

HEAT PUMP MODELLING FOR ANNUAL PERFORMANCE, DESIGN AND NEW TECHNOLOGIES

Thomas Afjei, Ralf Dott
Institute of Energy in Building, University of Applied Sciences Northwestern Switzerland
4132 Muttenz, Switzerland
thomas.afjei@fhnw.ch
http://www.fhnw.ch/iebau

ABSTRACT

In future highly energy-efficient buildings, heat pumps will play a key role. Hence, annual efficiency calculation and optimization by means of simulating heat pump heating and cooling systems are very valuable, especially if building and building technology are coupled.

This paper gives an overview on existing models and a categorization with pros and cons in terms of generic approach, validation and quality of documentation. Most common are simple performance map based models for seasonal performance factor calculations, sometimes improved by adding PT1 inertia for heating up and cooling down. Especially for heat pump design, refrigerant cycle based physical gray-box models can encourage new developments.

INTRODUCTION

Europe launched an ambitious program: 20-20-20 up to 2020, i.e. 20% less greenhouse gases, 20% higher energy efficiency and 20% more renewable energy until the year 2020. Ultra-low energy, near or net zero energy buildings are part of the solution. Many of them are equipped with heat pumps. Hence, heat pumps play an important role in many simulations, where building and building technology are coupled.

Purpose of heat pump modelling

Depending on the use of the heat pump model, there are mainly three different classes of models corresponding to the required level of detail and the amount of work accepted for their application. Table 1 shows a qualification of the three model classes according to their application, which are described in the following.

1. Calculation methods

The aim of calculation methods is to provide a fast but sufficiently precise calculation of heat pump system performance, in order to compare different heat pump products using a seasonal coefficient of performance (SCOP) or to calculate a building specific seasonal performance factor (SPF). Both, the best choice and most commonly used, are calculation methods using simplified performance maps and an energy or time related weighting of representative operating conditions.

2. Dynamic system behaviour

Going into more detail, the next step is dynamic analyses of whole heat supply systems. Therein, mostly performance maps for the heat pump behaviour are both in use and in most cases appropriate, but now with more detailed performance data. However, the time-dynamic influence of the boundary conditions, such as climate or user behaviour, is now considered as a temporal series of boundary conditions with fixed or dynamic time steps.

3. Heat pump design

The need in heat pump design processes is to optimize the heat pump unit on the level of the refrigerant cycle. Hence, the models need to calculate the refrigerant flows and states as well as to represent the heat pump components individually (evaporator, compressor, condenser and expansion valve) to be able to replace components and optimize the interaction of the heat pump components. Therein mostly component performance map models are in use. For further optimization of the individual heat pump components, e.g. the evaporator of an air-to-water heat pump, specialized physical models are necessary.

Table 1 General qualification of models

	CALCULATION METHODS		DYNAMIC SIMULATION		HEAT PUMP DESIGN MODELS	
	SCOP	SPF	quasi steady state model	dynamic effect model	refrigerant cycle model	heat pump component
Flexibility of use	--	-	o	+	++	o
Level of detail	--	-	o	o	+	++
Amount of work for application	--	-	o	o	++	++
Computation time	--	--	o	+	++	++
Required knowledge	--	-	o	+	++	x

-- = very low - = low o = medium
++ = very high + = high x = specialized

HEAT PUMP MODEL OVERVIEW

Calculation methods

In standards, mostly easy to use calculation methods are required for the seasonal performance factor of commonly used heat pumps. They are in use for the purpose of comparison between different heat pumps or with other heat generating technologies. Therein two different applications of calculation methods can be distinguished, product comparison with standard ratings and system evaluation applied to individual buildings. Both types of calculation methods usually use simplified performance maps of heat pumps while time and energy are taken into account by weighting factors for representative operating conditions.

Product comparison with standard ratings (e.g. SCOP) uses simplified calculation methods under uniform standard conditions and boundaries not related to a real but only to an artificial reference building and furthermore indicated only for a single or a restricted number of representative climatic conditions. For Northern America, the standard ANSI/AHRI 210/240-2008 defines the measurement conditions and calculation procedure of the seasonal energy rating for the heating period as heating seasonal performance factor (HSPF) and for the cooling period as the seasonal energy efficiency ratio SEER. In the European standardization there is up to now no finalized standard for a seasonal energy rating on product comparison level. Until now, product labeling and hence comparison is referred to single operating conditions by the coefficient of performance (COP) based on measurements according to EN 14511:2007.

The draft standard prEN 14825:2010, which is actually in development, will provide a standard rating by a SCOP for the heating season and by a SEER for the cooling season. The standard defines one reference climate for cooling and three reference climatic conditions for heating, i.e. average, warmer and colder climate. It defines furthermore one procedure for heat emission systems in cooling mode and three types of heat emission systems for heating, i.e. low, medium and high temperature application.

The comparison of an individual building technology solution with required minimum performance (e.g. SPF) or maximum primary energy consumption requires calculation methods that are more detailed and have as far as possible realistic boundary conditions, but also take into account some standard assumptions e.g. for the user behaviour. On the European level, this provides the standard EN 15316-4-2:2008 named "Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4-2: Space heating generation systems, heat pump systems". Therein, the calculation method bases on a temperature class approach (bin-method shown in Figure 1), where representative operating conditions are weighted with individually de-

derived factors based on time and energy. This could be extended by a ventilation system and eventually installed solar components that are considered by subtracting the fraction of the heat recovery / solar input of the fraction of space heating (SH) or domestic hot water (DHW, in figure 1 also W is used) energy, respectively, to be covered by the heat pump. The fraction of back-up energy is calculated by an energy balance, which is evaluated by the running time of the heat pump. The heat pump fraction is subsequently weighted with the respective COP of the bin derived by the testing, and added-up. An energy weighting of the heat pump and back up delivers the SPF for the SH mode. In a similar way, the SPF of the DHW-mode is calculated and energy weighting of both SPF numbers delivers the overall SPF of the unit.

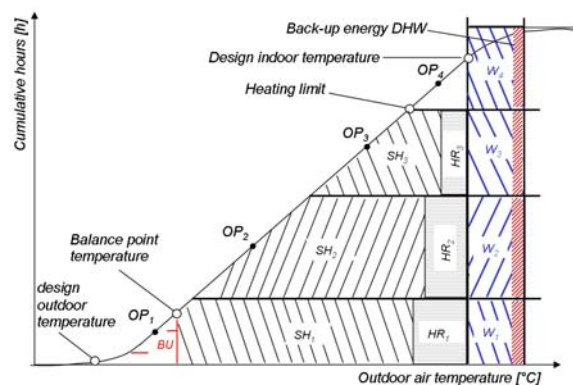


Figure 1 Principle of the bin method for space heating, domestic hot water and ventilation with heat recovery systems (OP - operating point, BU – back-up, HR – heat recovery unit) (from Affei et al. 2007)

The described static calculation methods are well suited for known components in known system configurations that are covered by the respective calculation procedure and allow for a very fast and sufficiently precise result for a broad group of users. However, they are also restricted to the above-mentioned application and are not suitable for new system configurations, applications or an extrapolation of the application range.

Dynamic system simulation

For the evaluation of new more sophisticated system concepts, a more detailed modelling is required to be able to consider system dynamics or to evaluate the systems under varying boundary conditions. Therein the interaction of heat loads like building or domestic hot water demand with heat storages and heat sources, e.g. borehole heat exchangers or solar heat, play a key role for the evaluation of the system behaviour over long-term periods like full years or short-term periods to evaluate for example the control behaviour. Empirical models are quite widespread, because the representation of the component behaviour in the system is sufficiently precise and furthermore the required data of individual products are mostly available. Physical models, or better mod-

els based on physical effects, are rather available for less complex components like solar collectors or borehole heat exchangers, but not for such complex units as heat pumps since the required computation time rises significantly if solving the states and flows of the refrigerant cycle for each simulation time step.

Quasi steady state performance map models as for example described in Afjei 1989 are the most widespread heat pump models in dynamic simulation programs like e.g. TRNSYS, ESP-r, Insel, EnergyPlus, IDA-ICE or Matlab/Simulink Blocksets. Therein, a restricted number of sampling points from performance map measurements are used either to interpolate in-between those points or to fit a two-dimensional polynomial plane. These models use the inlet-temperature of the heat source to the heat pump and the desired outlet-temperature on the heat sink side of the heat pump to calculate the thermal output of the heat pump and its electricity demand. Figure 2 shows an exemplary COP performance map of an air-to-water heat pump.

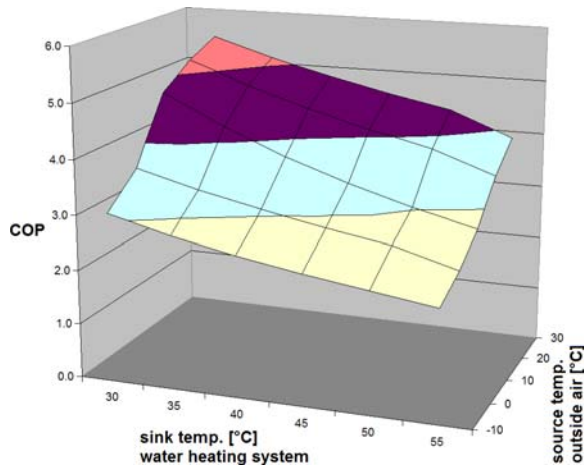


Figure 2 Exemplary COP performance map of an air-to-water heat pump from Dott et al. 2011

Typical implementations of quasi steady state performance map models for heat pumps in simulation software packages are for example the TRNSYS Types 504, 505, 665 and 668 from the TESS library 2011. Furthermore, these kinds of models can also represent a more complex heat pump, if they still have defined characteristics like a two stage heat pump that has been described by Afjei et al. 1997.

Usually only the standard measurements according to EN 14511:2007 are available as input for this kind of simulation models. For water source heat pumps in minimum two testing points, representing typical heat source conditions, i.e. 10 °C / 15 °C, and for brine source heat pumps in minimum three representative testing points, i.e. -5 °C / 0 °C / 5 °C, are available. For air source heat pumps also in minimum three testing points are available, i.e. -7 °C / 2 °C / 7 °C. The corresponding heat sink temperatures represent the desired outlet temperature, i.e. the supply temperature to the heating system. EN 14511:2007 defines standard rating points at 35 °C e.g. for floor

heating, 45 °C e.g. for radiator heating in low energy buildings and an application rating point at 55 °C, e.g. for other radiator heating systems. Both, look-up table and polynomial fit models, represent the behaviour quite well in the range of the given sampling points. However, if extrapolating the range, the user has to check the results carefully, since the gradients at the boundary of the modelled performance map do not necessarily correspond to the gradient of the real heat pump. This effect has especially shown relevance on the one hand for very low sink temperatures, e.g. in ultra-low energy houses with thermally active building elements using flow temperatures in the range of 25 °C to 30 °C, and on the other hand for air source heat pumps. Air source heat pumps show a significant drop in performance at source temperatures below 5-7 °C. There, the moisture in the heat source air can start forming ice on the evaporator, which needs to be defrosted, if too much ice has formed. This effect decreases the COP of the heat pump by a few decimal points below the mentioned source temperatures. Therefore, especially for air source heat pumps, the gradients at the boundary of the measured performance maps are very important for extrapolating the performance map in simulation. The measurements according to EN 14511:2007 include the icing / defrosting effect for air source heat pumps by averaging the performance measurements over operating periods that include heat pump operation with and without icing. Hence, it will be considered correctly for annual performance simulations, but the short time dynamic effect influencing the control or flow temperature fluctuations will not be represented.

Dynamic effects like described above for icing / defrosting can be an extension to quasi steady state models. The effect of icing / defrosting has been described by Afjei 1989 based on a semi empirical model approach. Therein, the COP reduction due to the icing effect could be considered separately as addition to performance map based on compressor data, where icing / defrosting is not taken into account. Furthermore, the model described in Afjei 1989 includes extensions for thermal inertia in condenser or evaporator. A PT1 element with defined time constant represents the dynamic effect during the start-up or shutdown of the heat pump. The first order differential equation models the heating of the heat pump components and the pressurising of the heat pump cycle as well as releasing the stored heat after shutdown.

In conclusion, the dynamic system simulation models rely also on reference measurements according to EN 14511:2007, like the calculation methods, which are very well suited for most applications that operate in the range of the underlying measurement data. Important is a good fitting of the performance map to the real characteristic that could be achieved by look-up tables as well as with polynomial fits. Compared to static calculation methods, dynamic system simu-

lations consider time dynamically and hence are free to combine a wider range of surrounding system components. Furthermore, they are able to implement additional short time dynamic effects.

Heat Pump Design models

The most sophisticated class are models for the design of heat pumps. Their aim is to be able to build up a theoretical heat pump out of known component characteristics, i.e. for condenser, expansion valve, evaporator and compressor. Therefore, they need to represent the interaction of the internal heat pump components on the refrigerant cycle level and calculate the refrigerant states and flows. On the heat pump assembly design level, the interaction of the refrigerant cycle components, their sizing and the refrigerant types are of main interest. Again, performance map models are mostly used, but now performance maps for the refrigerant cycle components. For the optimization of single components, some models go even to a more detailed level to optimize the component design and behaviour like e.g. icing of an air-to-water heat pump evaporator or inverter driven compressors. On this level of detail, physical models are sometimes in use or, albeit rarely, 3D-CFD models.

Refrigerant cycle models aim to optimize the heat pump by choosing the right components for evaporator, compressor, condenser and expansion valve or by integrating additional components into the refrigerant cycle like subcooler, desuperheater or internal heat exchanger. Therein, the aim is mainly not to optimize components in detail but to exchange components and find the right components for an optimized interaction. Hence, the refrigerant cycle models require performance characteristics of the heat pump components from measurements, where a good representation of their behaviour in the heat pump cycle is important. These models are again mainly quasi steady state models, but now on component level.

The development of refrigerant cycle models starts from rather simple representations of the refrigerant cycle. Therein, pressure drop is neglected and hence constant refrigerant temperatures are assumed over the evaporator and condenser and they are calculated based on the average logarithmic temperature difference in the heat exchangers. A heat transfer characteristic of the heat exchangers (NTU-model) allows for the adaption to varying mass flow rates, flow temperatures and compressor capacity. Hence, the evaporator and the condenser are calculated with one averaged heat transfer efficiency, summarizing the parts of the heat exchanger, where desuperheating, subcooling or superheating of the refrigerant takes place or only separating the desuperheating part in the condenser. Fourth degree polynomials can be found for the evaporator and condenser NTU-models. The compressor model bases on a characteristic of its isentropic efficiency.

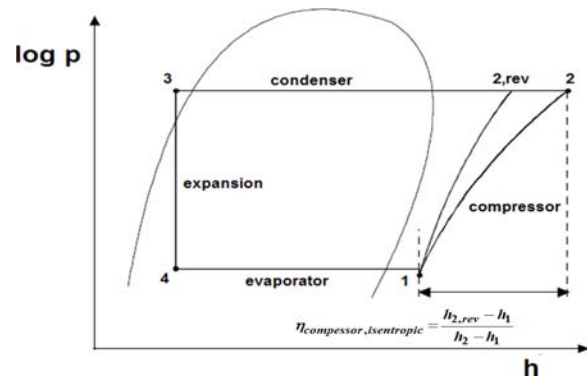


Figure 3 Heat pump process in log-p-h-diagram with isentropic compressor efficiency

The isentropic efficiency is calculated as quotient of the isentropic enthalpy difference for an ideal and hence reversible compression divided by the real, polytropic enthalpy difference, c.f. Figure 3. The models used for the isentropic efficiency range from an assumed constant isentropic efficiency to 3rd degree polynomials of the compression ratio over the compressor. For the volumetric efficiency of the compressor, 2nd degree polynomials or exponential functions of the compression ratio can be found. The process in the expansion valve is commonly assumed adiabatic, since the heat transferred across the small surface is negligible. All parameters of the above-mentioned curve fit models have to be derived from measurements or catalogue data. The above-described models can be found for example in detail in Hornberger 1994 or Schrapf 2001.

The development of refrigerant cycle models in detail goes into more detailed representations of heat pump components, the integration of more physical effects and the consideration of time dynamic effects. For example, Bühring 2001 extended in his doctoral thesis the commonly used quasi-static heat pump design models with the integration of two condensers in series, an air source evaporator considering icing on the evaporator with a simplified model and with an internal heat exchanger. With using two condensers in series it is necessary to differentiate the parts of the condenser where the refrigerant desuperheats, condenses or subcools, to be able to calculate the heat flows in the two heat exchangers that work with only one expansion valve behind both. Thus every condenser, and accordingly every evaporator, is calculated with a moving boundary, which defines the three named parts of the heat exchanger dynamically. The model for icing on the air source evaporator adds the heat gain caused by the phase change of the humidity in the air and enables the model to estimate the point in time when defrosting would be necessary. An internal heat exchanger in the refrigerant process can enhance the heat pump performance by transferring heat from the condensed refrigerant at high pressure before the expansion valve to the evaporated refrigerant at low pressure before the compression. Albert et al. 2008 furthermore examined the icing on air source evaporators in detail and devel-

oped a model for the formation of ice on the evaporator of an air source heat pump depending on its geometry, surface structure and the state of the entering moist air.

Alternatively to the above-mentioned isentropic efficiency model for the compressor, a polynomial equation fit for the compressor performance map using the same model as described for the heat pump performance in Afjei 1989 gives an equal good representation of the compressor characteristic. Ohyama et al. 2008 gives an overview on the last year's development in capacity controlled scroll compressor technology. Especially the capacity controlled scroll compressors improved their performance due to development of enhanced electronic controlled interior permanent magnet motor technology. Although most of the compressor models were developed for single speed compressors, some empirical curve fit models for part load operation have been described e.g. in Bühring 2001, Jin 2002 or Jin et al. 2003.

Afjei 1993 went a step further and examined the behaviour of inverter driven scroll compressors in detail. In his doctoral thesis, he describes a model for an inverter driven scroll compressor and the determining effects for its efficiency, i.e. the fixed volume ratio, leakages, friction, motor losses, inverter losses and shell losses. One essential characteristic is the fixed built-in compression ratio or volume ratio of the scroll compressor. Especially air source heat pumps work over a wide range of operating conditions with varying refrigerant pressure ratios between condenser and evaporator. Only in one operating point, this external pressure ratio of the heat pump cycle is equal to the internal pressure ratio of the scroll compressor. In all other operating points over- or under-compression lead to reduced efficiency. Madani et al. 2011 describe a similar approach for the compressor model in a capacity controlled ground source heat pump system. One other essential influence on the inverter compressor efficiency is the part load efficiency of the electric motor and the inverter. The development in the motor technology from inverter driven induction motors (IM) over surface permanent magnet synchronous motors (SPMSM) to interior permanent magnet synchronous motors (IPMSM) over the last 15 years combined with inverter improvements has resulted in a significant enhancement of the part load efficiency (c.f. Figure 4). In today's high efficiency capacity controlled compressor motors, the motor efficiency stays above 90% over the part-load operating range.

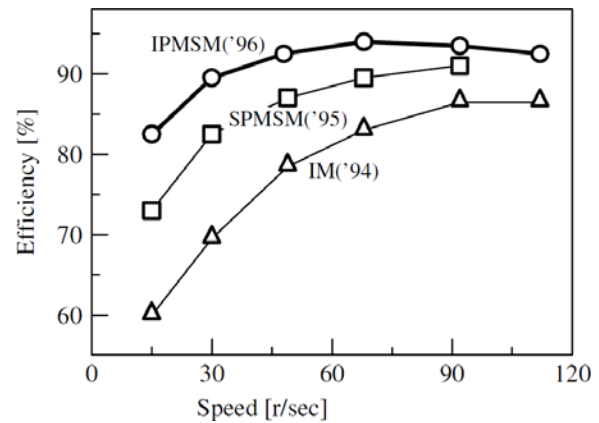


Figure 4 Comparison of compressor motor efficiency from Ohyama et al. 2008

Jin 2002 and Brandemuehl et al. 2009 conducted an extensive literature review on heat pump and chiller models and available manufacturer data. They further developed the found model descriptions and defined heat pump models based on a parameter identification methodology for the fit to catalogue data. The aim of this approach is to benefit from the detailed compressor model developments and derive a model that is still suitable for annual performance simulations. For this, the most important parameters of detailed heat pump design models are identified and the detailed calculation methods are simplified so far, that a parameter estimation procedure can fit these parameters from manufacturer catalogue data. Although, the derived approach requires only catalogue data from manufacturers, the model results are as precise as detailed design models and furthermore may be extended beyond the catalogue data with a more stable and precise extrapolated prediction.

Heat pump design models usually include models on the refrigerant cycle level. They are in use and suited mainly for the design of heat pumps, requiring a high level of knowledge, computation time and amount of work for application and on the other hand delivering very specialised or detailed results. In newer developments, the experiences with heat pump design model application lead to improved dynamic system simulation models in the form of complex models that are easier to access by parameter identification techniques or give advice for the integration of relevant effects into empirical models.

VERIFICATION

Calculation method against field measurements

The results of the calculation according to EN 15316-4-2:2008 have been compared to field measurements in Afjei et al. 2007 (also published in Wemhoener et al. 2008) for a ventilation compact unit air source heat pump (SH, DHW and ventilation mode). The measured performance has been compared to the calculated values for the different operation modes and different system boundaries. For the calculation the local outdoor temperature conditions monitored with the field monitoring equipment have been used as input data for the calculation in order to refer to the same boundary conditions as in field testing, which is necessary for validation purposes. Figure 5 gives an overview of the results.

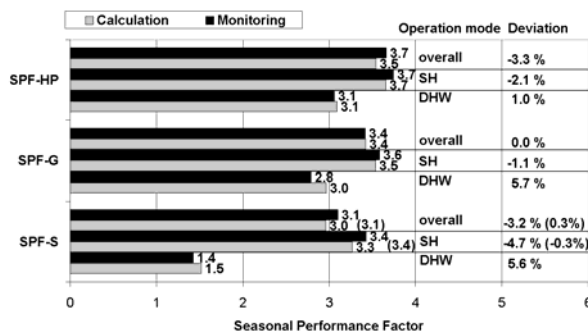


Figure 5 Seasonal performance factors of the field monitoring compared to the calculated values (from Afjei et al. 2007)

Since controller settings are usually not known in detail and therefore cannot be evaluated, two operation modes of the circulating pump on the sink side have been considered for the calculation of the electricity demand: pump is running when heat pump is on and pump is running throughout the whole heating period, which is given by the values in brackets. However, the impact on SPF values is marginal. Regarding the comparison of the field monitored performance to the calculation results, the space heating part is generally reproduced better than the DHW part. For the DHW calculation a constant daily consumption has been assumed, while in reality the tapping volumes are not so evenly spread over the year. Further differences occur due to control effects. For instance, the back-up heating supports the DHW heating by the heat pump after large draw-offs to accelerate the hot water availability. These controller settings are too case-specific to be reproduced by a hand calculation.

Calculated overall seasonal performance factor values deviate in the range of $\pm 4\%$ from measured values, which is a satisfactory result for a hand calculation, where certain simplifications are inevitable. The single operation modes show with $\pm 6\%$ a slightly higher deviation from the measured results for this ventilation heat pump compact unit.

Calculation method against dynamic simulations

The same calculation method according to EN 15316-4-2:2008 has been compared in Dott et al. 2011 to dynamic simulations for a brine-to-water heat pump coupled to a low energy single family house. Exemplary results of this study for the generator seasonal performance factor (SPF-G) are shown in Table 2. The SPF-G assesses the energetic quality of the heat generation and is calculated as sum of the generated heat divided by the required expenditure in the heat generator including the expenditure for the heat source. In Dott et al. 2011, on the one hand the functions space heating (SPF-G_H) and domestic hot water preparation (SPF-G_W), as described in the standard, and on the other hand an additional SPF-model for a passive cooling function (SPF-G_C) with borehole heat exchanger passively coupled to the low temperature floor heating system have been considered. The calculation model for the passive cooling function only considers a daily heat storage of the space cooling heat rejected into the ground and the increase in the heat pump source temperature in domestic hot water operation, whereas the dynamic simulation uses a detailed physical model of the borehole and the surrounding ground for all operation modes. The calculation method without passive cooling according to EN 15316-4-2:2008 leads to very good agreement of the SPF-G compared to detailed simulation results.

generator SPF <u>without</u> passive cooling			
	simulation	bin-method	
SPF-G _H	4.4	4.6	5%
SPF-G _W	3.3	3.2	-3%
SPF-G _{HW}	4.0	4.1	2%
generator SPF <u>with</u> passive cooling			
	simulation	bin-method	
SPF-G _H	4.4	4.6	5%
SPF-G _W	3.5	3.5	-1%
SPF-G _C	12.9	12.6	-2%
SPF-G _{HWC}	4.7	4.8	2%

Table 2 Comparison of the generator SPF from dynamic simulation and calculation method (from Dott et al. 2011)

The added passive cooling function, neglecting the heat storage effect on the space heating operation in the calculation method, leads to equal good agreement like without heat injection into the borehole and confirms the simplification to neglect the effect of heat injection into the borehole on the winter heat withdrawal. The increase of the domestic hot water seasonal performance factor by the heat injection from passive cooling could be reproduced by a very simple calculation model based on a short time adiabatic ground heat storage model. The performance factor of the passive cooling could be reproduced with good agreement also by a simplified calculation based on average electric power consumption as long as the assumption of full cooling need coverage is valid.

Validation of heat pump design models

Bühring 2001 conducted a validation of the derived detailed heat pump design model. Therein, the bi-quadratic polynomial curve fit for the compressor thermal capacity and electric power consumption according to Afjei 1989 reaches deviations smaller than 0.4% compared to manufacturer data. The electricity consumption and the heat capacity of the whole heat pump model achieve inaccuracies smaller 5%.

Jin 2002 compared the results of the derived parameter estimation model and of a quasi-static performance map model against catalogue data. Table 3 shows the results as relative error.

Catalog Data Used	RELATIVE ERROR					
	Maximum (abs. value)		average (abs. value)		RMS	
	Par. est.	EQ -fit	Par. est.	EQ -fit	Par. est.	EQ -fit
	electric power consumption					
234 points	16.1%	29.1%	4.2%	6.3%	5.8%	7.8%
16 points	22.0%	33.5%	4.2%	6.9%	5.8%	8.5%
	heating capacity					
234 points	8.2%	38.5%	2.5%	7.4%	3.1%	9.7%
16 points	8.8%	49.2%	2.8%	12.6%	3.4%	16.5%

Table 3 Comparison of the relative error by parameter estimation (Par.est.) and equation-fit (EQ-fit) models (from Jin 2002)

CONCLUSION

This paper gives an overview on heat pump models in literature that are well described to be implemented in a program. A categorization with pros and cons in terms of generic approach and validation has been carried out as well. Therein the whole spectrum of heat pump models is addressed, from simple calculation methods as used in standards over dynamic system simulation model up to heat pump design models.

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