Performance assessment of solar thermally activated steel sandwich panels with mineral wool core for industrial and commercial buildings

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Abstract. Steel sandwich panels are well established, cost effective components for the construction of structural and functional envelopes in buildings. They offer not only design flexibility but also great potential for active solar energy use. This contribution investigates the performance of solar thermal active panels with mineral wool core featuring heat exchangers comprising different designs and materials by means of FEM simulations. The simulation model is validated against measurements of fabricated test specimen. Specific results are reported for a large-sized test specimen with a grey finishing (RAL 7043) where the conversion factor η₀ ranges between 0.46 and 0.56. The results imply that higher figures of 0.73 and higher are theoretically achievable with a more efficient use of the panel area and improved manufacturing.

1. Introduction

The yearly installed area of sandwich panels insulated by polyurethane (PUR) or mineral wool core amounts up to 14 million m² in Germany and up to 130 million m² in Europe (as of 2010) [1]. In addition to a wide range of architectural design options, these large areas offer great potential for active solar energy use. Building regulations in regard to energy efficiency and sustainability are gradually becoming more restrictive even for the commercial and industrial sectors. To comply with these, a classic approach is to simply increase the thickness of the insulation. An alternative approach is the active energetic use of the façade. In the case of solar thermal activation, heat exchangers may be integrated into the sandwich panel and used to harvest solar and environmental heat for space heating and cooling or supplying production processes, typically in combination with heat pumps (Figure 1). Today, the thermal use of solar energy by means of building-integrated, uncovered collectors is not a widespread application. Among the few existing ones are façade and roof elements made of steel or aluminum by the Dutch manufacturer Triple Solar [2] and by the Austrian company WAFF Fassadensysteme [3], besides several R&D projects in this field [4] [5] [6].

Against the backdrop of these works, the present R&D project “Building integrated solar steel sandwich panel with mineral wool core for industrial and commercial buildings” investigates improved solutions for steel sandwich panels, focusing on cost reduction and on the efficient integration into the energy supply system. We pursue the approach to replace PUR as the core material by mineral wool and use new designs of the heat exchanger as well as alternative solutions to copper tubes. The performance potential of different geometries is determined by using finite element method (FEM) simulations. Measurements of fabricated specimens are used to evaluate smaller and larger designs. The thus validated simulation models provide the basis for further optimization.
2. Heat transfer in solar-thermally activated cover shells
To predict the efficiency of heat exchanger designs comprising different materials and geometries, we model the heat transfer in activated sandwich panel cross sections in 2D by means of FEM simulations. We investigate constructions with round tubes, tubes with an optimized D-profile, a bead-geometry and constructions with an additional heat transfer plate (HTP), according to a well-established approach in low-temperature solar thermal applications [7] (Figure 2).

![Schematic assembly of a solar thermally activated sandwich panel with inner cover shell (a), insulation (b), tubes (c) and outer cover shell (d)](Image)

![Cover shell constructions used in the simulations: round (a), bead (b), D-profile (c) and heat transfer plate (d)](Image)

The considered pipe materials are copper, steel, polybutylene (PB) as well as composite designs (polyethylene / aluminum). The heat transfer plates are made of aluminum or steel and attached to the cover shell using a one-component adhesive based on PUR or a transfer adhesive tape based on acrylate. The evaluation criterion is the internal heat transfer coefficient \( U_{\text{int}} \) defined by

\[
U_{\text{int}} = \frac{\dot{Q}}{A_{\text{abs}}(T_{\text{abs}}-T_{\text{fi}})}
\]

\( \dot{Q} \) denotes the heat flow, \( T_{\text{abs}} \) the mean absorber temperature, \( T_{\text{f}} \) the arithmetic mean between fluid inlet temperature \( T_{\text{in}} \) and outlet temperature \( T_{\text{out}} \) and the cover shell area is denoted by \( A_{\text{abs}} \).

### 2.1 Simulation of the heat transfer
The essential technical data of the investigated configurations are summarized in Table 1. The fin width (i.e. the tube spacing) \( w_{\text{r}} \) is varied by use of a parametric study (83, 91, 100, 111, 125, 143 and 167 mm).

![Table 1. Thickness \( s \), outer diameter \( d_a \) and assumed thermal conductivity \( \lambda \) of the components.](Table)

| Component                  | Layer               | \( s \) (mm) | \( d_a \) (mm) | \( d_a \) (mm) meander | \( \lambda \) (W/(m*K)) |
|----------------------------|---------------------|--------------|----------------|------------------------|--------------------------|
| Cover sheet                | Color coating       | 0.065        | -             | -                      | 0.20                     |
|                            | Galvanized steel    | 0.8          | -             | -                      | 50                       |
|                            | Protective coating  | 0.010        | -             | -                      | 0.80                     |
| Pipe                       | Copper              | 0.35         | 6.0           | 12.0                   | 337                      |
|                            | Steel               | 1.20         | 8.0           | 14.0                   | 50                       |
|                            | Polybutylene        | 1.30         | 8.0           | 14.0                   | 0.20                     |
|                            | Composite material  | 2.20         | 10.0          | 16.0                   | 0.43                     |
| heat transfer plate        | Aluminum            | 0.50         | -             | -                      | 225                      |
|                            | Steel               | 0.32         | -             | -                      | 50                       |
| adhesive                   | Polyurethane        | 0.30         | -             | -                      | 0.35                     |
|                            | Acrylate tape       | 0.05         | -             | -                      | 0.24                     |

The heat carrier mass flow (30% glycol-water mixture) is set to \( \dot{m}_f = 30 \text{ kg/(hr*m²)} \) and the average fluid temperature is set equal to the ambient temperature \( (T_f = T_a = 25 \text{ °C}) \). These model parameters are kept constant. The model considers laminar flow conditions at Reynolds numbers between 220 and 1350, depending on the hydraulic diameter of the tubes. The heat transfer coefficient between pipe wall and fluid is calculated according to VDI Heat Atlas (constant pipe temperature) [8]. The simulation results provide \( U_{\text{int}} \) as a function of \( w_{\text{r}} \). For the meander-type design, the results for copper and PB pipes are displayed in Figure 3. The results for the harp register with a fin width of \( w_{\text{r}}=100 \text{ mm} \) are 1.6 W/(m²K) to 7.8 W/(m²K) below the respective figures of the meander-type construction.
Figure 3. Simulation results of $U_{int}$ as a function of $w$, with copper (a) or PB (b) tubing (meander).

The conversion factor $\eta_0$ (efficiency at $T_a = T_d$) depends on the optical properties of the outer coating (absorption coefficient $\alpha$), the heat loss coefficient $U_l$ and the internal heat transfer.

$$\eta_0 = \alpha \cdot F'$$  \hspace{1cm} (2)

with $F'$ being the collector efficiency factor accordingly defined by

$$F' = \frac{U_{int}}{U_{int} + U_L}$$  \hspace{1cm} (3)

We calculate the equivalent fin efficiency for a fin width of $w_f = 100$ mm considering an exemplary grey cover plate (RAL 7043) which has an absorptivity of $\alpha = 0.886$. The results comply with a heat loss coefficient for well insulated, uncovered collectors of $U_L = 10$ W/(m²K). This value is confirmed by measurements of large-sized specimens (Chapter 3). The performance of cover shells with steel tubes (57.3% to 74.1%) is comparable to that of copper tubes (+0.4% to -1.1%). On the contrary, the use of plastic (-9.6% to -25.4%) or composite materials (-7.2% to -21.2%) leads to a significant reduction of performance. Due to the reduced thermal conductivity of steel cover shells, the use of heat transfer plates is further pursued. The two adhesive solutions prove to have only a negligible impact on the results.

2.2 Experimental investigation

To validate the model, we experimentally investigate different absorber fin specimens manufactured in cooperation with the project partner Schmöle GmbH. A selection of the tested fins is listed in Table 2.

| Configuration | Pipe material   | Pipe shape   | $d_a$ (mm) | $s$ (mm) | $w_f$ (mm) | Heat transfer plate, adhesive            |
|---------------|-----------------|--------------|------------|----------|-----------|------------------------------------------|
| V1.0          | Copper          | D-profile    | 12         | 0.35     | 135       | Aluminum, acrylate tape                  |
| V2.0          | Copper          | Round        | 12         | 0.35     | 135       | Aluminum, acrylate tape                  |
| V3.0          | Copper          | D-profile    | 12         | 0.35     | 135       | Steel, acrylate tape                     |
| V4.0          | Polybutylene    | D-profile    | 12         | 1.3      | 135       | Aluminum, acrylate tape                  |
| V7.0          | Composite       | Round        | 12         | 2.2      | 135       | Aluminum, acrylate tape                  |
| V9.0          | Steel           | Round        | 12         | 1.0      | 135       | Aluminum, acrylate tape                  |
| V11.0         | Copper          | D-profile    | 12         | 0.35     | 135       | Aluminum, polyurethane                   |

The experimental determination of the internal heat transfer coefficient value employs calorimetric measurements established for solar absorbers at ISFH: an individual absorber fin is heated at the peripheral edges by means of heating bands while cold water is pumped through the absorber pipe [9]. A cross-section of the fin instrumented with temperature gauges is shown in Figure 4. We use PT100 temperature sensors mounted on the absorber at predetermined positions. During testing, the fin is heated...
by an electrical heating power $P$ corresponding to an equivalent solar irradiation of $G = 1000$ W/m$^2$. The mass flow of the water is set to 85 kg/hr, ensuring a turbulent flow regime in this case.

![Heating element and Temperature sensors](image)

**Figure 4.** Cross-section of the heated fin with positions of heating element and sensors.

Under these conditions, the heating power

$$Q = m_f \cdot c_p (T_{\text{out}} - T_{\text{in}})$$

(4)

is used to calculate $U_{\text{int}}$ from equation (1):

$$U_{\text{int}} = \frac{m_f c_p (T_{\text{out}} - T_{\text{in})}}{A_{\text{abs}} (T_{\text{abs}} - T_{\text{out}} + T_{\text{in}})}$$

(5)

The mean absorber temperature is related to the temperature field of the absorber fin and must be calculated from the integral value, divided by the specific area. We employ the FEM model for this purpose. The model is fitted against the discretely measured temperature signals and the mean absorber temperature is then calculated from the fitted function.

2.3 Model validation

The absorber fin test rig explained above is used to validate the FEM model. This model incorporates a number of parameters responsible for the heat transfer across the contact interfaces of adjacent materials, which are partly in loose contact. To mimic the contact between the tube and the HTP or cover plate a generic air layer is used. The correspondent thermal resistance is proportional to the layer thickness, which serves as a free fit parameter. This parameter helps to compensate manufacturing tolerances as well as uncertainties regarding the thermal conductivity of the used materials. The resistance lies between $1.5 \cdot 10^4$ and $44 \cdot 10^4$ $\text{m}^2\text{K} / \text{W}$, which corresponds to a resistance layer thickness in the range between 0.004 mm to 0.115 mm (mean value: 0.034 mm). For comparison, the resistance for aluminum surfaces with a roughness of 10 $\mu\text{m}$ under a contact pressure of 100 N/m$^2$ is approximately $2.75 \cdot 10^4$ $\text{m}^2\text{K} / \text{W}$, which is equivalent to an air layer with a thickness of 0.007 mm [10]. This method yields a FEM model comprising maximal local temperature misfit of below 2 K to the measured data (see Figure 5a). Based on the simulated results of the internal heat transfer coefficient, we calculate fin efficiencies $\eta_0$ between 55% (V4.0) and 73% (V1.0; V11.0) for a fin width of 135 mm (see Figure 5b).

![Simulation and measured temperatures](image)

**Figure 5.** Selection of simulated and measured temperatures (a) and calculated fin efficiencies (b).
3. Performance evaluation of large-sized test panels

We designed and fabricated large-sized sandwich panel specimens with dimensions of 1020 x 2000 mm ($A_{sh} = 2.04$ m²), a 150 mm thick mineral wool insulation and meander registers with a pipe spacing of 100 mm. The connection between the HTP and the cover shell is made by the double-sided adhesive tape based on acrylate (V1.D, V2.D and V3.D) or by the one-component PUR-adhesive (V4.D), using the same PUR-adhesive for the application of all absorbers on the sandwich modules. Table 3 lists the realized configurations.

| Configuration | Pipe material | Pipe shape | $d_s$ (mm) | $s$ (mm) | $w_r$ (mm) | Heat transfer plate |
|---------------|---------------|------------|------------|----------|------------|---------------------|
| V1.D          | Copper        | D-shaped   | 12         | 0.35     | 100        | Aluminum, acrylate tape |
| V2.D          | Copper        | D-shaped   | 12         | 0.35     | 100        | Steel, acrylate tape  |
| V3.D          | Stainless steel | D-shaped | 10         | 1.00     | 100        | Steel, acrylate tape  |
| V4.D          | Copper        | D-shaped   | 12         | 0.35     | 100        | Aluminum, polyurethane adhesive |

3.1 Experimental evaluation

We tested the manufactured specimens in regard to performance and serviceability in accordance with ISO 9806 for solar thermal collectors [11]. The investigations were carried out under laboratory conditions in a solar simulator (irradiance $G = 880$ W/m² or $G = 980$ W/m² (V4.D), mass flow (water) $m = 120$ kg/hr (turbulent) and ambient temperature $T_{amb} = 26$ °C). These indoor measurements have to be validated with running outdoor measurements, since the artificial conditions of the solar simulation cannot completely reproduce the ambient conditions. We report the following conversion factors $\eta_0$ for still air (wind speed $v$ below 1 m/s):

- V1.D: 53.3%
- V2.D: 52.3%
- V3.D: 45.6%
- V4.D: 56.3%

The temperature and wind dependent efficiency of the collector can be represented by the following equation.

$$\eta = \eta_0(1 - b_1 \cdot v) - (b_1 + b_2 \cdot v) \frac{T_H - T_a}{\dot{q}}$$  (6)

Note that this equation applies only, if the sky temperature is equal to the ambient temperature. The heat loss coefficients are comparable for all measured specimens. The linear heat loss coefficient $b_1$ (7.37 to 8.87 W/(m²K)) and the wind dependent coefficients $b_2$ (0.82 to 1.37 J/(m³K)) and $b_3$ (0.057 to 0.085 s/m) show small deviations. The large difference in the conversion factor from the results obtained with the absorber fins can be attributed to the relatively large non-activated area (30%) of the panels. It is noticeable that configuration V2.D (52.3%) has only a slightly lower efficiency than V1.D, although we expect a clearer difference due to the poorer thermal conductivity of the steel HTP. In contrast, the conversion factor of the almost identical configuration V4.D is with 56.3% significantly higher. Furthermore, the significantly lower conversion factor of V3.D compared to V2.D cannot be fully explained by the use of stainless steel tubes. Thermographic analysis of the specimens reveal a less homogeneous glued connection between the heat exchanger and the cover shell for V1.D and V3.D, which explains the suboptimal heat transfer.

3.2 Performance simulation

Based on the FEM model for the individual absorber fin, an extended model is set up to simulate the behavior of the larger test panels. It is validated against the respective experimental data. The panels are simulated using the environmental conditions from the solar simulator. The convective heat losses are modelled by a wind speed $v$ dependent function according to [12]. The radiation losses are calculated by considering the radiative heat exchange to the ambient temperature. The resulting windless losses...
(v ≈ 0.5 m/s) are approx. 10 W/(m²K), which complies with the experiments. The simulations result show, that the configurations V1.D and V4.D (Aluminum-HTP) have the highest efficiency with 58.3% and 58.4% respectively, as expected. The performances of V2.D and V3.D (steel HTP) are respectively 6.3 and 6.6 percentage points lower (52.0% and 51.7%). The simulation results thus confirm the expectations as well as the suboptimal production of the specimen V1.D and the specimen V3.D.

4. Conclusion
This paper presents the theoretical and experimental evaluation of the heat transfer in solar-thermally activated steel sandwich panels. Based on successfully validated FEM simulations we can predict the correspondent thermal performance. The results show that the use of heat transfer plates in steel sandwich elements is crucial. In addition to copper and aluminum, steel can also be used as a material for both pipes and heat transfer plates. The use of plastics for the pipes, on the other hand, is critical as expected due to their low thermal conductivity. The production process of the activated sandwich panels plays an important role with regard to the thermal performance, since it is responsible for the connection of the heat exchanger. In our tests, the conversion factor could be 10% lower if the production was not optimal. The non-activated edge of the panels has also a strong influence on the performance. Based on the achieved results, the design will be optimized with respect to material input (costs) and performance (energy gain). A meaningful design ultimately depends on the running evaluation of the panels as part of the overall heat supply system.

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