CONDENSATION HEAT GAINS ON UNGLAZED SOLAR COLLECTORS IN HEAT PUMP SYSTEMS

E. Bertram[1], J. Glembin[1], J. Scheuren[1], G. Rockendorf[1]

[1] Institut für Solarenergieforschung Hameln/ Emmerthal, Am Ohrberg 1, 31860 Emmerthal, Germany,
Corresponding Author: e.bertram@isfh.de.

Abstract

The performance of ground coupled heat pump systems with borehole heat exchangers can be significantly improved by unglazed solar collectors. In this combination, collector operation temperatures below the ambient air level and hence additional condensation heat gains occur. For an investigated and monitored system in Limburg, Germany, the condensation yield was determined to 3.7% or 19 kWh/(a m²). Thereby, condensation shows a significant dependency on the season. During the summer months, only 0.8% of the total collector yield is induced by condensation, while in winter it increases to 13%. The implementation of an established condensation model to the collector model according to EN 12975 is demonstrated. As main result the investigated the heat pump system performance is only marginally improved due to the condensation heat gains.

Introduction

Unglazed collectors offer low temperature heat at low costs. Consequently, they are well applicable in combination with heat pump systems. According to the low temperature level the collector is frequently operated below dew point temperature. This results in additional heat gains through condensation. Several publications predict a wide range for these additional condensation gains. For the Swedish climate, a fraction of 10 to 25% of the collector heat gains have been stated depending on the operating temperature, with a tendency to a higher yield increase in humid climates [1]. In comparison, for Germany lower condensation heat gains of less than 10% are predicted [2]. No prediction is made about the influence of the additional gains on the annual performance factor of the whole heat pump system.

In this context, a new collector model for TRNSYS including the condensation effect has been developed and implemented. Applying the model, simulation results have been generated, which could be compared with the measurement data of a monitored system. Subsequently, a system simulation study allows investigating the influence of condensation gains on the annual performance factor. For all systems discussed the unglazed collector is a second heat source in addition to a borehole heat exchanger.

Unglazed Collector Model with Condensation

The thermal behaviour of an unglazed collector is commonly described by the EN 12975 model [5]. This description disregards heat gains through condensation for operation temperatures below the dew point temperature. Therefore, the characterization by EN 12975 is extended in the developed model by an extra term, which describes condensation effects. The applied condensation model [2] describes the additional heat gain through condensation $\dot{q}_{cond}$ according to eq. (1). It has been validated for a metal roof collector in [3].

$$\dot{q}_{cond} = -\frac{R_{Air}}{R_D} \cdot \frac{h}{p_0 \cdot c_{Air}} Le^{-\frac{2}{3}} \cdot U_{conv} \cdot (p_s(T_{abs}) - p_s(T_d)) \tag{1}$$

R_{Air}, R_D	Specific gas constants of air and water vapour in kJ/(kg K)
h	Enthalpy of evaporation of water in kJ/kg
p_0	Local standard air pressure mbar
c_{Air}	Specific heat capacity of air kJ/kgK
Le	Lewis number
U_{conv}	Convective heat loss coefficient of the collector in W/m²
p_s	Pressure of saturated water vapour in mbar
T_{abs}, T_d	Absorber temperature, dew-point temperature in °C
$\dot{q}_{cond}$	Condensation heat flow in W/m²

The condensation heat flow $\dot{q}_{cond}$ calculated by eq. (1) is an additional term to the thermal collector heat flow with dry surface conditions $\dot{q}_{coll,dry}$ calculated by the collector performance according to EN 12975 in eq (2).

$$\dot{q}_{coll,dry} = \left(\eta_0 \cdot (1 - b_u \cdot w) - \frac{T_m - T_{amb}}{G''} \cdot (b_1 + b_2 \cdot w) \right) \cdot G'' \tag{2}$$

$\dot{q}_{coll,dry}$	Collector heat flow with dry surface conditions according to EN 12975 in W/m²
η_0	Conversion factor
w	Air speed in m/s
b_u, b_1, b_2	Performance loss coefficients according to EN 12975 in s/m, W/(m²K), Ws/(m³K)
T_{amb}	Ambient air temperature in °C
T_m	Mean fluid temperature in the collector (arithmetic mean of inlet and outlet) in °C
G´´	Net solar irradiance (reduced by infrared radiation balance) in W/m²

The collector heat gain $\dot{q}_{coll,fluid}$ including condensation is then obtained by a heat balance for the collector fluid eq. (3).

$$c_{p,coll} \frac{\partial T_m}{\partial t} = \dot{q}_{coll,dry} + \dot{q}_{cond} - \dot{q}_{coll,fluid} \tag{3}$$

$c_{p,coll}$	Heat capacity of the collector in kJ/(kg K)
$\dot{q}_{coll,fluid}$	Usable collector heat flow in W/m²

Important to notice is, that $\dot{q}_{coll,dry}$ refers to the mean fluid temperature whereas the condensation heat gain according to eq. (1) refers to the (mean) absorber temperature. The latter is determined with eq. (4) by calculating the absorber surface temperature T_{abs} from the average collector fluid temperature T_m and the internal thermal conductivity of the collector U_{int}.

$$T_{abs} = \frac{\dot{q}_{coll,fluid}}{U_{int}} + T_m \tag{4}$$

U_{int}	Internal thermal conductivity of collector in W/(m² K)

Typical values of U_{int} for unglazed collectors range from 45 W/m²K for metal roof collectors with a conversion factor η_0 of 0.60 to 200 W/m²K for plastic rubber collectors with a conversion factor of 0.87. Hence, the internal thermal conductivity U_{int} has to be taken into account in order to determine the absorber temperature. For the typical U_{int} values given, even a small collector heat flow of 200 W/m² results in a comparatively large temperature difference. Using the typical values for U_{int} the difference between the mean absorber and the mean fluid temperatures for the rubber collector is 1 K and for the metal roof collector 4.4 K. With the temperature difference the saturated vapour pressure of the absorber temperature $p_s(T_{abs})$ increases, and hence, as T_{abs} is below T_d, the driving partial pressure difference $(p_s(T_{abs}) - p_s(T_d))$ is reduced. Moreover, this influence of a low U_{int} value even increases in cases with higher collector heat flows.

Therefore, it is to conclude, that the internal thermal conductivity of a collector has to be taken into account, if condensation effects are to be determined. Exceptions can be made for collector designs with very high U_{int} values. Here the occurring temperature differences between fluid and absorber temperature are negligible. Further, this correlation implies that collectors with a high U_{int} are most suitable for measuring the effects of condensation [3].

The described model is implemented in a TRNSYS type for unglazed solar collectors. Herewith a tool is available that allows investigating condensation effects in systems in the course of the year.

Determination of condensation heat gains from system measurement

The yearly condensation heat gain is investigated on a monitored heat pump system. The system consists of 44 m² unglazed metal roof collector, a 16 kW heat pump with a heat production of 36 MWh/a, and 14 tube in tube borehole heat exchangers of 17 m depth each. The heat pump supplies heat for space heating and domestic hot water in a large single-family dwelling in Limburg, Germany. All meteorological and energetic data of the collector field are measured on-site in a resolution of one minute time steps. The measurements include collector in- and outlet temperatures, collector flow rate, long wave irradiance, air velocity, hemispherical solar irradiance, ambient air temperature and humidity. The unglazed collector serves heat as a second heat source and is connected in series to the borehole. Its energy is supplied only to the heat source side of the heat pump. A simple hydraulic scheme is displayed in Figure 1. Further results of the system are presented in [4]. The condensation heat gain is investigated over a period of one year from November 2006 to November 2007.

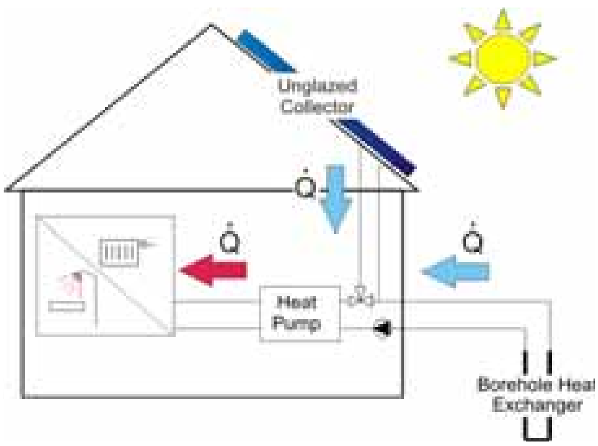

Fig. 1: Scheme of the monitored pilot plant in Limburg.

The direct measurement of the additional condensation heat gain $\dot{q}_{cond}$ is impossible, because of the interference between the condensation heat gain and the heat gains through convection and radiation. Therefore, the determination of the condensation heat is carried out by a comparison of the measured meteorological and energetic data and the collector model presented above, being aware to the fact, that this indirect method has a higher uncertainty.

The model parameters of the collector have been determined within an additional EN 12975 test in the ISFH test lab. The measured data of the pilot plant have been applied to the model in one minute time steps. As a result the model provides the simulated collector outlet temperature $T_{out,sim}$, the absolute collector yield $\dot{q}_{coll,sim}$, and the condensation yield $\dot{q}_{cond,sim}$ in the dynamic course of the year. This simulated outlet temperature and the yield of the collector hold two important characteristics. First, they indicate the accordance of the complete model to the measurements by comparison of the simulated values $T_{out,sim}$ and $\dot{q}_{coll,sim}$ to the measured collector outlet temperature $T_{out,mea}$ and collector heat gain $\dot{q}_{coll,mea}$. Second, the model allows to specify the condensation heat gain $\dot{q}_{cond,sim}$ as part of the absolute collector yield $\dot{q}_{coll,Sim}$. A schematic description of the procedure is shown in Figure 2. With this setup the condensation heat gain for the collector in combination with a heat pump for one year of operation has been determined. For the calculation of the condensation heat gains according to eq. (1) the convective loss coefficient is required. It is retrieved from a collector heat balance and the measured performance data.

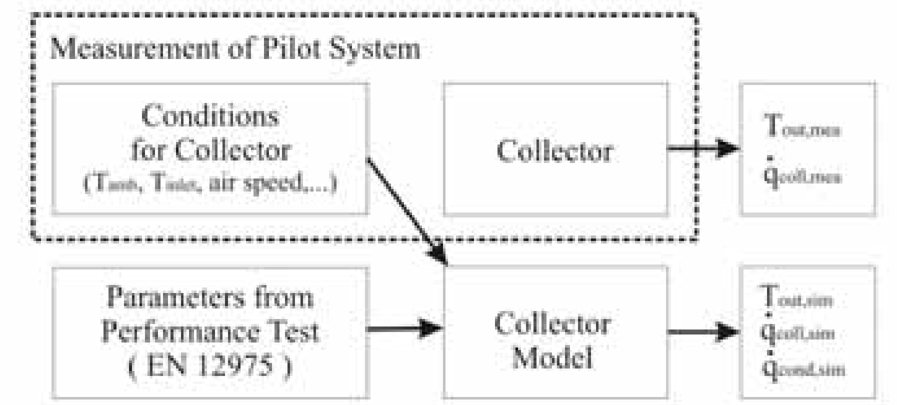

Fig. 2: Schematic procedure to determine the condensation gains in the monitored system from the measured data. The comparison of outlet temperature Tout and collector yield $\dot{q}_{coll}$ indicates the accordance of simulation. The yield through condensation is determined within the simulation model.

The results of the method described are displayed in Figure 3 for the measured pilot system. For the collector a yield $\dot{q}_{coll,mea}$ of 557 kWh/m² is measured. The simulated yield $\dot{q}_{coll,sim}$ for the total period is 7.5% lower and differs up to 20% for the monthly yields, especially in summer. The reasons for these deviations are assumed to result mainly from the measurement of air speed and ambient temperature. Although they have been measured both in the collector plane, the local sensors, which have been located at the side and not in the centre of the collector field, are not representing the real mean values for a 44 m² collector field for all weather conditions. Correspondingly, differences for the ambient air temperature and air speed occur between the measured and the surface relevant values. This heat transfer effect of the flowing air is influenced by the direction, speed and turbulence of the ambient air. Furthermore, the real ambient air temperature over a large collector area is higher than the measured air temperature at one side of the collector field. Therefore, model deviations can not be eliminated and separated from this general measurement problem.

As already discussed, the collector and condensation model has been validated separately [3] on a test rig for exactly the same collector type used in the system discussed. Accordingly, it is concluded that the simulations will reveal the fraction of condensation heat flow in good accordance with reality.

Figure 3 (left) shows the typical behaviour of an unglazed collector in a heat pump system with solar regeneration of a borehole heat exchanger. During winter low collector yields appear, while in summer high collector yields are obtained, from which the major part is used to regenerate the borehole heat exchanger. The influence on the system of the additional heat is mainly caused by the improved system stability with a regenerated borehole heat exchanger. The achieved improvements for the annual performance factor (APF) of the heat pump are about 0.2 to 0.4 with a tendency to further improvement for borehole heat exchanger fields or boreholes that are designed too small [4].

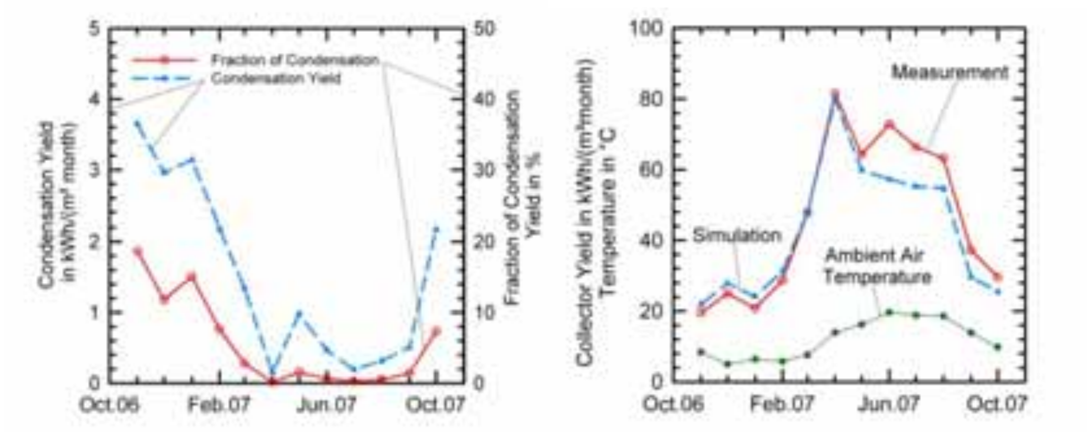

Fig. 3: Measurement and simulation results for the pilot system (left) and determination of collector yield from condensation (right), monthly values from Nov. 2006 to Oct. 2007

Figure 3 (right) shows the collector yield by condensation and its fraction of the total collector yield. As to be expected from the total collector yield the results reveal a very non-uniform distribution of condensation in the course of the year, which is discussed in terms of average values for three characteristic periods: summer (May to Aug.), winter (Nov. to Feb.) and autumn/spring. In summer, the collector yield reaches its maximum, with an average monthly value of 57 kWh/(m² month), while the condensation gains are negligible with 0.8%. The difference to operation in spring (March to April) and autumn (Sept. to Oct.) is small. Apart from a small decrease in the average monthly yield to 47 kWh/(m² month) the condensation yield of 2.1% stays insignificant. In the winter months from November to February condensation is determined to a significant fraction of 13% of the total collector yield of 26 kWh/(m² month). The daily collector yield on some selected days shows a condensation fraction of above 30%. It is to conclude that in winter the collector yield is significantly influenced by condensation, while in summer the condensation effect on the collector is negligible. But, the absolute collector yield in winter during the heating season is small compared to the collector yield in summer.

In summary, a yearly condensation yield of 3.7% or 19 kWh/m²a results. So, even in the comparably mild and humid winter of 2006 to 2007 with an average ambient air temperature of 6.4°C, the condensation gains are only a small fraction of the total yield. However, this conclusion is only valid for heat pump systems with unglazed collector and borehole heat exchanger under similar conditions. Thus, changing the climate or system may lead to higher fractions of the condensation gains. For example a system with ambient air instead of the ground as heat pump source will result in lower collector temperatures during cold winter days and therefore more condensation heat gains. Here, even enthalpy gains through the phase change to frost are to be expected. However, as the measurement period has been a very mild and humid winter in Germany, the authors do not expect a significantly higher value for this system combination in central European climate than the measured condensation fraction of 13 % in winter. On the other hand, in a solar assisted heating system, these winter gains are of higher importance. Therefore, this should be verified in further investigations.

Simulation

System simulations are conducted using TRNSYS including the new collector model to investigate the impact of condensation on electricity savings for the heat pump. The results are discussed by comparison of the annual performance factor (APF) for identical simulations with and without respect to condensation. Hence, this allows the quantification of the condensation impact on the APF for different system. Auxiliary energy consumers like pumps are not included in the APF numbers discussed below.

The TRNSYS simulations are conducted with boundary conditions according to Table 1. For long wave radiation, humidity, and wind speed the values of the test reference years have been used.

Table 1: Boundary conditions for the TRNSYS system

Parameter	**Value**
Space heat demand	60 kWh/m^{2}a
Radiator heating	40°C
Domestic hot water, 4 persons	170 l/d at 45°C
Heat pump	7.5 kW thermal
Total heat demand	12 MWh/a
Collector tilt angle	45°, azimuth 0° (south)
Weather region	TRY 2004 (Kassel, central
Heat conductance of the soil	2 W/mK

Figure 4 displays the simulation results in the 20th year of operation, where the APF is displayed for two different collector areas as a function of borehole length. Smaller borehole heat exchangers and collector arrays lead to lower system temperatures and therefore to higher specific collector yields and higher fractions of condensation. For 10 m² collector and borehole heat exchangers of 70 m and 30 m 2.6% of 12 kWh/m² of condensation gains occur. In summary, the higher the system efficiency and therefore the operating temperature level of the system is, the less the condensation effect plays a role.

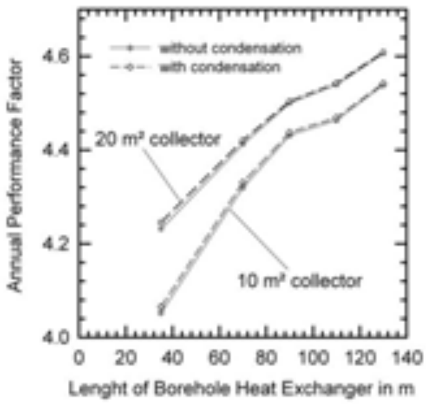

Fig. 4: Simulation results for the APF in a heat pump system with unglazed solar collector with and without respect to condensation effects.

Apart from these general correlations, the simulation confirms the marginal condensation influence in a heat pump system with unglazed solar thermal collector. Even with undersized borehole heat exchangers, and accordingly lower operating temperatures, no significant difference in the APF occurs. The maximum difference in the APF is 0.015 or correspondingly about 0.35% of the electricity consumption.

Conclusions

Our investigations, both measurements and simulations, at a solar assisted ground coupled heat pump system in central Germany demonstrate that this system is barely influenced by condensation heat gains. The improvement of the annual electrical heat pump performance factor (APF) of 0.015, or the low condensation fraction of the collector yield of 3.7% are very small figures. These improvements suggest for other systems, that the shift in the operation temperature through the additional condensation heat gain is small. Correspondingly, if condensation is respected, simulation results will not change substantially if compared to the characterization of unglazed solar thermal collectors with the EN 12975 model.

However, the results obtained are so far restricted to the system concept and configuration presented, located in central European climate. It has been shown that condensation significantly affects the absolute collector yield in the winter and even more strongly on particular days. Here, during the heating season, the condensation reaches its maximum. Therefore, condensation effects reduce the thermal load on the borehole heat exchanger. Further, the influence of condensation on the systems performance in an annual consideration might increase for warmer climates, different system configurations, collectors, or control strategies.

Finally, the impact of condensation is not restricted to the performance improvement. Other aspects of condensation as humidity problems in façade integrated collectors, corrosion, fouling, or the formation of ice possibly rise into focus with an increased dissemination of heat pump systems supported by unglazed collectors.

Acknowledgements

The results presented are based on the work of the project “UMSys” (October 2004 to June 2008) funded by Deutsche Bundesstiftung Umwelt (FN 21098). The authors are grateful for their support. The authors alone are responsible for the content of this publication.

References

[1] B. Perers, A Dynamic Collector Model for Simulation of the operation below the dewpoint in Heat Pump Systems, Eurosun (2006), Paper 040_ES06-T01-0405

[2] R Pitz-Paal, Diploma thesis (1988), Kondensation an unabgedeckten Sonnenkollektoren, Ludwig-Maximilians-Universität München

[3] W. Eisenmann, O. Müller, F. Pujiula, Metal Roofs as Unglazed Solar Collectors, Coupled with Heat Pump and Ground Storage: Gains from Condensation, Eurosun (2006), Paper ID 008_ES06-T01-0256

[4] E. Bertram, J. Glembin, J. Scheuren, G. Rockendorf, G. Zienterra, Unglazed Solar Collectors in Heat Pump Systems: Measurement, Simulation and Dimensioning, Eurosun (2008), Paper ID 235

[5] DIN EN 12975-2, Thermal solar system and components- Solar collectors- Part 2: Test Methods