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Determination Of The Operating Point And The Enthalpy Per Unit Surface Of A Cold Battery With Icy Water And A Double Heat Exchanger

B. Dieng[1],G. Jaw[2], A. Kane[3], I. Diagne[3] and G. Sissoko[3]
[1] Physics Section, UFR SATIC, UniversityAliouneDiop ofBambey, Senegal
[2] Regional Maritime University, Electrical Engineering Department Accra Ghana
[3] University Cheikh Anta Diop, Dakar, Senegal

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 are connected to a pipe of the same diameter. These pipes will permit the transit of the icy water coming from the flat-plate heat exchanger by gravitation towards the tubes of the second exchanger [1]. The good operation of this cold battery depends on the knowledge of its operating point. We are proposing a technique of determination of the operating point by using one of the two fluids (water or air) and the efficiencies [2, 3].The Knowledge of that operating point will enable us, through experimental means, determine the mean surface temperatureand then determine the mean surface enthalpy from the specific heat capacity at saturation obtained from the linearization of the entrance and exit air temperatures on the saturation curves.

I. Introduction

The battery is composed of two heat exchangers connected in series: the flat-plate and the shell-and-tube exchangers.

The axial suction of air by the fan passes in cross-flow 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 two heat exchangers of the cold battery.

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].

This condensation temperature is called equivalent mean surface temperature of the cold battery. That is the minimum air temperature if the heat transfer was ideal.

To know the performance of the battery, it is necessary to know its operating point and its mean surface temperature. Let us note that the knowledge of these parameters is necessary for the characterization of the cold battery.

The battery is conceived in such a way that the flat-plate heat exchanger operates in a cross-flow system while the shell-in-tube one operates in a counter flow system.

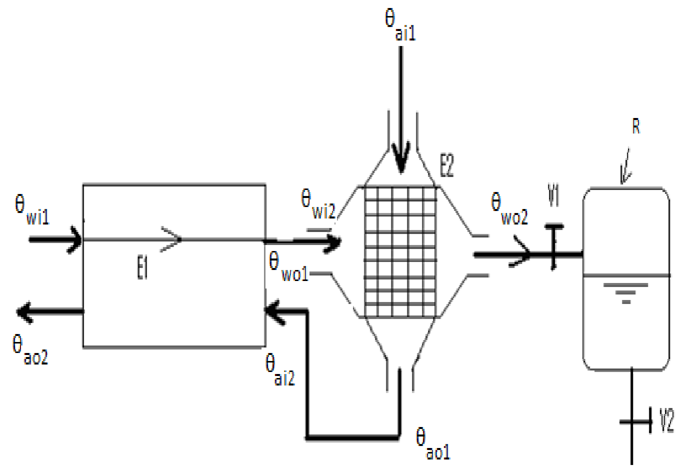

Figure 1: Cold battery with two heat exchangers

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. Hypotheses [4, 5]

- The variables (fluid temperatures) are dependent on their axial positions.
- The specific heat capacities are constant during the process.
- The heat transfers are all convective.
- The heat exchanger is thermally isolated from its surrounding.
- There is no heat loss during the transmission of fluids between the two heat exchangers.
- We assume that the enthalpy of air at saturation is a linear function of temperature within the range of temperatures in consideration.

III. Determination of the overall heat flux.

The overall flux is the sum of the two flux $\Phi1$ and$\Phi_2$because the two heat exchangers are in series.

$$\phi = \phi_1 + \phi_2 \qquad 4$$

Φ_1: flux due to the shell-in-tube exchanger.

Φ_2 : flux due to the flat-plate heat exchanger.

III.1. Determination of the battery efficiency.

By introducing the number of transfer units NTU $\left(NTU = \frac{U\ A}{C_{\min}}\right)$ and the ratio of the thermal rates $C = \frac{C_{\min}}{C_{\max}}$ [6], we can write the efficiency of the counter-flow heat exchanger as:

$$E_1 = \frac{e^{-NUT(1-C)} - 1}{Ce^{-NUT(1-C)} - 1} \qquad 7$$

And that of the cross flow one as:

$$E_2 = 1 - \exp\left(NUT^{0,22}\right)\chi \qquad 8$$

Where,

$$\chi = \frac{\exp\left(-R.NUT^{0,78}\right) - 1}{R} \qquad 9$$

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

$$E = \frac{\frac{1-E_1}{1-R.E_1}\frac{1-E_2}{1-R.E_2} - 1}{R\frac{1-E_1}{1-R.E_1}\frac{1-E_2}{1-R.E_2} - 1} \qquad 10$$

E is the efficiency of the double heat exchanger battery.

III.2. Determination of the operating point

The determination of the operating point consists of determining the flow rate of water or that of the air. This can be achieved by using one of the two fluids.

a. Using water to determine the operating point.

The determination of the operating point consists of determining the flow rate of water because the fan produces a constant flow rate of water.

The local heat transfer coefficient for the air does not vary because air has a minimum capacitive rate, then the NTU remains constant [7, 8]. However the overallheat transfer coefficient varies proportionally with surface area and with the water flow rate because of the variation of the local heat transfer coefficient on the side of water.

The efficiency at the operating point is obtained by maintaining a constant temperature of the water at the exit point:

$$\varepsilon = \frac{\phi_{req}}{C_{air}\left(\theta_{ae} - \theta_{ee}\right)} \qquad 11$$

WithΦ_{req} :is the required power to bring the water temperature to the desired value.

Moreover, the efficiency can be expressed in terms of NTU and R [10].

b. Using air to determine the operating point.

On the side of air the efficiencies at the operating point and the actual efficiency of the system are only function of the inlet and outlet temperatures of the two fluids. The determination of the operating point allows us to find the outlet temperature of water with the knowledge of the outlet air temperature and by fixing the water flow rate.

Considering the two cases, the point of intersection of the two efficiency graphs allows us to:

- In the first case to obtain the water flow rate at the operating point for a given temperature of water.
- In the second case to obtain the outlet water temperature for a given water flow rate.

Since our apparatus has a fixed air flow rate, we will utilize the water to determine its operating point.The intersection between the two graphs of efficiencies (10 and 11) gives us the flow rate of water at the operating point for a given outlet water temperature.We hereby present the graphs for the determination of the operating point of the double heat exchanger.

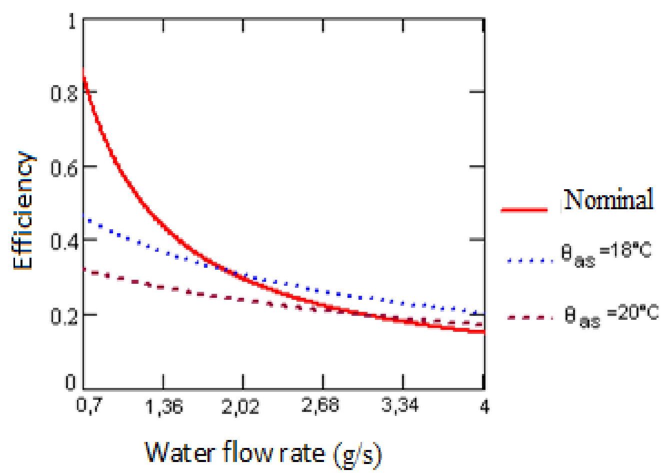

Figure 2:Graphs for the determination of the operating point (θ_{es} =15°C, θ_{ee} =30°C andθ_{ee}=0°C)

Therefore, the knowledge of the desired outlet water and air temperatures allows us to determine the water flow rate.For example, for an outlet water temperature of 15°C the flow rates are respectively 1,925 g/s for an outlet air temperature of 18°Cand 3,235 g/s for an outlet air temperature of 20°C.

III.3. Determination ofh_{ms}.

III .3.1. Behavior of humid air

We are now going to do the modeling of this apparatus in a humid mode, when the surface temperature of the cold battery is negligible compared with the dew temperature of air. The

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cooling enables the condensation of water vapor. We finally obtain a transformation of the form ACD. In practice we assume that the transformation is a direct transformation AD. The air comes out of the cooling system with a reduced water content and temperature [9, 10].

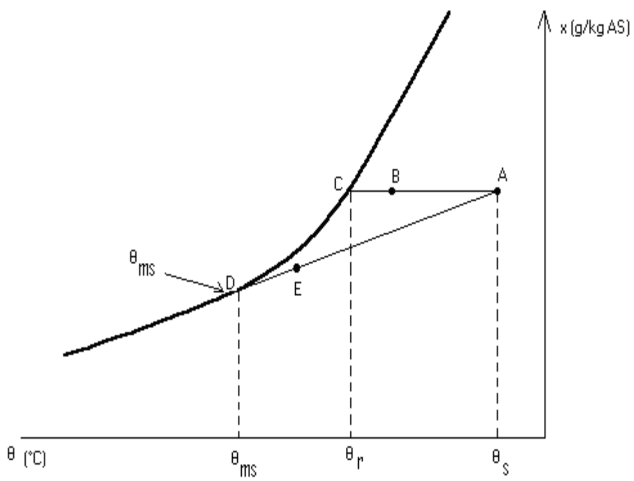

Figure 3: Variationof humid air on the Carrier diagram.

The temperature θmsof figure 3 (point D) is called equivalent mean surface temperature of the cold battery. That could be the minimum air temperature if the heat transfer processes were ideal.In reality the air comes out of the cooling system at an intermediary system between points A and B: that is at point E.
In this figure 2 we have noticed the alignment of the points A, E et D from which we can deduce respectively input conditions, output conditions and the surface temperature of the cold battery.

The slope of the line passing through these three points is given by:

$$\frac{P_{aeSat}-P_{asSat}}{\theta_{ae}-\theta_{as}}=\frac{P_{aeSat}-P_{msSat}}{\theta_{ae}-\theta_{ms}} \quad 12$$

This equation is also valid for other parameters like enthalpy, specific humidity etcThe pressure at saturation can be expressed in its classical form for low temperatures (4°C – 15°C) [7, 10] or in its linearized form.

In its classic form we have:

$$P_{Sat}=c_1\theta^2+c_2\theta+c_3 \quad 13$$

Where :

c_1 = 2,4101, c_2 = 35.537 et c_3 = 638.3347

But we can also linearize it and put it in the form:

$$P_{Sat}=c_4\theta+c_5 \quad 14$$

Where :

$$c_4=\frac{P_{aeSat}-P_{asSat}}{\theta_{ae}-\theta_{as}} \quad 15$$

and

$$c_5=P_{aeSat}-c_4\theta_{ae} \quad 16$$

Now equating the expressions 13 and 14 we obtain the quadratic equation:

$$c_1\theta^2+\left(c_2-c_4\right)\theta+c_3-c_4 = \quad 17$$

Solving this quadratic equation, we obtain the mean surface temperature.

$$\theta_{ms}=\frac{\left(c_4-c_2\right)+\sqrt{\left(c_2-c_4\right)-4c_1\left(c_3-c_5\right)}}{2c_1} \quad 18$$

This temperature should have been the exit temperature of air if the heat exchanger were ideal.The knowledge of that temperature will enable us calculate the enthalpy of air from equation 21.

The amount of water removed from the air is equal to the difference between the two absolute humidities in kg per kg of dry air, that is:

$$\Delta w=w_A-w_{ms} \quad 19$$

The necessary air flow rate is also given by:

$$\dot{m}_a=\frac{\phi}{h_A-h_{ms}} \quad 20$$

If we divide it by the air density we obtain the air flow rate.

III.3.2. Calculation of the mean surface enthalpy of air

The mean surface enthalpy of airh$_{ms}$correspond to enthalpy of saturated air at the mean surface temperature and is given by:

$$h_{ms}=c_{psat}\theta_{ms}+w_{ms}\left(L_v-c_{pv}\theta_{ms}\right) \quad 21$$

$\Theta_{ms}$is obtained from the linearized graph in figure 1.
W_{ms} is obtained when the humid and dry temperatures are known.

- **Determination ofc_{psat}.**

The enthalpy at the dew point is given by:

$$h_r=h_{ee.}+c_{psat}\left(\theta_r-\theta_{ee}\right) \quad 22$$

c_{pSat}can be estimated when the enthalpy of the dew point is known.

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The dew point is obtained by using the formula established by the HVAC associationin France (AICCAV) [11].

$$\theta_r = \frac{31,61}{\frac{1}{\log(P_v)} - 2,7877} - 0,13 \qquad 23$$

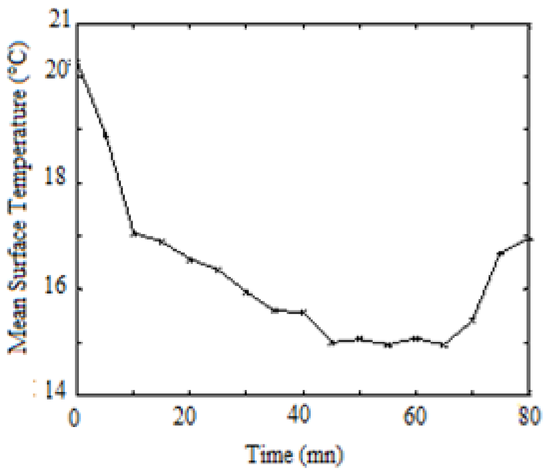
Using this formula we calculate $h_r$using:

$$h_r = c_{pa}\theta_r + w_{ae}\left(L_v - c_{pv}\theta_r\right) \qquad 24$$

And then we deduce the expression of $c_{pSat}$from equation 33.

$$c_{pSat} = \frac{h_r - h_{ee}}{\theta_r - \theta_{ee}} \qquad 25$$

The graphs of the mean surface temperature and mean surface enthalpy of the cold battery follow the same pattern. These two parameters enable us to characterize the cold battery in a humid mode.



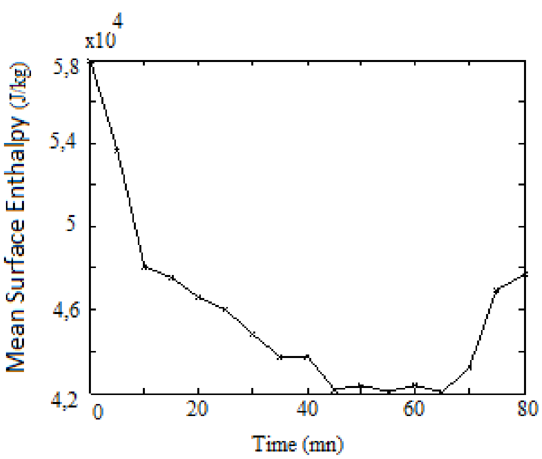


Figure 4: Variation of mean surface temperature and mean surface enthalpy with time

IV. Experimental results.

This is the experimental part of this work. It consists of switching on the battery for an optimal operating point and recording input and exit temperatures of air and using the proposed technique to obtain at any instant t, the mean surface temperature and then the mean surface enthalpy.

We are now presenting in figure 3 the variation with time of the mean surface temperature and the mean surface enthalpy of the constructed battery.

The mass of icy water introduced in the battery is: 2kg,

The air flow rate is: $\dot{m}_a$ =0.13kg/s and the water flow rate is $\dot{m}_e$ =0.5 g/s

V. Conclusion:

This work has enabled us to show a technique of determining the operating point of the battery by using the efficiencies. Based on the external conditions this operating point permits us to choose the most appropriate materials for the construction of the battery. The experimental determination of the mean surface temperature and the mean surface enthalpy enables us to obtain the parameters of the cold battery. With the knowledge of these experimental parameters we can deduce the local and global heat transfer coefficients of the cold battery in humid mode.

References

i. *B. Dieng : thèse d'etat mai 2008 ucad, etude et réalisation d'une batterie froide à double échange*

ii. *W. Maake, J. Eckert Et J. L. Cauchepin : « le nouveau polhman – manuel technique du froid », pyc édition, 1988.*

iii. *Andreieff De Notbeckg : « manuel du conditionnement d'air » tome ii, pyc edition, 1978.*

iv. *Carrier : manuel carrier 1ère et 2ème partie, carrier international ltd, new-york, carrier corporation 2ème edition, 1960*

v. *Dumilil. M : « air humide, technique de l'ingénierie 1999 » b 2 230-1.*

vi. *A. Beshkani And R. Hosseini "numerical modelling of rigid media evaporative cooler applied thermal engineering " 26 (2006) 636 – 643.*

vii. *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.*

viii. *P. Hollmuler, B. Lachal, F. Romerio et w. weber « habitat, confort et energie » colloque du cycle formation du cuepe 2002-2003.*

ix. *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*