Models of Sub-Components and Validation for the IEA SHC Task 44 / HPP Annex 38

Part A: Summary

A technical report of subtask C

Report C2 Part A – Final Draft

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IEA Solar Heating and Cooling Programme

The *International Energy Agency* (IEA) is an autonomous body within the framework of the Organization for Economic Co-operation and Development (OECD) based in Paris. Established in 1974 after the first "oil shock," the IEA is committed to carrying out a comprehensive program of energy cooperation among its members and the Commission of the European Communities.

The IEA provides a legal framework, through IEA Implementing Agreements such as the *Solar Heating and Cooling Agreement*, for international collaboration in energy technology research and development (R&D) and deployment. This IEA experience has proved that such collaboration contributes significantly to faster technological progress, while reducing costs; to eliminating technological risks and duplication of efforts; and to creating numerous other benefits, such as swifter expansion of the knowledge base and easier harmonization of standards.

The *Solar Heating and Cooling Programme* was one of the first IEA Implementing Agreements to be established. Since 1977, its members have been collaborating to advance active solar and passive solar and their application in buildings and other areas, such as agriculture and industry. Current members are:

Australia	Finland	Singapore
Austria	France	South Africa
Belgium	Italy	Spain
Canada	Mexico	Sweden
Denmark	Netherlands	Switzerland
European Commission	Norway	United States
Germany	Portugal	

A total of 49 Tasks have been initiated, 35 of which have been completed. Each Task is managed by an Operating Agent from one of the participating countries. Overall control of the program rests with an Executive Committee comprised of one representative from each contracting party to the Implementing Agreement. In addition to the Task work, a number of special activities—Memorandum of Understanding with solar thermal trade organizations, statistics collection and analysis, conferences and workshops—have been undertaken.

Visit the Solar Heating and Cooling Programme website - www.iea-shc.org - to find more publications and to learn about the SHC Programme.

Current Tasks & Working Group:

Task 36	*Solar Resource Knowledge Management*
Task 39	*Polymeric Materials for Solar Thermal Applications*
Task 40	*Towards Net Zero Energy Solar Buildings*
Task 41	*Solar Energy and Architecture*
Task 42	*Compact Thermal Energy Storage*
Task 43	*Solar Rating and Certification Procedures*
Task 44	*Solar and Heat Pump Systems*
Task 45	*Large Systems: Solar Heating/Cooling Systems, Seasonal Storages, Heat Pumps*
Task 46	*Solar Resource Assessment and Forecasting*
Task 47	*Renovation of Non-Residential Buildings Towards Sustainable Standards*
Task 48	*Quality Assurance and Support Measures for Solar Cooling*
Task 49	*Solar Process Heat for Production and Advanced Applications*

Completed Tasks:

Task 1	*Investigation of the Performance of Solar Heating and Cooling Systems*
Task 2	*Coordination of Solar Heating and Cooling R&D*
Task 3	*Performance Testing of Solar Collectors*
Task 4	*Development of an Insolation Handbook and Instrument Package*
Task 5	*Use of Existing Meteorological Information for Solar Energy Application*
Task 6	*Performance of Solar Systems Using Evacuated Collectors*
Task 7	*Central Solar Heating Plants with Seasonal Storage*
Task 8	*Passive and Hybrid Solar Low Energy Buildings*
Task 9	*Solar Radiation and Pyranometry Studies*
Task 10	*Solar Materials R&D*
Task 11	*Passive and Hybrid Solar Commercial Buildings*
Task 12	*Building Energy Analysis and Design Tools for Solar Applications*
Task 13	*Advanced Solar Low Energy Buildings*
Task 14	*Advanced Active Solar Energy Systems*
Task 16	*Photovoltaics in Buildings*
Task 17	*Measuring and Modeling Spectral Radiation*
Task 18	*Advanced Glazing and Associated Materials for Solar and Building Applications*
Task 19	*Solar Air Systems*
Task 20	*Solar Energy in Building Renovation*
Task 21	*Daylight in Buildings*
Task 22	*Building Energy Analysis Tools*
Task 23	*Optimization of Solar Energy Use in Large Buildings*
Task 24	*Solar Procurement*
Task 25	*Solar Assisted Air Conditioning of Buildings*
Task 26	*Solar Combisystems*
Task 27	*Performance of Solar Facade Components*
Task 28	*Solar Sustainable Housing*
Task 29	*Solar Crop Drying*
Task 31	*Daylighting Buildings in the 21st Century*
Task 32	*Advanced Storage Concepts for Solar and Low Energy Buildings*
Task 33	*Solar Heat for Industrial Processes*
Task 34	*Testing and Validation of Building Energy Simulation Tools*
Task 35	*PV/Thermal Solar Systems*
Task 37	*Advanced Housing Renovation with Solar & Conservation*
Task 38	*Solar Thermal Cooling and Air Conditioning*

Completed Working Groups:

CSHPSS; ISOLDE; Materials in Solar Thermal Collectors; Evaluation of Task 13 Houses; Daylight Research

IEA Heat Pump Programme

This project was carried out within the Solar Heating and Cooling Programme and also within the *Heat Pump Programme*, HPP which is an Implementing agreement within the International Energy Agency, IEA. This project is called Task 44 in the *Solar Heating and Cooling Programme* and Annex 38 in the *Heat pump Programme*.

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

Under the IA collaborative tasks or "Annexes" in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and work plans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex. The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

The IEA Heat Pump Centre

A central role within the IEA Heat Pump Programme is played by the IEA Heat Pump Centre (HPC). Consistent with the overall objective of the IA the HPC seeks to advance and disseminate knowledge about heat pumps, and promote their use wherever appropriate. Activities of the HPC include the production of a quarterly newsletter and the webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

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

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Visit the Heat Pump Programme website - http://www.heatpumpcentre.org/ - to find more publications and to learn about the HPP Programme.

Contents

1 About this report

This report has been established in an international cooperation in the framework of the IEA SHC Task 44 / HPP Annex 38 on solar and heat pump systems (T44A38). Subtask C of T44A38 deals with modeling and simulation of solar and heat pump systems and their components. Report C2 of subtask C gives an overview of recent developments and the state of the art in component modeling in this field. This report has been split into several parts:

Part A: Summary

Part B: Collectors

Part C: Heat Pumps

Part D: Storage

The summary presented in this Part A of Report C2 has been published in a paper at the Solar Heating and Cooling Conference SHC 2012 in San Francisco (Haller et al. 2012). The version presented in this report contains additional figures compared to the final printed version in the proceedings of the conference, as well as minor changes in the text, and an additional chapter on storage. It does not contain the editorial changes of the conference proceedings. The original paper reference is:

Haller, M.Y., Bertram, E., Dott, R., Afjei, T., Ochs, F. & Hadorn, J.-C., 2012. Review of Component Models for the Simulation of Combined Solar and Heat Pump Heating Systems. Energy Procedia, 30, p.611–622. DOI: 10.1016/j.egypro.2012.11.071.

http://www.sciencedirect.com/science/article/pii/S1876610212015871

2 Introduction

Combined solar and heat pump heating systems have been on the market for years and are the topic of a current joint task/annex of the International Energy Agency's Solar Heating and Cooling Programme (SHC) and its Heat Pump Programme (HPP), the IEA SHC Task 44 / HPP Annex 38 (T44A38) "Solar and Heat Pump Systems" (Hadorn 2011). The increasing use of these heating systems may substantially contribute to the reduction of CO_2 and other emissions from fossil fuels. This reduction is dependent on the electricity mix used to run these systems and on the seasonal performance factor of the system (SPF_{sys}), i.e. the ratio of heat output to electric energy use of the whole heating system. Heat pumps used for domestic hot water preparation (DHW) and space heating (SH) in central Europe use about 60-80% heat from the ambient and 20-40% electric energy (Miara et al. 2011). Solar thermal collectors can be used to further decrease the amount of electricity used since they can provide heat using only 1-2% electric energy input for a pumped system. Subtask C of T44A38 is dealing with the modeling and simulation of combined solar and heat pump heating systems. This paper gives an overview on the results of the survey on component models for these systems. Because of the large number of publications that can be encountered in this field starting from the 50ies of the last century (Jordan & Threlkeld 1954), a selection had to be made in order to cover some of the most important classical modeling approaches as well as important recent developments. A special focus has been laid on features that are new or of particular importance for the simulation of the annual performance of the combination of solar and heat pump heating systems such as:

- Solar collectors that are used as a heat source for heat pumps and thus may face operation below the temperature of the ambient air and especially also below the dew point of the ambient air, including also operation of uncovered collectors in the absence of solar irradiation (e.g. at night), with little heat gain per area and possibly also with lower mass flow rates than usual.
- Heat pumps that receive heat from solar thermal collectors and thus possibly face higher temperatures on the evaporator and also higher variability of the temperatures available for the evaporator.
- Ground heat exchangers / heat storage that are not only used for heat extraction but also for re-charging by solar thermal collectors.

3 Solar thermal collector models

Within T44A38, models for the simulation of solar thermal collectors have been reviewed by Bertram et al. (2013). Solar thermal collectors that are applied in heat pump systems are usually either covered or uncovered flat plate collectors or vacuum tube collectors that are operated at temperatures below 100 °C. One of the most well-known approaches for the simulation of the steady state or instantaneous area specific heat gain rate of such a solar thermal collector is given by eq. (1) (Duffie & Beckman, 1991, p. 278).

$$\dot{q}_{gain} = F_R \left[\dot{S} - u_L \left(\vartheta_{in} - \vartheta_{amb} \right) \right] \tag{1}$$

In this equation, F_R is the mass flow dependent heat removal factor, $\dot{S}$ is the absorbed solar radiation per unit area and time, and u_L is the overall heat loss coefficient of the collector. However, standard test procedures in use today (e.g. ISO 9806-1 1994, EN 12975-2 2006) use the even more simplified or approach that is based on empirical values for the efficiency at zero temperature difference (η_0) and the heat loss coefficients a_1 and a_2 as shown in eq. (2):

$$\dot{q}_{gain} = \eta_0 \cdot G - a_1 \cdot \left(\vartheta_{ref} - \vartheta_{amb} \right) - a_2 \cdot \left(\vartheta_{ref} - \vartheta_{amb} \right)^2 \tag{2}$$

Where G is the area specific irradiance on the collector plane, and ϑ_{ref} is either defined as the inlet temperature of the collector or as the average of the inlet and outlet temperatures. Additionally, the quasi-dynamic collector efficiency equation in EN12975 includes a number of additional effects such as different incident angle modifiers for direct and diffuse radiation, the influence of wind speed, the influence of the thermal capacitance, and the influence of long wave radiation exchange. From the point of view of the operation as a heat source for heat pumps, several aspects are put on debate concerning the approach presented in eq. (2) or in the quasi-dynamic collector efficiency equations in EN12975:

- Latent heat gains of condensation or sublimation of water vapor on the absorber surface when the absorber is operated below the dew point are not included.
- The term $a_2 \cdot \left(\vartheta_{ref} - \vartheta_{amb} \right)^2$ suggests increasing heat losses even at collector temperatures below the ambient air. It should be noted that it is generally recommended to set $a_2 = 0$ for uncovered collectors, although this recommendation may not always be appropriate.
- For uncovered solar collectors with selective coatings that are available on the market (Thissen 2011) dew on the absorber surface changes the emissive properties of the surface and thus the parameters determined for the collector model from measurements without dew on the surface.
- For the operation of uncovered collectors as ambient air heat exchangers without solar irradiation the definition of an efficiency based on the solar irradiation is not useful. Furthermore, for the resulting small heat gains from ambient air it is tempting to operate the solar collectors with mass flow rates that are much smaller than the ones used for standard testing. For these small mass flow rates the simplifying assumption of a linear increase of the temperature of the fluid between the inlet and the outlet that is often assumed when applying eq. (2) may not be justified.
- For the simulation of photovoltaic-thermal absorbers, so called PV/T collectors, a subtraction of the photovoltaic yield from the available solar radiation that can be converted into heat has to be added.

Condensation heat gains have been included in several collector models reported in the literature (Massmeyer & Posorski 1982, p.10; Keller 1985, p.11; Pitz-Paal 1988; Soltau 1992; Morrison 1994; Eisenmann et al. 2006; Frank 2007; Bertram et al. 2010; Perers 2010). A common feature of these models is that the condensation heat gain is based on the theory of heat and mass transfer as presented in standard textbooks. Usually the model equations include a convective heat transfer coefficient h_{conv}, the relative humidity of the ambient air Φ_{amb}, the phase change enthalpy of water $\Delta \mathrm{h}_{lat}$, and the difference between the water vapor load of the ambient air and the water vapor load at the surface of the absorber. However, the models differ in the assumption of the temperature at which the maximum water vapor load at the absorber surface is evaluated. Eq. (3) shows an example from Bertram et al. (Bertram et al. 2010) where the saturated water vapor pressure p_{sat} is evaluated at the – physically correct – surface temperature ϑ_{surf}. Because the surface temperature is usually not available from standard tests on solar thermal collectors, it is estimated with eq. (4). In this equation, Θ is used for the conversion of the heat transfer coefficient to the (partial pressure difference based) mass transfer coefficient with the help of the Lewis number for air.

$$\dot{q}_{lat} = h_{conv} \cdot \Delta \mathrm{h}_{lat} \cdot \Theta \cdot \left[\Phi_{amb} \cdot p_{sat}\left(\vartheta_{amb}\right) - p_{sat}\left(\vartheta_{surf}\right) \right] \tag{3}$$

$$\vartheta_{surf} = \vartheta_m + \dot{q}_{gain+} / u_{int} \tag{4}$$

In eq. (4), the surface temperature is calculated using a collector parameter for the overall heat transfer coefficient between the fluid and the absorber surface u_{int}. In order to avoid the necessity to estimate the surface temperature, Perers has presented a model where the saturated water vapor load of the air ν_{sat} is evaluated at the mean fluid temperature ϑ_m, and possible overestimation of condensation gains by this assumption are corrected with the empirical factor c_{lat} (Perers 2010). This factor also includes the conversion from heat transfer coefficient to (water vapor load difference based) mass transfer coefficient and is assumed to be constant.

$$\dot{q}_{lat} = \mathrm{c}_{lat} \cdot h_{conv} \cdot \Delta \mathrm{h}_{cond} \cdot \left[\Phi_{amb} \cdot \nu_{sat}\left(\vartheta_{amb}\right) - \nu_{sat}\left(\vartheta_{m}\right) \right] \tag{5}$$

It can be shown that for identical values of h_{conv} and rather large values for u_{int}, a value for c_{lat} can be found such that both approaches deliver the same result (Fig. 1a). For smaller values of u_{int}, the onset of condensation is shifted to lower fluid temperatures for the model based on eq. (4), and no value can be found for c_{lat} that would result in equal results of the two models (see Fig. 1b).

The implementation of these condensation models in TRNSYS Type 136 (Perers 2010) and Type 202 (Bertram et al. 2010) were compared with results from field measurements of fully irrigated unglazed metal cushion collectors for different real weather conditions in Yverdon-les-Bains (Switzerland) by Bunea et al. (2012). The results were then compared to the field measurements of unglazed collectors. The amount of condensation was measured by collecting the condensate underneath each unglazed collector. The incertitude of the condensation heat gains measurements was estimated to 8%. For both models the condensation heat gains agreed well with the measurements.

The influence of wind on the convective heat transfer coefficient of the absorber is an often discussed topic with large uncertainties and a wide range of different models both for the estimation of local wind speed based on meteorological wind speed and for the estimation of the effect of local wind speed on the convective heat transfer. A review of wind convection coefficient correlations has been presented (Palyvos 2008). Theoretically, in the absence of wind, the natural convection heat transfer coefficient of a cooled plate facing upwards is dependent on the inclination of the plate. Philippen et al. (2011) have performed parallel measurements of the heat gain on a fully irrigated metal cushion absorber with selective coating and an identical absorber without selective coating inclined at different angles and operated below the temperature of the ambient air at night. Although at wind speeds < 1 m/s significantly higher heat gains were achieved with higher angles of inclination, the evaluation of the pyrgeometer measurements revealed that these higher heat gains must be attributed to higher long wave irradiance from the field of view of the absorber. After subtraction of this influence, no significant dependency of the convective heat transfer coefficient on the inclination of the absorbers was detected. Within the same study it was also shown that heat gains from the selective coated absorber were higher as long as there was no dew on the surface, and equal to the non-selective coated absorber when there was dew on the surface.

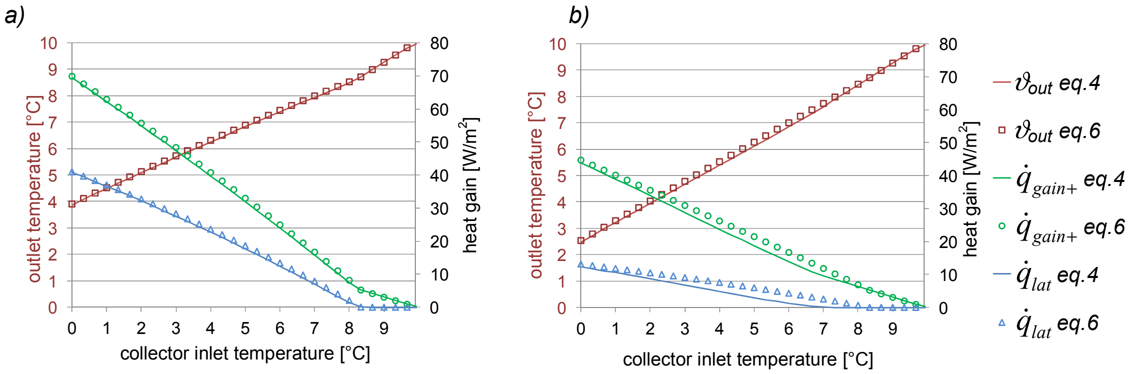


Fig. 1. comparison of total heat gain, condensation heat gain and outlet temperature simulated based on eq. (4) and based on eq. (6) with ambient temperature ϑamb = 10°C, and relative humidity Φamb = 0.9; (a) for large values of uint = 10'000 W/(m2K), (b) for small uint = 8 W/(m2K). Source: SPF Institut für Solartechnik.

The evaluation of the influence of rain or frost on uncovered absorbers used as a heat source for heat pumps has received little attention so far and no model was found that included these effects. Within T44A38 further model validations will be carried out as well as system simulations for the determination of the sensitivity of system performance on the inclusion of additional terms in the collector simulation models.

Models for the combination of photovoltaic modules with solar thermal application, so called PV/T collectors, have been reviewed in Charalambous et al. (2007) and Zondag (2008). A model extension for uncovered PV/T collectors that is an extension to the thermal model in EN12975 has been presented by Stegmann et al. (2011). The general recommendation to carefully check the compatibility of the material and collector design with possible water vapor condensation and wetting of cold parts applies for all non-hermetically sealed collectors that are operated below the temperature of the ambient air, especially also for PV/T collectors as an electric device.

4 Heat pump models

Simulation models for heat pumps have been reviewed for T44A38 by Dott et al. (2013). A review on heat pump and chiller models has also been given by Jin & Spitler (2002). 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. For the evaluation of new more sophisticated system concepts, a more detailed modeling 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 behavior over long-term periods like full years or short-term periods to evaluate for example the control behavior.

Empirical black box models are quite widespread, because the representation of the component behavior in the system is sufficiently precise and furthermore the required data of individual products are mostly available. Physical models, or better models 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 for solving the states and flows of the refrigerant cycle for each simulation time step. Quasi steady state performance map models 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, as e.g. described in Afjei (1989) and implemented in the simulation software Polysun (Marti et al. 2009). 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. An exemplary performance map for the coefficient of performance (COP) of an air-to-water heat pump is shown in Fig. 2. The extension of black box steady state models for the inclusion of dynamic effects such as for icing / defrosting and for the thermal inertia in the condenser or evaporator has been described e.g. in Afjei (1989).

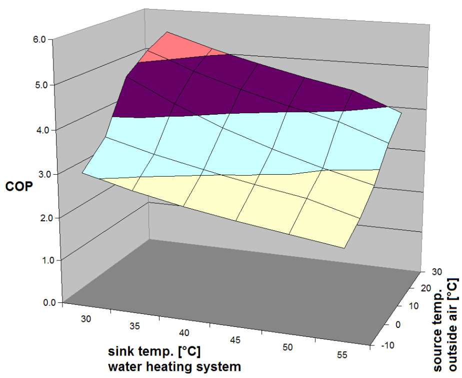


Fig. 2: Exemplary COP performance map of an air-to-water heat pump (Source: (Dott et al. 2012)).

More complex models are available that calculate the performance of the heat pump based on the performance of the compressor and the overall heat transfer coefficients of the evaporator and of the condenser (Jin & Spitler 2002; Bühring 2001; Bertsch & Groll 2008;

Sahinagic et al. 2008; Madani et al. 2011; Heinz & Haller 2012). The compressor may thereby be simulated based on assumptions for the volumetric and isentropic efficiency or based on a performance map that can be obtained from the manufacturer of the compressor. These models have the advantage that they are more flexible and can thus be used to study changes in the heat pump circuit such as the inclusion of two evaporator heat exchangers in series – one for the use of air as a heat source and the second one for the use of brine from a solar heat source, and/or an additional desuperheater to provide DHW while the heat pump delivers space heat. These additional model features may justify the higher computational effort that is needed to compute the thermodynamic states of the refrigerant in the heat pump cycle iteratively.

For the modeling of heat pumps that can take heat from solar collectors for the evaporator and may thus run on higher source temperature levels than usual, special attention has to be paid in order not to overestimate the performance of the heat pump in this application:

A very simply approach for a black box model is to assume that the COP of the entire heat pump is a more or less constant fraction of the thermodynamically maximum possible value for ideal heat pump cycles, the Carnot-efficiency. However, an extrapolation of COP values with this approach to temperature lifts that are much lower or much higher than the ones for which this model is calibrated cannot be expected to produce reliable results due to the fact that changing operating conditions and thus changing pressure ratios of the refrigerant lead to lower exergetic efficiencies with a fixed internal pressure ratio of e.g. scroll compressors especially at low temperature lifts of only a few K. Furthermore, if the temperature lift is taken as the difference between the temperatures of the heat source and the heat sink it has to be taken into account that this temperature lift is not equal to the temperature lift between the evaporation and the condensation of the refrigerant due to the temperature difference within the respective heat exchangers. Thus, an extrapolation of the heat pump performance in particular to low temperature lift operation - that may result from the use of solar heat - using a constant fraction of the Carnot-efficiency calculated from the temperatures of the heat source and of the heat sink is likely to overestimate the heat pump performance quite significantly (Fig. 3a).

Some physical heat pump models are based on the assumption that the temperature difference for the superheating after the evaporation of the refrigerant is a constant value. However, a comparison with measurements performed on an air source heat pump has shown that this assumption may overestimate the performance of the heat pump significantly when the source temperatures are increasing, which is of particular importance when solar heat increases the evaporation temperature above the usual level (Fig. 3b).

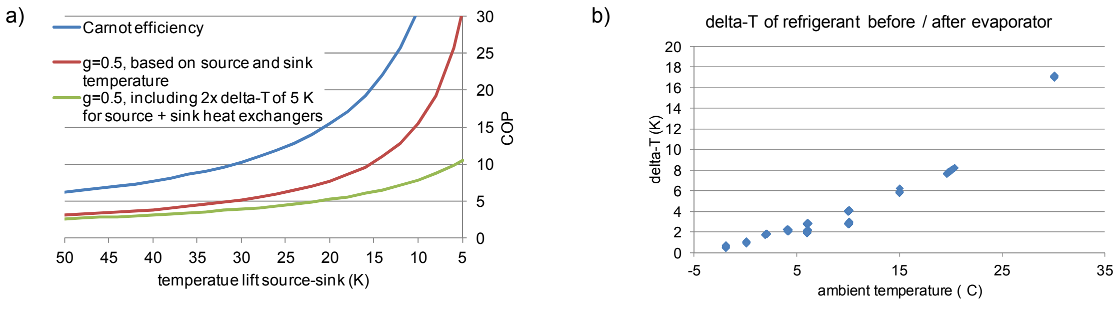


Fig. 3: a) Estimated COP of a heat pump based on the Carnot efficiency, source and sink temperature lift, and exergetic efficiency g. b) Measured superheating for variable source temperatures for an air-to-water heat pump with thermostatic expansion valve, source: (Heinz & Haller 2012).

Simulation models for capacity controlled heat pumps have been presented by several authors (Hiller & Glicksman 1976; Krakow et al. 1987; Afjei 1993; Lee 2010; Vargas & Parise 1995; Madani et al. 2011). However, only little data is available for the validation of capacity controlled heat pump models. In general, the lack of data availability for the parameterization of physical models and especially also for models of capacity controlled heat pumps is currently limiting the use of these models to heat pump development projects or project where extensive additional performance data measurements can be performed in the laboratory.

5 Ground heat exchange models

Models for the simulation of ground heat exchangers have been reviewed for the IEA T44A38 by Ochs et al. (2013). Additional reviews for vertical ground heat exchangers (VGHX) are available (Yang et al. 2010). Ground heat exchangers are used to extract heat from the ground at depths ranging from a few meters (shallow ground heat exchangers, SGHX) to some hundreds of meters (VGHX). Models for ground heat exchangers can be distinguished in (a) Finite Element Methods, (b) Finite Volume Methods, (c) Differential Methods, (d) Capacitance-Resistance Models, (e) Analytical, empirical models (response functions, g-functions), and (f) combinations of the above. Due to their higher demand on computation time, 3D FVM and FEM models are today only rarely used for annual simulations. However, they may still be the appropriate tools e.g. when effects of ground water flow (Bauer 2012) or moisture transport phenomena (Ramming 2007) are investigated.

One fundamental difference between VGHX and SGHX is the influence of seasonal temperature variations at the ground's surface that affects considerably the performance of SGHX but may be neglected for most VGHX systems where on the other hand the geothermal gradient may play a more important role (Fig. 4). Consequently, the simplifying assumptions that can be made in order to increase simulation speed differ quite substantially between models for VGHX and models for SGHX.

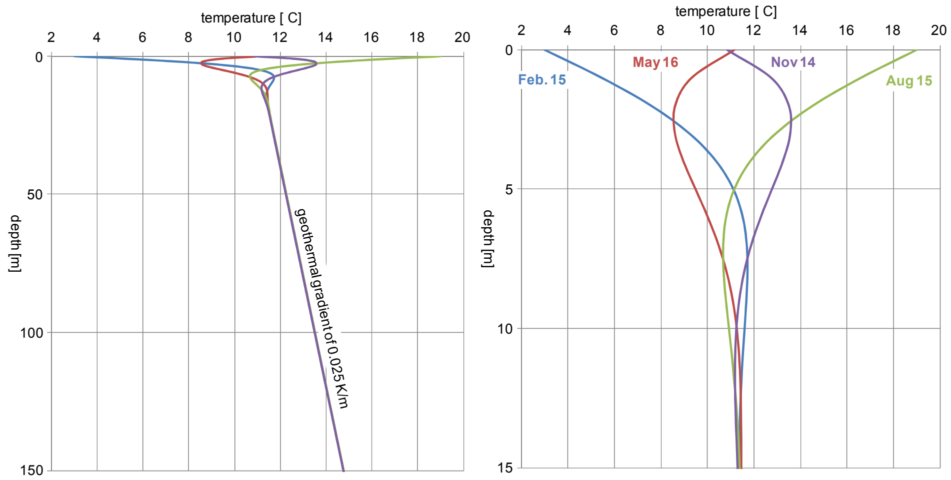


Fig. 4: Temperature of the undisturbed ground for a selected northern hemisphere climate as a function of the depth and time of the year for shallow depths (left) and medium depths (right).

The review on simulation models and their application leads to the following conclusions on simplifying assumptions:

Ice formation is of relevance for most SGHX (Glück 2009; Ramming 2007) whose design and sizing is usually such that return temperatures to the ground may be below 0°C for several weeks in the year and that are usually placed in moist ground. However, ice formation is usually not of importance for VGHX that are designed for operation above 0 °C for most of the year. This may be different if the operating conditions are below 0 °C for longer time periods (Eslami-nejad & Bernier 2012) and a large fraction of the VGHX is placed in moist ground or groundwater areas.

The geothermal gradient should be taken into account for VGHX, but usually not for SGHX. In many simplified VGHX models the geothermal gradient is not considered as a gradient but rather as an average temperature increase of the undisturbed ground, which may underestimate the performance of deep boreholes with insulation of the return line as e.g. reported in (Meggers et al. 2012).

The average moisture of the ground is of importance for both SGHX and VGHX because it affects the thermal conductivity and heat capacity of the soil. Ramming (2007) concludes that taking into account moisture transport phenomena and/or the time variable influence of rain does not change the simulation results for SGHXs significantly. Sealing of the ground surface however might change the moisture content of the soil on the long term. It has also been claimed that recharging the ground with temperatures above a certain level may induce changes in the long-term moisture of the ground and lead to the reduction of overall ground conductivity. No detailed information about this concern was found in the scientific literature.

5.1 Vertical ground heat exchangers

Three dimensional (3D) models for the simulation of VGHX can be found in Bauer (2012) and Li & Zheng (2009). For VGHX, a common approach for reducing calculation time is to split the calculation into a far field problem and a near field problem. The near field is affected by short term changes in heat extraction as well as by heat transfer between the upward flowing and the downward flowing fluid and is solved on a small time step basis. The far field problem determines the temperature at the outer boundary of the near field after a certain amount of time based on the superposition of analytical solutions for constant heat extraction over time. This temperature only has to be recalculated at longer time intervals of days or even a week.

Analytical solutions for heat extraction from VGHX are easy to use and very efficient in terms of computational time. Most analytical solutions are based on simplifications such as constant ground conductivity and diffusivity, as well as homogeneous temperatures of the ground before the start of heat extraction. Three main analytical solutions have been presented for the development of temperatures with time at any distance from a borehole with constant heat extraction or constant temperature. Of these three models, the cylindrical source model (CHS) and the infinite line source model (ILS) do not account for the limited extension of the borehole and thus the regeneration of the ground temperatures from above or below the extension of the borehole (Yang et al. 2010). This may not be a problem for short term estimations, but it may lead to an underestimation of the performance of the ground heat exchanger in the case of long term heat extraction, and to an overestimation of the performance in the case of long term net heat injection. For this reason, the finite line source model (FLS), that was proposed for the simulation of VGHX by Claesson & Eskilson (1987) is the preferred analytical model for most applications today. Based on this finite line source model, an analytical solution for the temperature at the middle of the borehole length at any distance of the borehole has been presented by Claesson & Eskilson (1987). A solution for deriving the average temperature over the length of the borehole has been presented by Lamarche & Beauchamp (2007). By temporal superposition, also pulsed extractions and injections and thus variable extraction/injection profiles can be simulated and by spatial superposition, the influence of neighboring boreholes can be accounted for. However, short time responses and heat transfer effects between the upward and downward flowing fluid in the borehole cannot be covered by these analytical solutions.

A popular approach for the determination of the temperature at a given distance of the borehole after a time of constant or – by temporal superposition – variable heat extraction is to use g-functions that were proposed by Eskilson (1987). The concept of g-functions is based on the fact that the non-dimensional time response of a linear system to a step change will be identical for systems with similar boundary conditions. The g-functions themselves can be derived from analytical solutions of line source models, or from numerical simulations using FV, FE, or FD approaches. Once a g-function is known, the temperature difference between the outer margin of the borehole and the undisturbed ground after a time of known constant heat extraction can be calculated based on the thermal properties of the ground. Although Eskilson presented an analytical solution of the FLS theory, he used a detailed 2-D simulation model (the SBM model described below) for the determination of the g-functions, including also the temperature response of multiple boreholes by spatial superposition. These functions were stored in a data base in order to avoid time expensive computations every time a calculation is performed. This implies a lack of flexibility because g-functions need to be pre-calculated for each configuration and a new configuration cannot be solved without this pre-processing task. The analytical solution of the FLS problem presented by Lamarche & Beauchamps (2007) may be a very efficient way to overcome this problem.

For VGHX, the work conducted at Lund University has been used as a reference for many years (Claesson & Eskilson 1987; Hellström 1991). The superposition borehole (SBM) model has been developed by Eskilson (1986) for the calculation of heat extraction from borehole fields. The three-dimensional temperature field around the boreholes in the ground is calculated by superposing two dimensional axi-symmetrical numerical solutions from each borehole. Today, different branches exist for this model implemented into the software TRNSYS (Pahud 2012; Nussbicker-Lux et al. n.d.). A validation of the model with measured data has been carried out by (Nussbicker-Lux et al. n.d.).

For densely packed and equally distributed borehole fields that are used for ground heat storage, the duct storage model (DST) has been presented that treats the whole borehole field as a near field that can be sub-divided into the region between the boreholes that is simulated with a 2D FD approach and the region within the boreholes that is solved differently (Hellström 1989).

The EWS model has originally been developed for single boreholes by Huber & Shuler (1997). The EWS model simulates the earth in a radius of about 2-3 m around the borehole based on a one dimensional finite difference approach with the Crank-Nicholson algorithm. In its original version, the temperature at the outer boundary of this cylinder is determined by an analytical solution based on the ILS theory. Later, this part has been replaced by g-functions of Eskilson and at the same time the model has been extended from single borehole calculation to multiple boreholes (Huber & Pahud 1999). This model is today also implemented in the system simulation software Polysun (Marti et al. 2009) and in the Carnot Blockset for Matlab/Simulink (CARNOT 2009).

The near field problem may be divided into the simulation of the region outside the borehole, i.e. between the borehole and the far field, and the region within the borehole. An overview on different models for both problems has been given by Yang et al. (2010).

5.2 Shallow ground heat exchangers

SGHX may be of quite different shape such as shown in Fig. 5. Due to the shallow depth (usually well below 5 m) SGHX are strongly influenced by weather conditions such as variation of the ambient temperature, solar radiation and long-wave radiation as well as rain and snow (including thawing). In addition, freezing of the soil next to the pipes may play an important role. The knowledge of the relevant parameters for the mechanisms mentioned above is usually poor.

Three dimensional (3D) models for the simulation of SGHX have been presented in Raumming (2007) and in Wu et al. (2010). Giardina (1995) describes a finite difference model (available for TRNSYS as TESS Type 556) that simulates a buried horizontal pipe in the middle of a cylinder of earth represented by several capacitance-nodes in radial direction and in the axial direction of the pipe. Another approach has been to use a 2D finite difference model corresponding to a vertical cut normal to one collector pipe's path that is mirrored at the boundary to the earth segment of the next parallel pipe (Tarnawski & Leong 1993; Ramming 2007; Glück 2009; Esen et al. 2007). In these models, the capacitance nodes are usually not divided along the fluid's path (in contrast to the model by Giardina). For the calculation of heat transfer to and from the ground, the arithmetic average temperature of the inlet and the outlet of the fluid are used. The model of Ramming (2007) accepts time-dependency of ground properties such as moisture, water infiltration, etc. However, the author concludes that the soil properties in 1.5 m depth change only over long time-periods, and therefore constant values can be used for one year. Glück (2009) argues based on a rough estimation of the influence of precipitation that the effect of heat input into the ground by precipitation can be neglected, and precipitation is therefore only a factor that effects the long-term water content and thus the heat transfer coefficient and latent energy changes (water/ice) within the ground.

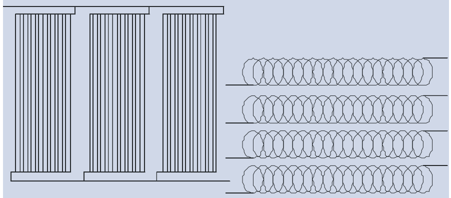
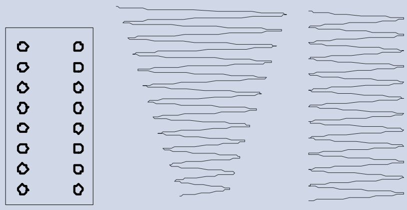
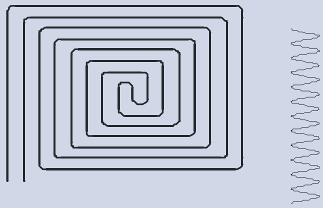
Type	figure	scheme
horizontal ground heat exchanger capillary collector meander/harp		
trench heat exchanger compact absorber basket / helix		
building integrated heat exchanger energy pile, basement/ wall absorber		

Fig. 5: Selection of available ground heat exchangers (Ochs & Feist 2012a).

An analytical model based on g-functions is suggested by Cauret & Bernier (2009). Due to the linear character of the governing equations, it is claimed that spatial superimposition and temporal superimposition can be applied in a similar way as for VGHX.

Piechowski (1999) solves the heat and moisture transfer equations for a horizontal U-pipe. He demonstrated that partial linearization of the heat and moisture diffusion equations does not result in any significant change in the simulation results as compared with the fully non-linear form of those equations. A simulation tool that is especially designed for the simulation of energy piles is PILESIM2 (Pahud 2007), a software that is based on the TRNSED feature of TRNSYS. Wu et al. (2010) used the commercial CFD software package FLUENT to predict the thermal performance of a portion of horizontal-coupled slinky and straight heat exchangers. Double spiral coil ground heat exchangers have been simulated by Bi et al. (2002). Simulation models for air to earth heat exchangers that can be used for preheating of ambient air that is used for building ventilation have been presented by several authors (Bojic et al. 1997; Mihalakakou et al. 1994; Santamouris et al. 1995; Tzaferis et al. 1992).

As shown by Ochs & Feist (2012a) most configurations of ground heat exchangers can be modeled with 1D (R-C) models. For brine and water driven systems discretization along the path of the fluid is not necessary. Instead the fluid-ground coupling can be modeled as a semi-isothermal heat exchanger. Ground heat exchangers with a more complex geometry such as trench or basket collectors and construction integrated systems have to be modeled in 2D (or 3D). FEM is usually the tool for complex geometries. With the PDETOOL, Matlab provides functions and interactive tools to solve PDE problems of the form:

$$d \cdot u' - div\left(c \cdot grad(u)\right) + a \cdot u = f \qquad (6)$$

where u is the dependent variable (depending on time and position). Applied to heat transfer, d is the volumetric heat capacity $d = \rho \cdot c_p$, c is the thermal conductivity, a is the convective heat transfer coefficient and f the source term. Using the Method of Lines partial differential equations can be transformed in ordinary differential equations, which can be solved with Matlab/Simulink. Thus, finite element models can be directly coupled to the building and system simulation. The heat capacity method is applied to account for freezing (Ochs & Feist 2012b).

6 Models for artificial storage devices

Models for artificial storage devices such as water storage, PCM storage and artificial ground storage systems are not treated in this paper but have been reviewed in a report for T44A38 by Sunliang & Siren (2013).

In order to overcome the mismatch of heat production and consumption, the thermal storage is often equipped with the solar and heat pump system. Currently, there are mainly two types of thermal storage depending on the phase state of the thermal mass. If the thermal mass experiences a phase change process during the operating process, the storage is categorized into the type of latent energy storage. On the other hand, if the thermal mass keeps in the same phase state during the operating process, the storage is categorized into the type of sensible energy storage.

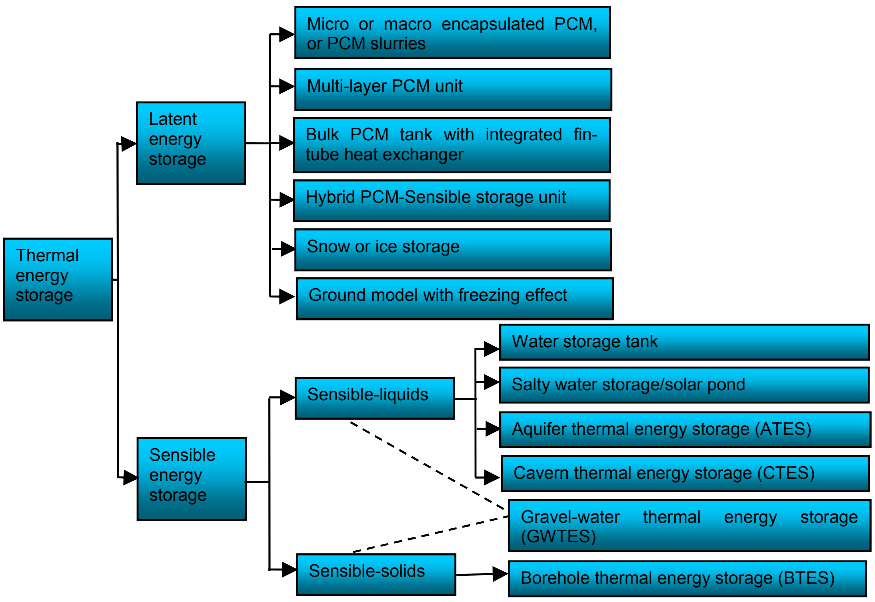


Fig. 6: The categories of thermal energy storage models listed in this report. The sensible energy storage part of the figure is referred to the figure in (Sunliang, 2010).

As shown in Fig. 6, the latent energy storage can be further classified into six types: micro or macro encapsulated phase change material (PCM) or PCM slurries, multi-layer PCM unit, bulk PCM tank with integrated fin-tune heat exchanger (HX), hybrid PCM-sensible storage unit, snow or ice storage, and ground model with freezing effect. The macro-capsulated PCM storage module is to encapsulate the PCM inside the macro-shaped capsules, such as balls, cylinders, and cubic boxes (Regin et al. 2009). The micro-encapsulated PCM is to encapsulate the PCM inside the micro-sized (BASF, 2009) or even nano-sized (Dupont, 2007) capsules. The micro-encapsulated PCM can be evenly mixed with sensible heat transfer fluid (such as water) to form PCM-slurries (Schranzhofer et al, 2006). Multi-layer PCM unit and bulk PCM tank with integrated fin-tube heat exchanger are developed to increase the heat transfer area between the working fluid and the PCM storage unit (Brousseau et al, 1996; Simard et al, 2003; Streicher et al, 2008). The hybrid PCM-sensible storage unit is mostly referred to the water tank storage immersed with capsulated PCM storage modules (Ibanez et al, 2006; Bony et al, 2007). Snow and ice are phase change materials both based on water (Skogsberg et al, 2001; Stewart et al, 1994). Ice is the solid phase of the pure water, while snow is a mixture of ice, air, and other particles (sand, soil, pollutions, etc). Ground model is normally treated as the sensible energy storage, but if freezing effect of surrounding soil is considered, it should be classified into the latent energy storage (Eslami-nejad et al, 2012).

On the other aspect, sensible energy storage is traditionally used storages without phase change process. It can be classified into two types: sensible-liquids and sensible-solids. Sensible-liquids are based on liquid thermal mass, whereas sensible-solids are based on solid thermal mass. For the sensible-liquids storage, water storage tank is most commonly used (Klein, 1976; Newton, 1995; Kuhn et al, 1980). Salt gradient solar pond is also included in the sensible liquids type due to its typical salt and temperature gradient, which is very different from pure water storage tank (Ouni et al, 1998; Ouni et al, 2003; Maozhao et al, 2006). Furthermore, the sensible-solids storage for the solar and heat pump application is often the rock bed thermal storage (Hughes et al, 1976). Moreover, the aquifer thermal energy storage (ATES), cavern thermal energy storage (CTES), gravel-water thermal energy storage (GWTES), and borehole thermal energy storage (BTES) (Novo et al, 2010; Bauer et al, 2010) all belong to the sensible energy storage. However, the models of ATES, CTES, GWTES, and BTES are not specifically focused in this chapter.

7 Discussion

The review of component models for the simulation of combined solar and heat pump heating systems has revealed a large number of models and model options for the simulation of solar thermal collectors, heat pumps and ground heat exchangers. For some models validation has been performed with measured data and it has been shown that due to the particularity of the combination of solar thermal collectors with heat pumps effects may have to be taken into account that are usually not included in annual performance simulation models of other systems that use either solar thermal collectors or heat pumps.

For solar collectors, several aspects may have to be taken into account when they are operated as a heat source for heat pumps, possibly also acting as air source heat exchangers, and with operating temperatures that are below the temperature of the ambient air and the dew point of the ambient air. Several models including validation are available for heat gains due to condensation of water vapor on uncovered collectors. Other aspects such as frosting and changes in the heat transfer coefficient due to frosting or changes in emissive and absorptive properties due to dew or frost on the surface are generally not included in current modeling tools.

A wide range of heat pump models from empirical COP performance maps to detailed simulation of the refrigerant cycle can be found. Weak points in most models seem to be the lack of validation of low temperature lift applications that may result from the use of heat from solar collectors, as well as a lack of data and validation for the parameterization of heat pump models for capacity controlled compressors and heat pumps that have become the standard air source heat pumps for space heating and domestic hot water preparation in many countries.

For the simulation of ground source heat exchangers the models range from detailed 3D FEM or FVM simulation for special applications to Capacitance-Resistance Models and analytical / empirical models –that demand less computation time and may be sufficient for most general applications. The most widely used models seem to the combination of 2D and/or Capacitance-Resistance Models for near field problems including the borehole itself with analytical and empirical solutions for the far field influence of heat extraction or injection over longer time periods. The accuracy of the long term temperature development prediction is not only important for heat extraction, but also for net heat injection (e.g. solar recharging). There is a lack of information on the concern that solar heat injection into the ground might lead to moisture migration with possible drying out and subsequent loss of conductivity and performance.

Nomenclature

$\dot{q}$ area specific heat gain rate, W/m^2

ϑ temperature, °C

Subscripts

amb ambient air

$gain$ heat gain by the collector

$gain+$ heat gain including latent gains from condensation

lat latent heat gain / loss of the collector

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