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# EXERGO-ENERGETIC ANALYSIS OF A SENSIBLE HEAT STORAGE SYSTEM IN POROUS MEANS

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**Abstract.** *The possibility of efficiently storing energy in thermal form has been pointed out as a competitive advantage of plants using the Concentrated Solar Power (CSP) technology route compared to photovoltaics. Storage compensates to some extent for the intermittent nature of the solar source and gives flexibility to the operator allowing dispatch at more advantageous times both economically and operationally. There are different storage systems for CSP plants that depend on the technology adopted in the plant. In this work, an exergo-energy study was carried out for a thermal energy storage (TES) system that uses air as heat transfer fluid flowing through a porous medium with optimized geometry for heat transfer. Approximate energy equations for the air and the porous matrix were proposed to enable the study with different configurations of the TES. The aerodynamic parameters of the flow were obtained through simulations in the CFD platform Star-ccm+ that also served to validate the proposed approximations. A numerical model based on the approximated equations was developed. The model discretizes the domain in finite volumes, solving the equations of solid and fluid energy conservation and fluid mass conservation, using the upwind scheme for advective and time implicit term. Simulations of the loading/unloading cycle were performed for the TES of lengths 6m and mass flux in the porous medium of* $0.001\ kg/m^2s$*. The initial conditions (temperature of the porous medium) were set at the beginning of the simulation. The initial (porous medium temperature) and boundary conditions were considered based on the operational data of the Julich CSP Plant in Germany. To validate the proposed model, the air discharge temperature at Julich in Germany was compared with that of the present model, where at Julich proposed an acceptable temperature range between 640-680℃ and in the present work, the respective temperature obtained was 664℃. The total exergy destroyed in charge/discharge cycles for different configurations was calculated and has a value of* $8.1x10^3 J$*. The results show different values of exergy destroyed due to the heat exchanges between fluid and solid depending on the operating conditions. Therefore, an optimization of parameters should be applied for the specific operating conditions of each project to reduce losses that may compromise the profitability of the system as a whole.*
***Keywords:*** *Thermal load. Thermal discharge. Exergy. Heat Transfer*

## 1. INTRODUCTION

Addressing the current climate crisis necessitates a focus on enhancing energy conversion processes' efficiency. studies carried out through numerical simulations using software such as matlab, star-ccm+, among others, as well as experimental research, share a common goal: improving the efficiency of equipment or processes. these endeavors span academic and industrial settings.

When the objective is to enhance equipment efficiency, the exergetic analysis of equipment or processes emerges as a crucial tool. it leverages the first and second laws of thermodynamics to quantify the capacity to perform work, referred to as exergy. in this context, exergy serves as the upper limit for converting a given amount of energy into mechanical work via a thermal machine. exergy analysis, which contrasts the exergy involved in a process with the work performed, highlights the missed opportunity for a more efficient process.

In scenarios involving energy storage, like concentrating solar power (csp) plants, exergy loss is inevitable as it's dissipated in heat exchange processes. nonetheless, this loss can be mitigated. exergy is intertwined with processes featuring energy conversion, encompassing: (i) irreversibilities in real processes, (ii) the conversion of thermal energy into work, and (iii) constraints on useful energy. these processes are closely tied to heat transfer, fluid flow, and thermal energy storage, whether sensible or latent heat.

In the context of heat transfer analysis due to fluid flow over equipment, thermal power plants have employed air as a heat transfer fluid, owing to its cost-effectiveness when coupled with the sun's inexhaustible energy source. within this context, this study aims to investigate a thermal energy storage system using an exergetic numerical model. the research focuses on the charging processes during the day and the subsequent discharge for a 2-hour period, specifically when there is no available energy source. thermal energy storage, specifically the sensible heat variety, relies on the transfer of heat to a storage medium, allowing its later use in heating, cooling applications, or electricity generation when conventional energy sources are unavailable.

## 2. SCHEMATIC OF THE PLANT UNDER ANALYSIS

The study was carried out for the Julich CSP plant, where heliostats concentrate the incident solar irradiance on an absorber, consisting of a porous medium through which air passes to be heated. The heated air feeds a Rankine cycle and the excess is diverted to an energy storage system capable of storing energy to support the steam cycle at full load for 1.5 hours (Heneke et al 2008; Koll et al., 2009 Zunft et al., 2011).

(Pitz-paal,2014) heliothermal plant, also known as a concentrating solar plant, is a power generation facility that uses mirrors or lenses to concentrate sunlight at a single point or in a heat transfer tube. This concentrated heat is then used to heat a fluid, such as thermal oil or molten salt, which in turn is used to generate steam and drive a turbine that produces electricity. Heliothermal power plants are a more advanced form of solar energy compared to traditional photovoltaic solar panels, as they can store heat and generate electricity even when the sun is not shining. This makes them a viable option for continuous power generation. Heliothermal plants can be classified into different types, including central tower, parabolic and parabolic disk, each with its own system for collecting and concentrating sunlight.

In Figure 1, the solar panels reflect and concentrate the heat in a receiver, where in this equipment the air is heated due to the solar concentration up to a temperature of 680ºC. Part of the heated air is stored in the sensible heat store and the other part is fed into the steam power cycle through the boiler, in the superheated steam section.

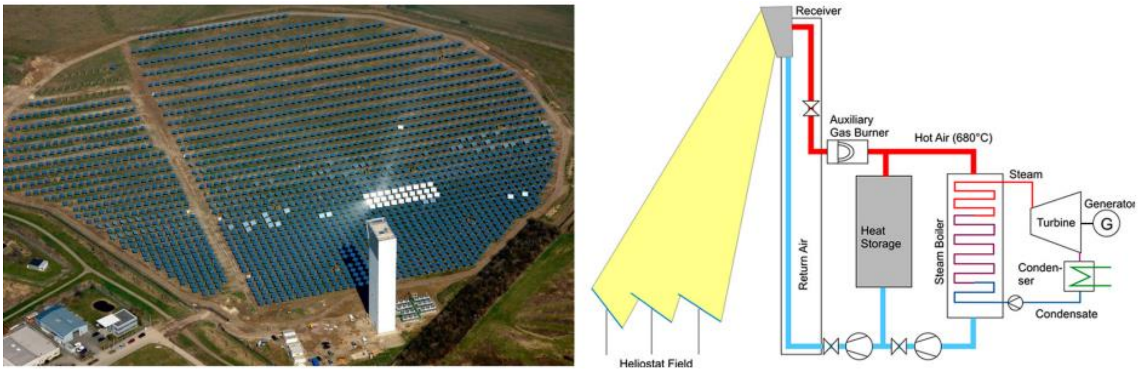


Figure 1 - Solar field on the left and steam power plant scheme on the right

## 3. METHODOLOGY

In this work, a porous solid matrix with parallel channels with sinusoidal geometry as shown in Figure 2 was considered for energy storage. In the thermal load, while the temperature difference between the fluid and the solid is favorable for the fluid, the solid raises its temperature and stores sensible heat. In the case of thermal discharge, where the solid will be at a higher temperature than the fluid, the heat exchange has the opposite direction, and the thermal energy will be transferred to the fluid as it flows through the channels of the porous matrix.

According to the literature, the main objective of the porous structure is to increase as much as possible the heat transfer coefficient between the fluid and the solid storage material. And this phenomenon is achieved by increasing the contact area between the flow of air and the solid, thus, each channel was defined with a circular section of variable and periodic diameter. In a research carried out by Nishimura et al. (2003), Rush, Newell and Jacobi (1999) and Russ and Beer (1997a), sinusoidal-type sections present a higher heat transfer coefficient than a constant diameter section (smooth tube).

With this information, the numerical model consists of two equations, one for the fluid and the other for the porous matrix (solid), which were obtained through an energy balance for both phases, considering that there is no thermal equilibrium. Thus, the following equations aim to show the temperature distribution of the air flow, Eq. (1) and of the solid, Eq. (2).

Fluid:

$$\varepsilon \cdot \left(\rho_f c_{p,f}\right) \frac{\partial T_f}{\partial t} + c_{p,f} G \frac{\partial T_f}{\partial x} = h_v\left(T_s - T_f\right) \tag{1}$$

Sólid:

$$(1-\varepsilon) \cdot \left(\rho_s c_{p,s}\right) \frac{\partial T_s}{\partial t} = h_v\left(T_f - T_s\right) + (1-\varepsilon) k_s \frac{\partial^2 T_s}{\partial x^2} \tag{2}$$

where: $T$ is the temperature, subscript $f$ is for fluid (air) and $s$ is for sólid, $G$ is the mass flow, $e$ is the porosity, $h_v$ is the heat transfer coefficient, $c_p$ is the specific heat, $k_s$ is the thermal conductivity, $\rho$ is the density.

Equations (1) and (2), after validation with Star-ccm, where the parameters for effective Nusselt number for the proposed geometry were defined, were solved by finite volume method, with implicit formulation in time and "upwind" technique for the interpolation of the advective term.

The equation (1) allows it to be solved sequentially from the first to the last finite volume. The solution for each volume is performed by Newton-Raphson since the dependence of thermophysical properties on temperature makes the equation for each volume non-linear. The equation (2) results in a tridiagonal system and can be solved using sparse matrix solution techniques. The systems of equations for solid and liquid are coupled and must be solved simultaneously, through an iterative process. For each step in time, first solve the equation (1) for the fluid, then (2) for the solid, the cycle is repeated until the

To achieve the stationary cyclic regime, considering the operation of a Heliothermal Plant with a thermal energy storage system (TESS) as suggested by (ZUNFT,2011), in which it is charged during the day with the surplus energy from the solar field, and discharges at night, prolonging the operation of the plant, consecutive cycles were simulated from an initial condition until the distribution temperatures of the solid in subsequent cycles are close.

Applying the boundary conditions in equations (1) and (2) which are

- The inlet temperature, $T_{f,in}$, and the mass flow G of the fluid are constant at all times;
- Adiabatic condition for the fluid at the exit of the domain, $\frac{dT_f}{dx=L} = 0$;
- Adiabatic condition for the solid at the entrance and exit of the domain, $\frac{dT_s}{dx=L} = 0$;

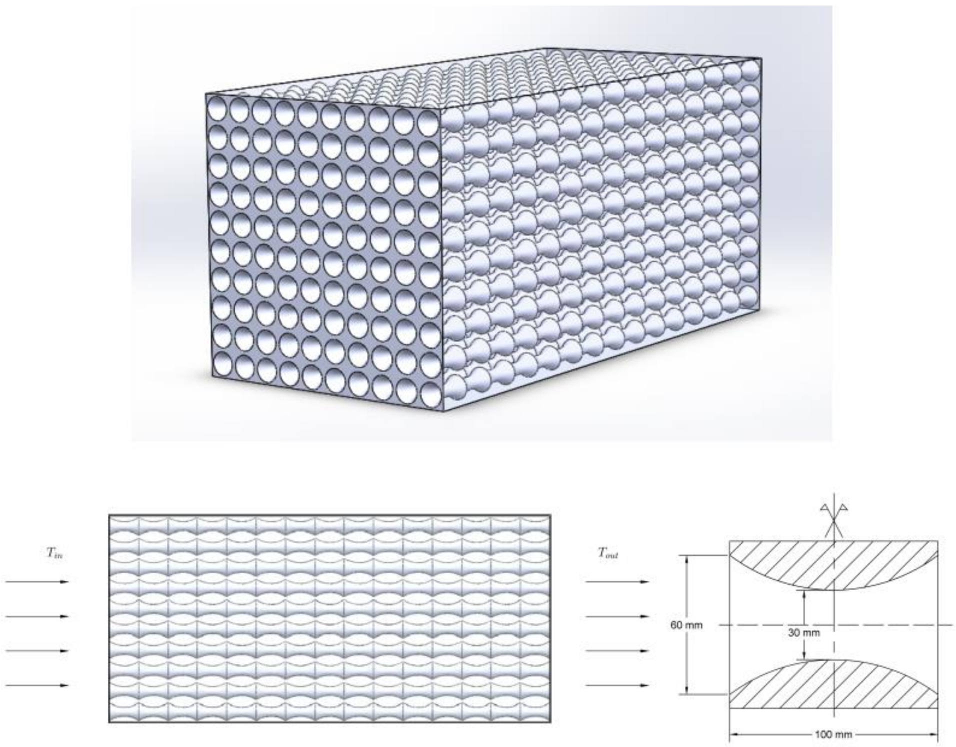


Figure 2 - Thermal storage model under study

From the values obtained in Equations (1) and (2) for temperature distribution in the solid and in the air, it is possible to make an exergetic analysis given by the following equations.

$$X_{mass,in} = \dot{m}_{air,in}\left\{\int_{T_0}^{T_{in}} c_{p,air}dT - T_0\left[\int_{T_0}^{T_{in}} \frac{c_{v,air}}{T_f}dT + R.ln\left(\frac{\rho_{air,0}}{\rho_{air,in}}\right)\right]\right\} \quad (3)$$

$$X_{mass,out} = \dot{m}_{air,out}\left\{\int_{T_0}^{T_{out}} c_{p,air}dT - T_0\left[\int_{T_0}^{T_{out}} \frac{c_{v,air}}{T_f}dT + R.ln\left(\frac{\rho_{air,0}}{\rho_{air,1,out}}\right)\right]\right\}_t \quad (4)$$

$$X_{air} = \sum_{i=1}^{ncv} \left\{ \varepsilon \, Vol_{\,i} \rho_{air,1,i} \left[ \int_{T_0}^{T_{air,1,i}} c_{v,air} dT + P_0 \left( \frac{1}{\rho_{air,1,i}} - \frac{1}{\rho_{air,0}} \right) - T_0 \left( \int_{T_0}^{T_{air,1,i}} \frac{c_{v,air}}{T} dT + R.\ln\left( \frac{\rho_{air,0}}{\rho_{air,1,i}} \right) \right) \right] \right\} \tag{5}$$

$$X_{sol} = \sum_{i=1}^{ncv} \left\{ (1-\varepsilon) \, Vol_{\,i} \rho_{sol} \left( \int_{T_0}^{T_{sol,i}} c_{sol} dT - T_0 \int_{T_0}^{T_{sol,i}} \frac{c_{sol}}{T} dT \right) \right\} \tag{6}$$

$$X_{destroyed} = X_{dest,conv} + X_{dest,cond} \tag{7}$$

$$X_{dest,conv} = \sum_{i=1}^{ncv} \left\{ \left[ \left( \frac{T_0}{T_{sol,i,avg}} - \frac{T_0}{T_{air,i,avg}} \right) \right] \left[ hA\left(T_{air,i} - T_{sol,i}\right) dt \right] \right\} \tag{8}$$

$$X_{dest,cond} = \sum_{i=1}^{ncv} \left\{ \left[ \left( \frac{T_0}{T_{sol,i,avg}} - \frac{T_0}{T_{sol,i-1,avg}} \right) \right] \left[ \frac{-kA}{dx} \left(T_{sol,i} - T_{sol,i-1}\right) dt \right] \right\} \tag{9}$$

where: $X_{sol}$ is the exergy in the solid, $X_{air}$ is the exergy in the air flow, $X_{dest,conv}$ is the exergy destroyed in the convection process, $X_{dest,cond}$ is the exergy destroyed by conduction, $T_0$ it's room temperature, 25°C, R is the general gas constant 287 kJ/kg K.

## 4. RESULTS AND DISCUSSIONS

To obtain the results of the air and solid flow temperature distribution, the numerical model was run with the operational data shown in table 1.

Table 1 - Test conditions

| | |
|---|---|
| Air inlet temperature | 800 K |
| Initial temperature of the porous matrix | 400 K |
| Porous matrix length | 6 m |
| mass flow | 0.001 $kg/m^2s$ |
| Room temperature | 300 K |
| Simulation time on load | 12 hours |
| Download simulation time | 2 hours |
| Cycles | 6 |

When operating at steady state, the cycles must follow the following sequence: charge-discharge-redistribution. Redistribution takes place between the end of the 2-hour-discharge and the beginning of the new charge, which will only happen the following morning. However, at first it is not known what the initial temperature distributions in the charging and discharging processes would be when repeated cycles is reached. Therefore, the simulations are performed from a charging phase with a uniform temperature distribution as initial condition followed by multiple cycles until the temperature distribution at the beginning of the charge is the same in consecutive cycles. Figure 3 shows the temperature profiles at the end of charging and discharging for 6 cycles. Note that the temperature distributions of the last two cycles (green and cyan in the figure) are very close, characterizing the steady state of the cycles.

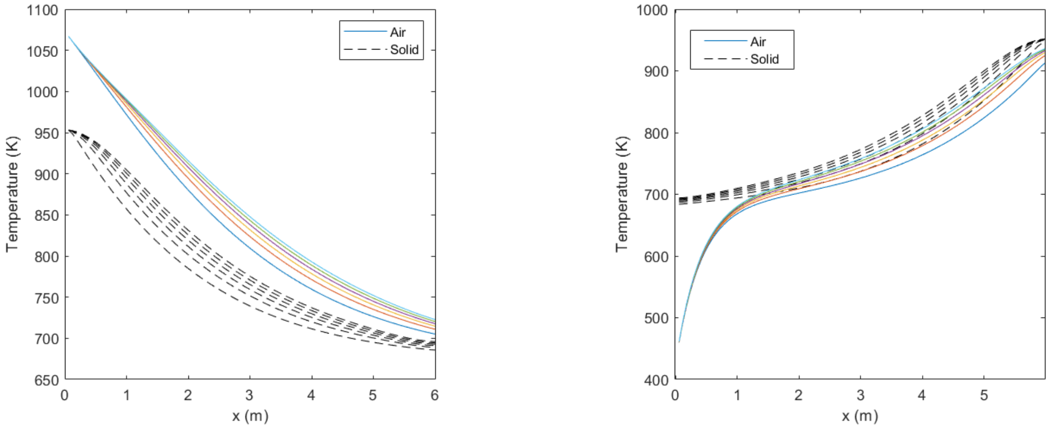


Figure 3- Temperature distribution in charge (a) and discharge (b)

With the conditions shown in table 1 and through the results obtained in Figure 3, the main result of the present work, in effect of comparison, was the air discharge temperature, where 664 ℃ was obtained during 2 hours of thermal discharge, while in the experimental work carried out in Julich by (Zunft et al., 2011), obtained an acceptable temperature range between, 640-680 ℃, as shown in Figure 4. This is probably due to small flow rate.

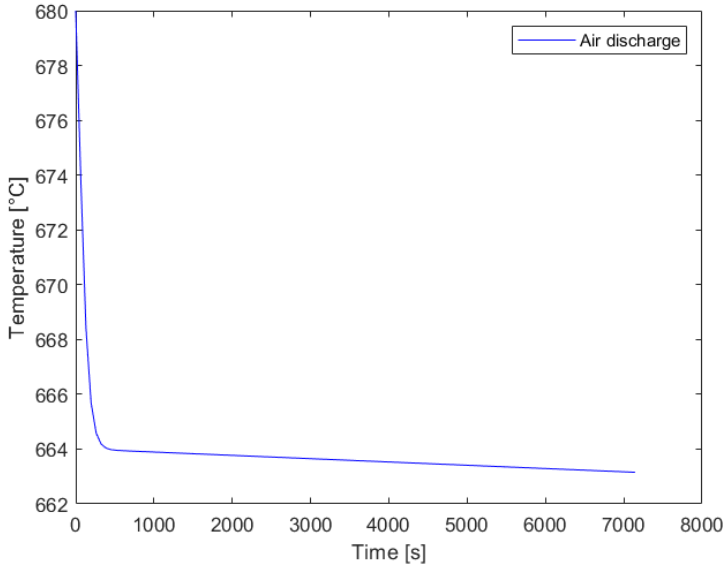


Figure 4 - Air discharge temperature distribution

According to Figure 4, it is observed that between the time from 0 to approximately 200 seconds, when the simulation (operation) starts, the temperature distribution has a drop from 680 to 664 ℃, this was due to the air that was inside the porous matrix discharges in the cycle until this temperature enters a steady state, in the case of 664 ℃ throughout the simulation. Figure 5 below shows the temperature distribution in its last thermal cycle, which is interesting because it is at the end of this cycle that it shows the thermal behavior inside the storage, both of the air and of the solid and the redistribution.

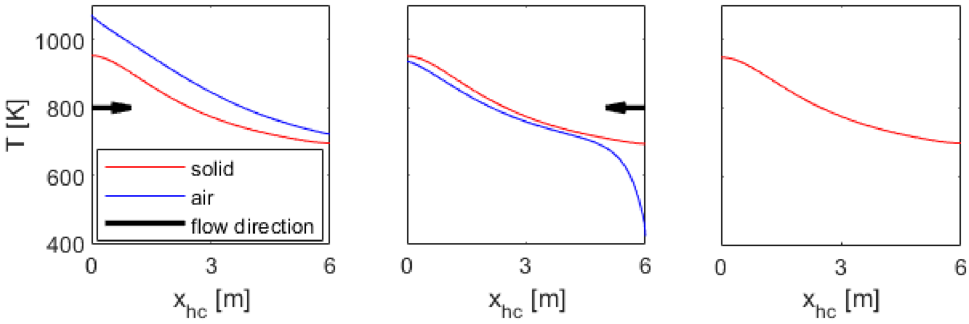


Figure 5 - Temperature distribution in the load (a), thermal discharge (b) and redistribution referring to the last cycle (c).

According to the results of Figure 5 thermal charge, discharge and redistribution as a function of the length of the porous matrix, it is notable that the matrix absorbs heat during the thermal load process, (a), considering that the initial temperature of the porous matrix is 300 K and it rises to 950 K at $x = 0$ during 12 hours of operation develops, when the temperature difference is appreciable, it means that part of the thermal energy transported by the air in the flow is not being transferred to the matrix, which indicates some inefficiency in the process.

In the thermal discharge process, (b), when the air flow is inverted and enters the porous matrix at $x = 6$, according to the arrow, the fluid recovers part of the heat left by it in the load and heating the matrix, causing there is a better heat exchange between the solid and the air. In the discharge, the solid has a higher temperature than the air in the whole process and has little difference, which shows that the heat transfer is not complete and this was due to the fact that in the charging process the solid was heated and thus reduced the temperature difference between the phases. It is also observed that at the end of the discharge, the air temperature presents a considerable reduction, this phenomenon was due to all the air that was contained in the store leaving it, leaving only the temperature of the solid. In the redistribution, (c), there is no air flow inside the storage and only internal heat transfer until the thermal load process again, which starts when the sun "rises" and starts to heat the solar panels, as shown in Figure 1.

According to Zunft et al., (2014), a better heat transfer between the phases would allow the transfer fluid to lose heat faster, resulting in the same temperature of the fluid and the porous matrix, and consequently, a better efficiency. From the analysis of the temperature distribution of the air flow and the defined porous matrix, it is possible to make an exergy analysis of the entire process, exergy accumulated and destroyed in both processes.

With the exergy values shown in table 2, it is noted that in the thermal load, the exergy accumulated in the porous matrix and in the air flow, has the same exergy value, this value is the same because the air is in full contact with the porous matrix due to the constant air flow of 12 hours with a constant temperature of 800K at the entrance of the matrix, and through the heat exchange occurred in the process, the exergetic value that "leaves" the air (hot source) "enters" the matrix (cold source), hence equality. In the accumulated exergy in the air flow, the thermal discharge is shown with a lower value than in the thermal load, this is due to the temperature difference between the air and the porous matrix being smaller in the discharge, as shown in Figure 3. This value in the air discharge represents that the heat transported in the air, part of it is not being transferred to the porous matrix, which indicates some degree of inefficiency of the process.

Table 2 – Sum of exergy values

| | Thermal load | Thermal discharge |
|---|---|---|
| Accumulated exergy in the porous matrix [J] | $8.8x10^6$ | $1.4x10^9$ |
| Accumulated exergy in the airflow [J] | $8.8x10^6$ | $7.5x10^4$ |
| Exergy lost [J] | $6.5x10^3$ | $1.6x10^3$ |

The exergy destroyed in the thermal loading and unloading process shows that in the load its loss is greater. This is because the air flowing and transferring heat to the porous matrix has a value of 1060K, at $x = 0$, greater than the temperature of the porous matrix, 950 K, according to Figure 3 in its respective cycle. And at the end of the last cycle, in thermal loading and unloading, the total lost exergy was $8.1x10^3$J, that is, during the entire process of 12 hours of loading, 2 hours of unloading and redistribution, this value represents the amount of work, or power that the system no longer obtains. This number is directly linked to the temperature variation between the porous matrix and the air flow, where the greater the difference, the greater the loss of exergy. For thermal analyzes linked to exergy, as seen in the work, the closer the phase temperatures, the better the process efficiency and lower cost.

## 5. CONCLUSION

According to the results obtained and the proposed methodology, using the experimental power plant at value (CSP) in Julich, proposed by (Zunft et al., 2011) and with a thermal charge and discharge heat store of the honeycomb type honey (porous media) proposed by (Adasme, 2018), it was possible, through a computational code written in MATLAB, to verify its consistency in relation to the air discharge temperature, where in the present work it was found 664℃ during 2 hours of thermal discharge, and in the experimental work carried out in Julich in Germany, an acceptable temperature range between 640-680℃ was established with the respective 2 hours of thermal discharge process.

It was also verified the accumulated and lost exergy in the thermal load and discharge and its total value, where through its results it is possible to conclude that there is a loss of exergy due to the temperature difference, which shows that there is a need for a more development towards reduction of the temperature difference between the phases, and to transfer most of the temperature carried by the air to the porous matrix, thereby increasing the efficiency of the cycle. It is also possible to conclude that if there was a better transfer of heat from the air to the porous matrix, that is, their temperatures closer to each other, less loss of exergy would be recorded, and with that, used to convert it into work in the cycle CSP.

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## 7. RESPONSIBILITY NOTICE