NEW ABSORBER MANUFACTURING AND MATERIALS – CHALLENGES FOR ABSORBER DESIGN AND EVALUATION

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1. Introduction

State-of-the-art solar thermal absorbers are mostly based on sheet-and-tube constructions. Laser welding, ultrasonic welding or soldering are common technologies to connect the absorber sheet with the fluid channels. For future absorbers both new manufacturing and materials are key issues for innovation and cost savings. PVT collectors for example need rigid, one-side-flat absorbers. If metals are used, potential alternative manufacturing technologies are in most cases based on sheet metal forming such as roll-bonding, hydroforming or deep-drawing. The main difference to state-of-the-art is that with these technologies absorber and channel form a union, i. e. the thermal bottleneck of a sheet-tube connection is not relevant anymore. Moreover, the number of channels does not have a big influence on the absorber costs, and there is much more flexibility regarding the geometry of both the channel paths and their cross sections. Fraunhofer ISE has been working on this topic together with industry as well as research partners for some years now (Hermann et al., 2010). Examples are the European project BIONICOL (aluminium roll-bond absorbers, Hermann et al., 2011), the German project STAHLABS (steel absorbers produced by roll cladding and hydroforming, Koch et al., 2011) and work done in direct contact with manufacturers willing to enter the solar thermal market. The question arising with the gained flexibility in design is how the channel pattern and the channel cross sections should look like with respect to high thermal efficiency as well as low pressure drop, taking the material and the sheet thickness into consideration. One example for work on channel patterns is the so-called FracTherm® algorithm (Hermann, 2005). But even if standard meander or harp patterns are used, it is still a challenge to design and assess absorbers within the framework of the boundary conditions given by the manufacturing technology and the material.

2. Examples of alternative absorber designs in comparison with state-of-the-art

Sheet-and-tube constructions (Fig. 1, left) are commonly used in solar thermal absorber technology. One advantage of this construction is that sheet and tube wall thickness and material can be different (e. g. copper tube attached to aluminium sheet). Thus the functions of heat and fluid transport can be separated, which means that e. g. the tube wall thickness can be adjusted to the internal pressure independently of the absorber sheet. However, the thermal bottleneck of this construction is the junction between sheet and tube, and moreover the mechanical load in this area is challenging, especially if materials with different thermal expansion coefficients are used and sudden changes in temperature occur (thermal shock). Therefore joining technologies should ensure both a good thermal and mechanical performance. Today ultrasonic or laser welding are mostly used. Another disadvantage of sheet-and-tube constructions is that the channel design is not very flexible: conventional harp and meander absorbers are easy to be realized, but more complex geometries might only be possible at much higher costs.

An alternative to the previous concept is to build absorbers which form a unity of sheet and channel, which the authors will name "integrated absorber" within this paper (Fig. 1, right). It is obvious that a disadvantage is that in this case both wall thickness and material of sheet and channel are not independent: the channel wall thickness is half of the sheet thickness (provided that two sheets of identical thickness are bonded). This automatically leads to relatively thick absorbers if a certain resistance against internal pressure in the channel is to be guaranteed. Moreover, in most cases it will not be possible to use sheets which are pre-coated with a selective coating. However, apart from these drawbacks it is obvious that the integrated absorber has many advantages: both thermal and mechanical contact are much better, the channel distance can be reduced without significant additional costs, and there is a high flexibility in design. Thus it is very easy to optimize channel designs in order to increase the collector efficiency factor *F'* or to adjust it to various boundary conditions. An example for alternative designs are FracTherm® structures developed at Fraunhofer ISE which allow for adjusting the channel pattern also to non-rectangular absorber geometries (Fig. 2).

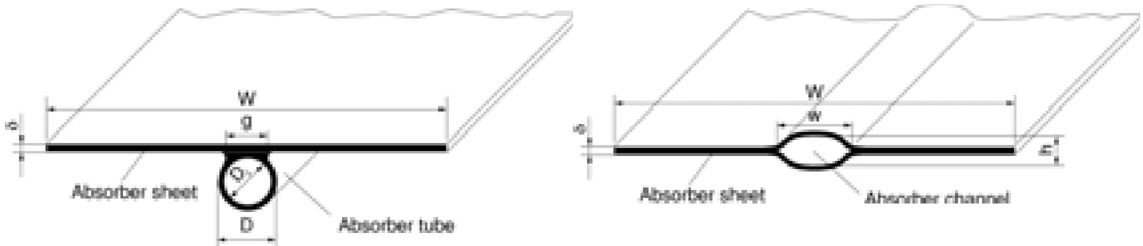

Fig. 1: Schematic sketches of sheet-and-tube absorber (left) and integrated absorber (right)

Fig. 2: Part of a solar collector with an aluminium roll-bond absorber featuring a FracTherm® channel structure (left); drawing of a triangular FracTherm® absorber (right)

Fig. 3 shows channel samples of an aluminium roll-bond panel and a roll-cladded steel panel with copper layers. The latter is produced by two roll-cladding steps and a final hydroforming process. In the first step two thin copper sheets are roll-cladded to a steel core plate. In the second step two of these hybrid plates are roll-cladded again, but this time the bonding only occurs partially in the areas which are not covered by a separating channel layer applied to one of the sheets before. Finally, the structure is pressed into a forming tool, which determines the final cross-sectional geometry, using a water-oil emulsion. This is the main difference to the roll-bond process which does not use forming tools, but only a flat plate determining the final height of the channels. Moreover, roll-bonding uses air instead of a water-oil emulsion. The difference of the resulting channel cross sections can easily be seen from Fig. 3.

Fig. 3: Channel samples of an aluminium roll-bond panel (left) and a roll-cladded steel panel with copper layers (right)

3. Freedom of design and resulting challenges

The freedom of design is an important issue in the context of alternative production processes and materials. On the one hand there are many more possibilities in comparison with fin-and-tube constructions. On the other hand the question is how a good channel design should look like. Fig. 4 shows two different boundary conditions for this design process: a 2-D channel design for panels with constant height (e. g. for the classical aluminium roll-bond process) and a 3-D channel design (e. g. for roll-cladding and hydroforming of steel). There are different reasons for possible restrictions of the channel height, e. g. material properties (formability), process temperature (influence on material properties), standard applications (e. g. refrigerators) or costs (no need of forming tool). A main disadvantage of a constant, small channel height is that large cross-sectional areas can only be realized with very wide channels. This is especially necessary for channels which are intended for large volume flow rates and low pressure drops such as header channels of an absorber. The usual roll-bond solution is a wide channel with an “island pattern” which is necessary in order to avoid deformation (Fig. 5). However, it can be shown that such a flat rectangular channel (“islands” neglected) with a typical roll-bond height of 3 mm would need to be extremely wide (e. g. 350 mm for a mass flow rate of 150 kg/h) in order to obtain a pressure drop which is as low as in a conventional 20 mm header tube. Moreover, it is difficult to find appropriate solutions in order to connect several of such header channels in parallel.

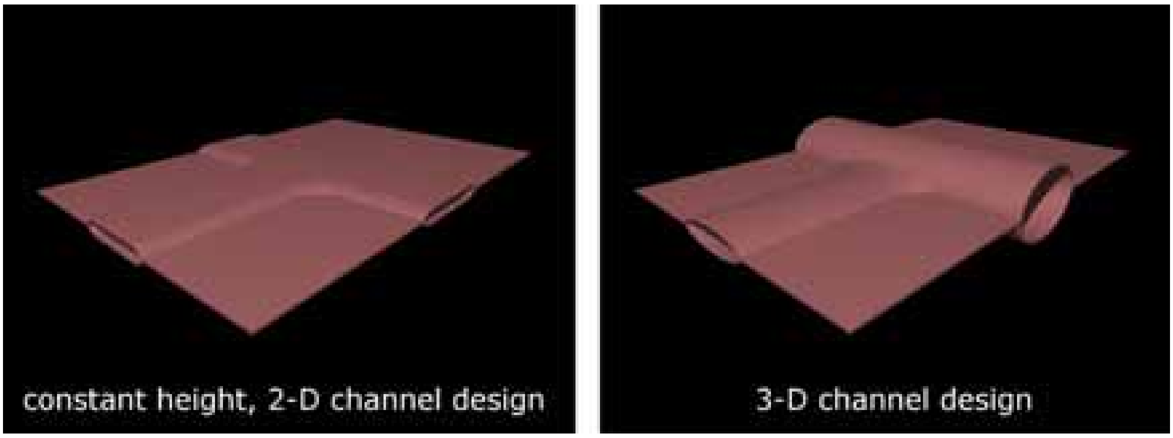

Fig. 4: Different degrees of freedom: only 2-D for constant height (left), 3-D if height is not fixed (right)

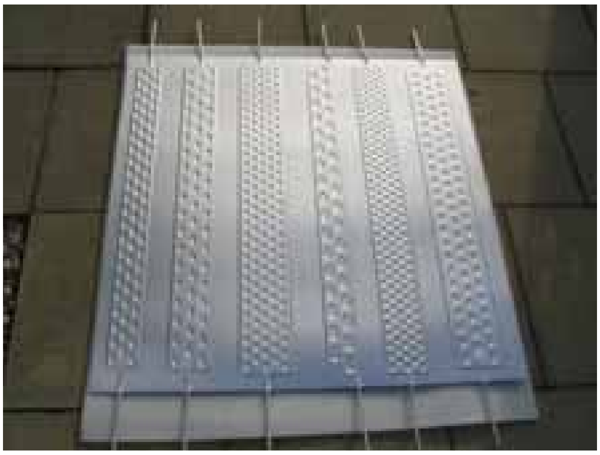

Fig. 5: Different possible roll-bond header channels

If a further degree of freedom is added – which means that the design is not restricted to a constant height anymore –, it becomes much easier to realize optimized channel structures. Fig. 6 gives an impression of what might be possible. But still the question remains: What does “optimized” mean? Should the channels be close to a tube with circular cross section? Or should they rather tend to be rectangular? Is it better to have a lot of channels with small cross-sectional areas or only few with larger ones? In order to answer all of these questions it is necessary to first have a look at the properties of a solar absorber which are influenced by its channel geometry and also at the boundary conditions given by the material and the bonding technology.

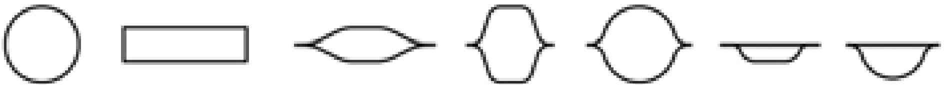

Fig. 6: Different potential cross sections of absorber channels (principle sketches)

4. Influences on the design process

The design process is quite complex because it has an influence on thermodynamics, fluid dynamics and mechanics and vice versa (Fig. 7). Moreover, all of these three key issues also influence each other.

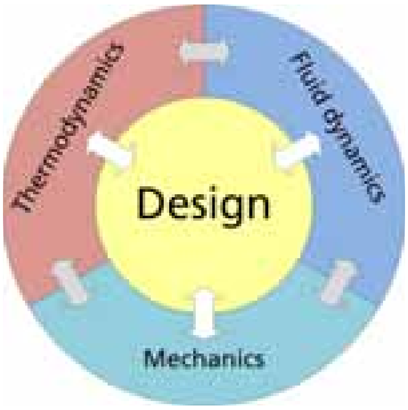

Fig. 7: Influences on the design process

The interdependency of design, thermodynamics, fluid dynamics and mechanics can be explained by the following example:

The width of an absorber channel with constant height is to be increased. This will lead to an increase of the cross-sectional area and therefore a decrease of the flow speed (assuming the same volume flow rate). The increase of the width will have a positive influence on the thermal efficiency, since the distance between the edges of two channels will become smaller, however the heat transfer in the channel will be reduced due to the decrease of the flow speed which again has a negative influence on the thermal efficiency. Concerning fluid dynamics the effect of a wider channel will be a reduction of the pressure drop and thus of the hydraulic power needed to transport the fluid. However, concerning mechanics the risk of deformation under internal pressure rises.

These are only some – not all – effects of a slight change of just one design parameter out of many. In order to get an estimation of the sensitivity of such changes, a quantitative analysis is necessary.

Since the purpose of a solar absorber is to gain thermal energy, we will first focus on its thermal efficiency which can be assessed by the well-known collector efficiency factor *F'*. For conventional sheet-and-tube constructions Duffie and Beckman (1991) have established the following equation:

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L[D+(W-D)F]}+\frac{1}{C_b}+\frac{1}{\pi D_i h_{fi}}\right]} \qquad \text{(eq. 1)}$$

where

$$F = \frac{tanh[m(W-D)/2]}{m(W-D)/2} \qquad \text{(eq. 2)}$$

and

$$m = \sqrt{U_L/k\delta} \qquad \text{(eq. 3)}$$

The geometry parameters W, D, D_i and δ can be taken from Fig. 1, left (Duffie and Beckman assume that $g = D$). U_L is the overall heat transfer coefficient, C_b is the bond conductance, k is the thermal conductivity of the absorber sheet and h_{fi} is the heat transfer coefficient between the tube wall and the fluid.

For the integrated absorber in Fig. 1, right, we propose to adapt the formula as follows:

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L[w+(W-w)F]}+\frac{1}{Ph_{fi}}\right]} \qquad \text{(eq. 4)}$$

where

$$F = \frac{tanh[m(W-w)/2]}{m(W-w)/2} \qquad \text{(eq. 5)}$$

and

$$m = \sqrt{U_L/k\delta} \qquad \text{(eq. 6)}$$

The diameter D is replaced by the internal channel width w, and the heat transfer coefficient h_{fi} is multiplied by the general internal perimeter P. We propose to use the internal width w instead of the external one for two reasons: First, we assume the absorber temperature to be nearly constant above the fluid, but on the sides there must still be a temperature gradient in order to induce a heat transport to the fluid. Second, the internal width w is easier to determine.

For an ideal rectangular channel P can directly be calculated from the channel width w and height h:

$$P = 2(w + h) \qquad \text{(eq. 7)}$$

However, for a real cross section P has to be determined from either a drawing or from measurements of produced channels.

In eq. 4 the bond conductance C_b has completely disappeared because there is no connection between sheet and tube anymore since sheet and channel form a unit ("integrated absorber"). It can directly be seen that this leads to a smaller denominator in eq. 4 (if all other parameters keep the same) and thus to a higher value of F'. It should be noted that the absorber and channel design has a direct or indirect influence on the parameters W, w, δ and h_{fi} and thus also on the collector efficiency factor F'.

The heat transfer coefficient h_{fi} is influenced by the hydraulic diameter D_h and the flow speed v. Both D_h and v again depend strongly on the cross-section of the channels and are also relevant for the pressure drop, which brings us to the next important issue.

The pressure drop of a straight pipe or – more generally spoken – a straight channel can be calculated as follows:

$$\Delta p = f \cdot \frac{l}{D_h} \cdot \frac{\rho}{2} v^2 \qquad \text{(eq. 8)}$$

where f is the friction coefficient, l the channel length, D_h the hydraulic diameter, ρ the fluid density and v the mean flow speed. f is a function of the Reynolds number Re:

$$Re = \frac{vD_h}{\nu} \qquad \text{(eq. 9)}$$

where ν is the kinematic viscosity. It is obvious that D_h and v again play an important role; therefore their quantitative dependence on a general cross-sectional geometry should be described:

$$D_h = \frac{4A}{P} \qquad \text{(eq. 10)}$$

where A is the cross-sectional area and P the wetted perimeter. The mean flow speed is

$$v = \frac{Q}{A} \qquad \text{(eq. 11)}$$

where Q is the volume flow rate.

We finally see that both for the calculation of the collector efficiency factor *F'* and the pressure drop Δp the cross-sectional area *A* and the wetted perimeter *P* are the most important values determined by design. Fig. 8 shows an ideal rectangle as well as a possible real cross section of a fluid channel. In order to describe and calculate a real cross section, we introduced geometry factors which describe the cross-sectional area and the perimeter in relation to the corresponding values of an ideal rectangle:

$$f_A = \frac{A}{A_R} \qquad \text{(eq. 12)}$$

and

$$f_P = \frac{P}{P_R} \qquad \text{(eq. 13)}$$

Thus e. g. the hydraulic diameter becomes

$$D_h = \frac{4A}{P} = \frac{4f_A A_R}{f_P P_R} = \frac{f_A}{f_P} \cdot D_{h,R} = \frac{f_A}{f_P} \cdot \frac{2wh}{w+h} \qquad \text{(eq. 14)}$$

Using these geometry factors the calculation of Δp and *F'* can easily be adapted to changed cross-section designs with given internal width and height. All formulae are based on rectangular channels and adjusted by the introduction of the area correction factor f_A and the perimeter correction factor f_P. It is useful to give the correction factors as functions of height *h* over width *w*: $f_A(h/w)$ and $f_P(h/w)$. Apart from these pure geometric corrections, which can easily be calculated or measured e. g. from a scanned real channel, there are also more effects on the hydraulic and thermal behavior which can only be determined empirically. An example is the correction factor φ which is used in order to correct the laminar friction factor (see eq. 8) of a circular tube:

$$f = \varphi \cdot \frac{64}{Re} \qquad \text{(eq. 15)}$$

In the literature the correction factor φ is given e. g. for rectangular channels as a function of *h/w*. However, for non-rectangular channels with special shapes it is usually necessary to determine it by measurements. It would be very useful to introduce a general correction factor φ *(h/w)* independently of the individual cross section; however more work has to be done in order to find out the correct correlation between the friction factor and the cross-sectional geometry.

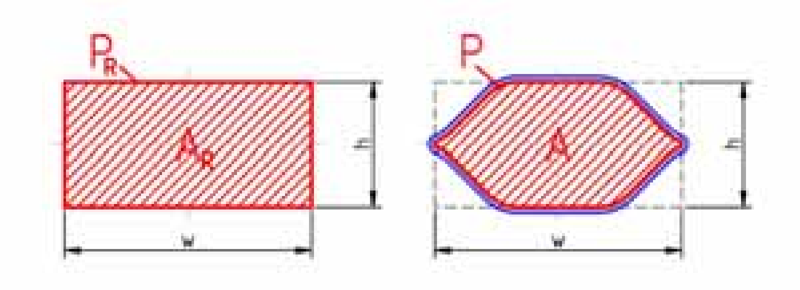

Fig. 8: Rectangle and real cross section with cross-sectional area A and perimeter P

Up to now we only regarded straight channels. However, an absorber channel structure consists of curved channels (meander design, FracTherm® design) and/or T-pieces (harp design) or more complex bifurcation geometries (FracTherm® design). The pressure drop of these components has to be taken into consideration if the total pressure drop and the volume flow distribution are to be calculated. The additional pressure drop can be calculated by multiplying the dynamic pressure by a pressure drop factor ζ:

$$\Delta p = \zeta \cdot \frac{\rho}{2} v^2 \qquad \text{(eq. 16)}$$

ζ values have to be determined experimentally; they depend on the channel geometry and – for diverging or converging elements such as T-pieces – from the volume flow ratio of both channels. Even for standard geometries already numerous ζ value tables exist (Idelchik 1994). Weitbrecht et al. (2002) have shown that ζ values of T-pieces also very strongly depend on the Reynolds number. For 180° elbows or T-pieces with circular or rectangular cross sections data for ζ values exist, but usually not in dependence of the Reynolds number. If we now increase the degree of freedom concerning the design of the channel structure, there is no chance to find appropriate ζ values in the literature. This is especially true for FracTherm® bifurcations and channel parts with a change in height (see Fig. 4, right). Therefore the ζ values for such channel structures have to be determined by own measurements or CFD (Computational Fluid Dynamics) simulations.

If we look at the sample cross section in Fig. 8, right, an important question is whether such a channel cross section is really producible. The boundary conditions of the production process, e. g. metal forming, have to be considered. For example different strains and sheet thickness reductions depending on material, sheet thickness and channel geometry occur in a hydroforming part (Fig. 9). There are maximum values which may not be exceeded. Thus given mechanical boundary conditions are restrictions for the design and vice versa: a given design can lead to a mechanical restriction with respect to operation of the absorber; e. g. the chosen channel cross-section may define the maximum internal pressure allowed in the solar thermal system.

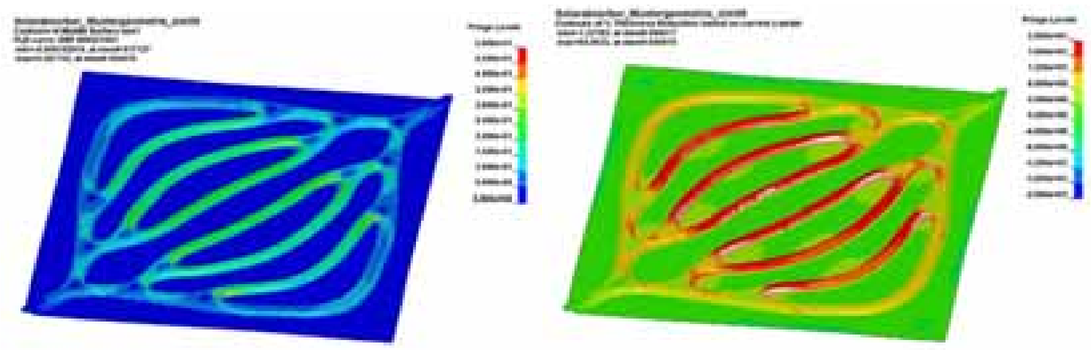

Fig. 9: Simulated effective plastic strain (left) and sheet thickness reduction (right) of a sample FracTherm® channel structure to be produced by hydroforming (Koch et al. 2011; simulations carried out by Institute of Forming Technology and Lightweight Construction, TU Dortmund)

5. Fluid dynamics test facility

As mentioned in section 4, experimental and/or CFD simulations are necessary in order to be able to carry out fast analytical calculations of a complex integrated absorber. Concerning experiments both quantitative (measurements) and qualitative (visualizations) investigations are necessary. In order to do this, Fraunhofer ISE has developed a fluid dynamics test facility (Fig. 10).

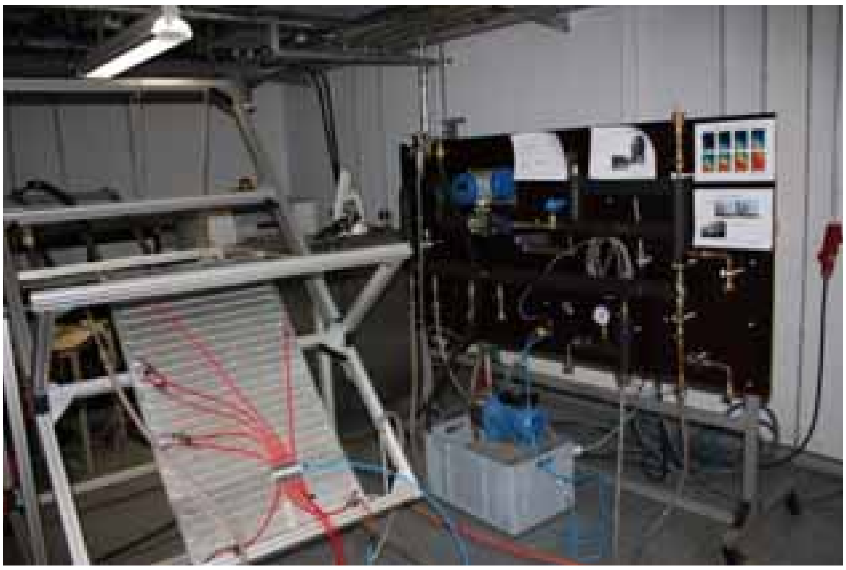

Fig. 10: Part of fluid dynamics test facility at Fraunhofer ISE

Qualitative investigations can be done either by thermography (visualization of flow distribution using warm water as a tracer medium, Fig. 11) or by hydrogen bubbles (Fig. 12). For the latter an own construction has been developed: since it is important to ensure an appropriate lighting all over the channel structure, we use a fluorescent collector material (originally also an early development of Fraunhofer ISE) in order to collect light on a big surface and emit it concentrated at the walls of the channels which have been cut out of the material using waterjet cutting.

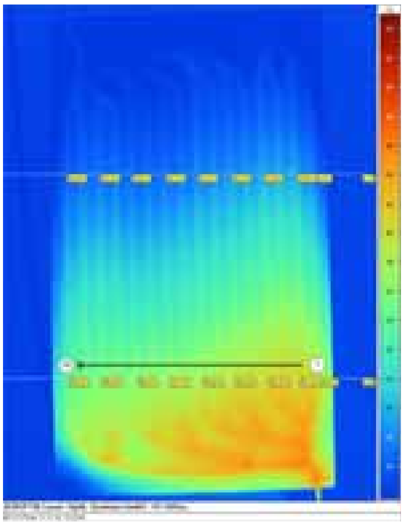

Fig. 11: Thermography picture of a sample absorber

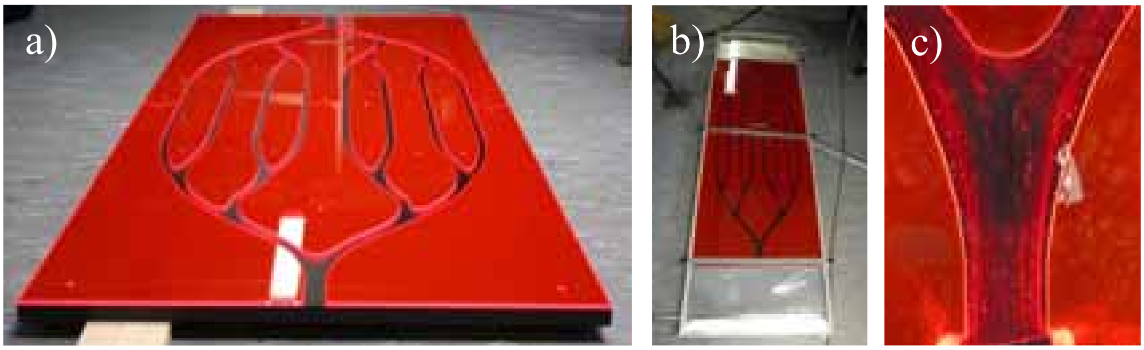

Fig. 12: Sample absorber structure made of a fluorescent collector (a), installation for H_2 bubble visualization (b) and streamlines at a FracTherm® bifurcation (c)

The example shown in Fig. 12 is a scaled model with channels having a rectangular cross section. The H_2 bubbles are directly produced in front of the inlet channel by hydrolysis. Fig. 12 c shows that with this method it is possible to visualize streamlines in the channels. This is especially useful for a better understanding of what happens at T-pieces, Y-pieces, FracTherm® bifurcations or any other structures distributing a fluid flow. The experiments complement CFD simulations (Fig. 13) very well.

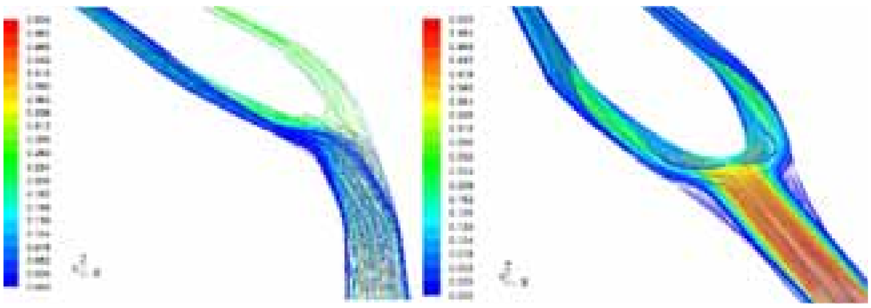

Fig. 13: CFD simulations of FracTherm® bifurcations

Quantitative investigations can be done with pressure drop measurements of real absorbers. Apart from conventional pressure drop measurements of a complete absorber at different volume flow rates we also prepare absorbers with a number of pressure measuring points. Thus it is possible to locate pressure drops within an absorber structure and use the measured values to calculate the ζ value of the relevant component (e. g. an elbow). In the same way correction factors φ of straight channels with a special cross section not known in the literature can be determined. Both ζ and φ values are necessary in order to be able to carry out analytical calculations of an integrated absorber.

Fig. 14: Sample meander panel with a number of pressure measuring points

6. Thermal and hydraulic calculations of sample absorber designs

In order to get an impression of the effect of changes in design on the thermal efficiency and the pressure drop we carried out a study based on some simple assumptions. Fig. 15 shows five integrated absorber designs with different channel cross sections. Type 3 acts as a reference; its geometry is taken from a real roll-bond absorber. The geometry factors as well as the correction factor φ are known; therefore analytical calculations can be carried out. In order to obtain the same fluid volume of the absorber and thus the same heat capacity (which has an influence on the annual collector yield), all other designs have the same cross-sectional area as the reference absorber. In contrast to the producible roll-bond absorber the other designs are idealized in order to find out which of them should rather act as a prototype for further investigations. Type 1 features a circular, type 2 a square and type 4 a rectangular (flat) cross section. Type 5 is an absorber with a fluid gap between two sheets. Due to the given cross-sectional area the gap distance becomes extremely small. All five absorbers are regarded to be ideal harp absorbers with uniform flow distribution (header channels neglected). The distance between the middle of the channels is identical for types 1-4; type 5 only features one channel (the gap).

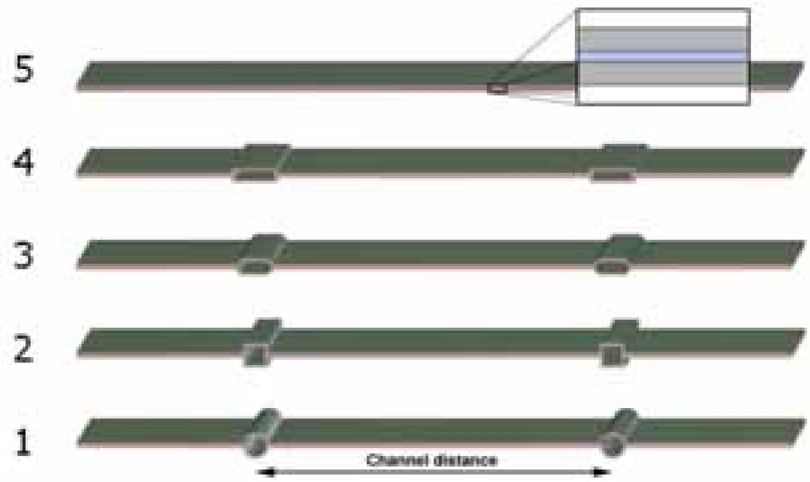

Fig. 15: Absorber designs with different types of channel cross section

The assumptions for the calculations are as follows:

- Perimeter (only type 3): 24.9 mm
- Cross-sectional area (for all types, taken from type 3): 28.2 mm
- Correction factor φ (only for type 3): 1.27
- Thermal conductivity (stainless steel): 15 W/(mK)
- Sheet thickness between channels: 0.5 mm
- Overall heat transfer coefficient U_L: 3.5 W/(m²K)
- Absorber size: 1 m x 1 m = 1 m²
- Fluid: water
- Temperature: 80 °C
- Mass flow rate (total absorber): 72 kg/(m²h)
- Flow condition: laminar
- Cross-sectional area per channel remained the same for all variations of the channel distance

For this sample study we used stainless steel as an absorber material since it has quite a poor thermal conductivity (about 4 % of copper). On the one hand we wanted to show that despite this poor value it is possible to reach thermal efficiencies which are competitive to state of the art. On the other hand we also wanted to point out that such efficiencies can only be reached if the channel distance is significantly reduced. For a conventional sheet-and-tube construction this would lead to additional costs due to the increased number of tubes and the effort for joining them to the absorber sheet. For an integrated absorber the costs remain almost the same (depending on the tool). However, it has to be considered that the channel wall thickness is usually half of the absorber sheet thickness; therefore a minimum sheet thickness is necessary in order to withstand the internal pressure in the channels during operation. It might be necessary that for certain integrated absorbers the maximum internal pressure in the solar system has to be reduced (e. g. for aluminium roll-bond absorbers) in order to prevent deformation.

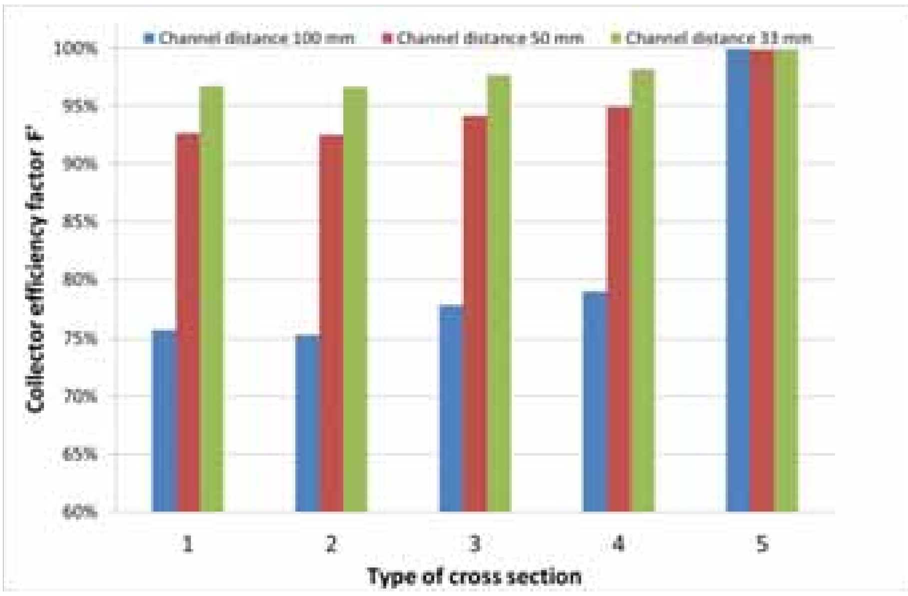

Fig. 16: Calculated collector efficiency factor F' for different types of cross section and different channel distances

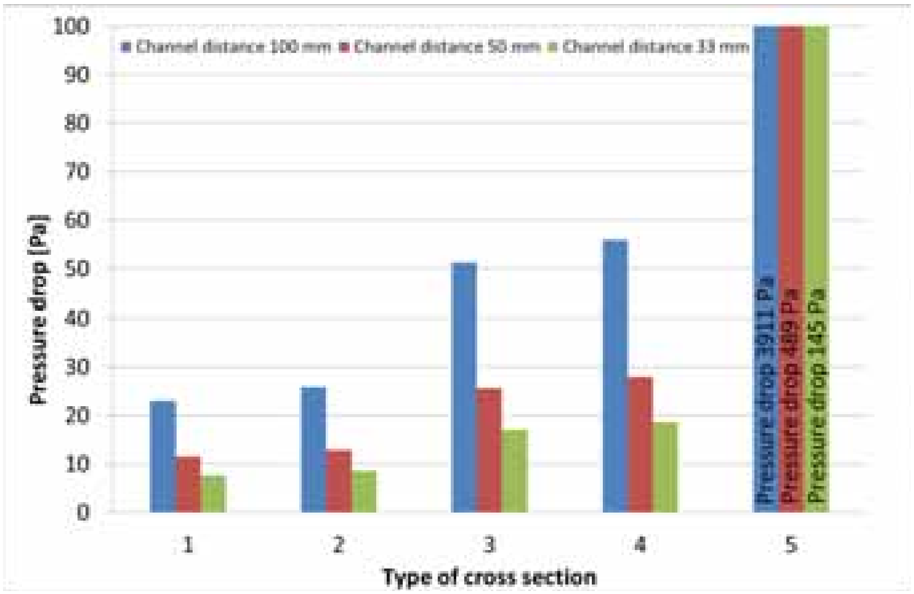

Fig. 17: Calculated pressure drop Δp for different types of cross section and different channel distances

Fig. 16 shows that all types of absorbers need a reduction of the channel distance in order to reach *F'* values which are significantly higher than 0.9 (which should be reached in order to be competitive). It can be seen that the flatter, wider types 3 and 4 lead to slightly higher *F'* values. The reasons are a smaller distance between the channel borders and a higher internal heat transfer coefficient h_{fi} which again is influenced by the smaller hydraulic diameter. In type 5, where the channel width becomes equal to the absorber width, the mentioned effects are maximized. It has to be pointed out that according to the literature (VDI-Gesellschaft, 2006) there are no standard equations for the calculation of heat transfer coefficients h_{fi} in non-circular channels at laminar flow conditions. So actually the hydraulic diameter can only be used for turbulent flows. However, it is expected that qualitatively the result will remain the same.

Concerning the pressure drop calculation (Fig. 17) we can state that the increase in thermal efficiency is not for free: types 3 and 4 (high aspect ratio) show a much higher pressure drop (about factor of 2) than types 1 and 2 (aspect ratio of 1). For the narrow gap in type 5 the pressure drop becomes extreme due to the boundary condition of the same overall cross-sectional area as the other types. The reduction of the pressure drop with decreasing channel distance is caused by the fact that the cross-sectional area per channel remained constant while the number of channels and thus the overall cross-sectional area was increased.

Since the hydraulic power needed to drive the pump is just a small percentage related to the thermal power it is of course a question how to assess the increase of the pressure drop in relation to the gain in thermal efficiency. However, our conclusion is that as soon as the channel distance is small enough, the gain in thermal efficiency is only minor, while the increase in pressure drop is significant. Therefore we recommend focusing on a large number of channels with nearly circular cross section, because the hydraulic effort for just a small increase in thermal efficiency cannot be justified. It makes more sense to increase the efficiency by a further decrease of the channel distance if possible. This again means that if the production process and the material offer the possibility to manufacture cross sections which are rather nearly circular than wide and flat, the circular design should be favored (Fig. 3, right instead of Fig. 3, left). A gap design with the same fluid volume turns out to be unrealistic; so this solution can only be used if a larger volume and thus a higher thermal capacity can be accepted. Nevertheless it should be kept in mind that for such a construction there must be concepts which ensure a uniform flow distribution as well as stability against internal pressure.

7. Conclusion and outlook

The paper shows that absorber concepts based on an "integrated" approach (absorber and channels form a unit) offer new possibilities for design and manufacturing. At the same time new challenges arise due to the complex interdependency of design, thermodynamics, fluid dynamics and mechanics. In order to be able to carry out analytical calculations of the thermal efficiency and the pressure drop, it is necessary to do more simulations and experiments since there is not enough information available in the literature. A new fluid dynamics test facility at Fraunhofer ISE offers possibilities to carry out both qualitative and quantitative investigations of original absorbers and scaled models. A simple study of an ideal harp absorber with different channel cross sections and varied channel distance was presented. It turned out that depending on absorber sheet material and thickness an integrated absorber should feature rather small channel distances and channel cross sections which tend to be circular.

Fraunhofer ISE will continue working on alternative absorber materials and manufacturing technologies with a focus on a better understanding and characterization of the thermal and hydraulic behavior of integrated absorbers in order to finally be able to predict and optimize their overall performance. The fluid dynamics test facility as well as CFD simulations will be the main tools to reach these goals. If new approaches which differ from flat absorbers based on sheet metal are to be investigated, the proposed equations to calculate the efficiency factor F' will not be applicable anymore. In this case three-dimensional multiphysics software and comprehensive experiments will be needed in order to elaborate appropriate modeling tools.

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