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Thermal simulation and efficiency of a hermetically sealed flat plate collector with a fully adhesive edge bond

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Abstract

This research work deals with the thermal analysis of a hermetically sealed flat plate collector with a gas-filled cavity between absorber and glazing. Published scientific work in this field is discussed and compared to own laboratory testing and simulation results. The impact of the mechanical absorber deflection for small gap sizes between absorber and glazing on the collector efficiency are investigated. A parameter study for gas-filled collectors with a fully adhesive edge bond was conducted. Four different types of functional models were built and tested to validate the simulation model. Impacts affecting the thermal efficiency such as the absorber deflection of solar collectors with a fully adhesive edge bond were analysed and evaluated. A system simulation was conducted analysing the components temperatures and the annual collector yield.

Keywords: *solar thermal; absorber; gas-filled; edge bond;*

1. Introduction

In a standard flat plate solar collector, the highest heat loss is caused via the front loss (Tabor 1958, Beikircher 2009). This loss, in turn, can be split up into a radiative and a convective part. Modern collectors are equipped with a high selective coating with absorbance values of 0.95 respectively emission values of less than 0.05. This means 95 % of the short-wave sunlight is absorbed and only 5 % of the heat radiation (long-wave) emitted. Consequently, the highest loss in a flat plate collector with a high selective coated absorber is caused by the convection between absorber and glazing (Platzer 1988, Beikircher 2009, Tabor 1958). Consequently, reducing the convective heat transfer is a promising approach to increase the efficiency.

A common and successful method to reduce the convective front heat loss is the use of transparent insulation material. In particular, a transparent component is put between absorber and the ambient retarding the convection process. The measures can be classified according to their geometrical position to the absorber (Platzer 1988, Kaltenbach 2003) – structures parallel and perpendicular to the absorber, chamber structures and quasi-homogenous structures. The exhibit for a collector with a structure parallel to the absorber is the ordinary glazed flat plate collector. Yet, there were several approaches to use two or more layers to suppress free convection. Usually, plastic foils or glasses were used as layer material. Remarkable boosts in efficiency were calculated and demonstrated (Rommel et al. 2003, Beikircher 2010, Föste 2013). Absorber parallel structures such as multi-covers are finding infrequent use in solar collectors.

Structures perpendicular to the absorber are, amongst others, honeycomb structures. A wide field of valuable publications on the convective heat transfer itself and on honeycomb structures have been published. There might be still some challenges concerning the long-term stability as well as problems during stagnation but a decent performance level was demonstrated (Hollands 1965, Tabor 1969, Buchberg et al. 1976, Kessentini et al. 2013). However, due to the demanding requirements in solar collectors concerning the long-term stability and costs, these measures are not widely spread. Chamber structures are closely linked to absorber perpendicular and parallel structures and are usually made out of multiple wall sheets. The method

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retards convection in a similar way to the other structures mentioned above but results in a deterioration of the solar transmission coefficient. Since the chambers are usually made out of plastics the negative effects of ageing and long-term stability applies here as well as on the honeycomb insulation.

Svendsen and Jensen (1987), Svendsen (1992) respectively Nordgaard and Beckman (1992) investigated the use of monolithic silica aerogel in solar collectors. Aerogel is a highly porous solid body in which more than 95 % of the volume is air. Hence, it is an outstanding insulation material. Up to date, the use of aerogel insulation in solar collectors is linked with no economic advantages, as the material costs are exorbitant.

Beside transparent insulation between absorber and glazing, the possibility of evacuating this gap was analysed as well. In theoretical studies, an efficiency of 41 % at an operating temperature of 150 °C was calculated (Eaton and Blum 1975). Benz and Beikircher (1999) concluded in a laboratory testing that the convective heat coefficient is reduced by up to 65 % in the continuum range compared to air. Buttinger et al. (2009, 2010) analysed an evacuated compound parabolic collector (CPC) and confirmed efficiencies above 50 % for working temperatures of more than 150 °C. A significant drawback of these collector types can be seen in the complex design, the high requirements concerning mechanical loads and sealing methods, which prevents a low-maintenance lifetime.

A gas-filled interspace between glazing and absorber is a promising approach in reducing the convective heat loss. Vestlund et al. (2009) analysed the thermal performance of a gas-filled solar collector at ambient pressure. In their work, a parameter analysis was carried out including the variation of the inclination angle and the gas type. Based on simulations, a considerable reduction of the overall heat loss by 20 % was shown. Despite of the discussed paths, gas-filled collectors at ambient pressure are not yet as explicitly analysed as the other discussed approaches. The charm of this measure is that there is no additional hardware needed. In return, an efficiency on the level of a well-designed collector with a honeycomb structure or even an evacuated flat plate collector needs to be proven. The aim needs to be in a collector design, which is suitable for a highly automated production with a decent performance level and a cost structure competitive to a conventional solar collector.

2. Design of a collector with an all-round supported absorber with a fully adhesive edge-bond

Within this paper, a collector with an all-round supported absorber with a fully adhesive edge bond was analysed in terms of its thermal behaviour respectively convective heat loss. The basic collector construction was intended to be simple and as similar to an insulated glazing unit as possible in order to produce the collector on a highly automated insulated glazing unit production line. A butyl sealant (primary sealing) was applied directly on the absorber sheet. Subsequently, it was bonded to the glazing to achieve a gastight cavity. During the assembly process of the absorber and the glazing, the gap was filled with an inert gas to lower the convective heat loss. Fig. 1 shows a schematic cross section of the proposed collector design.

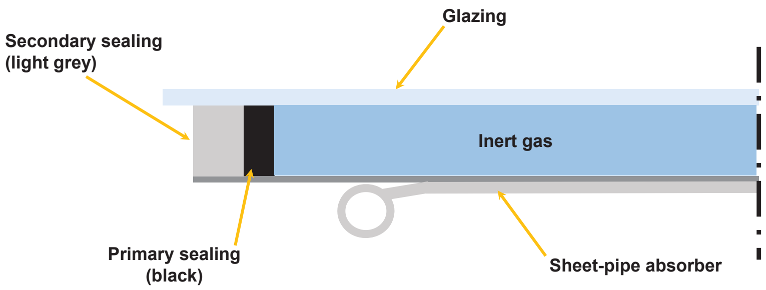

Fig. 1: Schematic cross section of the assembly group of absorber and glazing

In this work, it was favourable to achieve a short distance between absorber and glazing. On the one hand, a shorter distance between absorber and glazing reduces the adhesive costs, which cannot be ignored, as those are a considerable part of the total collector costs. On the other hand, a lower convective heat loss might be achievable by adjusting the gap size to the transition region between conduction and convection, i.e. Nusselt number is still 1 respectively very close to 1. Furthermore, it needs to be considered that less volume is enclosed between absorber and glazing and, thus, a lower mechanical load is applied on the collector

components due to the pressure change (Riess et al. 2014, Vestlund et al. 2009, 2012). Conventional flat plate collectors are designed to have a gap size between 25 and 35 mm whereas in this approach an absorber glazing distance at first of 10 mm for Argon was intended. In a later stage of the research work, gap sizes of up to 20 mm were considered and analysed as well. Even though Xenon and Krypton are allowing very short spacing distances in comparison to Argon or air (Vestlund et al. 2009), those inert gases are of no practical interest. Whilst Krypton is by a factor of 100 times more expensive than Argon, Xenon is even more costly than Krypton. An Argon filled interspace with a gap size of 15 mm for an aperture area of 2 m² causes costs of about 0.06 €. In theory, Krypton enables a shorter distance between absorber and glazing cutting the filling costs down to about 3.5 €. However, own results showed that it is not rational to achieve shorter gap sizes than 15 mm with conventional sheet-pipe absorber (Riess et al. 2014). Beyond that, it needs to be analysed whether the higher filling costs can be justified by an increased annual yield. Consequently, Argon was chosen as a filling gas in this approach whereas air served as the main reference. However, Krypton was used in some considerations as a further reference to Argon and air. Xenon was of no further interest due to its uneconomic pricing and very limited availability.
The principal theory concerning the convective heat loss in a flat plate collector is not discussed in this paper but can be found in (Eismann 2015, Hollands et al. 1976, Tabor 1959).

3. Thermal collector model

Over the decades, there have been plenty contributions in the modelling of flat plate solar collectors. In the early 1940s, Hottel and Woertz (1942) analysed the thermal performance of flat plate solar collectors and presented an analytical calculation approach of the collector efficiency. In 1958, their analytical approach was refined by Tabor (1958). In the same year, Hottel and Whillier (1958) published a linear efficiency model of the correlation between test results and analytical description. Cooper and Dunkle (1980) introduced a non-linear flat plate collector model in which the efficiency is plotted in dependence of the difference between ambient temperature and the mean fluid temperature to the solar irradiation. Klein et al. (1974) presented their investigations on the transient collector behaviour and pointed out the differences to a steady state collector model. Ultimately, the collector simulation model was refined by two heat capacities resulting in a two-node model. Matuska and Zmrhal (2008) as well as Koo (1999) published a collector simulation model that can be downloaded and used without a license fee. The mentioned models are sophisticated and established simulation approaches. A drawback of those approaches is that the component temperatures are not available at all or only very limited. In the case of a hermetically sealed flat plate collector with a fully adhesive edge bond new materials are being used. Moreover, the adhesive is by its nature sensitive to high temperature loads, i.e. temperature magnitude, period and occurrences. Against this backdrop, beside the collector efficiency it was essential to analyse the thermal loads on the components. Finally, in the mentioned freeware simulation models a gas-filled cavity cannot be simulated. Therefore, the simulation model by Reiter et al. (2015) was adapted for the conducted thermal analysis. This collector model was implemented for the development of full polymeric collectors and, hence, the temperature loads were in the focus of Reiter's research (2014). The principle equations, assumptions and simplifications used in this model can be found in Reiter et al. (2015) and, thus, are not further described. However, the convective heat transfer between absorber and glazing was modified. Reiter et al. (2014, 2015) used an approach to calculate the convective heat transfer between absorber and glazing from Matuska and Zmrhal (2009). In contrast to this, the widely used convection theory according to Hollands et al. (1976) was applied in this approach at first.
During the research program it turned out that, there is a considerable discrepancy between the applied convection theory and the laboratory testing results. The thermal influences that led to this deviation are discussed within the next chapters.
The convective heat transfer is amongst other parameters affected by the inclination angle of the collector. However, the inclination effect was not of interest within this research program. Hence, the standard inclination for testing and simulation was always set to 45°.
The inert gas properties as well as the change of the material properties such as viscosity or density depending on the present gas temperature were approximated by polynomials and can be found in the VDI Heat Atlas (Stephan 2010).

4. Thermal simulation according to the convection theory after Hollands et al. (1976)

The efficiency tests of the functional model were conducted on the institute's own indoor solar simulator according to the standard DIN EN 12975-2 (2006). During the complete research programme all parameter relating to the collector efficiency were referenced to the aperture area – unless otherwise stated.

Even though deviations between simulation results and measured efficiency curves were expected, the deviations were surprisingly high. Fig. 2 shows the simulated efficiency curve and the tested efficiency conducted on an indoor solar simulator. The collector parameters are found in Tab. 1.

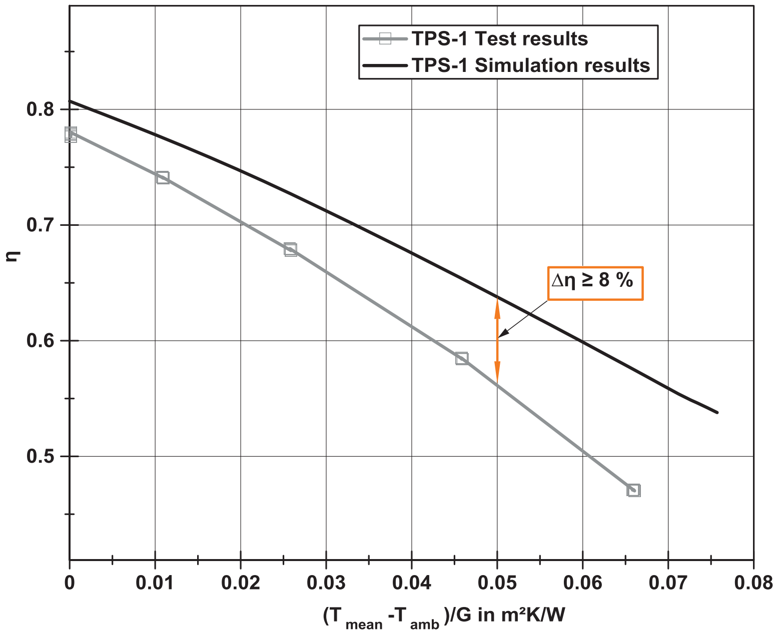

Fig. 2: Comparison between the first simulation model result of a hermetically sealed and gas-filled collector (convection theory according to Hollands et al. 1976) and the measured collector efficiency of the corresponding experimental collector

Tab. 1: Collector parameters used in both laboratory testing and simulation

Parameter	Value
Optical efficiency η_0	0.782
Linear loss coefficient a_1 in W/m²K	3.215
Quadratic loss coefficient a_2 in W/m²K^2	0.023
Aperture area in m²	1.9
Initial gap size in mm	10
Insulation thickness (back) in mm (mineral wool; λ = 0,04 W/mK)	40
Insulation thickness (side) in mm	--
Absorber type	Sheet-pipe
Absorber piping	Harp
Material absorber sheet	Aluminium
Material absorber piping	Aluminium
Gas filling	Argon

The simulated efficiency curve shows a higher performance over the complete operating range. Close to η_0 the deviation between the curves is only 2.4 %. Even though η_0 is called the optical efficiency there is also the fin efficiency F' contributing to this value and, thus, the collector heat loss is affecting this parameter as well. With rising collector working temperatures, the deviation between the two curves increases. For a typical working point of a space heating supporting solar system of $(T_{mean} - T_{ambient}) / G = 0.05$ m²K/W the deviation is more than 8 %. Reiter et al. (2015), however, validated the simulation model for a conventional collector with a double-harp absorber and a gap size of 30 mm. According to their results, the model showed a good correlation between laboratory testing and simulation results. Within this research study, the simulation model code was modified in terms of the convective heat transfer, smaller gap sizes and gas fillings. Consequently, it can be concluded that the deviation is rather an underestimation of the actual convective heat loss or a failure in the experimental setup than a bug in the model code. At first, it seemed plausible to examine the experimental setup to exclude possible mistakes. As the institute's solar simulator is regularly checked by testing the same reference collector, it was ensured that the test rig was not faulty. Therefore, the collector was further analysed. The gas concentration of the functional model was measured thrice – right after its production, two hours later and 24 h after the production. All measurements gave values well over 95 % Argon concentration. The collector test was conducted 36 h after the production. It can be, therefore, concluded that a high Argon concentration (< 95 %) was in the interspace. However, several observations were made on the absorber. The thin absorber sheet showed a considerable deflection during collector operation. Instead of the intended 10 mm gap between absorber and glazing a mean distance between the two plates of only 6 mm was measured. It turned out that the initial absorber shape has a significant impact on the absorber deflection as its shape is magnified during collector operation (Riess et al. 2014). Consequently, this means absorber areas that had already a shorter distance than 10 mm to the glazing are getting even closer during collector operation and vice versa. Those absorber deflections were hard to spot by a visual inspection through the glazing due to the dark absorber coating and the glass reflections. Instead, holes were drilled through the back of the insulated back plate to conclude on the actual distance of the absorber to the glazing. This was possible as the collector dimensions, e.g. glass thickness and total collector height, were known and measurable. This investigation led to the result that the absorber deflection has a considerable effect on the difference of the collector efficiency prediction. Fig. 3 clarifies the impact of a small deviation in the gap size on the convective heat transfer respectively the collector efficiency for collectors with a small gap size, i.e. within the transition area between heat conduction and convection.

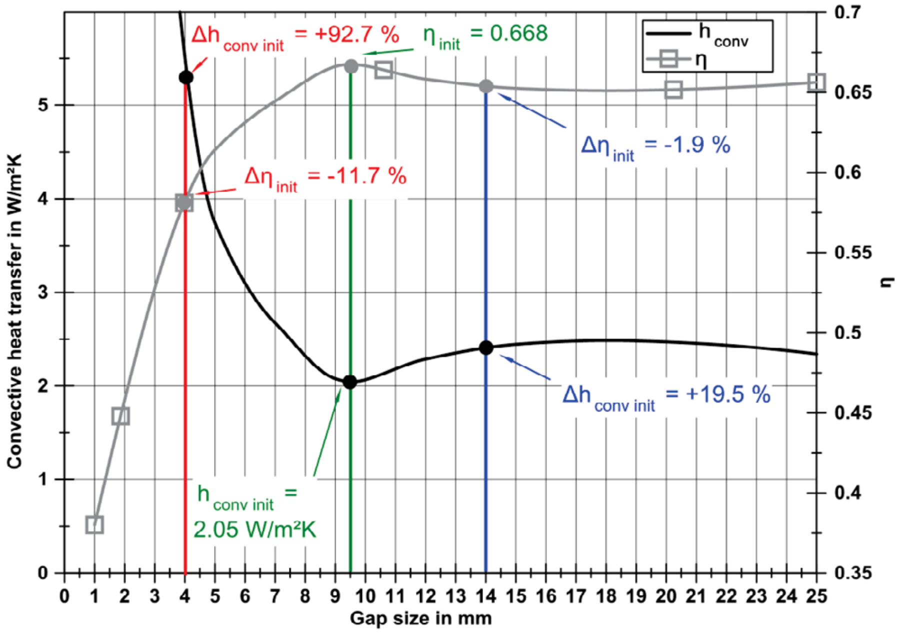

Fig. 3: Simulated change of collector efficiency depending on the gap size after the convection theory of Hollands et al. (1976) for a mean fluid temperature difference of 60 K

For a mean fluid temperature difference of 60 K and Argon as gas filling, the local minimum in the convective heat transfer is at 9.5 mm. If the gap size is decreased by only 5.5 mm the convective heat transfer is increased by almost 93 %. Ultimately, this means an efficiency loss in this typical working point of 11.7 %. In contrast to this, an increased interspace of 5.5 mm to 14 mm is followed only by a slight increase of the convective heat loss of 19.5 % which correspondents to an efficiency loss of less than 2 %. However, the interaction between absorber deflection and convective heat transfer cannot be implemented in to the applied thermal collector model. Reiter et al. (2015) showed that the simulation model was valid for conventional flat plate collector with a normal gap size; it is doubtful whether it is possible to predict the

precise convective heat loss for such short distances between absorber and glazing due to the absorber deflections. Furthermore, it is problematic if not impossible to adjust the gap between the glazing and a conventional sheet-pipe absorber to the local minimum of the convective heat transfer.
As a model validation with this experimental was not conductible another set of functional models with a wider spacing between absorber and glazing and different absorber types was produced and tested.

5. Discussion of different convection theories and model validation

Bartelsen et al. (1993) measured with an own experimental collector under various inclinations the convective heat transfer between absorber and glazing down to gap sizes of 15 mm and compared the results to the one published by Hollands. Their results showed a 25 to 60 % higher convective heat loss than the results computed according to Hollands. Föste (2013) measured the convective heat loss of a collector with a double-glazing unit. In his approach, a collector prototype was equipped with temperature sensors along the glass panes and the absorber. Subsequently, the absorber piping was perfused by hot water (> 150 °C) and the temperature difference between inlet and outlet was measured to conclude on the total collector heat loss. His results showed a deviation of the convective heat loss coefficient between the values derived by the literature and the measured results by up to 32 % between absorber and glazing unit respectively up to 15 % in the glazing interspace. Recently, Eismann (2015) published an extended correlation for the convective heat transfer in the cavity between absorber and glazing. In his study, 22 standard collectors were modelled based on the specifications provided by producers respectively test reports. As the common convection correlations were stated for isothermal plates, Eismann's focus was to extend the Hollands' equation for the convective heat loss for non-isothermal absorbers with a high selective coating. This is in particular of interest as the Hollands equation is only valid for Rayleigh numbers up to 105. The Rayleigh number, however, are by a factor of three higher in the case of coated absorber surfaces (Eismann 2015). According to the author, those facts are resulting in an underestimation of the convective heat loss in solar flat plate collectors. Eismann extended in his approach the convection equation according to Hollands by fitting it to the equation of ElSherbiny et al. (1982). This results in a better accordance for selective coated absorber surfaces.
For the author's own model validation, an experimental test was setup in which the convective heat transfer between absorber and glazing was measured. The collector 'TPS-AlCu20' was, therefore, in addition to the already applied 55 mm thick mineral wool further insulated on the back and on the side with polystyrene (t = 80 mm, λ = 0.038 W/m²K). The collector was equipped with temperature sensors to conclude on the thermal collector losses (Fig. 4).

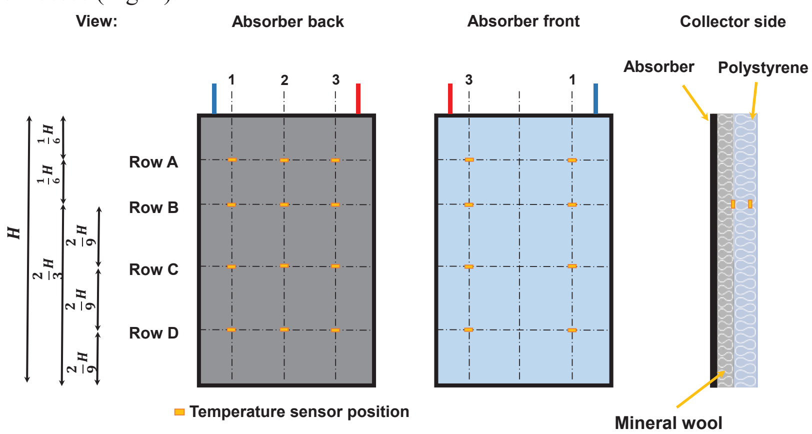

Fig. 4: Sensor positions on the experimental collector

Subsequently, the functional model was put under the solar simulator to measure the efficiency for certain collector inlet temperatures, i.e. 70, 80 and 89 °C. The testing rig is equipped with two pyranometers, a magnet-inductive flow meter and two matched PT100 sensors to log the collector inlet and outlet temperature. Hence, the useful collector energy respectively collector loss can be calculated. The conduction loss via the back insulation and the radiative absorber loss is calculated based on the collector temperature measurements, the ambient temperature and the known physical collector properties. The sidewall losses are neglected, as the collector side surface is small compared to the collector's aperture area. Beyond that, the collector sides are very well insulated and, thus, considered as adiabatic. The collector model was enhanced by the convection equation according to Eismann (2015):

$$Nu = 1 + 1.44 * \left(1 - \frac{1708}{Ra\emptyset + 1708R_c}\right) * \left(1 - (\sin(1.8\emptyset))^{1.6} \frac{1708}{Ra\emptyset + 1708R_c}\right) + ((\frac{Ra\emptyset + 5830R_c}{5830})^{0.39} - 1))(1 + C * R_c)$$

Eismann (2015) suggests a C-value of 0.29 with a mean deviation of ±0.17. For the here conducted calculation C was set to 0.46 whereas R_c was 1. Subsequently, the author's own convection loss measurements were compared to the calculation results according to the equations of Hollands et al. (1976) and Eismann (2015) in Fig. 5.

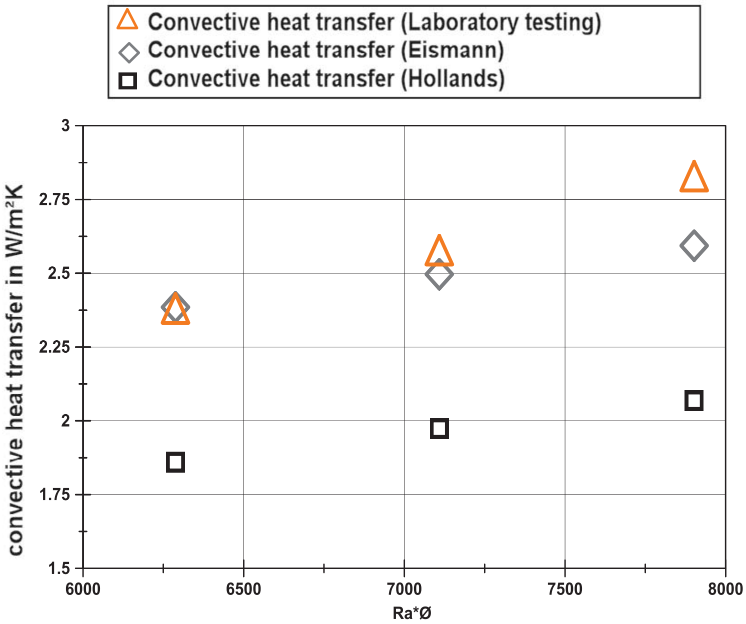

Fig. 5: Comparison of the calculated heat transfer according to Eismann and Hollands et al. and the own test results

The own results differ clearly from Hollands' approach. However, the measured deviations are in a similar range with the results obtained by Bartelsen et al. (1993) and Föste (2013). Beyond that, the results for the convective heat loss are in good accordance with the values from Eismann. The measured points with an inlet temperature of 70 °C and 80 °C are differing only -0.9 % respectively 2.5 % from the values derived by Eismann's approach. Yet, the value for the point with an inlet temperature of 89 °C differs by 7.8 %. Based on this experimental setup the used collector model was enhanced by the extended convection equation. Fig. 6 shows the tested results and the simulated efficiency curve according to Eismann's respectively Hollands' approach of the collector TPS-AlCu20 (Tab. 2).

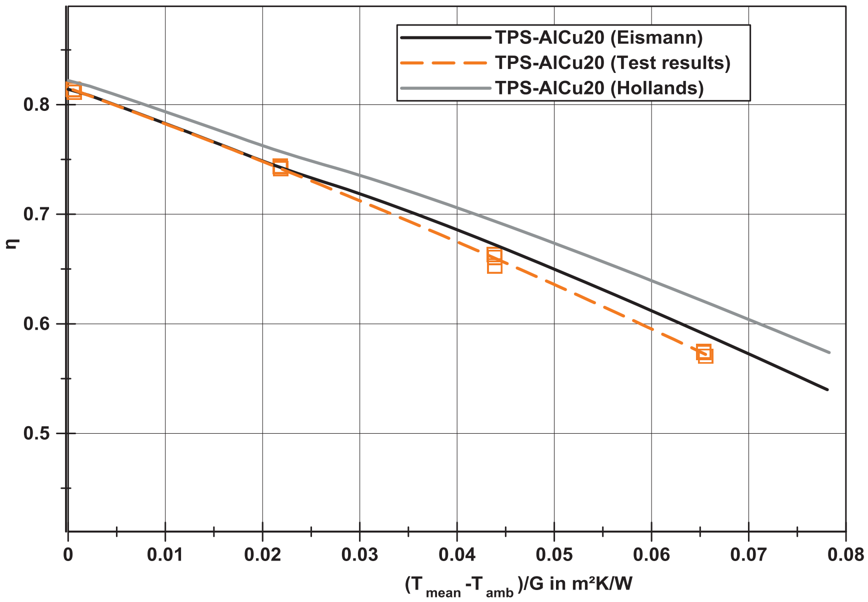

Fig. 6: Model validation by comparison of the measured and simulated collector efficiency with a calculation approach according to Eismann (2015) respectively Hollands et al. (1976)

Tab. 2: Used collector parameters in simulation

Parameter	**Value**
Aperture area in m²	1.84
Absorber type	Sheet-pipe
Absorber piping	Double-harp
Material absorber sheet	Aluminium (Al)
Material absorber piping	Copper (Cu)
Absorption absorber	0.95
Emission absorber	0.05
Transmission glazing	0.94
Emission glazing	0.94
Thermal conductivity Aluminium in W/m²K	235.0
Thermal conductivity Copper in W/m²K	390.0
Thermal conductivity mineral wool in W/m²K	0.035
Thermal conductivity glass in W/m²K	0.84
Glass thickness in mm	3.2
Thickness absorber sheet in mm	0.5
Fin width in mm	99
Outer diameter riser in mm	7
Absorber length in mm	1,857
Absorber width in mm	990
Insulation thickness in mm	50
Gap size in mm (unless stated otherwise)	20
Gas filling	Argon / Krypton

For the typical operation point of 0.05 m²K/W the deviation of the simulated values are 7.5 % (Hollands) respectively only 3 % (Eismann). The deviation of the complete temperature difference between the measured values and the simulated values is shown in Fig. 7.

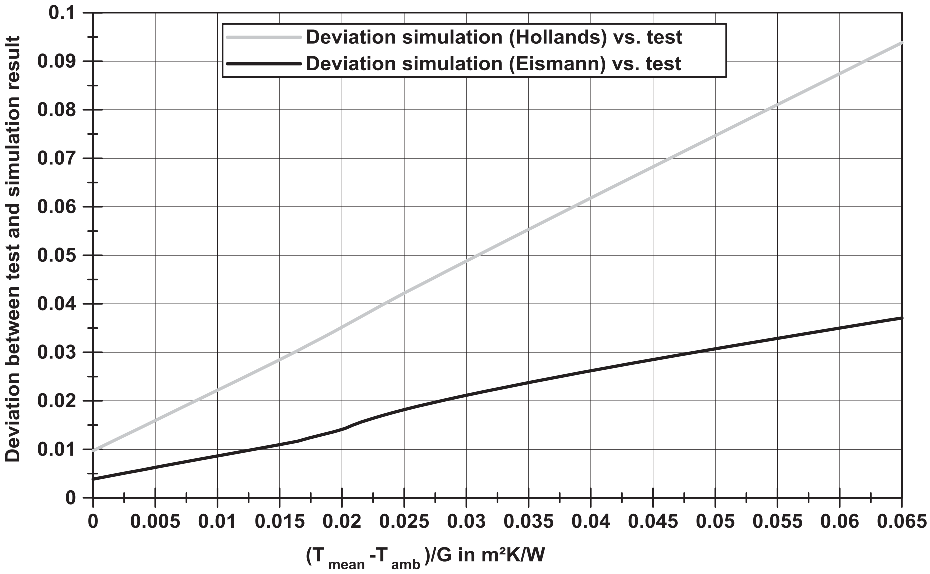

Fig. 7: Deviation between simulation and measurement results

The smallest deviation for the Hollands' approach is about 1 % in the region of the optical efficiency. Again, this is owed to the impact of the fin efficiency factor F' that affects this value. As the total collector losses are included in F' and the convection losses are slightly different, a rather small deviation exists. However, with increasing operation temperatures the deviation gets more obvious. For an ordinate value of 0.065 m²K/W the difference is almost 9.5 %. The extended correlation after Eismann shows only a deviation of 0.5 % for values close to η_0 and 3.5 % at max for higher operating temperatures. The deviation with increasing temperature ranges were also observed in the own conducted convective heat loss measurement. As the mean collector gap size varies, the convective losses differ as well. Ultimately, this behaviour is not included in an equation.

Beside the satisfying results of the efficiency comparison, the measured component temperatures showed as well a good correlation to the simulated results. In this second test setup, the functional model TPS-AlCu20 without the additional polystyrene insulation was equipped with the temperature sensors and put under the solar simulator. The collector's absorber and glazing temperatures were measured for three different inlet temperatures (70, 80 and 89 °C). During this steady-state test, the average insolation was 881.7 W/m² with a mean wind speed across the glazing of 2.5 m/s and an ambient temperature of 24.9 °C. In addition to the varying inlet temperatures, a further measurement series in dry stagnation (I = 879.8 W/m²; v_{wind} = 2.5 m/s) was conducted. Finally, the measured temperatures for absorber and glazing were averaged and compared with the simulation results. The deviation between the stagnation test and the simulated values were between -1.7 up to +0.1 % for the absorber respectively -1.7 to -0.2 % for the glazing. As for the good correlation between test and simulation results, the model is considered to be working properly and, thus, validated.

6. Efficiency analysis, thermal loads and annual yield of a gas-filled collector

Derived by the parameter study above, a gap size of 20 mm for an efficiency comparison between gas-filled collectors (Krypton, Argon) and an identical but vented collector was conducted (Fig. 8).

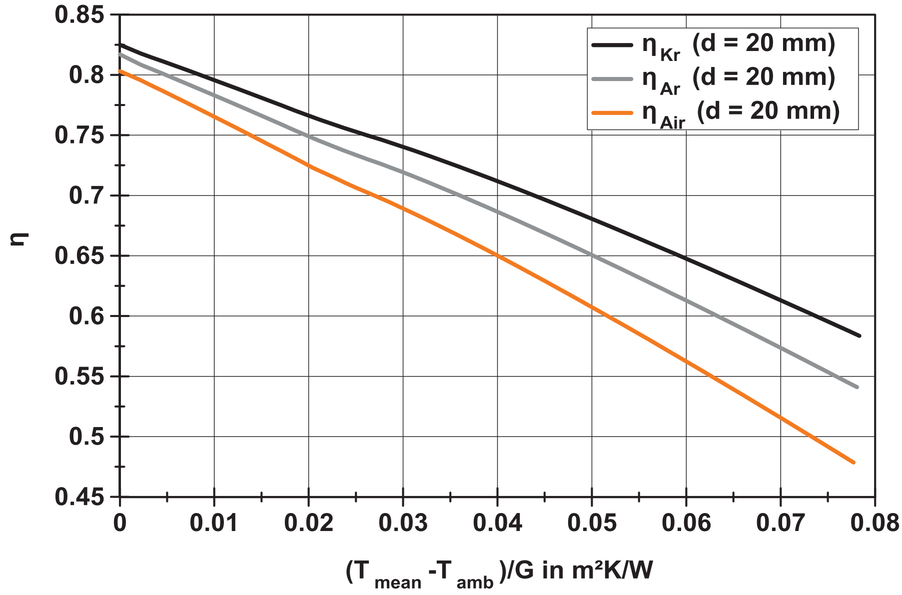

Fig. 8: Efficiency comparison of an Argon and Krypton filled respectively vented collector with a gap size of 20 mm (convection theory according to Eismann (2015))

The collector setting with the Krypton filling outruns the two other simulation results over the complete temperature range. For an ordinate value of 0.05 m²K/W, the efficiency difference between the Krypton and vented collector results in a 7.2 % higher performance whereas the second gas-filled collector version (Argon) lacks 3 % compared to the Krypton variant but outperforms the vented collector by 4.3 %. The predicted efficiency increase by Vestlund (2009, 2012) cannot be confirmed.

Beside the analyses of the collector's efficiency, a system simulation in Carnot (Hafner et al. 1999), an open source toolbox in Matlab / Simulink, and a simulation of the maximum temperature in dry stagnation was conducted. Therefore, the introduced collector model was embedded in a system simulation to derive the magnitude and occurrences of thermal loads during a normal system operation in Germany, maximum temperature during dry stagnation and the annual yield. For this evaluation, a typical solar thermal system for a single-family house in Germany was chosen and implemented in Matlab / Simulink. Since the maximum temperature occurs at the fin end (Duffie and Beckman 2009) and the edge bond was applied in this region, these absorber temperatures were analysed for the thermal load on the primary sealing. However, the edge bond temperatures are slightly lower than the simulated absorber temperatures due to higher edge losses. Yet, it is assumed that this effect is negligible as of the thick side insulation (20 mm). Tab. 3 comprises the used system parameters.

Tab. 3: Applied parameters in the system simulation

Parameter	
Location	Wurzburg (Germany)
Building	Single-family house
Annual heat energy demand in kWh	16,239
Space heating in kWh	2,539
Domestic hot water in kWh	13,700
Heating	Floor heating
Furnace	Oil (15 kW)
Domestic hot water storage volume in litre	130
Buffer storage volume in litre	1,000
Solar collectors in m²	14.4
Collector azimuth	South
Collector parameter TPS-AlCu20	
Optical efficiency	0.815
Linear heat loss coefficient in W/m²K	3.19
Quadratic heat loss coefficient in W/m²K²	0.009
Collector parameter Reference collector	
Optical efficiency	0.815
Linear heat loss coefficient in W/m²K	3.52
Quadratic heat loss coefficient in W/m²K²	0.012
Mass flow in l/m²h	40

Fig. 9 shows the maximum temperatures that occurred during a regular system operation throughout the year.

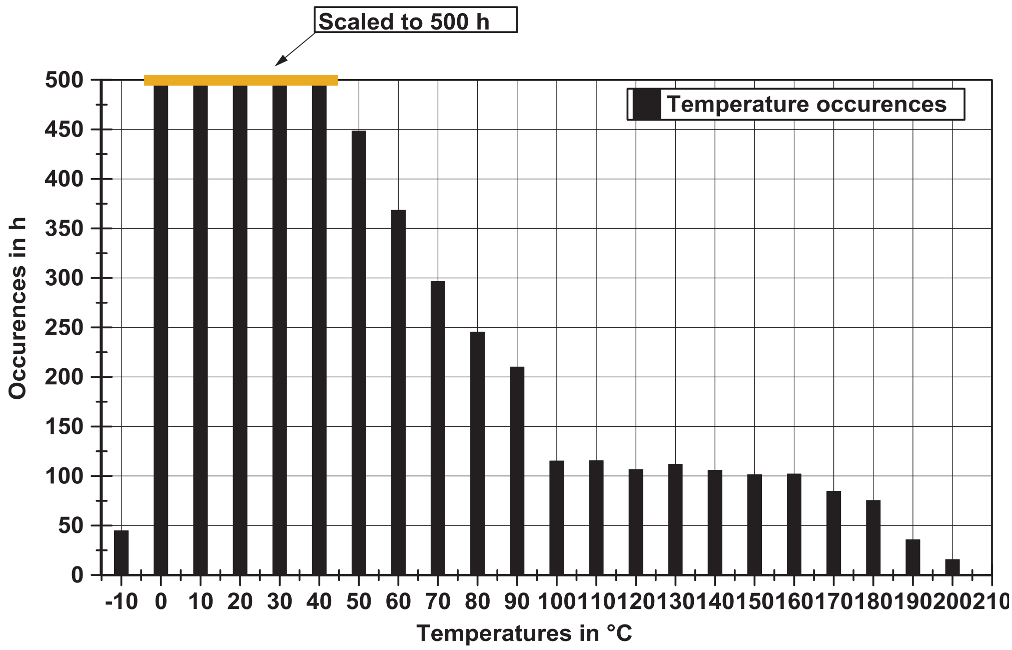

Fig. 9: Temperature load and duration of the absorber temperature during a year in system operation in Wurzburg, Germany

The adhesive service temperatures are not available in public. The temperature range between 100 and 130 °C were defined as short service temperature; however, the idle time cannot be stated. Temperatures exceeding 130 °C are critical and leading to an accelerated ageing effect. In total a dwell time for an absorber temperature between 100 to 130 °C of 449 h respectively 520 h for temperatures above 130 °C were detected. This means 5.9 % of the complete collector operation time the short-term service temperature was exceeded. The higher temperatures, i.e. > 200 °C, are badgering the adhesive more than temperatures close to

130 °C. A temperature amplitude between -13 °C and up to 207 °C was derived by the system simulation. Unlike the high temperatures, this minimal temperature is not afflicted with an ageing effect. To what dimension the temperature magnitude and periods exceeding the service temperature are affecting the material needs to be thoroughly tested, e.g. in outdoor tests.

The gas-filled solar collector achieved an annual solar yield of 5,386 kWh, which corresponds to a specific solar yield of 374 kWh/m²a. In contrast to this, the annual yield of the reference collector is 5,525 kWh (352 kWh/m²a). Further studies with Krypton as gas filling will be conducted as the Argon filled variant only shows a 6 % higher annual yield per collector area than the reference collector.

7. Summary

An existing collector simulation model was adapted, enhanced for gas-filled solar collectors and validated by laboratory testing. The convective heat transfer between absorber and glazing was analysed and compared to recent findings from other scientists. The results confirm the extended convection correlation after Eismann (2015) and show that the use of the convection calculation approach in solar collectors according to Hollands et al. (1976) underrates the heat loss. An efficiency increase of more than 4 to 7 % at a collector working temperature of 0.05 m²K/W for gas-filled collectors was demonstrated. Compared to the values stated in the literature, the efficiency increase of a gas-filled solar collector falls short of expectations (Vestlund 2009, 2012). Short gap sizes of less than 15 mm are critical due to the absorber deflection and, thus, not recommended.

In a system simulation, the thermal loads on the absorber edges and the annual yield of an Argon filled collector were derived. Whether the calculated temperature loads are critical for the adhesive needs to be analysed in a following paper. However, this collector type outruns a vented collector in terms of the annual yield by 6 %. Finally, the proposed collector is produced on a highly automated production line that offers a cost effective assembly for mass production (Riess et al. 2014a, b).

8. References

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