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Experimental Evaluation of Volumetric Solar Absorbers – Ceramic Foam vs. an Innovative Rotary Disc Absorber Concept

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Abstract. This paper presents the experimental evaluation of two volumetric solar absorber concepts. It compares the thermal performance of conventional foam absorbers with that of an innovative rotary disc absorber design. The moving absorber showed advantages regarding air temperature stability at the outlet and better operability compared to the foam absorber. Unfortunately, it is shown that the innovative rotary disc design in its present form is not competitive, neither in terms of thermal performance, nor regarding mechanical reliability and robustness. The future task is therefore to optimize the mechanical design regarding thermal fatigue, in order to obtain a robust and durable rotary absorber that can be thoroughly evaluated experimentally.

INTRODUCTION

Solar thermal power, also known as concentrated solar power (CSP) or solar thermal electricity (STE) can be considered as a highly promising technology when it comes to dispatchable and thus grid-friendly supply of renewable electricity. This is due to the possibility of thermal energy storage (TES), the key advantage over other renewable technologies (such as wind or photovoltaic), which enables the decoupling between solar energy collection and electricity production. Given the abundant amount of solar power available for terrestrial solar collectors (≈85 PW) [1], which exceeds the current world's power demand (≈15 TW) several thousand times [1], CSP is a highly promising alternative to conventional fossil-fuel or nuclear technology, setting new standards in terms of environmental impact, sustainability, safety, and thus quality of life.

Unfortunately, currently the cost of electricity generation for CSP (≈14 c€/kWh [2]) is still clearly above conventional technology and other renewables (wind and photovoltaic reach 6 c€/kWh on average [2]). It is thus clear that the cost of electricity generation for CSP has to be decreased in the near future to facilitate its large-scale commercial deployment. On the one hand, the cost reduction will have to be achieved via optimized solar concentrating and receiver systems, considerably reducing manufacturing costs (high volume) and increasing optical/thermal performance. On the other hand, further cost reduction needs to be achieved by introducing advanced power plant configurations and highly efficient power cycles.

A very promising power cycle configuration is of course the combined cycle, which is currently reaching about 60% of thermal efficiency [3] in conventional fossil-fired power generation. It is obvious that a solar powered combined cycle has a big potential regarding efficiency improvement with respect to current state-of-the-art single-

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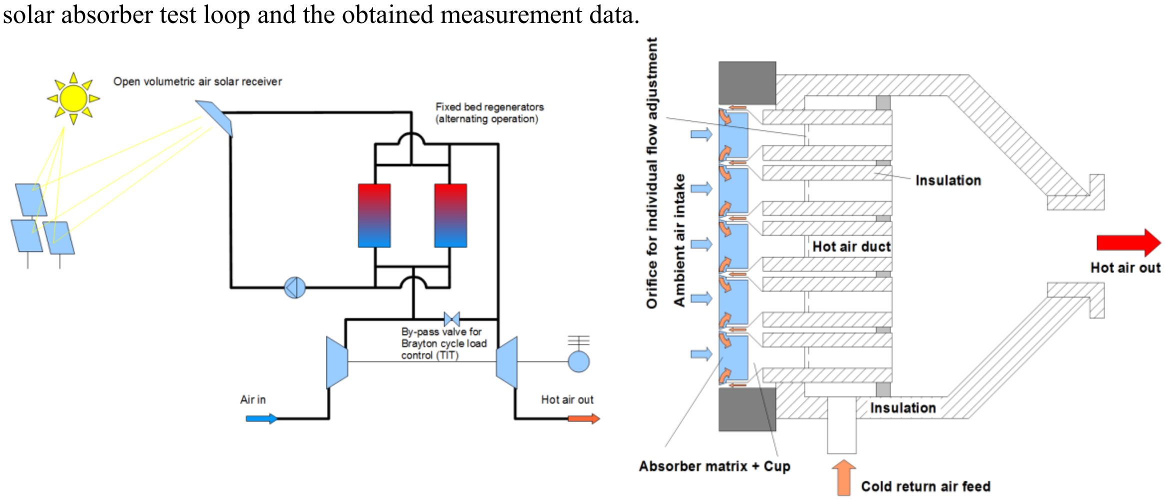
cycle CSP plants that achieve thermal-to-electric conversion efficiencies between 30 and 40% [4] (turbine inlet temperatures below 565 ºC).

Cleary, the key component of a solar powered combined cycle is the solar receiver unit, which has to provide heat to a pressurized air stream coming from the gas turbine's compressor. Several previous research projects have already endeavored to design such a demanding component, which has to operate under very high solar flux (≈ 0.5 - 1 MW/m^2), at high temperatures (> 900 ºC), and in addition at pressures of up to 10 bar [5-13]. However, this very cumbersome task has not been brought to commercial readiness so far for thermodynamically interesting operating temperatures (> 1000 ºC).

In the framework of the CAPTure [14] project, a new solar receiver design approach is being investigated, which decouples the high temperature and the high heat flux part (solar receiver) from the high pressure part (compressed air stream of the Brayton cycle) via an air-air regenerative heat exchanger (see Fig. 1a). The project aims at applying a network of fixed-bed regenerative heat exchangers working in alternating modes (non-pressurized heating period, pressurized cooling period), in this way avoiding the design challenges of a pressurized air receiver. Thus, the open volumetric air receiver technology [15-17] can be applied (see Fig. 1b). It is envisaged to reach 1000 ºC of turbine inlet temperature in purely solar-driven operation, providing receiver thermal efficiencies of 85%.

During the first project phase, the CAPTure project analyzes two promising solar absorber configurations in parallel; (i) a ceramic foam absorber option (porous SSiC - pressureless sintered Silicon Carbide) and (ii) an innovative rotary disc absorber configuration where the disc material is solid SSiC with high density. Then, after the first experimental evaluation of both design options at small scale, the better/more mature configuration will be chosen for the final solar receiver design. It should be emphasized that all relevant criteria, such as the thermal and fluid-dynamic performance parameters as well as manufacturing costs and operability will be taken into account in the decision process.

This paper aims at presenting the experimental procedure and measurement results of the test campaign performed at the solar research facility Plataforma Solar de Almería (PSA) in the south of Spain. The first part of the paper will briefly present the analyzed solar receiver design options. The second part will present the experimental solar absorber test loop and the obtained measurement data.



(a) (b)

FIGURE 1. CAPTure solar powered hot air turbine unit (a); Solar receiver subunit (b)

SOLAR ABSORBER DESIGNS FOR THE CAPTURE OPEN VOLUMETRIC AIR RECEIVER

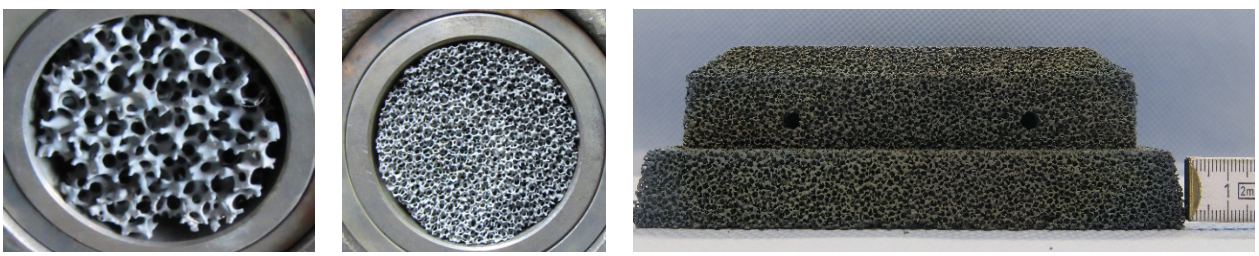
The solar receiver design is based on a modular approach, as suggested by previous research projects [15, 16], where the volumetric air solar receiver unit is typically composed of an array of cups, wherein each cup contains the solar absorber matrix (see Fig. 1b). The modular design is required in order to adjust the mass flow locally (one orifice for each cup) according to the given solar flux map [18]. The aim is to achieve the same air outlet temperature for all cups. Thus, cups with higher incident flux need to have higher mass flow rates than cups with lower incident solar flux. The incident solar flux can be considered homogenous for one cup (which is small enough – 140 mm x 140 mm), however not for the receiver. Given the heliostat aiming point strategy, the flux is typically higher in the center of the receiver, and lower at the boundaries.

Within the CAPTure project, two absorber designs are evaluated in parallel: a conventional absorber based on cellular ceramics (foam) and an innovative rotary disc design. Both designs will be described in the next sections.

The Ceramic Foam Absorber Design

Ceramic foam is a positive image of a polymer foam of chosen pore density [19, 20]. During the first production process, the polymer foam, which is usually polyurethane, is submersed in ceramic slurry. Then, excess slurry is removed from the wetted foam via squeezing and kneading. Next, the wet foam is dried, and then heated to high temperatures, where the polymer base vaporizes or burns and the ceramic sinters [19, 20]. Due to this process, the final shape of the ceramic foam is intrinsically dependent on both the geometry of the polymer foam and the coating thickness of the ceramic slurry. Thus, it makes sense to use already existing characterization standards for polymer foam, since this defines the base geometry. Nevertheless, due to different shrinkage behavior of the ceramics applied, the final cell density of the ceramic foam may differ significantly from its polymer foam base.

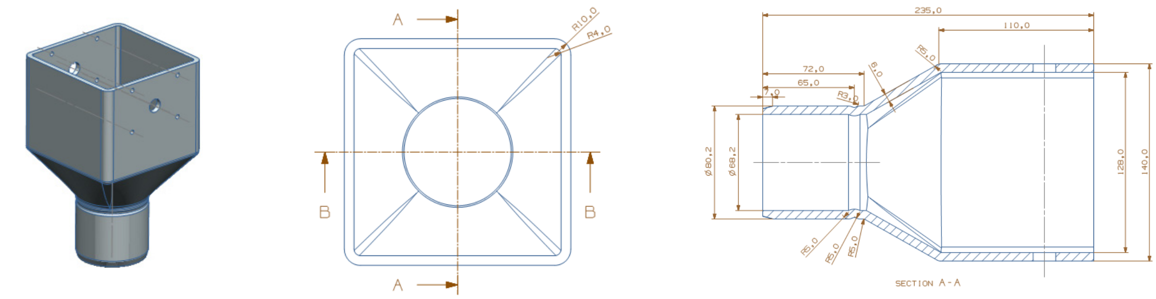
Polymer foams are typically characterized via the parameter cell or pore density according to the norm ASTM D 3576-77, which is based on a mean value of number of cells on a specified linear distance. The unit is thus pores per inch (PPI) [20]. The higher the number is, the more pores or cells exist, i.e. the smaller are the cells. The lower the number is, the fewer pores or cells exist, i.e. the larger are the cells. Polyurethane foam is commercially available between 8 and 90 PPI [20]. Figure 2 shows two porous SSiC (pressure-less sintered Silicon Carbide) foam samples of different cell densities (10 PPI sample (a), 30 PPI sample (b)), placed in a metallic tube for laboratory-scale testing purposes [21].



(a) (b) (c)

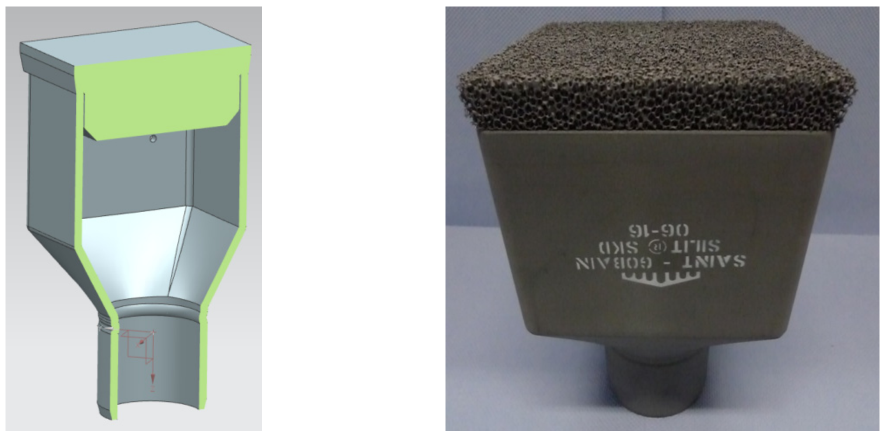
FIGURE 2. SiC foam absorber samples - (a) 10 PPI sample 30 mm diameter, (b) 30 PPI sample 30 mm diameter, (c) 30 PPI sample for cup-level testing – 50 mm thickness and 10x10 mm frustum

The present paper will deal with the test results on cup level. One cup is the modular subunit of the solar receiver (see Fig. 1b). In order to allow the testing of both absorber types with one cup design, a universal cup design has been defined. Due to the requirements of the rotary disc design (discussed in the next section), the cup needs to be deeper (110 mm – see Fig. 3c), than typically necessary for foam or honeycomb receivers. For the ceramic foam absorber option, only the width of the cup (140 mm) and the wall thickness (6 mm) is relevant. The foam absorber has been designed such that it covers the complete width of the cup (protection of the cup's front edges against direct incident solar flux) and provides sufficient total thickness in order to guarantee thermal equilibrium between air and solid at the rear of the absorber (see Fig. 4). In addition to this, a frustum shape at the rear of the absorber may homogenize the air speed over the entire cross section [22], leading to a homogenous absorber front temperature and thus better efficiency. The foam parameters as cell density and porosity have been optimized for thermal efficiency, with the result that the porosity should be as high as possible and the cell density should be around 30 PPI (flat optimum) [21]. The planned experiments are expected to confirm these findings. Table 1 shows the tested configurations, varying total absorber thickness, rear shape (frustum or flat) as well as cell density and porosity.

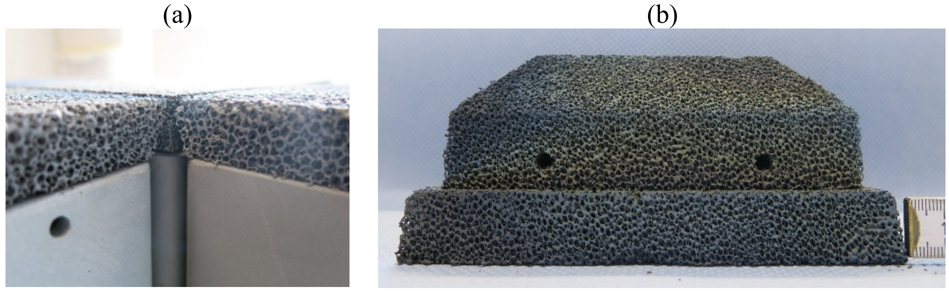


(a) (b) (c)

FIGURE 3. Universal cup design - (a) 3-D view of cup, (b) Front view of cup (c) Section A-A

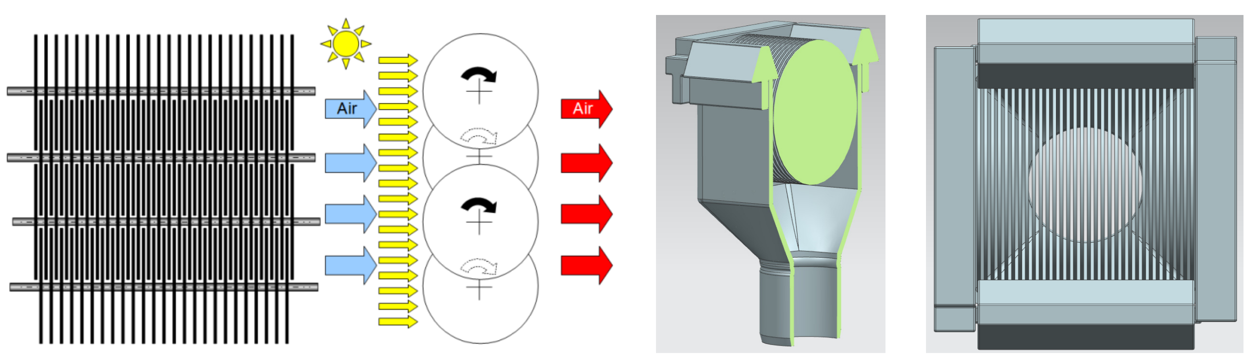


(a) (b)



(c) (d)

FIGURE 4. Universal cup and foam absorber - (a) 3-D view of cup and foam absorber, (b) Picture of the real cup assembly, (c) Detailed view of cup foam arrangement, (d) 30 PPI sample for cup-level testing – 65 mm thickness and 20x20 mm frustum



(a) (b) (c)

FIGURE 5. Rotary disc absorber concept (a), Rotary disc prototype 3-D section (b), Rotary disc prototype front view (c)

TABLE 1. SSiC foam absorber configurations tested.

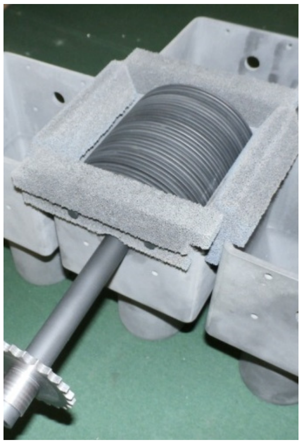
Sample label	Nominal cell density (PPI)	Rear shape - Frustum (mm x mm)	Absorber thickness (mm)	Total porosity (%)
RF-130	30	10 x 10	50	90.1
RF-131	30	10 x 10	50	91.9
RF-126	15	10 x 10	50	90.2
RF-127	15	10 x 10	50	92.3
RF-134	30	10 x 10	65	89.4
RF-135	30	10 x 10	65	91.6
RF-138	30	10 x 10	65	90.3
RF-139	30	10 x 10	65	92.5
RF-128	15	10 x 10	65	90.3
RF-129	15	10 x 10	65	91.2
RF-132	30	Flat	50	89.7
RF-133	30	Flat	50	92.1
RF-137	30	20 x 20	65	88.8
RF-136	30	20 x 20	65	91.5

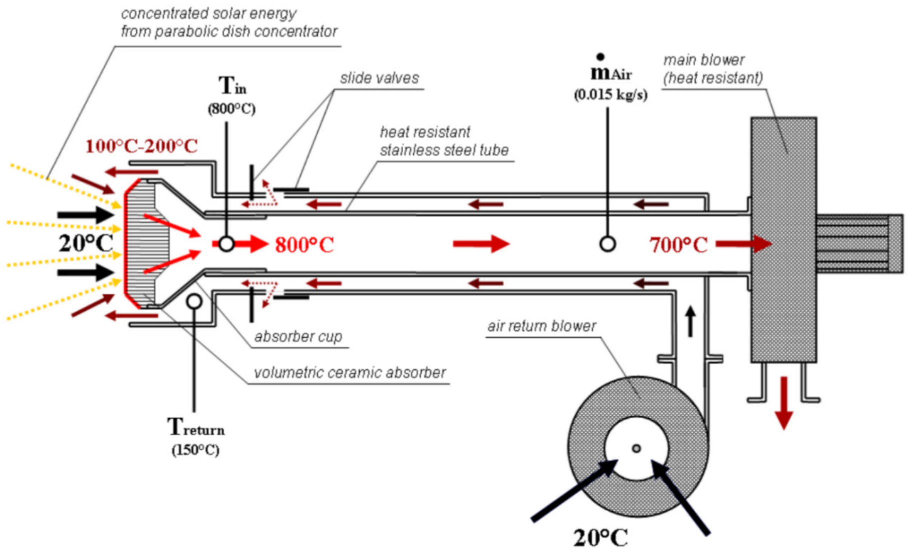
The Innovative Rotary Disc Absorber Design

The second absorber design option is an innovative approach, introducing a rotary movement of the absorber from the irradiated domain towards the rear of the absorber (air outlet), and vice versa. The idea is to introduce a movement to the absorbing structure, so that the hot parts can be moved to the backside while transferring the heat to the air, reducing thermal losses to the environment. A conceptual scheme of the idea is shown in Fig. 5a. It is expected from the rotation that the mean absorber front temperature is reduced, compared to a fixed cellular ceramic absorber. This configuration should also enable higher solar flux densities without overheating the absorber matrix, i.e. avoiding the generation of hot spots or flow instabilities. However, the conventional so-called "volumetric effect" cannot be achieved as the temperature profile of the discs is axially symmetric (only a function of distance from the center) and is typically higher for the disc edges than for the center regions. Note that the speed of rotation needs to be sufficiently high to avoid thermal fatigue, whilst respecting mechanical stability limits (avoiding vibrations and resonance).

Related to the heat transfer area of the absorber, an important factor is the ratio between the area in contact with the fluid and the aperture area of one representative channel. At previous projects, where honeycomb absorbers were used, a typical value for this ratio is 100. Hence the conventional honeycomb has 100 m^2 of heat transfer area for each square meter of channel inlet cross-sectional area. We define this area ratio as contact-to-inlet area ratio γ. In order to achieve similar values of γ for the rotary disc design, the diameter of the discs has been chosen as big as possible (126 mm) when keeping the modular receiver concept based on cups (see Fig. 3), and the thickness of the discs and the spacing in between has been set to 2 mm. This also keeps the design reasonable simple, i.e. avoiding multiple shafts per cup.

Figure 5b shows a 3-D section view of the rotary disc receiver prototype on cup level. Figure 5c displays the front view of the absorbing plane. It is important to note that ceramic foam pieces are placed at the borders of the absorber in order to protect the cup from direct solar flux and improve air recirculation. Figure 6a displays the final manufactured rotary disc receiver prototype. The material of the discs and the driving shaft is solid SSiC with high density. The thermal properties as conductivity and specific heat capacity at a temperature of 1000 ºC are 40 W/(m K) and 1100 J/(kg K), respectively. The solar absorptivity is 0.85, the thermal emittance is 0.8.





(a) (b)

FIGURE 6. Manufactured rotary disc receiver prototype (a); Test loop configuration at PSA [23] (b)

THE EXPERIMENTAL LOOP INSTALLED AT A PARABOLIC DISH CONCENTRATOR

The above described absorbers have been tested at cup-level at an experimental air loop (see Fig. 6b) installed at a parabolic dish concentrator with a dish diameter of 8.5 m and focal distance of 4.1 m. In order to achieve a mean solar flux of about 800 kW/m^2 and a considerable reduced peak flux of around 1600 kW/m^2, the solar absorber surface has been placed at distance of 1.032 m away from the focal plane. The experimental air loop (Fig. 6b) consists of an air duct, where the receiver cup is placed at the entrance. At the rear of the duct, a heat resistant blower induces the air flow through the solar absorber and the air duct, releasing hot air to the ambient. In order to simulate air recirculation at the cup borders, a coaxial tube is placed with respect to the main receiver duct and a second blower forces a counter-current air stream through the annulus. The air stream in the annulus is heated from ambient conditions to a certain recirculation temperature, which is a function of absorber outlet temperature, air mass flow and slide valve positions (see Fig. 6b). The air temperature is measured with type K thermocouples located at the transversal section corresponding to the absorber outlet. The air flow through the main absorber duct is measured with a differential pressure flow meter (orifice plate) provided by Intra-Automation GmbH.

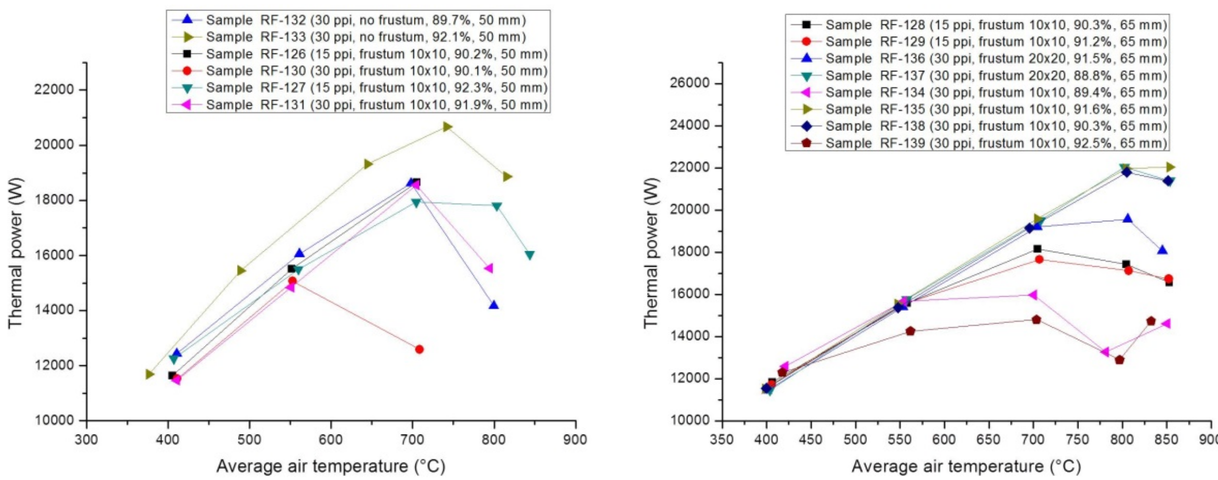
EXPERIMENTAL RESULTS

Figure 7 and Fig. 8 display the compiled measurement results from the above described experimental loop. The thermal power gained by the air stream (mass flow times enthalpy difference) is plotted over the absorber outlet temperature achieved at steady-state conditions. It has been decided to plot the thermal power instead of absorber efficiency, because unfortunately, the solar flux measurement could not be made accurately enough due to the continuous solar tracking of the system and variable weather conditions which affect the irradiance distribution during the test.

Figure 7a shows the results of the foam absorbers with 50 mm thickness (see Table 1). Figure 7b presents the results of the foam absorbers with 65 mm thickness. Comparing foams of same geometry (frustum or not) and same PPI value, shows that foams with higher porosity perform better. Foams with frustum seem to have a better thermal behavior than flat foams. Furthermore, 30 PPI foams perform better than 15 PPI foams. Note that these conclusions are based on the important additional parameter of direct normal irradiance (DNI) during the experiments, which is however not shown in the plots. It is clear that only foams can be compared that were tested under very similar DNI boundary conditions. In addition, the obtained experimental results have high uncertainty, producing mostly qualitative results, for comparative purposes only. Unfortunately, this is a mayor disadvantage of the experimental setup, due to the lack of resources at the testing facility.

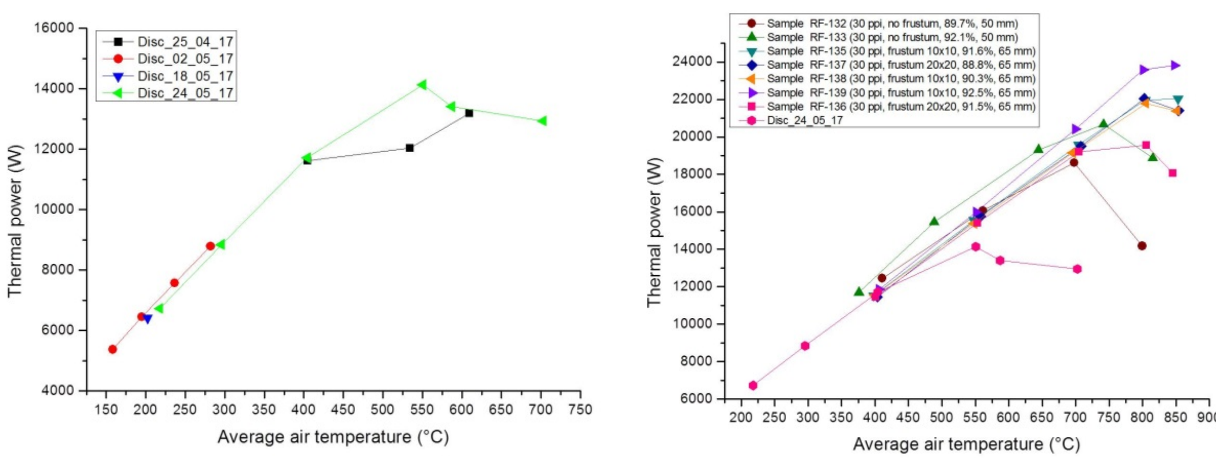
Figure 8a shows the measurement results of the rotary disc absorber concept. Here, no design or disc parameter variations are made. Unfortunately, due to several mechanical problems of the drive train and breakage of discs during the experiments, only four usable experimental data sets could be generated. Only two experiments could be completed, reaching relevant absorber outlet temperatures of up to 700 ºC at steady-state conditions. Higher temperatures could not be reached due to failure of either the discs or the drive train. When comparing the peformance of the rotary disc concept with the conventional foam absorber, the thermal performance of the rotary disc concept is clearly worse. Nevertheless, due to the movement of the rotary disc absorber, the outlet temperature is more stable during operation and it shows in principle better operability under variable weather conditions.

Figure 8b shows a summary of the best absorber samples out of the 3 previously described figures. Foam samples RF-139 and RF-135 are the best performing ones (30 PPI and high porosity).



(a) (b)

FIGURE 7. Thermal power - outlet temperature of 50 mm foams (a); Thermal power - outlet temperature of 65 mm foams (b)



(a) (b)

FIGURE 8. Thermal power - outlet temperature of rotary disc absorbers (a); Thermal power - outlet temperature of best foam and disc absorbers (b)

CONCLUSIONS AND OUTLOOK

This paper presents the experimental evaluation of two volumetric solar absorber concepts. It compares the thermal performance of conventional foam absorbers with that of an innovative rotary disc absorber design. Unfortunately, it is shown that the innovative rotary disc design in its present form is not competitive, neither in terms of thermal performance, nor regarding mechanical reliability and robustness. Nevertheless, the moving absorber showed advantages regarding air temperature stability at the outlet and better operability under variable weather conditions compared to the foam absorber. The future task is therefore to optimize the mechanical design regarding thermal fatigue, in order to obtain a robust and durable rotary disc absorber that can be thoroughly evaluated experimentally. Regarding the performance of the foam absorbers, it can be concluded that higher porosity performs better, and that 30 PPI foams perform better than 15 PPI foams under comparable boundary conditions. The influence of the remaining parameters could not be confirmed due to experimental uncertainties.

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