FIELD TEST BASED VALIDATION OF A HEAT PUMP MODEL FOR THE DETERMINATION OF SEASONAL PERFORMANCE FACTORS

Jörn Ruschenburg, Fraunhofer Institute for Solar Energy Systems ISE, Heidenhofstr. 2, 79100 Freiburg, Germany

Tomislav Ćutić, Fraunhofer ISE

Jeannette Wapler, Fraunhofer ISE

Marek Miara, Fraunhofer ISE

Abstract: In the residential sector, heat pumps are applied for domestic hot water and space heating. Simulations are widely used to plan such installations or for more general research in the field of heat pumps. Their advantages are low expenditure of time and costs. In contrast to simulations, field tests are conducted to evaluate the operation of heat pumps under real-life conditions.

Validation of simulation models is mandatory to guarantee a sufficient quality, i.e. the confirmation that a computerised model represents reality within specified limits of accuracy. In the presented paper, the field monitoring results of three ground-source installations are utilised for the validation of a black-box heat pump model. The model is similar to TRNSYS Type 201, but implemented in IDA ICE and then modified to allow variable mass flows and to handle the difficulties caused by rampant polynomials. As overall result, deviations between 1 % and 27 % are seen on monthly basis, with simulations usually resulting in higher values when compared to field monitoring results. The overall result appears as convincing, taking into account typical inaccuracies of laboratory and field tests as well as tolerances during heat pump production.

Key Words: heat pumps, model, black box, simulation, verification, validation, field test, monitoring, seasonal performance factor

1 INTRODUCTION

Heat pump models can be used during planning processes to predict the performance of heat pumps with electrically driven compressors[1], or in research and development to analyse the component behaviour, for example a heat pump integrated in different system layouts. For such purposes, usually black-box models are applied. They provide a sufficient level of detail – i.e. the dependency of the heat pumps' performance on source and sink temperature, probably also on mass flow – and their parameterisation does not require more than a limited set of parameters, i.e. no information about refrigerants, heat transfer, thermodynamic cycles, etc.

Afjei and Dott (2011) stated that such models are most wide-spread in dynamic simulation programs like TRNSYS, ESP-r, EnergyPlus, IDA ICE and MATLAB/Simulink. The heat pump

[1] henceforth referred to just as heat pumps

performance, in particular its coefficient of performance (COP) and heating power, is usually implemented by means of polynomials. The operational range is then covered by these polynomials, which comprises interpolation and extrapolation. Afjei and Dott categorise such models “performance map models”. The polynomials are derived from parameters, mostly provided by standardised laboratory test results, i.e. according to EN 14511 or EN 255 prior to this. They are usually found in laboratories’ publications or in the heat pumps’ data sheets published by manufacturers.

Thus, it is possible to parameterise heat pump models exclusively with laboratory test results. However, the conditions in the laboratory are not mirrored in the field – the fact that heat pumps are tested under steady-state conditions is the most obvious proof. The start-up phase, for example, is characterised by full electricity consumption and a thermal power that requires many minutes until reaching nominal level. The method of choice to analyse technical systems under real-life conditions is field monitoring. One example for heat pumps is the “WP Monitor” project (Miara et al. 2014).

Comparing simulation results to field monitoring results is a good possibility to assess the quality of modelling and simulation. It is a very important detail that boundary conditions in the field, e.g. source temperatures, show a wider variance compared to test rigs. In the following section, the whole process will be further described and labelled as validation.

2 METHODS

Before the methods are discussed, some definitions regarding modelling and simulation as well as performance assessment are to be presented for clarification.

2.1 Nomenclature

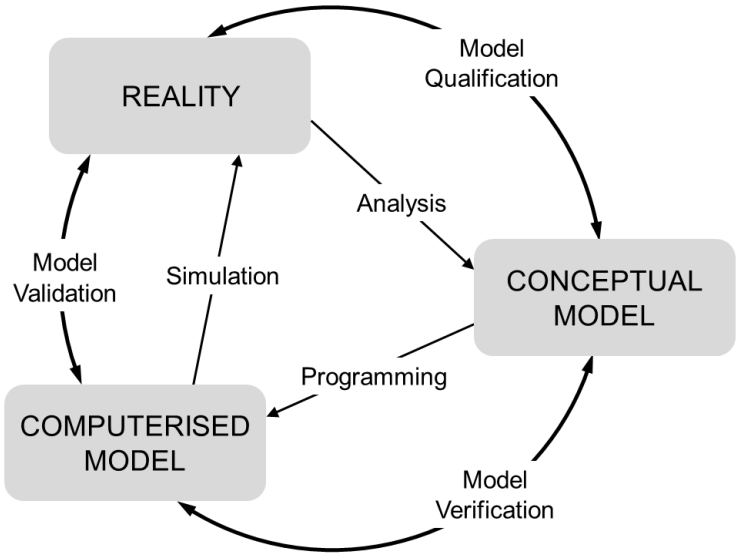


Figure 1: Phases of modelling and simulation and the role of verification and validation (created according to Schlesinger 1979 as cited in Oberkampf and Trucano 2002)

Both laboratory testing procedures and performance map models analysed by Afjei and Dott (2011) treat heat pumps as black boxes characterised by efficiency and thermal power depending on source and sink temperatures. By doing so, no information about refrigerants, heat transfer, thermodynamic cycles and other technical aspects is required. According to Figure 1, this abstraction is called model qualification, and the black box is one example of a conceptual model. This applies both to laboratory tests and calculative methods. If standardised tests are used to analyse a specific heat pump, the obtained results can be used to parameterise the performance maps of computer models, thus called parameters.

After programming and implementing a computerised version of this model, verification confirms that the computerised model represents the conceptual model. Validation, in turn, shows that a simulation carried out with the computerised model matches reality. The process of verification and validation may result in modifications and iterations before the actual model is finalised (Oberkampf and Trucano 2002).

Regarding the assessment of heat pumps installed in heating systems, boundaries must be defined. As for the efficiency, the coefficient of performance (COP) definition of EN 14511 is applied. To evaluate on daily, monthly or annual basis, the seasonal performance factor of the heat pump according to Malenković et al. (2012) is chosen, called SPF_{HP}, which has an identical system boundary but considers influences like dynamics and standby.

2.2 Procedure

The used simulation software is described in the following subsection. Then, the applied model is presented focusing on similarities and differences when compared to other models.

In a first step, verification is achieved by simulating one specific heat pump with the same operating points and mass flows that were arranged during laboratory testing. This heat pump was mainly selected because the number of tested operating points exceeds those presented in the standard. Based on the test results, standard operating points are selected to parameterise the model, whereas surplus operating points – expected to be accessible for only very few of the market-available heat pumps – allow the verification of extrapolating behaviour in a second step.

Field measurements results are used as reference for validation. It is known that this reference comes along with inaccuracies and therefore is not representing reality as shown in Figure 1. Taking into account that laboratory measurements, simulation and field measurements all having inaccuracies (to be quantified in Section 3.2), it is obvious that validation is different from perfect agreement in the mathematical sense. For comparing the simulation results with field measurement results, the model is parameterised with the corresponding standardised test data and then fed with the measured time series of boundary conditions, i.e. source temperature, sink temperature, mass flows and control signal.

Three independent installations, anonymised as A, B and C and presented in Section 2.5, are used in order to place the validation on a broader base. Each is evaluated for the whole year of 2012. They are selected from a large field test project with approximately 100 installations. This field test was conducted for the determination of seasonal performance factors, identification of optimisation possibilities, preparatory analyses regarding smart metering and enhancing dissemination and acceptance. Against this background, the sensor systems were selected and adjusted to show a precision and sample rate that is low when compared to laboratory equipment. The time resolution of measurement and thus simulation is one minute, though the priority during measurement was integral accuracy, not time-resolved accuracy. The analysis is therefore restricted to annual, monthly and daily basis. To avoid an analysis of the most dynamic effects, which are possibly not registered suitably by the applied sensors, the evaluation is further limited to ground-source heat pumps, i.e. brine/water heat pumps, which are not subject to defrosting and similar effects.

2.3 Simulation Environment

The software used for all simulations throughout this paper is IDA ICE, version 4.6. Generally, it can be used for modelling and simulating buildings and HVAC systems including hydraulics and control. IDA ICE is validated against other simulation software as well as real test buildings (Crawley et al. 2005, Sahlin et al. 2004, Achermann and Zweifel 2003).

The source code of all models can be viewed by the user. Editing is possible as well as implementing completely new models. This way was chosen for the heat pump model, which

was written in NMF (neutral model format), the equation-based and most commonly used modelling language for IDA ICE models (Sahlin 1996).

2.4 Presentation of the Applied Model

Following a black-box approach, COP and heating power are calculated as functions of temperatures at source and sink side. These functions are known from Afjei and Wetter (1997) as TRNSYS Type 201 or Type 204, or earlier, from Fischer and Rice (1981). They can be described as second-degree, mixed polynomials with 6 unknowns each. Laboratory test results for 6 operational points are required for parameterisation. In mathematical terms, an equation system with 6 unknowns has to be solved during pre-processing. The data can be acquired easily if the testing institution follows the complete procedure described by EN 14511 and if the manufacturer is willing to make the results available to the public.

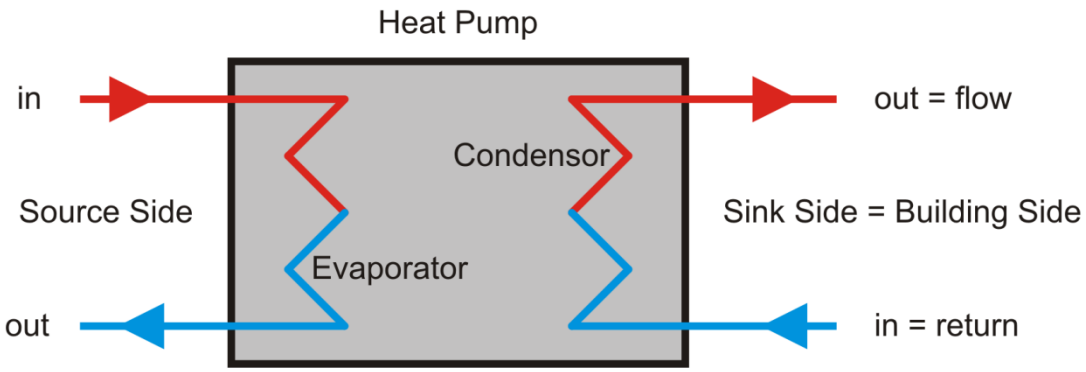


Figure 2: Heat pump seen as black box

The heat pump as black box is shown in Figure 2, with the outline shown in black to be literally understood as black box. Its inner elements – condenser, evaporator, compressor, refrigerant, etc. – and their physical behaviour are of no relevance for the conceptual model. Figure 2 also defines inlet and outlet nomenclature.

In the following step, the electric power consumption is calculated by reversing the COP definition, i.e. as ratio between heating power and COP. Similarly, the outlet temperature on source side and the return temperature on sink side can be calculated because mass flow and power on both sides are known, and the complementary temperature is already given as input variable. These equations, as well as the modelling of the dynamic start-up phase by a first-order equation, here with a time constant of 120 s, are identical to Afjei and Wetter (1997).

It is interesting to know that the IDA ICE software and its equation-based programming language NMF allow defining either of the complementary temperatures as input and output variables. Instead of providing the flow temperature to calculate the return temperature, it is just as well possible to calculate the flow temperature out of the given return temperature.

Though the TRNSYS Type 201 model considers mass flows to be equal to the value measured during standardised testing and thus constant, in today's heating systems with speed-controlled circulation pumps mass flows different from the standard values may occur. A modification as presented by Pahud and Lachal (2004) and validated by Pärisch et al. (2014) is therefore introduced to the model. According to this method, the black-box heat pump characteristics (COP and thermal power $\dot{Q}$) do not depend on source inlet and sink outlet temperature, as indicated by EN 14511 and the TRNSYS Type 201 model, but on the mean temperatures of source and sink side, shown by the following equations. Changes in

temperatures and heating power affect each other, and thus, the solver is required to recalculate each time step until convergence is met.

$$\mathrm{COP} = \overline{\square_{\mathrm{source}}} \cdot k_1 + \overline{\square_{\mathrm{sink}}} \cdot k_2 + \overline{\square_{\mathrm{source}}}^2 \cdot k_3 + \overline{\square_{\mathrm{sink}}}^2 \cdot k_4 + \overline{\square_{\mathrm{source}}} \cdot \overline{\square_{\mathrm{sink}}} \cdot k_5 + k_6 \quad (1)$$

$$\dot{Q}_{\mathrm{sink}} = \overline{\square_{\mathrm{source}}} \cdot l_1 + \overline{\square_{\mathrm{sink}}} \cdot l_2 + \overline{\square_{\mathrm{source}}}^2 \cdot l_3 + \overline{\square_{\mathrm{sink}}}^2 \cdot l_4 + \overline{\square_{\mathrm{source}}} \cdot \overline{\square_{\mathrm{sink}}} \cdot l_5 + l_6 \quad (2)$$

Because polynomials tend to skitter away when extrapolated (cf. Afjei and Dott 2011), the model is further modified by piecewise definitions. Polynomial interpolation and extrapolation is used for an operational range limited by the tested source temperatures extended by 5 K. If, for example, the maximum source temperature among the parameters is given by B5/W35[2] or B5/W45, both COP and heating power will be simulated according to their polynomials for source temperatures up to 10 °C. Outside this range, extrapolation for COP and heating power is calculated in a linear way, taking the mean slope between specific centre parameters, namely B–5/W45 and B5/W45. This approach is certainly not physically correct but applied to avoid the volatility of polynomials. Additional equations are needed for this modification, though no empirical or other additional parameters.

2.5 Presentation of the Evaluated Heat Pump Installations

All heat pump installations analysed in this paper were monitored within the project "WP Monitor" (Miara et al. 2014), organised by the Fraunhofer Institute for Solar Energy Systems. The installations will be presented in an anonymous way using project-internal numbering. Out of the 83 installations within the project, 3 are examined within this paper. The selection process that led to this small number is based on several, independent effects and described in the following:

The first criterion is the availability of laboratory test data. Caused by the modelling approach explained above, comprehensive test data is needed for the parameterisation of the heat pump model. While all this data corresponds to standard EN 14511, laboratory testing is not required for all desired operational points, for example when complete type series are tested. Furthermore, it is not mandatory for manufacturer to make all the test results public.

The second criterion is the availability of specific field monitoring data. As the project providing the data base aims at different purposes (determination of seasonal performance factors, identification of optimisation possibilities, preparatory analyses regarding smart metering, enhancing dissemination and acceptance), the installed measurement and data logging equipment are not always sufficient (regarding logging frequency, positioning, etc.) for the validation procedure. One example is the ambient air temperature, which is essential for the validation of air-source heat pump models but not for SPF determination in the field.

All of the analysed heat pumps are installed in recently built (semi-)detached houses, covering space heating via floor-heating systems as well as the heating of domestic hot water (DHW). Table 1 further characterises the installations. It is a coincidence that solar thermal collectors are featured twice, as solar combinations was no focus of the project. None of them does include modulating technology. The area-specific heat consumption of these buildings is presented in Table 3 within the results section further below, each of which close to 45 kWh/m²/a and thus being characteristic for an energy-efficient building standard.

[2] compare EN 14511 for naming convention

Table 1: Characterisation of the evaluated heat pump installations

no.	COP	heating power in kW	nominal conditions and standard	source	solar thermal collectors	thermal storages	year of installation	geographical region
A	4.6	10.2	B0/W35 EN 14511	borehole heat exchanger	no	DHW buffer, 400 L	2007	North German Plain
B	4.7	10.6	B0/W35 EN 14511	near-surface collector	yes, for heating and DHW	combined, 750 L	2009	German Central Upland
C	4.4	10.1	B0/W35 EN 14511	borehole heat exchanger	yes, for DHW	heating buffer, 600 L, DHW buffer, unknown size	2008	German Central Upland

3 RESULTS

3.1 Verification

According to the chosen nomenclature, verification is the confirmation that a computerised model represents a conceptual model. Both simulation model and laboratory testing procedure treat heat pumps as black boxes to be characterised by COP and thermal power depending on source and sink temperatures. This black box is in fact the conceptual model. Testing results are used as input parameters for the computerised model, and thus the reference for the verification.

The steady-state COP is used to visualise the verification results in Figure 3. Heating power and electricity consumption, which are computed likewise, behave accordingly and are thus not shown. The laboratory test results are connected with solid lines. Here, the subset used to parameterise the model is highlighted with oversized symbols. The simulation results from the polynomial model – for the presented verification, the TRNSYS Type 201 model would behave identically – appear as dashed lines.

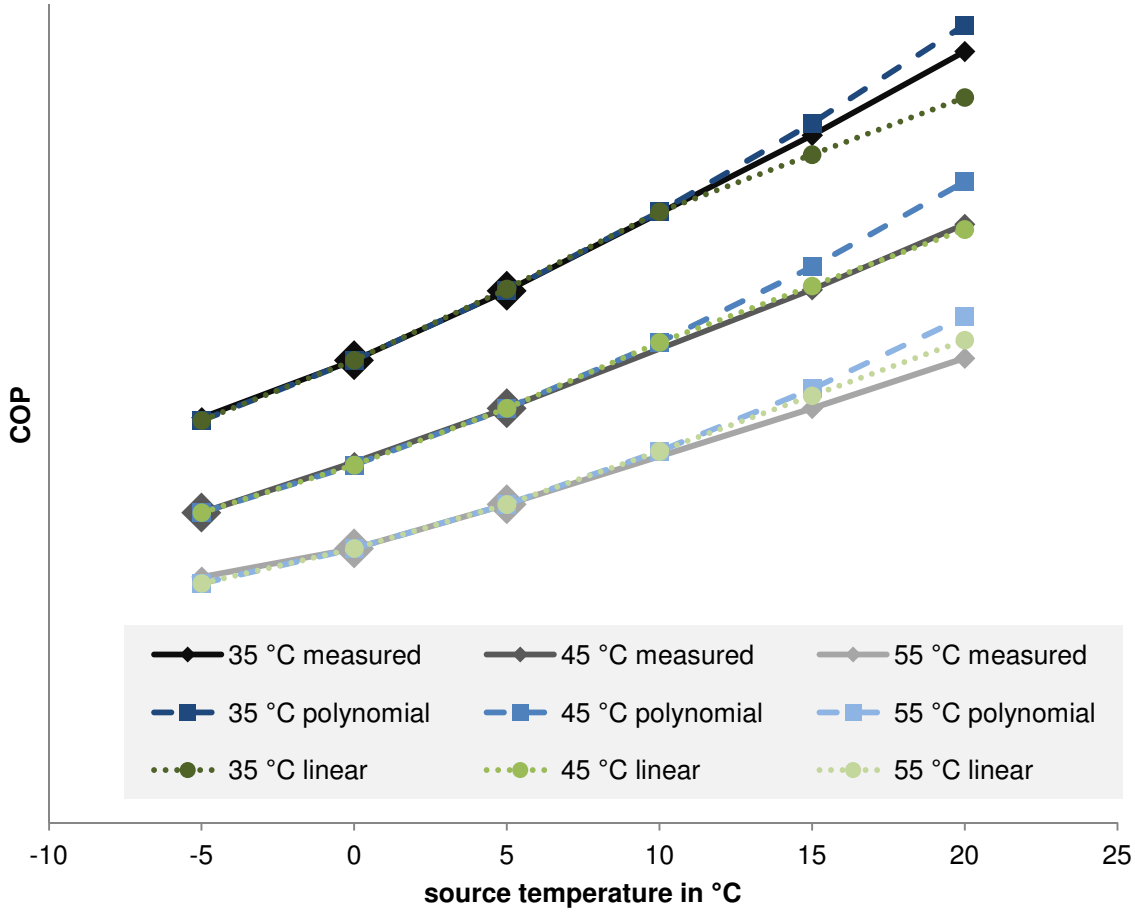


Figure 3: Steady-state COP as function of source and flow temperatures. Standardised measurement results, polynomial model (Afjei and Wetter 1997) simulation results, piecewise defined model (linear extrapolation approach) simulation results. Axis of ordinate is depicted qualitatively to anonymise test data.

It can be observed that the polynomial reproduces the highlighted parameterisation points with highest precision. In fact, deviations are caused by numerical reasons only and thus amount to less than 0.01 percent. When interpolating, the polynomials provide accurate results, with deviations remaining below one percent throughout. When extrapolating, in contrast, the polynomials systematically overrate the performance by up to 8 percent (for B20/W55). These findings confirm the observations of Afjei and Dott (2011) in a mild form.

Results for the piecewise (polynomial-linear) model are shown as dotted lines in Figure 3. Like intended, the results are identical to those of the purely polynomial model for source temperatures up to 10 °C (5 °C as maximum source temperature tested plus 5 K polynomial extrapolation).

For flow temperatures of 45 and 55 °C, the modified model achieve higher accordance between measured and simulated data than the original model. For a flow temperature of 35 °C, the COP is underrated by the modified model, whereas it was overrated before. This outcome may not appear as quantifiable improvement, but there is one overall benefit to be kept in mind: The model restrains even on more rampant polynomials. The verification is considered successful against this background. A more elaborated approach would be a flow-temperature dependent extrapolation, still linear but based on two variables, or other more sophisticated approaches demanding additional parameters.

3.2 Validation

Table 2 shows the comparison between SPF_{HP} obtained from field measurement and simulation. In fact, results for three independent installations are shown on monthly and annual basis. The relative deviation D of the SPF_{HP} is defined by Equation 3, using indices "m" for measurement and "s" for simulation. The question whether the tabled values are acceptable is answered in the following by comparing them with uncertainties met in measurement.

$$D_{\mathrm{SPF_{HP}}} = \frac{SPF_{\mathrm{HP,s}} - SPF_{\mathrm{HP,m}}}{SPF_{\mathrm{HP,m}}} \tag{3}$$

Table 2: Field measurement (index m) and simulation results (index s) for SPF_{HP} and relative deviation, for all three installations on monthly and annual basis

no.	A			B			C		
	$SPF_{HP,m}$	$SPF_{HP,s}$	$D_{SPF,HP}$	$SPF_{HP,m}$	$SPF_{HP,s}$	$D_{SPF,HP}$	$SPF_{HP,m}$	$SPF_{HP,s}$	$D_{SPF,HP}$
Jan	3.73	3.71	–0.5 %	4.10	4.67	13.9 %	4.46	4.40	–1.3 %
Feb	3.76	3.72	–1.1 %	3.93	4.50	14.5 %	4.19	4.13	–1.4 %
Mar	3.34	3.55	6.3 %	3.98	4.55	14.3 %	4.38	4.32	–1.4 %
Apr	3.11	3.33	7.1 %	4.00	4.51	12.8 %	4.39	4.33	–1.4 %
May	2.62	3.04	16.0 %	3.87	4.40	13.7 %	4.02	4.00	–0.5 %
Jun	2.20	2.72	23.6 %	3.07	3.67	19.5 %	3.66	3.67	0.3 %
Jul	2.17	2.75	26.7 %	2.88	3.28	13.9 %	2.99	2.98	–0.3 %
Aug	2.23	2.75	23.3 %	1.59	1.80	13.2 %	2.30	2.32	0.9 %
Sep	2.73	3.17	16.1 %	4.26	4.79	12.4 %	4.12	4.14	0.5 %
Oct	3.13	3.45	10.2 %	4.44	5.08	14.4 %	4.59	4.55	–0.9 %
Nov	3.42	3.62	5.8 %	4.10	4.70	14.6 %	4.58	4.51	–1.5 %
Dec	3.55	3.63	2.3 %	4.07	4.59	12.8 %	4.44	4.37	–1.6 %
Year	3.32	3.49	5.1 %	4.05	4.61	13.9 %	4.33	4.27	–1.2 %

The results appear inhomogeneous since deviations vary between installation and season and in sign. Installation A shows the most explicit seasonal variance, installation B is characterised by a constant bias with additional rather small seasonal variance and System C appears to be the most balanced one though still with some seasonal influence regarding the sign. Interpretation is possible when looking at all the inaccuracies of the validation process. Though no complete propagation of uncertainty is presented, the main influences can be identified and quantified to some extent:

- For standardised laboratory testing, there are a number of uncertainties to be accepted. The influence of measurement quality on the inaccuracy of COP values has been analysed by Ertesvåg (2011) for air/water heat pumps. The results show relative uncertainties for COPs in the range of ±7…9 % when tested according to

EN 14511. Though brine/water heat pumps were not scope of that publication, similar deviations are anticipated because of similar uncertainties in measurements.

- The heat pump in the field is actually not identical to the heat pump tested in the laboratory and its data sheet. Tolerances during manufacturing of compressors, for example, can cause deviations of up to ±5 % in efficiency.
- During simulation, the tolerance for convergence is set to ±2 %. Further numerical discrepancies are considered insignificant.
- For field measurement, the uncertainty is difficult to quantify. First, because of dynamic influences like the reaction time of sensors. Second, because mass flow and temperature differences vary from installation to installation. Even with identical measurement equipment, inaccuracies may thus vary as well. While this topic is still subject to further investigation, systematic errors for exemplary cases are given by values up to ±10 % (Miara et al. 2014). These inaccuracies are mainly caused by thermal measurement rather than electric measurement.

Still, the results may appear inhomogeneous when comparing installations. The seasonal effects require a more detailed analysis. As shown in Figure 4, the effect is caused by differences in space heating and DHW measurement. In this example, DHW measurement is less congruent. The seasonal variance is explained by the fact that two different heat meters with potentially different systematic deviations are used. The following paragraph explains why deviations can generally be expected to be higher in the DHW mode.

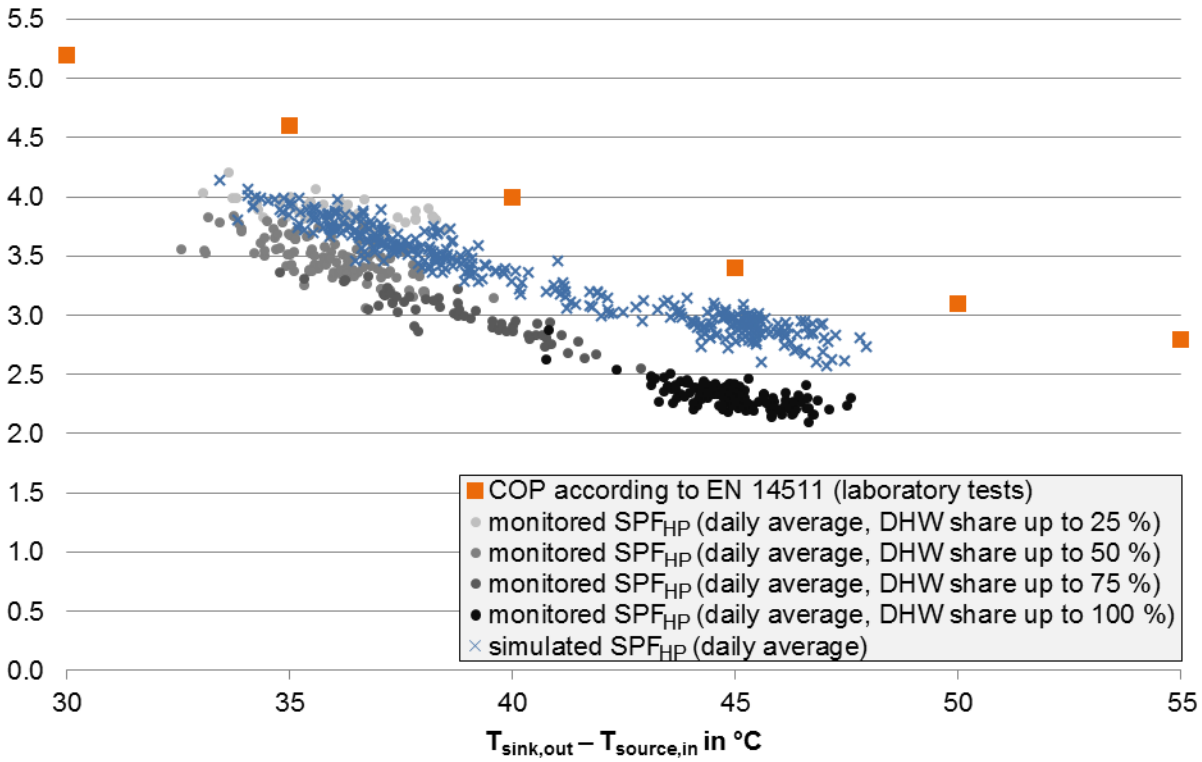


Figure 4: Laboratory test, field monitoring and simulation results for installation number A

Heat meters actually measure volume flow, for instance of water through the condenser, and temperature difference, in this case between inlet and outlet of the condenser. For any heat pump in heating mode, thermal power for low flow temperatures is higher than thermal power for high temperatures. This results – if the volume flow is considered constant – in a smaller temperature difference for high-temperature DHW heating. Consequently, a given constant deviation in temperature difference measurement has a more significant impact on the DHW thermal power measurement when compared with (relatively) low-temperature space

heating. The overall deviation on monthly base is thus a question of weighting the consumption with the systematic deviation of the respective heat meter. This can be achieved with the help of Table 3.

Table 3: Heat consumption (Q, indices SH for space heating, and DHW for domestic hot water) (measured at the building side of the heat pump), area-specific consumption (q), DHW share

no.	A	B	C
Q_{SH} in kWh/a	9069	8935	12720
Q_{DHW} in kWh/a	5713	5856	2645
q_{SH} in kWh/m²/a	48	45	45
————	39 %	40 %	17 %

The counterexample less variant deviations is given by installation number C presented in Figure 5. Here, there are separate heat meters for space heating and DHW as well, but their systematic deviation is of more uniform range.

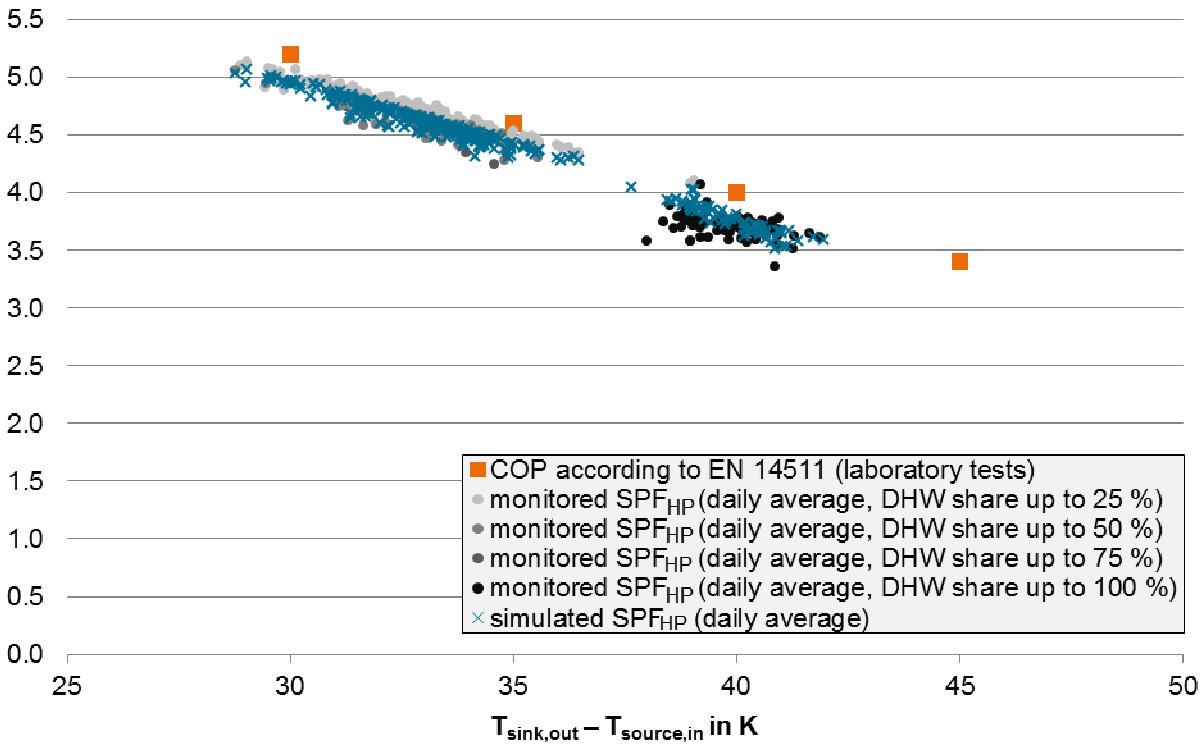


Figure 5: Laboratory test, field monitoring and simulation results for installation number C

The measurement equipment of installation C is not necessarily of higher quality, i.e. higher *guaranteed* accuracy. It has to be pointed out that systematic deviation is described by ranges. With the inaccuracies justified and seasonal effects explained, the validation – as approached in this paper – is considered successful.

4 DISCUSSION, CONCLUSION AND OUTLOOK

The heat pump model presented and validated in this paper is an IDA ICE adaption of TRNSYS Type 201 with two significant modifications. First, the mass flow may now be

different from standard level. Second, polynomial extrapolation was mostly changed to linear extrapolation with the aim to gain stability and accuracy.

Regarding verification, it was shown that the linear extrapolation approach does not necessarily result in more accurate results. Still, the approach is a reliable opportunity when dealing with less homogeneous polynomials. More precise and yet stable approaches are certainly possible, though at the same time likely to demand more product-specific input parameters.

The standard EN 14511 was frequently updated during the last years. One suggestion for the next update could be the introduction of higher source temperatures to be tested for brine heat pumps, e.g. B15 or even B20. On the one hand, extended testing causes extended costs. On the other hand, the heat pumps' behaviour at higher source temperatures might be of interest when solar thermal or other (in relative terms) high-temperature energy is used as source (cf. Ruschenburg et al. 2013). Similarly, dynamic test sequences would help the modelling and parameterisation, though again at the price of higher costs.

During validation, it is important to remember the inaccuracy of the "reality" provided by field test and laboratory test results. Even without complete propagation of uncertainty, it becomes clear that systematic inaccuracies of 30 % and more are possible. It is one possible solution to put the evaluation on a broader basis, i.e. a higher number of installations. This would enhance statistical significance. Another approach could be high-precision field tests, consequently with higher efforts.

ACKNOWLEDGMENTS

The Federal Ministry of Economics and Technology (BMWi) of Germany granted funding for this research under the administrative numbers 03ET1028A and 0327401A. This support is gratefully acknowledged and enables Fraunhofer ISE to participate in Annex 38 of the IEA Heat Pump Programme (HPP), also organised as Task 44 of the Solar Heating and Cooling Programme[3].

The authors would like to thank the participants and funders of the "WP Monitor" project, especially for kindly supporting this particular work with the required laboratory test data.

We wish to express our thanks to the members of the heat pump research group at Fraunhofer ISE, notably Danny Günther, Robert Langner and Sebastian Helmling, who have acquired, processed and kindly provided the most essential monitoring data. Last but not least, many thanks to Peter Engelmann for proofreading.

REFERENCES

Achermann M. and G. Zweifel 2003. „Praxisnahe Validierung von Simulationsprogrammen, Beitrag zu IEA Solar Task 22 – Schlussbericht", Bundesamt für Energie, Bern, Switzerland.

Afjei T. and R. Dott 2011. "Heat pump modelling for annual performance, design and new technologies", *Proceedings of Building Simulation 2011 – 12th Conference of International Building Performance Simulation Association*, Sydney, Australia.

Afjei T. and M. Wetter 1997. "Compressor heat pump including frost and cycle losses – Model description and implementing into TRNSYS", Zentralschweizerisches Technikum Luzern, Horw, Switzerland.

Crawley D.B., J.W. Hand, M. Kummert and B.T. Griffith 2005. "Contrasting the capabilities of building energy performance simulation programs", US Department of Energy, Washington (DC), USA.

[3] http://task44.iea-shc.org/

EN 14511:2013. “Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling”, European Committee for Standardization (CEN), Brussels, Belgium.

Ertesvåg I.S. 2011. “Uncertainties in heat-pump coefficient of performance (COP) and exergy efficiency based on standardized testing”, *Energy and Buildings*, Volume 43(8), pp. 1937–1946.

Fischer S.K. and C.K. Rice 1981. “A steady-state computer design model for air-to-air heat pumps – report”, Oak Ridge National Laboratory, USA.

Malenković I., S. Eicher and J. Bony 2012. “Definition of main system boundaries and performance figures for reporting on solar heat pump systems – technical report”, Vienna, Austria.

Miara M., D. Günther, R. Langner and S. Helmling 2014. “The outcomes and lessons learned from the biggest in Europe monitoring campaign of heat pumps in family dwellings in Germany”, *Proceedings of the 11th IEA Heat Pump Conference.* Montréal (Québec), Canada.

Oberkampf W.L. and T.G. Trucano 2002. “Verification and validation in computational fluid dynamics”, *Progress in Aerospace Sciences*, Volume 38, pp. 209–272.

Pahud D. and B. Lachal 2004. »Mesure des performances thermiques d’une pompe à chaleur couplée sur des sondes géothermique à Lugano – rapport final«, Office fédéral de l’énergie, Bern, Switzerland.

Pärisch P., O. Mercker, J. Warmuth, R. Tepe, E. Bertram and G. Rockendorf 2014. “Investigations and model validation of a ground-coupled heat pump for the combination with solar collectors“, *Applied Thermal Engineering*, Volume 62, pp. 375–381.

Ruschenburg J., S. Herkel and H.-M. Henning 2013. “A statistical analysis on market-available solar thermal heat pump systems“, *Solar Energy*, Volume 95, pp. 79–89.

Sahlin P. 1996. “NMF handbook. An introduction to the neutral model format”, Stockholm, Sweden.

Sahlin P., L. Eriksson, P. Grozman, H. Johnson, A. Shapovalov and M. Vuolle 2004. “Whole-building simulation with symbolic DAE equations and general purpose solvers”, *Building and Environment*, Volume 39(8), pp. 949–958.

Schlesinger S. 1979. “Terminology for model credibility”, *Simulation*, Volume 32(3), pp. 103–104.