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Solar Heat Pump Standard Assessment Model

Jacob van Berkel[a]*, Onno Kleefkens[b], Felix Lacroix[c]

a Entry technology Support BV, Spoorbaanweg 15, 3911 CA Rhenen, The Netherlands
b Phetradico, Boskamp 16, 3828 VR Hoogland, The Netherlands
c RVO, Croeselaan 15, 3521 BJ Utrecht, The Netherlands

Abstract

Due to their extremely high COP's, solar thermal systems may advantageously be combined with heat pump systems. Though these systems are now entering the market, the lack of open energy performance assessment tools has been acknowledged by the Dutch authorities (RVO), and has led to initiation of this project.

The investigated systems typically comprise a relatively small (< 5 kWth) heat pump, combined with a 200 liter DHW-tank and around 5-25 m^2 of solar collector, in combination with a ground- or ambient air heat source. Modeling of the entire system is done conform the regulating Dutch Standards for Energy Performance of Buildings: 1) To determine the rationale of the system combination and 2) To assess the energy performance of the combined system.

The model developed in this research is versatile and straightforwardly programmed in Excel, and is used already to prepare Declaration of Conformity of commercial heat pump systems.

The paper addresses: 1) Modeling of the relevant physical phenomena and 2) Results of a practical example Solar Thermal Heat Pump system.

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Keywords: Solar Assisted Heat Pump, Energy Performance Model

1. Preface

Does a residential heat pump system benefit from extension with solar collector? What is the benefit, and how can it be assessed and accepted in standard energy assessment procedures?

On forehand the benefit of combining solar collectors with heat pumps indeed seems clear: A heat pump with high Coefficient of Performance (COP~3-4), during sunny periods combined with a solar thermal system with an even higher COP~30-50, would result in a more efficient system. The question is whether the improved efficiency justifies the increased system costs and whether a similar efficiency increase could be achieved also in a different manner, possibly at lower costs.

This project is focused on compact solar thermal heat pump systems, comprising compact stores (around 200 liter) and moderate collector areas (5-15 m^2), typically systems in which a solar system assists the heat pump system, see Van Berkel [1] and [2]. This investigation is complementary to work done on an international level i.e. IEA SHC Task 44 HPP Annex 38 [3], where projects tend to focus on relatively large systems with >1 m^3 storage tank and > 15 m^2 solar collector area.

2. Introduction

* Corresponding author. E-mail address: entry@xs4all.nl

The combination a solar and heat pump system could be done rudimentary, as with two parallel and separate systems, or more elegantly as an integrated system, with a shared storage tank with back-up heating.

Schematically, figure 1 provides the layout of a system with three principal ambient heat sources: 1) Ground heat exchanger; 2) Ambient air and 3) Solar collector. In practice a solar collector is combined either with a ground source or ambient air heat source into one system.

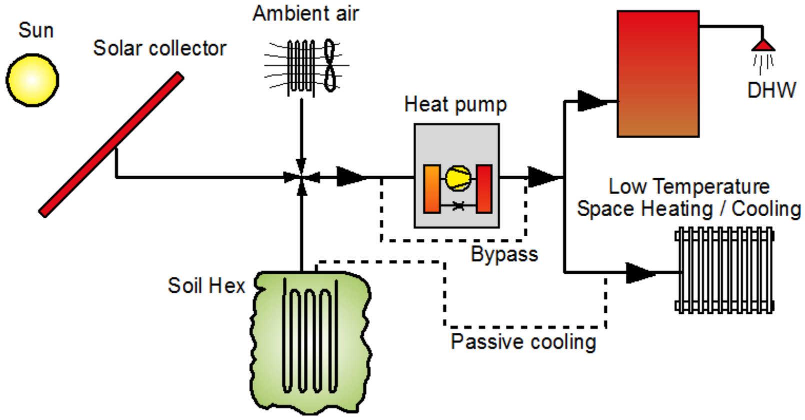

Fig. 1. Solar assisted heat pump system (generic layout)

Normally the three heat sources provide thermal energy to the evaporator of the heat pump. The heat pump however can be bypassed e.g. when the solar collector is able to provide thermal energy at sufficient temperature directly to the space heating- or DHW. In case a ground heat exchanger is used, the building can be cooled during summertime by circulation of water through the soil heat exchanger. In general, the solar collector is also used to regenerate the ground heat source, which cools down during normal operation of the heat pump.

In effect the various combinations ask for an integrated solar thermal and heat pump system, and immediately raises the question regarding control strategy. In summary the questions within this project are:

- What are the quantitative benefits (energy performance) of the combined system? How to evaluate the system in the Energy Performance Standards: In the Netherlands: NEN7120 [4].
- Regarding control of the combined system: How (at what temperature level) must the solar collector operate for optimal performance?

Due to the complex boundaries, meteorological conditions, space and DHW- heating loads, above questions require numerical simulation; Theoretical analysis (calculus) cannot provide the necessary details and experimental investigation is too expensive and cumbersome.

3. Standard Energy Assessment Model

A first question regarding to numerical simulation relates to the program environment. As the project is publicly funded, the results including the program should be available for anyone and free to use. This led to the selection of Microsoft Excel as the program environment, though it is acknowledged that other (closed and more expensive) options like Trnsys, Polysun and Matlab in principle are easier to use.

Regarding the program set-up, it must be noted that: 1) Parameters (meteorological conditions, space heating and DHW-loads) vary from hour to hour and possibly also within the hour and 2) Thermal capacities are relevant especially when a thermal store is integrated in the system e.g. DHW-storage and ground heat Source. Acknowledging these features, choices are made regarding the program principles:

- Time stepping
- Meteorological conditions and solar collector model
- Space- and DHW heat load, DHW-tank
- Heat pump performance modeling and auxiliary energy demand
- System control

These topics to be elaborated in the remainder of this chapter.

3.1. Time stepping

Time stepping is fundamental; Momentary quantities must be integrated time-wise to arrive at annual quantities (e.g. for energy consumption). The accuracy of the numerical integration depends on the time step and integration procedure.
Unlike methods (like (pr)EN15316-4-3 (F-chart) and the Dutch Standard) which use a stationary (time averaged) method on a monthly time basis, for accuracy here a time-dependent method is used with an hourly time step. This method is also used in e.g. prEN15316-5. Simulation is done with hourly input, with time-stepping (temporal discretization) throughout a whole year (8760 hour). Every time step the transients (d/dt) are calculated and used to assess the temperature of the next time step (Euler explicit time stepping).

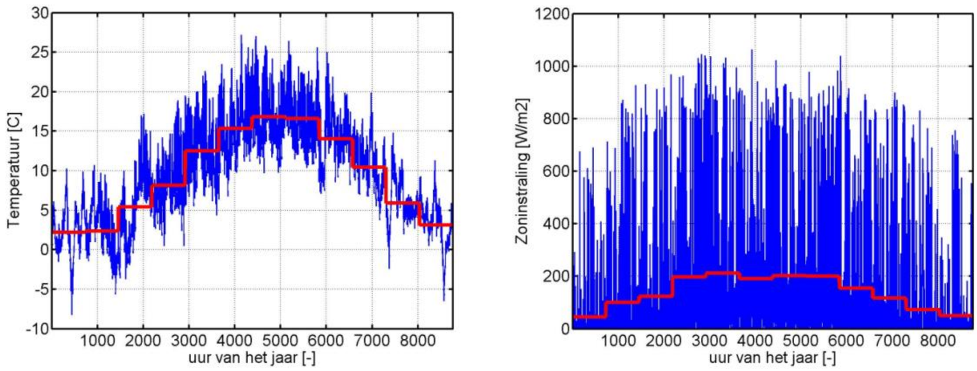

Fig. 1. Hourly (blue line) and monthly averaged (red line) values for temperature and solar irradiation, location De Bilt (NL).

The current method is different from the binned temperature method, which is also an hourly method (without time integration however). As solar thermal heat pumps depend on too many independent (non correlated) variables and thermal capacities, the binned method cannot be used. The current hour-by-hour method is less complex, but takes more (8760) program lines.

3.2. Meteorological conditions and solar collector

As input for meteorological conditions the Standard NEN 5060-A2 [5] is used, providing a wide variety of meteorological parameters on an hourly basis. For a covered/uncovered collector the most important parameters are: 1) Solar irradiation (global, diffuse- and direct on a horizontal plane and normal to the sun); 2) ambient dry bulb temperature and 3) wind (velocity and direction). Parameters like humidity, enthalpy and precipitation in this case are not taken into account. As an example, in figure 1 hourly data for ambient temperature and global solar irradiation is shown. The solar irradiation components are recalculated to a total incoming radiation on an arbitrary collector plane, using the well known irradiation model of Perez [6], with a coefficient for ground reflection of 0.2. Figure 2 gives the most important collector heat fluxes taken into account in this case.

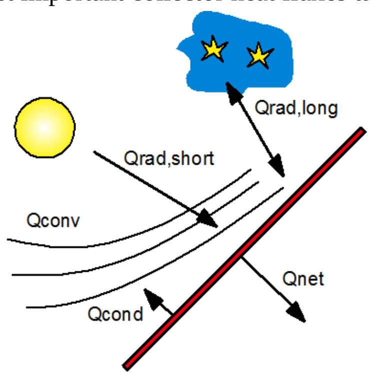

Fig. 2. Ambient heat fluxes from and towards a covered or uncovered collector.

The exchange of thermal energy to and from the collector, in this case, is modeled following Bunea [7]:

$$\dot{Q} = IAM\,\eta_0\,G - c_1(T_m - T_a) - c_2(T_m - T_a)|T_m - T_a| - c_3 u(T_m - T_a) - c_4\sigma(T_m^4 - T_s^4) - c_5\frac{dT_m}{dt} - c_6\,u\,G \quad (1)$$

With:

$\dot{Q}$	net thermal energy flux, in W/m^2;
IAM	Incidence Angle Modifier [-], accounting for the inclination of the sun relative to the collector plane.
η_0	dimensionless optical efficiency of the collector;
G	total solar irradiation directed towards the collector, in W/m^2;
c_1	factor representing the (conductive) thermal loss of the collector, in W/(m^2.K);
T_m	momentary mean collector temperature, in ºC;
T_a	momentary ambient (dry bulb) temperature, in ºC;
c_2	factor representing the convective, temperature dependent thermal loss, in W/(m^2.K^2);
c_3	factor representing the wind-dependent thermal loss of the collector, in J/(m^3.K);
u	wind speed, in m/s;
c_4	factor representing the (long wave radiation) thermal loss of the collector.
σ	Stefan–Boltzmann constant for radiation, in W/(m2.K4)
T_s	radiation temperature of the hemisphere, in K
c_5	factor for thermal capacity of the collector, in J/(m^2.K)
t	time, in s
c_6	factor representing the effect of wind on the optical efficiency of the collector, in s/m

The radiation temperature of the hemisphere is not a standard variable in the meteorological database, and, following De Vries [8], here derived from the ambient temperature by:

$$T_s = 0.0552\,T_a^{1.5} \quad (2)$$

3.3. Alternative heat sources: Ambient air and ground

The temperature of the ambient air is simply provided by the meteorological data, see chapter 3.2 and figure 1. For the ground source, the Dutch Energy Performance Standard (NEN7120) discerns between two cases: 1) a closed loop heat exchanger, exchanging heat by conductance and convection in the ground and an open loop (ground water) heat source. Temperatures depending on the time of the year are given in figure 3.

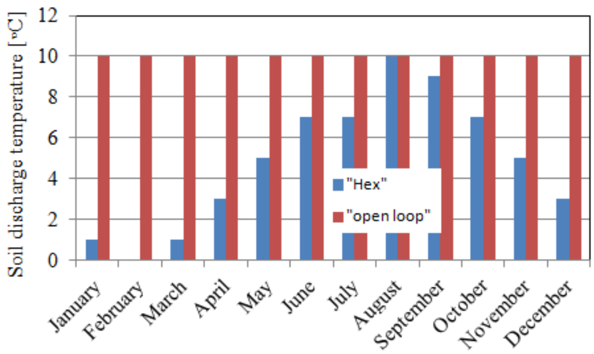

Fig. 3. Discharge temperatures of ground thermal energy sources:

3.4. Space- and DHW-heating: Energy and temperature demands

Following the Standard NEN7120, it is assumed that below a threshold ambient temperature, temperatures of the heating emitter and required heating power both vary linearly with ambient temperature, see figure 4.

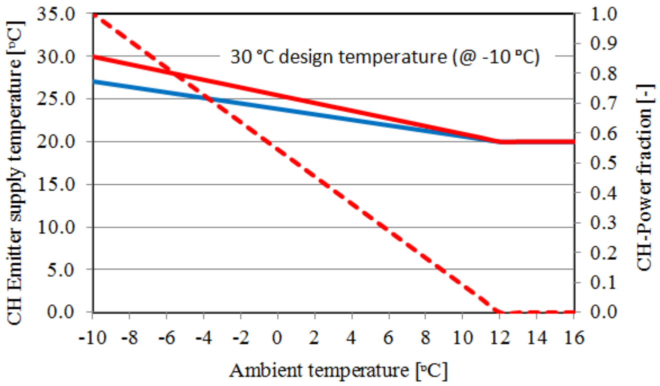

Fig. 4. Example: CH emitter supply (red solid line) and return temperature (blue solid line), depending on ambient temperature. The dashed line gives the thermal power demand, as a fraction of the maximum power required at -10 °C. In this example, no space heating demand is assumed above +12 °C threshold ambient temperature.

Using the linear dependence and degree-hours principles, power demand for every hour of the year is calculated from the annual heat demand. The specific demand of course depends on the building type and climate. For the Dutch climate and residential buildings a number of threshold and supply and return temperatures are selected in the Energy Performance Standard.

For the DHW-demand, arbitrary (hourly) standard demand profiles can be adopted. The example here is based on a Dutch profile with 14 GJ/year, with 49 draws distributed over the day. As some draws take place within the hour, the profile is remapped into an hourly pattern, displayed in figure 5.

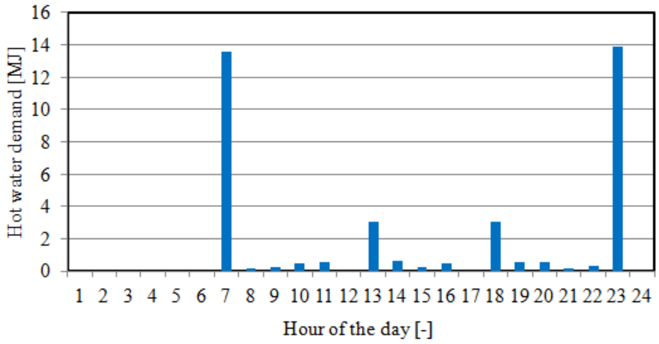

Fig. 5. Example: DHW demand: Example distribution throughout the day. In total 14 GJ/year

As shown in figure 5, the DHW-demand profile is dominated by two large (shower) draws at the beginning and end of the day. As stated, any (hourly) profile can be adopted and implemented in the Excel-program.

The temperature of the DHW delivered to the distribution system is prescribed by the Standard NEN 1006, stating that at any time hot water must be delivered at 55 - 60 °C (depending whether it is a recirculation- or non-recirculation system), unless periodic (weekly) system disinfection at high temperature takes place. The example in this paper is based on delivery of 60 ºC hot water supply temperature.

For peak shaving purposes, a DWH-tank is generally incorporated in the system. Modern systems use a stratified tank in which hot water accumulates at the top and cold water at the bottom of the tank. For accuracy in conceptual modeling of the system, the tank frequently is divided in 2-100 separate layers each having a particular temperature.

The Excel model developed earlier by Van Berkel [1] adopted 10 fixed volumes, variable temperature layers, which gave adequate accuracy. For implementation of the segment model (which needs evaluation for conduction and convection (displacement) every time step, macros were built in the Excel-program, that worked fine, but however was complex and not user-friendly.

For clarity and robustness, here the "plug flow" concept is adopted comprising two variable-volume, fixed-temperature layers. The two (hot and cold) water layers are assumed to be perfectly stratified and non-mixing. Though on an energetic basis this concept is equivalent to a perfectly mixed tank concept, the stratified version resembles better the reality encountered in short term storage tanks.

Adopting this strategy does not require intermediate evaluation and corresponding macros. In the program, the volume of the hot water layer straightforwardly is calculated on the basis of the begin-volume [liters]; DHW-energy demand [MJ], power supply by the heat pump or (directly) solar collector [W] and finally the tank heat loss [W].

3.5. Heat pump performance and auxiliary energy demand

The heat pump is modeled according to the Energy Performance Standard NEN7120. Assumed is an on/off heat pump with Coefficient of Performance (COP) and maximum thermal power (Pth) bi-linearly depending on weighted averaged evaporator and condenser temperatures, see figure 6.

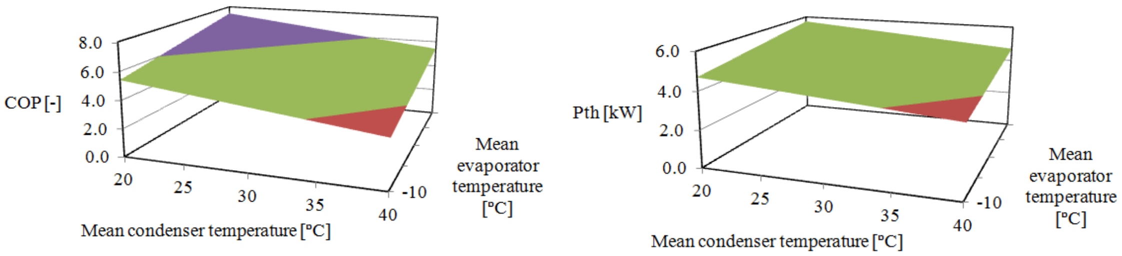

Fig. 6. Example: Left: Coefficient of Performance and (right) maximum thermal power of a heat pump, depending on mean supply (evaporator) and mean discharge (condenser) temperatures.

In the program, both COP and Pth are modeled with coefficients determining the flat planes. Control parameters determine whether the heat pump can be in operation, or is switched "off" due to e.g. exceeded temperature limits (too low evaporator or too high condenser temperature).

Using above model rules, the heat pump is considered to provide power for space heating (H) or DHW-heating (W). For compliance with the Energy Performance Standard, the heat pump performance data COP and Pth must be tested and measured according to the European Standards EN 14511 (on/off-machines) or EN14825 (inverter machines). Care must be taken that for delivery of high temperature heat (60 °C), the heat pump must be tested also at an adequate high temperature.

Given the COP of the heat pump, depending on energy supply and demand temperatures, the momentary electrical power demand can be calculated and summarized throughout the whole year.

Auxiliary energy is required to drive external pumps (heat pump internal pumps are incorporated in the COP), an auxiliary heating element (in case the heat pump cannot provide thermal energy) and system control.

Regarding the electrical element it is assumed that it is activated whenever the DHW tank energy content drops below 25% (e.a. 50 liters of hot water).

Pumps re-circulate space heating emitter fluid, solar collector fluid or (if present) the ground source fluid (water, brine or glycol). Their annual energy demand is calculated from the respective power demand and the operation time (for space- and DHW-heating separately).

3.6. System control

To further explain the system control in a general sense, the system as shown in figure 7 is adopted. It comprises a solar collector, with in addition a ground heat source as an alternative heat source. Without loss of generality, the alternative ground heat source can be replaced by e.g. an ambient air heat source.

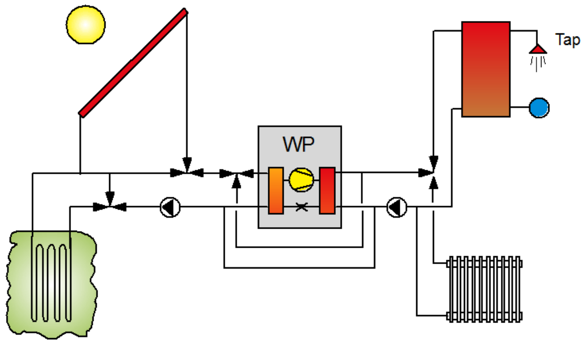

Fig. 7. Example system with solar collector and ground heat source.

The system can be switched in different configurations ("modes") by 3-way valves. The particular mode very much depends on the temperature that the solar collector can deliver to the heat pump system. Here it must be noted that the potential temperature of the solar collector depends on: 1) ambient conditions (solar irradiation, wind, temperature) and 2) on the mass flow rate and collector inlet temperature: In effect the collector mean temperature T_m. This dependency is governed by equation 1.

Table 1 gives the respective modes of operation:

Table 1. Solar collector/heat pump operation modes

Solar collector status	System mode
Collector cannot deliver temperatures above the ground source temperatures: Collector is switched off	"0"
Collector can, for a part, deliver required evaporator power at ground source temperature: Collector parallel to ground source at same temperature.	"1"
Collector can deliver full evaporator power, but not at space- or DHW-load temperature: Ground source switched off, collector feeds heat pump evaporator.	"2"
Collector can deliver space- or DHW-load at required temperature: Ground source and heat pump both switched off (collector directly feeds load, bypassing the heat pump).	"3"

If necessary and in case no heat is required for space- or DHW-preparation, the solar collector can be further switched on for regeneration of the ground source. For further explanation, as an example, figure 8 gives the modes of operation, depending on arbitrary solar irradiation and temperatures.

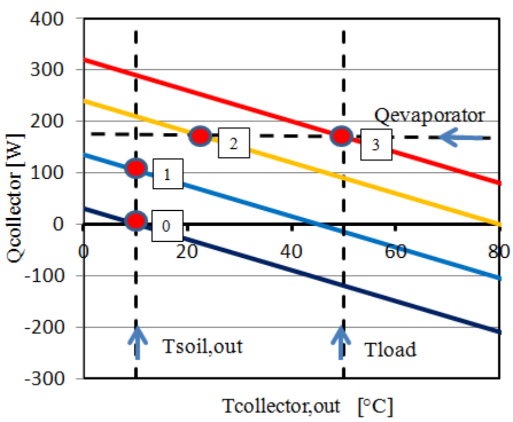

Fig. 8. System modes of operation, depending on solar irradiation level and temperatures. Shown is thermal power of a 1 m^2 collector depending on collector exit temperature, for 4 arbitrary levels of solar irradiation (dark blue, blue, yellow and red). Also indicated by vertical dashed lines are the reference temperatures for ground source exit temperature and heat load temperature demand. The horizontal dashed line denotes the heat load of the evaporator. The red dots indicate points of operation.

Clearly seen in figure 8 is that the collector power increases with solar irradiation but decreases with higher collector temperature.

Regarding system control it must be mentioned that though the program uses parameters that are generally not known in practice (e.g. solar irradiation), the principles of the control system could be identical (using e.g. collector exit temperature instead). Furthermore it is mentioned that a more advanced control of the collector addresses the exergy content of the collector heat flow (in terms of 2nd law of thermodynamics). Elaboration of this more advanced control strategy however is outside the scope of this paper.

4. Sample system results.

For convenience, compactness and ease of interpretation, results will be presented of an arbitrary, very straightforward system, using an ambient collector as the sole heat source, see figure 9.

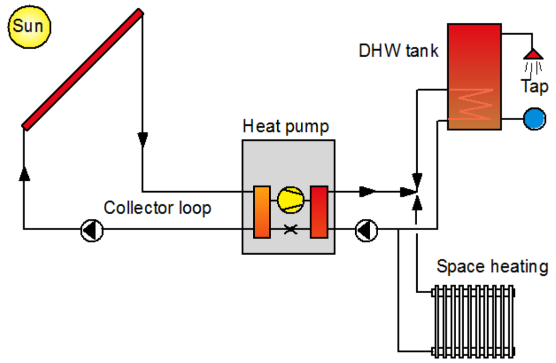

Fig. 9. Sample system

The collector is the sole heat source, always feeding thermal energy into the evaporator. This system is a simplification of the generic system explained earlier, thus operating exclusively in system mode "2".

Figure 10 gives details of an arbitrary (but realistic) example of the system components.

Specifications Heat Pump: EN 14511 Test Results					
Tcond;out	Tcond;in	Tevap;in	Tevap;out	COP	P(CH)
[°C]	*[°C]*	*[°C]*	*[°C]*	*[W/W]*	*[W]*
35.00	30.00	0.00	-5.00	5.00	4500
45.00	40.00	0.00	-5.00	4.00	4000
45.00	40.00	10.00	5.00	5.00	4500

Hot water tank:		
Tank Volume	180	[Liter]
Set temperature hottop	60	[°C]
Tank ambient temperature	20	[°C]
Heat loss factor	0.75	[W/K]
Set point max tank temperature	80	[°C]
Set point Heat pump on	0.50	[-]
Set point E-heater	0.25	[-]
Annual Hot Water Demand	7.5	[GJ]

Specifications Ambient heat Exchanger		
Collectoroppervlakte	20.0	m2
Inc. Angle Modifier (IAM)	1.00	[-]
Collector optical efficiency ho	0.70	[-]
Conduction heat loss coeff c1	20.00	W/K
Convection heat loss coeff c2	0.00	W/(K^2)
Wind loss factor c3	5.00	W/(K m/s)
Radiation loss c4	0.70	-
Thermische capaciteit c5	30000.00	J/(m2.K)
Wind loss factor opt. eff. c6	0.05	s/m

Space heating demand		
Threshold temperature	12	[°C]
CH-supply temperature (@ -10 C	30	[°C]
Space heating load	45.5	[GJ]

Auxiliary Energy		
Solar circulation pump	50	[W]
CH-circulation pump (H, W)	50	[W]
System control	5	[W]

Fig. 10. Sample system component specification

The Standard Assessment Model evaluates all relevant system parameters hour by hour, and can be used for initial system analysis and optimization. Figure 11, for example gives the mean temperature of the uncovered solar collector, in case it used for DHW-preparation.

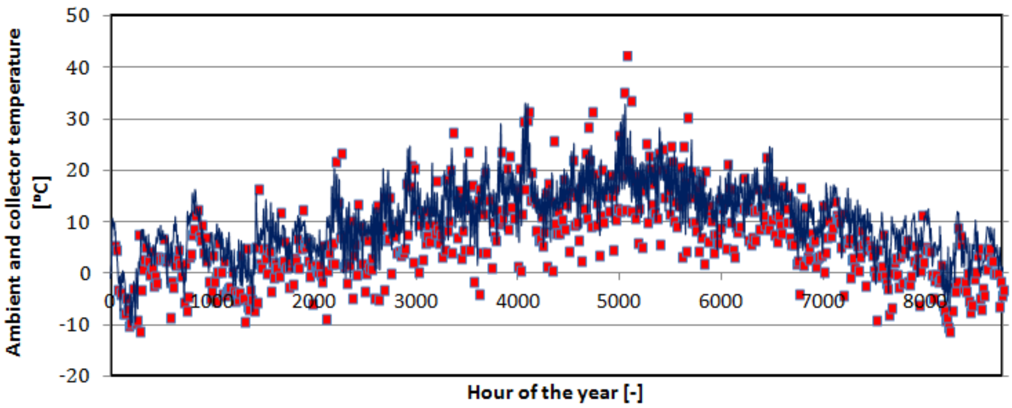

Fig. 11. Sample system. The red dots give the mean temperature of the collector in case it is used for water heating. The blue solid line is the ambient air temperature.

As shown in figure 11, during wintertime, in absence of high solar radiation levels, the mean collector temperature is generally below the ambient temperature. In summertime however, the collector temperature frequently lies above the ambient temperature.

As a further illustration of the optimization potential of the model, figure 12 gives the overall generation efficiency of the system, defined as: Total amount of thermal energy delivered at the heating and hot water system, divided by the total amount of electricity necessary to drive the system (compressor, pumps and control).

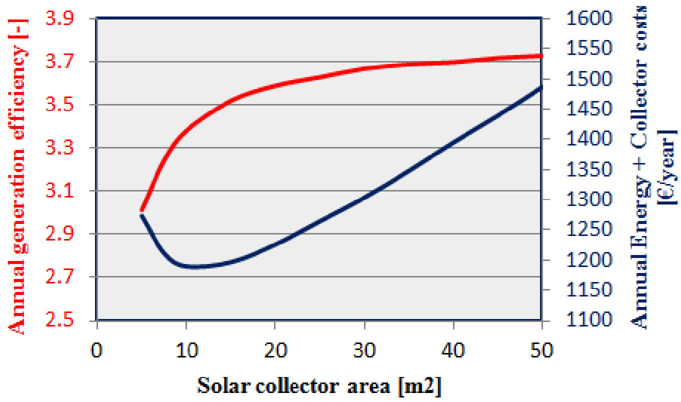

Fig. 12. Sample system. The red line indicates Annual generation efficiency; The blue line the annual electricity + collector depreciation costs (assuming: 25 ct/kWh electricity costs; 100 €/m^2 collector costs and 10% annual amortization)

Figure 11 indicates that increasing the collector area initially results in a sharp increase of annual efficiency, as a result of increasing collector and therefore heat pump evaporator temperatures. For small collector areas, the collector temperature is well below the ambient temperature. The added value of larger collector areas however decreases as at higher temperature the collector more and more looses thermal energy to the environment.

The economical optimal size of the collector depends on the benefits of higher efficiencies (in terms of reduced electricity costs) in relation to added collector costs. The blue line in figure 12 gives the marginal sum of electricity costs and collector costs, assuming 25 ct/kWh electricity costs; 100 €/m^2 collector costs and 10% annual amortization. Other costs, for the heat pump etc., are supposed to be constant. Given these assumptions, the optimal size of the collector is around 10 m^2 (in this particular case).

Main result of the Standard Assessment Model is that the annual generation efficiency now can be quantified. The performance figure can be used in the relevant Energy Performance Standards, thus recognizing the specific potentials of the combined Solar and Heat Pump Systems.

5. Further work.

As shown in this paper, the Standard Assessment Model is ready to be applied to specific systems. Important step in the assessment procedure is configuration-specific validation of the model. This work is now underway.

6. Conclusions and recommendations

- An open and versatile Standard Assessment Model is built for determination of the energy performance of combined solar thermal heat pump systems. The program is built in Excel and evaluates the system status for a year, hour-by-hour. The processing time needed to run the model is less than a second.
- Regarding modeling of the system components 1) meteorological conditions and solar collector; 2) heat pump; 3) DHW-tank; 4) heating and hot water energy and temperature demands, the model is built according to the principles of the European Energy Performance Standards.
- The model can be used for initial system analysis and optimization. An example shows that for an average house (around 50 GJ total heating load) the optimal size of the solar collector could be in the range of 10-20 m^2, depending on the precise assumptions regarding energy- and component costs.
- Following validation, the model may serve as a tool for evaluation of the energy performance e.g. within the Dutch NEN7120 standard. The model is open and available for anyone interested in combined Solar Thermal Heat Pump Systems.

Acknowledgements

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References

[1] Van Berkel, J, Solar Assisted Heat Pumps, Joint Government & Industry project, Commissioned by Ministry of Economic affairs, program DEN executed by the Dutch Enterprise Agency RVO, Project DENB128018, report may 1st, 2014.

[2] Van Berkel J, NEN7120 Conformity of Solar Assisted Heat Pumps, Commissioned by Ministry of Economic affairs, program DE executed by the Dutch Enterprise Agency RVO, Project DE14000040, report July 25th, 2016

[3] Hadorn, J.C., Solar and Heat Pump Systems for Residential Buildings, Wiley, 2015

[4] NEN 7120+C2/C3 (nl), Energy Performance of Buildings, including addenda.

[5] NEN 5060 (nl), Dutch Climatic conditions

[6] Perez, R., Seals, R., Ineichen, P., Stewart, R., & Menicucci, D. 1987, Solar Energy, 39, 221.

[7] Bunea et al., Mathematical modeling of unglazed solar collectors under extreme operating conditions, LESBAT + DTU, 2015.

[8] Vries, D. de, Design of a thermal photovoltaic/thermal combi-panel, Proefschrift Technische Universiteit, 1998.