IOSR Journal of Engineering
e-ISSN: 2250-3021, **p**-ISSN: 2278-8719,
Vol. 2, Issue 12 (Dec. 2012), ||V1|| PP 49-54

Charaterization of a cold battery with double heat exchanger in a dry mode

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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 [1,2]. The characterization of that cold battery will be done in a dry regime where there is only one heat exchanger and the quantity of vapor contained in the air does not varied and also the surface temperature of the battery is greater than the dew temperature of air [3 ; 4; 5].**

Keywords - Battery, Heat exchanger, Temperature, Condensation, Cross flow, Counter flow

NOMENCLATURE

Symbol	**Description**	**Unit**
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	Heat Capacity	J/kg°C
$\dot{m}_e$	Flow rate of water	kg/s
$\dot{m}_a$	Flow rate of air	kg/s
U	Coefficient d'échange thermique par convection	$W.m^{-2}.K^{-1}$
S	Surface	m^2
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
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
r: relative to the dew point of air
t : relative to tube
p : relative to the flat-plate

I. INTRODUCTION

The axial sucksion 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 to heat exchangers of the cold battery.

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 $\Delta\theta_{lm}$ and NTU-ε techniques [5; 6] and for a given operating point in a dry regime.
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

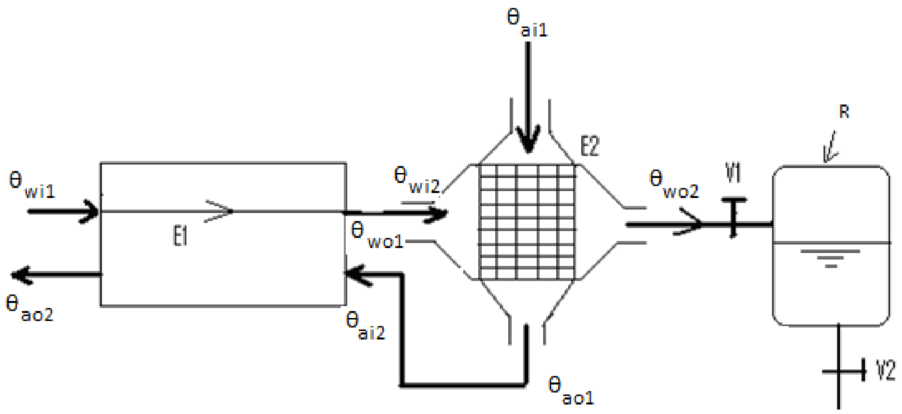

Figure 1: Cold battery with two heat exchangers

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 DRY REGIME

1. Determination of the 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 à operating in counterflow
The overall flux which is the sum of the two flux is then given by::

$$\phi = \phi_1 + \phi_2 \qquad 1$$

therefore:

$$\phi = U_P S_p \frac{(\theta_{a02} - \theta_{wi1}) - (\theta_{ai2} - \theta_{w01})}{\ln\left(\frac{(\theta_{a02} - \theta_{wi1})}{(\theta_{ai2} - \theta_{w01})}\right)} + U_t S_t F \frac{(\theta_{a01} - \vartheta_{wi2}) - (\theta_{ai1} - \theta_{w02})}{\ln\left(\frac{\theta_{a01} - \theta_{wi2}}{\theta_{ai1} - \theta_{w02}}\right)} \qquad 2$$

The correction factor F linking the crossflow and the counterflow is given by charts or correlation equations.
The overall flux can be written as:

$$\phi = U_{eq} S \,.Y \frac{(\theta_{a02} - \theta_{wi1}) - (\theta_{ai1} - \theta_{w02})}{\ln\left(\frac{(\theta_{a02} - \theta_{wi1})}{(\theta_{ai1} - \theta_{w02})}\right)} \qquad 3$$

Y is the correction of our double heat exchanger with respect to the counterflow operation. Y is obtained from equations (2) and (3). We will propose the experimental determination using the input and output conditions of the fluid and the characteristics of the heat exchanger which is the objective of this paper

III.2. Determination of the efficiency of the battery.

By introducing the number of transfer units NTU $\left(NTU = \frac{U\ A}{C_{\min}} \right)$ and the ratio of the thermal rates of heat flow $C = \frac{C_{\min}}{C_{\max}}$, 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:

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

ε is the efficiency of the battery with two heat exchangers

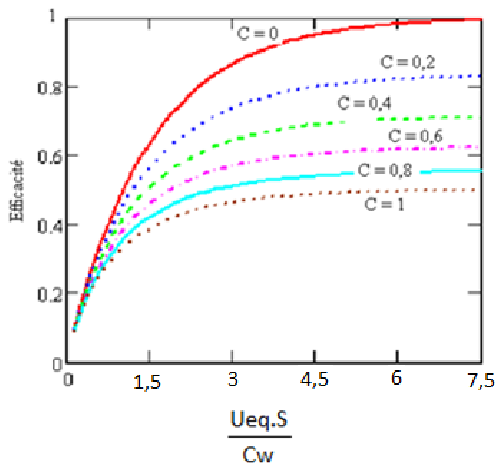

Figure 2 : Diagramme of determination of the effective heat transfer coefficient

This diagram enables us to determine the effective heat transfer coefficient of the cold battery by using the operating point and the measured input and output temperatures of the battery. For that reason we start by determining the efficiency and resolve it on the grapgh in order to obtain the heat transfer coeficient

3. Determination of the heat exchange coefficients and the Y correction factor.

The globall and local heat transfer coefficients of the battery are determined for a given operating point

3.1. Global heat transfer coefficient

For a given operating point and the input and output temperatures of the battery known, the heat transfer coefficients can be obtained from figure 1.

3.2. Determination of the Y correction factor

Knowing the global heat transfer coefficient we can calculate the Y correction factor given by equation 10.

$$Y = \frac{C_a(\theta_{ai1}-\theta_{a02})}{.U_{eq}S\,\dfrac{(\theta_{ai1}-\theta_{w02})-(\theta_{w02}-\theta_{wi1})}{\ln\left(\dfrac{\theta_{ai1}-\theta_{w02}}{\theta_{w02}-\theta_{wi1}}\right)}} = \frac{C_w(\theta_{w02}-\theta_{wi1})}{U_{eq}S\,\dfrac{(\theta_{ai1}-\theta_{w02})-(\theta_{w02}-\theta_{wi1})}{\ln\left(\dfrac{\theta_{ai1}-\theta_{w02}}{\theta_{wi1}-\theta_{w02}}\right)}} \qquad 10$$

3.3. Local heat transfer coefficient on the air side.

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 [7, 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:

$$\varepsilon_{\inf} = 1 - e^{-2NUT} \qquad 11$$

With

$$NUT = \frac{U_a A_a}{C_{\min}} \qquad 12$$

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:

$$\varepsilon_{\inf} = \frac{C_a(\theta_{ao}-\theta_{ai})}{C_{\min}(\theta_r-\theta_{ai})} \qquad 13$$

We then deduce the local transfer coefficient on the side of air by a combination of equations 11, 12 and 13 and the 'expression of NTU.

$$U_a A_a = -\frac{1}{2}\dot{m}_a c_{pa} \ln(1-\varepsilon_{\inf}) \qquad 14$$

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.

3.4. Local heat transfer coefficient on the water side

This coefficient is deduced from the global heat transfer coefficient from air side and from the characteristics (ε, λ) of the materials used in design of the battery.

The global heat transfer coefficient is obtained by using the electrical analogy [10, 11].

$$\frac{1}{U_{eq}} = \frac{1}{U_a} + \frac{1}{U_w} + \frac{e_p}{\lambda_p} + \frac{e_t}{\lambda_t} \qquad 14$$

U_a equivalent external heat transfer coefficient (air side)

U_w equivalent internal heat transfer coefficient (water side)

e_p and λ_p are respectively the thickness and the thermal conductivity of the flat plate heat exchanger.

e_p and λ_p are respectively the thickness and the thermal conductivity of the shell and tube heat exchanger.
The exit conditions of the battery could be obtained from the previous equations.

IV. EXPERIMENTAL WORK

This part of the work consists of validating this technique by an experiment for which the battery is filled with water and thermocouples placed within in order to measure the temperature at different points of the heat exchanger. The simulation is done by using MATLAB.

- The air temperatures at the entrance and exit of the cross flow heat, the latter gives simultaneous the air temperatures at the entrance of the flat plat heat exchanger;
- The air temperature at the exit of the flat plat heat exchanger.
- The water temperatures at the entrance and exit of the heat exchanger.
- The rate of flow of water in the heat exchanger (obtained from the operating point).
- Evaluation of the logarithmic mean temperature difference
- Determination of F from tables of values
- Calculation of the effective global heat transfer coefficient
- Determination of the correction factor Y
- Determination of the local heat transfer coefficient on the side of air
- Determination the local heat transfer coefficient on the side of water

These results will be compared with the classical method used to calculate the equivalent heat transfer coefficient of the battery composed of two heat exchangers (the flat plat and the shell and tube one). This experiment has been realized in a $9m^2$ room with an initial air temperature of 38°C and a relative humidity of 45% with a battery filled with 3 liters of tap water at the temperature of 20^oC with a flow rate of 1g/s and an air flow rate of 0.13 $kg.s^{-1}$. Since the temperature variations are very small we have taken some values and calculated the parameters.
The experimental results are presented in this table.

Table 1 : Experimental results

Time	**Measured temperatures**												
T (mn)	θ_{ai1}	θ_{a01}	θ_{Ch}	θ_{wi}	θ_{w0}	□	**Y**	**Ueq (exp)**	ε_{inf}	**Ua**	**Uw**	**Ueq (calcul)**	**Erreur %**
5	37	35,5	34	20	23	0,78	0,89	9,19	0,67	5,63	2,93	8,79	4,3
15	36	35	32,5	21	25	0,79	0,90	10,03	0,69	6,33	2,98	9,83	2
30	33,5	31	29	23	27	0,80	0,92	10,86	0,72	6,74	3,07	10,4	4,1
50	30,7	30	28,5	25	29	0,76	0,90	10,45	0,70	6,62	3,07	9,98	4,5

The table gives the local heat transfer coefficients (on the sides of air and water) and the global ones. It also gives the efficiencies and the Y factors generated by a Matlab programme. The results obtained are compared to the ones generated by the classical method and we arrive at a conclusion that the proposed method is satisfactory because the margin of errors does not exceed 5%. The results also show that the best global coefficients are the ones that correspond to the best efficiencies and are obtained over a period of 30 minutes time from which the battery will attain its lowest exit air temperature and the highest exit water temperature.

V. CONCLUSION

In this work we are able to present the machine designed for the purpose of cooling a room by using tap or well. We have also presented a new method of determination of the parameters of the battery. This model is based on the Log-mean –temperature difference applied to a battery operating in a dry mode. This technique is based on the determination of the operating point of the battery and we have proposed a method of determining that point. From the experimental results and a MATLAB programme we have determined the local heat transfer coefficients (for air and water), the global coefficients, the efficiencies and the Y factors. The results obtained and compared with the classical method are satisfactory given the fact that the margin of error does not exceed 5%.

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