., Klaassen, C., Holtz, M. 2001. Empirical validation of Iowa energy resource station building energy analysis simulation models, IEA Task 22, Subtask A, 2001.