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THEORETICAL-EXPERIMENTAL ANALYSIS OF A FINNED TUBE HEAT EXCHANGER COMPOSED OF WAVY FINS USED IN THERMAL SYSTEMS AS CONDENSER

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Abstract. *This work consisted in submitting a finned tube heat exchanger composed of wavy fins to nine test conditions in an experimental apparatus in order to determine which condition this equipment can operate with greater thermal hydraulic performance. Thus, in each test the Colburn j and Fanning friction (f) factors calculated experimentally were compared with two empirical correlations, where it was observed that these correlations did not represented with good accuracy both factors at the same time, but only one of them, where the Wang et al., 1997 correlation represented the experimental data with a maximum deviation of 17.97% for the Colburn j factor and the Kim et al., 1997 correlation represented the experimental data with a maximum deviation of 29.30% for the f factor. The thermal hydraulic performance of heat exchanger tested was evaluated using the j/f parameter, where it obtained higher performance in the 3 test condition. The mass flow rate of air passing through the device has been calculated using the nozzle plate method and the energy balance method held in the exchanger and it was observed a good approximation between the methods, where the maximum deviation was 30% and the minimum of 5%. Thus, this work contributed to projects of finned tube heat exchangers that reduce energy consumption in thermal systems.*

Keywords: *Heat Exchangers, Finned Coil, Thermal Hydraulic Performance.*

1. INTRODUCTION

In any society, the economic activity, in particular industrial, needs energy to its full operation. In the 1970's, due to the oil crisis, a framework of energy use rationalization was originated, especially in those countries whose economy had a greater dependence on oil imports, such as Brazil, which only in the 1980's began to arise attention in relation to energy waste. Thus, from this decade, the measures for energy use rationalization were maintained, fruit of strategic and economical maturing, and an awareness of the environmental problems caused by the production, transport, conversion and energy use.

In Brazil, the high consumption of electrical energy in thermal systems is due to thermodynamic irreversibility inherent in the heat transfer processes and the refrigerant flow inside the refrigeration system components, especially the heat exchangers. One of the main types of heat exchangers is the finned tube heat exchanger, which is widely used in the fridge sector for applications of heating and cooling the air. The increase in the heat transfer from the air side, without increasing significantly the pressure loss, is great interest to improve the thermal hydraulic performance of heat exchangers and thus reduce the energy consumption of refrigeration systems.

In this way, this work consisted in submitting a finned tube heat exchanger composed of wavy fins to nine test conditions in an experimental apparatus in order to determine which condition this equipment can operate with greater (j/f) thermal hydraulic performance.

2. CORRELATIONS TO CALCULATE THE COLBURN J AND FANNING FRICTION FACTORS

The heat transfer in problems involving finned tube heat exchanger is adequately correlated by Eq.1, known as Colburn j factor.

$$j = St \cdot Pr^{2/3} = \frac{h}{G \cdot cp} \cdot Pr^{2/3} \tag{1}$$

Where **h** [W/m^2] is convection heat transfer coefficient in the air side, **Cp** is specific heat at constant pressure in air side. The variable **G** is the mass speed and is calculated by Eq. 2.

$$G = \dot{m}/A_c \tag{2}$$

The variable $\mathbf{A_c}$ [m^2] is free flow area of heat exchanger in study and $\dot{m}$ [kg/s] is mass flow rate. The variable **Pr** is Prandtl number and is calculated by Eq. 3. The variable **μ** [kg/s-m] is viscosity and **k** [W/m-K] thermal conductivity.

$$Pr = \mu \cdot c_p / k \tag{3}$$

The Fanning Friction Factor, **f**, is calculated by Eq.4. The variables v_e, v_s and v_m are respectively specific volume in the input, output and medium. The variable $\sigma = A_c / A_f$. The variables A_{total} and A_f are respectively total and frontal areas. Finally, ΔP is total pressure loss of air when it passes through the heat exchanger.

$$\Delta P = \frac{G^2 v_e}{2} \left[\left(1 + \sigma^2\right) \left(\frac{v_s}{v_e} - 1 \right) + f \frac{A_{total}}{A_c} \frac{v_m}{v_e} \right] \tag{4}$$

3. ANALYSIS OF HEAT TRANSFER: WAVY FIN SURFACE

Among the heat transfer intensification geometries, one of the most used is wavy fin (known as corrugated).

Wang et al., 1997 stated that heat transfer coefficients for the wavy fin showed an increase from 55 % to 70 % compared to plane fin. However, the friction factor is even greater, ranging from 66 % to 140 %.

Wang et al., 1999-e stated that heat transfer characteristics are strongly related to the corrugation angle (θ), with the waffle height (P_d) and the wavy length (P_o).

Wang et al., 2011 observed on their various investigations that the herringbone type wavy fin when compared with plane fin produces a significant increase of pressure loss for fin spacing lower than 1.6 mm. They also observed an increase in heat transfer of 5% to 10% for a corrugation angle lower than 20°.

Elshafei et al., 2010 investigated experimentally the heat transfer and pressure loss characteristics in wavy channels of parallel plates and made the following statements: The heat transfer and pressure loss average coefficients increased respectively by a factor of 2.6 to 3.2 and 1.9 to 2.6 for wavy channels of parallel plates. At the same time, the authors observed that the friction factor increases with increased channels spacing.

Kim and Youn (2013) stated that a heat exchanger compound of a wavy fin with waffle height of 2.79 mm has a heat transfer coefficient 4% higher and a pressure loss 10 % higher compared to a heat exchanger compound of a wavy fin with waffle height of 2.39 mm.

Wang et al., 1997 made the following statements to a staggered tube arrangement: For $Re_{Dc} > 900$ and $N_f > 1$, the Colburn j factor increases slightly with decreasing of number of tube rows and the friction factor practically does not vary with the number of tube rows. For $Re_{Dc} < 900$, they observed that both the friction factor is lower as the Colburn j factor is higher for the heat exchanger with the lowest number of tube rows.

4. EXPERIMENTAL PROCEDURE

The test took place according to the experimental procedures suggested by ASHRAE 41.2 and ASHRAE 33 standards, where were collected the data to calculate of the friction factor and Colburn j factor. The experimental apparatus consists of a data acquisition system, pressure transducers, thermocouples, flow meter, a sensor that measures the relative humidity, an air circuit (cross-section area of wind tunnel is 0.5734 m^2) and a water circuit.

The tested heat exchanger has copper tubes with smooth inner surface, aluminum fins of type herringbone wavy and its main characteristics are show in Table 1.

The heat exchanger in study was subjected to nine test conditions, in order to know the thermal and hydraulic behavior, to observe the degree of approximation of the experimental data obtained with some empirical correlations and finally determining in which of these nine test conditions the exchanger operated with greater heat exchange producing the lowest pressure loss possible, i.e., higher thermal hydraulic performance (j/f).

Thus, for each of these nine tested conditions was established a certain value in revolutions per minute (rpm) for the fan and the particular value in hertz (hz) for the pump, as can be seen in Table 2.

Table 1 - Geometric characteristics of the heat exchanger under study.

D_a [al/m]	N_t	N_f	D_o [mm]	D_c [mm]	E_{ea} [mm]	E_t [mm]	E_l [mm]	E_a [mm]	P_o [mm]	θ [°]
370	12	2	15,87	16,48	2,5	37,5	33,0	0,3	3,0	14,0

Nomenclature: E_{ea} (Fin spacing), D_o (tube outer diameter), D_c (collar diameter), E_a (fin thickness), θ (Corrugation angle), P_o (wavy length), D_a (fin density), E_t (transverse tube pitch), E_l (longitudinal tube pitch), N_t (Number of tubes), N_f (Number of longitudinal tube rows)

Table 2 - Operational condition of experimental tests.

Test condition	*Fan*		*Pump*	
	Frequency (rpm)	*Percentage of capacity*	*Frequency (hz)*	*Percentage of capacity*
1	2050	100%	60	100%
2	2050	100%	45,3	75%
3	2050	100%	30,2	50%
4	1536	75%	60	100%
5	1536	75%	45,3	75%
6	1536	75%	30,2	50%
7	1030	50%	60	100%
8	1030	50%	45,3	75%
9	1030	50%	30,2	50%

In all these experiments the wind tunnel worked with the nozzle plate operating with only an open nozzle throat whose diameter is 80 mm. The following parameters were measured: volumetric water flow through the pump, relative humidity of the air, air pressure loss when passing through the exchanger and from the nozzles plate, temperature of the water that enters and exits the exchanger and the temperature of air in and out of the exchanger.

The mass flow rate of the air was determined by method of energy balance and the nozzle plate, according to ASHRAE 33 standard. The methodology for calculating the expanded uncertainty is described in the (C.H. Paula, 2016) dissertation.

Finally, the thermal hydraulic performance of the exchanger was calculated through a program made in Engineering Equation Solve software.

5. RESULTS AND DISCUSSION

The experimental data concerning the parameters mentioned above were collected and treated and the values for the expanded uncertainty for each were determined. Subsequently, both (j_{author}) Colburn j factor and (f_{author}) Fanning friction factor were determined for the heat exchanger in study.

The Colburn j and Fanning friction factors were also calculated for the same exchanger, but this time using three distinct empirical correlations, as shown in Table 3. These correlations were Wang et al., 1997, Wang et al., 2002 and Kim et al., 1997, whose main characteristics are illustrated in Table 4.

The thermal hydraulic performance was calculated for each test condition which the heat exchanger in study was submitted and the result found is illustrated in Figure 1.

According to this figure, the 3 test condition obtained the largest thermal hydraulic performance between the tested and the 7 test condition obtained worst performance.

Table 3 – Colburn j factors and Fanning friction calculated for the tested conditions.

Test	j_{author}	j_{Wang} (1997)	j_{Wang} (2002)	j_{KIM} (1997)	f_{author}	f_{Wang} (1997)	f_{Wang} (2002)	f_{KIM} (1997)
1	0.0073 ±0.0004	0.0082 ±0.0011	0.0028 ±0.0001	-	0.40±0.23	0.11±0.01	0.17 ±0.06	-
2	0.0072 ±0.0004	0.0076 ±0.0007	0.0027 ±0.0001	-	0.32±0.13	0.11±0.01	0.14 ±0.03	-
3	0.0070 ±0.0005	0.0074 ±0.0006	0.0027 ±0.0001	-	0.29±0.10	0.10 ±0.01	0.13 ±0.03	-
4	0.0086 ±0.0007	0.0095 ±0.0024	0.0029 ±0.0002	-	0.66±0.65	0.13±0.03	0.25 ±0.20	-
5	0.0085 ±0.0007	0.0091 ±0.0015	0.0029 ±0.0002	-	0.56±0.38	0.13±0.02	0.23 ±0.10	-
6	0.0081 ±0.0006	0.0085 ±0.0009	0.0028 ±0.0001	-	0.43±0.18	0.12±0.01	0.19 ±0.05	-
7	0.0110 ±0.0008	0.0134 ±0.0049	-	0.65± 0.14	2.04 ±0.27	0.18±0.06	-	1.44±0.38
8	0.0105 ±0.0008	0.0123 ±0.0030	-	0.62± 0.09	1.5 ±0.13	0.17 ±0.04	-	1.36 ±0.24
9	0.0103 ±0.0008	0.0109 ±0.0013	-	0.57± 0.04	0.97 ±0.42	0.15 ±0.02	-	1.24 ±0.11

Table 4 – Colburn j factors and Fanning friction calculated for the tested conditions.

Correlation	Tested Heat Exchanger	D_c (mm)	N_f	E_{ea} (mm)	E_l (mm)	E_t (mm)	P_o (mm)	θ (graus)
Kim et al., 1997	32	9.53 a 12.7	1 a 4	1.57 a 4.09	19.05; 22; 27.5	25.4; 31.8	3.67 a 5.5	5.7° a 34.7°
Wang et al., 1997	18	10.3	1 a 4	1.69 a 4.8	19.05; 29.4	25.4 a 29.4	4.76; 9.525	15.22° a 17.48°
Wang et al., 2002	61	7.66 a 16.85	1 a 6	1.21 a 6.43	12.7 a 33	21 a 38.1	3.175 a 8.25	5.3° a 18.5°

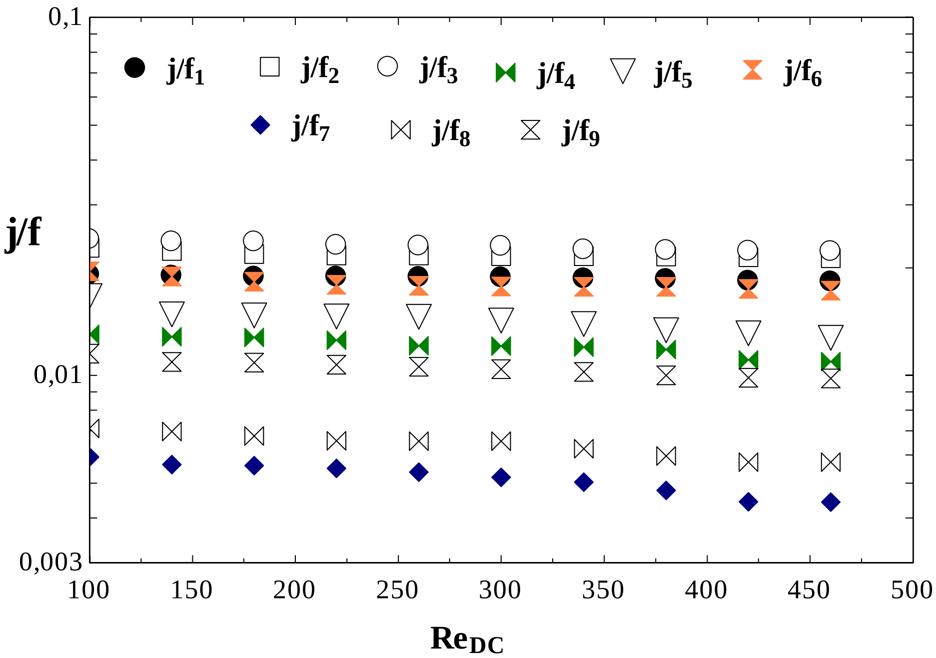

Figure 1. Result of the thermal hydraulic performance for the nine tested conditions.

The comparison between the values provided by the correlations in relation to the values calculated by the author for the 3 test condition can also be observed in Figure 2.

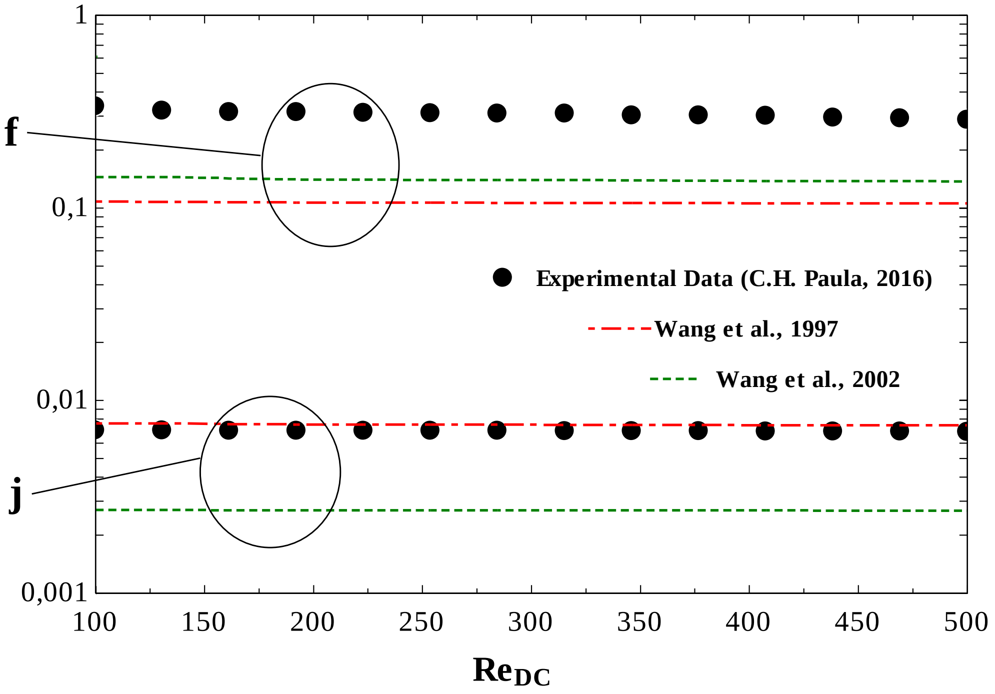

Figure 2. Comparison of Wang et al., 1997 and Wang et al., 2002 with experimental results for 3 test condition.

For the test condition illustrated above was observed that the Wang et al., 1997 correlation agreed very well with the experimental results for the j factor compared with Wang et al., 2002 correlation, but both correlations didn't approximate very well the f factor. This contrast is due to the differences in the geometrical aspects of the heat exchangers.

The mass flow rate of air passing through the device was calculated using the nozzle plate method and the energy balance method held in the exchanger and it was observed a good approximation between the methods, where the minimum and maximum deviations were 5 % and 30 % respectively, as can be seen in the Figures 3 and 4.

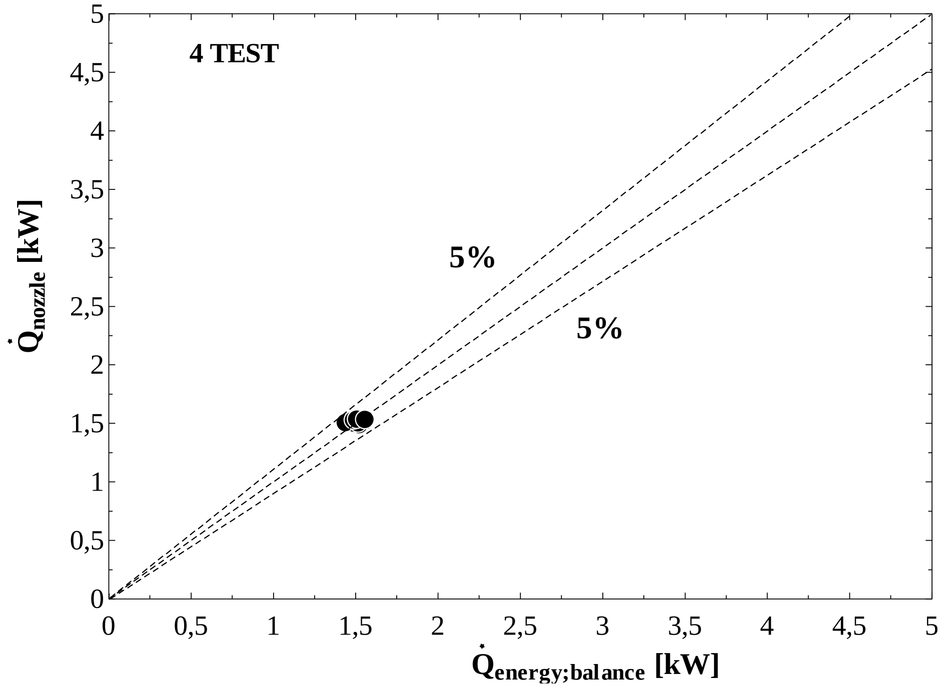

Figure 3. Comparison between the results of the heat transfer rate obtained by the two methods, for 4 test condition.

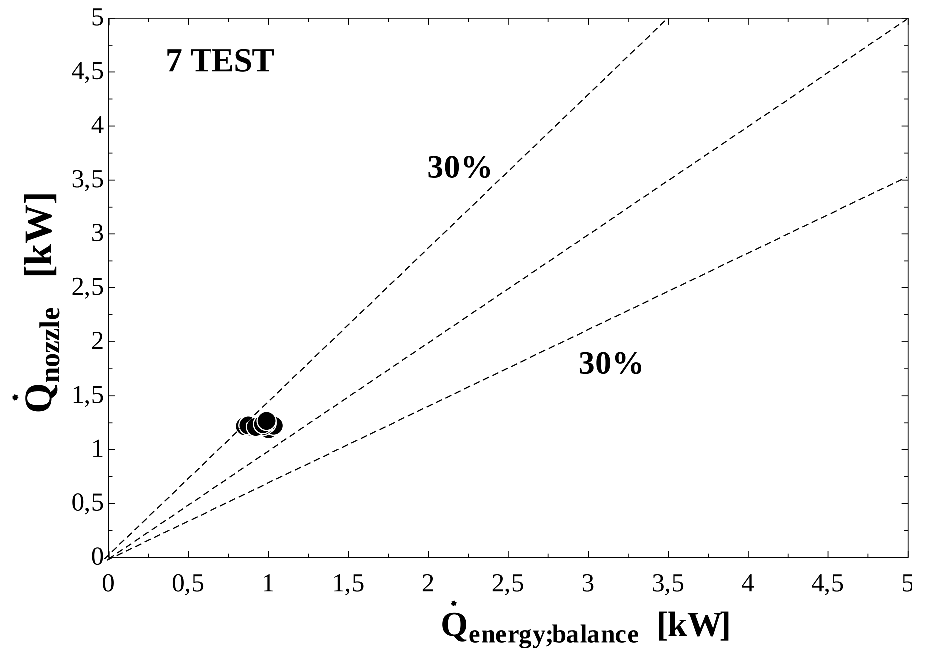

Figure 4. Comparison between the results of the heat transfer rate obtained by the two methods, for 7 test condition.

Table 5. Correlations for wavy fin used in this article.

Wavy Fin: Wang et al., 1997	
$j = \dfrac{1.201}{\left[\ln\left(Re_{Dc}^{\sigma}\right)\right]^{2.921}}$ $\sigma = A_c / A_f$	$f = \dfrac{16.67}{\left[\ln\left(Re_{Dc}\right)\right]^{2.64}} F^{-0.096} N_f^{0.098}$ $F = \left(A_{total} / A_{pri}\right)$
Wavy Fin: Kim et al., 1997	
For $N_f \geq 3$: $j_3 = 0.394 \cdot Re_{Dc}^{-0.357} B^{-0.272} T^{-0.205} X^{-0.558} Y^{-0.133}$	$f_a = 4.467 \cdot Re_{Dc}^{-0.423} B^{-1.08} T^{-1.08} X^{-0.672} \cdot \left(A_{sec} / A_{total}\right)$ **Where**: $B = (E_t / E_l)$, $T = (E_{ea} / D_c)$, $X = (P_o / P_d)$, $Y = (P_d / E_{ea})$
Wavy Fin: Wang et al., 2002	
$f = 0.228 \times Re_{Dc}^{f1} (tg\theta)^{f2} \left(\dfrac{E_{ea}}{E_l}\right)^{f3} \left(\dfrac{E_l}{D_C}\right)^{f4} \left(\dfrac{D_C}{D_h}\right)^{0.383} \left(\dfrac{E_l}{E_t}\right)^{-0.247}$ **Where**: $f1 = -0.141 \left(\dfrac{E_{ea}}{E_l}\right)^{0.0512} (\tan\theta)^{-0.472} \left(\dfrac{E_l}{E_t}\right)^{0.35} \left(\dfrac{E_l}{D_h}\right)^{0.449 \cdot \tan\theta} N_f^{-0.049 + 0.23 \tan\theta}$ $f2 = -0.562 \left(\ln(Re_{Dc})\right)^{-0.0923} N_f^{0.013}$, $f3 = 0.302 \cdot Re_{Dc}^{0.03} \left(\dfrac{E_t}{D_c}\right)^{0.026}$ $f4 = -0.306 + 3.63 \tan\theta$	$j = 0.0646 \cdot Re_{Dc}^{j_1} \cdot \left(\frac{D_c}{D_h}\right)^{j_2} \left(\frac{E_{ea}}{E_t}\right)^{-1.03} \left(\frac{E_l}{D_c}\right)^{0.432} (\tan\theta)^{-0.692} N_f^{-0.737}$ $j_1 = -0.0545 - 0.0538 \tan\theta - 0.302 \cdot N_f^{-0.24} \cdot \left(\frac{E_{ea}}{E_l}\right)^{-1.3} \left(\frac{E_l}{E_t}\right)^{0.379} \left(\frac{E_l}{D_h}\right)^{-1.35} \tan\theta^{-0.256}$ $j_2 = -1.29 \left(\frac{E_l}{E_t}\right)^{1.77 - 9.43 \tan\theta} \left(\frac{D_c}{D_h}\right)^{0.229 - 1.43 \tan\theta} \cdot N_f^{-0.166 - 1.08 \tan\theta} \left(\frac{E_{ea}}{E_t}\right)^{-0.174 \ln(0.5 N_f)}$

6. CONCLUSIONS

After analyzing the results obtained for the nine tested conditions it was observed that the 3 test condition (higher fan frequency and lower pump frequency) has a higher thermal hydraulic performance. It was also observed that for the tested conditions the Wang et al., 1997 correlation represented with a good degree of approximation the Colburn j factor, with a maximum percentage deviation of 17.97 %. For the Kim et al., 1997 correlation represented with a moderate degree of approximation the Fanning friction factor (f), with a maximum percentage deviation of 29.30 %.

The method of calculating the mass flow of air using the nozzle plate in relation to the energy balance method performed on the heat exchanger had a good agreement with each other, because for most of the tests the zero deviation lines were below 20 %.

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