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Modelling heat and mass transfer in solar evaporation systems

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Abstract

A two-dimensional axisymmetric model of a solar evaporator was developed, based on a rigid biosourced open-cell foam floating on the surface of water contained in a beaker, including a set of boundary conditions and water transport in the porous medium. Experiments were carried out to validate the model for two cases, with and without a layer of insulating material around the beaker. Several concrete cases of experimental solar evaporation studies recently reported in the literature, with or without insulation, radiation intensification, etc., have then been studied in the light of our model. We have thus been able to compare the observed trends with those of the numerical model in order to explain the results reported in the literature and, in some cases, to find a geometric optimum. The effect of ambient conditions on the evaporation rate was investigated, and significant changes were observed by modifying solar input, air temperature and air humidity. We also showed that increasing the evaporation area and using an insulating layer between the water and the evaporative material increases the evaporation performance.

Keywords: Simulation, Biosourced material, Solar energy, Solar evaporation, Porous media, Heat and mass transfers

1 Introduction

In modern society, energy plays a major role, and its development as we know it today is only possible with access to a significant amount of resources. Nowadays, improving the effectiveness of systems and producing energy from more sustainable sources is more relevant than ever. Water evaporation, for example, is a subject of great interest [1] and a wide variety of materials have recently been developed to produce clean water from any source, even polluted water and seawater, without the need for an energy grid. Evaporation indeed occurs with or without solar flux at the water surface, but it can be enhanced by adding a floating porous medium.

Water evaporation systems (WESs) are composed of one or more materials, sometimes synthesised or manufactured from a natural resource, whose characteristics can be modified to improve evaporation efficiency [2]. In general, studies aim to improve materials properties, *e.g.* to increase the absorption of sunlight, to improve the energy conversion of light into heat or to improve the transport of water to the surface. Thermal conductivity can also be modified to reduce transfer to bulk water and thus conserve energy at the surface where evaporation occurs. In addition, the resistance of materials to clogging has been studied in seawater evaporation processes, where salt can affect long-term performance [3]. Changes in geometry have also been made to exploit heat exchange from the air to the surface instead of letting the transfer occur from the hot evaporation surface to the air [4].

Besides the large variability of solar evaporation systems mentioned above, a common problem encountered when reading the literature is that the protocol for measuring experimental performance is not identical in all publications. For example, some studies report mass loss measurements under a transparent bell jar and others do not. The bell jar has the effect of increasing the temperature and humidity of the air around the material compared to an open air system. In addition, tests can be carried out in different locations around the

world with different humidity and temperature conditions. In some cases, an insulating layer can be used to support the evaporation material, which would not necessarily be able to float on its own, and to provide thermal insulation with the bulk water. Examples of such a system have been published based on pomelo peel [5] or plant stems [6]. All of these differences lead to more or less favourable conditions for the evaporation of water, regardless of the material.

The interest in proposing a numerical model that takes into account the mass and heat transfer that underlies water evaporation is the ability to accurately capture the behaviour of such WESs, and to explore the effect of experimental parameters such as those presented above. Whether it is the layout of the system, the geometry of the material or even the properties of the water within the material, it is possible to examine all of these impacts on the overall performance and behaviour in order to better understand how to design an efficient evaporation system. Furthermore, although numerical simulations have been used in the past for solar evaporation systems, they were limited to predicting certain aspects such as temperature distribution or water diffusion [7–9]. As far as we know, a general model taking into account the different types of heat exchange between the system and the air, the transport of water in the material and the evaporation from the water surface is missing in the literature.

In this work, we developed a model based on experimental results obtained under strictly controlled conditions, using a mimosa tannin foam that we have synthesised and used to improve water evaporation. Two cases were experimented, with and without insulation around the vessel containing the water to be evaporated, to limit heat exchange at the boundaries. These two cases were simulated, as the physics at the boundaries are different, and were validated using the same model apart from the system geometry and a few parameters. Various cases from the open literature have also been considered, allowing the effects of various operating conditions and materials geometry to be explored, and to which our model has been applied. Although the results between the literature experiments and our

simulations cannot be directly compared due to the different materials used, the main trends could be successfully compared.

Therefore, our model should be particularly interesting in revealing the optimal geometry parameters, such as the dimensions of the insulation to be used beneath a material, or to study the impact of the insulation provided by synthetic foams used to allow the flotation of the material. Prior to our research, a guideline for the main experimental conditions, such as ambient temperature, relative humidity, geometry or position of the different elements, had been discussed [10]. Here, these conditions are investigated to describe their impact on the final evaporation rate and efficiency results, as our model takes into account the ambient air temperature, relative air humidity, air convection type and solar flux.

2 Materials and Methods

2.1 Evaporation material

Firstly, a bio-based foam was prepared, from which water evaporation experiments were conducted to validate our model. This foam was synthesised from tannin, a natural extract from the bark of the mimosa tree (*Acacia Mearnsii*), using a process described elsewhere [11]. The material was obtained by physical foaming, which involves producing a gas *in situ* in a thermosetting resin during its polymerisation. **Table 1** shows all the ingredients used in the process, indicating their relative amounts and their role in the formulation. Briefly, tannin is first dissolved in water under magnetic stirring and then mixed with furfuryl alcohol. Next, diethyl ether and a 37 wt. % aqueous solution of formaldehyde are added and the mixture is carefully homogenised. Finally, a 65 wt. % aqueous solution of para-toluenesulphonic acid (pTSA) is added under stirring, after which the mixture expands rapidly while polymerising at the same time. Once fully expanded and cured, the foam is dried at room temperature for one week; further details are given elsewhere [12]. The foam is then cut to the proper dimensions, as shown in **Fig. 1** (about 58 mm in diameter and 15 mm thick).

Table 1. Standard formulation of tannin-based foam.

Ingredients	Role in the formulation	Mass (g)
Mimosa tannin extract	Base of the resin	60
pTSA (65 wt. % in water)	Acidic catalyst	22
Furfuryl alcohol	Co-monomer	21
Formaldehyde (37 wt. % in water)	Crosslinker	14.8
Water	Solvent	12
Diethyl ether	Blowing agent	6

2.2 Experimental protocol and measurement methods

The experiment consists in illuminating a beaker (60 mm diameter), filled with tap water and with the foam floating on the surface, with a solar light simulator (SOLAR LIGHT LS-1000). The experimental setup is shown in **Fig. 1**. A Sartorius ENTRIS weighing scale accurate to the milligram is used to measure the mass of the whole system, and it is thus possible to determine the amount of evaporated water by difference. T-type thermocouples are used to measure temperatures at the interface between the evaporation material and the water on the one hand, and within the evaporation material on the other hand. A thermal camera (OPTRIS PI) pointing at the surface of the evaporation material is used to measure its temperature locally. Air temperature and relative humidity are measured using a sensor placed in the room. To explore the effect of heat exchange from the sides, a second experiment was also conducted with insulating foam around the beaker. Therefore, case 1 discussed below will refer to the case without insulation and case 2 to the case with insulation.

Fig. 1. Experimental setup used to measure the evaporation rate of water with a dark porous foam floating on the surface. The 15 mm-thick, 58 mm diameter tannin foam used for this purpose is shown in the lower right-hand corner.

In order to compare it with the numerical model, the experiment was carried out under three different solar concentrations: 0.7, 1.0 and 1.3 sun (1 sun being equal to 1 kW m^{-2}) for about one hour for each illumination. This allowed the evolution of heat and mass to be studied as a function of solar concentration. The system efficiency was calculated using Eq. 1:

$$\eta = \frac{\dot{m} h_v}{C_{opt} P_{s,0}} \quad \text{Eq. 1}$$

where $\dot{m}$ ($\text{kg m}^{-2} \text{ h}^{-1}$) is the evaporation rate, C_{opt} (sun) is the solar concentration, and $P_{s,0}$ (1000 W m^{-2}) is the power density of the average solar light on earth. Typical results in the literature present the evaporation rate and efficiency for a given specific solar power. Assuming a theoretical efficiency of 100 % and with 1 sun illumination contributing only to

the evaporation of water, whose enthalpy of vaporisation is 2450 kJ kg^{-1} under STP conditions, $\dot{m}$ is equal to $1.47 \text{ kg m}^{-2} \text{ h}^{-1}$, which therefore represents the theoretical limit of evaporation. Sometimes in the literature, evaporation in the dark is subtracted from the evaporation rate with illumination to take into account only evaporation due to light. It is useful to note that Eq. 1 does not take into account the heat exchange with the air, but only the solar illumination input and the evaporation rate output. This is deliberate, as Eq. 1 is most often used in the literature, and this is how we can compare our results with what has already been published.

3 Physical model

A 2D axisymmetric model was written using COMSOL Multiphysics 5.6. Bulk water and evaporation material were modelled as rectangles (in blue and red, respectively, in **Fig. 2**). This basic geometry was adapted to cases 1 or 2 of the study, i.e., without or with an insulating layer around the beaker, respectively. In this model, the upper surface is illuminated by a solar radiation, is cooled by convection with air, and evaporates water. On the sides and bottom, there are only convection and conduction, respectively.

Fig. 2. Types of heat exchange at the boundaries of the system: (a) case 1: without insulation; and (b) case 2: with insulation.

3.1 General equations

In order to simulate the heat and mass transfers existing during the evaporation process, energy and mass conservation equations were used to obtain the temperature, T , (Eq. 2) and the volume fraction of liquid in the pores, ε_l (Eq. 3).

The conservation of energy equation describes conductive heat transfer and temperature changes over time. The energy associated with mass transfer is small and due to the very slow kinetics of this phenomenon, it has therefore been neglected in this study. This conservation of energy equation is applied to calculate the temperature in the beaker and in the insulator (corresponding to case 2).

$$\rho_j c_{p,j} \frac{\partial T}{\partial t} + \vec{\nabla}(-\lambda_j \vec{\nabla} T) = 0 \quad \text{Eq. 2}$$

where ρ_j (kg m⁻³) is the density, $c_{p,j}$ (J kg⁻¹ K⁻¹) is the heat capacity, and λ_j (W m⁻¹ K⁻¹) is the thermal conductivity of component j (water w , evaporation material m or insulating layer i).

The mass transfer is described by a phenomenon of water diffusion in the porous material and is not solved in water or in the insulated medium. A transient equation is used in order to allow the material to dry out and thus reduce surface evaporation when it is no longer saturated. The equation is obtained by keeping the water liquid inside the material (Eq. 3).

$$\rho_w \frac{\partial \varepsilon_l}{\partial t} + \vec{\nabla} \left(-D \frac{\varepsilon_l \rho_w \rho_w}{\varepsilon_s \rho_s} \vec{\nabla} \varepsilon_l \right) = 0 \quad \text{Eq. 3}$$

where ε_l (dimensionless) is the water volume fraction in the evaporation material, ε_s (dimensionless) is its skeleton volume fraction, ρ_w (kg m⁻³) is the density of liquid water, ρ_s (kg m⁻³) is the skeleton density of the evaporation material, and D (m² s⁻¹) is the diffusion coefficient of water in the evaporation material.

3.2 Boundary conditions

In this section, the boundary conditions used are specified. It should be noted that thermal continuity conditions are used for the heat equation for all internal boundaries.

3.2.1 Interface between the air and the material

The heat exchange at the surface (Eq. 4) takes into account convection, incoming light radiation, as well as losses by emission and by water evaporation (**Fig. 2**). The latter involves the surface evaporation, $\dot{m}_{surf}$ (kg m⁻² h⁻¹), multiplied by the energy consumed to raise the temperature of water and the enthalpy of vaporisation of water, h_v (J kg⁻¹).

$$-\vec{n} \cdot (-\lambda_m \vec{\nabla} T) = h_c (T_a - T) + \dot{m}_{surf} (h_v + c_{p,w} (T_{w,i} - T)) + P_{s,0} C_{opt} + E_m \sigma (T_a^4 - T^4) \quad \text{Eq. 4}$$

with λ_m (W m⁻¹ K⁻¹) the thermal conductivity of the evaporation material, h_c (W m⁻² K⁻¹) the convection coefficient at the surface, $c_{p,w}$ (J K⁻¹ kg⁻¹) the heat capacity of water, $T_{w,i}$ (K) the initial temperature of water, $P_{s,0}$ (1000 W m⁻²) the standard solar surface power, C_{opt} (sun)

the solar concentration (used to achieve different values of solar power), E_m (dimensionless) the emissivity of the material and σ ($5.67 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$) the Stefan-Boltzmann constant, and $-\vec{n}$ the unit vector normal to the boundary.

To calculate the convective heat transfer coefficient, the following equation was used:

$$h_c = \frac{\lambda_{air}}{L_c} Nu \quad \text{Eq. 5}$$

where L_c is the characteristic length of the surface (which corresponds to the diameter for the top surface and the height for the side), and Nu is the Nusselt number (plane surface assumption), which can be calculated as follows:

$$Nu_{forced,top} = 0.3387 \frac{Pr^{\frac{1}{3}} Re^{\frac{1}{2}}}{\left(1 + \frac{0.04683}{Pr}\right)^{\frac{1}{4}}} \quad \text{Eq. 6}$$

In Eq. 6, Re is the Reynolds number and Pr the Prandtl number, such that:

$$Pr = \frac{\mu}{\alpha} \quad \text{Eq. 7}$$

in which α is the thermal diffusivity and μ the dynamic viscosity of the fluid, here the air. The Reynolds number is calculated using the following equation:

$$Re = \frac{u L_c}{\nu} \quad \text{Eq. 8}$$

with u the velocity of air (m.s^{-1}) and ν is kinematic viscosity ($\text{m}^2.\text{s}^{-1}$).

Since h_c is fitted for the case without any forced convection, the Nusselt number for the case of natural convection, $Nu_{natural}$, is determined using Eq. 5. Then, a Nusselt number $Nu_{combined}$, combination of both forced and natural convection, can be calculated [13]:

$$Nu_{combined} = (Nu_{natural}^4 + Nu_{forced}^4)^{\frac{1}{4}} \quad \text{Eq. 9}$$

At the upper surface, the water loss in the diffusivity equation is equal to the evaporation rate:

$$-\vec{n} \left(-D \frac{\varepsilon_l \rho_w \rho_w}{\varepsilon_s \rho_s} \vec{\nabla} \varepsilon_l \right) = \dot{m}_{surf} \quad \text{Eq. 10}$$

Finally, since water evaporation comes from the difference between the mass concentration of the surface and the air, then $\dot{m}_{surf}$ is calculated as follows [14]:

$$\dot{m}_{surf} = -k_m \frac{M}{R} \left(\frac{P_w^{v,sat}}{T} a_w - H_R \frac{P_a^{v,sat}}{T_a} \right) \quad \text{Eq. 11}$$

where M ($18 \times 10^{-3} \text{ kg mol}^{-1}$) is the molar mass of water, R ($8.314 \text{ J mol}^{-1} \text{ K}^{-1}$) is the universal gas constant, $P_w^{v,sat}$ (Pa) and $P_a^{v,sat}$ (Pa) are the water vapour pressures at the local temperature T (K) and at the air temperature T_a (K), respectively, a_w is the water activity, H_R (%) is the relative humidity and k_m (m s^{-1}) is the convective mass exchange coefficient, calculated using the Colburn analogy [13]:

$$k_m = \frac{h_c}{1000} \quad \text{Eq. 12}$$

$P_w^{v,sat}$ and $P_a^{v,sat}$ are calculated from the Clapeyron formula as follows:

$$P_x^{v,sat} = P_0 \exp \left(\frac{h_v}{R} M \left(\frac{1}{T_0} - \frac{1}{T} \right) \right) \quad \text{Eq. 13}$$

where $P_x^{v,sat}$ is the water vapour pressure at the temperature of the environment, P_0 is the standard pressure of 101 325 Pa and T_0 is the boiling point of water (373.15 K).

3.2.2 Interface between the air and the side of the system

Heat exchange between the air and the wall of the beaker involves convection and radiation losses:

$$-\vec{n} \left(-\lambda_j \vec{\nabla} T \right) = h_{c,s} (T_a - T) + E_s \sigma (T_a^4 - T^4) \quad \text{Eq. 14}$$

where λ_j ($\text{W m}^{-1} \text{ K}^{-1}$) has the same meaning as in Eq. 1, $h_{c,s}$ ($\text{W m}^{-2} \text{ K}^{-1}$) is the convection coefficient at the side, and E_s (dimensionless) is the emissivity of the side.

For forced convection at the side of the system, the following Nusselt expression is used assuming a vertical cylinder:

$$Nu_{forced,side} = 0.3 + \frac{0,62Re^{1/2}P_r^{1/3}}{\left[1 + (0,4/P_r)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re}{282}\right)^{5/8}\right]^{4/5} \quad \text{Eq. 15}$$

3.2.3 Interface between the weighing scale and the bottom of the beaker

At the interface between the weighing scale and the bottom of the beaker, only a convective heat flow was assumed, formalised by:

$$-\vec{n} \cdot (-\lambda_w \vec{\nabla} T) = h_{c,b}(T_a - T) \quad \text{Eq. 16}$$

where $h_{c,b}$ ($\text{W m}^{-2} \text{K}^{-1}$) is the convection coefficient at the bottom.

3.2.4 Interface between the water and the evaporation material

At the interface between the water and the evaporation material, as the latter is considered saturated, the liquid fraction is fixed at a value equal to the porosity, φ , and thermal continuity is assumed, i.e.:

$$\varepsilon_l = \varphi \quad \text{Eq. 17}$$

3.3 Simulation parameters

Table 2 lists the input parameters used in the simulation, and their symbols, definitions, units and values considered are also given.

Table 2. Input parameters used in the simulation.

Symbols	Definitions	Units	Values
$\lambda_w, \lambda_s, \lambda_a, \lambda_i$	Thermal conductivities of water, skeleton of the evaporation material, air, insulating material	$\text{W m}^{-1} \text{K}^{-1}$	0.6, 0.32, 0.0262, 0.043
φ	Porosity of the evaporation material	-	0.955
$c_{p,w}, c_{p,s}, c_{p,i}$	Heat capacities of water, evaporation material and insulating material	$\text{J K}^{-1} \text{kg}^{-1}$	4180, 1800, 871
D	Coefficient of diffusion of water inside the material	$\text{m}^2 \text{s}^{-1}$	10^{-8}
ρ_w, ρ_s, ρ_i	Density of water, skeleton density of the evaporation material, and bulk density of the insulating material	kg m^{-3}	1000, 1550, 30

σ	Stefan-Boltzmann constant	$\text{W m}^{-2} \text{K}^{-4}$	5.67×10^{-8}
R	Universal gas constant	$\text{J mol}^{-1} \text{K}^{-1}$	8.314
M	Molar mass of water	kg mol^{-1}	0.01801
P_0	Standard pressure	Pa	101325
T_0	Boiling point of water	K	373.15
$h_c, h_{c,s}, h_{c,b}$	Heat transfer coefficients at the surface, the side and the bottom	$\text{W m}^{-2} \text{K}^{-1}$	5, 5, 1
E_m, E_s	Emissivities of the material and of the side of the beaker	-	0.8, 0.8

3.3.1 Water activity

Fig. 3 shows the water activity, a_w , as a function of water uptake (wt.%). a_w was obtained from the experimentally obtained moisture sorption isotherm, taken from a previous study [15], and corresponds to the partial vapour pressure of water divided by the standard vapour pressure of water. From the knowledge of the moisture content inside the material, it was possible to obtain the partial vapour pressure of water with the water activity, and thus to calculate the evaporation rate using Eq. 11. Since the evaporation rate is related to the amount of water at the surface, the WES performance is limited if there is not enough water at the surface. At 31% water uptake, the water activity is 1, and remains constant for higher water uptakes.

Fig. 3. Water activity in the tannin foam as a function of water uptake (after data from [15]).

3.3.2 Thermal conductivity

Since the porous evaporation material is partially filled with water, its average thermal conductivity λ_m ($\text{W m}^{-1} \text{K}^{-1}$) must be calculated taking into account the thermal conductivities of the material skeleton, water and air:

$$\lambda_m = \lambda_w \varepsilon_l + \lambda_s \varepsilon_s + \lambda_a \varepsilon_g \quad \text{Eq. 18}$$

where ε_s is the volume fraction of solid in the porous evaporation material, ε_g is the corresponding volume fraction of gas such that $\varepsilon_g = 1 - \varepsilon_s - \varepsilon_l$, and λ_w , λ_s and λ_a ($\text{W m}^{-1} \text{K}^{-1}$) are the thermal conductivities of water, material skeleton and air.

3.3.3 Heat capacity

The mass heat capacity of the material, $c_{p,m}$, varies much with the presence of water but the effect of air on $c_{p,m}$ can be neglected. Therefore, the volumetric heat capacity was calculated as follows:

$$\rho_m c_{p,m} = \varepsilon_l \rho_w c_{p,w} + \varepsilon_s \rho_s c_{p,s} \quad \text{Eq. 19}$$

where ρ_m (kg m^{-3}) and $c_{p,m}$ ($\text{J kg}^{-1} \text{K}^{-1}$) are the density and the mass heat capacity of the simulated material filled with water, respectively, and $c_{p,s}$ ($\text{J kg}^{-1} \text{K}^{-1}$) is the mass heat capacity of the dry, porous, evaporation material.

3.3.4 Vaporisation enthalpy

The enthalpy of vaporisation of water, h_v (kJ kg^{-1}), changes slightly with the temperature according to [16]:

$$h_v = 1.91846 \times 10^3 \left(\frac{T}{T - 33.91} \right)^2 \quad \text{Eq. 20}$$

3.3.5 Numerical model

Calculated variables T and ε_l were obtained using the commercial software COMSOL Multiphysics 5.6. First, a preliminary parametric study on the triangular mesh was carried out in the first study case (*i.e.*, without insulation) for three meshes comprising 2569 elements (mesh #1), 2873 elements (mesh #2), and 5553 elements (mesh #3). Since the impact of mesh size on the results were extremely limited, mesh #2 with 2873 elements was used. **Fig. 4a** shows a mesh consisting of 2873 elements for the first study case and 5431 for the second (**Fig. 4b**), both refined at the interfaces between water and evaporation material, as well as the interface between air and the evaporation material, with 100 elements. The rest of the mesh is created automatically by the software, depending on the physics involved. Using a desktop computer with a quad-core Xeon E5 coupled with 64 GB of RAM, the model for a system without insulation is calculated in 1 minute and 5 seconds, while the model for a system with insulation around the beaker is calculated in around 1 minute and 40 seconds. The simulation required 90 calculation steps, with a step time self-determined by the software.

Fig. 4. Mesh used for the water evaporation system: (a) without insulation (case 1); and (b) with insulation (case 2).

4 Results and discussion

In order to validate the numerical model, the results of the calculations were compared with the results of the experiments obtained with the floating tannin-based foam, for the two cases, i.e., with and without insulation foam. Then, different cases from the open literature were examined with the proposed model, and the model was modified if necessary, taking into account the different parameters considered in these former studies. This includes ambient air temperature, relative humidity, solar power density, convection coefficient, and system geometry. Overall, the objective is not to reproduce the same results from the literature but to compare trends.

4.1 Validation of the model

Evaporation experiments were conducted without and with insulation (i.e., corresponding to case 1 and 2, respectively) around the beaker containing the floating tannin foam, under different solar inputs. To validate the simulation, the external conditions were imported as input (T_a and H_r) and the duration was set to that of the experiment. The sunlight power was set to three different values during the same test: 0.7 sun, then 1 sun and finally 1.3 sun, each for a duration of 3300 seconds for the non-insulated configuration and 3500 seconds for the insulated configuration. Changing the illumination in the middle of the test allows the evolution over time of parameters such as temperature and mass loss to be validated when the power input changes. The results of the simulation were compared to the average of three experiments and are shown in **Fig. 5**.

4.1.1 Non-insulated configuration

Fig. 5a shows the vertical section of the configuration without insulation, where 1, 2 and 3 are the temperature measurement points, and **Fig. 5b** and **Fig. 5c** show the experimental and predicted results.

Fig. 5. Revolution cross-section of the model water evaporation system in the case where this system is (a) non-insulated, and (d) insulated; experimental (circles) and simulated (lines) temperatures for (b) non-insulated and (e) insulated system; and water mass evolution for the (c) non-insulated, and (f) insulated system.

The experimental evaporation rates are equal to 0.51, 0.70 and 1.01 kg m² h⁻¹ with efficiencies of 49, 47 and 52 % for 0.7, 1 and 1.3 sun, respectively. In comparison, the calculated evaporation rates are equal to 0.48, 0.74 and 1.03 kg m² h⁻¹ with efficiencies of 46, 50 and 53 % for 0.7, 1 and 1.3 sun, respectively. The difference between the simulated mass of water and the experimental measurements is no higher than 5% (see **Fig. 5c**). The three different solar inputs are clearly visible on the temperature curves, showing three successive jumps. The agreement between measured and calculated temperatures is better in the evaporation material (point 2) and at the interface between the latter and the water (point 3) than at the surface (point 1) (see **Fig. 5b**). This discrepancy at point 1 is attributed to the setting of the thermal camera (position, emissivity, measurement area) and the estimation of fitted parameters in the model such as the convection coefficient. The average differences

between the simulated and experimental temperatures in the first case study are indeed, from top to bottom, 1.01, 0.69 and 0.44 °C, which shows that the model reproduces the experiment fairly closely.

4.1.2 Insulated configuration

Fig. 5d, e and f are equivalent to **Fig. 5a, b and c**, respectively, the only difference being the layer of insulating material around the beaker. The experimental evaporation rates are equal to 0.56, 0.82 and 1.19 kg m² h⁻¹ with efficiencies of 53, 55 and 61 % for 0.7, 1 and 1.3 sun, respectively. In comparison, the calculated evaporation rates are equal to 0.54, 0.88 and 1.25 kg m² h⁻¹ with efficiencies of 52, 59 and 64 % for 0.7, 1 and 1.3 sun, respectively. The average difference between experimental and simulated masses of water is 8%, so it is higher than in the non-insulated configuration. The differences for the three temperatures were 2.71, 1.31 and 0.64 °C for points 1, 2 and 3, respectively (**Fig. 5d**). As in the previous case (**Fig. 5b**), and for the same reasons, the difference between experimental and simulated surface temperature is higher (point 1 in **Fig. 5d**) than for the other two temperatures, in the bulk (point 2) and at the interface (point 3), which were recorded with thermocouples. However, the evolution of the water mass over time is fairly well predicted by the model for the three illuminations tested, as well as the transient thermal evolution that occurs when the input sunlight changes.

The numerical model has therefore been validated and we can conclude that it can be used to simulate water evaporation in the following. This validation was done on the basis of the results obtained with tannin foam, but this material did not show particularly interesting performances compared to other materials reported in the literature, and for which the evaporation rate were close to 1.3 kg m⁻² h⁻¹ [17]. We now extend the application of this model to these more efficient materials in terms of solar evaporation.

4.2 Study of different cases using the model

Among the large number of evaporation results for different materials and systems in the literature, some phenomena and techniques are discussed repeatedly. The aim of this section is to test the behaviour of our model when applied to the cases seen in the literature. Those cases can be a good source of inspiration for the design of an efficient evaporation system. For all studies considered, unless otherwise specified, the temperature of the air was set at 20 °C with a relative humidity (H_R) of 50 %. The initial temperature of all components was also set at 20 °C.

4.2.1 Impact of ambient conditions

The previous section 4.1.2 has shown the significant impact of the beaker insulation on the system performance, which indicates a dependence of the performance on the ambient conditions. In order to better understand their impact, several parametric studies on the exchanges between the air and the material (i.e., air temperature, relative humidity, convection and solar input) were carried out using the numerical model. The evaporation performance was calculated from the final results, which correspond to a steady state.

4.2.1.1 Air temperature

The first parameter that has an impact on the evaporation rate is the ambient air temperature, which can be easily measured and controlled. In the simulation, we set the relative humidity at 50 % with natural convection, and a parametric study was conducted on the air temperature.

Fig. 6a shows the simulated evaporation rates for different values of air temperature, and the corresponding evaporation efficiency. With increasing temperature, the evaporation rate increases significantly, leading to changes of about $0.15 \text{ kg m}^{-2} \text{ h}^{-1}$ and 10 % efficiency for a temperature change of 10 °C, which may be a reasonable change over a whole day.

Evaporation in the dark is also likely to change, as it is also shown in the same figure for three different temperatures (taken from [4]).

Fig. 6. Simulated evaporation rate as a function of: (a) air temperature at 50% H_R , with the corresponding experimental evaporation in the dark (taken from [4]); and (b) relative humidity of the air at 20 °C. In each case, some data of experimental evaporation rate in the dark are also given: (a) at 3 different temperatures at 50% H_R ; and (b) at 3 different H_R at 23.6 °C (from [4]).

4.2.1.2 Relative humidity

The relative humidity (H_R) is considered in Eq. 11 and it also has an impact on the evaporation rate at 20 °C, which was studied by varying the H_R from 10 to 90 %, with a 10 % step. **Fig. 6b** shows the evaporation rate and efficiency as a function of H_R . Lower H_R increases the evaporation rate and efficiency, for instance the evaporation rate increases from 0.60 kg m⁻² h⁻¹, with an efficiency of 40 % at 90% H_R , to 0.72 kg m⁻² h⁻¹, with an efficiency of 49% at 10 % H_R . Although H_R is a factor that has a significant impact on evaporation rate and efficiency, it is easier to control and can simply be recorded as an ambient parameter. The effect of H_R at 23.6 °C on the evaporation rate in the dark is also plotted for three H_R values for reference (from [4]).

4.2.1.3 Heat exchange with air by convection

Insulating the system increases evaporation performance by increasing the temperature and thus the evaporation rate [18], as shown when validating our model in section 4.1.2. Although convection increases the heat exchange of the system with the air [19], which means that there is more heat loss (Eq. 4, Eq. 14 and Eq. 16), convection also increases the evaporation rate F_m (Eq. 11 and Eq. 12) with or without solar illumination. Here, three types of forced convection, with and without insulation, are tested by adjusting the convection coefficients to three air flow rates. These three flow rates are 0.15, 0.30 and 0.6 m s⁻¹ which corresponds to a heat exchange coefficient at the evaporation surface h_c of 6.8, 9 and 12.4 W.m⁻².K⁻¹. **Fig. 7** shows the surface evaporation rates in the dark and under 1 sun illumination with their corresponding efficiencies as a function of the air flow rate, with and without insulation around the beaker.

Fig. 7. Simulated evaporation rate: (a) without, and (b) with insulation for several air velocities. $F_{m,dark}$ and F_m are evaporation rates in the dark and under 1 sun, respectively. Heat exchange flux (such that losses take negative values) at different material surfaces: (c) without and (d) with insulation, as a function of air velocity.

Forced convection improves water evaporation performance in the dark and in the light, which means that although heat loss increases with convection, the evaporation rate benefits more. Insulation around the beaker also improves evaporation in darkness and light. Despite slightly reducing the slope of the trend line (compare **Fig. 7b** and **Fig. 7a**), beaker insulation gives the same results by increasing convection. As shown in **Fig. 1c** and **Fig. 7d**, the insulation reduces the amount of heat exchange between the material and the bulk water as well as between the material and its side, which is divided by 3 in the insulated case. The

global heat exchange between the air and the evaporation surface is lower in the insulated case than in the non-insulated case. Considering that the solar input is the same, this is consistent with the evaporation results. The more water evaporates, the more heat is transferred to the air, and the less heat reaches the material surface. Overall, insulation can only reduce the heat loss from the sides of the beaker, , which does not have as much impact on evaporation as convection at the surface of the material. Nevertheless, convection may improve evaporation in a large open area when a much larger evaporation surface is used, so it is an interesting factor to consider when evaluating evaporation performance.

4.2.1.4 Solar power and efficiency

One of the most mentioned phenomena in the literature is the increase in evaporation efficiency with increasing solar energy [20,21]. Many papers show results with illuminations above 1 sun as solar energy input [22–28]. **Fig. 8** shows evaporation rate and efficiency results from the literature as a function of solar power concentration. The more power there is on the WES, the higher is the efficiency.

Fig. 8 Evaporation rates (E.R.) and evaporation efficiencies for different solar concentrations, as reported in the literature: (a) and (c) for a wood-based material [20] (reproduced from Ref. [20] Copyright (2017), with permission from Wiley); and (b) and (d) for an improved bamboo-based material [21] (reproduced from Ref. [21] Copyright (2020), with permission from Elsevier).

Based on these articles, **Fig. 9a** shows the simulated evaporation rate and efficiency for our system. The efficiency increases from 49 to 67 % when the illumination increases from 1 to 7 sun and so the evaporation rate increases from 0.7 to 6.9 kg m⁻² h⁻¹. It is important to note that, according to Eq. 1, if the efficiency had remained constant, the evaporation rate at 7 suns would have been 4.9 kg m⁻² h⁻¹ (only 7 times the evaporation at 1 sun). With a 15 % increase in efficiency from 1 to 7 sun for our system, **Fig. 8c** showing a 12 % increase and **Fig. 8d** a 14

% increase for both systems from the literature, the increase in efficiency with the solar power concentration is consistent.

Fig. 9 (a) Simulated evaporation rate and simulated efficiency for different solar illuminations. (b) Evolution of the simulated evaporation rate F_m with surface temperature, according to Eq. 11 and Eq. 13 (blue line), with the results of temperature and evaporation rate at different solar illuminations (black dots).

Natural convection alone cannot be the only cause of the increase in efficiency and, although the enthalpy of vaporisation also changes with the temperature according to Eq. 20, it produces only a slight increase in the evaporation rate. The main cause of the increase in efficiency is the surface temperature, which increases steadily with increasing illumination. As $P_w^{v,sat}$ is a function of temperature (Eq. 13), the latter plays a major role in the evaporation rate (Eq. 11). **Fig. 9b** shows that the evaporation rate increases exponentially with surface temperature while it increased linearly with illumination (**Fig. 9a**).

4.2.2 Increase of the exchange area with air

Several studies have shown that increasing the evaporation area effectively increases the performance of the system [4,7,29]. One of its consequences is a decrease in surface

temperature, due to the fact that sunlight is then scattered over a larger area and the exchange with air is higher, thus reducing the local heating and thus the temperature. This technique has already been explored in the literature [4], as shown in **Fig. 10**, with a carbon-coated paper placed on a polystyrene foam. By increasing the receiving area (**Fig. 10d** and **Fig. 10e**) the evaporation rate also increases (**Fig. 10c**).

Fig. 10 (a) Geometries used in the literature for increasing the exchange area; (b) surface temperature of the flat surface ($\theta = 180^\circ$); (c) experimental evaporation rate for the different surfaces (dots) and the theoretical limit of evaporation at 1 sun (lines); (d) surface temperature when $\theta = 37.8^\circ$; and (e) surface temperature when $\theta = 22.9^\circ$. Reproduced from Ref. [4] Copyright (2018).

Based on this work, we have modified the geometry of our model to represent the increase in surface area in the simulation. A conical geometry, not identical but close to the geometry illustrated in **Fig. 10a**, was chosen for its simplicity to be captured by the axisymmetric model. With such a geometry, we can easily conduct a parametric study on the height of the cone. For that purpose, a variable N has been created, which is written as follows:

$$N = \frac{h}{r} \quad \text{Eq. 21}$$

where h (m) and r (m) are the height and the radius of the base of the cone, respectively. This allows the cone height to be easily fixed in the simulation and to produce a parametric study with integer values. The solar power, initially 1 kW m^{-2} at the base of the cone, is decreased when the tip of the cone is stretched as this increases the area. A coefficient k (dimensionless) is defined to account for the corresponding scattering of the solar power. It represents the increase in surface area when changing from a flat surface to a conical surface and is calculated as follows:

$$k = \frac{S_{base}}{S_{cone}} = \frac{\pi r^2}{\pi r \sqrt{r^2 + (Nr)^2}} = \frac{1}{\sqrt{1 + N^2}} \quad \text{Eq. 22}$$

where S_{base} (m^2) is the area of the base of the cone and S_{cone} (m^2) is the lateral area of the cone. Therefore, the power concentration at the surface of the cone reads:

$$P_m = P_{s,0} k \quad \text{Eq. 23}$$

where P_m (kW m^{-2}) is the surface solar power arriving on the material, and $P_{s,0}$ (kW m^{-2}) is the surface solar power arriving from the sun. A with the decrease in power concentration, the evaporation rate increased, as the evaporation rate is still determined by the flat surface that receives the sunlight (S_{base}). **Fig. 11** shows the conical geometry used in the simulation with the parameters introduced previously, for three values of N . With the increase of the parameter N , k decreases, which reduces the solar power density.

Fig. 11 Conical geometry used in the simulation, here shown for three parameters N : 0 (flat), 2 and 3.

Fig. 12a shows the simulation results of evaporation rates and efficiencies by varying N from $N = 0$ (flat surface) to $N = 8$, applying the model presented here. The temperatures are also shown for each simulated cone. It can be observed that the evaporation rate and the efficiency increase linearly with N while the surface temperature decreases. This observation is contrary to the previous study, which showed that the evaporation flux increases with the surface temperature. It can also be observed that the evaporation rate decreases with increasing the area of the cone (**Fig. 12b**). The calculation of the performance of the system, concerning evaporation, is not related to the exchange area but to the area at the base of the structure. By calculating the flux related to the base, we observe an increase in the evaporated

flux as a function of N and therefore of the exchange area (**Fig. 12b**). It is also observed that the ratio $1/k$ between the exchange area and the area at the base increases linearly with N (**Fig. 12c**), which shows the direct relationship between the evaporation area and the evaporated flux related to the base of the cone. Overall, the increase in evaporation area is beneficial, since it doubles the efficiency from $N = 0$ to 8, passing from 44 to 88%. It should be noted that these efficiencies were calculated using Eq. 1, which does not take into account the heat exchange with the air, but only the solar illumination input and the evaporation rate output.

However, such an improvement could have limitations, for example, a slender geometry with $N = 8$ could be difficult to produce. Another possible geometry could be a recessed spherical area, which might be easier to manufacture [30]. With such a geometry, the surface allows the same kind of reduction of heat loss with air, but also a reduction of radiation loss, since the light beam is partially reflected inside the sphere. Such a system could achieve $1.65 \text{ kg m}^{-2} \text{ h}^{-1}$ for a solar illumination of 1 sun [30].

505 **Fig. 12.** (a) Simulated evaporation rate (left axis) and efficiency (right axis) calculated using
506 Eq. 1, for different values of N ($N = 0$ corresponding to a flat surface), and surface
507 temperature (colour bar code) as a function of geometry change; (b) evaporation rate
508 calculated with the base area as a reference and using the cone area as reference; and (c)
509 variations of k and $1/k$ with the increase of surface area.

4.2.3 Insulation under the material

Due to the natural shape of some materials, it is not possible to use them alone for evaporation and therefore an additional material must be used to maintain the evaporative surface above the water [5,31–35]. This can be a standard foam for thermal insulation, which has the ability to float on water but also allows less heat exchange with water due to its very low thermal conductivity. This was done by Xu et al. [31], whose system is shown in **Fig. 13a**, where a mushroom was placed on a polystyrene foam support. The path taken by the water to evaporate is in the centre of the insulation foam, where a hole has been made to allow the mushroom to touch the water. Similarly, **Fig. 13b** shows another type of WES that has paper wrapped around an insulating foam [36] to allow water to be transported through the sides of the system to an evaporation surface at the top.

Fig. 13 (a) Mushroom-based water evaporation system on a floating polyurethane foam support, reproduced with permission from Ref. [31]. Copyright 2017, Wiley-VCH; (b) a WES using paper around an insulation foam, reproduced from Ref. [36]. Copyright (2016), with permission from National Academy of Sciences.

Inspired by these two uses of insulating foam, we studied two configurations, one where the insulator is located in the core of the evaporation material, with water transport taking place at the sides, and the other where the insulator is located around the material, with water

transport taking place through a hole in the foam. These configurations are shown in **Fig. 14a** and **Fig. 14b**, respectively.

Fig. 14. Simulated temperature of the system with the insulation layer located: (a) in the core, with peripheral water transport, and (b) at the periphery, with a central hole for water transport. Both configurations have the same radius and height ratio of the supporting foam to the evaporation material.

Thus, the evaporation material is positioned on the insulating/floating foam, and water transport is allowed either through a central hole or through the periphery. To study the impact of the foam, parametric studies were conducted on the effects of varying the radius (R_i) and the height (h_i) of the insulating foam. N_r and N_h represent the ratios of the radius and heights of the foam to those of the evaporation material (see Eq. 24 and Eq. 25 and **Fig. 15**).

$$N_r = \frac{R_i}{R_t} \quad \text{Eq. 24}$$

$$N_h = \frac{h_i}{h_t} \quad \text{Eq. 25}$$

where R_t and h_t are the radius and the height of the evaporation material, respectively. The higher N_h and N_r , the larger the foam and the more space it takes under the material. $N_r = 0$

543 with $N_h = 0$ thus correspond to the case without insulation layer. The variations of N_r for the
544 two configurations are represented in **Fig. 15**.

545 Adding an insulation layer under the evaporation material should theoretically improve
546 performance as more heat is retained on the surface. However, the addition of insulation
547 limits the transport of water to the surface, so a balance must be struck between heat and
548 water management. The simulation shows that increasing the size of the insulation, both in
549 radius and in height, increases evaporation to a certain point, where it reaches its maximum,
550 and then water transport limits the performance. The diffusion coefficient of water in the
551 evaporation material plays an important role in the dimensions of the insulation material that
552 can be used.

Fig. 15. Top: Cross-section of revolution of the evaporation system shown in **Fig. 14b** with insulation at the periphery for: (a) $N_r = 0.1$ and $N_h = 0.6$, and (b) $N_r = 0.9$ and $N_h = 0.6$; (e) system with its notations, and (f) results of evaporation rate for different values of N_r and N_h for insulation at the periphery. Bottom: Cross-section of revolution of the evaporation system shown in **Fig. 14a** for: (c) $N_r = 0.1$, and $N_h = 0.6$, and (d) $N_r = 0.9$ and $N_h = 0.6$; (g) system and its notations, and (h) results of evaporation rate for different values of N_r and N_h .

Fig. 15a, b and **e** show the simulation performed when insulation at the periphery was considered. **Fig. 15f** shows the increase in performance with the values of N_h and N_r , up to certain points where the insulation takes up too much space and therefore not enough water can be transported to the upper surface, thus limiting the evaporation rate. **Fig. 15c, d** and **g**

show the case of insulation in the centre. In this case, as shown in **Fig. 15h**, increasing the size of the insulating foam in both directions increases the evaporation, and as in the first case, up to a certain maximum after which it no longer increases. However, in this case, the evaporation rate starts to decrease at higher values of N_h and N_r . This is due to the peripheral transport of water, which spreads over a larger area compared to the hole in the centre. Therefore, to obtain the same kind of proportion between the volume of the evaporation material and the volume of the insulator, the radius of the insulating foam must be higher. Nevertheless, this geometry, with a thin layer around the insulator, is limited by the possibilities of preparation. A sheet of paper, like the one presented above, is feasible but a foam-like product like our tannin foam would be impossible to cut so thin. Overall, the peripheral insulation with $N_r = 0.9$ and $N_h = 0.4$ achieves an evaporation rate of $0.87 \text{ kg m}^{-2} \text{ h}^{-1}$, while the central insulation with $N_r = 0.95$ and $N_h = 0.9$ gives $0.85 \text{ kg m}^{-2} \text{ h}^{-1}$.

These observations may be clues to making evaporation systems more efficient, since the thermal conductivity of evaporation materials is strongly influenced by the amount of water inside their porous structure. Having an additional insulating layer to prevent heat exchange with water results in a higher evaporation rate, when water supply to the surface is not a problem. However, this can also be an obstacle when comparing materials. Again, it is essential to consider the whole system, with all the parameters and appropriate geometry, to study its evaporation efficiency.

5 Conclusion

A two-dimensional axisymmetric model, validated for temperature and water mass evolutions by experiments on a solar water evaporation system based on tannin foam, has made it possible to study the impact of different operating parameters on the performance of this type of system. As present, the tannin foam is of no interest in this type of system. This is

due to the fact that the open porosity of the foam is saturated with water, which considerably limits its insulating properties.

We have shown that air temperature, relative air humidity and the type of air convection around the system all have an impact on the evaporation rate. When the air temperature increases from 20 to 50 ° C, the evaporation rate is doubled, while a large change in relative humidity has very little impact on performance. In addition, a system where the beaker is insulated increases the performance of our material by 15 % on average, and increasing the convection at the evaporation surface significantly increases the evaporated flux. This study on the experimental conditions shows the need to approach the experimental conditions recommended by Li et al. [10] with the air temperature of 25 °C and the relative air humidity of less than 60 %, all in a breeze-free environment. Additionally, the cone study shows that increasing the exchange area allows the evaporation rate of the system to be increased, which is a considerable potential improvement for future development.

Finally, the addition of a thermal insulation layer to the evaporation material, with two possible positions for the water transport channel, either at the centre or at the periphery of the axisymmetric system, proved effective, providing good thermal insulation to prevent energy from going to the water while not impeding water transport. With thermal insulation at the periphery, N_r of 0.9 and N_h of 0.4 are the best parameters, while N_r of 0.95 and N_h of 0.9 is the optimal combination when the insulation is in the centre of the system.

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