

Characterization of a cold battery with iced water and double heat exchanger in a humid mode.

[1]B. Dieng and [2]G. Jaw

[1]University of Bambey, UFR SATIC, UB, Physics Section, Bambey, Senegal

[2] Regional Maritime University, Electrical Engineering Department Accra Ghana

Abstract :

The cold battery is a heat exchanger between two fluids, air (secondary fluid) and iced water (primary fluid).

The cold battery is composed of two heat exchangers in series, one of which is made up of flat-plate in galvanized steel serving as a reservoir for the iced water and the other one a copper shell-and-tube exchanger with aluminum cooling blades. The two heat exchangers a connected pipe of the same diameter. These pipes will permit the transit of the iced water coming from the flat-plate exchanger by gravitation towards the tubes of the second exchanger. These two heat exchangers are incorporated in a galvanized container coupled with a centrifugal fan for the improvement of the thermal comfort. The water, after passing through the two heat exchangers is stored in an adiabatic reservoir and will serve as a water fountain [1].

The modeling will be done in a humid mode that is the temperature of the surface of the battery is very low compared with the dew temperature of air. The cooling allows the condensation of water vapor [2].
Key words: Battery, Heat exchanger, Temperature, Humidity, Condensation, Cross flow, Counter flow

NOMENCLATURE

Symbol	Description	Unit
A	Surface	m^2
c_{pw}	Specific heat of water at constant pressure	$J.kg^{-}1.K^{-1}$
c_{pa}	Specific heat of air at constant pressure	$J.kg^{-}1.K^{-1}$
c_{pv}	Specific heat of water vapour at constant pressure	$J.kg^{-}1.K^{-1}$
C	Heat Capacity	J/kg ℃
h	Specific Enthalpy	J/kg
HR	relative humidity	%

Lv	Latent heat of vaporization of water	J/kg
$\dot{m}_e$	Flow rate of water	kg/s
$\dot{m}_a$	Flow rate of air	kg/s
Psat	Saturated Pressure	Pa
U	Convective heat transfer coefficient	$W.m^{-2}.K^{-1}$
w	Specific humidity of air	kg/kg as
θ	Temperature	°C
Φ	Heat flux	W

Indices :

a : air

ai : air flowing into the cold battery

ao :air flowing out of the cold battery

cond: relative to the condensation film

wi : water flowing into the cold battery

wo : water flowing out of the cold battery

ext or e : relative to the air side of the heat exchanger

int or i : relative to the water side of the heat exchanger

nom: relative to the nominal operating point

sat: relative to saturated air, by extension the fictitious point representing water.

r: relative to the dew point of air

t : relative to tube

p : relative to the flat-plat

I. Introduction

The axial suction of air by the fan passes in crossflow the shell-and-tube heat exchanger in which flows an iced water before being pushed radially by the fan under the flat-plate containing the iced water. The global heat transfer is the sum of the heat exchanged by the heat exchangers of the cold battery.

The modeling will be done in a humid mode, when the surface temperature of the battery is very low compared with the dew temperature of air. The cooling process will cause the water vapour to condense.

The battery is conceived in such a way that the flat-plate heat exchanger operates in in counterflow and the shell and tube. Heat exchanger in crossflow, these characteristics will be determined by the h_{lm} and NTU-ε techniques [3;4] and for a given operating point.

Therefore the apparatus we have built is composed of the following elements:

- Two heat exchangers (flat-plate and shell-and –tube) in series and connected by pipes.
- A centrifugal fan
- An insulator in Polythene
- A frame in galvanized steel
- A water reservoir in steel
- Two valves

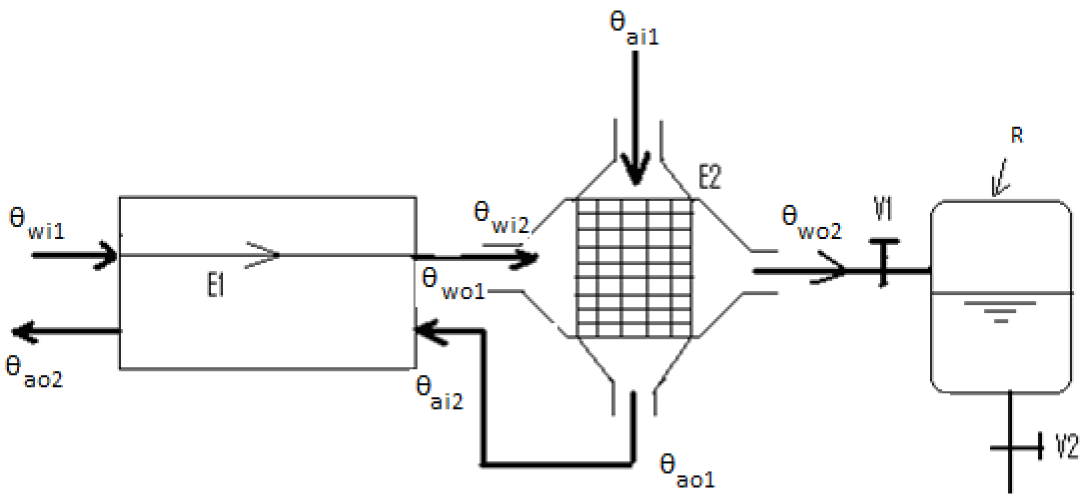


Figure 1: Cold battery with two heat exchangers

E1: Flat-plate heat exchanger

E2: Tube heat exchanger

V1 : Flow rate regulator valve

V2 : Inlet valve to the water fountain

R : Water reservoir

II. HYPOTHESIS

• The variables (fluid temperatures) are mainly dependent on their axial position.

• The heat capacities of the two fluids are constant during the transformations.

• The heat transfer processes are not convective,

• The heat exchanger is thermally isolated from the external environment.

• There are no heat losses during the transfer of fluid between the two heat exchangers.

• We assume that the enthalpy of the saturated air is a linear function of temperature in the intervals considered

III. MODELLING IN HUMID MODE

The aggregated method of transfer of heat and mass in an enthalpy change between air and water is used. This technique aims substituting the two thermodynamic forces with only one force derived from the enthalpy of humid air. It becomes then possible to apply the calculation techniques developed for the heat exchangers by use of this unified transfer of heat and mass [3].

III. 1 Heat transfer between air and condensation film

The heat transfer between the air and the condensation film is given by:

$$d\phi = \frac{U_{te}.dA_e}{C_{pa}}.\left(h_a - h_{cond.sat}\right) \qquad 1$$

III.2. Heat transfer between the condensation film and water

If we neglect the condensation film because of the high conductivity of the materials used and the water, compared with the convective processes [4, 5], the heat transfer between the condensation film and the water can expressed as:

$$d\phi = \frac{U_{ti}.dA_i}{C_{psat}}\left(h_{cond.sat} - h_w\right) \qquad 2$$

III.3. Local heat transfer between air and water

Considering the expression of the heat flux in the different heat transfers air - condensation film- water, we design then a direct heat exchange between air and water and take into consideration the expression of the two resistances in series , one resistance being the convective transfer between air and the condensation film (U_e) and the other the convective transfer between the battery and the water (U_i).

The global conductance global can then be written as by taking into consideration the heat transfer conductance and the surfaces of exchange:

$$\frac{1}{U\ .dA} = \frac{C_{pa}}{U_{te} dA_e.} + \frac{C_{psat}}{U_{ti}.dA_i} \qquad 3$$

dA is the Log-Mean Temperature Difference.

Then the heat transfer is affected by two resistances:

- A resistance to the convection between the internal surface of the battery and the fluid (1)

- A resistance to the convection between the external surface of the battery and the fluid (2)

In the case of the heat exchangers with the condensation film, the surface areas of convection are identical [6] therefore the equation 3 can be written as:

$$\frac{1}{U} = \frac{c_{psat}}{U_{ti}} + \frac{c_{pa}}{U_{te}} = \frac{1}{U_i} + \frac{1}{U_e} \qquad 4$$

The units of enthalpy change coefficients U_{ti} and U_{te} are in kg/s

IV. Determination of the overall heat flux.

With the hypotheses of calculation, the overall heat flux will be the sum of flux Φ_1 and Φ_2 because the two heat exchangers are in series

$$\phi = \phi_1 + \phi_2 \qquad 5$$

Φ_1 : heat flux due to the shell-and-tube exchanger operating in crossflow

Φ_2 : heat flux due to the flat-plate exchanger àoperating in counterflow

The overall flux which is the sum of the two flux is then given by ::

$$\phi = U_t A_t F \frac{(h_{ai1} - h_{wo2}) - (h_{ao1} - h_{wi2})}{\ln\left(\dfrac{h_{ai1} - h_{wo2}}{h_{ao1} - h_{wi2}}\right)} + U_p A_p \frac{(h_{ao2} - h_{wi1}) - (h_{ai2} - h_{wo1})}{\ln\left(\dfrac{h_{ao2} - h_{wi1}}{h_{ai2} - h_{wo1}}\right)} \qquad 6$$

The correction factor F linking the crossflow and the counterflow is given by charts or correlation equations.

The overall flux can be written as:

$$\phi = U\ A\ Y \frac{(h_{ao2} - h_{wi1}) - (h_{ai1} - h_{wo2})}{\ln\left(\dfrac{h_{ao2} - h_{wi1}}{h_{ai1} - h_{wo2}}\right)} \qquad 7$$

Y is the correction of our double heat exchanger with respect to the counterflow operation. Y is obtained from equations (6) and (7).

V. Determination of the efficiency of the battery.

By introducing the number of transfer units NTU $\left(NTU = \dfrac{U\ A}{C_{\min}}\right)$ and the ratio of the thermal rates of heat flow $C = \dfrac{C_{\min}}{C_{\max}}$, we can write the efficiency of the counterflow heat exchanger as :

$$\varepsilon_1 = \frac{e^{-NTU(1-C)} - 1}{Ce^{-NTU(1-C)} - 1} \qquad 8$$

And that of the crossflow as:

$$\varepsilon_2 = 1 - \exp\left(NTU^{0,22}\right)\chi \qquad 9$$

where

$$\chi = \frac{\exp\left(-R.NTU^{0,78}\right) - 1}{R} \qquad 10$$

Knowing the two efficiencies we can find the relationship between the three efficiencies:

$$\varepsilon = \frac{\dfrac{1-\varepsilon_1}{1-R.\varepsilon_1}\dfrac{1-\varepsilon_2}{1-R.\varepsilon_2} - 1}{R\dfrac{1-\varepsilon_1}{1-R.\varepsilon_1}\dfrac{1-\varepsilon_2}{1-R.\varepsilon_2} - 1} \qquad 11$$

ε is the efficiency of the battery with two heat exchangers

VI. Determination of the heat transfer coefficients.

For a given operating point, that correspond to the flow rates of water and air [1,6], The overall heat transfer coefficient can be determined when the local heat transfer coefficients of air and water are known.

The total power transferred for the cold battery is then given in humid mode by equation 7.

This total power can also be determined from the states of the fluids, air and water and by considering the system described.

$$\phi = C_a\left(h_{ao} - h_{ai}\right) = C_w\left(h_{woSat} - h_{wiSat}\right) \qquad 12$$

With the defined specific heat capacities $C_a = \dot{m}_a$ et $C_w = \dot{m}_w \dfrac{c_{pw}}{c_{pSat}}$

The overall enthalpic coefficient is then from the equations 7 et 12. Solving these equations we arrive at:

$$U\ .A = \frac{C_a\left(h_{ao} - h_{ai}\right)}{Y\dfrac{\left(h_{ao} - h_{woSat}\right) - \left(h_{woSat} - h_{wiSat}\right)}{\ln\left(\dfrac{h_{ai} - h_{woSat}}{h_{woSat} - h_{wiSat}}\right)}} = \frac{C_{\min}\left(h_{woSat} - h_{wiSat}\right)}{Y\dfrac{\left(h_{ao} - h_{woSat}\right) - \left(h_{woSat} - h_{wiSat}\right)}{\ln\left(\dfrac{h_{ai} - h_{woSat}}{h_{woSat} - h_{wiSat}}\right)}} \qquad 13$$

From this expression we can deduce the overall heat transfer coefficient provided that the input and output conditions and the rates of heat flux of the fluids are known.

VI.1. Determination of the local heat transfer coefficient on the side of air.

To determine the local heat transfer coefficient on the side of air, we consider the fictitious cold battery having an infinite flow rate of water and giving the same operating conditions to the air. The external heat transfer coefficient for the cold battery depends only on the operating conditions on the side of air and the geometry of the battery [8, 9].

Because of the internal resistance nullified by the infinite rate of flow of water, the considered battery gives a homogeneous and constant temperature. The considered temperature is called mean surface temperature which is higher than the dew temperature of air.

By letting $\dot{m}_w c_w$ approach infinity ($\dot{m}_w c_w \to \infty$) ($\Delta\theta_w$ =0) in the expression of the overall efficiency of the cold battery (11), we obtain the following efficiency for the fictitious battery :

$$\varepsilon_{\inf} = 1 - e^{-2NUT} \qquad 14$$

This efficiency of the fictitious battery of infinite capacity can be defined as the ratio between the total power transferred and the maximum transferable power in the ideal case:

$$\varepsilon_{\inf} = \frac{C_a\left(h_{ao} - h_{ai}\right)}{C_{\min}\left(h_{ms} - h_{ai}\right)} \qquad 15$$

h_{ms} corresponding to the enthalpy of saturated air at the mean surface temperature is given by :

$$h_{ms} = c_{pa}\theta_{ms} + w_{ms}\left(L_v - c_{pv}\theta_{ms}\right) \qquad 16$$

Where θms is the mean surface temperature

We then deduce the local transfer coefficient on the side of air by a combination of equations 14, 15 et 16 and the 'expression of NTU.

$$U_a A_a = -\frac{1}{2}\dot{m}_a c_{pa} \ln\left(1 - \varepsilon_{\inf}\right) \qquad 17$$

The calculated transfer coefficient on the side of air for the fictitious battery can assimilated to that of the real battery because this battery has been chosen in such a way that the operating conditions on the side of air are unchanged. The transfer coefficient depends only on these conditions. In addition the fictitious operating conditions on the side of air do not arise from an arbitrary choice but from a physical limiting case.

VI.2. Local Transfer Coefficient on the side of water:

This coefficient is deduced from the overall transfer coefficient and from the local transfer coefficient on the side of air by equation 4.

VI.3. Calculation of input and output conditions of air and water.

VI.3.1. On the side of air

The input enthalpy of air is calculated given by the following equation, provided that the temperature is known..

$$h_{ai} = C_{pa}\theta_{ai} + w_{ai}(L_v + C_{pv}\theta_{ai}) \qquad 18$$

For the output :

$$h_{ao} = c_{pa}\theta_{ao} + w_{ao}(L_v + c_{pv}\theta_{ao}) \qquad 19$$

w_{ai} et w_{ao} are obtained when the dry and humid temperatures at the input and output are known.

VI.3.2. On the side of water

The expression of the input and output enthalpy of water will be calculated by using equation [10]

$$h_w = \frac{\theta}{0{,}2374 + 4{,}015 \times 10^{-5}\theta - 2{,}721 \times 10^{-7}\theta^2} \qquad 20$$

VI.3.3. Calculation of c_{psat}.

The enthalpy of air at the dew point is given by :

$$h_r = h_{wiSat.} + c_{psat}\left(\theta_r - \theta_{wi}\right) \qquad 21$$

C_{psat} can be estimated when the enthalpy of the corresponding dew point is known.

Th dew point is obtained by using the formula found and recommended by AICCAV (l'Association des Ingénieurs en Chauffage Conditionnement d'Air et de Ventilation en France) [11].

$$\theta_r = \frac{31{,}61}{\frac{1}{\log(P_v)} - 2{,}7877} - 0{,}13 \qquad 22$$

Using this formula we calculate hr by using the following equation:

$$h_r = c_{pa}\theta_r + w_{ai}\left(L_v - c_{pv}\theta_r\right) \qquad 23$$

And we deduce the expression of c_{pSat} from equation 21.

$$c_{pSat} = \frac{h_r - h_{wiSat}}{\theta_r - \theta_{wi}} \qquad 24$$

The application of these techniques of determination of the characteristics of the cold battery requires the knowledge of the input and output temperatures of the fluids, the mean surface temperature and the rate of heat flux of the fluids.

These temperatures are measured experimentally by placing thermocouples at different points of the cold battery.

IV. Calculation procedures.

1. Calculation of the overall transfer coefficient U.A based on the enthalpies for each operating in the humid zone.
   a. Estimate C_{psat} by considering the dew point of air at the input temperature of water (24)
   b. Calculate the input and output enthalpies of the fluids.
   c. Evaluate the enthalpy Mean-log Temperature Difference
   d. Determine the correction factors F and Y.
   e. Determine the overall enthalpy transfer coefficient from the total power transferred.
2. Determine the transfer coefficient on the side of air Ue.Ae by considering a fictitious cold battery with an infinite capacity on the side of water.

   a. Calculate h_{ms} from the mean surface temperature as the intersection of the straight line given by the input conditions and the saturation curve.

   b. Determine the efficiency of the fictitious battery.

   b. Calculate the local transfer coefficient on the side of air $U_e.A_e$ by using the relation NUT-ε to obtain a battery of infinite capacity.
3. Calculate the transfer coefficient on the side of water $U_i.A_i$ from the overall transfer coefficient.

V. Experimental results and discussion.

This part consists of the experimentation of the battery by placing thermocouples at various points to measure the temperature differences.

In the following figures we present the variation of the parameters of the fluids and the characteristics of the battery with respect to time.

The mass of iced water introduced is 2kg, the flow rate of air flow is $\dot{m}_a$ =0.13kg/s and the flow rate of water is m_w =0.5 g/s

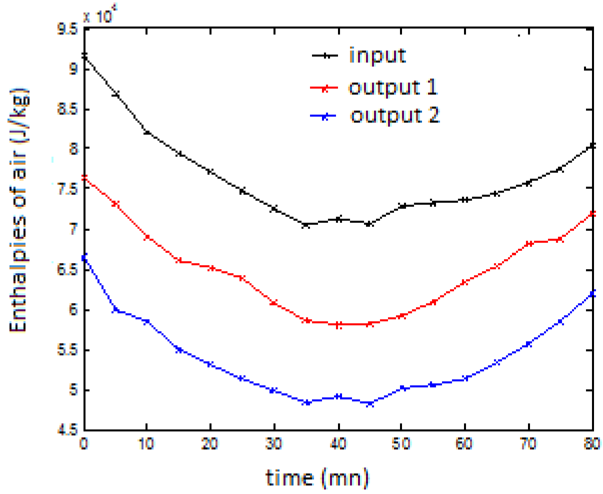


Figure 2 : Evolution of the enthalpies of air

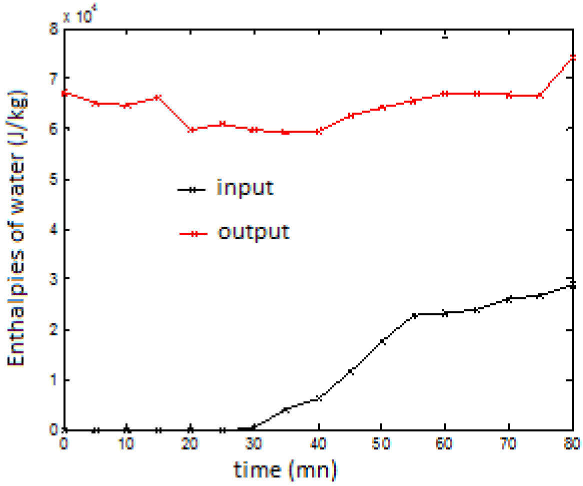
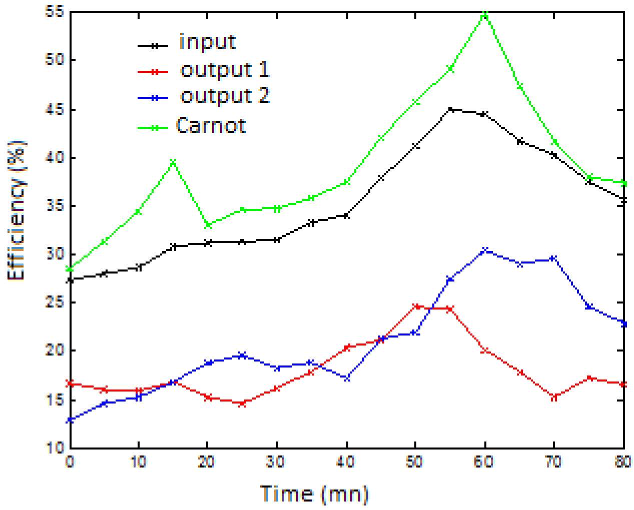


Figure 3: Evolution of the enthalpies of water

The input enthalpy of air is higher than the output enthalpies on the two exchangers because the air releases heat to the cold fluid (iced water).

Knowing these enthalpies will permit to calculate the efficiencies of the two heat exchangers and that of their combination.



Figures 4 : Efficiencies in terms of time.

The cold battery presents a good efficiency because of the fact that its efficiency tends towards that of CARNOT. This efficiency has improved because of the series arrangement of the heat exchangers.

We can now present the graphs of the overall and local heat transfer coefficients obtained from Δhlm and the NTU-ε methods.

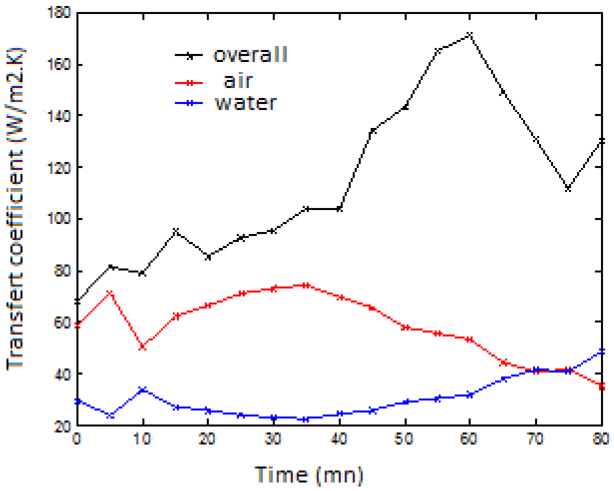


Figure 5: Evolution of the transfer coefficients in terms of time

The heat flux is an additive quantity but the transfer coefficients are not additive, this is due to the two observed phenomena: counterflow for flat-plate exchanger and crossflow for the shell-and-tube exchanger.

We present in figure 6 the graphs of the variations of the local transfer coefficients on the side of air obtained by means of the proposed calculation technique and the equations found in the literature.

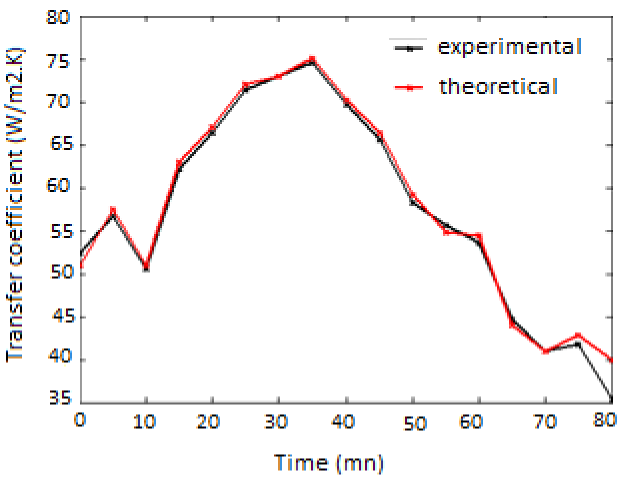


Figure 6: Variation of the local transfer coefficients in terms of time

The results obtained from the proposed method are very closed to the theoretical results. This validates the proposed technique. The contribution to the overall transfer coefficient by the air is to a large extend dominant compared with the contribution of the water.

VII. Conclusion:

This work has enabled us to present a machine that presents a double advantage, because it allows us to cool water and serve as a water fountain. We have also presented a method of calculation the can be used to determine the parameters of the battery. This model is based on the usage of the Log-Mean Temperature Difference applied to the battery operating in a humid mode. This technique is utilized for the determination of the operating point of the battery and we have proposed a method of determining that point. The local coefficients (for air and water) and global and the efficiencies are determined.

An experimental work has enabled us to validate this technique for the characterization of the cold battery with a double heat exchanger.

References

[1] B. DIENG : Thèse d'Etat Mai 2008 UCAD,
« Etude et réalisation d'une batterie froide à double échange couplée à un ventilateur pour l'amélioration du confort thermique dans l'habitat »

[2] W. MAAKE, J. ECKERT et J. L. CAUCHEPIN : « Le nouveau Polhman – Manuel technique du froid », Pyc édition, 1988.

[3] Threlkeld, J. L. *Thermal Environmental Engineering*, 2nd Edition 1970, Englewood Cliffs : Prentice-Hall, Inc, pp 254-270

[4] GAYE S., NIANG F., CISSE I. K., ADJ M., MENGUY G. et SISSOKO G., Caractérisation des propriétés thermiques et mécaniques du béton de polymère recyclé, Journal des Sciences (2001), Vol.1, N°1, pp.53-66

[5] L. M. VOUMBO, B. DIENG, S. TAMBA, S. GAYE, M. ADJ et G. SISSOKO
Automatisation de la mesure de la conductivité et de la diffusivité par la méthode des boites Journal des Sciences (2007), Vol.7, N°4, pp.73-86.

[6] CASARI AICVF, 1998 guide n°10

[7] CARRIER : Manuel Carrier 1ère et 2ème partie, Carrier International LTD, New-York, carrier corporation 2ème Edition, 1960

[8] DUMILIL. M : « Air humide, Technique de l'ingénierie 1999 » B 2 230-1.

[9] ASHRAE Handbook of Fundamentals, SI Edition, American Society of Heating, Refrigerating and Air Conditioning Engineers, 1994

[10] BOUTELOUP. J, LE GAY M et LIGEN. J. « Distribution des fluide hydraulique – Aéraulique ». Tome 2, 2002, 311p Edition parisienne ISBN 2-86-243-062-5.

[11] J. R. CAMARGO, C. D EBINUMA AND J. L. SILVEIRA
"Experimental performance of a direct evaporative cooler operating during summer in a Brazilian City".
International journal of Refrigeration 20 (2005) 1124 – 1132