18th Brazilian Congress of Thermal Sciences and Engineering
November 16-20, 2020 (Online)

ENC-2020-0088

NUSSELT NUMBER EXPERIMENTAL CORRELATION FOR FORCED CONVECTION IN FINNED ASTM TUBES

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Abstract. *The forced convection heat transfer on extended surfaces is largely applied in heat exchangers. These exchange surfaces have geometries dimensioned to promote controlled heat exchange between two or more fluids. Tubes with finned surfaces are recommended in heat exchangers that involve gaseous external flows. To design the surfaces of thermal exchanges, correlations are used to obtain the Nusselt number and thus calculate the coefficient of convective heat transfer. Classic correlations found in the heat transfer textbooks cover only plain tubes and simple surfaces, while the heat exchangers literature presents experimental values available to a very limited and standardized class of diameters, fins and pitch of tubes. The objective of this work is to experimentally determine the forced convection heat transfer for finned tube samples in different situations and, based on these results, to propose a specific experimental correlation for the air flow variation applicable in this type of finned tube. Samples of finned tubes are tested in a wind tunnel available. The correlation obtained from the experimental results provides consistent values. In the particular case of plain tubes, results from the correlation are significantly in accordance with the results obtained from the classical Zukauskas correlation, as shown by Pearson's chi-square test.*

Keywords: *Forced convection, heat exchanger, finned pipes, ASTM tube, experimental correlation, wind tunnel.*

1. INTRODUCTION

Heat transfer by forced convection in finned tubes is a very active research field with direct applications in the industrial sector. Research in this field involves numerical studies (Morales and Loredo, 2018; Petracci et al., 2016) and experimental ones, such as the studies conducted by Martinez et al. (2010) and Anoop et al. (2015).

This research results from the lack of information in the literature on the convection heat transfer coefficients of a specific type of extended surface, here called non-standard finned tube. The experimental correlation for forced convection in finned tubes provided by London and Kays (1964) has been used for over half a century for standardized finned tubes; such correlations do not apply to the ASTM standard tubes currently available in several countries, including Brazil, which causes difficulties in the design of heat exchangers involving gas flow.

The experimental approach is important to treat heat transfer problems, especially for complex geometries, such as finned tubes even with advances in computational and numerical methods. Anoop et al (2015) mentions that the accuracy of a numerical model depends on several aspects, for example, the specification of the contour conditions, the quality of the mesh used to delimit the computational domain, the turbulence model used, among others. Thus, many times, the effort to design and execute experiments is smaller than the effort to implement a precise computational model, as well as the computational time required to solve the problem.

The objective of this work is to experimentally determine the forced convection heat transfer for finned tube samples in different situations and, based on these results, to propose a specific experimental correlation for the airflow around this type of finned tube.

2. METHODOLOGY

2.1 Experiment description

The tests were performed in the Fluid Laboratory of the Department of Chemistry and Energy, in the School of Engineering of the Universidade Estadual Paulista, Guaratinguetá (SP). Figure 1 shows the open circuit wind tunnel (blower type) consisting of an electric motor (1.47kW, 60Hz, 1730 RPM) with a frequency inverter coupled to a fan. The incident flow is stabilized by a honeycomb, enters the test section and is discharged into the atmosphere. The test section is 200 mm wide by 200 mm high, resulting in a hydraulic diameter D_h = 200 mm. The length of the test section is Cs = 980 mm. Martinez et al. (2010) in their study considers that if $C_s > 1.5D_h$, the measurements are not affected by the boundary layers formed on the walls of the test section.

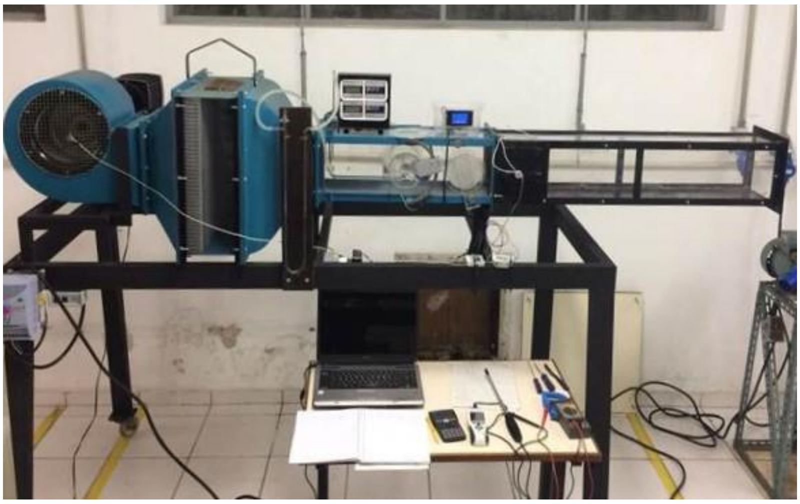


Figure 1. Open circuit wind tunnel

For simulating the heat exchange conditions in the laboratory, samples of finned tubes were designed and building from a smooth tube to which the fins are added (Tab. 1; Fig. 2). The material used is ASTM A-53 Gr.B carbon steel. In each sample was inserted a 180 W cartridge-type electrical resistance (220 V/60Hz; Fig. 3).

Table 1. Samples dimensions.

Sample	Pitch(mm)	tube surface area (m²)
1	Plain tube	0,0134
2	15	0,0651
3	10	0,0914
4	5	0,1700

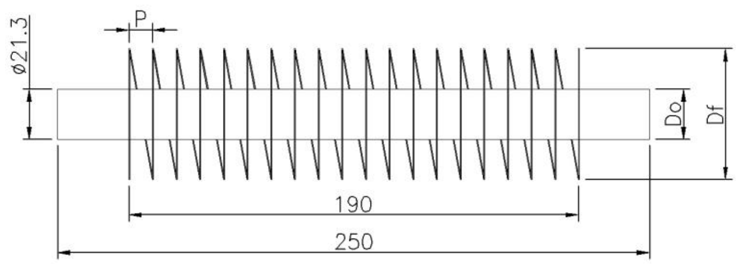


Figura 2. Samples design.

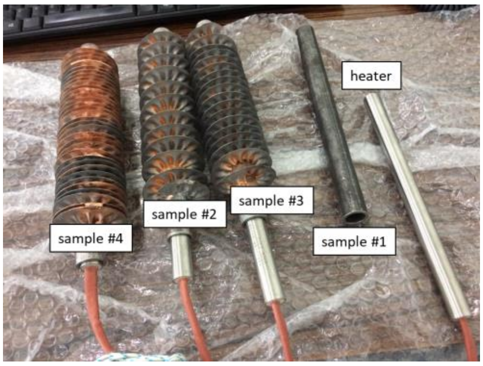


Figure 3. Samples with heaters inside.

Any apparatus close to the finned tube can cause disturbances in the flow that interfere with the measurement values, as described by Chimres et. al (2018). Here, type J thermocouples are used to measure the air temperature upstream of the test section and the temperature on the surface of the finned tube (Fig. 4). The thermocouple is positioned in the drag region (or tail region) to minimize disturbances in the flow provided by the sensor. The measurements are made through a universal module DigiRail-2A NOVUS through a USB-i485 converter for with RS485 communication system. To measure the dynamic pressure of the upstream flow, a pressure tap installed on the duct wall and a probe with its axis perpendicular to the surface are used. The flow velocity is related to the dynamic pressure as Eq. (1), which is the difference between the stagnation pressure and the static pressure. This difference is read on the inclined pressure gauge, with a Δh scale in millimeter already corrected for the inclination, using water as the gauge fluid. The power dissipated by the electrical resistance during the heating process of the finned tube is measured by a wattmeter. Table 2 describes the quantities measures and the respective instruments.

$$u_\infty = \sqrt{2g\rho_{H_2O}\Delta h/\rho_{air}} \tag{1}$$

Table 2. Quantities measured and respective instruments.

Quantity	Instrument	Uncertainty
Flow temperature	Type J, class 1 thermocouple	±1.5 °C*
Surface temperature	Type J, class 1 thermocouple	±1.5 °C*
Dynamic pressure	Pitot and static ports	±0.5 mm**
Energy dissipated	Wattmeter	±0.5 %**

* According to Brazilian Standard ABNT NBR 12771

** According to the respective manufactures

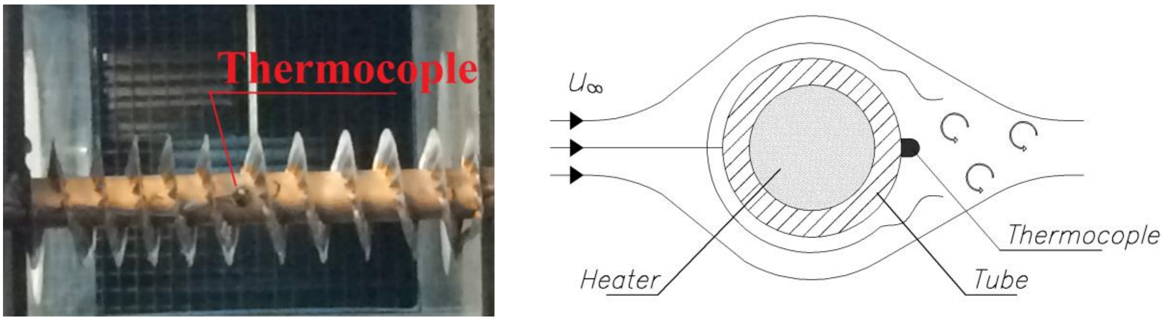


Figure 4. Left: Image of the finned tube with the thermocouple positioned on the surface. Right: Schematic representing the thermocouple in the wake region of the flow.

2.2. Mathematical formulation for development of a characteristic correlation

For typical radial diameter geometries the experimental correlations have the form of Eq. (2), similar to the Hilpert (1933) correlation for perpendicular external flows on isothermal cylinders. In this study, Equation (2) is adapted to take into account different combinations of D_f fin diameter and p fin pitch (Fig. 2; Tab. 1). Equation (3) is adapted to the airflow so that the Prandtl number is implicitly considered in the constant $B = CPr^n$. Equation (3) is then the resulting adaptation of Eq. (2).

$$\overline{Nu}_D = CRe_D^m Pr^n \tag{2}$$

$$\overline{Nu}_D = BRe_D^m \left(\frac{D_f}{p}\right)^s \tag{3}$$

The Nusselt number and the Reynolds number are obtained through several experiments with the four available samples tubes and from there the coefficients *B*, *m* and *s* in Eq. (3) are determined by the method of Stoecker (1989). For the steady state and neglecting the heat transfer by radiation, the convective coefficient is calculated using Eq. (4), derived from the energy balance of the sample, that is, the heat generated in the electric heater is dissipated by convection. Thus, Eq. (4) results in an average convective coefficient.

$$\bar{h} = \frac{q_e}{A_s(T_s - T_\infty)} \tag{4}$$

The convection heat transfer rate is described using Eq. (4), with $\bar{h}$ the average convection heat transfer coeficient, T_∞ the air flow temperature, T_s the surface temperature, A_s the surface area of the finned tube heat exchange and q_e is the electrical power dissipated by the electrical resistance. All these quantities are measured, $\bar{h}$ is calculated and the average Nusselt number can be determined using Eq. (5), with the thermal conductivity of air k at the film temperature. In finned tubes, the characteristic length can be the tube diameter, the fin diameter or the fin pitch, according to Morales and Loredo (2018). Here, the tube diameter is chosen. The values obtained for Nusselt number are then compared to other correlations available in the literature.

$$\overline{Nu}_D = \frac{\bar{h}D}{k} \tag{5}$$

2.3. Uncertainty analysis

The uncertainty analysis was obtained through the method of Taylor and Kuyatt (1994), employed by Ozturk et al. (2019) and Annop et al. (2015) in the performance analysis of fins of heat exchanger similar to the proposed work. Thus, the uncertainty in the estimation of the convective coefficient Eq. (6) and Nusselt number Eq. (7) for the experiments is given by:

$$u_{\bar{h}} = \sqrt{\sum_{i=1}^{3}\left(\frac{\partial \bar{h}}{\partial x_i}u_{x_i}\right)^2};\ x_1 = q_e;\ x_2 = A_s;\ x_3 = T_s - T_\infty \tag{6}$$

$$u_{\overline{Nu}_D} = \sqrt{\sum_{i=4}^{5}\left(\frac{\partial \overline{Nu}_D}{\partial x_i}u_{x_i}\right)^2};\ x_4 = \bar{h}\ ;\ x_5 = D \tag{7}$$

An average deviation was obtained for the convective coefficient of ± 3,41% and for Nusselt number of ± 2.76%. This uncertainty obtained in the Nusselt number is within the range of uncertainties calculated by Morales and Loredo (2018) who obtained values between 2.6% and 2.9% for the same range of Reynolds.

3. RESULTS AND DISCUSSION

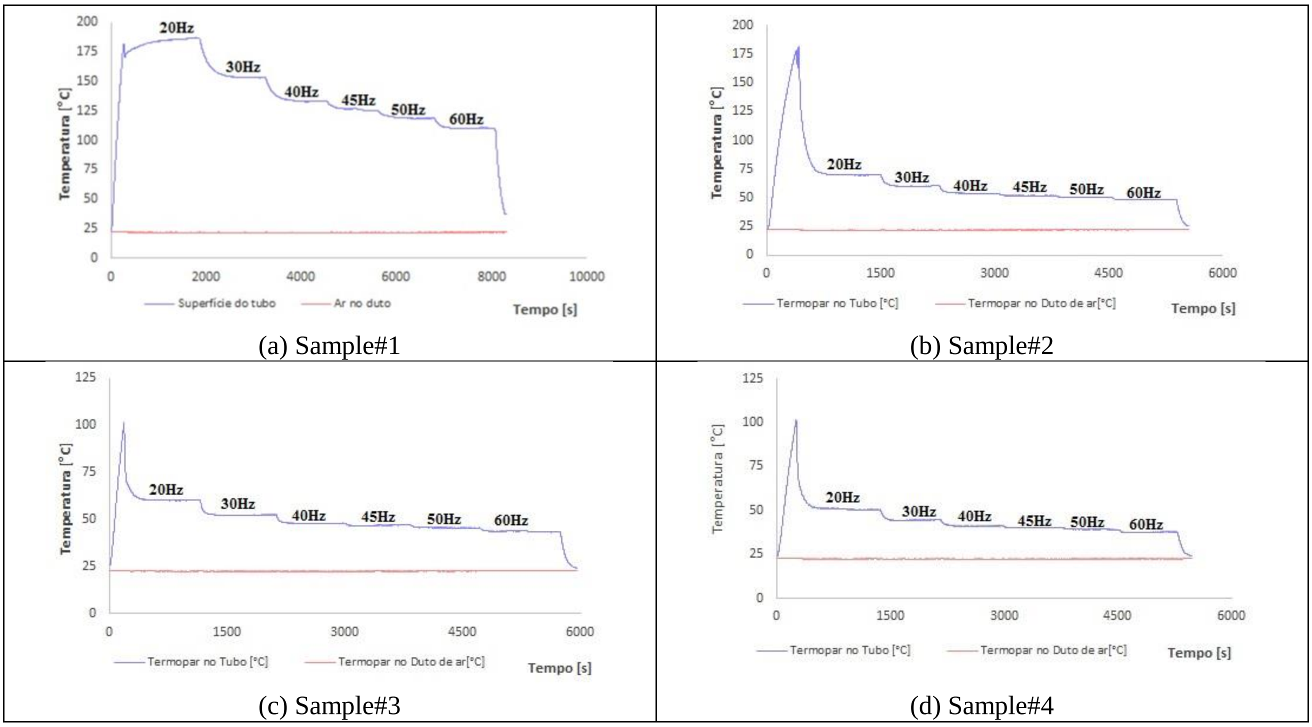
3.1 Messured quantities from tests

Each sample tube was tested for six different flow rates corresponding to different speeds pre-established in the fan frequency inverter (20, 30, 40, 45, 50 and 60 Hz). The pressure and flow temperature data for each sample are collected and presented in Tab. 3.The surface temperatures of the tube and the bulk flow are measured at 1-second intervals and recorded in a file in csv format. Thus, at least 6000 temperature values are obtained for each tube, from which the temperature curves are generated as a function of time (Fig. 5).

Table 3. Flow temperature and dynamic pressure* for each sample

			Sample #			
			1	2	3	4
20 Hz	$T\infty$	(°C)	22,0	21,9	22,5	22,6
	Δh	(mm)	2,5	2,5	2,5	2,5
30 Hz	$T\infty$	(°C)	21,9	21,9	22,4	22,5
	Δh	(mm)	6,0	5,5	6,0	5,5
40 Hz	$T\infty$	(°C)	21,9	21,9	22,5	22,4
	Δh	(mm)	11,5	11,5	11,0	10,5
45 Hz	$T\infty$	(°C)	22,0	22,0	22,5	22,4
	Δh	(mm)	14,5	14,5	14,0	13,5
50 Hz	$T\infty$	(°C)	22,0	22,0	22,5	22,4
	Δh	(mm)	18,5	18,0	17,5	17,0
60 Hz	$T\infty$	(°C)	22,1	22,1	22,6	22,5
	Δh	(mm)	26,5	26,0	25,5	24,5

* Dynamic pressure is measured in mmH20



(a) Sample#1 (b) Sample#2

(c) Sample#3 (d) Sample#4

Figure 5. Surface temperature profile for each sample.

3.2 Convective heat transfer coefficient

In the experimental investigation conducted by Annop (2015) with finned serrated tubes, the air inlet temperature and air velocity was measured in the test section, after attaining the steady state. Okbaz et al. (2020) observed that the fluid flow and heat transfer are steady after 200th measurement. When keeping constant the test parameters, Zhang (2019) reported that the flow reach steady state if the variation amplitude was under 0.5% per minute and temperature variation amplitude was within 0.5 per minute. Then the experimental data could be collected.

During the tests, pre-defined fan rotations were used for all samples. For each rotation, a minimum time of 20 minutes was maintained in the steady state condition, taking the readings in the last 5 minutes, equivalent to 700th measurements.

The flow velocity and Reynold number corresponding to each of these intervals are shown in Tab. 4. Then, the convective coefficient (Eq. (4)) and Nusselt number (Eq. (5)) are calculated. The Nusselt and Reynolds numbers are calculated with air properties taken at the film temperature $T_f = (T_s + T_\infty)/2$. The results for Nusselt number are shown in Tab. 5.

Table 4. Flow velocity and Reynolds number* for each tube.

			Sample #			
			1	2	3	4
20 Hz	$u\infty$	(m/s)	6,40	6,40	6,40	6,40
	Re_D	( )	5769	7758	7959	8180
30 Hz	$u\infty$	(m/s)	9,91	9,49	9,92	9,49
	Re_D	( )	9675	11810	12607	12340
40 Hz	$u\infty$	(m/s)	13,72	13,72	13,43	13,12
	Re_D	( )	14022	17410	17285	17226
45 Hz	$u\infty$	(m/s)	15,40	15,40	15,15	14,87
	Re_D	( )	16117	19643	19562	19590
50 Hz	$u\infty$	(m/s)	17,4	17,16	16.94	16,69
	Re_D	( )	18476	21973	21954	22054
60 Hz	$u\infty$	(m/s)	20,83	20,63	20,45	20,04
	Re_D	( )	22630	26588	26682	26566

* $Re_D = u_\infty D/\upsilon$

Table 5. Convective coefficient and Nusselt number.

Sample#	A_s (m²)	ω (Hz)	q_e (W)	T_s (°C)	T_∞ (°C)	$\bar{h}$ (w/m²K)	$\overline{Nu}_D$ ( )
1 (plain tube)	0.0134	20	175	186.5	22.0	79.4	54.1
		30	176	153.1	21.9	100.1	70.9
		40	176	134.5	21.9	116.6	84.5
		45	175	125.2	22.0	126.5	92.7
		50	175	119.4	22.0	134.1	99.0
		60	175	110.4	22.1	147.9	110
2 p = 15mm	0.0651	20	181	69.6	21.9	58.3	45.9
		30	183	60.3	21.9	73.2	58.4
		40	182	53.5	21.9	88.5	71.2
		45	180	51.8	22.0	92.8	74.9
		50	180	50.4	22.0	97.4	78.7
		60	180	48.0	22.1	106.8	86.6
3 p = 10mm	0.0914	20	175	60.2	22.5	50.8	40.5
		30	175	52.4	22.4	63.8	51.4
		40	175	48.0	22.4	75.1	60.9
		45	176	46.9	22.5	78.9	64.1
		50	176	45.6	22.5	83.4	67.8
		60	173	43.2	22.6	91.9	75.0
4 p = 5mm	0.1700	20	171	50.5	22.6	36.1	29.1
		30	173	44.7	22.5	45.8	37.3
		40	175	41.2	22.4	54.8	44.8
		45	174	40.2	22.4	57.5	47.2
		50	174	39.1	22.4	61.3	50.3
		60	174	37.9	22.5	66.5	54.7

The Nusselt number obtained for the plain tube is compared to the Nusselt calculated from the established correlations of Zukauskas (1972) expressed in Eq. (8) for interval $1000 \lesssim Re_D \lesssim 200000$ (Tab. 6). It should be noted that Zukauska's correlation requires fluid properties taken at the flow temperature, except for Prandtl, which must also be taken at the surface temperature. In this work, on the other hand, the properties of the fluid must be taken in the film temperature, which explains the differences in Reynolds numbers for the same flow rate.

$$\overline{Nu}_{D,z} = 0.26 Re_D^{0.60} Pr^{0.37} \left[\frac{Pr(T_\infty)}{Pr(T_s)}\right]^{1/4} \tag{8}$$

The Pearson's chi-square test indicates a probability greater than 99,99% of having statistically significant correlation between Nusselt number obtained for single tube and Nusselt number calculated from the Zukauskas formula, which is the normally accepted criterion. Thus, the proposed experiment is capable of generating reliable results and it is reasonable to extrapolate this confidence for the case of finned tubes.

Table 6. Comparison with Zukauska's correlation

$Re_{D,z}$ ( )	$Pr(T_s)$ ( )	$Pr(T_\infty)$ ( )	$\overline{Nu}_{D,z}$ ( )	$\overline{Nu}_D$ ( )	Δ (%)
8904	0,6979	0,7077	53,8	54,1	-0,60
13800	0,6982	0,7077	70,0	70,9	-1,39
19105	0,6987	0,7077	85,0	84,5	0,60
21444	0,699	0,7077	91,1	92,7	-1,78
24221	0,6993	0,7077	98,0	99,0	-0,98
28976	0,6997	0,7077	109,1	110,4	-1,18
			$\chi 2$	0,9999	

Plotting values of $\overline{Nu}_D/(D_f/p)^s$ and Reynolds number on a log-log graph results in straight lines, as shown in Fig. 6, making possible to find the coefficients *m* by means of linear regression. This exponent *m* associated with the Reynolds number corresponds to the slope of the lines, the average of the exponents obtained in Fig. 6 is assumed, resulting in *m* = 0.519. The coefficients *B* and *s* of Equation (3) are obtained using an Excel supplement function for hypothesis testing called Solver, in which the objective is to find an optimal value for *B* and *s* that corresponds to a minimum error. The application of the Solver function results in *B* = 0.776 and *s* = -0.418. Thus, the proposed correlation for external forced convection in ASTM tubes with fins perpendicular to the air flow is expressed by Eq. (9), being valid for Pr ≈ 0.7 and $5000 \lesssim Re_D \lesssim 27000$. The experiments done by Mangrulkar et al. (2020) and Gonzalez et al. (2019) conducted to a similar Reynolds number range.

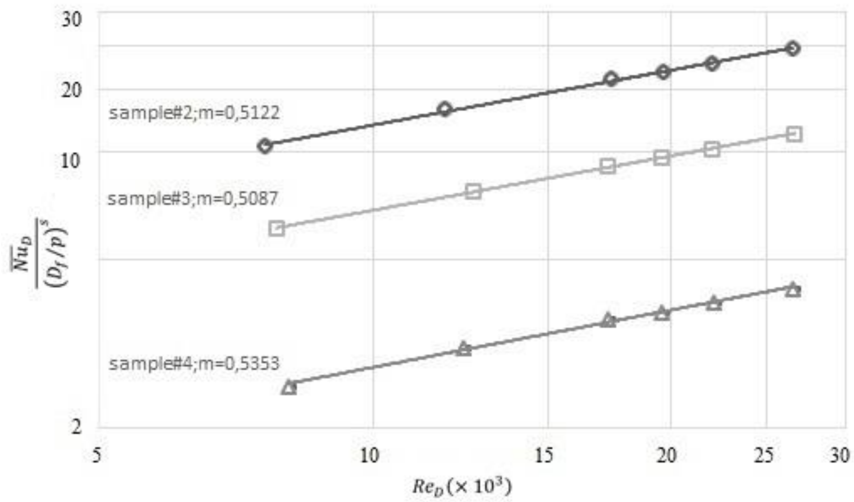


Figure 6. Variation of Nusselt values with Reynolds number for each finned tube sample

It is important to inform that the above experimental analysis was performed for a single finned tube and with the flow perpendicular to the tube axis. The correlation proposed for external forced convection in ASTM tubes with non-standard fins perpendicular to the air flow is expressed by Eq. (9).

$$\overline{Nu}_D = 0,776 Re_D^{0,519} \left(\frac{D_f}{p}\right)^{-0,418} \tag{9}$$

Once Equation (9) is obtained, the values of samples 2, 3 and 4 in Tab. 1 are considered. The calculated results are compared with the results generated experimentally, as shown in the Tab. 7.

Table 7. Comparison between Equation (9) and other experimental data.

Sample#	Re_D	$\overline{Nu}_D$ Experimental	$\overline{Nu}_D$ Proposed	Δ (%)
2	7758	45,9	46,5	1,3%
	11810	58,4	58,2	-0,4%
	17410	71,2	70,3	-1,3%
	19643	74,9	75,4	0,7%
	21973	78,7	79,5	1,0%
	26588	86,6	87,9	1,4%
3	7959	40,5	39,8	-1,8%
	12607	51,4	50,4	-2,0%
	17285	60,9	59,6	-1,8%
	19562	64,1	63,4	-1,0%
	21954	67,8	67,4	-0,7%
	26682	75,0	74,3	-0,9%
4	8180	29,1	30,2	3,5%
	12340	37,3	37,3	0,0%
	17226	44,8	44,5	-0,7%
	19590	47,1	47,5	0,7%
	22054	50,3	50,6	0,5%
	26566	54,7	55,6	1,7%
		$\chi 2$	< 0,9999	

3.3. Limitations and further developments

The limitations of the study presented for using Eq. (9) are: the use of just one fluid (air), the Prandtl number values are limited to $Pr \approx 0{,}70$, the temperature range is $20\ °\mathrm{C} \lesssim T_s \lesssim 200\ °\mathrm{C}$, the range of Reynolds number is limited to $5000 \lesssim Re_D \lesssim 27000$ and the availability of different finned tube profiles and materials. According to Equation (7) it should be expected an uncertainty about ± 2.76% in Eq. (9). However, two other sources of errors in Eq. (9) comes from disregarding radiation heat transfer and considering fin efficiency equal to 1.

For future developments it is suggested to test other finned tube profiles, varying the material, the diameter of the tube, the thickness, the temperature, the height and the fin pitch, thus making it possible to obtain a more comprehensive correlation.

4. CONCLUSION

A sample plain tube (without fins) and three different samples of finned tubes, with different pitches, were tested to experimentally determine the convective coefficient of heat transfer. Then, Nusselt numbers are calculated to develop a generalized experimental correlation. The experimental Nusselt number was compared to the classic Zukauska’s correlation for plain tubes. An acceptable error range was obtained and it was found that the experiment is capable of generating reliable results, making it possible to extrapolate this confidence to the finned tubes. The Person's chi-square showed that there is a greater than 99.99% probability of having a statistically significant correlation between the results generated by the proposed correlation and the results obtained by the classical Zukauska’s correlation. The analysis of the data of the finned tubes samples, combined with exponential equation adjustment techniques, allowed to generate the generalized experimental correlation and to determine errors from the uncertainty analysis. The generalized correlation for finned tubes obtained in this work is capable of calculating the Nusselt number values with good precision, providing reliable parameters in the design of heat exchanger that uses finned tubes. This work contributes with the addition of technological value to the industrial sector, facilitating the use of materials available in the Brazilian market, as is the case with standard ASTM tubes. The creation of a relative simple and robust calculation methodology applicable to other finned tube profiles provides a reliable ground for validating future numerical and computational modeling.

5. ACKNOWLEDGEMENTS

The authors are grateful to the support of:

FERLEX Company, for making the sample material available.

Electrical Engineer Gabriel Dallacqua, sales consultant at Novus Automation, for his support and training in the use of data acquisition sensors.

Mr. Rodolfo dos Santos and Ms. Livia Vitória de Souza Batista, for their valuable assistance with the experiments.

Prof. Mauricio Araujo Zanardi and Mr. Antonio Augusto Moretti Rizzato, who designed and built the wind tunnel.

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