NUMERICAL ANALYSIS OF PERFORMANCE IN A FINNED-TUBE EVAPORATOR

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Abstract. *The purpose of this paper is to analyze accuracy of different softwares with different correlations for heat transfer and pressure drop applied to solve the performance of an evaporator. In this paper experimental data are compared with the solution given by EVSIM and the EVAP-COND. Both programs are based on a tube-by-tube approach which allows complex geometries for the heat exchangers, modeling refrigerant distribution between these circuits according to the pressure drop correlations. Those programs are able to simulate performance of an evaporator coil with non-uniform, transverse to the tubes, one-dimensional air distribution. Each tube is associated with refrigerant parameters and specific air mass flow rate, inlet temperature and humidity. Advantages, disadvantages and viability like accuracy and easiness for use are discussed along the paper.*

Keywords: *Evaporator, EVSIM, EVAP-COND, analysis.*

1. INTRODUCTION

This paper describe a analysis of different programs to predict the performance of a finned-tube evaporator. Considering that the evaporator is one of the most important components in air-conditioning, find the ideal design regarding the efficiency is necessary. The fin-and-tube evaporator geometry is being more complicated, and the refrigerant used is frequently modified due to environmental reasons.

The numerical analysis is becoming more relevant because the experimental studies are much more time costing and expensive. Therefore, in the moment, numerical investigation is a good alternative to reduce costs. Many numerical models have been presented in this intention by many researchers.

Fisher and Rice (1981) developed a general heat pump model that includes a evaporator model. This model considers the heat exchangers as made of parallel circuits and consequently utilizes effectiveness versus NTU correlations.

Domanski and Didion (1983) presented a computer model for a heat pump that includes a evaporator model based on a tube-by-tube approach. That model is able to analyze the thermal performance of each tube separately considering the refrigerant temperature at the tube and the average air temperature for all tubes in a given row. The selection of tube for thermal performance evaluation is opposite to the refrigerant flow. This scheme causes a unrealistic imposition of the same refrigerant conditions at the outlet of each circuit. The air mass flow rate is assumed the same for all tubes.

In sequence of those models was done by Domanski (1989) a evaporator model, EVSIM, also based on tube-by-tube approach. A new ability of that model was to simulate performance of an evaporator coil with non-uniform, transverse to the tubes, one-dimensional air distribution, see Fig. (1).

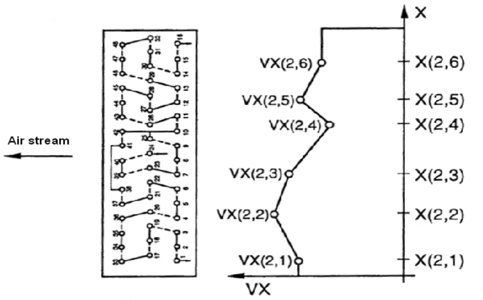

Figure 1. One-dimensional air distribution

The selection of the tubes is from inlet to outlet resulting in a more realistic model. Refrigerant distribution is based on pressure drop in each circuit and the pressure drop correlation is described in Pierre (1964). Refrigerant superheat at different circuit outlets can be different depending on the heat gained and mass flow in each circuit (Domanski, 1989).

EVAP-COND version 2.3 is a package which contains simulation models for predicting thermal performance of air-to-refrigerant finned-tube evaporators and condensers. Nowadays that program is a used a lot due the easiness to operate and the actual correlations utilized for heat transfers and pressure drop, moreover the recent version is flexible at the fin type and geometry. In the software EVAP-COND, the author Piotr Domanski indicate reports that describe the models used in the EVAP-COND v2.3: Domanski (1999), Domanski (1991) and Domanski and Didion (1984).

A comparison between EVSIM and EVAP-COND are done in section 4. Two different evaporative heat transfer correlations are inserted in EVSIM by the authors and these results are compared between themselves and with EVAP-COND results.

As can be seen in Fig. 2, the coil simulated in this paper is designed with 3 rows and 16 tubes per row. The length of the tubes is 454 mm, inner diameter 9,22 mm, outside diameter 10,01 mm, distance between tubes in the same row 25,4 mm, distance between the rows 22,23 mm, smooth inner tube, thermal conductivity of the tube 0,386 kW/(m°C), wavy fin thickness 0,2 mm and center to center distance between fins are 2 mm which thermal conductivity corresponds to 0,2216 kW/(m°C). For EVAP-COND was possible to consider a fan thermal power equal to 100 W, but in EVSIM it is not allowed.

2. EVAP-COND V2.3

EVAP-COND v2.3 (2008) is also a tube-by-tube approach like EVSIM, however the more recent program uses renovated correlations for heat and mass transfer.

Following air side heat transfer correlations are used in EVAP-COND: For flat fins is used Wang et al. (2000), for wavy fins Wang et al. (1999a), for slit fins Wang et al. (2001), for louver fins Wang et al. (1999b) and for the fin efficiency is used the Schmidt method, described in McQuiston and Parker (1982).

Refrigerant side heat transfer correlations in EVAP-COND are: McAdams for single-phase heat-transfer coefficient for smooth tubes, described in ASHRAE(2001), Jung and Didion (1989) up to 80% quality, for quality range 80%-100% linear interpolation between heat transfer coefficient values for 80% and single-phase saturated vapor, or instead Jung and Didion the user can set the correlation used to Thome (2005).

Evaporation heat transfer coefficient, rifled tube, the correlation selected for the smooth tube with a 1.9 enhancement multiplier was suggested by Schlager et al. (1989).

About the pressure drop, for single-phase in a smooth tube, Blasius (1912) is used. And for two-phase pressure drop Muller-Steinhagen and Heck (1986) is the correlation used.

Return bend pressure drop, single-phase and two-phase, for a smooth tube are used Chisholm (1983) and Idelchik (1986). The length of a return bend depends on the relative locations of the tubes connected by the bend.

Single-phase and two-phase pressure drops for a rifled tube is solved with smooth tube correlations with a correction accounting for a smaller hydraulic diameter.

The main disadvantage in EVAP-COND is the fact that the source code is closed. However the program is very easy to utilize and it is able to use many different refrigerants. The coil design is mounted directly from the user similar whats sketched in Fig. 2. In EVSIM the same geometry should be written in a “txt” file.

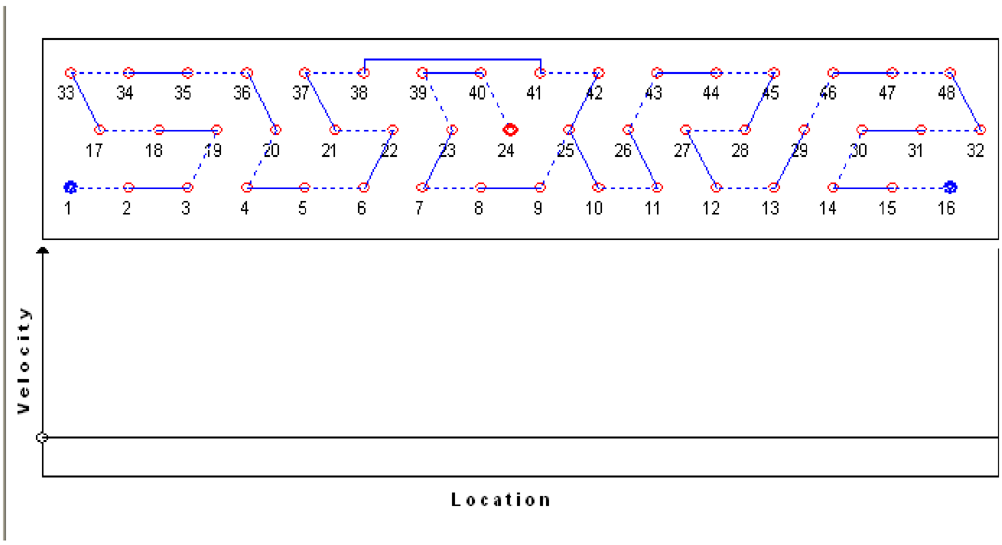

Figure 2. Picture from EVAP-COND representing the coil geometry

3. EVSIM MATHEMATICAL MODEL (Domanski, 1989)

EVSIM is detailed described in the report (Domanski, 1989). Analyzing a separated tube of the evaporator, the Peclet equation is used:

$$Q=U\cdot A\cdot \Delta T_m \tag{1}$$

In Eq. (1), A represents the heat transfer surface area, U is the overall heat transfer coefficient, ΔT_m stands for log mean temperature difference (LMTD) between the heat exchanging fluids, and Q the heat transfer rate.

In agreement with Threlkeld (1970) and others, LMTD if one of the fluids maintain a constant temperature the formula used is Eq. (2):

$$\Delta T_m=\frac{t_2-t_1}{\ln\frac{T_2-t_1}{T_1-t_2}} \tag{2}$$

And when temperature of both fluids change the Eq.(2) turns into Eq (3):

$$\Delta T_m=\frac{t_2-t_1}{\ln\frac{\frac{T_1-T_2}{t_2-t_1}}{\frac{T_1-T_2}{t_2-t_1}+\ln\frac{T_2-t_1}{T_1-t_1}}} \tag{3}$$

In Eqs. (2) and (3) T represents the temperature of one fluid, and t represents the temperature of the other fluid. Subscripts 1 and 2 stand for the inlet and outlet of the tube, respectively.

Finally energy conservation for both fluids is formulated:

$$Q=m\cdot(i_2-i_1) \tag{4}$$

$$Q=m\cdot c_p\cdot(T_2-T_1) \tag{5}$$

Where Q is the heat transfer rate, m is the mass flow rate, i is the enthalpy and T the fluid temperature.

The Eqs. (1) to (5) are the fundamental equations applied in the model for calculate the heat transfer rate between the air and refrigerant, in two-phase flow of the refrigerant or superheated vapor flow. If a phase changing is present it is necessary identify flow patterns in the tubes and associate to the fraction of tube length where it is present. To obtain the refrigerant pressure drop, the model employs different correlations in region of two-phase flow and superheated refrigerant flow. The calculus of heat transfer considers two different flow patterns in the two-phase flow region, considering annular flow (quality up to 0,85), and misty flow for qualities between 0,85 – 1,00. The correlations used are presented below.

3.1 Refrigerant two-phase flow regime

For annular flow the heat transfer rate (Domanski, 1989):

$$Q=m_a\cdot c_{p,a}\cdot(t_i-T_i)\cdot(1-\exp(\frac{-U\cdot A_o}{m_a\cdot c_{p,a}})) \tag{6}$$

The heat transfer rate calculated, Q, results in a quality for the refrigerants greater than 0.85, the heat transfer rate in annular regime:

$$Q=m_r\cdot(i_{r,85}-i_i) \tag{7}$$

The fraction of tube length with quality up to 0.85 is calculated with the following equations:

$$ANULAR=\frac{m_r\cdot(i_{r,85}-i_{r,i})}{m_a\cdot c_{p,a}\cdot(t_i-T_i)\cdot(1-\exp(-\frac{U\cdot A_o}{m_a\cdot c_{p,a}}))} \tag{8}$$

For misty flow, considering the air flow, the heat transfer rate:

$$Q = m_a \cdot c_{p,a} \cdot (1 - ANULAR) \cdot (t_i - T_i) \cdot (1 - \exp - (\frac{U \cdot A_o}{m_a \cdot c_{p,a}})) \tag{9}$$

If the heat transfer rate is calculated by the equation above the result is a greater refrigerant enthalpy than the saturated vapor, $i_{r,v}$, the heat transfer rate in misty flow region considering the refrigerant flow:

$$Q = m_r \cdot (i_{r,v} - i_{r,i}) \tag{10}$$

The fraction of tube length with misty flow, XDRY, may be calculated by the following equation:

$$XDRY = \frac{m_r \cdot (i_{r,v} - i_{r,i})}{m_a \cdot c_{p,a} \cdot (1 - ANULAR) \cdot (t_i - T - i) \cdot (1 - \exp(-\frac{U \cdot A_o}{m_a \cdot c_{p,a}}))} \tag{11}$$

Single-phase flow (superheated vapor):

$$Q = m_r \cdot c_{p,r} \cdot (t_i - T_i) \cdot (1 - \exp(-\frac{(1 - ANULAR - XDRY) \cdot m_a \cdot c_{p,a}}{m_r \cdot c_{p,r}} \cdot (1 - \exp(-\frac{U \cdot A_o}{m_a \cdot c_{p,a}})))) \tag{12}$$

From Eq. (6) to Eq. (12) the following nomenclature is used: A_o represents the exterior total area of the tube humid by the air, $c_{p,a}$ the air specific heat at constant pressure, $c_{p,r}$ the refrigerant specific heat at constant pressure, $i_{r,i}$ the refrigerant enthalpy at the tube inlet, $i_{r,v}$ the enthalpy of refrigerant saturated vapor, $i_{r,85}$ the refrigerant enthalpy at flow quality equal 0.85, m_a the air mass flow rate associated with the tube, m_r the refrigerant mass flow rate in the tube, t_i the air temperature upstream of the tube, T_i refrigerant temperature at tube inlet, ANULAR length fraction of the tube with flow quality up to 0.85, XDRY length fraction of the tube with flow quality within the range 0.85 – 1.00, and U the overall heat transfer coefficient.

3.2 Refrigerant pressure drop in a tube

Pressure drop is consequence of friction, momentum change and gravity. The gravitational pressure drop is not considered by EVSIM. Pressure drop in enhanced surfaces is obtained employing correlations of smooth tubes and multiplying it to correction factors.

3.2.1 Single-phase Pressure drop

3.2.1.1 Smooth tubes

The frictional pressure drop is obtained from Fanning equation through Fanning friction factor like what is done in Eq. (13) and (14):

$$\frac{dP}{dL} = \frac{2 \cdot f \cdot G^2}{D_i \cdot \rho} \tag{13}$$

$$f = 0{,}046 \cdot Re^{-0{,}2} \tag{14}$$

Pressure drop due to momentum change is calculated with:

$$\frac{dP}{dL} = -G^2 \cdot \frac{dv}{dL} \tag{15}$$

In above relations, *P* stands for the pressure, *L* the coordinate along the tube axis, *G* the refrigerant mass flux, D_i the tube inner diameter, v the refrigerant specific volume, *Re* the Reynolds number and f the Fanning friction factor.

3.2.1.2 Tubes with enhanced surfaces

The frictional pressure drop is calculated by Eqs.(13) and (14) applying a multiplying factor 1.5 to the pressure drop obtained from Eq. (16). This correction factor increases in 50 % the result obtained from EVSIM for the two-phase heat transfer coefficient for these surfaces.

3.2.2 Two-phase pressure drop with evaporation

3.2.2.1 Smooth tubes

The frictional pressure drop calculated for two-phase flow with evaporation may be calculated through the correlation proposed by Pierre (1964), based on his experiments with R-12 and R-22. The Pierre correlation combines the effects of momentum change and of friction in a unique equation:

$$\Delta P = (f \cdot \frac{L}{D_i} + \frac{\Delta x}{x}) \cdot G^2 \cdot v_m \tag{16}$$

In Eq. (16) f represents the friction factor calculated from Eq. (17), x the mean quality, Δx the quality change, $v_m = v_L + x_m.(v_V - v_L)$, where v_m is the mean specific volume, v_L the specific volume of liquid phase, v_V the specific volume of gas phase and x_m the mean vapor quality.

$$f = 0{,}0185 \cdot [\frac{K_f}{Re}]^{0,25} \tag{17}$$

Where:

$$K_f = \frac{J \cdot i_{fg} \cdot \Delta x \cdot g_c}{L \cdot g} \tag{18}$$

$$Re = \frac{G \cdot D_i}{\mu_L} \tag{19}$$

In Eq. (18) J is a conversion factor, J = 0.10213 (kg.m/J), g is the gravitational acceleration constant, g = 9.8 (m/s^2), and g_c = 9.8 ($m.kg/(kgf.s^2)$), dimensionless constant.

3.3 Overall heat transfer coefficient for finned-tubes

3.3.1 Dry tube

Overall heat transfer coefficient, U, in a dry finned-tube can be obtained from resistances between the refrigerant and the air, Threlkeld (1970), Domanski (1989):

$$U = [\frac{A_o}{A_{p,i}} + \frac{A_o \cdot x_p}{A_{p,m} \cdot k_p} + \frac{A_o}{A_{p,o} \cdot h_{tf}} + \frac{1}{h_{c,o} \cdot (1 - \frac{A_f}{A_o} \cdot (1-\phi))}]^{-1} \tag{20}$$

Where A_f is the fin surface area, A_o the total exterior surface area exposed to air, $A_{p,i}$ the pipe inside surface area, $A_{p,m}$ the pipe mean surface area, $A_{p,o}$ the pipe outside surface area, $h_{c,o}$ the convection heat transfer coefficient at the exterior surface, h_i inside tube heat transfer coefficient, h_{tf} thermal conductance of the pipe-to-fin contact, k_p thermal conductivity of pipe material, x_p thickness of pipe wall. And the fin efficiency ϕ is determined with Eq. (21):

$$\phi = \frac{T_{f,m} - T_a}{T_{f,b} - T_a} \tag{21}$$

Where T_a stands for the air temperature, $T_{f,b}$ the fin base temperature and $T_{f,m}$ mean fin temperature.

First and fourth term of Eq. (20) represent the inside and outside convection heat transfer resistances, respectively. The second term is the conductive heat transfer resistance through the tube wall and the third term is the contact resistance between the fin base collar and the outside tube surface.

3.3.2 Wet tube

The wet tube analysis is useful to evaporator model when the tube surface temperature is below the dew point of air.

Consequently, humidity is removed from the airstream by condensation on evaporator surface. If evaporator temperature is above the freezing point (simulation condition present in EVSIM) the condensate flows down under the influence of gravity.

The heat transfer rate between the airstream and the water surface is described by Eq. (22):

$$dQ=[h_{c,o}\cdot(T_a-T_w)+h_{D,o}\cdot(w_a-w_w)\cdot i_{fg,w}]\cdot dA_o \tag{22}$$

The first term represents the sensible heat transfer and the second means the latent heat transfer. In atmospheric pressure the Lewis number:

$$Le=\frac{h_{c,o}}{h_{D,o}\cdot c_{p,a}} \tag{23}$$

where normally in close to 1, Threlkeld (1970) and others. Considering that the fin efficiency approximates to the ratio of the moisture content differences:

$$\phi=\frac{w_a-w_{f,m}}{w_a-w_w} \tag{24}$$

The Eq. (22) , so, turns into the following model for the tube with flat fins:

$$dQ=h_{c,o}\cdot\left(1+\frac{i_{fg,w}\cdot(w_a-w_w)}{c_{p,a}\cdot(T_a-T_w)}\right)\cdot\left(1-\frac{A_f}{A_o}(1-\phi)\right)\cdot(T_a-T_w)\cdot dA_o \tag{25}$$

In the Eqs. (22-25): A_f symbolize the fin surface area, A_o the total external area, $c_{p,a}$ the specific heat of air, $h_{c,o}$ the convection heat transfer coefficient in the air-side (forced convection), $h_{D,o}$ the air-side mass transfer coefficient, $i_{fg,w}$ the latent heat of condensation for water, T_a the temperature of air, T_w the temperature of liquid water at the fin base, w_a humidity ratio of air, w_w the humidity ratio of saturated air at T_w temperature.

The one-dimensional heat conduction through the condensate film can be written by the equation:

$$dQ=h_w\cdot\Delta T_w\cdot dA_o \tag{26}$$

Where the heat transfer coefficient for the condensate film is given by Eq. (27):

$$h_w=\frac{k_w}{\delta} \tag{27}$$

Where k_w is the thermal conductivity of water, ΔT_w is the temperature difference across the condensate film and δ is the thickness of condensate film.

Utilizing the Eqs. (22) and (26) and referring to the Eq. (20), the subsequent relation for the overall heat transfer coefficient for a wet finned-tube is obtained:

$$U=\left[\frac{A_o}{h_i\cdot A_{p,i}}+\frac{A_o\cdot x_p}{A_{p,m}\cdot k_p}+\frac{1}{h_L}+\frac{A_o}{A_{p,o}\cdot h_{tf}}+\frac{1}{h_{c,o}\cdot\left(1+\frac{i_{fg,w}\cdot(w_a-w_w)}{c_{p,a}\cdot(T_a-T_w)}\right)\cdot\left(1-\frac{A_f}{A_o}(1-\phi)\right)}\right]^{-1} \tag{28}$$

Where the symbols used are defined according to the Eq. (20) and (24).

The actual model accounts for the impact of moisture condensation on the heat transfer in these directions (Domanski, 1989):

a . The layer of condensation offers additional heat transfer resistance (third term of Eq. (28)).

b. The air-side air heat transfer resistance decreases due to the effect of condensation (fourth term of Eq. (28)).

c. The air-side heat transfer coefficient, $h_{c,o}$, increases since it is sensitive to external surface geometry and the air flow Reynolds number.

d. Fin efficiency decreases as h increases.

More details involving fin and their heat transfer employed in the present version of the EVSIM may be seen in Domanski (1989), which is available in the NIST (National Institute of Standards and Technology) website .

3.4 Inside heat transfer coefficient

The single-phase forced convection heat transfer coefficient, h_{sp}, for a smooth tube is calculated considering the Dittus-Boelter correlation, (Domanski, 1989; Incropera, 2008) .

$$h_{sp}=\frac{0,023\cdot Re^{0,8}\cdot Pr^{0,4}\cdot k}{D_i} \tag{29}$$

In Eq. (29) *Re* is the Reynolds number, *Pr* is the Prantl number, *k* is the thermal conductivity of the refrigerant vapor and D_i is the tube inner diameter.

3.4.1 Tubes with enhanced surfaces

The heat transfer coefficient for enhanced inner surfaces is calculated in EVSIM by multiplying h_{sp}, obtained from Eq.(29), by a correction factor equal to 2.0. This correction factor chosen is an average of the heat transfer enhancement reported by Khanpara et al. (1987) for micro-fin tubes with R-22.

3.4.2 Two-phase flow with Evaporation

The refrigerant flow with evaporation is subdivided, in EVSIM model, in two flow patterns; annular flow and misty flow. The quality value of 0,85 was chosen as the division point between the two flow patterns.

3.4.3 Smooth tubes, annular flow

The correlation developed by Gungor et al. (1986) is used in the original EVSIM code, published in Domanski(1989), to calculate the evaporative heat transfer coefficient for annular flow in internally smooth tubes. This correlation was developed based on a databank which includes 4300 points by 28 authors using 7 different fluids. The kind of the correlation is coherent with Chen (1966) which recognizes two distinct mechanisms for the heat transfer: nucleate boiling and forced convection. The two-phase evaporation heat transfer coefficient, h_{an}, is given by a weighted average of the single-phase heat transfer coefficient, h_l, and the pool boiling heat transfer coefficient, h_{pool}, that accounts for the nucleate boiling contribution to the heat transfer:

$$h_{an}=E\cdot h_{sp}+S\cdot h_{pool} \tag{30}$$

$$h_l=\frac{0,023\cdot Re^{0,8}\cdot Pr_l^{0,4}\cdot k_l}{D_i} \tag{31}$$

$$h_{pool}=55\cdot P_r^{0,12}\cdot(-\log P_r)^{-0,55}\cdot M^{-0,5}\cdot q^{0,67} \tag{32}$$

$$E=1+24000\cdot Bo^{1,16}+1,37\cdot X^{-0,86} \tag{33}$$

$$S=(1+1,15\cdot 10^{-6}\cdot E^2\cdot Re^{1,17})^{-1} \tag{34}$$

In the case of a horizontal tube and a Froude number, Fr, smaller than 0.05, E and S must be multiplied by E_2 and S_2, respectively:

$$E_2=Fr^{(0,1-2\cdot Fr)} \tag{35}$$

$$S_2=Fr^{0,5} \tag{36}$$

The symbols used in Eqs.(30) to (36) have the following meaning:

$$Re=\frac{G\cdot(1-x)\cdot D_i}{\mu_l} \tag{37}$$

$$Fr=\frac{G^2}{\rho_l^2\cdot D_i\cdot g} \tag{38}$$

Where *Re* stands for the liquid Reynolds number, *Fr* the Froude number, D_i the tube inside diameter, *G* the refrigerant mass flux, *g* the acceleration of gravity, *M* the molecular weight, P_r the reduced pressure, Pr_l the liquid Prandtl number, *q* the heat flux, *x* the flow quality, μ_l the liquid dynamic viscosity, ρ_l the liquid density, *Bo* the boiling number and *X* represents the Martinelli parameter in accordance with Gungor et al.(1986). In the results of the present report, a comparison is done between results obtained from correlation: Gungor et al. (1986) and Liu et al. (1991). This second correlation differs from the first only in the coefficients *E* and *S*. The other relations are the same used in Gungor et al. (1986). In Liu et al. (1991) the terms *E* and *S* turns into:

$$E=[1+x\cdot \mathrm{Pr}_l\cdot(\frac{\rho_l}{\rho_v}-1)]^{0,35} \qquad (39)$$

$$S=(1+0{,}055\cdot F^{0,1}\cdot \mathrm{Re}_L^{0,16})^{-1} \qquad (40)$$

The Liu-Winterton correlation was implemented by the authors in EVSIM code. The analysis of its influence, the comparison with EVAP-COND results and experimental data are the scope of this study.

4. RESULTS

In Table (1) is presented a comparison between the total capacity for a evaporator obtained from: EVSIM using Gungor-Winterton correlation, EVSIM using Liu-Winterton correlation, EVAP-COND v2.3 and the experimental results from Chwalowski et al. (1989); and its discrepancies.

Table 1. Comparison between EVSIM results, EVAP-COND and experiment.

Air flow rate (m^3/min)	Tsat (°C)	Experiment (W)	EVSIM Gungor-Winterton (W)	EVSIM Liu-Winterton (W)	EVAP-COND (W)	Discrepancy(1) EVSIM Gungor-Winterton (%)	Discrepancy(1) EVSIM Liu-Winterton (%)	Discrepancy(1) EVAP-COND (%)
32.18	7.2 ± 0.2	7269	7618	7476	7910	4.8	2.85	8.82
22.84	7.2 ± 0.2	6434	6358	6220	6700	-1.18	-3.33	4.13
21.28	7.2 ± 0.2	6285	6117	5990	6450	-2.67	-4.69	2.63
15.96	7.2 ± 0.2	5415	5006	4881	5410	-7.55	-9.86	-0.09
11.49	7.2 ± 0.2	4421	4019	3912	4370	-9.09	-11.51	-1.15
27.99	10.0 ± 0.2	5381	5673	5547	5840	5.43	3.08	8.53
14.83	10.0 ± 0.2	3781	3747	3646	4020	-0.9	-3.57	6.32
13.95	10.0 ± 0.2	3915	3596	3496	3850	-8.15	-10.7	-1.66
12.45	10.0 ± 0.2	3449	3325	3228	3550	-3.6	-6.41	2.93
7.44	10.0 ± 0.2	2410	2257	2172	2370	-6.35	-9.88	-1.66

(1)Discrepancy (%) = [(simulation - experiment) * 100] / experiment

Where *Tsat* is the refrigerant saturation temperature at coil outlet, the total capacities are expressed in Watts and discrepancies in percentage form.

The experimental conditions are: air temperature = 26.0 ± 0.2 °C; air relative humidity = 0.51; refrigerant superheat at evaporator outlet = 4.4 ± 1.4 °C; Refrigerant 22 as the operating fluid. The evaporator is a single-slab and the airstream is perpendicular to the slab. More detailed experimental conditions are described in Chwalowski et al. (1989).

A mean of the discrepancies have been done and the results are: 4,97 % using EVSIM with Gungor-Winterton correlation, 6,59 % using EVSIM with Liu-Winterton correlation and 4,73 % using EVAP-COND v2.3.

The formula used to calculate the mean of the discrepancies is a simple arithmetic mean of the absolute values obtained from discrepancies.

The Figure (3) and (4) are representations of the capacities present in Tab. (1). The Fig. (3) is responsible for Tsat equal to 7.2 °C and the air flow rate variates from 11.49 m^3/min to 32.18 m^3/min. In Fig. (4) Tsat is equal to 10.0 °C and the air flow rate variates from 7.44 m^3/min to 27.99 m^3/min.

Both figures, as the Tab. (1), show that EVAP-COND program present a much greater accuracy when the air flow rate is lower.

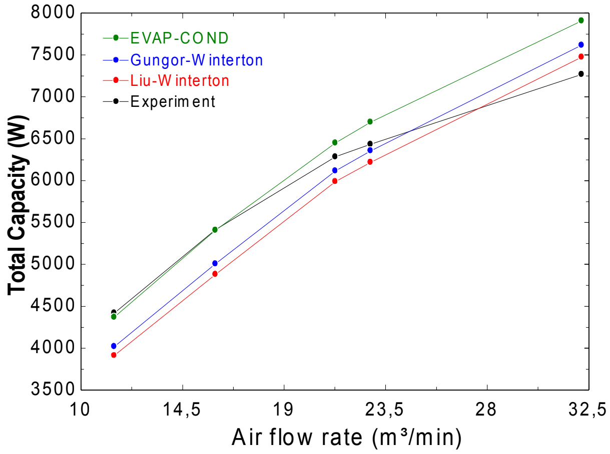

Figure 3. Capacities for refrigerant saturation temperature equal 7.2 °C

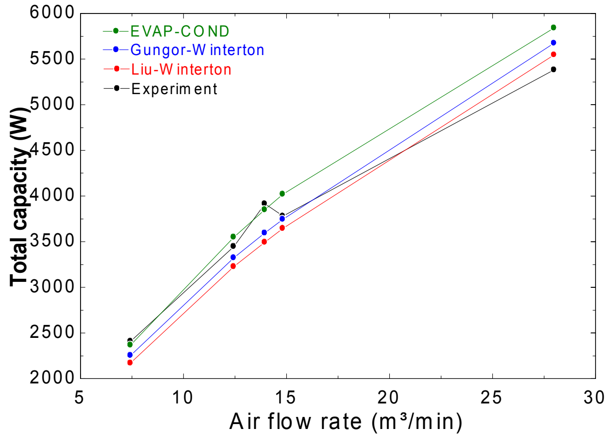

Figure 4. Capacities for refrigerant saturation temperature equal 10.0 °C

5. CONCLUSIONS

The three methods simulated demonstrate reasonable precision in predicting the total capacity of the evaporator.

Beyond accuracy, the EVSIM source code is known and published attached to Domanski (1989) together to the logic of all processes sketch, for advanced users is interesting because is possible for the user to change the code and adequate the program to his necessities.

However, the fact that EVAP-COND v2.3 has a interface and a lot of functionalities, that facilitates the operation, is a good advantage against EVSIM.

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