ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

ANALYSIS OF PERFORMANCE IN TEST BOARD OF FINNED-TUBE HEAT-EXCHANGER

First Author's Name Leandro Cristino Oliveira Pereira
Second Author's Name Paulo Eduardo Lopes Barbieri
Institution and address
Universidade Federal dos Vales do Jequitinhonha e Mucuri, Campus JK, Rodovia MGT 367 – Km 583, nº 5000 – Alto da Jacuba. Diamantina – MG, Brasil , cep 39100-000
Centro Federal de Educação Tecnológica de Minas Gerais – CEFET-MG – Av. Amazonas, 7675 – Nova Gameleira – Belo Horizonte – MG, Brasil , cep 30510-000

e-mails leandro.pereira@ict.ufvjm.edu.br ; barbieri@des.cefetmg.br

Abstract. *This paper consist in set up the parameters of heat transfer and pressure drop to compact heat exchangers of the coil type for thermo systems, with the purpose of optimize designs, for a reduction of energy wear. For this, Will be bulding a experimental apparatus with focus of produce experimental dates. This experimental apparatus is based in the Ashare 33 norm. The Calibration analysis will purpose, such as the error analysis. The gives will compared with the literature, to validate the test.*

Key words: experimental apparatus; thermal hydraulic performace, finned-tube heat-exchanger

1. INTRODUCTION

The pursit of energy efficiency need of science research for development of inteligence forms to save energy. No only involving the devices necessaries to generation or energy consumption, but the own process, with often, need be reinvented due the new requeriments of world, such as, machines and equipments that do not harm the environment. In this context, the study of finned-tube heat-exchangers used of air treatment, presents as devices for be investigation to obtain the energy eficiency.

Analysis the thermal performance of heat exchangers use the rate of heat transfer equation and the 1ª law of thermodinamic. However, due the many variables, such procedure forbid a graph description and immediate ralation among one. But, this variables can be joint in the small number of dimensionless parameters with allow that representation. Which this intention arise the efectivess method – NUT (ε - NUT) shows for Kays and London (1998).

Heat-exchanger that a fluid change of phase, condenser or evaporator, C* tends to zero, once that maximum thermal capacity tends to infinity, because one fluid change of phase and the temperature still constant. This case, all geometries shows C*=0, so that alls equations tend a expression:

$$\varepsilon = 1 - e^{NUT} \tag{1}$$

The estimate thermo-hydraulic of finned-tube heat-exchanger air cooled is held by experimental procedures, to obtain empiric relations to assist the projects and especification such as heat-exchangers, contributing to advance of energy eficiency in thermal systems.

Zoghbi Filho (2004) makes a test-board to raise the thermal-hidraulic performance of a group of coils. This board consists of two circuits: the air circuit and water circuit. The objective of air circuit is to impose control conditions of flow and temperature for air. The water hot circuit establishes desired conditions of pressure and temperature, as the flow that loop in inner of coils.

Zoghbi Filho (2004) also used a program for calculate the performance of coils by the ε-NUT method and comparete with correlations of literature. He concluded that: The effectiveness increase with increasing of number of rows, for a value determined of NUT, that due increase of heat transfer area. The coils tested, with number of rows over more than two, the results of simulation program underestimated the correlation. This diference was assigned to the altenate arragement (quincunce), which promotes diferences between a inlet tempeature and outlet of fluids, which do not see on linear arrangement.

The correlation established by Zoghbi Filho (2004) are based on experimental of heat transfer and pressure drop of coils, given by the expressions:

$$j = C_1 \, Re_{DH}^{C_2} \tag{2}$$

$$f = C_3 Re_{DH}^{C_4} \tag{3}$$

Zoghbi Filho (2004) concluded that both the empirical works such as numerical suggest that there are geometrics effects on performance of coils associate with thermo hydraulic mechanism of air flow. Thus, the coefficients C_1 until C_4 were obtained from esperimental results, collected for test board, mode to incorporate the geometric effects on thermal hydraulic performance of coils.

The Zoghbi’s review involved a deep study about the influence of geometrics parameters on thermal hydraulic performance of finned-tube heat transfers. He developed a experimental apparatus to collect data to compare with numerical simulation and the correlations propose. Zoghbi Filho (2004) investigate the thermal hydraulic mechanisms occurring due a form of flow and its influence on coil performance, beyond study the geometric effects of geometric of different surface.

Dong et al (2007) did a study of compact heat transfers with louver fin by experimental apparatus to obtain correlation for one. This study the tested fins are the multi-louvered. Due it’s characters and great thermal hydraulic performance.

Dong et al (2007) used too the ε-NUT method on your study obtain this expression:

$$\varepsilon = 1 - \exp \frac{NTU^{0,22}}{C^* \times \left[\exp\left(-C^* NTU^{0,78} - 1 \right) \right]} \tag{4}$$

The focus of Dong’s study is the effect of geometric parameters on performance of heat transfer. Then he did several tests for check these effects. The parameters varied are: fin density, fin height, fin length and opening angle.

ISSN 2176-5480

Among these Dong et al (2007) found the fin density is the main parameter that affect the global coefficient of heat transfer.

Other research about heat-exchanger was realized by Barbosa et al (2009). He found the effect of number of rows on thermal hidraulic performance, for coils as used evaporators with fin pitch lower than 5mm. He found the performance of pump is related with capacity of heat transfer with evaporator containing fins with lenght lower and more number of rows. That study investigation the effects of number of rows and the great distance between fins (fin pitch) to avoid the ice formation. A big problem on compact heat exchangers used on domestic coolers. He also built a experimental apparatus for data collected necessary, as standard ASHARE 37 (1998).

Barbosa et al (2009) also observed the pressure drop is related with the flow, the pressure drop increase with increased of flow. The influence of number of rows on pressure drop is showed in the coil tests, in wich it has almost the same number of fins, but different numbers of row. The heat transfer global coefficient increase with the number of rows due the increase of surface area. This form, there is a decrease of friction factor and Colburn factor with increase of Reynolds number of air as expected. The form of correlations j and f, showed that equation (5) and (6), reflect the experimental observation of Barbosa et al (2009) as for number of rows, showing the strong influence on pressure drop and the heat transfer coefficient. However, this parameter was considered only in the correlation of *f*.

$$j = 0,6976\,Re_a^{-0,4842}\varepsilon^{-0,3426} \tag{5}$$

$$f = 5,965\,Re_a^{-0,2948}\varepsilon^{-0,7671}(N_f\,/\,2)^{-0,4436} \tag{6}$$

Barbosa et al (2009) found that the energy of pumping depends of heat transfer capacity, as calculated by the empirical correlation. On low heat transfer capacity, all evaporator can supply the cooler need with the pump energy, moreover, the pump performance is related with the heat transfer capacity with evaporator of smaller width, this means more number of rows. With heat transfer low, indicating that less number of rows add to pumping performance without increase the heat transfer.

2. EXPERIMENTAL APPARATUS

The experimental apparatus was built to simulate real situations of work in which the coils are subjected. It's composed of two circuits: air circuit and hot water circuit. The air circuit aims impose control condictions of velocity and temperature of air. The duct systems (fig.1) consists of tree valves; two honey cone straightener; the test section were is located the heat transfer; and the mensuring devices of temperature, deferencial pressure and air flow, according ASHRAE 33 (2000). Above this section there is a return duct for future air treatment.

The hot water circuit allows the desired condictions of enthalpy (pressure and temperature) and fluid flow that circulate in inner tube (fig.2). Consist a pump of 0,75CV; a needle valve; a flowmeter; a water reservoir with electrical resistence, this reservoir is isolated to avoid heat losses. The eletrical resistences is actuated by switch, controlled by a signal of a resistence thermometer, the PT100.

ISSN 2176-5480

The test board has a acquisition system (Agilent 34980A), fundamental for the collection and processing of data. All out signs of transducer of temperature and flow send at acquisition system. Consist for two plates analog digital converter and two terminals of connection installed on computer PC Pentium 980MHz. The plate converter has the function of transform the analogical sign to digital sign.

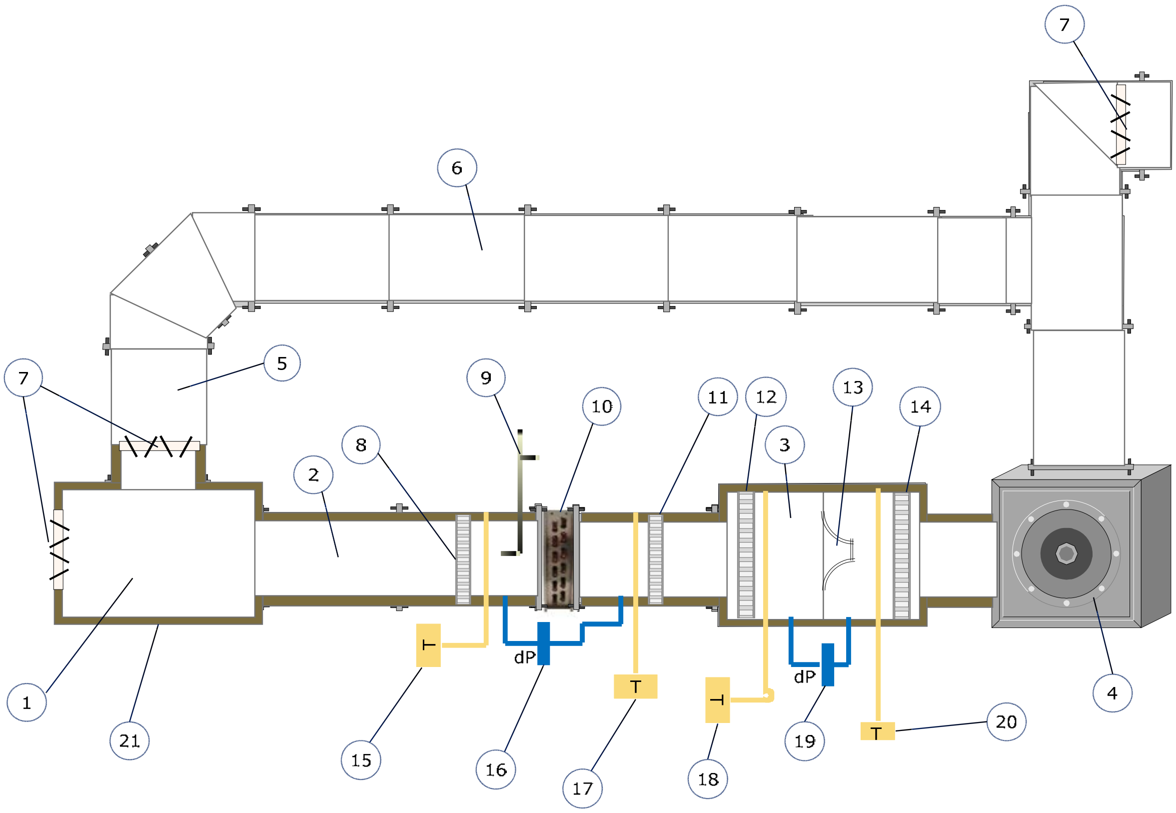


Figure 1 – schematic air circuit of test-board

1-mixing chamber, 2-test section, 3-nozzle chamber, 4-fan, 5-return duct, 6- return section, 7-damper, 8-honeycomb, 9-pitot tube, 10- coil 11- honeycomb, 12- honeycomb, 13- nozzle plate, 14- honeycomb, 15-mensuring temperature, 16-differencial pressure transducer, 17- mensuring temperature, 18- mensuring temperature, 19- differencial pressure transducer, 20- mensuring temperature, 21-therma isulating.

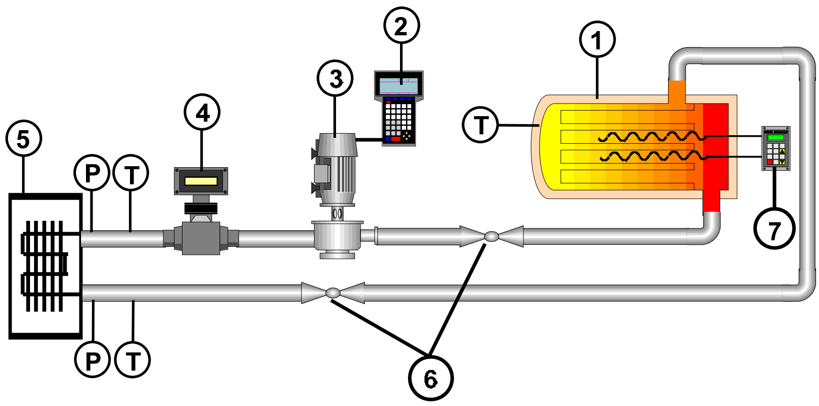


Figure 2 – schematic water circuit

ISSN 2176-5480

1-water tank, 2-variable frequency drive, 3-pump, 4-flow mensuring, 6-gate, 7-electrical resistence, T- mensuring temperature, P- mensuring pressure.

The tests on board provide the value of air flow, through the pressure differential mensuring on nuzzle chamber, water flow, drop pressure in coil, middle temperature at inlet and outlet of air and water. So, with this dates, is possible calculate the thermal hydraulic performance of coil, expressed in j factor for Reynolds and f factor for Reynolds.

The meansuring of temperature is doing on entrance of heat Exchange by thermocouples. Is fixed five before of heat exchanger and five after the one (figure 3). Which the middle those thermocouples is calculated the energy balance.

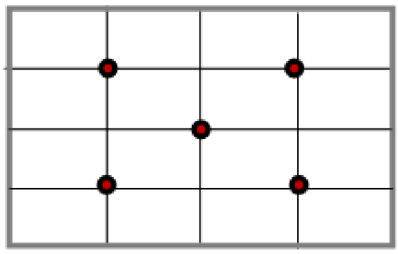


Figure 3 – Temperature meansuring station

The calibrating of instruments need to ensure the reliability of results. The error analysis was taken based the procedure for ABNT (2003) and ASME (1985), this is considered the combined uncertainty.

3. TREATMENT OF RESULTS

To determine the global effective of finned surface need to determine the effectiveness of own fin. The estimate of fin effectiveness with geometric developed to get up the thermal hydraulic performance makes complex the calculus of effectiveness. The methodology used to estimate the value of convective coefficient that incorporate the global effectiveness of coil. To determine the convective coefficient is made a series of calculates as departing from mensuring of parameters obtained to test board.

The global performance of finned-tube surface (η) is related which heat rate as the use of fin to offer, on coil air cooling, that performance is principal parameter to determinate your capacity of heat exchange, but is difficult avaliation, because the complex geometric. This paper was used the development by Schimidt (1949) apud Zoghbi Filho (2004), were the fin is divided on regions of influence of tubes, the fin is considered circular of section rectangular of same area. This is empirical method. The heat coefficient is considered constant on fin surface. The problem is find the (η_a) for the rectangular fin. The Schimidt (1949) study this problem that simple form, he based of

ISSN 2176-5480

selection of one fin with R_{eq} (ratio equivalent) which the performance is same of rectangular fin.

To the calculation of performance thermal hydraulic the parameters more important to define are the areas of heat exchange and flow. To the heat exchange was used the total area, which is characterized by sum of primary area and secondary area. The primary area is considered the area of heat exchange of tube surface without fin area. The secondary area is the fin surface, that is bigger area of heat exchanger, with the goal of increase of heat exchanger. The areas related with flow are frontal area, which is the area of section view in the front of heat exchanger, or the area of duct where the heat exchanger is mounted. Calculated multipling the length o heat exchanger by your height. The area free flow (Ac), is the area where the air flow passes by the heat exchanger without be blocked, also can be called minimum area, due to be only small space understood between the fins and the tubes.

One of the goals this work is available the energy used for the fan, on the air side that related with the pressure variation caused by the coil. Are three parameters related with the pressure drop: pressure drop related with friction; pressure drop related with acceleration; pressure drop related with inlet and outlet of air. The equation (7) shows the de pressure drop of air side due the effects of friction and acceleration obtained by Kays and London (1998).

$$\Delta P_{ar} = \frac{G_{max}^2}{2\rho_e} \cdot \left[\underbrace{f \cdot \frac{A_e}{A_c} \cdot \frac{\rho_e}{\rho_m}}_{\text{Friction}} + \underbrace{\left(1+\sigma^2\right)\left(\frac{\rho_e}{\rho_s} - 1\right)}_{\text{acceleration}} \right] \quad (7)$$

In which σ is the ratio between free area of flow (A_c) and the front area of coil (A_f),ρ_e is the specific mass of air in inlet, ρ_s is the specific mass of air in outlet, ρ_m is the middle specific mass of air, G_{max} is the mass velocity of air calculate on maximum velocity of air. The first term of equation represents the pressure drop to friction and the second term represents de pressure drop to acceleration.

With the mensuration of parameters, pressure, temperature and flow, can be the effectiveness of heat exchanger that is the performace. This calculus is realized over the equation (1), which is a fundamental equation and need of parameters of water temperature and air temperature. Otherwise is to equation (7) on air side and the equation (8) on inlet side of water, established for ESDU (2003) specifically for coils. That equation were used on the calculus of effectiveness fin.

$$\varepsilon = \frac{1}{C^*}\left[1 - \left(1 + C^* k^2\right) \times e^{-2kc^*}\right] \qquad k = 1 - e^{(-NUT/2)} \quad (8)$$

$$\varepsilon = \left[1 - \left(1 + \frac{k^2}{C^*}\right) e^{-2k/C_r}\right] \qquad k = 1 - e^{(-NUT \times C_r/2)} \quad (9)$$

The inlet temperature and outlet temperature of air side, and the water side too are fundamental to this parameter, and also the relation between the thermal capacity, represented by C*. With the values NUT, get calculate the total resistance, hence determining the external resistance of air side and the external convective coefficient, since the

ISSN 2176-5480

internal resistance and the wall can be evaluated. In Works involving the thermal performance is established a criterion to defined the best coil. For attend the specific condition of momentum conciliate with the heat transfer each project, and the restriction of space and cost, aims to the thermal performance based the first and second law of thermodynamic. The parameters j and f are used this criterion, that refer the selection of coils over the hydraulic of coefficient of heat transfer joint to air side due the drop pressure, Zoghbi Filho (2004).

The test board of CEFET-MG was construction to tests of finned-tube heat-exchanger, with the objective of evaluate the thermal hydraulic performance related of air side. So this experimental apparatus can evaluate the operation conditions shows on table 1. All out signs of transducer is sends to acquisition system.

Table 1 - Operating conditions evaluated by experimental setup

Paramater	**Variable**	**Sensor**	**Uncertainty**
Air velocity	3 a 5 m/s	anemometer	0,6 m/s
relative humidity	0 a 100%	Relative Humidity Sensor	2%
Inlet temperature of air	Conditions outside air	Thermocouple type T	0,638 ºC
Outlet temperature of air	Conditions outside air	Thermocouple type T	0,705 ºC
Water flow	0 a 1,3 m³/s	electromagnetic	0,12 %
Inlet temperature of water	35°C a 45°C	Thermocouple type T	0,747 ºC
Outlet temperature of water	35°C a 45°C	Thermocouple type T	0,756 ºC
Drop pressure of air side		diaphragm	0,003143 mbar

The physical parameters have been obtained indirectly. These parameters are related by a corresponding function and physical parameter directly. In this context, there is an error that affects the final measure. For this error is made a specific system of measurement to reliably. Using the equation (10), Taylor and Kuyatt(1994).

$$u_c^2(y) = \sum_{i=1}^{n}\left[\frac{\partial f}{\partial x_i}\right]^2 u^2(x_i) \tag{10}$$

Each has its measured parameter measurement error, some parameters such as flow rate, have more impact on the measurements j e f due to its variable is not a linear function. These parameters should be analyzed with more discretion.

The uncertainty analysis friction factor f is related to the area and geometrical parameters of air density shown in Equation (11). Other parameters involved are analyzed separately because they are indirect parameters.

ISSN 2176-5480

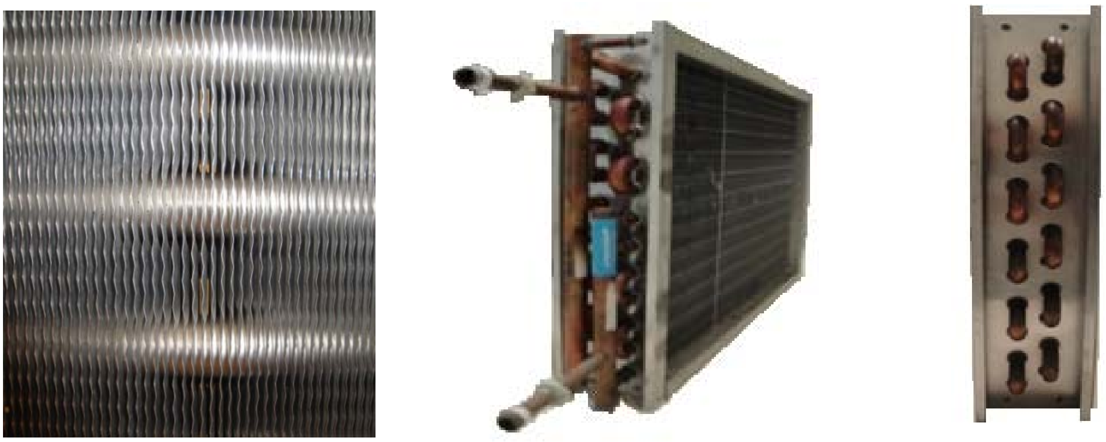
$$u_c(f) = \left[\begin{array}{l} \left(\frac{\partial f}{\partial A_l} \cdot u(A_l)\right)^2 + \left(\frac{\partial f}{\partial A_e} \cdot u(A_e)\right)^2 + \left(\frac{\partial f}{\partial \rho_e} \cdot u(\rho_e)\right)^2 + \left(\frac{\partial f}{\partial \rho_m} \cdot u(\rho_m)\right)^2 \\ + \left(\frac{\partial f}{\partial \rho_s} \cdot u(\rho_s)\right)^2 + \left(\frac{\partial f}{\partial \Delta P} \cdot u(\Delta P)\right)^2 + \left(\frac{\partial f}{\partial G_{máx}} \cdot u(G_{máx})\right)^2 + \left(\frac{\partial f}{\partial \sigma} \cdot u(\sigma)\right)^2 \end{array} \right]^{\frac{1}{2}} \tag{11}$$

4. RESULTS AND DISCUSSION

The objective of the experimental test is to evaluate the thermal-hydraulic performance of a finned coil. The results were processed using the software EES. For the first test was only considered the friction factor, and the experimental results confronted with the correlations available in the literature by Wang et al (1999) and Wang et al (2002).

Experimental tests were performed by varying the air velocity in front of coil 1.4 m / s to 3.0 m / s, and the dry bulb temperature of the air at entry ranged from 23 ° C to 30 ° C and realtiva humidity of 44% to 56%, since no air treatment. These three flow tests were used for the hot water, 0.8 m³ / s 1.05 m³ / s 1.5 m³ / three inlet temperatures of the water in the tubes of the serpentine, 35 ° C, 40 ° C and 45 ° C, totaling one hundred twenty six operating conditions, for each flow of hot water were tested three inlet temperatures for hot water in the coil and for each temperature were tested 12 speeds the air inlet.

The coil tested was manufactured by TROX in Brazil, built with copper pipes with a diameter of 15.80 mm (5/8 "), with a smooth inner surface, the type corrugated fins (herringbone wavy fin) and casting aluminum sheet . The coil has a face area of 940 mm by 610 mm with two rows of tubes in a staggered and triangular profile. as shown in Figure 3.



(a) (b) (c)

Figure 4 – Test of coils: (a) corruged fin. (b) face and side of coil. (c) left side

The geometrical parameters of the coil tested are presented in Table 2.

Table 2 - Geometric characteristics of the finned coil tested.

Da [al/m]	**Nt [-]**	**Nf [-]**	**De [mm]**	**Dc [mm]**	**Eea [mm]**	**Et [mm]**	**El [mm]**	**Ea [mm]**	**Po [mm]**	**θ [°]**	**Lt [mm]**	**Lh [mm]**	**Le [mm]**
370	12	2	15,8	16,4	2,5	37,5	33,0	0,3	3,0	14,0	940	610	125

In which D_a is density of fins, N_t is number of tubes, N_f is number of rows of tubes, D_e is external diameter, D_c is collar diameter, E_{ea} is distance between fins, E_t is transversal distance of fin, E_l is longitudinal distance of fin, E_a is fin thickness, P_o is wave height, θ is angle of wavy, L_t lenght of heat exchanger, L_h is height of heat exchanger, L_e is depth of heat exchanger.

The drop pressure is the parameter that was measured directly through the differential pressure transducers in the test section. The uncertainty of this parameter is associated with the uncertainty of the instrument, which is around 2%.

It is observed in Figure 5 that as the air velocity on the face of the coil increases, there is an increased pressure drop and it is also verified that variations in temperature and hot water flow does not affect the behavior of pressure drop.

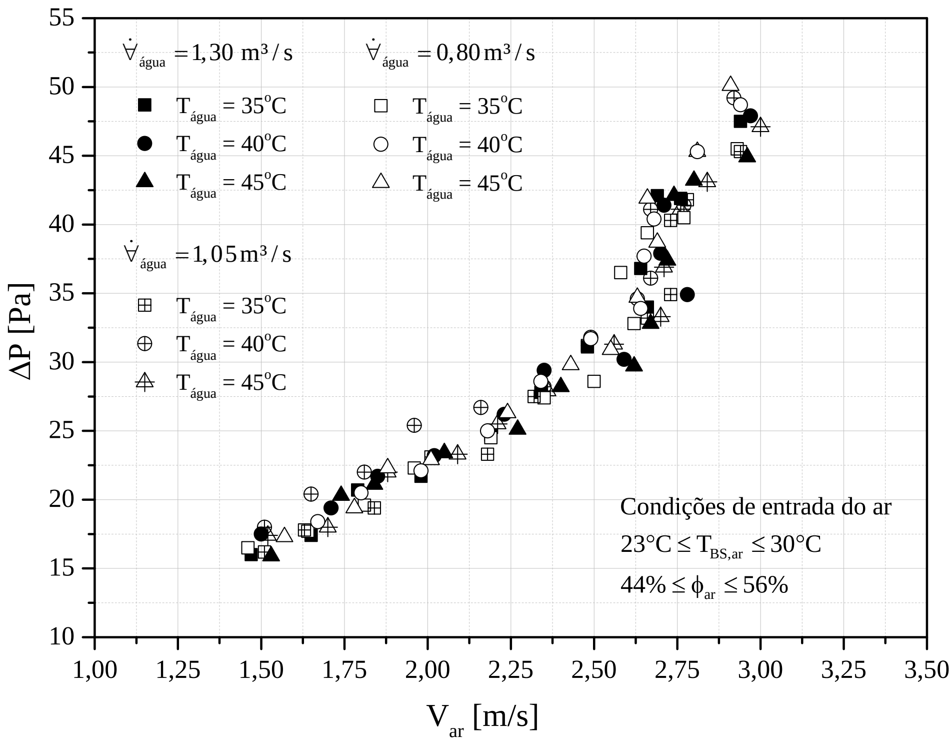


Figure 5 – Pressure drop for the air-side coil tested as a function of airspeed

ISSN 2176-5480

The friction factor (f) was determined according to the procedure presented in Chapter 4, which shows how to treat the results of this work. This factor is used to calculate the pressure drop, and thus define the energy needed to drive the hood or fan.

The results obtained by treatment results, equation (7) are shown in Figures 6 and 7. The results of the factor f are shown in relation to the Reynolds number which is a function of the diameter of the collar. The diameter of the collar has been chosen for its simplicity and because of this parameter as used in the literature for this type of coil.

Figure 6 shows the friction factor for obtained for three flow and temperature of the hot water that circulates through tubes that such variations have little interference in the value of friction factor. It was also observed that as the Reynolds number increases, there is a reduction in friction factor, and for Reynolds numbers of about 5000 there was a small increase in friction factor. Such behavior can be related to air flow through the channels of the fins and the formation of recirculation zones.

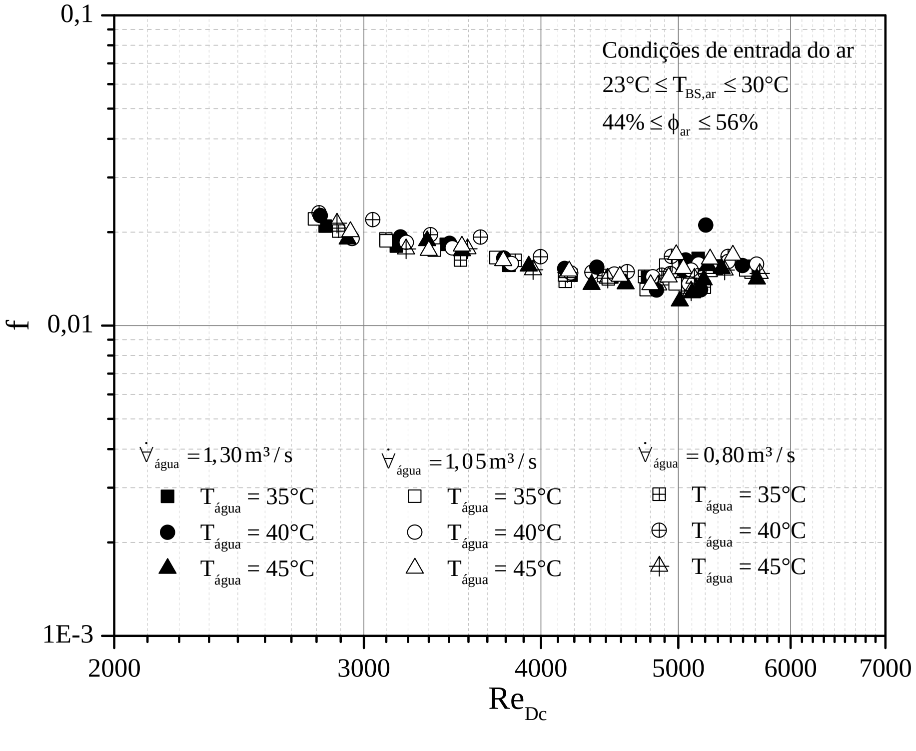


Figure 6 – F factor as compared with the different water flows

To validate the experimental setup, we compare the results obtained from literature correlations. It is noteworthy that in the literature there are many options for this type of correlation coil with corrugated surface and outer diameter of the tubes greater than 13.0 mm. The correlations that are best fit correlations proposed by Wang et al (1999) and Wang et al (2002), which were compared with the experimental friction factor, as can be seen in Figure 7.

The correlation of Wang et al (1999) as a function of the geometric parameters is given in Equation (12), together with their related factors.

ISSN 2176-5480

their related factors.

$$f = 0,05273\mathrm{Re}_{Dc}^{f1}\left(\frac{P_d}{P_o}\right)^{f2}\left(\frac{E_{ea}}{E_t}\right)^{f3}\left(\ln\left(\frac{A_e}{A_p}\right)\right)^{-2,726}\left(\frac{D_h}{D_c}\right)^{0,1325} N^{0,02305} \tag{12}$$

Where,

$$f1 = 0,1714 - 0,07372\cdot\left(\frac{E_{ea}}{E_l}\right)^{0,25}\ln\left(\frac{A_o}{A_t}\right)\left(\frac{P_d}{P_o}\right)^{-0,2} \tag{13}$$

$$f2 = 0,426\left(\frac{E_{ea}}{E_t}\right)^{0,3}\ln\left(\frac{A_o}{A_t}\right) \tag{14}$$

$$f3 = \frac{-10,2192}{\ln(\mathrm{Re}_{Dc})} \tag{15}$$

Similary correlation was used in Wang et al (2002) as a function of the geometrical parameters shown in Equation (16), together with their related factors.

$$f = 0,228\times\mathrm{Re}_{Dc}^{f1}\left(\tan\theta\right)^{f2}\left(\frac{E_{ea}}{E_l}\right)^{f3}\left(\frac{E_l}{D_C}\right)^{f4}\left(\frac{D_C}{D_h}\right)^{0,383}\left(\frac{E_l}{E_t}\right)^{-0,247} \tag{16}$$

Where,

$$f1 = -0,141\left(\frac{E_{ea}}{E_l}\right)^{0,0512}\left(\tan\theta\right)^{-0,472}\left(\frac{E_l}{E_t}\right)^{0,35}\left(\frac{E_l}{D_h}\right)^{0,449\cdot\tan\theta} N^{-0,049+0,23\tan\theta} \tag{17}$$

$$f2 = -0,562\left(\ln(\mathrm{Re}_{Dc})\right)^{-0,0923} N^{0,013} \tag{18}$$

$$f3 = 0,302\cdot\mathrm{Re}_{Dc}^{0,03}\left(\frac{E_t}{D_c}\right)^{0,026} \tag{19}$$

$$f4 = -0,306 + 3,63\tan\theta \tag{20}$$

It can be seen in Figure 7 the correlation proposed by Wang et al (1999) underestimates the friction factor correlation whereas proposed by Wang et al (2002) overestimates the friction factor. It is noteworthy that the correlation of Wang et al (1999) has been proposed for coils with a pitch between corrugations higher than the present study. The correlation of Wang et al (2002) already shows the equivalence of all parameters. Thus, according to Wang et al (2002) expects a dispersion of ± 15% correlations used in relation to the experimental results.

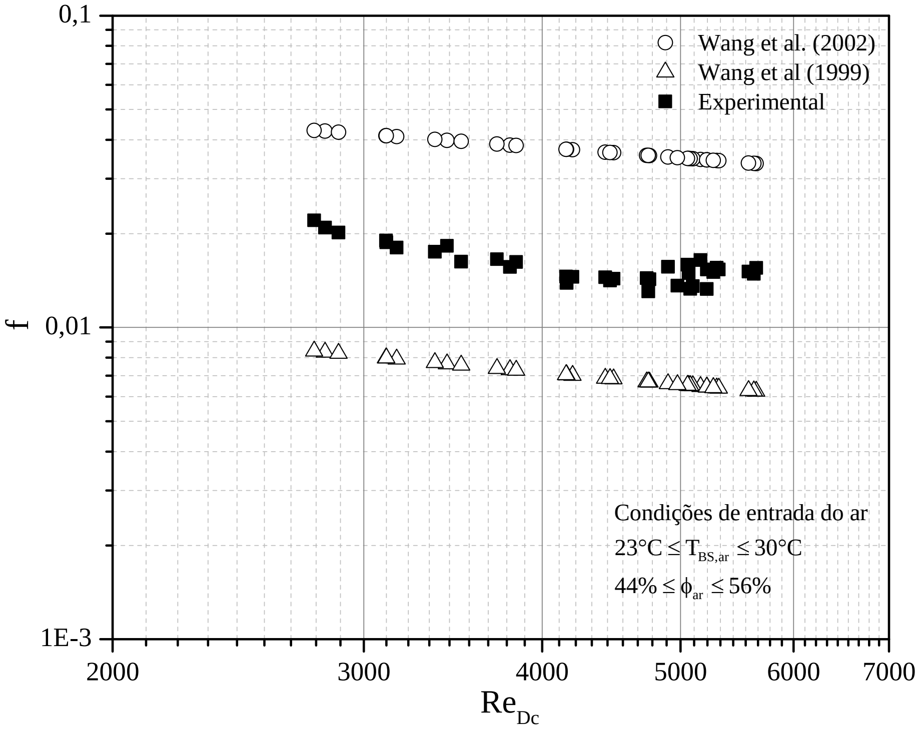


Figure 7 – Comparison between the experimental dates and friction factor correlation of Wang et al (1999) and Wang et al (2002) as a function of Reynolds number

The correlation of Wang et al (1999) presented a friction factor on average 55% lower than that obtained experimentally, while Wang et al (2002) presented a friction factor on the average 135% greater than the one obtained experimentally. Such behavior may be associated with the sensitivity of the correlations on the geometrical parameters of the heat exchanger. This is because for the coil geometrical parameters tested were obtained by direct measurement equipment, since the coil designs were not available. Thus, for certain parameters, such as thickness of the fin, fins spacing, angle and pitch of corrugation measurement is hampered thereby generating source of uncertainty.

Another aspect to be noted is related to the differential pressure transducer used to measure pressure drop, which has a measuring range from 0 to 5.0 kPa very large compared to the pressure drop expected for this type of coil that would be around 0 to 0.2 kPa. Thus, there is no need for replacement of such transducer in order to adapt it to the measurement range.

However, despite the discrepancy between the values for the friction coefficient experimentally obtained and those calculated by the correlations observed in Figure 7 similar behavior, that is, as the Reynolds number increases, there is a reduction in the friction factor.

Despite the uncertainty generated in obtaining direct some geometrical parameters, the present study evaluated the uncertainty associated with the friction factor, which was estimated as discussed in Section 3, using the calculation

ISSN 2176-5480

software EES (Engineering Equation Solver), whose method for calculating the uncertainty propagation follows the one proposed by Taylor and Kuyatt (1994).

Figure 8 shows the experimental friction factor as a function of Reynolds number incorporating uncertainty bar for friction factor. It is observed the propagation of uncertainty due to the uncertainties of the transducers has little effect on the friction factor, ranging from the minimum value up to ± 0.00114 ± 0.00193 maximum value, with the mean percentage ± 8.75%.

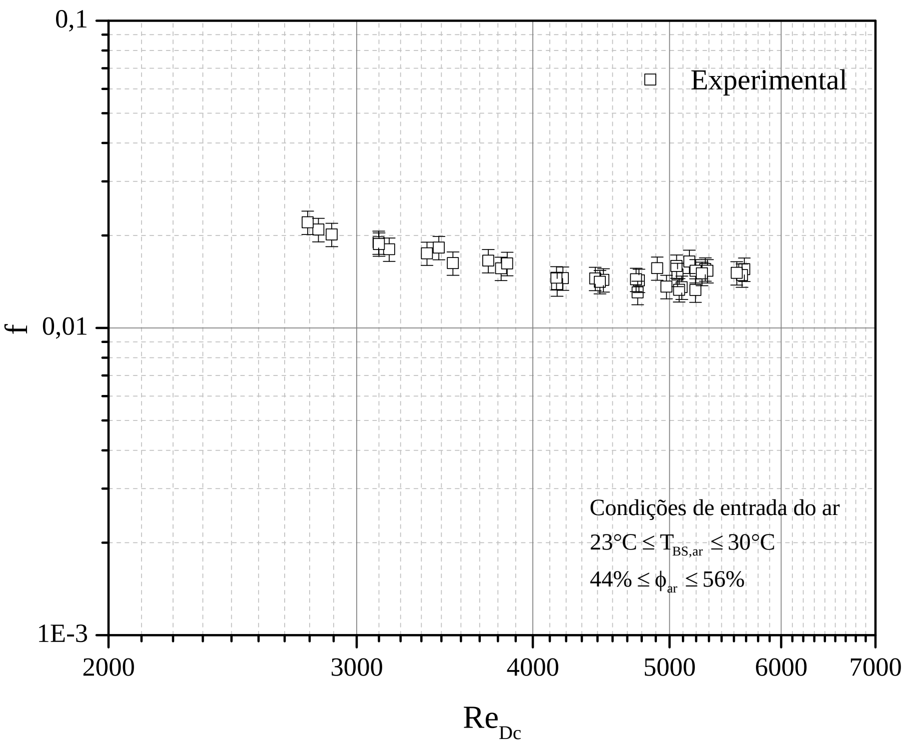


Figure 8 – experimental factor friction as a function of Reynolds number incorporating uncertainty bar

In the calculation it was observed that the most significant factor for the definition of uncertainty is the airspeed. This analysis becomes important to define which instrument is critical in test equipment, and that should receive greater attention and investment. The standard ashare 33 (2000) indicates two distinct ways to measure the air flow, which improves the reliability of the measurement of this parameter, and an instrument must be of the firstborn (nozzle flow), precisely because it is the parameter that most affects the measurement of this type of test.

In the case of correlations were noted that the geometric parameters are responsible for the increase in error. Some particularly affect significantly as spacing between fins and fin thickness. The angle of ripples is related to the corrugation pitch and amplitude ripple, these parameters are also difficult to measure and may influence the result of the friction factor calculated by the correlations. friction calculated by correlations.

5. CONCLUSION

This work covered the entire system for the calculation of thermo-hydraulic performance of a heat exchanger finned coil type, involving the construction and operation of an experimental apparatus for testing to obtain experimental data in order to allow the calculation of the factor friction factor f.

The present study also evaluated the literature correlations for calculating the friction factor f and the Colburn j factor for comparison with experimental results. Among the studied correlations showed that only postponed by Wang et al (1999) and Wang et al. al (2002) have proved suitable for the analysis of the results since it met the operating conditions and geometry of the coil tested. With the test for the friction factor performed, it was possible to reach the following conclusions:

- The treatment of the results was accomplished by experimental tests, thus characterizing a methodology to assess the thermo-hydraulic performance of corrugated coils.
- The pressure difference increases with increasing speed of the face.
- The initial temperature of the hot fluid and the flow of this fluid does not affect the friction factor.
- The correlations used in the comparison with experimental data showed no good agreement that such behavior may be related mainly to measure the pressure drop across the coil, since the transducer used has a very wide range of measurement in relation to fall expected pressure.
- Another factor related to the correlations for the friction factor is that even between the two correlations evaluated it was found inconsistent results, as compared to the experimental data correlation overestimated friction factor while the other underestimated.

The uncertainty in the friction factor with respect to the parameters measured this around ± 8.75%.

6. REFERENCES

[1] Zoghbi Filho, João. Theorical and Experimental evaluation of the Air-side thermo-hydraulic perfomance in heat exchangers air coils type. 2004. 320p. Doctoral thesis. Universidade de São Paulo, Escola de Engenharia de São Carlos, São Carlos.

[2] Chen, Yongpan. Error analysis of heat transfer for finned-tube heat-exchanger test-board. Control Systems for energy efficiency and comfort, Shenzen, Vol. V-7-3, November, 2006.

[3] ABNT, INMETRO (2003), Guide to the expression of uncertainty in mensurement, third edition in portuguese lenguage – Rio de Janeiro, Rio de Janeiro, 2003.

[4] ANSI/ASME PTC 19.1, (1985), Instruments and Apparatus, Part 1 – Measurement Uncertainty, American society of mechanical engineers, New York, NY.

ISSN 2176-5480

[5] ASHARE STANDARD 33, (2000), Method of Testing Forced Circulation Air Cooling and Air Heating Coils, American society of Heating, Refrigeration and Air condioting Engineers, Atlanta, GA.

[6] Dong, Junq, Heat transfer and pressure drop correlations for the multi-louvered fin compact heat exchangers. Energy conversion and Management, Shanghai, nº 8, pg 1506 - 1515, january, 2007

[7] Barbosa, Jader, A study of the air-side heat transfer and pressure drop characteristics of tube-fin 'no-frost' evaporators. Applied Energy, Florianópolis, nº 86, pg 1484 - 1491, 2009.

[8] Wang, C. C., Heat transfer and Friction Characteristics of Typical Wavy Fin-and-Tube Heat Exchangers. Experimental Thermal and Fluid Science, New York, nº 14, pg 174 - 186, 1997

[9] Wang, C. C., Data reduction for air side performance of finned-and-tube heat exchangers. Experimental Thermal and Fluid Science, New York, nº 21, pg 218 - 2226, 2000.

[10] Wang, C. C., Empirical correlations for heat transfer and flow friction characteristics of herringbone wavy fin-and-tube heat exchangers. International Journal of Refrigeration, Taoyuan, nº 25, pg 673 - 680, 2002

[11] Kays, W. M., London, A. L., Compact Heat Exchagers. New York : McGraw-Hill,3ªed.,1998., 335p.

[12] Taylor B.N. and Kuyatt, C.E., Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results, National Institute of Standards and Technology Technical Note 1297, 1994

7. RESPONSIBILITY NOTICE

The author(s) is (are) the only responsible for the printed material included in this paper.