PROPOSAL FOR A PERFORMANCE CALCULATION AND EVALUATION PROCEDURE FOR SOLAR COOLING APPLICATIONS

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1. Abstract

The emerging market for solar thermal cooling applications raises the need for a standardised performance evaluation of the thermally driven device as well as the whole systems. Such a procedure will also allow the comparison between different thermally driven devices, solar cooling systems among each other as well as solar cooling systems with renewable or conventional competing technologies. An additional benefit could be the chance to be able to develop procedures to predict average system performance based on standardised values.
While there exist a lot of different standards for vapour compression chillers (e.g. EN14511, EN14825), no standard for thermally driven devices covers all cases so far. They do not take into account the particularities of these emerging technologies and use differently defined boundary conditions. This paper will summarize the work on standardisation of thermally driven heat pumps/chillers done within SHC Task 38 and HPP Annex 34 as well as first national approaches and propose a standard site-dependent performance calculation method open for discussion and which will be one of the central work for the new IEA SHC Task 48 on “Quality Assurance and Support Measures for Solar Cooling”.

2. Performance figures

Compared to e.g. gas boilers or electrical compression heat pumps or chillers the performance evaluation for thermally driven cooling and especially solar cooling is more complex, since different kind of driving energies have to been taken into account and rated. A solar cooling system, needing a large solar collector to generate the driving heat, will very often make use of this collector to generate heat for other purposes in the non cooling season, raising the need to distinguish different operating modes and how to take this into account in the overall seasonal performance evaluation. Also will a solar cooling system often be used only to cover the base load (for economical reasons) and will be combined with a backup system, which than can be installed either on the hot side or the cold side of the chiller. Another difference between conventional and solar cooling systems is that not only the load and the heat rejection temperature varies, but also the driving energy will vary in power and maybe even in the temperature level. Therefore a first step is to identify the most important figures for performance evaluation and comparison with other systems.
One of the most used figures is the thermal coefficient of performance (COP) or energy efficiency ratio (EER) of the chiller, often additionally with the needed electrical auxiliary power, but with different system boundaries. Extending this from machine to a system COP/EER and even to a seasonal performance figure offers a variety of possibilities which will be discussed in the next chapters.

3. System boundary definitions

The analysis of the available standards for heat pumps showed that generally three system boundaries and four performance figures are commonly used. In Figure 1, system boundaries for a reference thermally driven heat pump TDHP system with the respective performance figures are shown.

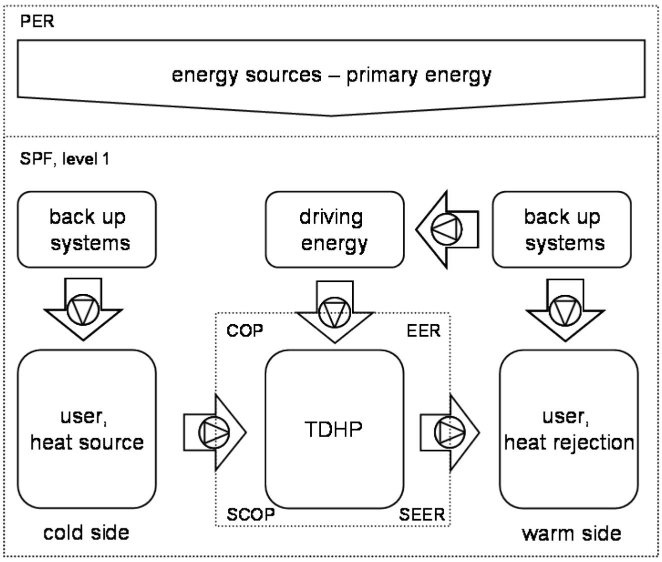


Figure 1: Temperature and solar irradiance bin matrix for the calculation of the solar input into the solar cooling system

The system boundary for the coefficient of performance (COP) and the seasonal COP (SCOP) for heating as well as the energy efficiency ratio (EER) and the seasonal EER (SEER) for cooling are drawn around the TDHP unit. These figures take into account only the energy inputs and outputs of the heat pump itself, which are obtained from the measurements on a test rig under specified, mostly steady state operating conditions. Following the practice of [1], the energy inputs and outputs are corrected for the energy consumption of the liquid pump and its influence on the heat transfer media, respectively.

The seasonal performance factor (SPF), at its highest level (as defined in [3]), takes into account the total system energy outputs and inputs of all system components. The SPF, however, does not take into account the source of the consumed energy, i.e. electricity or driving heat. The primary energy ratio (PER) takes into account the type of the energy source, an important figure for the calculation of the energy costs or the environmental impact of the system, for example. For the SPF and the PER, the energy inputs and outputs are obtained from the monitoring results. Further performance figures might also be of interest, such as the solar fraction or fractional energy savings, but are not the topic of this paper. The basic ideas are presented in e.g. [2] for solar cooling systems specifically and [3] for all thermally driven applications.

Unlike the electrically driven heat pumps (EDHP), the TDHP consume both electricity and thermal energy. By defining separate thermal and electrical performance figures, additional information on the unit or the system behavior is provided.

3.1 COP and EER

As described in the previous chapter, the COP and the EER are obtained from the laboratory measurements under defined, mostly steady state operation. These figures represent the efficiency of the unit at its nominal capacity under specified operating conditions. The corrections for the liquid pumps are defined in analogy to [1]. The upper sign (addition or subtraction) in the equations 1 through 4 applies for the case when the liquid pump is integrated into the heat pump unit, the lower sign for the case when it is not.

$$COP_{th} = \frac{\dot{Q}_H \pm \dot{Q}_{lp}}{\dot{Q}_{in,unit}} \quad (1) \qquad EER_{th} = \frac{\dot{Q}_C \pm \dot{Q}_{lp}}{\dot{Q}_{in,unit}} \quad (2)$$

$$COP_{el} = \frac{\dot{Q}_H \pm \dot{Q}_{lp}}{P_{in,unit} \pm \sum_i P_{lp,i}} \quad (3) \qquad EER_{el} = \frac{\dot{Q}_C \pm \dot{Q}_{lp}}{P_{in,unit} \mp \sum_i P_{lp,i}} \quad (4)$$

3.2 SCOP and SEER

The SCOP and the SEER represent the calculated seasonal efficiencies for a reference application and a reference climate, based on measurements on a test rig under steady state conditions at different loads. Unlike the COP and the EER, they represent the efficiency of the unit for time dependent conditions (e.g. oscillating source temperature, changing supply temperature etc.) for a certain period of time, usually a year.

The method proposed in this paper is based on the temperature bin method already used in standards for EDHP, e.g EN 15316-4-2 [4] or prEN 14825 [5]. The method is based on the division of the cumulative annual frequency of the outside dry-bulb temperature into temperature classes called bins. For every bin, an average operating condition is defined. It is then assumed, that the heat pump unit operates under this condition for the entire temperature range covered by the bin. A full description of the method, as well as some open questions can be found e.g. in [6] and [7]. The method has been also adopted as a basis for the calculation of SCOP and SEER for TDHP in IEA HPP Annex 34, as reported in [2].
The SCOP and the SEER are determined from equations 5 -8.

$$SCOP_{th} = \frac{(Q_H + Q_{DHW})}{\frac{Q_H}{SCOP_{on,th}} + \frac{Q_{DHW}}{PF_{DHW,on,th}} + Q_{aux}} \quad (5) \qquad SEER_{th} = \frac{Q_C}{\frac{Q_C}{SEER_{on,th}} + Q_{aux}} \quad (6)$$

$$SCOP_{el} = \frac{(Q_H + Q_{DHW})}{\frac{Q_H}{SCOP_{H,on,el}} + \frac{Q_{DHW}}{PF_{DHW,on,el}} + E_{aux}} \quad (7) \qquad SEER_{el} = \frac{Q_C}{\frac{Q_C}{SEER_{on,el}} + E_{aux}} \quad (8)$$

$SCOP_{on}$ and $SEER_{on}$ represent the efficiencies in the active heating or cooling mode, respectively. In this mode, useful energy output is delivered to the user. PF_{DHW} represents the efficiency of the unit while producing DHW. Q_{aux} and E_{aux} represent the auxiliary energy inputs like controls or the energy consumption in the stand-by mode.

For many applications of TDHP, an analogue definition of SCOP and SEER to the one used for their electrically driven counterparts can be used. This is the case in applications, where the main driving energy comes from a source which can basically follow the demand instantly, e.g. gas mains, district heating or waste heat. In this case, the temperature of the heat source, as well as its availability can be taken as constants. Thus, the $SCOP_{on}$ and $SEER_{on}$ can be determined from equations 9-12 which depend only on the ambient temperature. The total energy output, given in the enumerator, is determined from the sum of the heating demands in every temperature bin. This figure includes energy inputs both from the TDHP and the back-up units. The terms in the denominators represent the energy input to the unit, including all auxiliaries as well as the energy consumption of the reference back-up system.

$$SCOP_{on,th} = \frac{\sum_{j=1}^{n} h_j \cdot \dot{Q}_H(T_j)}{\sum_{j=1}^{n} h_j \cdot \left(\frac{\dot{Q}_{H,TDHP}(T_j)}{COP_{th}(T_j)} \right)} \quad (9) \qquad SEER_{on,th} = \frac{\sum_{j=1}^{n} h_j \cdot \dot{Q}_C(T_i)}{\sum_{j=1}^{n} h_i \cdot \left(\frac{\dot{Q}_{C,TDHP}(T_{ji})}{EER_{th}(T_j)} \right)} \quad (10)$$

$$SCOP_{on,el} = \frac{\sum_{j=1}^{n} h_i \cdot \dot{Q}_H(T_j)}{\sum_{j=1}^{n} h_i \cdot \left(\frac{\dot{Q}_{H,TDHP}(T_j)}{COP_{el}(T_j)} + P_{bu}(T_j) \right)} \quad (11) \qquad SEER_{on,el} = \frac{\sum_{j=1}^{n} h_i \cdot \dot{Q}_C(T_j)}{\sum_{j=1}^{n} h_i \cdot \left(\frac{\dot{Q}_{C,TDHP}(T_{ji})}{EER_{el}(T_j)} + P_{bu}(T_j) \right)} \quad (12)$$

However, in the case of solar cooling, the availability of the driving energy is dependent on the climate conditions, ambient temperature and solar radiation for the chosen climate. In this case, the calculation procedure has to be extended. This will be discussed in Chapter 4.

4 Proposal for a simplified method for the calculation of SEER of the chiller in solar cooling systems

Despite the complexity of the problem to determine the performance of a given system, a simplified method to calculate the SEER of a cooling system would help to compare different solar cooling systems and conventional systems. Therefore this paper proposes an extension of the simplified temperature bin method described in Chapter 3 to take into account the special situation of a solar thermal cooling system.
The method described in Chapter 3 uses only one parameter (ambient air temperature) for the determination of the seasonal efficiency of the chiller. This approach gives good performance estimations if this performance is mainly a function of ambient temperature, as it is the case with electrically driven heat pumps or direct fired TDHP. However, for thermally driven chillers operating in solar cooling systems, the availability and quantity of solar energy is a crucial parameter to evaluate their performance under realistic operating conditions. Thus, the temperature bin method has to be extended with the information about the solar radiation for the considered climate.

4.1 Calculation of solar input to the system

For every temperature bin, the information on solar radiation can be added. As the same hourly mean value of the ambient temperature can coincide with a high mean solar irradiance or a very low one, it was decided to apply solar irradiation bins in the same manner as it was done for the temperatures. Thus, the one-dimensional bin method is being extended to a two-dimensional one. This 2D temperature-radiation bin matrix has the format shown in Figure 2. With the 2D bin matrix , the solar energy input to the system can be estimated for every combination of ambient temperature and solar irradiation, following eq 13, thus, the overall solar energy available for the chiller is described by eq 14:

$$Q_{sol,i,j} = \begin{cases} \eta_{coll,i,j} \cdot G_i \cdot A_{spec} \cdot h_{i,j} \\ 0, \ \forall Q_{sol,i,j} < 0 \end{cases} \quad (13) \qquad Q_{sol} = \sum_{i=1}^{n} \sum_{j=1}^{m} Q_{sol,i,j} \quad (14)$$

In the first step of the method development, an assumption is made, that the energy coming from the solar system has exactly the temperature level needed for the respective operating condition of the chiller. The influence on the method accuracy of this assumption has still to be investigated, considering the uncertainty of the method itself and the general purpose at this stage.

	G_1	G_2	…	G_i	G_{i+1}	…	G_n
T_1	$h_{1,1}$	$h_{2,1}$	…	$h_{i,1}$	…	…	$h_{n,1}$
T_2	…						…
…	…		…				…
T_j	…			$h_{i,j}$			…
T_{j+1}	…				…		…
…	…					…	…
T_m	$h_{i,m}$	…	…	…	…	…	$h_{n,m}$

Figure 2: Temperature and solar irradiance bin matrix for the calculation of the solar input into the solar cooling system

The collector efficiency can be calculated for every 2D bin from the quadratic efficiency equation:

$$\eta_{coll,i,j} = \eta_0 - a_1 \cdot \frac{T_m - T_j}{G_i} - a_2 \cdot \frac{(T_m - T_j)^2}{G_i} \quad (15)$$

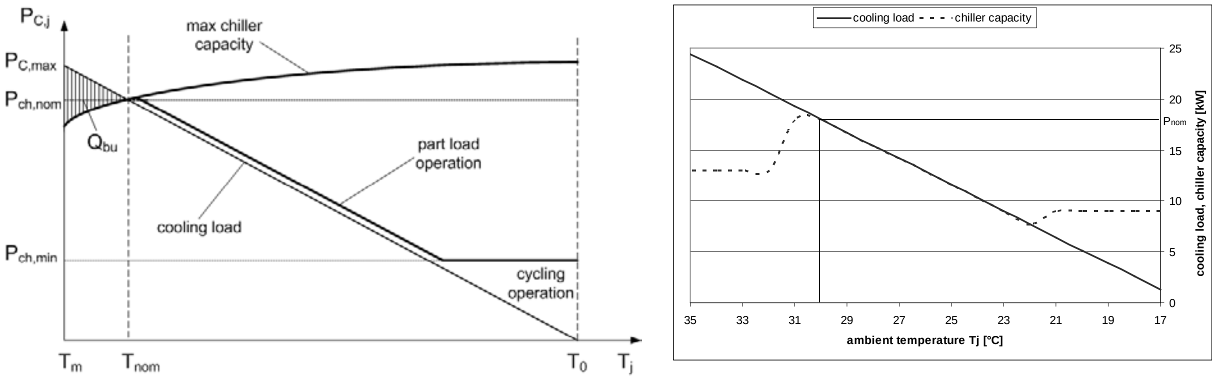
For the estimation of the energy provided by the solar cooling system, the collector area is an important factor. This area used in a solar cooling system will generally depend on the application (building type), the local climate, the chiller type and requirements and on the cooling strategy. However, for a standardised calculation procedure, a generally applicable value, depending on the nominal capacity of the chiller, should be used to ensure comparable conditions for all chillers. Here, the following formula for the calculation of the specific collector area is used, as proposed in [8]:

$$A_{spec} = \frac{1}{\overline{G} \cdot \eta_{coll,nom} \cdot EER_{nom}} \quad (16)$$

4.2 Calculation of chiller performance

For the calculation of the chiller performance, a number of thermal and electrical EER values for different operating conditions are needed, as described in Chapter 3. In reality, the chiller can operate both at full capacity and under part load conditions. If both the driving temperature and the temperature of the chilled water are assumed constant under given conditions, as well as the respective mass flows of the energy carriers, the chiller performance is mainly depending on the part load condition and the temperature of the heat rejection system. With this simplification, the number of measurements needed as input to the calculation is reduced to a reasonable number. However, it might be feasible to define more temperature levels for the chilled water outlet, as well as corresponding mass flows if the method is to be used for standardised calculation and labelling, as it is done for electrically driven air-conditioners in EN14511 or EN14825.

For the calculation of the seasonal performance of the chiller, a cooling load has to be assumed. Following the practice of standards like EN 14825, the cooling load is defined as a linear function of the nominal chiller capacity $P_{ch,nom}$ corresponding to the ambient temperature T_{nom} and the zero cooling load for the first temperature bin (T_0) not included in the cooling season, Figure 3a. Other values on the line are interpolated or extrapolated from these two. In this way, the maximum cooling load $P_{C,max}$ is also given.
The interception of the chiller characteristic curve giving the maximum chiller capacity under given conditions and the cooling load is the nominal operation point of the chiller. Note that T_{nom} refers to the heat rejection temperature T_{HR}. As previously described, T_{HR} is constant in a certain ambient temperature range, which means that $P_{ch,nom}$ is not clearly defined. This has to be overcome by specifying $T_{HR,j}$ for every T_j in the future.
For smaller cooling loads, the chiller works under part load conditions until the minimum capacity $P_{ch,min}$ corresponding to a certain cooling load P_C as a function of T_j is reached. For lower cooling demand, the chiller has to operate intermittently. This operation range is considered in the calculation method with a cycling coefficient C_c. Both the thermal and the electrical efficiency of the chiller are diminished because of thermal losses and additional electricity consumption while the chiller is in the standby mode, not delivering useful energy.



(a) (b)

Figure 3: Approximation of the cooling load for given climate and nominal chilling power in theory (a) and calculated from assumed data and climate conditions with the proposed method (b)

From the operation conditions of the chiller (temperature level, part load condition, cycling operation), the needed thermal driving energy is calculated interpolating between the measurement data from a defined matrix of operation points.

The temperature of the heat rejection circuit has a large influence on the chiller efficiency. Due to different designs and control strategies of the heat rejection system, it is very hard to give a generalized dependency of the water/brine temperature T_{HR} entering the condenser and absorber as a function of the ambient temperature T_j. As a first proposal, T_{HR} was assumed as constant within a certain temperature range, Table 1.

Table 1: Assumed temperatures of the heat rejection at the inlet to the condenser/absorber as a function of ambient temperature

T_j [°C]	<22	22-26	27-31	32-36	>36
$T_{HR,j}$ [°C]	25	28	30	33	35

The values given are only a rough estimation, more work is needed to evaluate real systems and to find appropriate correlations for heat rejection technologies (cooling tower, dry cooler, etc.). It would be beneficial to have the same number of bins both for ambient temperature and heat rejection temperature. This will be further discussed e.g. in IEA SHC Task 48.

The available driving energy for the chiller is limited by the energy supply from the solar system on one hand, and the limited capacity of the chiller for cooling loads exceeding its maximal capacity on the other hand (Figure 3b). The area where the difference between the cooling load and the chiller energy output has to be covered by a back-up unit is shown as Q_{bu} in Figure 3a. For the proposed method, the back-up unit is considered to be electrically driven chiller supplying exactly the needed amount of energy to meet every time the cooling load. For the first calculation a constant EER of the back up unit was used. However, a temperature dependent EER, assumed as average value for market available products, will be used in future. Since many installations use a back-up unit on the hot side of the chiller (e.g. a gas burner), this option will also be added in the next developments of the method.

The energy needed from the back up unit for a temperature bin *j* is calculated as:

$$Q_{bu,j} = \begin{cases} \text{if } Q_{C,ch,j} \geq Q_{C,j} \text{ then } 0 \\ \text{if } Q_{C,ch,j} < Q_{C,j} \text{ then } Q_{C,j} - Q_{C,ch,j} \end{cases} \quad (17)$$

The overall electricity consumption is obtained by summing up the electricity consumptions of the chiller itself and the back up system:

$$Q_{el,tot} = \sum_{j=1}^{m} \left(Q_{el,ch,j} + Q_{bu,j} \right) \tag{18}$$

The total electrical *SEER*, including the back-up system, is calculated according to eq 8 and the $SEER_{el}$ for the chiller only, excluding the back-up system, might be interesting for the comparison between different products. The Thermal SEER for the chiller is calculated from the energy supplied by the unit divided by the driving energy supplied to the unit:

$$SEER_{el,tot} = \frac{Q_C}{Q_{el,tot}} \quad (19) \qquad SEER_{el,ch} = \frac{\sum_{j=1}^{m} Q_{C,ch,j}}{\sum_{j=1}^{m} Q_{el,ch,j}} \quad (20) \qquad SEER_{th,ch} = \frac{\sum_{j=1}^{m} Q_{C,ch,j}}{\sum_{j=1}^{m} Q_{dr,ch,j}} \quad (21)$$

Finally, the solar cooling fraction F_{sol} can be defined as the amount of energy provided by the solar driven technology divided by the total energy demand, if the back-up unit provides cooling energy directly to the user:

$$F_{sol} = \frac{Q_{C,ch}}{Q_C} \tag{22}$$

The calculation method was tested using available data for the performance of the system components. For the chiller, example data based on technical specifications of market available absorption chillers were used. For the collector efficiency, coefficients η_0=0.8, a_1=3.3 and a_2=0.015 corresponding to a good quality flat plate collector were assumed, just for an example. The mean collector temperature T_m was set at 85°C, which corresponds to the assumed chiller's nominal operating conditions. The global solar irradiation was calculated for an exact southern orientation and an incidence angle of 30°. Climate data for Palermo, Italy were used, which were obtained from [9]. The bin matrix was made for 1K temperature bins and five bins were used for global irradiance.
For the calculation of the EER_{el}, the nominal electricity consumption of the chiller unit and the calculated electricity consumptions of the liquid pumps for the secondary side of the generator, absorber/condenser and evaporator circuits, as described in Chapter 3.1, were calculated based on market available units, The pump efficiency was assumed to be at 50%.

Using assumed data for the collector and the chiller, the cooling load and the corresponding chiller capacity over the ambient temperature are calculated, Figure 3b. For ambient temperatures higher than the nominal condition, the chiller capacity cannot meet the load and the back-up chiller has to deliver the difference. Reaching its minimum part load at about 36% of P_{nom} and ambient temperature of 22°C, the chiller has to start operating intermittently. For this operation mode, extra electricity consumption is considered by introducing an EER diminishing factor to consider thermal losses and increased electricity consumption for stand-by mode. The curve for the chiller capacity corresponds well to the theoretical one from Figure 3a. In the area of high and low ambient temperatures, the capacity remains constant due to the assumption of constant inlet temperatures to the condenser and absorber. An increase of capacity in the area of minimum part load condition is an effect of both lower heat rejection temperature and switching from part load operation for temperatures >22°C and intermittent operation for lower temperatures.

For the assumed climate, chiller performance data, solar collector performance and specific solar collector area A_{spec}, the calculation method implemented in a spread sheet yielded the following performance figures:

$SEER_{th}$	$SEER_{el,tot}$	$SEER_{el,ch}$
0.67	7.81	16.22

5 Conclusions and further steps

A calculation method is proposed for the estimation of the seasonal performance of a thermally driven chiller operating in a solar cooling system. The method is based on a two dimensional bin method derived from existing standardised procedures. A number of assumptions have to be made for a transparent comparison of different chillers.

- Temperature level on the generator inlet and the water mass flow rate: there are no limits regarding the temperature level. Nominal temperatures provided by the manufacturers are used, also for the testing of the chiller;
- Temperature on the evaporator outlet and the water mass flow: constant values both for the temperature and mass flow are used. For a standardised method, predefined values should be used, for example 7/12°C or 18/23°C as in EN14511, and prEN14825;
- Temperature on the condenser/absorber inlet and the water mass flow: the temperature of the heat rejection unit is a function of the ambient temperature. Appropriate correlation has to be developed, preferably for a number of different technologies.
- Thermal and electrical EER for different operating conditions: Generally, a chiller will often operate under part load conditions and these conditions are generally not covered by manufacturers curves and specification data

Since this is just a proposal at an early stage of discussion, much more work has to be done to evaluate the results of this method for several systems. The assumptions made have to be checked for viability and are open for discussions and extensions.

Further steps:

- Consideration of different temperature levels of the solar energy delivered to the generator due to changing operating conditions;
- Correlation for the EER of the backup unit depending on the ambient temperature (and possibly time of the year for ground coupled units);
- Development of a more accurate correlation between the ambient temperature and heat rejection temperature;
- Consideration of more chilled water temperatures;
- Differentiation of part load controls: driving temperature and/or mass flow, heat rejection temperature and/or mass flow;
- Optimisation of the number of bins, both for temperature and for solar irradiation;
- Investigation of options to specify the optimum value for A_{spec};
- Possibility to extend the method to the whole solar cooling system, including different configurations.
- Possibility to extend the method to a multi purpose whole solar cooling systems, used out of summer period as a solar heating and DHW system.

This paper just aims to open the discussion about a simplified method for the performance evaluation of a solar thermal cooling system. The method described here is only a proposal open for discussions and will have to be validated. Further steps in this direction will be done within the new IEA SHC Task 48 on "Quality Assurance and Support Measures for Solar Cooling", starting in October 2011 for a 3,5 years duration till 2014.

Nomenclature

A_{spec}	specific collector area, in m^2/kW
E	electrical energy, in kWh
F_{sol}	solar cooling fraction
G	solar radiation, in kW/m^2
h	temperature bin hours
η_{coll}	collector thermal efficiency
P	electrical power, in kW
Q	heat energy, in kWh
$\dot{Q}$	heat power in kW
T	Temperature, in K

Subscripts

aux	auxiliary
bu	back-up
ch	chiler
C	cooling
DHW	domestic hot water
el	electrical
H	heating
lp	liquid pump
ref	reference
sol	solar
th	thermal
tot	total

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