EXPERIMENTAL ANALYSIS OF AN EVAPORATIVE CONDENSER

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Abstract. *This work presents an experimental analysis for an evaporative condenser built in small scale, keeping the geometric similarity to real size equipment, made by a Brazilian manufacturer. The small scale condenser has a bundle of 210 copper tubes, with 6mm of external diameter arranged in 35 columns and 6 rows, assembled inside a glass enclosure to allow for the water and air flows visualization. The condenser operates under different water and air mass flow rates and uses R-22 as the refrigerant fluid. From the measured data, studies on the heat transfer are performed and the obtained experimental values are compared to those found in the literature. Such analysis is one of the main goals of this work. The R-22 local heat transfer coefficient is determined as a vapor quality function and compared to an average coefficient. The applied methodology consists on the determination of the flow pattern map, followed by the calculation of the transition regions based on the void fraction concept. The flow patterns are classified as fully stratified and the condensation model assumes that two types of heat transfer mechanisms occur within the tubes: film condensation and convective condensation. The results show the agreement of the some correlations present in the literature with the measured experimental data. The values obtained for the local heat transfer coefficient inside the tubes at the condensation zone present similar results to those mean values calculated for the vapor quality of about 50%. For the single phase of superheated vapor and subcooling liquid the coefficient did not present significant difference between local and mean values.*

Keywords: *Heat transfer coefficient, evaporative condenser, convective condensation, film condensation*

1. INTRODUCTION

The analysis of equipments for fluid refrigerant condensation in large size refrigeration systems, especially those operating with R-717 and R-22 are of great importance and arouse interest, since condensers with better performances can promote systems with lower initial and operational costs.

The goal of this work is to study the heat transfer in an evaporative condenser, as well as to evaluate the relationship among some of the measured quantities during its operation.

The correct mathematical modeling offers great advantages when used for operation and performance analysis of equipments, becoming easier the establishment of important correlations. Several works, using this approach have been developed with the purpose of better understanding the heat and mass transfer phenomena that occurs in evaporative condensers and cooling towers.

A similar methodology was found in Facão (1999), who developed an experimental work on a 10 kW nominal capacity cooling tower with indirect contact, obtaining heat and mass transfer coefficients. The results were similar to those found by Parker and Treybal (1961) and Niitsu *et al*. (1967).

Centeno (2005) determined experimentally the capacity of an evaporative condenser operating with R-717 in a Facility using as reference the ANSI / ASHRAE 64-1995 standard. Results were compared to those calculated according to Parker and Treybal (1961) formulation.

Flores (2005) also performed a mathematical modeling of an evaporative condenser applying the ε-*NUT* method using data from measurements of quantities referents to refrigerant fluid flow (R-717) and air flow which allowed for the determination of the equipment performance by both refrigerant and air sides. Some parameters were identified in order to achieve a better equipment performance.

Nakalski (2006) compared the deviation presented by those two noticed models (Centeno, 2005 and Flores, 2005), concluding that the one used by Flores (2005) presented a higher deviation than the procedure applied by Centeno (2005).

There are several environmental conditions in which the equipment can be exposed when a real system is considered. Thus, measurements become harder to be done and the variables uncertainly are increased. For that, an evaporative condenser in a small scale is built allowing laboratory experiments over different simulated environment conditions.

2. EXPERIMENTAL FACILITY

Parallel to the assembling of the small scale evaporative condenser, an experimental facility is built in order to perform controlled tests (Fig. 1). R-22 is used as the refrigerant fluid and it flows internally along the tubes of the evaporative condenser, a shell and tube evaporator and a liquid receiver, in a closed loop. Inside the evaporator tubes, hot water flows from an auxiliary heating device, placed at the top of the facility. Water is heated by a 2.7 kW (nominal power) electric element. R-22 is evaporated on the outer side of the tubes of this exchanger.

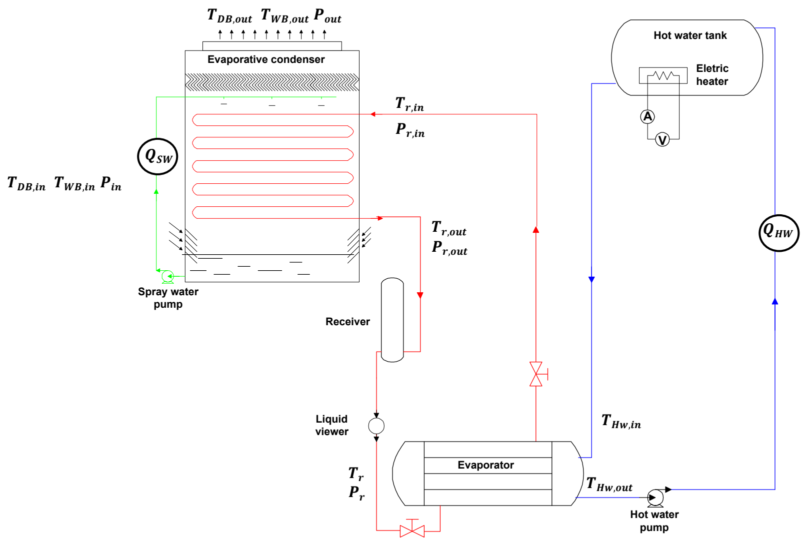


Figure 1. Test facility scheme

The red, green and blues lines represent, respectively, the R-22, spray water and hot water circuits. The measuring positions of more significant variables also are showed in this figure, and the symbols that represent them are: V (electrical tension, in volts), A (electrical current, in amperes), T_{DB} and T_{WB} (dry and wet bulb temperatures, in ^{o}C), T (R-22 and water temperatures, in ^{o}C), P (pressure, in bar) and Q (volumetric flow rate, in m^3/h).The refrigerant flow rate is driven by the heat transfer rate in the evaporator. The control of the hot water temperature on the water tank is performed by a PID controller.

The reason for choosing R-22 as refrigerant fluid is due to its lower toxicity in comparison to R-717, and also because R-22 does not attack copper materials.

2.1. Evaporative condenser

The evaporative condenser is built with geometric similarity to a commercially manufactured one, running on R-717 as refrigerant fluid in industrial refrigeration systems, with a scale factor of 1/4. This evaporative condenser and test facility can be seeing in Figs. 2a and 2b, respectively.

a) **b)**

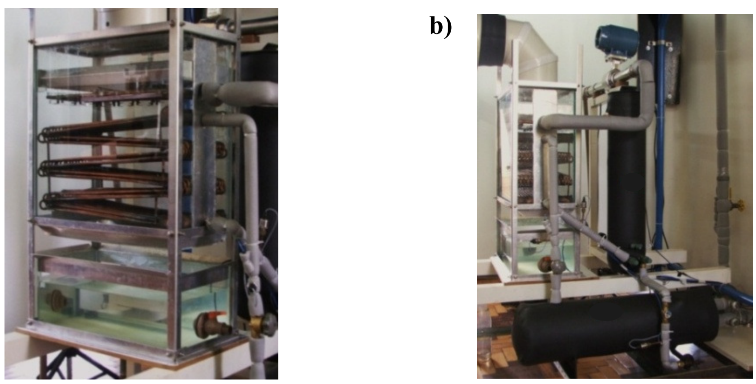


Figure 2. a) small scale evaporative condenser; b) experimental facility

Coil is built on copper tubes of 6.35mm external diameter (corresponding to commercial diameter of 1/4in), 35 columns of 6 tubes in a staggered arrangement, joined in a distributor in the top and in a collector in the bottom, with a total heat transfer area of 2.17 m^2.

A copper water spray distributor is placed above the coil, with 36 holes allowing for a uniform distribution of the spray over the coil. The spray water flow rate is controlled by a bypass in the pump circuit. The drift eliminator was made in aluminum with spacing of 1.8 mm.

The air flow leaves the condenser through 200 mm nominal diameter conduit connected to a centrifugal fan. The air flow rate is controlled by a frequency inverter acting on an electric driver. The water sump is made on glass and has a volume of 25 l. The water make up is completed at the end of each measure sample, providing sump water with more homogeneous temperatures and a steady state operation.

2.2 Measurements

All measured points are referred on Fig.1, and are described according to their circuit relations. Starting with the air flow circuit, the ambient air dry and wet bulb temperatures ($T_{DB,in}$ and $T_{WB,in}$) are measured by two PT-100 sensors. The inlet and outlet air pressure (P_{in} and P_{out}), as well as dry and wet outlet bulb temperatures ($T_{DB,out}$ and $T_{WB,out}$) are measured with a Vaisala PTU – 303 meter. The volumetric air flow rate is determined after a Venturi tube.

For the R-22 circuit, the inlet and outlet temperatures ($T_{r,in}$ and $T_{r,out}$) and pressures ($P_{r,in}$ and $P_{r,out}$) are measured for the superheated vapor and subcooling liquid, respectively. Also, pressure and temperature of the refrigerant fluid are measured in the pipe between the liquid receiver and the evaporator, mainly to verify the accuracy of this measurements and check the adiabatic behavior of the system. These temperatures are measured by thermocouples type J and the pressures with pressure transducers.

Spray water circuit includes the measurement of the water spray temperature T_w, measured inside the water sump. This same measurement is done right before the sprinkling nozzles, and results showed to be similar to those at the sump. Water spray flow rate (Q_{sw}) is measured by a flow meter.

Hot water circuit includes the measurement of the inlet and outlet hot water temperatures ($T_{Hw,in}$ and $T_{Hw,out}$) in the evaporator by thermocouples type J, and its flow rate (Q_{Hw}).

Finally, the refrigerant fluid mass flow rate ($\dot{m}_r$) is determined by a mass and energy balance around a control volume placed on the evaporator. The heat transfer involved in the process can also be determined by the electric power delivered to the hot water electric heater and by the air side, to compare results and ensure the measurement reliability.

Table 1 presents a summary of the instrumentation, as well as the measurement uncertainty of each instrument.

Table 1. Uncertainty of the instruments

Variable	Measuring instrument	Uncertainty
$T_{DB,in}$, $T_{DB,out}$, $T_{WB,in}$	PT 100	± 0,2 °C (at 20 °C)
$T_{WB,out}$	HUMICAP 180 sensor	± 1% (0 to 90%) ±1,7% (90 to 100%)
P_{in}, P_{out}	BOROCAP sensor	± 0.45hPa
$T_{Hw,in}$, $T_{Hw,out}$, $T_{r,in}$, $T_{r,out}$, T_r	Thermocouple type J	± 0.2 °C
$P_{r,in}$, $P_{r,out}$, P_r	Pressure transducer	± 1%
Q_{sw}	Flow Meter	± 0.5%
Q_{Hw}	Hydrometer	± 2%
$\Delta P_{venturi}$	Manometer	± 3%

The temperature sensors calibration was made in a stabilized thermal bath (thermostatic), within a range of 15 to 45°C, using as reference a calibrated PT-100 thermoresistance. The absolute deviation found for all sensors is about ±0.2°C to the PT -100 reference sensor.

Venturi tube was calibrated using the methodology described in Delmé (1983). Velocities were measured by a Pitot tube in different radial positions along the cross tube section. Figure 3 shows the air flow rate versus pressure drop in this Venturi tube.

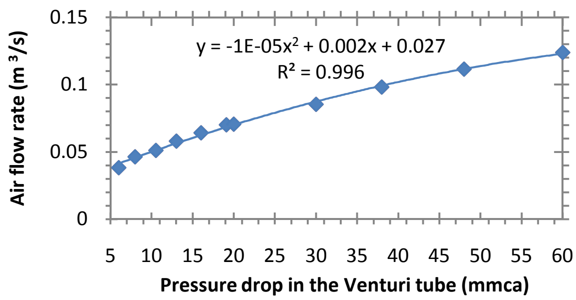


Figure 3. Venturi tube calibration curve.

Figure 4a displays the influence of condensation temperature and wet bulb temperature on the mass transfer between moisture air and water through the condenser, after experimental data from the facility. The wet bulb temperature of the inlet moisture air has a significant influence on the humidity ratio, unlike the trend of the condensation temperature.

Consequently the enthalpy changing (Fig. 4b) suffers a similar impact of these two variables, increasing with condensation and decreasing with web bulb temperature.

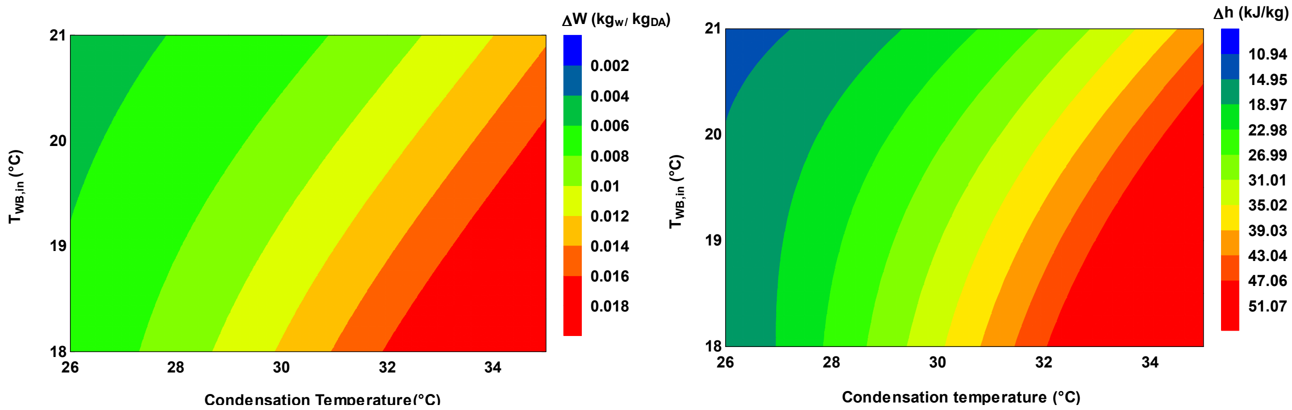


Figure 4. a) Humidity ratio increasing into condenser; b) Enthalpy increasing into condenser

3 Heat transfer rate

To evaluate the thermal capacity of an evaporative condenser, ambient heat transfer rate, or rejected heat $\dot{q}$ (kW), must be determined. This heat transfer rate can be calculated by two ways. The first one is through of the thermal balance involving the water spray stream and the air flow, as represented in Eq (1) (ASHRAE, 2005).

$$\dot{q} = \dot{m}_{air}\left(h_{air,out} - h_{air,in}\right) - \dot{m}_{wmu}h_w \tag{1}$$

where $\dot{m}_{air}$ is the air mass flow rate across the evaporative condenser (kg/s), $h_{air,in}$ and $h_{air,out}$ are, respectively, the air specific enthalpies in the inlet and outlet of evaporative condenser (kJ/kg), $\dot{m}_{wmu}$ is the make up water mass flow rate (kg/s) and h_w is the sump water enthalpy (kJ/kg).

The second one is establishing a thermal balance with the refrigerant and make up water flows (ANSI/ASHRAE, 1995):

$$\dot{q} = \dot{m}_r\left(h_{r,ent} - h_{r,sai}\right) - \dot{m}_{wmu}(h_w - h_{wmu}) \tag{2}$$

where $\dot{m}_r$ is the refrigerant mass flow rate (kg/s), $h_{r,in}$ and $h_{r,out}$ are, respectively, the refrigerant specific enthalpies (kJ/kg) in the inlet and outlet of tube bundle and h_{wmu} is the make up water specific enthalpy (kJ/kg). From Equations (1) and (2), results the heat transfer rate and the refrigerant mass flow rate.

After the heat transfer is determined, the experimental overall heat transfer coefficient, U_{exp} (Wm^{-2} $°C^{-1}$) can be calculated, through equations 3 and 4

$$U_{exp} = \frac{\dot{q}}{A.\Delta T_{exp}} \tag{3}$$

$$\Delta T_{exp} = T_{cond} - T_w \tag{4}$$

where A is the outside surface area of tubes (m^2) and, T_{cond} and T_w are the refrigerant fluid condensation and spray water temperatures (°C), respectively.

In this work, the overall heat transfer coefficient is determined for three heat transfer zones: dessuperheating, condensation and subcooling. In all zones, the heat transfer rate is determinate experimentally (Eq. (1)) and theoretically by correlations. The dessuperheating surface area is estimated at 1/6 of the total surface area, as well as for the subcooling heat transfer surface. Figure 5 shows a scheme of refrigerant, spray water and air flows over an elementary cross section of the tube. For simplicity, the refrigerant and spray streams flow in the same sense.

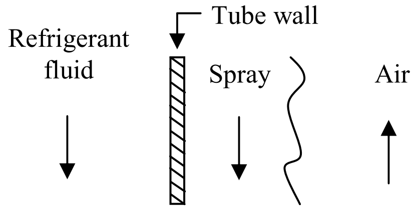


Figure 5. Schematic water, R-22 and air flows

The overall heat transfer coefficient is calculated for the region depicted in this Figure by Equation (5), in respect to the outer diameter d_{ext} (ASHRAE, 2000)

$$U = \frac{1}{\frac{d_{ext}}{d_{\text{int}}}\left(\frac{1}{h_{\text{int}}}\right) + \frac{d_{ext}}{d_m}\left(\frac{L}{k_T}\right) + \frac{1}{h_{ext}}} \tag{5}$$

where k_T is the tube thermal conductivity (W m^{-1} °C^{-1}), d_{int} and d_{ext} are the internal and external tube diameter (m), respectively, d_m is the mean tube diameter (m), L is the tube thickness (m), h_{int} is the heat transfer coefficient between refrigerant fluid and the internal tube surface (W m^{-2} °C^{-1}) and h_{ext} is the heat transfer coefficient between external tube surface and the spray water (W m^{-2} °C^{-1}).

3.1 Refrigerant fluid heat transfer coefficient

The internal heat transfer coefficient can be evaluated as an average or a local value. The local value must be determinate with the properties of the refrigerant fluid that are dependent mainly on the temperature of the single phase case, as well as in dessuperheating and subcooling regions. On these two last regions, the internal heat transfer coefficient (h_{int}) can be determinate using the Dittus- Boelter (1985, reprinted version) correlation that results in:

$$Nu = \frac{h_{int} d_{int}}{k} = 0.023 Re^{4/5} Pr^{n} \tag{6}$$

where k is the refrigerant thermal conductivity (W m^{-1} ^{o}C^{-1}), *Nu, Re* and *Pr* are the Nusselt, Reynolds and Prandtl numbers. The exponent of the Prandtl number (n) is 0.3 for dessuperheating and subcooling zones.

For the average heat transfer coefficient, the thermophysical properties in Eq. (6) are evaluated at the refrigerant fluid mean temperature, taken at the inlet and outlet considered regions.

The flow map pattern, that is a function of the void fraction, is a mandatory information in order to estimate the local heat transfer coefficient in the two phase zone, as it changes with vapor quality.

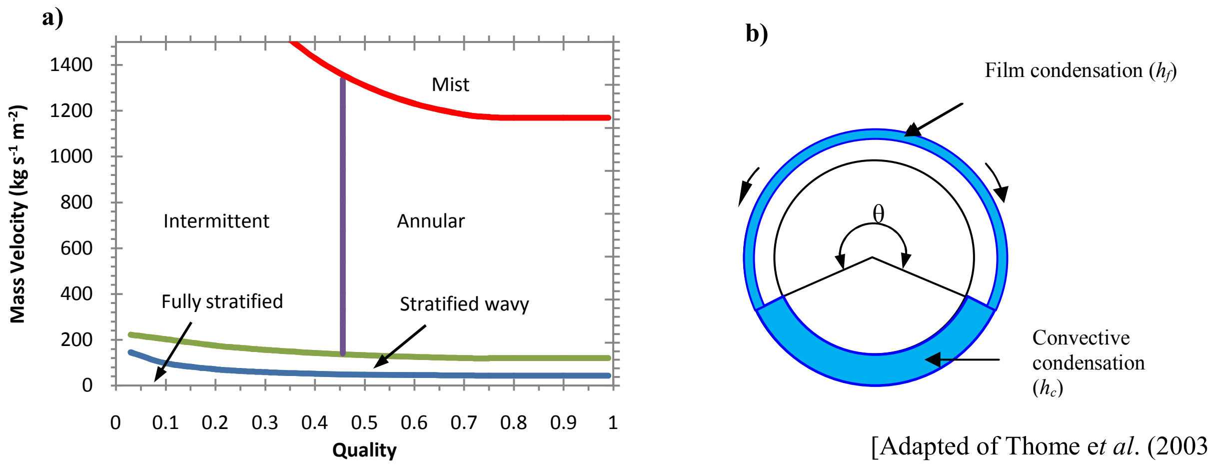
El Hajal *et al*. (2003) proposed a new version of a two phase flow pattern map for condensation inside horizontal plain tubes, based on the original work of Kattan *et al*. (1998) for flow boiling model, completed by Thome *et al*.(2003) to determinate the local heat transfer coefficient.

Presently, flow patterns are classified as: fully stratified flow, stratified wavy flow, intermittent flow, annular flow, mist flow and bubbly flow. However, considering that condensation in this work is a gravity controlled process, the flow occurs at very low mass velocities and a fully stratified flow in all measuring sample is verified. In this case, when the saturated vapor enters in a condensation zone, it forms a liquid layer in the tube bottom and a condensing film around the upper tube perimeter. Thus, the heat transfer coefficient must account for these two transference mechanisms.

Figure 6a shows a flow map pattern while Fig 6b shows the convective condensation (where is applied h_c) and film condensation (where is applied h_f) and their respective heat transfer surface area. All measuring samples of this study presented mass velocities lower than limit transition between fully stratified flow and stratified wavy flow. Therefore, all analysis and equations will be here performed for fully stratified flows.

The θ angle (rad) in this case is the stratified angle and defines the regions where the heat transfer coefficients will be of the film condensation type or of the convective condensation type. Thus, for a tube radius r, the internal heat transfer coefficient is:

$$h_{int} = \frac{r\theta h_f + (2\pi - \theta) r h_c}{2\pi r} \tag{7}$$



[Adapted of Thome *et al.* (2003)]

Figure 6. a) Flow map pattern for R-22 inside tubes condensation; b) Heat transfer coefficient and their respective perimeters.

A non iterative equation can be used to determine the θ angle, as a function just of void fraction, ε

$$\theta = 2\pi - 2\left[\pi(1-\varepsilon) + \left(3\frac{\pi}{2}\right)^{1/3}\left[1 - 2(1-\varepsilon) + (1-\varepsilon)^{1/3} - \varepsilon^{1/3}\right] - \frac{1}{200}(1-\varepsilon)\varepsilon\left[1 + 4\left((1-\varepsilon^2) + \varepsilon^2\right)\right]\right] \quad (8)$$

The void fraction ε is a logarithmic mean between homogeneous void fraction (ε_h) and the non homogeneous drift flux void fraction model of Rouhani and Axelsson (1970) (ε_{ra}). These void fraction methods are represented by following equations:

$$\varepsilon_h = \left[1 + \left(\frac{1-x}{x}\right)\frac{\rho_L}{\rho_v}\right]^{-1} \quad (9)$$

$$\varepsilon_{ra} = \frac{x}{\rho_v}\left(\left[1 + 0.12(1-x)\right]\left[\frac{x}{\rho_v} + \frac{1-x}{\rho_L}\right] + \frac{1.18(1-x)\left[g\sigma(\rho_L-\rho_v)^{0.25}\right]}{G{\rho_L}^{0.5}}\right)^{-1} \quad (10)$$

$$\varepsilon = \frac{\varepsilon_h - \varepsilon_{ra}}{ln\left(\frac{\varepsilon_h}{\varepsilon_{ra}}\right)} \quad (11)$$

where ρ_l and ρ_v are, respectively, the liquid and vapor specific masses (kg/m^3), x is the vapor quality, g is the acceleration of gravity (m/s^2), σ is the surface tension (N/m) and G is the mass velocity (kg s^{-1} m^{-2}).

Knowing the tube internal cross sectional area A (in m^2) and the void fraction, the cross sectional area of tube occupied by liquid and vapor (A_L and A_v, in m^2) and liquid thickness (δ, in m) can be determined from

$$A_L = A(1-\varepsilon) \quad (12)$$

$$A_L = A\varepsilon \quad (13)$$

$$\mathrm{A_L} = \frac{1}{8}(2\pi - \theta)\left[\mathrm{d_{int}} - (\mathrm{d_{int}} - 2\delta)^2\right] \quad (14)$$

Convection heat transfer coefficient is defined as:

$$h_c = 0.003 Re_L^{0.74} Pr_L^{0.5}\frac{k_L}{\delta}f_i \quad (15)$$

The interfacial roughness factor (*fi*), the Reynolds and Prandtl liquid numbers (Re_L and Pr_L) are calculates by

$$Re_L = \frac{4G(1\text{-}x)\delta}{(1\text{-}\varepsilon)\mu_L} \quad (16)$$

$$Pr_L = \frac{4G(1-x)\delta}{(1-\varepsilon)\mu_L} \quad (17)$$

$$f_i = 1 + \left(\frac{u_v}{u_L}\right)^{1/2}\left[\frac{(\rho_L-\rho_v)g\delta^2}{\sigma}\right]^{1/4}\left(\frac{G}{G_{strat}}\right) \quad (18)$$

where μ_L is the liquid dynamic viscosity (Ns/m^2), k_L is the liquid thermal conductivity (W m^{-1} ^{o}C^{-1}), C_{pL} is the liquid specific heat, G_{strat} is the stratified flow transition mass velocity (kg s^{-1} m^{-2}), u_L and u_v are the liquid and vapor mean velocities (m/s), respectively. The equations (19), (20) and (21) are used in order to calculate G_{strat}, u_L and u_v

$$G_{strat} = \left[\frac{(226.3)^2\left(\frac{A_L}{d_{int}^2}\right)\left(\frac{A_V}{d_{int}^2}\right)^2 \rho_v(\rho_L-\rho_v)\mu_L g}{x^2(1-x)\pi^3}\right]^{1/3} + 20x \quad (19)$$

$$u_v = \frac{Gx}{\rho_v\varepsilon} \quad (20)$$

$$u_L = \frac{G(1-x)}{\rho_L(1-\varepsilon)} \quad (21)$$

The film condensation heat transfer coefficient (h_f, in W m^{-2} ^{o}C^{-1}) that occurs in the upper perimeter of tube is a mean value of this perimeter and is determinate by the Eq. (21), in which the thermophysical properties are evaluated in the mean value between tube wall and refrigerant saturation temperatures (T_p and T_{sat}, respectively).

$$h_f = 0.728\left[\frac{\rho_L(\rho_L-\rho_v)gh_{fg}k_L^3}{\mu_L d_{int}(T_{sat}-T_p)}\right]^{1/4} \quad (22)$$

in this equation h_{fg} is the latent heat of vaporization of the R-22 (J/kg^{-1} ^{o}C^{-1}).

The average heat transfer coefficient is given by the equation (Chato, 1962)

$$\bar{h}_{int} = 0.555\left[\frac{\rho_L(\rho_L-\rho_v)gh'_{fg}k_L^3}{\mu_L d_{int}(T_{sat}-T_p)}\right]^{1/4} \quad (23)$$

where h'_{fg} is the latent heat of vaporization of the R-22 corrected (J/kg^{-1} ^{o}C^{-1}) which seeks to include the effect of the temperature reduction of the condensed liquid below of the saturation temperature and is determinate by

$$h'_{fg} = h_{fg} + \frac{3}{8}c_{p,L}(T_{sat} - T_p) \quad (24)$$

3.2 Spray water heat transfer coefficient

The heat transfer coefficient between external tube surface and spray water was calculated from some well known correlations available in literature and were used to compare the results obtained from correlations with those obtained from experimental data.

Tovaras *et al*. (1984) (appud Zalewski and Gryglaszwski, 1997) proposed a correlation for this heat transfer coefficient as a function of water Prandtl number (Pr_w), water and air Reynolds number (Re_w and Re_{air}). This correlation for water flowing downstream across the horizontal tubes has the following form:

in the range: $690 < Re_{air} < 3000$, $Nu_w = 3.3 \times 10^{-3}\, Re_w^{0.3} Re_{air}^{0.15}\, Pr_w^{0.61}$;
in the range: $3000 < Re_{air} < 6900$, $Nu_w = 1.1 \times 10^{-2}\, Re_w^{0.3}\, Pr_w^{0.62}$;
for $Re_{air} > 6900$, $Nu_w = 0.24\, Re_w^{0.3}\, Re_{air}^{-0.36}\, Pr_w^{0.66}$
where:

$$Re_w = \frac{4\Gamma}{\mu_w} \quad (25)$$

$$Re_{air} = \frac{u_0 d_{ext}\rho_{air}}{\mu_{air}} \quad (26)$$

In these equations μ_w and μ_{air} are the dynamic viscosities of water spray and air (kg s^{-1} m^{-1}), ρ_{air} is the specific mass of the air (kg/m^3), u_0 is the air velocity in the smallest cross section (m/s), Γ is the spray water mass flow rate per unit length of tube (kg s^{-1} m^{-1}). The external heat transfer coefficient (h_{ext}) is given by:

$$Nu_w = \left(\frac{\nu_w^2}{g}\right)^{1/3} \frac{h_{ext}}{k_w} \tag{27}$$

In this equation k_w is the spray water conductivity ($Wm^{-1}\ ^oC^{-1}$) and it is available for $160 < Re_w < 1360$ and $4.3 < Pr_w < 11.3$.

The Parker and Treybal (1961) studied 5 different situations obtaining for evaporative condenser case a correlation available for temperatures between 15 and 70 ^oC and $1.4 < \Gamma/_{dext} < 3.0$ kg/s^{-1} m^{-2} which is given by

$$h_{ext} = 704(1.39 + 0.022T_w)\left(\frac{\Gamma}{d_{ext}}\right)^{1/3} \tag{28}$$

Mizushina *et al*. (1967), developed a correlation to a greater range of applicability than Parker and Treybal correlation (1961), $0.2 < \Gamma/_{dext} < 5.5$ kg/s^{-1} m^{-2}

$$h_{ext} = 2102.9\left(\frac{\Gamma}{d_{ext}}\right)^{1/3} \tag{29}$$

The correlation proposed by Leidenfrost e Korenic (1982), was developed for a in line bundle tube with a external diameter of 15.9 mm and is quite similar to Eq. (29)

$$h_{ext} = 2064\left(\frac{\Gamma}{d_{ext}}\right)^{0.252} \tag{30}$$

Niitsu *et al*. (1967), studied evaporative coolers with bare and finned tubes finding for the last one larger heat transfer coefficients for $0.5 < < \Gamma/_{dext} < 3.2$ kg/s^{-1} m^{-2}

$$h_{ext} = 990\left(\frac{\Gamma}{d_{ext}}\right)^{0.46} \tag{31}$$

Very similar correlation was proposed by Dreyer and Erens (1990):

$$h_{ext} = 2843\left(\frac{\Gamma}{d_{ext}}\right)^{0.384} \tag{32}$$

4. RESULTS AND DISCUSSION

The heat transfer coefficients were calculated after the above presented correlations using experimental data from the facility.

Figure 7a shows results for the external heat transfer coefficient h_{ext} calculated by 6 correlations (eq 27 to 32) with data from a set of 39 experimental samples. Results display a significant difference among then, reaching around 1830 $Wm^{-2}\ ^oC^{-1}$. The lower values were obtained for Niitsu *et al*. (1967) and Tovaras *et al*. (1984) correlations followed by the Parker and Treybal (1961). Dreyer and Erens (1990) correlations as well as Leidenfrost and Korenic (1982) and Mizushina *et al*. (1967) correlations displayed the higher values.

Figure 7b shows the overall heat transfer coefficient U for a set of 5 experimental samples, calculated first by Eq. (3) and furthermore using the same correlations for h_{ext} on Eq. (5). R-22 heat transfer coefficient for the two phase flow was taken as an average value. The better agreement was verified to Niitsu *et al*. (1967) and Tovaras *et al* (1984).

All results concern the condensation region of the evaporative condenser, because negligible differences were found in the dessuperheating and subcooling zones.

The differences between average and local heat transfer coefficients for internal flow (h_{int}) in the dessuperheating and subcooling regions are negligible, and the adoption of any previous procedures will result in a good agreement with experimental values. Taking as an example, a sample in which the superheating vapor and condensation temperatures for the R-22 are, respectively 40.6 and 32.7^oC, the difference between the higher and lower values found for h_{int} was 0.42 $Wm^{-2}\ ^oC^{-1}$, evaluating the refrigerant thermophysical properties at the local temperature. Comparing this higher value with the one calculated considering R-22 thermophysical properties at the bulk temperature, the difference found was 0.29 $Wm^{-2}\ ^oC^{-1}$ (approximately 0.3%). As a consequence, the overall heat transfer coefficient also has a negligible variation.

The same considerations were done for subcooling zone, where the subcooled liquid temperature is 27^oC. In this case the difference between the highest and the average convection heat transfer coefficients resulted on a difference lower than 1% for U value, being also negligible.

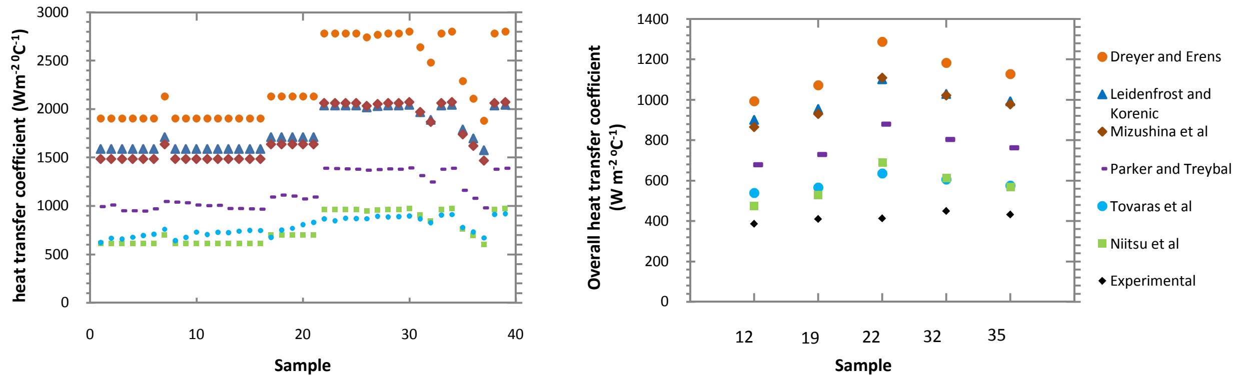


Figure 7. a) external heat transfer coefficient (h_{ext}); b) overall heat transfer coefficient (U)

The comparison between the R-22 average and local h_{int} in the condensation region can be seen in the Fig. 8a. Both ways of calculation result in very close values for vapor quality of approximately 0.6. However, for vapor quality close to 0, h_{int} was almost half of the average one. For vapor quality close to 1, the difference between these coefficients reached almost 23%. Another interesting aspect is that the local coefficient h_{int} calculated for extreme vapor qualities values (close 0 or 1) displayed a difference of the order of 2100 Wm^{-2} $^{o}C^{-1}$.

Figure 8b shows the overall heat transfer coefficient U calculated using three different approaches: predefined quality conditions from 0 to 1 (with a step of 0.25), with average heat transfer coefficient and experimentally. For all these cases, U was determinate with the aid of the 6 studied correlations for the external heat transfer coefficient. One sample was arbitrarily chosen to represent the problem, showed in the Figures 8a and 8b.

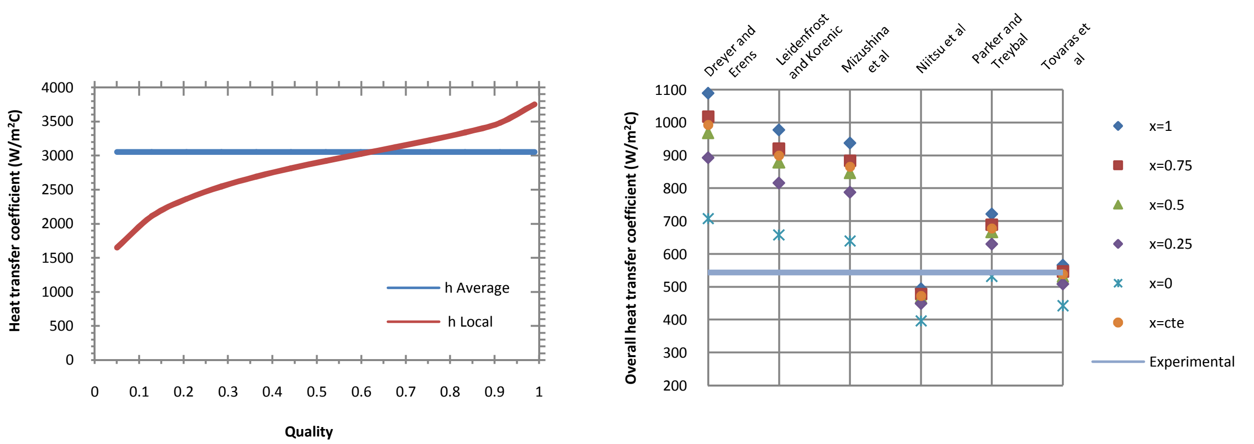


Figure 8: a) Local and average internal heat transfer coefficients; b) Overall heat transfer coefficient for 6 correlations for exteral heat transfer coefficient

The better agreement for U with experimental results is found when h_{ext} is calculated with Tovaras *et al*. (1984), Niitsu *et al*. (1967) and Parker and Treybal (1961) correlations. For these correlations, the best agreement is found for vapor quality around 0.5, corresponding to the average U.

5. CONCLUDING REMARKS

A small evaporative condenser was investigated using data obtained from tests performed in laboratory. Some correlations were used to determine the refrigerant fluid and spray water heat transfer coefficients with the purpose of calculating the overall heat transfer coefficients. In the condensation tube region, the average and local h_{int} coefficient presented very close results for vapor qualities about 0.6 and resulted in a good agreement between U and U_{exp} when combined with correlations proposed by Niitsu *et al*. (1967) or Tovaras *et al*. (1984) to the tube - water heat transfer coefficient (h_{ext}). Dessuperheating and subcooling regions did not present significant differences between average and local refrigerant heat transfer coefficients and all correlations used by h_{ext} resulted in a good agreement between U and U_{exp}.

The influence of condensation temperature on the humidity ratio and enthalpy changes of the moisture air shows that greater condensation temperatures lead to larger capacities of the equipment. On the other hand, it contributes to decrease the performance coefficient of the refrigeration cycle in which this equipment often works. At lower wet bulb temperatures the energy gain is larger for all the tests realized.

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