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A novel CSP receiver based on airlight energy technology - optimization of the thermal insulation system by means of CFD analysis

S.A. Zavattoni\textsuperscript{a}, A. Gaetano\textsuperscript{a}, D. Montorfano\textsuperscript{a}, M.C. Barbato\textsuperscript{a,\ast}, G. Ambrosetti\textsuperscript{b}, A. Pedretti\textsuperscript{b}.

\textsuperscript{a}Department of Innovative Technologies, SUPSI, Manno 6928, Switzerland.
\textsuperscript{b}Airlight Energy Manufacturing SA, Biasca 6710, Switzerland.

Abstract

The thermal behavior of the novel CSP receiver based on Airlight Energy technology was analyzed by means of accurate 2D steady-state computational fluid dynamics (CFD) simulations. Afterwards, its thermal insulation design was numerically optimized with the aim of minimizing the heat losses. Energy and radiation transport equations were numerically solved, using the finite-volume method approach, with Fluent code from ANSYS.

In this innovative receiver design, air is used as heat transfer fluid (HTF) allowing to go beyond the operating temperature limits of conventional HTFs. However, air receivers need a larger heat transfer area and flow cross-section compared to the most common oil and molten salts receivers. Thus, the insulation becomes technically challenging. Due to a large thermal capacity, the use of conventional insulating material is not the best solution for air-based receivers. Instead, thanks to the advantages offered, thermal radiation shields were selected as main receiver insulation system.

The analysis of the CFD simulations results drove the various modifications of the thermal insulation design; hence, a total of three versions were studied achieving, at the end, the final optimized solution which is implemented in the first full-scale 3.9 MW\textsubscript{th} pilot plant under construction in Ait Baha (Morocco).

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\ast Corresponding author. Tel.: +41 058 666 6639.
E-mail address: maurizio.barbato@supsi.ch

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1. Introduction

With the aim of overcoming the limitations of conventional parabolic trough in concentrating solar power (CSP) systems for large-scale solar electricity production, Airlight Energy has developed a completely new CSP solution based on an inflated mirror trough collector with precast fiber-reinforced concrete sustaining structure [1].

A novel approach has also been used for the receiver design, which includes an additional concentration stage, in turn coupled to a string of cavity receivers. Air is used as heat transfer fluid (HTF), which, besides being inexpensive and environmentally friendly, is optimally suited for high temperature operation, above 650 °C, and enables the use of a low-cost rock bed thermal energy storage (TES) system [2].

The use of air as HTF offers various advantages; however, air receivers, in comparison with the most common oil and molten salts ones, need a larger heat transfer area and flow cross-section. For these larger components the insulation becomes cumbersome and technically challenging.

The deployment of a conventional micro-porous insulating material is not a favorable solution for air-based receivers because, due to its relevant thermal capacity, it would lead to longer transient conditions.

Affordability and low heat capacity are among the main advantages that a thermal radiation shields system may offer as thermal insulation technology.

After the first numerical modeling and sensitivity analysis, performed to find the optimal design in terms of shields geometry, number and pitch to reduce the heat losses [3], and thanks to a satisfactory experimental proof of concept, performed with a 9 m ducts with a 0.4 m internal pipe diameter, the thermal radiation shields were selected as main receiver thermal insulation system for the first full-scale 3.9 MWth pilot plant under construction in Ait Baha (Morocco).

2. Working principle of the novel CSP receiver based on Airlight Energy Technology

A schematic of the novel receiver design is given in Fig. 1. Solar energy is collected and focused by the primary optics, a multi-layer stack of aluminized polymer membranes, towards the receiver. A secondary hyperbolic concentrator allows to avoid potential dispersion of energy directing the solar radiation into the receiving cavities through a glass window. The glass external surface is treated with a broadband anti-reflective (AR) coating to minimize reflection losses. Furthermore, the glass window, being interposed between the secondary optics and the cavity array, allows also to reduce convective losses.

The overall geometric concentration ratio obtained with this system is in the order of 97 Suns. The core of this innovative receiver is well represented by the helically coiled heat exchangers (r.h.s. of Fig. 1), which constitute the receiving cavities. The high density solar energy hits the inner surfaces of the cavities and is then gathered, in the form of thermal energy, by the heat transfer fluid (air) which is fed into the receiver from the bottom, by two feeding pipes, at low temperature (120°C) and pressure slightly above the atmospheric. Air at high temperature (up to 650°C), coming from all the cavities, is then collected into a single 450 mm diameter run-back pipe and used to feed both the steam generators and the high temperature rock-bed TES system.

The whole receiver and the inflated mirrors are separated from the external environment by a thin ethylene tetrafluoroethylene (ETFE) top-sheet widely used in building applications. Beyond the intrinsic advantages provided such as remarkable corrosion resistance, temperature resistance, self-cleaning and good optical properties (total transmittance for solar radiation of 92%), the ETFE layer allows to minimize the mirrors reflecting reduction, due to dust deposition, creating a clean and controlled environment.

In the Ait Baha (Morocco) pilot plant, a total of three full-scale collectors will be built. The overall collector length is 211.68 m, obtained assembling 36 standard modules which bring the total mirror surface to 2,053 m².
3. CFD modeling and numerical details

The effectiveness of the different thermal insulation designs was studied by means of 2D steady-state CFD simulations with the hypothesis of dominant heat transfer contribution occurring in the radial direction of the receiver cross-section and only a negligible fraction in the longitudinal one.

The internal heat transfer mechanisms, accounted for by the CFD simulations, are: conduction and radiation, the latter solved with the Discrete Ordinates (DO) model [4, 5]. Due to the geometric limitations of these simplified models, buoyancy-driven effects were not considered. This assumption is justified by the fact that convective heat transfer is negligible into both the internal part of the cavities and the thermal radiation shield separation volumes. For the latter, as reported in [3], the distance between shields was kept small enough to prevent the onset of intense convective motions.

Heat losses towards the external environment are given by means of convective and radiative heat transfer. Wind does not directly affect the convective heat transfer coefficient (HTC) of the receiver external surfaces since the ETFE top-sheet protects them from the external environment. Based upon preliminary computations, the temperature of this controlled environment was considered equal to 343 K. As far as concerning its optical properties [6], albeit ETFE is not completely opaque to long-wave radiation, the fraction of energy transmitted into this range is very small and therefore it can be safely neglected. With this assumption, the temperature used to compute the radiative heat transfer was set equal to the ETFE equilibrium temperature (343 K) instead of the sky temperature (278 K).

The CFD simulations were performed on a Linux Cluster with AMD multicore processors. All model equations were solved with second order upwind numerical schemes [7]. Convergence was considered achieved when energy and radiation intensity residuals were below $10^{-10}$ and $10^{-6}$ respectively.

3.1. Physical properties of the materials involved

Thermo-physical properties of air (specific heat, thermal conductivity and viscosity) were assigned as piecewise linear interpolations of tabulated data available in the literature [8]. The numerical solution of governing equations was performed with the “pressure-based” approach, which assumes that mass density depends on temperature and on a fixed pressure reference value [9].

Instead, thermal properties of solid materials (rock-wool, polyurethane foam, ceramic, steel and aluminum), were obtained from the manufacturers data sheets and assigned as constant values. Thermal conductivities of Microtherm® and Borofloat® glass were modeled as piecewise linear profile as indicated by the suppliers [10, 11].
4. First receiver thermal insulation design

The thermal insulation of the receiver was firstly designed with a combination of micro-porous insulating materials (points 1, 3 and 4 on the l.h.s. of Fig. 2) and a total of 9 radiation shields with a pitch of 18 mm (point 2 on the l.h.s. of Fig. 2).

The detailed description of the insulating materials adopted is reported hereafter. The reference for the numeration is the l.h.s. of Fig. 2.

1. Polyurethane foam to isolate and to protect the radiation shields from the external environment;

2. Aluminum thermal radiation shields;

3. Microtherm® (overstitched panel and Super G) to insulate the hottest regions of the receiver. The overstitched panel is used to cover the run-back pipe avoiding thus to observe a too high temperature of the most internal radiation shield. Instead, the Super G is used as housing for the receiving cavities.

4. Rockwool to insulate the bottom part of the receiver, as well as the feeding pipes, from the external environment.

However, the main receiver thermal insulation system, used to insulate almost 3/4 of the overall receiver cross-section, is represented by the thermal radiation shields. The latter are made of thin reflective aluminum foils leading to a superficial emissivity between 0.07 and 0.1. Calcium-silicate spacers are used as framework to keep the radiation shields in place.

Because of its high relevance in enhancing the harvesting of incoming solar radiation, special attention shall be spent for the secondary optics. The external surfaces of this high-efficiency optics are coated with a thin aluminum film maximizing their reflectivity behavior. In order to maintain the favorable optics characteristics, the coating cannot withstand temperatures higher than 390 K; therefore, a water-based cooling system is provided to maintain the aluminum structure, of the secondary optics, into a safety temperature range.
4.1. CFD modeling details and boundary conditions

The computational domain, of the first thermal insulation design, was built exploiting the symmetric characteristic of the receiver; hence, as reported on the r.h.s. of Fig. 2, only half of the cross section was discretized with a grid of almost 310,000 quadrilateral cells.

A list of the main boundary conditions applied to the model, as well as the modeling assumptions, are reported hereafter to provide a better understanding on the strategy followed to perform the CFD simulations under the worst-case conditions:

- A fixed temperature boundary condition was assigned to the external surfaces of: the run-back pipe (923.15 K), the feeding pipes (393.15 K) and the receiving cavities. The latter were divided into four smaller segments describing in a more realistic and detailed way the effect of temperature variation into the cavities.

- Heat losses towards the external environment are given by convective heat transfer, with a free-stream temperature of 343 K and a conservative convective HTC of 8 W/(m²·K), and radiative heat transfer, with an external radiation temperature of 343 K.

- The water-cooling circuit was modeled with an equivalent convective heat-sink wherein the free-stream temperature and the convective HTC were properly computed assuming a constant surface heat flux condition [8].

- A total absorbed heat flux of 21.2 W/m was applied onto the internal surfaces of the secondary optics. The heat flux was computed according to the optical characteristics of the primary and the secondary concentrators with a reference DNI of 1,000 W/m².

- Due to the small thickness in comparison to the overall length, thermal shields were modeled as thickness-less foils with emissivity values ranging from 0.07 to 0.1 on both the sides.

4.2. CFD simulations results

The overall heat losses of the first receiver thermal insulation design, integrated upon the whole receiver cross-section, are about 2,574 W/m; 46.2% of which are the heat losses towards the external environment. The remaining 53.8% represents the power removed by the water-cooling circuit. However, it is worth to note that, besides the power removed by the cooling circuit, the other major contribution of heat losses is due to the radiative/convective heat transfer from the external glass surface of the cavities.

As far as concerning the thermal shields, all of them are below the threshold temperature value, given by the limit of the material, of 820 K. The highest shields temperature, of about 730.2 K, is observable in correspondence of the first and most internal one.

Secondary optics shows an average temperature of about 480 K, well above the fixed limit. The reasons are manifold: (1) mechanical contact between glass external surface and secondary optics structure; (2) the presence of thin structural steel plates which, due to the relatively high steel thermal conductivity, establish a favourable way for heat transfer from the hottest internal regions towards the colder external surfaces; (3) the presence of other thin steel plates directly connected to the secondary optics and the internal region of the cavity.

Figure 3 depicts the CFD simulations results obtained in terms of temperature distribution of the receiver cross-section with a magnification of the cavity/secondary optics region.
4.3. Further numerical investigation

Accounting for the limits aforementioned, a second CFD simulation, based upon the first receiver thermal insulation design but with some changes on the components properties, was performed. The result obtained confirmed the possibility of reducing the overall heat losses, and the secondary optics temperature, acting mainly on the separation of the mechanical contact between secondary optics and external glass surface and the removal of the steel plates which directly link the secondary optics and the internal region of the cavity.

The combination of these modifications allowed to reduce the overall heat losses to a total of 2,182.5 W/m (87.7% heat losses towards external environment and 12.3% power removed by the cooling cycle) and the average temperature of the secondary optics to 357 K. Figure 4 shows the temperature distribution in the case of modified design. It has however to be intended as ideal case because the thermal contact between glass external surface and secondary optics was replaced with an adiabatic condition and the structural steel plates, linking the internal region of the cavities with the secondary optics, were replaced with the same neighbor insulating material.
5. Second receiver thermal insulation design

Thanks to the information obtained with the first analysis, a modified version of the receiver thermal insulation was proposed. The main changes were:

- increased number of radiation shields and reduced pitch, from the original 9 to the actual 17 and from 18 mm to 10 mm respectively, keeping unchanged the receiver external diameter;
- reduced number and cross-section area of the structural steel plates surrounding the receiving cavities;
- separation of the secondary optics from the external glass surface;
- increased cross-section area of the secondary optics water-cooling system;
- different arrangement of insulating materials;
- employment of a cheaper and stiffer aluminum-extruded framework for the secondary optics.

Figure 5 shows a schematic of the second receiver thermal insulation design and a magnification of the relative computational domain.

![Schematic of the second receiver thermal insulation design](image1)

Fig. 5. schematic of the second receiver thermal insulation design (l.h.s) and relative computational domain (r.h.s.).

5.1. CFD simulations results

Since most of the modifications were focused in the receiving cavities and secondary optics region, as depicted on the r.h.s. of Fig. 5, the CFD analysis was limited to the most critical region and hence, only a quarter of the whole receiver cross-section was considered as computational domain. The latter was further improved including the air region in front of the secondary optics allowing to consider the effect of the mutual interaction of the reflecting surfaces for radiative heat transfer. The grid size was of about 476,000 quadrilateral elements.

A symmetry boundary condition was used either for the zone in between the two feeding pipes and for the thermal shields. For the latter, the strong assumption can be justified by the fact that, as observed from the previous simulations, far enough from the cavities, the heat flux through the shields is purely radial.

Figure 6 shows the CFD simulation result in terms of temperature distribution. The enhanced discretization of the computational domain allowed to observe the heat transfer between the external surface of the glass and the secondary optics. In this case, the average temperature of the latter is about 350 K with a peak of 364 K in the region close to the external glass surface.
As far as concerning the radiation shields, for this new design an average temperature of 900 K, well beyond the limit of the material, was observed for the most internal shields. In this case the shields emissivity was set equal to 0.1 for both the sides.

The power removed by the cooling circuit is comparable with that of the previous test case. This indicates that other improvements, such as wider separation between external glass surface and secondary optics and better insulation of the internal components, still need to be increased. A CFD investigation, in which the secondary optics surfaces were considered adiabatic in all the regions in contact with steel plates and internal structure, was performed to evaluate the possible reduction of power removed by the cooling circuit. An ideal 55% reduction was envisaged showing the relevance of the heat flux coming from the structure itself.

Fig. 6. temperature distribution of the second receiver thermal insulation design. Temperature values are in K.

Two other modifications were suggested analyzing the temperature distribution:

- The fluid region above the cavities shall be separated from the one between Microtherm®, protecting the run-back pipe, and the innermost radiation shield in order to avoid the onset of convective motions which would substantially affect the shields temperature and the overall heat losses.
- The further reduction, or complete removal, of structural steel plates which connect the hot internal receiver regions to the external surfaces.

6. Third receiver thermal insulation design

A third and last, improved receiver thermal insulation design was then proposed. The main variations with respect to the previous solutions are:

- complete removal of the structural steel plates connected to the external part of the receiver;
- larger separation of the secondary optics from the external glass surface;
- reduced surface of the secondary optics facing the glass window;
- improved geometry of the secondary optics cooling circuit;
- different arrangement of insulating materials between cavity and secondary optics;
- closing of the fluid regions above the cavities.
Figure 7 shows a schematic of the third version of the receiver thermal insulation design and a magnification of the relative computational domain.

![Fig. 7. schematic of the third receiver thermal insulation design (l.h.s) and magnification of the relative computational domain (r.h.s.).](image)

6.1. CFD simulations results

The third geometry was analyzed exploiting the symmetry of the receiver cross-section; therefore, only half of the latter was considered as computational domain. The domain was discretized with a mapped grid of almost 490,000 quadrilateral cells.

Figure 8 shows the result of the CFD simulation, in terms of temperature distribution, of the new receiver thermal insulation design with a magnification of the cavity/secondary optics region.

![Fig. 8. temperature distribution of the third thermal insulation design. Temperature values are in K.](image)

It is interesting to note that the removal of the structural steel plates, connecting the internal regions to the external surfaces, allowed to prevent a favorable ways for heat transfer; moreover, the closing of the fluid zones
above the cavity led to a noticeable temperature reduction, to a maximum of 820 K, of the most internal shield. Since this value is close to the limit of the material, it was decided to change the material of the three most internal shields, from the original aluminum to steel, in order to avoid the risk of failure.

The total heat losses of this last thermal insulation design is 2,340.7 W/m; 10.2% of which represent the total power per unit length lost through the radiation shields. The largest contribution, of about 56.6%, is given by the heat losses by radiation and convection from the external surface of both the glass window and the secondary optics structure. The latter showed a maximum temperature of about 355 K which can be easily sustained by the aluminum coating without losing its high-reflectivity characteristics. For this configuration, the total power removed by the water-cooling circuit is about 780 W/m (representing the 33.2% of the total heat losses).

7. Conclusions

The receiver thermal insulation design has been investigated, and optimized, by means of accurate 2D steady-state CFD simulations. The analysis of the results obtained drove the various modifications, in the thermal insulation design, aimed at minimizing the heat losses. Hence, a total of three different versions have been studied achieving, with the final optimized design, a reduction of the overall heat losses of 9% (about 233 W/m) with respect to the first solution.

Another noticeable result, achieved through the optimization of the receiver thermal insulation system, is relative to the secondary optics. The need of a water-cooling system is driven by the fact that the reflective aluminum coating, applied onto the external surfaces, cannot withstand temperatures higher than 390 K. Analyzing the CFD simulations results it was possible to observe, with the first design, a too high secondary optics temperature that would have led to the component failure. The examination of the receiver structural design allowed to obtain, with the optimized version, a noticeable reduction in terms of temperature and power removed by the water: from the original average temperature of 480 K and 1,385 W/m removed by the cooling circuit to the optimized 355 K and 780 W/m respectively.

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