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Analysis of the mechanical behaviour of an all-round fully adhesive supported absorber

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Abstract

This paper discusses the mechanical behaviour of an all-round fully adhesive supported absorber. Previous research is discussed and compared to the author's own results collected using both laboratory tests and simulation. The influence of the finite-element approach (i.e. geometrically linear or non-linear) was compared. The thermal deformation of a hyperstatic absorber was investigated. The relationship between thermal expansion and pressure change in a gas-filled collector is discussed.

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Keywords: Solar collector; Absorber; Thermal expansion; Deformation; Mechanical stress; Hyperstatic; Edge bond

1. Introduction

Adhesive bonding is increasingly being adopted in the design of flat-plate solar collectors. This enables highly automated production in a cost effective manner opening the path towards new design approaches for flat-plate solar collectors. Against this background, new materials and design approaches need to be scientifically analysed in terms of efficiency and life expectancy. As the use of bonding techniques in solar collectors is still in its infancy, little scientific work has been published so far. Today's adhesive technology allows advanced designs in which the absorber is supported in position by an all-round fully adhesive edge bond.

An all-round adhesive supported absorber has several technical advantages over more conventional designs. In this research, an approach from the insulated glazing industry was adopted, which is a two-stage adhesive method. This gives the collector a significant performance advantage due to the hermetically sealed 'interspace' between the absorber and glazing. Unlike un-sealed collector designs, environmental contaminants such as moisture and dust have no negative effects on the absorber surface of the proposed design. Another advantage of the two-stage

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adhesive application is the impermeability to gas of the interspace. Vestlund et al. [1] analysed theoretically the benefits of a solar collector with a gas-filled cavity at ambient pressure. Using simulation, they were able to show a reduction in the overall heat loss coefficient of more than 20%. By decreasing the interspace pressure below ambient conditions the thermal losses can be further minimized (Benz and Beikircher [2], [3]).

The disadvantage of a sealed interspace is the change of pressure in the cavity and the thermal deformation of the absorber during collector operation, resulting in a mechanical load on the absorber, the edge bond and the glazing. As the absorber is less rigid than the glazing, the absorber experiences the largest deformation.

To keep the collector design simple, neither an interspace pressure below ambient pressure nor an additional expansion tank were considered in this work.

In this work the mechanical behaviour of a hyperstatic absorber was simulated taking the thermal elongation of the absorber and the dependent pressure into account. Furthermore, the quality of a geometric non-linear and geometrically linear finite-element model is discussed and compared. A physical model of an all-round fully adhesive supported absorber was set up and investigated in laboratory tests.

Nomenclature

ΔV	Volume change
T	Temperature
p	Pressure
IGU	Insulated glazing units
FE	Finite element
h	Heat transfer coefficient
Subscripts	
Abs	Absorber
Amb	Ambient
C	Convection
Exp	Expansion
G	Gas
Over	Overpressure

2. Previous research in this field

In 2011, Vestlund et al. [15] discussed the movement and mechanical stresses in sealed solar collectors at ambient pressure. A finite-element model was used to examine the behaviour of the sealed collector and its components. A sheet-pipe absorber was modelled as a tray connected at its edges to the glazing. However, the connection between absorber and glazing was modelled as if there were a rigid support. In real applications an adhesive bead is used to achieve a sealed cavity, which also allows deformation. In addition, only a quarter of the absorber was analysed and a symmetrical deformation was assumed.

The parameter studies were conducted using a gas temperature range of 300 to 500 K, but the thermal expansion of the components was ignored, including that of the absorber. This was justified by calculations which showed only a minor influence in the volume change compared to the volume change due to the pressure rise. Unfortunately, these calculations were not explained further.

It should be noted that Vestlund et al. did not mention in their publications whether geometrical non-linearity effects were included. Considering the results of the finite-element analysis of Vestlund et al. it seems likely that a geometrically linear model was used.

In the view of the authors, Vestlund’s validation of his model is problematic. It was not explicitly mentioned how the validation was carried out, nor was the named reference accessible. In addition, the author’s own laboratory results do not correlate with the results given by Vestlund et al.

Neither the influence of a suitable edge bond nor the thermal expansion of the absorber was studied in the research. Instead, a geometrically linear model was used. This may be regarded as a weakness since these factors have a considerable influence on the deflection of the absorber and should therefore be included in studies.

3. Proposed new design of solar thermal collector

A main component of the proposed new collector design is the elastic fully adhesive edge bond. In many industry sectors, such as in the glazing or automotive sector, adhesive technologies are already well established. Nowadays, modern insulated glazing units (IGU) are held together entirely by adhesive. Such an IGU consists of at least two or more low-emissivity coated glass panes. For a further thermal loss reduction, the pane interspace is additionally flooded with an inert gas such as argon or less often krypton. A two-stage adhesive application is used for the IGU edge bond. The primary sealing is based on a modified butyl and functions as an impermeable gas sealant. As the butyl is not designed to withstand the considerable mechanical loads of e.g. structural glazing units, a secondary sealing is needed to add mechanical rigidity. This secondary sealing is based on a silicone adhesive. The significant advantages of this production technology are the very low cycle times as the adhesive application (which means it can be automatically applied by a robot) and, unlike conventional IGU, no metallic spacers are needed to be prepared in advance. The basic designs of an IGU and a flat plate solar collector are similar in that two or more layers are joined by adhesive. The use of adhesive technology allows a much higher level of automation than in current collector production lines, resulting in a more repeatable process with very high quality and low cycle times. More details on the state-of-the-art of solar-collector production are given in [8] and [9]. The positive influence of a gas-filled interspace is discussed in [10] and [11].

In accordance with the author's own simulation results, an interspace of 10 mm was chosen and flooded with argon in the assembled physical models; whereas for conventional, vented collectors a distance of 20 to 25 mm is used. The proposed design features a standard harp absorber made from aluminium and bonded to heat strengthened low iron glass. The short distance between absorber and glazing is attractive as less volume is enclosed in the interspace and less adhesive is needed. This results in lower mechanical loads and costs. The analysis of convective heat transfer h_c for an interspace between to plane parallel plates is given in [12] and [13]. It was concluded that there is a local minimum of h_c for smaller distances and a global minimum for greater gap sizes. Bartelsen et al. [14] investigated this phenomenon for flat-plate collectors down to a gap of 15 mm.

Despite the advantages of a gas-filled interspace, there are also some challenges. Due to the short gap of 10 mm and depending on the enclosed gas, the deformation of the absorber during collector operation is significant as it dominates the front heat loss. In the author's laboratory tests, absorber movements of more than 13 mm were observed. It is therefore preferable that the absorber deflects in a controllable way. To overcome this, find countermeasures and develop design guidelines, the mechanical behaviour of an all-round supported absorber was studied.

4. Methodology

This research focuses on the use of finite-element modelling to analyse the mechanical behaviour of an all-round fully adhesive supported absorber. An overview of finite-element simulation can be found in [4], [5], [6] or [7].

a. Modelling and constraints

Initially, an absorber model was set up in the finite-element programme ANSYS Workbench. The model is composed of a harp absorber made out of aluminium, an adhesive edge bond and the glazing (Fig 1).

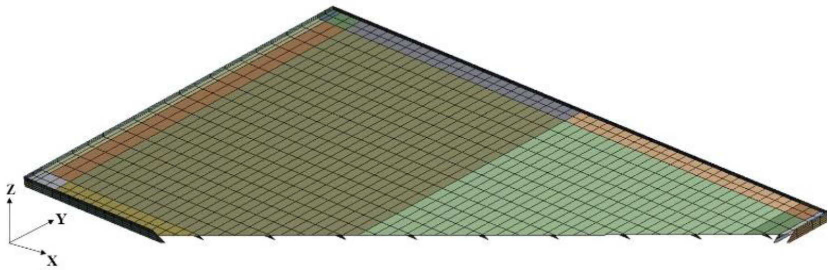

Fig. 1: Definition of the coordinate system and a cross section of the meshed model without the glazing

The model was built up in a CAD environment as a surface model and implemented in the FE programme. During the research it became clear that it was necessary to model two absorber types – an 'ideal' shaped absorber and a 'real' shaped absorber. The ideal shaped absorber was assumed to be completely plane parallel to the glazing without any initial deflections. In contrast to this, for the real shaped absorber the initial deflections of the absorber used in the physical model were taken into account. The edge bond is divided into two sections. The upper part connects the absorber and the glazing whereas the lower part connects the absorber and the back plate. In the simulation the insulated back plate was not modelled. At the lower edge of the edge bond the FE model is fixed supported. The elements used for the FE model were shells. It is assumed that there is no material plasticizing, i.e. linear material behaviour was used. Finally, the earth's gravitation field was taken into account for a collector inclination of 45°.

The ambient conditions, collector parameters and material properties can be seen in Table 1.

Table 1. Boundary conditions and parameters used in the finite-element simulation.

Ambient conditions		
Pressure	100	kPa
Temperature	295	K
Material properties		
Elastic modulus – *Glazing*	70	GPa
Poisson's ratio – *Glazing*	0.23	-
Elastic modulus – *Aluminium*	70	GPa
Poisson's ratio – *Aluminium*	0.33	-
Thermal expansion coefficient – *Aluminium*	$23 \cdot 10^{-6}$	K^{-1}
Geometric parameters		
Absorber thickness – *Aluminium*	0.5	mm
Absorber length	1825	mm
Absorber width	1100	mm
Fin width – w_{fin}	109	mm
Outer diameter – *Riser*	8	mm
Wall thickness – *Riser*	0.4	mm
Outer diameter – *Header*	18	mm
Wall thickness – *Header*	0.8	mm

b. Simplifications

In reality, a temperature gradient exists in the absorber, caused by thermal losses and the cooler collector inlet temperature at the bottom of the collector. However this gradient is small so the finite-element model assumes a uniform absorber temperature.

The piping was modelled as beams. This is justified since the beam section modulus and height are the same respectively as the original risers and headers.

The junction between header and riser were not modelled in this research, however in this joint high stress levels are reached. In the light of a collector life of 20 years and more, the stresses in these soldered or welded joints should be analysed. To retain correct stress dimensions, only the joint with its welded seam needs to be modelled with solid elements and applied with the internal forces. The internal forces are computed using a model similar to the one used in this paper. Considering that the paper is not focussed on fatigue, such an analysis is not carried out.

In the simulation, sufficient space between absorber and insulation is assumed to ensure an unimpeded absorber deflection.

Due to the relatively low temperature on the glazing and its smaller thermal expansion coefficient the lengthening of the glass has no major effect on the mechanical behaviour of the absorber. The elongation of the glazing with a length of 2 m for a temperature rise of 40 K would be less than 0.6 mm. The chosen glazing material properties are the ones for a single pane of safety glass.

Since the absorber is all-round supported and the thermal expansion of the absorber is considered, plate buckling can result, which might cause asymmetric deflections. Hence, a complete absorber was implemented in the finite-element programme. If only the pressure change were analysed, a centre line could have been used to reduce the total number of nodes, thus accelerating the computation.

Aluminium has been chosen for the absorber sheet and the piping, as it is likely that aluminium absorbers are going to replace more costly copper absorbers in the future. There are some advantages of copper compared to aluminium. The thermal expansion coefficient of copper is 28% less than that of aluminium. This leads to a lower temperature elongation compared to aluminium and thus to a less temperature-driven deflection. Collector constructions with an aluminium absorber sheet and copper piping are especially sensitive to thermal stress because of their different thermal expansion coefficient. The deflection of a conventional absorber, (i.e. non-hyperstatic), is well described in [16].

c. Geometric non-linearity

Vestlund et al. seem to have used a geometrically linear model in their study. A linear finite-element model calculates the deformation based on the initial stiffness matrix during the complete procedure. By contrast, by using a geometrically non-linear finite-element model the change of the stiffness according to the current deflection is considered. Hence, using a linear FE model for problems with 'large' deformations will give inaccurate estimations of either the load or deformation.

The mechanical behaviour of the absorber can best be modelled using plate theory. In relevant publications in this field, rules can be found regarding the use of a geometrically linear or non-linear model [17], [18]. Feldmeier [19, 20] analysed the mechanical behaviour of gas-filled insulated glazing units which have a layout similar to the proposed collector design. By simulation and testing, Feldmeier proved that at deflections exceeding the components' material thickness, an error of 100% occurs. As soon as the deflection exceeds the plate thickness the linear Kirchhoff plate theory becomes invalid. To simplify the design of IGU and to avoid failure in their design, the norm for overhead glazing [21] limits the deformation of the first pane to its material thickness resulting in a higher safety factor but also an oversizing of the product.

The author's own data from simulation and laboratory testing on the physical collector and simulation model showed that in the case of an all-round supported absorber the maximum deformation is 20 times larger than its thickness. Thus, a simulation model is needed which includes the effect of geometric non-linearity.

d. Laboratory tests

To validate the finite-element simulation two physical models were constructed and used to analyse the mechanical behaviour at certain collector operation points. In particular, the expansion volume and the shape of the deflected absorber were measured and compared to the simulation results. One of the physical models was equipped with a removable differential pressure transmitter allowing tests with a sealed and opened interspace. The tests were carried out using a solar simulator.

To measure the real deflection of the fully adhesive supported absorber the back plate and insulation of this collector were removed. Subsequent to this a matrix of 323 (Lateral: A – S; Longitudinal: 1 – 17) measuring points was applied on the absorber backside. The deflections measurements were taken just beside the risers respectively midway between two risers. It was assumed that there is only a movement in the z-axis of the absorber. The z-coordinate of the measured points was recorded twice for each test run – at ambient conditions and in the collector operation state. To get the point displacement in the z-direction a digital vernier caliper was used which was fixed on a supported traverse, assuring a constant zero level. The closest agreement between simulation and reality is given in the case of the expansion volume of the deflected absorber. By measuring ΔV_{exp} and T_G, the pressure change can be concluded.

There are some uncertainty factors in the measurement method which influence the calculated volume expansion of the absorber. As the absorber thickness is only 0.5 mm there will be always some sporadic buckling. Another element of uncertainty is the measurement method itself. However, throughout the work a precise optical metrology was used to validate the accuracy of the manually recorded displacements. The comparison showed a good accordance giving an uncertainty margin of 0.5 mm. These factors result in a total deviation of ± 0.85 litres in the volume determination. The expansion volume was deduced by the relative z-displacement of the 323 points.

The coordinates measured with the vernier caliper were rounded to the nearest 0.1 mm. The absorber temperature and glazing temperature were taken at 2/3 of the height and in the middle of the absorber. These points usually indicate the hottest spot on the collector.

5. Simulation results

The collector was modelled to analyse the mechanical behaviour of a hyperstatic absorber. Taking this model as a basis, a study could be carried out of the optimised collector parameters as well as methods for a controlled absorber deformation.

Initially, the dependency between Δp_{Over} and ΔV_{exp} was analysed for the ideal absorber (Fig. 2).

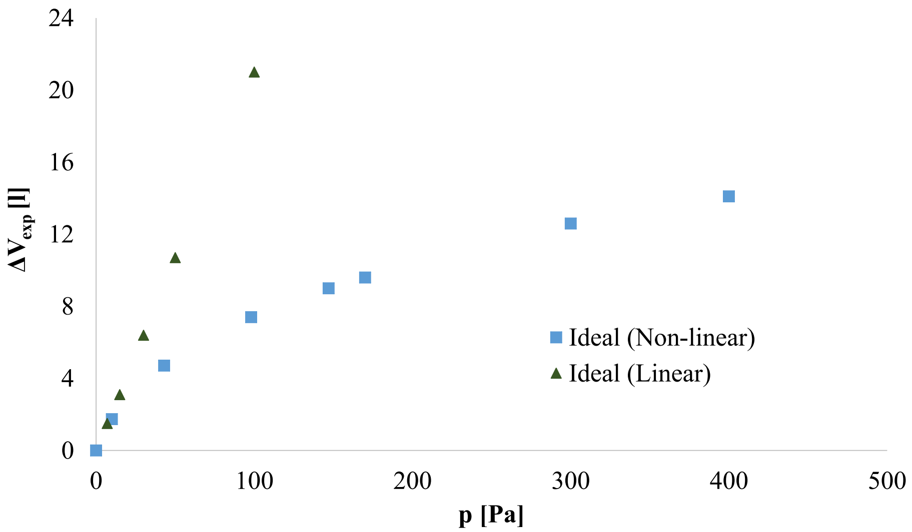

Fig. 2: Dependency between pressure load and expansion volume

The pressure was varied between 0 and 400 Pa. In Fig. 2 the increasing absorber rigidity of the geometrically non-linear model with higher pressure loads is clearly visible. For comparison the same study was conducted with a geometrically linear model. At very small deflections the approach of a geometrically linear model is in good agreement with the geometrically non-linear model.

Fig. 3 shows the deflection curve measured in the middle of the absorber in longitudinal direction.

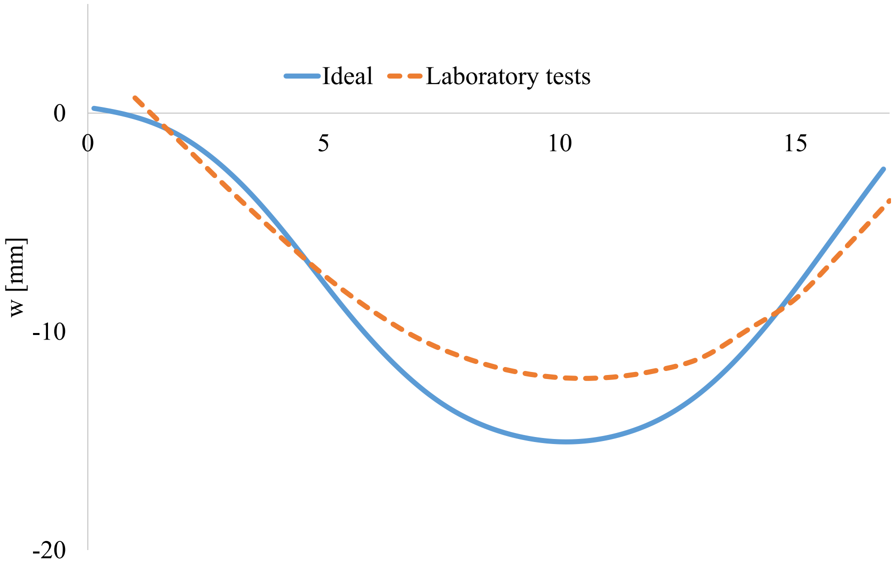

Fig. 3: Measured and simulated deflection curve

The shape of the simulation results is in a good correlation with the laboratory results. However, the test results obtained by laboratory tests are showing a more rigid behaviour than the simulation results.

Overall the idealised absorber shows a less rigid behaviour than the physical model. Also the real shaped absorber model shows some deviation to the laboratory testing. In this model only the initial deflections were implemented, the pre-stresses in the material caused by the welding were not included. This circumstance is probably one of the main reasons for the deviation.

By knowing the ambient conditions, the overpressure and the corresponding expansion volume, it is possible to calculate the mean gas temperature in the collector in this operation point. Assuming that an expansion volume of about 7 litres (measured) is needed at 170 Pa, a mean gas temperature of at least 367 K is required. The mean gas temperature is presumed to be midway between the absorber temperature and glazing temperature. Therefore T_{abs} is significantly higher than the mean gas temperature. However, at a temperature rise of only 74 K the absorber elongates by about 2.9 mm. This length variation needs to be compensated by a deflection in the z-axis as the absorber is supported on all four edges. In the simulation the absorber was heated from ambient conditions up to a uniform absorber temperature of 367 K. At this state an extra volume of about 12 litres was computed, and this was caused purely by the thermal expansion of the absorber. Fig. 4 clarifies this fact.

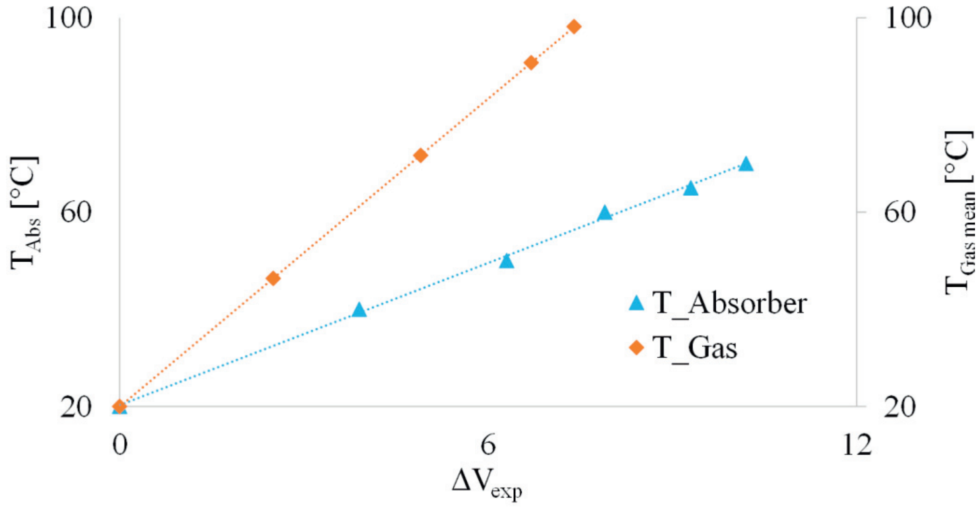

Fig. 4: Dependency of pressure change and thermal elongation on the expansion volume

If the pressure change were dominating the expansion volume the mean gas temperature would need to be higher than the absorber temperature, which is not possible in a normal operation point.

This leads to the conclusion that the thermal expansion is actually the driving factor and not the pressure rise. In fact, the expansion volume caused by the thermal elongation is compensating to an extent for the tendency towards overpressure. That means in a real application a pressure below ambient sets in. This theoretical result was also validated by laboratory tests with the physical collector (Fig. 5).

6. Laboratory results

In particular, three kinds of tests were carried out with the physical model. First the physical model was loaded by pressurising the interspace. In a second test only the thermal elongation at ambient pressure was measured. Finally, both loads were superposed, i.e. the collector was equipped with a pressure transmitter and sealed at ambient conditions.

Fig. 5 is in agreement with the results obtained by the FE simulation.

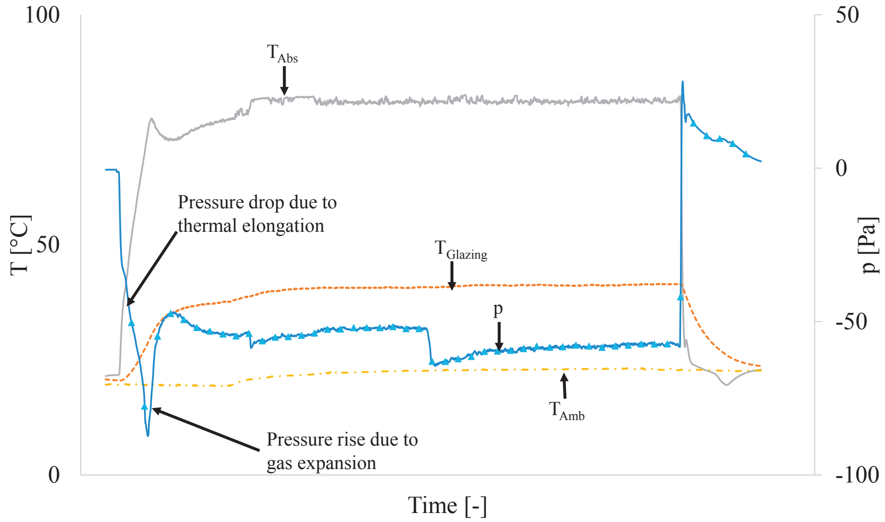

Fig. 5: Pressure change during a test run with sealed interspace

The pressure change is at first related to the absorber temperature. As T_{Abs} rises the pressure drops because of the volume expansion caused by the thermal elongation of the absorber. After some time the gas in the interspace heats up and the pressure decreases below ambient. At the end of the test run the pressure is again at ambient conditions.

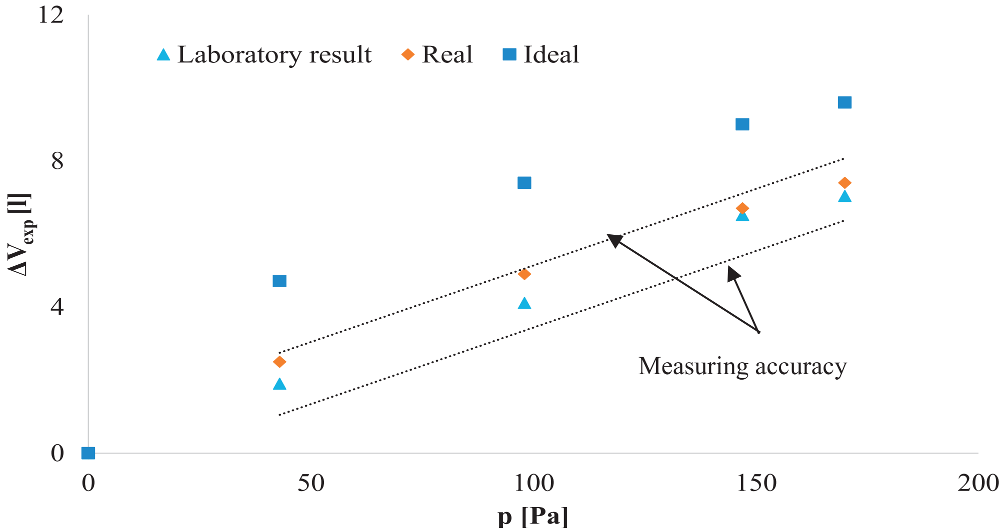

Fig. 6: Comparison of the simulation results and results obtained by laboratory testing

Fig. 6 shows the correlation between the ideal and real shaped absorber as well as the measured deflection of the physical model for a pressure between 43 and 170 Pa.

The graph shows a significant deviation between the ΔV_{exp} of the ideal absorber and the laboratory results. In reality the absorber has an initial deflection which dominates the deformation. A main cause of this is the welding process during the absorber production as the material is pre-stressed by structural changes. To obtain a better correlation between simulation and measurement the initial deflection of the absorber was taken into account for the

FE model. In fact, this implementation is in a very good agreement with the laboratory results. Also the measured deflection curve and the simulated one is in a superior accordance.

7. Conclusions

The key finding of this paper results from an analysis of the mechanical behaviour of an absorber that is all-round supported by a fully adhesive edge bond. As a result of this research, a physical model of a fully adhesive bonded flat-plate solar collector has been created. A finite-element model of the absorber was implemented and compared with test results collected in laboratory tests. Two different finite-element models - geometrically linear and non-linear – were investigated. The pressure change as well as the thermal elongation of the absorber for a temperature difference of 80K was taken into account.

In the case of an absorber which is all-round supported it is necessary to model an elastic edge bond. An idealised support such as a linear support will not reflect the real mechanical behaviour, especially for thermal elongations.

The results show that the model including geometric non-linearity is much closer to the reality. By contrast, using geometrical linearity will lead to an oversizing of the components, which increases costs.

It was shown that thermal elongation cannot be omitted in such simulations. In fact, the expansion volume created by the thermal elongation dominates the pressure change. In terms of the physical model a pressure below ambient rise was found, which shows that that the pressure variation can occur in both directions. By adjusting certain parameters the pressure variation (negative or positive) can be influenced. These influencing parameters have to be studied more in detail.

The mechanical behaviour of the real shaped absorber is almost identical to the behaviour measured in the laboratory tests. It can be assumed that the simulation model itself is correctly set up regarding the material parameters and constraints. The only difference between the ideally shaped absorber and the real shaped absorber is the initial deflection. It can be therefore concluded that the model of the idealised absorber can be used for further simulation studies. This is an important result as the initial deflection of every absorber differs.

Even though, the absorber deflection of the physical model was to the ‘back’ the absorber could have shown a deflection towards to the glazing as its deformation is strongly depended on the initial deflection. A pressure below ambient will reinforce this behaviour. For such a design approach it is important to prevent an uncontrolled deflection.

8. Outlook

This work was conducted as a basis for further scientific research work in the field of sealed flat-plate collectors. Parameter studies will be carried out identifying the optimal material and geometrical parameters for such collector approaches. These studies can be now conducted with the idealised absorber as it was proven that the simulation model was set up correctly. Linked with these studies corrective actions are analysed to control the absorber deflection which are also of interest for conventional vented collectors as smaller distances between absorber and glazing can be achieved.

In the next year, further laboratory tests are planned. In particular, different absorber types (roll-bond-absorber) and material pairing will be investigated and used to validate the FE simulation.

For such collector designs the components stress level are of great importance. To predict the stress dimensions a more detailed absorber (welding seams) model needs to be set up. However, the internal forces in the critical areas can be obtained based on this simulation and used for a further detail study. Finally, the stress in the edge bond and in the welding seams needs to be investigated to ensure a collector life time of at least 20 years.

A thermal performance study of the physical models is conducted as the thermal loss reducing inert gas in the interspace is a promising approach.

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