EXPERIMENTAL INVESTIGATIONS OF SERPENTINE-FLOW FLAT-PLATE COLLECTORS

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Abstract – Three different serpentine-flow flat-plate solar collectors were investigated experimentally. Beyond testing collector efficiencies, temperature distributions on the absorbers were measured with a high spatial resolution in order to allow a comparison with theoretical investigations from the literature. The two collectors called F and L have the same geometry. The tube is soldered to the absorber plate all through the serpines in collector F, whereas in collector L the bends of the tubing are not thermally connected with the absorber. F and L were measured outdoor under identical meteorological conditions. The efficiency of collector F is about 2 to 2.5 percentage points superior in the range of $\Delta T / G_T$ that was covered in the experiments. Another collector with a different geometry of the serpentine-shaped tubing was measured outdoor and indoor. A comparison of the measured absorber temperature distributions with a theoretical model published in 1989 shows a satisfactory agreement between theory and experiment for the central regions of the absorber, while in the outer regions significant differences occur.

1 INTRODUCTION

The new vacuum deposition technologies allow coil coating of solar absorbers with a maximum width of approximately 1,20 m. This makes it attractive to use a single absorber plate rather than several fins in parallel when constructing a flat plate collector. This investigation focuses on collectors with single-plate absorbers and a serpentine-shaped tubing.

When developing detailed theoretical collector models, one needs to know the temperature distribution on the absorber in order to determine the mean absorber temperature $T_{\text{mean}}$. Knowledge of $T_{\text{mean}}$ is necessary for the determination of the collector efficiency factor $F'$ and of several important heat transfer coefficients.

For the common fin-and-tube geometry, $T_{\text{mean}}$ can be calculated analytically when making some approximations which are usually justified for the practical case (see for example (Duffie and Beckman, 1991, pp. 268-282)). A serpentine-flow absorber has a far more complicated geometry. Consequently, coarser approximations need to be made in order to achieve analytical solutions for the absorber and fluid temperatures, cf. (Lund, 1989).

There are only a few publications that report on experimental results of serpentine-flow collectors. Especially, the authors do not know any publication which reports on detailed measurements of the temperature distribution on a serpentine-flow absorber.

We carried out extensive outdoor and indoor experiments with three serpentine-flow collectors. The measurements included a comparison of different techniques of connecting the tube with the absorber (tube bends soldered vs. not connected).

The temperature distribution on the absorbers was measured with a high spatial resolution. In sections 3 and 4, the measured temperature field will be compared with the predictions by Lund (Lund, 1989).

2 EXISTING THEORETICAL MODELS

Several papers were published between 1976 and 1988 which used matrix methods and partly numerical solutions to treat the differential equations governing the heat transfer in a serpentine-tube absorber (e.g. Abdel-Khalik, 1976, Zhang and Lavan, 1985, Akgün, 1988). In a more generalized approach, Lund (Lund, 1989) found an analytic solution for the absorber and fluid temperature distributions which will be summarized below.

The serpentine absorber is divided into n panels (regions where the tubes run parallel, cf. fig. 1a) and the border sections with the tube turns. The differential equations for the temperatures of absorber and fluid are solved analytically for one panel, and the whole absorber is considered as a series connection of n panels. The following approximations are made in order to gain an analytical solution: The collector efficiency curve is regarded as a linear function, heat conduction in x-direction (cf. fig. 1a) is neglected, and the boundaries of the panel ($x = 0$ and $x = B_p$) are considered as adiabatic. The first and last panel in the collector are not treated separately, although they geometrically differ from the rest.

The rise of the fluid temperature along the serpines is described by using the effectiveness - NTU method. The effectiveness $E_i$ of the straight part of tubing at the right border of panel i (cf. fig. 1a) is defined as the ratio of the temperature rise
in this part of the tube and the maximum possible temperature rise over this distance:

\[ E_p \left( x = B_p, y = -b \right) = \frac{T_F \left( x = 0 \right) - T_m}{T_m - T_F \left( x = 0 \right)}, \]

where \( T_m \) is the maximum absorber temperature, given by

\[ T_m = \frac{S A_{ap}}{U_L} + T_a. \]

In the same way, the effectiveness of the tube turn \( E_t \) is defined:

\[ E_t \left( x = B_p, y = -b \right) = \frac{T_F \left( x = B_p, y = +b \right) - T_F \left( x = 0 \right)}{T_m - T_F \left( x = 0 \right)}. \]

The effectiveness \( E_C \) of the whole collector is defined as

\[ E_C = \frac{T_{F,\text{out}} - T_{F,\text{in}}}{T_m - T_{F,\text{in}}}. \]

A relationship between \( E_{m}, E_t, \) and \( E_C \) is given:

\[ E_C = 1 - \left( 1 - E_t \right) \left( 1 - E_t \right)^a. \]

Another relation between \( E_{m} \) and \( E_t \) is

\[ E_p = 1 - \frac{1 + \sigma}{1 + \sigma} e^{-\rho(1 - e^{-2\sigma B})} E_t / 2. \]

where \( \sigma, \omega \) and \( \rho \) are abbreviations for expressions \(^1\) containing \( b, B, U_L, k_e, \delta \) and \( m c_F \) (see below).

\(^1\) For the sake of brevity, these complicated expressions - which are of no interest in the following - are not given here. For details, see (Lund, 1989).

\[ \delta, \text{the distance } 2b \text{ of the tubes, the widths } B \text{ and } B_p, \text{ the overall heat loss coefficient } U_L, \text{ the heat transfer coefficient } k_e \text{ which describes the heat transfer from the absorber base (point A in fig. 1b) to the fluid, the thermal effectiveness of the tube turn } E_t \text{ and the thermal conductivity } \lambda \text{ of the absorber material.} \]

### 3 OUTDOOR EXPERIMENTS

The technique of connecting the absorber plate with the fluid duct influences the collector efficiency factor \( F' \) and hence the thermal output of the collector. An experimental comparison of different connection techniques (e.g. clamping, soldering, ultrasound welding) for absorber stripes was carried out by Rockendorf, Falk and Wetzel (Rockendorf, Falk and Wetzel, 1996).

When regarding collectors with a serpentine-shaped tubing, the differences between different connection techniques may even be higher, because with some techniques (e.g. ultrasound welding), the tube bends usually cannot be connected with the absorber. Therefore, one objective of the experiments was to investigate this topic.

Moreover, the temperature field of the absorber was to be measured in order to allow a comparison with Lund’s theory.

#### 3.1 Experimental Setup

Two collectors, which are here named F and L, were built using cases of the commercially available VitoSol 100 type \( (A_{ap} = 1.61 m^2, \text{ Viessmann GmbH, Allendorf}) \). The single-plate absorbers \( (\delta = 0.2 \text{ mm}) \) have a highly selective TiNOX coating.

The collectors have serpentine tubes (outer / inner diameter 10 / 8 mm) of identical shape, soldered to the absorber plate (for further measures, cf. fig. 2). The only difference is that in the collector F (Fixed), the tube is soldered to the absorber plate all through the serpentining, whereas in the collector L (Loose) the bends of the tube are separated from the absorber plate by a thin wooden insulation of 1mm.

The collectors were mounted on a rack which was tracked around a vertical axis during the experiments. The collector tilt angle was 35°. In an open circuit, the collectors were connected in parallel. Hence, the collectors were compared under identical meteorological conditions and with the same fluid inlet temperature.

The volume flows through the two collectors were measured with magnetic-inductive flow meters. The inlet and outlet temperatures were measured with RTD’s (Pt100, four wire connection). The global and diffuse irradiance were measured in the collector plane (two CM11 pyranometers, shadow ring in a vertical position). The longwave radiation was measured with a CG1 pyrgeometer. The temperature distributions on the absorbers were measured with 81 thermocouples which were glued onto the backsides of the absorbers. Most of the sensors were on one panel of each collector in order to achieve a high...
The positions of the most important temperature sensors are shown in fig. 2 (lines (1) to (4)).

The experiments were carried out in Marburg (latitude 51 °) in summer 1999. The data were registered simultaneously by a data logger in an interval of 15 s and displayed on-line on a monitor. This made it easier to reach and detect stationary states.

3.2 Results
Over the whole measured range, the efficiency of collector F is about 2 to 2.5 percentage points higher than the efficiency of collector L (cf. fig. 3). Regarding the geometry in question (ratio length : width = 10:3), the thermal contact between tube bends and absorber hence clearly improves the collector efficiency, even though the distance between the bends and the edge of the absorber is quite small (33 mm, cf. fig. 2).

3.3 Comparison with theoretical predictions
The above mentioned quantities \( A_{ap}, \delta, 2b, B, B_p, T_a, \lambda, S, T_{F,0}, U_c, k_c \) and \( E_c \) must be known to allow the calculation of the theoretical temperature field on the absorber according to (Lund, 1989). The measures \( A_{ap}, \delta, 2b, B, \) and \( B_p \) are easy to determine, as well as the measured quantity \( T_c, \lambda \) can be found out from the producer of the copper sheets (although with a little uncertainty). The other quantities are far more complicated to determine.

For the determination of \( S \), the transmittance-absorptance product \( (\tau_0)_{a,b}(\theta) \) for beam radiation must be known for all incidence angles \( \theta \). \( \tau_0(\theta) \) was calculated from \( \tau_0(\theta=0) \) (which was given by the producer) using the Fresnel formulas. \( \alpha_a(\theta) \) was obtained from TiNOX. The incoming radiation is treated according to the Hay and Davies model (cf. Duffie and Beckman, 1991, pp. 96-97). The effective incidence angles \( \theta_{eff,d} \) for diffuse radiation and \( \theta_{eff,g} \) for ground reflected radiation were obtained by integrating numerically \( (\tau_0) \), over the appropriate range of incidence angles:

\[
(\tau_0)_{d,g} = \int \left[ (\tau_0)_g(\theta) \cos \theta \sin \theta \ d\theta \ d\phi \right]. 
\]

\( \theta_{eff,d} \) and \( \theta_{eff,g} \) are obtained from \( (\tau_0)_{d,g} \) by solving the equation \( (\tau_0)_{d,g} = (\tau_0)_g(\theta_{eff,d/g}) \) for \( \theta_{eff,d/g} \). - For a more detailed investigation on calculating incidence angle modifiers see (Uecker, Krause, Vajen and Ackermann, 2000).

\( T_{F,0} \) is calculated from the measured fluid temperature at line 4 (cf. fig. 2).

\( U_c \) has to be calculated from the collector efficiency curve. The collector efficiency curves measured outdoor could not be fitted with a quadratic function, but with a linear function the slope of which is \( F'U_c \). As a good approximation, \( F' \) can be determined from the intercept \( \eta_0 \) of the same curve:
\[ h_0 = F' (\tau_0(\theta=0)), \] where \((\tau_0(\theta=0)\) is known (see above), and \(h_0\) is a result of the linear fit.

\(k\) was measured at line 4 of collector F and was regarded to be the same in collector L.

\(E_t\) was determined indirectly: \(E_c\) (eq. (4)) was measured. Hence the equations (5) and (6) contain only two unknowns, namely \(E_p\) and \(E_t\), that can both be determined.

For collector L, \(E_t\) was defined as zero.

For the stationary states from the collector efficiency measurements, the temperatures along the lines 1 to 4 on the absorbers F and L (cf. fig. 2) were evaluated. The theoretical temperature distribution according to (Lund, 1989) was calculated in the above explained way. As an example, fig. 4 shows for both absorbers the difference between measured and calculated temperatures for a stationary state with 

\[ G_T = 1040 \text{ W/m}^2, \ T_{F,in} = 18.2°C, \ \frac{T_F - T_a}{G_T} = -0.004 \text{ m}^2 K/W, \]

\(\dot{m} = 51 \text{ kg/m}^2 \text{ h}.\) The clearest deviations between theory and experiment occur at the boundaries of the investigated panels (lines 1 and 4). On absorber F, the measured temperatures at the boundary with a bend (line 4) are lower than the calculated, i.e. the bend cools this end of the panel. At the panel boundaries without cooling bend (absorber F: line 1, absorber L: lines 1 and 4) the measured temperatures clearly exceed the theoretical values. This means that in these regions there is a heat flux from the border sections into the panel. This effect is stronger in absorber L because here the border sections of the absorber are - due to the missing thermal contact to the tubing - generally hotter than in absorber F.

In the central regions of the panels (lines 2, 3 and 2/3) the measured temperatures are slightly higher than theoretically predicted. The differences are in the order of magnitude of 1 to 2 K.

In autumn 1999, the weather allowed only very few outdoor measurements with collector H. Therefore, detailed experiments with this collector were carried out at the sun simulator SUSI 1 at the Institut für Solarenergieforschung Hameln-Emmerthal (ISFH). The collector slope was 45°. Two collector efficiency curves \((G_T = 821\) and 606 W/m²) were determined and fitted with quadratic functions, i.e. the overall heat loss coefficient \(U_L\) is dependent on the temperature:

\[ F'U_L = F'U_{L1} + F'U_{L2}(T_F - T_a). \] (8)
For the comparison of experiment and Lund's theory, $S$, $T_{F,th}$, $U_L$, $k_e$ and $E_t$ again had to be determined.

The angular distribution of the incoming radiation $G$ from the sun simulator SUSI 1 was measured by the ISFH. For common flat-plate collectors, the artificial radiation can be treated as if it was beam radiation coming from an equivalent incidence angle $\theta_{eq}$ of approximately 23 °(Sillmann, 2000).

With $\theta_{eq}$ and the given function for $(\alpha_0)$, (8) it was easy to determine $S$.

$T_{F,bo}$ was measured directly (cf. fig. 5, fluid temperature sensor at line 1).

$U_L$ was determined in the way described above, taking into consideration that $U_L$ had to be calculated individually for each stationary state, due to its dependence on temperature (cf. eq. (8)).

$k_e$ was measured at line 2 (cf. fig. 5).

$E_t$ was calculated the same way as for the collectors F and L. (The direct measurement of $E_t$ at line 4 (cf. fig. 5) is not very reliable, since the temperature rise is quite small.)

Again, the differences between measured and theoretically predicted temperatures were calculated. Fig. 6 shows the results for a stationary state with $G_T = 822$ W/m², $T_{F,in} = 17.3$ °C, $\frac{T_F - T_{in}}{G_T} = -0.0013 \, m^2 K/W$, $m = 48.9 \, kg/m^2 \cdot h$. The curves are in principal similar to those of collector F: again the measured temperatures at the panel boundary with a bend (line 4) are lower than calculated. Here the cooling effect of the bend is even stronger than in collector F, as the maximum difference is more than -6 K.

The measured temperatures on line 1 (without bend) are again higher than the theoretical values, but the effect is less pronounced than in collector F, and the shape of the curve is a bit different.

In the central regions of the panels (lines 2 and 3) the measured temperatures are lower than theoretically predicted. This is a clear difference compared to collector F.

![Fig. 6: Differences between measured ($T_{exp}$) and theoretically predicted temperatures ($T_{theor}$, according to (Lund, 1989)) along the lines (1) to (4) of the absorber of collector H for one stationary state](image-url)
The comparison of the measured absorber temperature distributions with the theoretical predictions shows that Lund's approximation of adiabatic panel boundaries is not fulfilled. On the other hand, when collectors with thermally connected bends are regarded, the errors at the two ends of each panel tend to compensate each other, so that the mean panel temperature might be approximately correct.

A more important shortcoming of Lund's model is that it is incomplete, because it is unable to make predictions on the thermal output of a serpentine-flow collector without feeding it with quantities that can only be obtained experimentally (especially \( U_L \) and \( E_L \)). Moreover, these quantities are difficult to determine even experimentally: For the calculation of \( U_L \) from the collector efficiency curve, the collector efficiency factor \( F' \) must be known as well, because from collector tests only the product \( F'U_L \) is determined. So one needs to know \( F' \) in advance, although it is an important quantity to be predicted by the collector model.\(^2\) (Strictly speaking, Lund's model predicts the value of the heat removal factor \( F_R \) instead of \( F' \)).

Measuring \( E_L \) directly is difficult due to the small temperature rise in a tube bend, and calculating it from \( E_{cL} \) as explained above results in considerable uncertainties because of the propagation of measurement errors.

Nevertheless, Lund's paper appears to be an important step on the way to a quantitative and appropriate treatment of serpentine-flow collectors. Probably it can be developed further with good success, e.g. by finding equations that express \( E_L \) as a function of the collector's geometry.

6 CONCLUSIONS

Three different serpentine-flow flat-plate collectors were investigated experimentally. Temperature distributions on the absorbers were measured with a high spatial resolution.

The two collectors \( F \) and \( L \) have the same geometry. They were measured outdoor under identical meteorological conditions. The tube is soldered to the absorber plate all through the serpentines in collector \( F \), whereas in collector \( L \) the bends of the tubing are not thermally connected with the absorber. The difference in collector efficiency is about 2 to 2.5 percentage points in the range of \( \Delta T / G_T \) that was covered in the experiments (-0.004 to 0.06 m²K/W).

Another collector with a different geometry, named \( H \), was measured outdoor and indoor.

A comparison of the measured absorber temperature distributions with the theoretical predictions by (Lund, 1989) shows that Lund's approximation of adiabatic panel boundaries is not fulfilled, whereas in the central regions of the panel, the agreement between theory and experiment is satisfactory.

An important shortcoming of Lund's model is that it is incomplete. Some quantities needed for the theoretical calculation can only be determined experimentally.

Therefore it appears desirable to develop improved theoretical models which describe serpentine-flow flat-plate solar collectors. One possible way could be to improve analytical models like Lund's, e.g. by analyzing the thermal behaviour of the tube bends. Moreover, detailed numerical investigations should also be taken into consideration.

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NOMENCLATURE

- \( A \) m² area
- \( B \) m absorber width
- \( B_p \) m panel width
- \( b \) m half distance of flow ducts
- \( c \) m heat capacity
- \( E \) collector capacity
- \( F \) collector / absorber \( F \) (fixed bends)
- \( F' \) collector / absorber \( F' \)
- \( F'' \) collector / absorber \( F'' \)
- \( G \) W/m² global irradiance
- \( H \) W/m² collector / absorber \( H \) (horizontal)
- \( k_e \) W/m²K heat transfer coefficient from the absorber base to the fluid (cf. fig. 1b))
- \( L \) W/m² collector / absorber \( L \) (loose bends)
- \( m \) kg/h or kg/m²h collector mass flow rate
- \( n \) number of panels
- \( S \) W/m² absorbed solar radiation
- \( T \) °C temperature
- \( T_{r,0} \) fluid temperature at the inlet of the regarded panel
- \( U_L \) W/m²K overall heat loss coefficient
- \( U_{L1} \) W/m²K part of \( U_L \) that is independent of the temperature (cf. equation (8))
- \( U_{L2} \) W/m²K² part of \( U_L \) that depends on temperature (cf. equation (8))

\(^2\) On the other hand, it has to be kept in mind that the same problem arises when modeling collectors with the common fin-and-tube geometry (cf. Duffie and Beckman, 1991, pp. 268-282).
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**REFERENCES**


