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Numerical study of the cooling of hot dry air by forced convection of a water film deposited on a cylindrical wall

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Abstract - *The costs of space cooling using conventional energy sources are prohibitive in Sahelian zones. We present a numerical study of the cooling of hot, dry air by forced convection through a cylindrical tube thermally insulated on its outer wall and bearing on its inner wall a thin absorbent cloth, constantly saturated with water. In hot and dry climate (as in Burkina Faso), we can (and it is useful to) lower the temperature of a water film [1], [2], [3], or a stream of air through evaporation with the use of low-cost systems. We present the results of a study of optimization of the production of cold moist air, according to the dimensions of the tubes.*

I. INTRODUCTION

The topic of this study involves the phenomena of simultaneous heat and mass transfer. These transfers take place between a liquid film distributed on a cylindrical wall and a flow of hot dry air. Then we have to study a complex phenomenon: the evaporation of a liquid film in a stream of air whose temperature and humidity vary gradually as it goes through the tube. Several studies have been conducted on this topic with different geometric configurations; R. Mansour et al. [4] conducted a numerical study of evaporation through natural convection of a liquid film flowing down an inclined wall, the results show that natural mass convection is predominant if proximal to the upper edge of the plate and increases with the angle of inclination (of the plate) from the horizontal. M. Feddaoui et al. [5] study the cooling through evaporation of liquid film placed in tubes and subjected to an air flow in mixed convection and in laminar flow; they show that the temperature at the air water interface decreases with the Reynolds number. The largest drop in temperature is observed with liquids with a high inlet temperature or low flows. J. Kou et al. [6] have measured surface temperature at the air-water interface which produced mass and heat transfer in mixed convection, using an infra-red camera (IR) for average wind speeds ranging from 0 to 4.0 $m.s^{-1}$, with an increment of 1.0 $m.s^{-1}$ / s... After statistical processing of the data obtained, they show that the average temperatures measured are closely dependent on the relation $Ra^{1/3}/(Re^{4/5} \times Pr^{1/3})$ in which Ra is the Rayleigh number, Re, the Reynolds number based on friction velocity, and Pr is the Prandtl number. [7] S. Ben Jabrallah et al. study heat and material transfer accompanying the evaporation of a film of water falling into a closed rectangular cavity with a factor of geometric shape equal to 10. The adhesion of the water film on the wall is ensured by the presence of a very thin cloth. The wall that supports the liquid film is heated at constant flow. A method based on mass and heat balances, obtained by integrating the equations of conservation of matter and energy on small increments in height, was developed. This method avoids the explicit resolution of the equations of momentum and energy. Exchanges in the liquid-gas interface are then characterized by the Nusselt number and Sherwood number.

We present a model of mass and heat transfer that is simple, in which the inner wall of the tube is covered with a fine fabric bearing the water film, and the outer wall insulated with a polystyrene cylinder. The latter is supplied with a small wick dipping into a small pan of water (Fig. 1).

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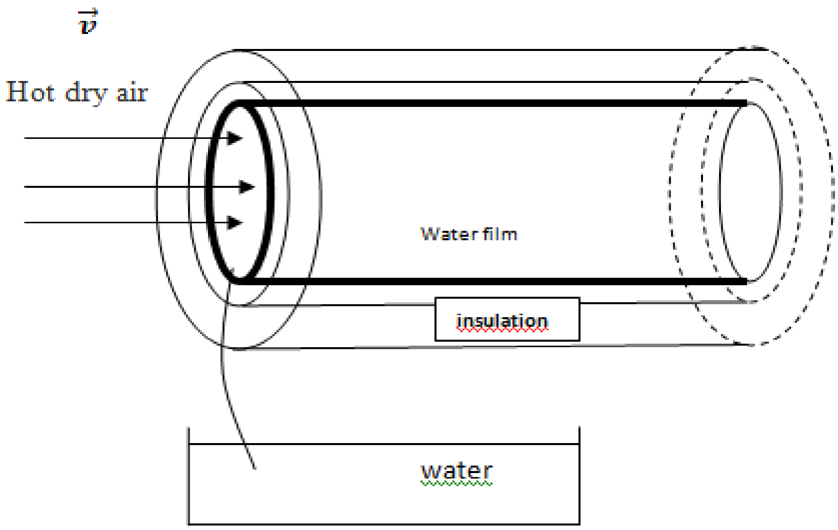

Fig. 1: Diagram of studied system

II. ANALYSIS

The thermal comfort zone is defined in Fig. 2. It is observed that the temperature of the wet bulb, corresponding to a volume of air whose temperature varies between 30 and 40 ° C, for a relative humidity between 10 and 40%, are still below 18 ° C. This area is therefore favorable for the development of an evaporative cooling system. In a hot and dry climate, the area of classic comfort (Givoni diagram) can be extended to the dashed area when we have at least a wind speed of 1 ms-1.

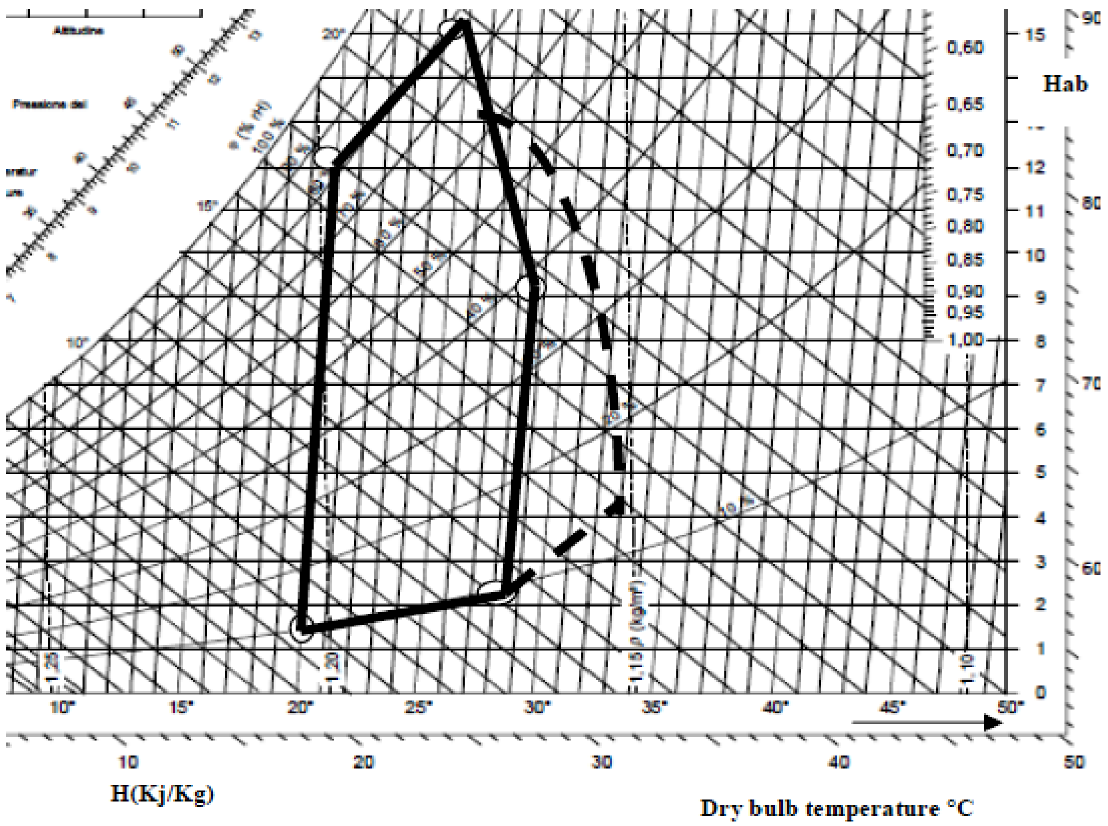

Fig. 1: Zones of thermal comfort in Sahelian climate

2.1. Assumptions

- To model our system, we use the following assumptions:
- It is considered that the tube wall is evenly wet throughout the process. The amount of water that evaporates is negligible compared to the volume of the water film.
- The system is cut into infinitely small slices according to the axis of the tube
- The temperature of incoming air in a slice is equal to the temperature of outgoing air in the previous installment.
- To assess heat balances, electric analogy is used.
- The temperature of the wall in a slice is uniform and constant.
- The average coefficients of heat and mass transfer are calculated for a film temperature representing the arithmetic average temperature between the wet bulb temperature and dry bulb temperature.

2.2. Physical model

Electrical analogy

We take a slice of our system that we cut along the axis of the tube. Fig. 3 shows the heat exchange between the hot dry air, the water film, the wall and the insulator.

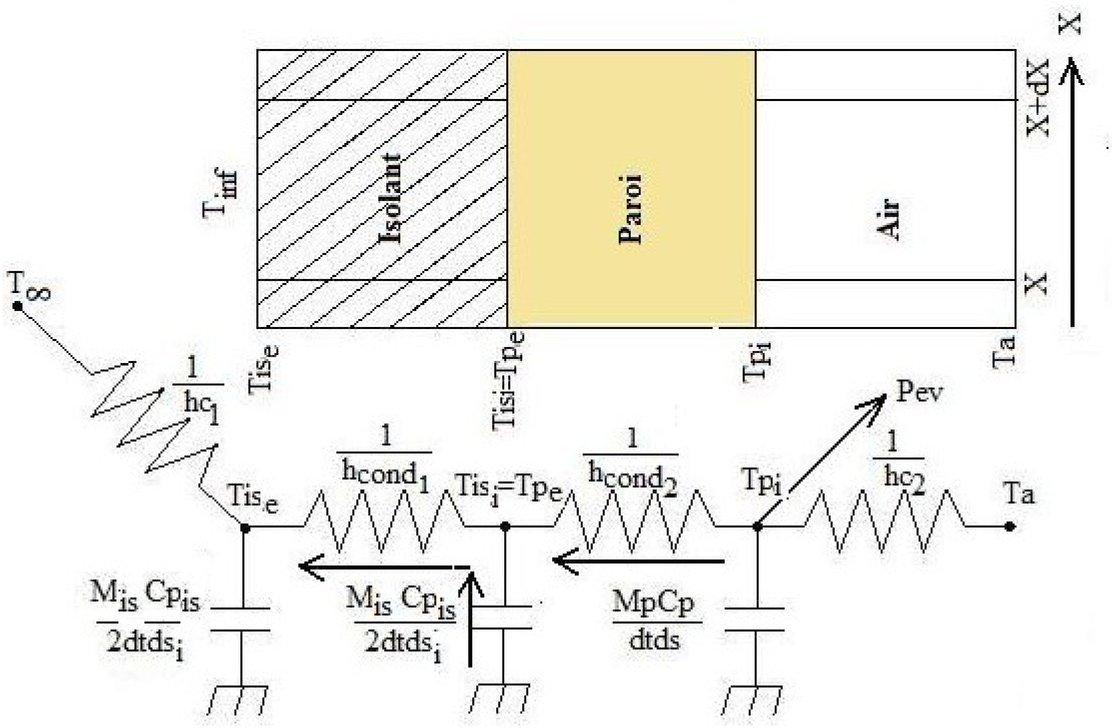

Fig. 2: figure of electrical analogy

2.3 Model of equations

The equation expressing the thermal balances are discretized by the method of finite differences. In these equations, n represents time and j space.

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a) Thermal balance in the upper half of the wall of isolation

$$\frac{M_{Pis}C_{Pis}}{2ds}\left(\frac{T_{ise\,j}^{\,n+1}-T_{ise\,j}^{\,n}}{dt}\right)=h_{c1}(T_{\infty}-T_{ise\,j}^{\,n+1})+h_{cond1}(T_{pe\,j}^{\,n+1}-T_{ise\,j}^{\,n+1}) \quad (1)$$

b) Thermal balance in the lower half of the wall of isolation or the upper half of the tube

$$\frac{M_{Pis}C_{Pis}}{2dsi}\left(\frac{T_{pe\,j}^{\,n+1}-T_{pe\,j}^{\,n}}{dt}\right)=h_{Cond1}(T_{ise\,j}^{\,n+1}-T_{pe\,j}^{\,n+1})+h_{cond2}(T_{pi\,j}^{\,n+1}-T_{pe\,j}^{\,n+1}) \quad (2)$$

c) Thermal balance for the wall of the tube

$$\frac{M_{p}C_{p}}{2ds}\left(\frac{T_{pi\,j}^{\,n+1}-T_{pi\,j}^{\,n}}{dt}\right)=h_{c2}(T_{a\,j-1}^{\,n+1}-T_{pi\,j}^{\,n+1})-h_{cond2}(T_{pi\,j}^{\,n+1}-T_{pe\,j}^{\,n+1})-\frac{\dot{m}_e L_V}{ds} \quad (3)$$

The equation expressing the evaporated mass flow is therefore:

$$\dot{m}_e=\frac{kds}{R_v.T_{pi\,j}^{\,n+1}}[P_{vs\,j}^{\,n+1}-P_{va\,j}^{\,n+1}] \quad (4)$$

d) Thermal balance of air going from slice j-1 to slice j

$$Ga(Cp_{as}+W_j^{n+1}.C_{pe})(T_{a\,j}^{\,n+1}-T_{a\,j-1}^{\,n+1})=-h_c 2.\pi.r.dx(T_{a\,j-1}^{\,n+1}-T_{pi\,j}^{\,n+1}) \quad (5)$$

e) Mass transfer in the air

$$Ga\left[\frac{W_j^{n+1}-W_{j-1}^{n+1}}{\Delta X}\right]=\frac{kds}{\Delta X.R_V.T_{pi\,j}^{\,n+1}}[P_{vs\,j}^{\,n+1}-P_{va\,j}^{\,n+1}] \quad (6)$$

2.5. Calculation of transfer coefficients

- The coefficient of convective mass exchange between the air and the wall can be calculated from the Sherwood number given by the expression below in a turbulent mode.

$$Sh=0.023R_e^{0.83}Sc^{0.33} \quad (7)$$

- The coefficient of convective heat exchange between air and the inner wall can be calculated by the equation of Colburn, below, since it is ($3000 \le \mathrm{Re} \le 9000$).

$$Nu=0.023\mathrm{Re}^{0.80}\,\mathrm{Pr}^{0.33} \quad (8)$$

III. RESULTS

Our simulations are done using a tube with an aspect ratio F so that 0.002 <F = d / L <0010. Naturally for an aspect ratio fixed form (0.002), there is a drop in temperature that increases as the humidity of the air is low, Fig. (4a) or Fig. (4b) and that the ambient temperature is high.

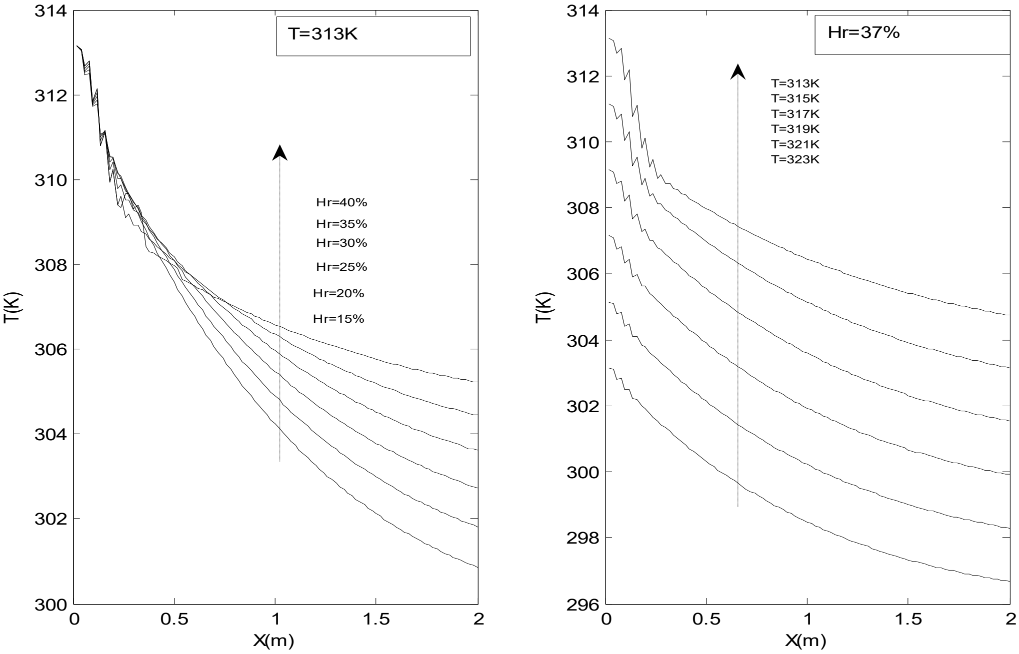

Fig.3: Variation in air temperature in the tube as a function of wet versus dry temperature

There are naturally fluctuating temperatures in the entrance area of the tube. The minima of air temperature are even lower when the Reynolds number is higher, within the selected range, which is consistent with the correlations of Nusselt number and Sherwood number (Fig. 5). But we notice that as soon as there is a turbulent regime, the increase in the Reynolds number does not have much effect on minima of temperatures.

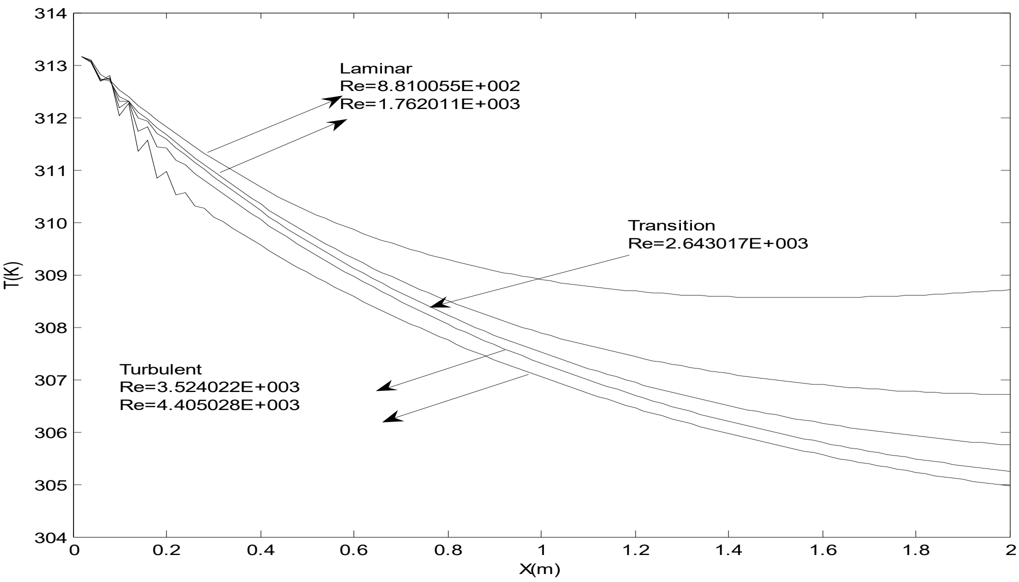

Fig. 4: Space evolution of the air temperature as a function of Reynolds number

It is observed that the air temperature decreases in the tube as long as it can transfer its sensible heat to the benefit of the latent heat coming from the water film. But for a given diameter, there is an optimum tube length to meet if we want to get an output of minimal temperatures. This optimum shifts slightly towards the positive x over time. Beyond this, the cold moist air cannot cool the wall with the liquid film. Then this latter transmits its latent heat without losing heat. Of course, there is an increase in temperature.

In a permanent state, the minimum in the figure of temperature remains above that of the wet bulb temperature (300K) in the case of Fig. 6.

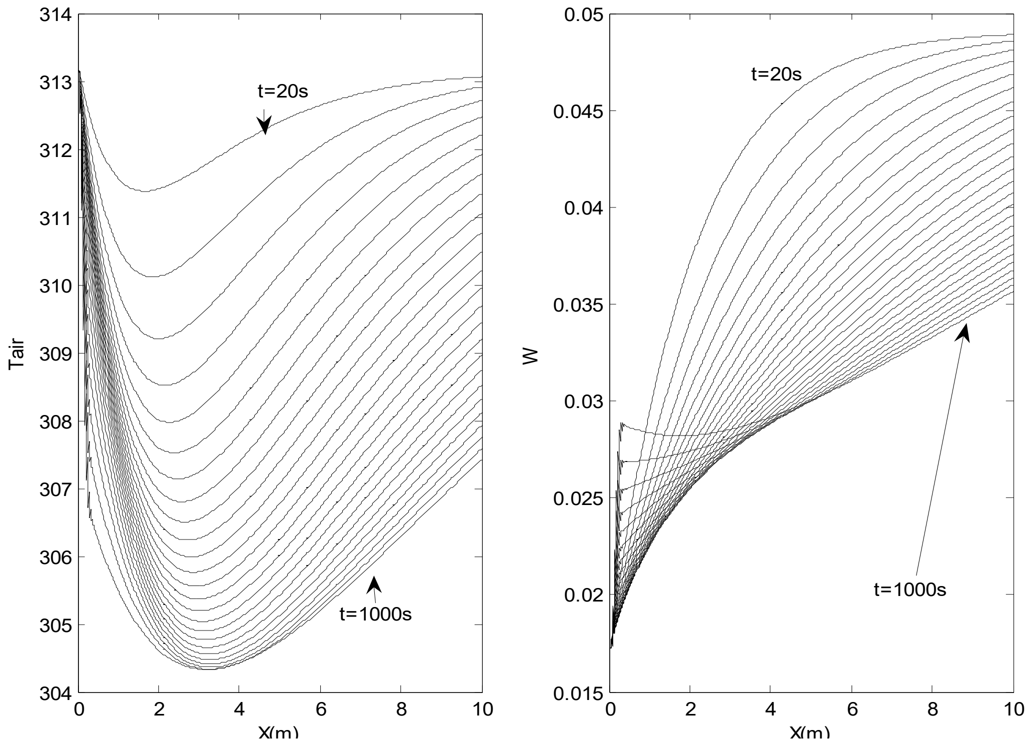

Fig. 5: Spatio-temporal evolution of temperature and humidity of the air in the tube: Re = 3.24 x 103, Ta = 40 ° C, RH = 37%, Th = 27 ° C

We observed on the curves of moisture around the permanent state, a sharp increase of moisture in the air at the entrance. This can be explained by the relatively large difference in temperature between the hot and dry air that reaches the wall that is already cold (Fig. 7).

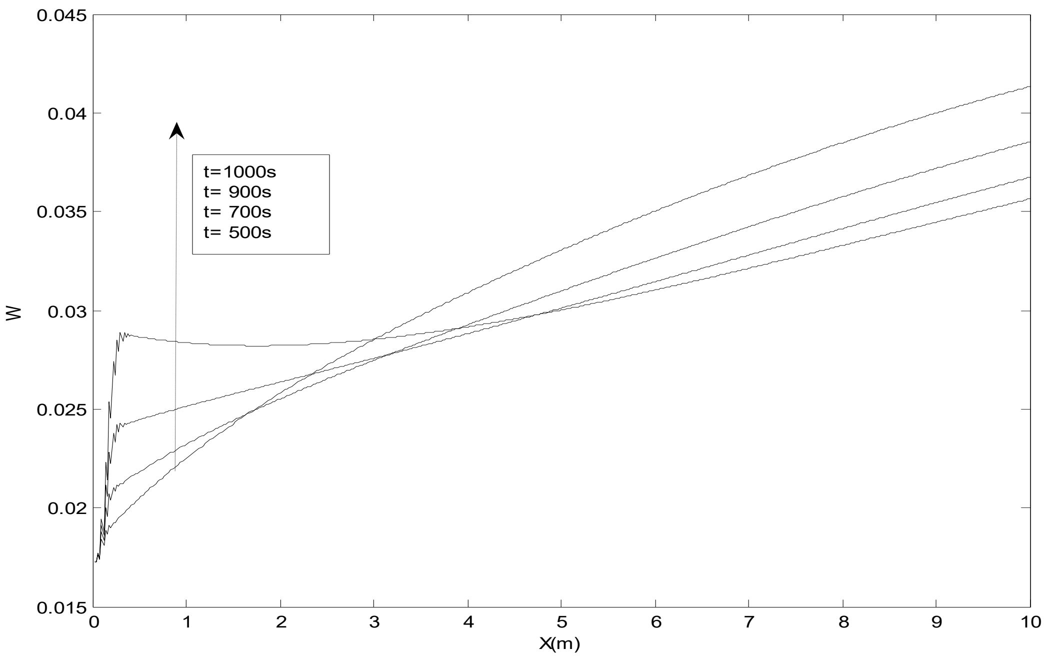

Fig. 6: Effect of the entrance of hot dry air over time Re = 3.24 x 103, Ta = 40 ° C, RH = 37%, Th = 27 ° C

In general, the curves of the amounts of humidity of the air decrease over time since the water film on the wall cools, reducing the evaporative power of the air.

Influence of tube diameter

We can define a physical quantity for the system, which takes into account the exchange surface, relative to the volume of air introduced. This quantity is the compactness which is the ratio of the exchange surface and the volume of air C = Se / Vair.

When representing the compactness as a function of the radius (Fig. 8) we see that it decreases exponentially when the radius of the tube increases.

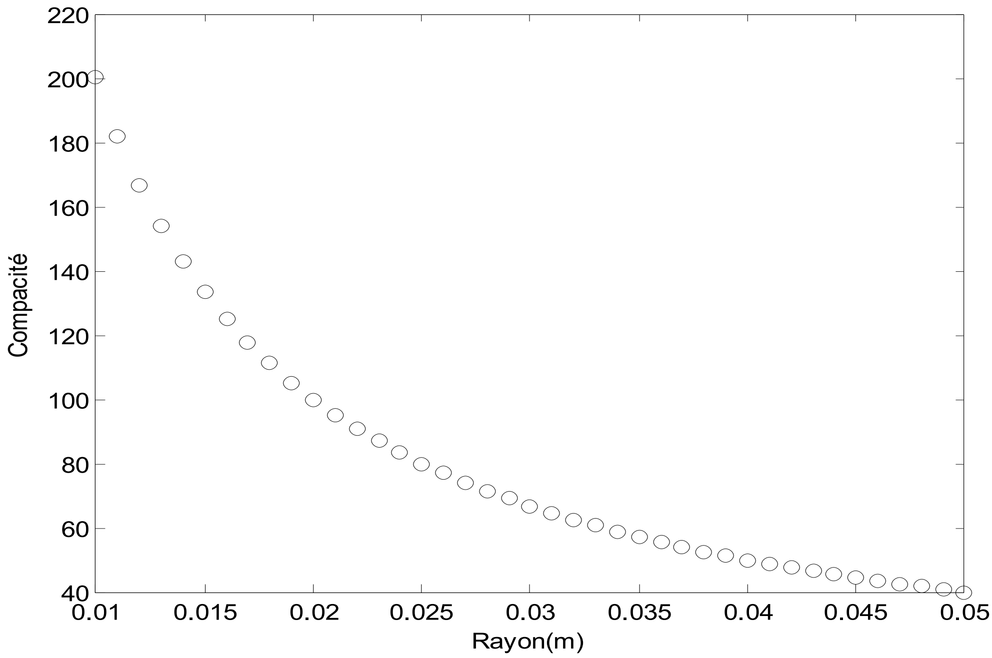

Fig. 7: Variation of compactness as a function of radius

For large compactness (small diameter), the volume of air involved is relatively small compared to the exchange surface. This small volume of air then quickly releases its sensible heat, which justifies the faster appearance of temperature minima with small diameters (Fig. 9).

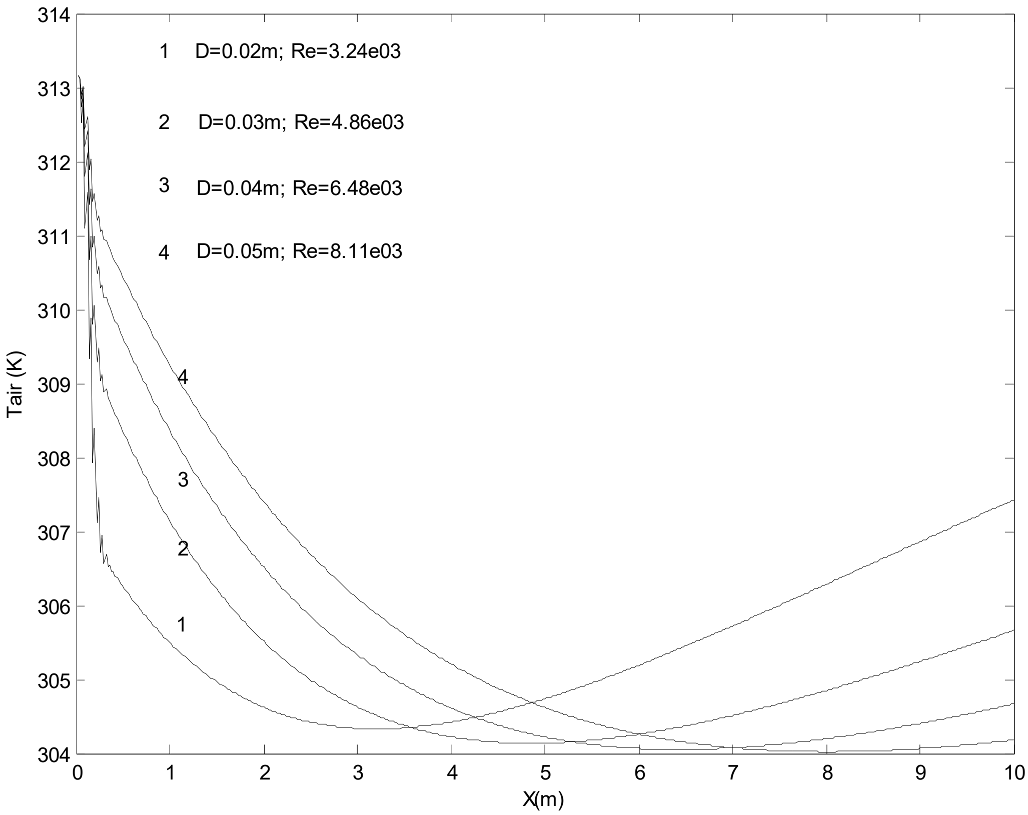

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Fig. 8: Evolution of air temperature in the tube as a function of the diameters

Ta = 40 ° C; Vair = 3m.s-1, RH = 37%, Th = 300K

The minima of temperatures are reached at the end of shorter routes with smaller diameters. But the lowest temperatures are obtained with larger diameters. We can then optimize the outlet temperature of our heat exchanger.

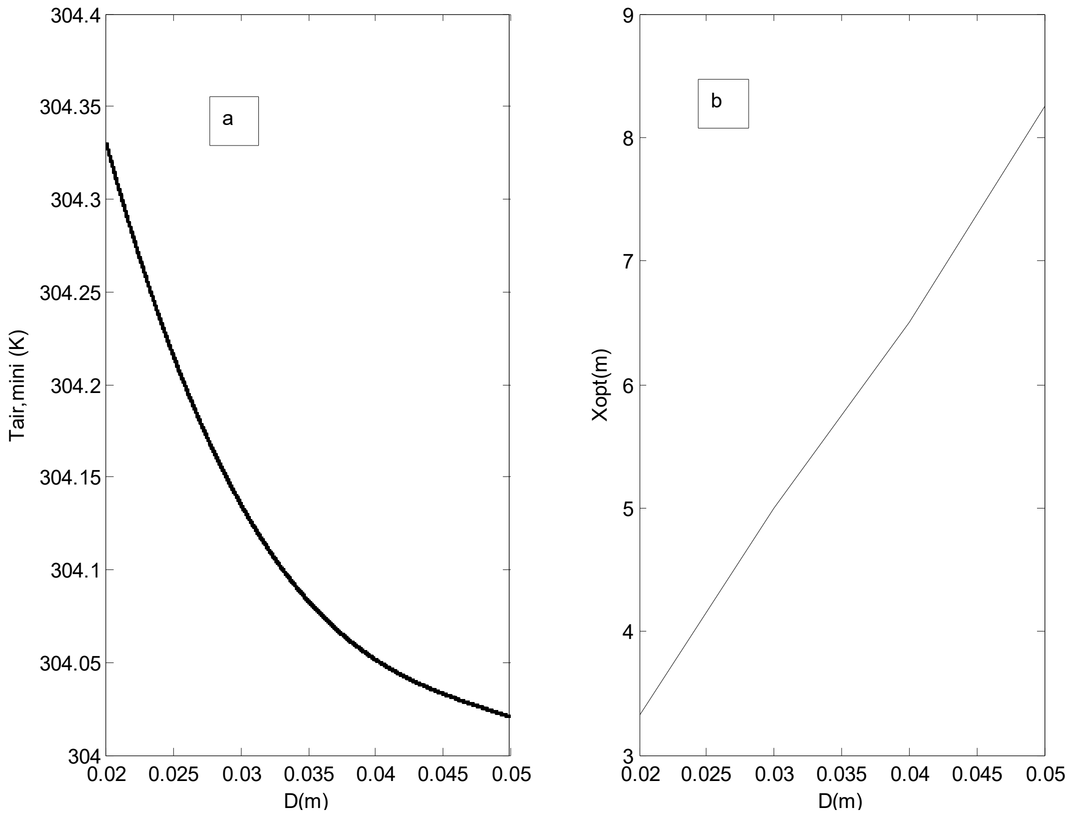

Fig. 9: a) Evolution of the minima of air temperature as a function of diameter:
Ta = 40 ° C, RH = 37%, Th = 300K
b) Evolution of the optimal distance of air cooling according to diameter:
Ta = 40 ° C, RH = 37%, Th = 300K

The minima of temperature remain very proximal, regardless of the selected tube diameters. Nevertheless, we notice a slight drop in temperature when the diameter increases within the limits of the selected range. Indeed, the compactness of the systems of large diameter pipes leads to larger evaporation capacities, and therefore a higher sensible heat transfer leading to lower temperatures. All these results are consistent with those in Fig. 10.

IV. CONCLUSION

The production of cold by evaporation in Sahelian climate (hot and dry) is used to design effective and simple systems of weather conditioning. Indeed one can ideally reach the wet bulb temperature. Actually decreases in temperatures remain below those of the wet bulb but belong to the comfort zone. The air is initially very dry. When it moistens, it helps to improve thermal comfort. The results show that it falls easily to 8 ° C below room temperature in less than ten minutes, with relatively low Reynolds numbers and is based on the performance of a thermal exchanger with the optimal length determined in Figure 4. The simulation results presented in this work, can justify the construction of a tubes heat exchanger with moistened walls, with a low power fan at the entrance. This type of exchanger should effectively contribute to decrease the energy cost of air conditioning.

NOMENCLATURE

$\dot{m}_e$	Mass flow of water	(kg .s^{-1})
C	Compactness (Se / VT)	(m^{-1})
C_P	Specific heat at constant pressure	(J.kg^{-1}.K^{-1})
$C_{P\,is}$	The specific heat at constant pressure, isolating	(J.kg^{-1}.K^{-1})
C_{Pi}	Specific heat at constant internal pressure	(J.kg^{-1}.K^{-1})
h_c	Coefficient of heat transfer by convection	(W.m^{-2} K^{-1})
h_{cond}	Coefficient of heat transfer by conduction	(W.m^{-2} K^{-1})
P_{va}	Partial pressure of steam room	(Pa)
P_{vs}	Saturating vapor pressure	(Pa)
K	Coefficient mass transfer	$k = \frac{Sh.D}{d}$
Ga	Air mass flow rate	Kg.s^{-1}
L_v	latent heat of vaporization	(J.kg^{-1})
M_p	Wall half mass	(Kg)
Nu	Average Nusselt number	$h = \frac{Nu.\lambda_{air}}{d}$
Mp_{is}	Mass of the insulating wall	(Kg)
Pr	Prandtl number	$\Pr = \frac{\nu}{\alpha}$
Re	Reynolds number	$\mathrm{Re} = \frac{Vd}{\nu}$
Rv	Perfect gas constant relative to water vapor	Rv = R / M
Sc	Schmidt number	$Sc = \frac{\nu}{D}$
Sh	Number of Sherwood	$Sh = \frac{kd}{D}$
T	Temperature	(°C)

Greek symbols

α	Thermal diffusivity	m².s^{-1}
λ	thermal conductivity	(Wm $^{-1}$ K $^{-1}$)
λ_m	Equivalent thermal conductivity	(Wm^{-1} K $^{-1}$)

Indices

∞	infinite
a	ambient
c	convection

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cond	conduction
e	water
is	insulation
ise	outer insulation
p	wall
pe	external wall
pi	Internal wall

REFERENCES

[1] S. Kam, J. Bathiébo, A.O. Dissa, K. Traoré, and L. Aurélien Numerical study of cooling water via an evaporatively cooled immersed porous tube, 2008, *International Journal of Ambient Energy, volume 29, Number 3, pp 115-125*

[2] S. Kam[1], J. Bathiébo[2], L. Zerbo[3], J. Soro[4], K. Traoré[5], P. Blanchart[6] and F. Kieno[7].
Experimental determination of some micro-structural and hydrous properties of a terra cotta and modelisation of hygrothermal transfers**.** 2009, *Journal of Applied Sciences 9 (19): 3424-3432, 2009*

[3] S. Kam[a], L. Zerbo[b], J. Bathiebo[a], J. Soro[c], S. Naba[d], U. Wenmenga[d], K. Traoré[b], M. Gomina[e], P. Blanchart[c] , Permeability to water of sintered clay ceramics
2009, *Applied Clay Science, Vol.46,4, pp 351-357*

[4] Rabii Mansour, Xavier Chesneau et Belkacem Zeghmati, Etude numérique de l'évaporation en convection naturelle d'un film liquide, ruisselant sur une paroi inclinée
Tanger, Maroc du 15 au 17 Novembre 2005 ;

[5] M'barek Feddaoui, El Mustapha Belahmidi ∗, Ahmed Mir, Abdelaziz Bendou, Numerical study of the evaporative cooling of liquid film in laminar mixed convection tube flows; *Int. J. Therm. Sci.* (2001) 40, 1011–1020

[6] J. Kou, K.P. Judd, J.R. Saylor, The temperature statistics of a surfactant-covered air/water interface during mixed convection heat transfer and evaporation Original Research Article *International Journal of Heat and Mass Transfer*, *Volume 54, Issues 15-16*, *July 2011*, *Pages 3394-3405*

[7] S. Ben Jabrallah, A. Belghith and J.P. Corriou - Convective heat and mass transfer with evaporation of a falling film in a cavity
International Journal of Thermal Sciences
Volume 45, Issue 1, January 2006, Pages 16-28.