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Proposal and Sizing of a Molten Salt-to-sCO2 Heat Exchanger in Supercritical Solar Thermal Power Plants

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Abstract. Solar Thermal Power Plants (STPPs), based on supercritical CO_2 (sCO_2) cycles, seem to be a promising alternative to increase the global solar-to-electric efficiency. The most conventional scheme for this technology is a molten salt (MS) central receiver, working at high temperature (above 700ºC), coupled to the sCO_2 cycle. For this scheme it is proposed a new design of the source heat exchanger that transfers the thermal energy from the molten salt to the CO_2: the Compact Honeycomb Heat Exchanger (CHHE), in which the molten salt goes through a larger circular duct that is surrounded by 6 smaller trapezoidal ducts, through which the sCO_2 circulates. This paper is focused in the thermal model of this new heat exchanger, and a thermo-economic optimization for a selected supercritical STPP.

INTRODUCTION

The Solar Power Gen3 Demonstration Roadmap has identified three potential layouts for the next generation of STPPs, based on the working fluid in the central receiver: molten salts, falling particles or gas [1]. All of these layouts are based on a supercritical cycle coupled to the solar field, so supercritical cycles seem to be a way to increasing the global efficiency of STPPs.

This work is focused on the molten salt path, particularly, in a molten salt central receiver system coupled to a supercritical cycle. One of the main challenges of this technology is the Heat Exchanger (HX) between the solar field and the supercritical cycle [1]. This heat exchanger must be compact enough to enhance the heat transfer of the supercritical phase; but, at the same time, the channel for the molten salt must be larger enough to avoid plugging issues.

The most conventional HX design is the Shell and Tube Heat Exchanger (STHX). Nevertheless, the STHX performance is worse as the tube thickness is greater due to the higher pressure; besides, it is probably that the molten salts through the shell can be kept retained in the baffles and interstices, also affecting the performance [2]. One solution to these problems is located the thermal energy source downstream the turbine, in the low-pressure side (85 bar), as in the supercritical cycle presented in [3].

Another design for this HX is a Printed Circuit Heat Exchanger (PCHE). Although this HX is the most proper for the supercritical phase, because it can withstand large pressure differences [4], the small dimensions of its channels (around 2 mm of diameter) can yield to plugging problems in the molten salt.

To minimize these problems, this work presents a design for the MS-to-sCO_2 HX. This design is based on a larger circular duct for the MS, to avoid the clogging, surrounded by six smaller trapezoidal ducts for the sCO_2. The repetition of this thermal unit in the plane yields to a honeycomb-like appearance, shown in Fig. 1. Because of that, this design is named Compact Honeycomb Heat Exchanger (CHHE).

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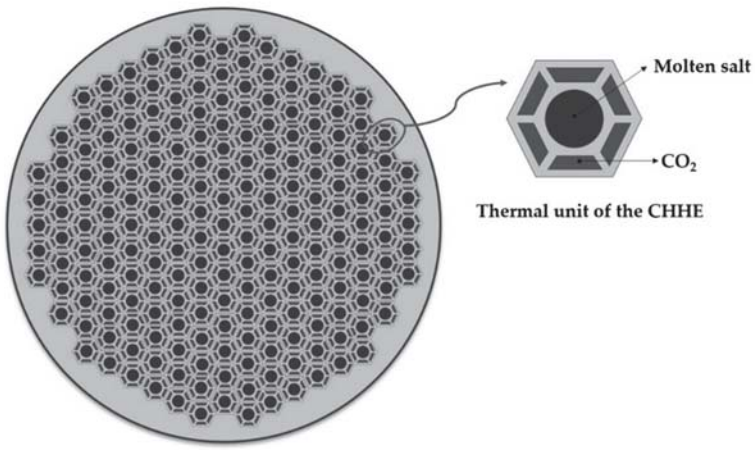


FIGURE 1. Cross section of t he Compact Honeycomb Heat Exchanger and thermal unit (Source: [5])

This CHHE has been already described in [5], where an optimization of the working conditions of this HX has been accomplished, by means of an exergy destruction minimization analysis. In this work, the performance conditions of the CHHE are fixed, as it is coupled to a supercritical recompression cycle, working at nominal conditions. Based on that hypothesis, the work has focused on the thermo-economic optimization of the geometric parameters and sizing of the HX.

This CHHE concept has been proposed in [6] as a compact shell and tube heat exchanger employed in aerospace applications, for lower thermal duties than those reported in this work. As said in [6], with a suitable manifolding, it can be achieved a counterflow configuration between the inner hexagonal "tubes" and the six rectangular channels surrounding each "tube".

THE COMPACT HONEYCOMB HEAT EXCHANGER

As said in the introduction, the proposed Compact Honeycomb Heat Exchanger consists of a larger circular duct, through which the molten salt circulates, surrounded by six smaller trapezoidal ducts for the supercritical CO_2. Dimensions of these channels will be optimized in the next section, for a particular STPP layout. Figure 2 shows the thermal unit of a CHHE, in which the molten salt duct is equal to half an inch (12.7 mm), and the trapezoidal channel for the CO_2 has been set to 5.7 mm. It will be demonstrated in the thermo-economic study that this geometry minimizes the investment cost.

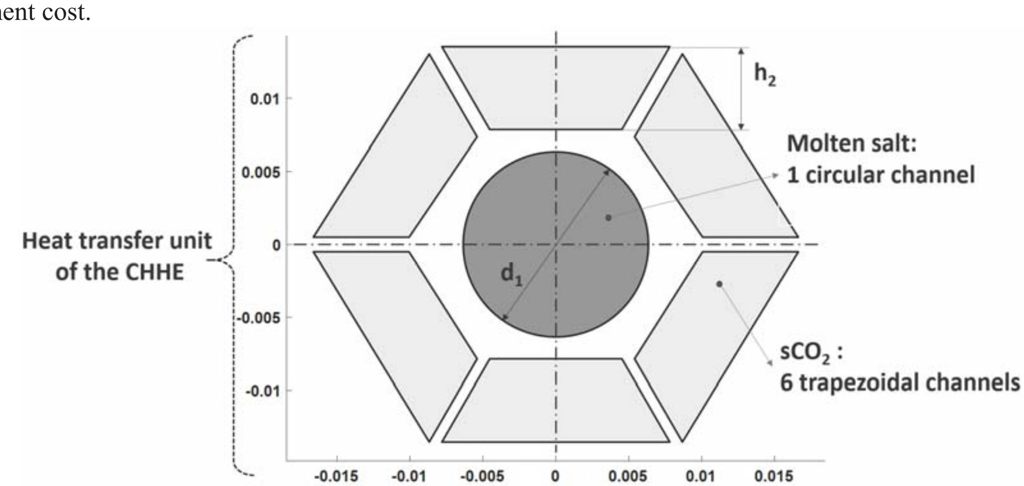


FIGURE 2. Dimensions of the thermal unit of the CHHE selected (scale in mm)

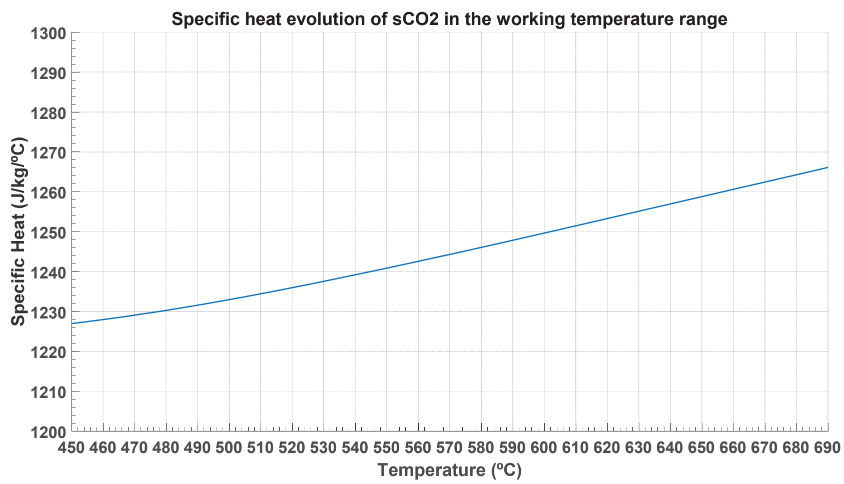
Previous to the thermal model, it was necessary to fix several parameters in the CHHE: the molten salt used, and the expressions or databases for determining the thermal properties of both fluids; the material of the heat exchanger; the working conditions in the supercritical STPP; and the minimum thickness between the MS and the CO_2, in order to guarantee mechanical resistance to pressure difference.

Regarding the molten salt, a ternary chloride salt has been selected as the heat transfer fluid in the solar field, as it exhibits a low melting point and a high thermal decompositions temperature, above 800ºC [7]. Table 1 summarizes the main thermal properties of this salt [3].

TABLE 1. Thermal properties of the ternary chloride molten salt $MgCl_2$/NaCl/KCl (Source: [3])

Thermal Property	Correlation
Specific heat (J/kg/ºC)	$c_p = 1180$
Density (kg/m3)	$\rho = 1899.3 - 0.43 \cdot T(ºC)$
Thermal conductivity (W/m/ºC)	$k = 0.5423 - 0.0002 \cdot T(ºC)$
Dynamic viscosity (Pa·s)	$\mu = 8.25 \cdot 10^{-6} \cdot \exp\left({}^{11874.71735}/_{1350.84595 + T(ºC)} \right)$

The CO_2 thermodynamic properties have been obtained from NIST database [8]. Although the thermal conditions of the CO_2 are supercritical, it is important to note that the thermal properties follow a linear tendency in a relatively small range, as the CO_2 is a very high temperature and far away from the critical point. These evolutions are shown in Fig. 3.



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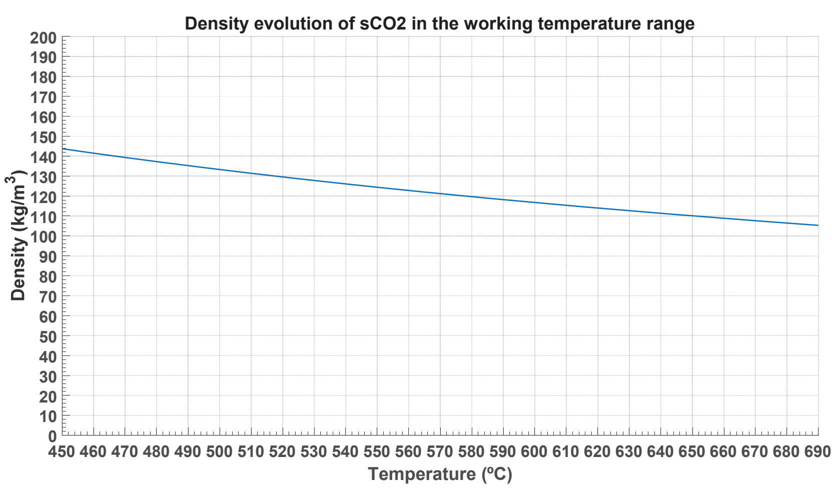

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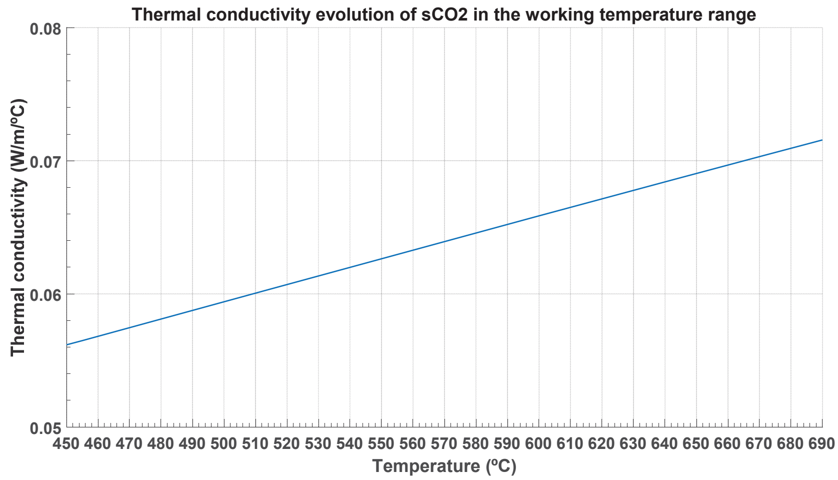

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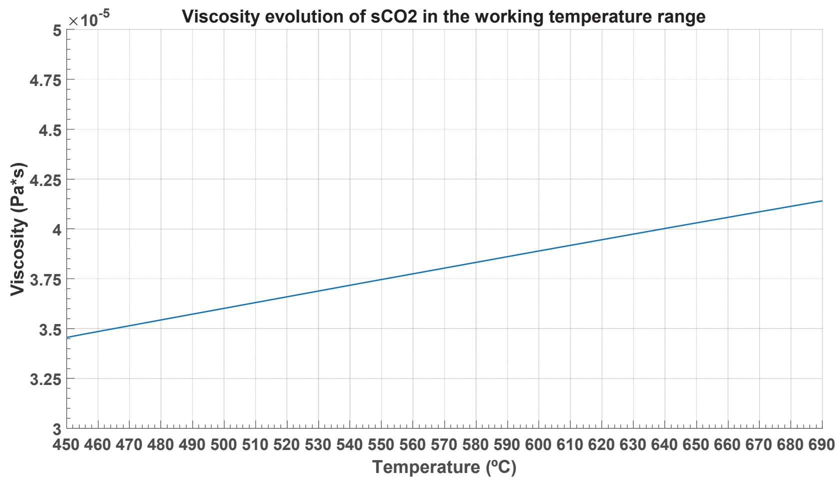

(d)

FIGURE 3. Thermal properties evolution for the supercritical CO2 between 460ºC and 690ºC: (a) Specific heat in J/kg/ºC; (b) thermal conductivity in W/m/ºC; (c) density in kg/m^3; and (d) dynamic viscosity in Pa·s (Source: [8])

The material of the CHHE is Haynes 242 (65% Ni – 8% Cr – 25% Mo). This material exhibits the following properties: a great withstanding to high temperatures and pressures, and a good corrosion resistance to ternary chloride molten salts, because of its high percentage of Molybdenum [9].

Regarding the working conditions, the layout considered incorporates a conventional recompression cycle of 50 MW_e. This layout will be explained in the optimization section, but it fixes the inlet / outlet CO_2 conditions in the CHHE to 520.5 ºC / 202.5 bar and 660 ºC / 200 bar, respectively. The pressure drop in the CO2 is set to 2.5 bar, that is an acceptable value for this type of heat exchanger working in a STTP [5]. The CHHE is considered as a balance counter-flow heat exchanger, and the temperature approach is fixed to 40ºC. So, the inlet / outlet conditions in the MS are 700 ºC / 25 bar and 560.5 ºC /22.6 bar.

At last the minimum thickness to withstand the pressure difference between channels is calculated by a mechanical method based on ASME codes [10].

Thermal Model of the Compact Honeycomb Heat Exchanger

The thermal model is based on a two dimensional thermo-fluid model, in which the CHHE is divided into N heat exchanger elements (HXEs), along the fluid direction, with the same thermal duty, $\dot{Q}_{HXE} = {}^{\dot{Q}}/_{N}$. The global heat transfer coefficient in every element, U_{HXE}, is calculated by Eq. (1).

$$U_{HXE} = \frac{1}{\frac{1}{h_{conv1}} + \frac{1}{U_w} + \frac{1}{h_{conv2}}} \tag{1}$$

In the above equation, U_w is the equivalent heat transfer coefficient along the thickness between channels, and h_{conv} is calculated by Gnielinski correlation [6], Eq. (2), as the flow is turbulent (Re > 2300) for both fluids.

$$Nu_{Dh} = \frac{(f_c/8) \cdot (Re_{Dh} - 1000) \cdot Pr}{1 + 12.7 \cdot \left(\sqrt{\frac{f_c}{8}}\right) \cdot (Pr^{2/3} - 1)} \cdot \left(\frac{Pr}{Pr_{si}}\right)^{0.11}$$

$$where:$$

$$f_c = [1.82 \cdot log(Re_{Dh}) - 1.64]^{-2} \tag{2}$$

In Eq. (2), f_c is the friction factor; Re_{Dh} is the Reynolds number based on the inner hydraulic diameter; Pr is Prandtl number at the bulk fluid temperature; Pr_{si} is the Prandtl number at the inner duct temperature, t_{si}. The use of Gnielinki correlation for supercritical CO_2 have been widely validated in other studies [11].

The pressure drop along the channels in the HXE is calculated by Darcy-Weisbach equation [6] for both fluids:

$$\Delta P_i = \frac{1}{2} \cdot f_{D,i} \cdot \left(\frac{L_{HXE}}{D_{h,i}}\right) \cdot \rho_{ave,i} \cdot u_{ave,i}^2 \tag{3}$$

Where D_h *(m)* is the hydraulic diameter of the duct; ρ (kg/m^3) is the average fluid density; u (m/s) is the average fluid velocity; and f_D is the Darcy friction factor; finally, the subindex i refers to the each fluid: molten salt or CO_2.

The results of the thermal model for a particular CHHE (MS channel diameter = 12.7 mm and CO_2 trapezoidal width = 5.7 mm) is shown in Table 2.

TABLE 2. Results from the thermal model of the CHHE

MS–to–sCO_2 CHHE	
Sizing and geometrical characteristics	
Number of modules in parallel	3
Number of modules in series	3
Shell diameter (m)	1.0194
Length of each module (m)	10.6432
MS circular channel diameter (m)	0.0127
sCO_2 channel width (m)	0.0057
Heat transfer area (m^2)	630.5962
Material	Haynes 242
Thermal characteristics	
Thermal power for each module (MW_{th})	11.22
Average U_{global} ($W/m^2/^oC$)	1342.8559
Temperature Approach (ºC)	40
Primary (Chloride Molten Salt)	
Maximum velocity (m/s)	2.0406
Inlet temperature (ºC)	700
Inlet pressure (bar)	25
Mass flow rate (kg/s)	204.5076
Outlet temperature (ºC)	560.5
Outlet pressure (bar)	22.6048
Pressure drop (bar)	2.3952
Average h_{conv} ($W/m^2/^oC$)	3612.8729
Secondary (sCO_2)	
Maximum velocity (m/s)	8.5167
Inlet temperature (ºC)	520.4975
Inlet pressure (bar)	202.4865
Mass flow rate (kg/s)	193.1400
Outlet temperature (ºC)	660
Outlet pressure (bar)	200
Pressure drop (bar)	2.4865
Average h_{conv} ($W/m^2/^oC$)	2526.5139
Costs	
Inversion Cost (Mio.$)	56.6979

As shown in Table 2, the total thermal duty of the heat exchanger is obtained by three modules in parallel and three modules in series. This is due to the manufacturing restrictions that could emerge from this new design. Although this concept is employed for lower thermal duties in aerospace applications, the dimensions required in this case can yield to new manufacturing challenges. In small heat exchangers, this type of structure could be manufactured by 3D printing, but in this case, it is possible that the original design would have to evolve towards cylindrical tubes for molten salts surrounded by a porous mesh through which the sCO_2 circulates. In a theorical approximation, the shell diameter is limited to 1 m and the ratio shell diameter to length must be greater than 1/10.

As a future objective, a CFD analysis of the CHHE will be accomplished, to study the thermo-mechanical stress along its structure.

THERMO-ECONOMIC OPTIMIZATION OF THE GEOMETRIC PARAMETERS OF THE CHHE

Integration of the CHHE in the Supercritical STPP

It has been considered the CHHE integrated into a STPP, as seen in Fig. 4. The supercritical cycle follows a conventional recompression layout of 50 MW_e, with dry cooling. The central receiver has been designed as a tubular cavity-type. The thermal storage consists of two tanks of molten salts with a solar multiple of 2.

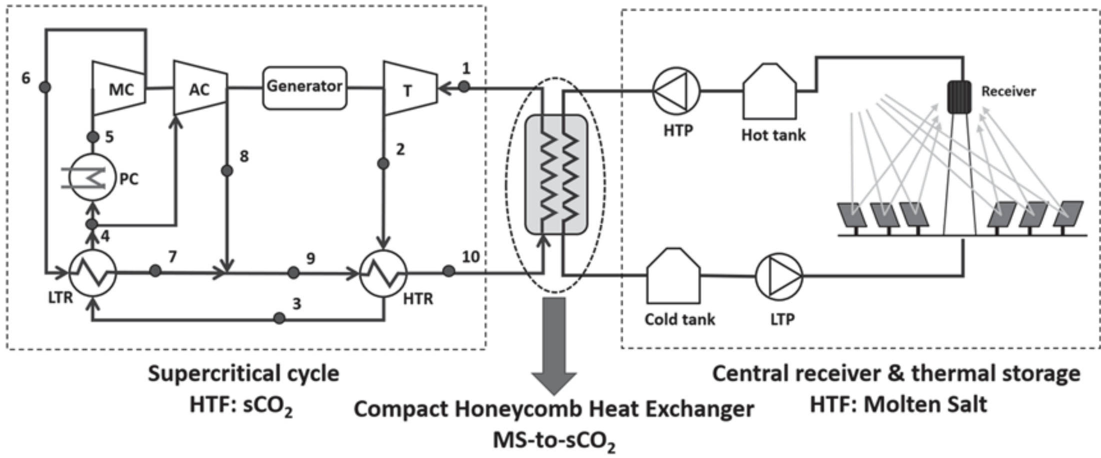

FIGURE 4. Scheme of the complete STPP with the CHHE between the solar field and the sCO_2 cycle (Source: [5])

As seen in Fig. 4, in the recompression configuration, there are two compressors: the main compressor (MC) and the auxiliary compressor (AC), that connect to the Low-Temperature Recuperator (LTR), and the High-Temperature Recuperator (HTR), respectively. The dry cooling is accomplished by a pre-cooler (PC). The CHHE is located upstream the turbine (T).

This cycle has been modeled and simulated in [3]. The thermodynamic properties of the state points marked in table 1 are summarized in Table 3.

TABLE 3. Thermodynamic properties of the state points of the recompression sCO_2 cycle

Recompression cycle			
Point	**P (bar)**	**T (˚C)**	**h (kJ/kg)**
1	200	688	701.3
2	86.2	574.1	566.5
3	85.8	224.2	158.4
4	85.4	122.9	39.09
5	85	50	-80.9
6	201.2	118.3	-41.57
7	200.8	219.6	117.4
8	200.8	212	107.2
9	200.8	217.7	114.9
10	200.4	545.6	522.9

The cycle power is equal to 50 MW_e, and the cycle efficiency is 49.57%, so the thermal power required in the CHHE is equal to 100.99 MW_{th}, distributed in 6 modules, as it shown in Table 2.

The operation conditions of the CHHE have been fixed, so only the investment cost is optimized; it has been considered a balanced heat exchanger, with a temperature approach (TA) equal to 40ºC and a sCO_2 pressure drop (dP_{sCO2}) of 2.5 bar. The geometric configuration of the CHHE has been optimized, in terms of the sCO_2 cross flow area vs. the molten salt cross flow area.

Thermo-Economic Optimization and Results

The investment cost of the CHHE, C_{CHHE}, has been estimated in a similar way to that employed in PCHEs [11]. The cost is calculated from the mass of the heat exchanger and the cost factor of the material (CM_{PCHE} = 120 \$/kg for Haynes 242), as seen in Eq. (4).

$$C_{CHHE} = M_{CHHE} \cdot CM_{CHHE} \tag{4}$$

The mass of the CHHE is calculated by means of the metal density (9050 kg/m^3 for Haynes 242) and the fraction of metal per m^3 of the heat exchanger, also called the core volume $V_{core,CHHE}$.

$$M_{CHHE} = \rho_{CHHE} \cdot V_{core,CHHE} \tag{5}$$

For the parametric analysis, two geometric parameters have been selected: the MS channel diameter (d_1), ranged between 0.5 inch (12.7 mm) and 1 inch (25.4 mm), and the CO_2 channel width (h2), with values from 5 mm to 10 mm. Results from this analysis are shown in Fig. 5.

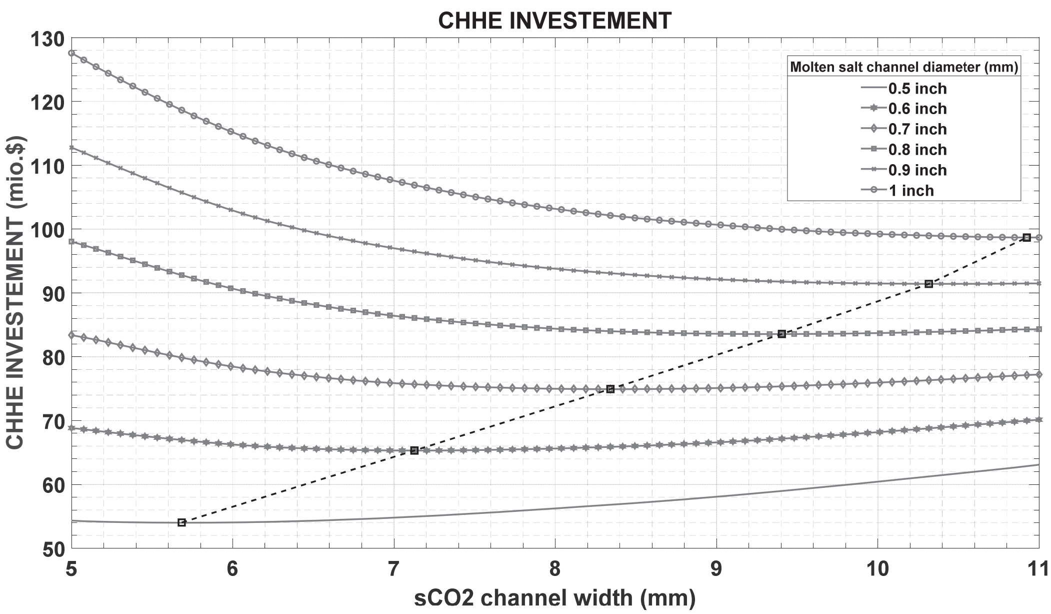

FIGURE 5. CHHE investment as function of the sCO_2 channel width (h_2), for different MS channel diameters (d_1)

The molten salt channel diameter (d_1) affects the CHHE investment more than sCO_2 trapezoidal channel width (h_2), as seen in Fig. 5. For each MS diameter, there is a sCO_2 width that minimizes cost (black dotted line in Fig. 5). These minima correspond to a similar ratio between cross flow areas, approximately 3.5, and a global heat transfer coefficient from 1200 to 1300 W/m^2/ºC.

CONCLUSIONS

This work presents a new design of heat exchanger to transfer heat from molten salt to supercritical CO_2. This heat exchanger is located between the solar field and the power cycle in a supercritical STPP, and it is one of the key elements of these layouts. The design presented is based on the honeycomb structure, and it consists of a larger circular duct for the molten salt, surrounded by six smaller trapezoidal channels for the CO_2. The larger dimensions for the molten salt duct are intended to avoid the plugging problems of this viscous fluid.

An economic optimization of the geometric parameters of this heat exchanger has been accomplished. For that, the operation conditions of the heat exchanger have been fixed. Results show that minimum investment costs are achieved with minimum ducts dimensions, that is, minimum MS channel diameter and minimum CO_2 trapezoidal channel width. The minimum MS diameter is limited to 0.5 inch (12.7 mm), to avoid clogging issues.

As a future work, a CFD analysis of the CHHE should be done, to study the thermo-mechanical stress along its structure. Besides, it is necessary to analyze the flow distribution in the channels and if there are problems in the temperature gradients, when facing CO_2 channels.

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REFERENCES

1. M. Mehos, C. Turchi, J. Vidal et al., 2017. *Concentrating Solar Power Gen3 Demonstration Roadmap* (No. NREL/TP--5500-67464, 1338899).
2. S. Kakaç, H. Liu and A. Pramuanjaroenkij, 2012. *Heat Exchangers: Selection, Rating, and Thermal Design,* Third Edition. CRC Press, Hoboken.
3. J.I. Linares, M.J. Montes, A. Cantizano and C. Sánchez, 2020. “A novel supercritical CO2 recompression Brayton power cycle for power tower concentrating solar plants.” *Applied Energy* 263, 114644.
4. R. Le Pierres, D. Southall, and S. Osborne, 2011, “Impact of Mechanical Design Issues on Printed Circuit Heat Exchangers.” *Proceedings of SCO2 Power Cycle Symposium*, University of Colorado at Boulder - University Memorial Center, CO.
5. M.J. Montes, J.I. Linares, R. Barbero and B.Y. Moratilla, 2020. “Optimization of a New Design of Molten Salt-to-CO2 Heat Exchanger Using Exergy Destruction Minimization.” *Entropy* 22, 883.
6. J.E. Hesselgreaves, 2017. *Compact heat exchangers: selection, design, and operation*, Second edition. ed. Elsevier/BH, Amsterdam.
7. C.S. Turchi, J. Vidal, M. Bauer, 2018, “Molten salt power towers operating at 600–650 °C: Salt selection and cost benefits.” *Solar Energy* 164, 38–46.
8. NIST database https://webbook.nist.gov/chemistry/.
9. H. Sun, J. Wang, Z. Li, P. Zhang, and X. Su, 2018. “Corrosion behavior of 316SS and Ni-based alloys in a ternary NaCl-KCl-$MgCl_2$ molten salt.” *Solar Energy* 171, 320–329.
10. ASME Boiler and Pressure Vessel Committee, American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Committee, Subcommittee on Pressure Vessels, 2010. *Rules for construction of pressure vessels.* an international code VIII, Division 1 VIII, Division 1. American Society of Mechanical Engineers, New York, N.Y.
11. V. Dostal, 2004. *A supercritical carbon dioxide cycle for next generation nuclear reactors.* Thesis (Sc. D.). Massachusetts Institute of Technology, Dept. of Nuclear Engineering.