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INFLUENCE OF ROUGHNESS ON THE TUBE HEAT TRANSFER SUBMITTED TO QUENCHING

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Abstract. *This work aimed to analyze, on a permanent basis, the influence of the surface finish on the heat transfer of a steel tube, submitted to the immersion tempering process. The proposed methodology consisted of heating a steel tube internally using electrical resistors with a maximum power of 36 kW. The tests were conducted under rotations varying between 10 and 60 rpm, water temperature equals to 25°C and surface temperature of the tube varying between 90 and 115ºC. The experimental data were evaluated using correlations proposed in the literature. Initially, it was seen that the roughness increase implies a more efficient thermal exchange, however with the temperature increase this influence was reduced to all rotations. In addition, the influence of roughness increased with the speed of rotation of the test tube. From the preliminary evaluation of the experimental data, through correlations provided in the literature, it was possible to propose two correlations for the mixed convection, which are better suited to the empirical data. These equations presented average relative percentage error (Emed) equals to 3.8% and 2.3%. Correlations were estimated from experimental data with Re ranging between 100,610 - 229,116 and Gr ranging between* $7.2x10^9$ *–* $1.63x10^{10}$.

Keywords: *Heat transfer coefficient, Surface roughness, Quenching.*

1. INTRODUCTION

Heat treatment can be defined as the set of heating and cooling operations to which steels are submitted, under controlled conditions of temperature, time, atmosphere and cooling speed, in order to change their mechanical properties and confer certain characteristics. Chiaverini (2005) lists influential factors in heat treatments and highlights cooling as the most important factor, since it is responsible for effectively determining the microstructure, and consequently, the final properties of the steel.

Quenching is a heat treatment aimed at the formation of bainite and martensite and prevents the obtaining of microstructures such as ferrite and perlite. In general parameters, this process can be divided into three stages: vapor phase, nucleated boiling and convective cooling. In the first stage, a steam wrap is formed at the workpiece interface as soon as the process starts; as the steam has low thermal conductivity, it acts as an insulator. Thus, the heat transfer in this phase is low and mainly by radiation. As the temperature decreases, the steam wrap becomes unstable and collapses, initiating the nucleated boiling process. Due to vaporization, the heat transfer at this stage is faster and greater, and persists until the surface temperature drops below the boiling point of the cooling medium. In the final stage of the process, heat transfer occurs through convection (Totten et al., 1993).

The microstructural evolution processes, such as the decomposition of austenite, are thermally conducted and are consequently directly affected by the intensity of the cooling employed. Different cooling paths lead to different phase transformation products, such as ferrite, perlite, bainite and/or martensite (Nobari, 2014). Thus, the industrial importance of mathematical models of heat transfer is highlighted, such as those developed by Chen (2016) and Gomes (2017), which allow to predict or monitor the temperature profiles of steel tubes submitted to the heat treatment of quenching. The quality and validity of these mathematical models are associated with the heat transfer coefficient, which is influenced by several factors, including the surface finish of the product to be quenched.

Cope (1941) carried out work in which three tubes were tested, their internal surfaces being artificially rough. The device used was a parallel flow heat exchanger, the working fluid was water and with Reynolds (Re) ranging from

2,000 to 60,000. The results indicated that in the transition region between laminar and fully turbulent flow, the roughness can increase this coefficient to a value considerably higher than those for smooth tubes.

Nikuradse (1950) carried out a complete study of the effect of roughness on friction and speed distribution, through a series of experiments with the internal flow in rough tubes for grains of sand. The measurements showed that the velocity distributions are slightly dependent on the Reynolds number (Re), but are more dependent on the relative roughness. In addition, it was found that: for small Re, the resistance factor (λ) values are similar to those of smooth tubes; for intermediate Re, λ increases for an increasing number of Re; and for large Re, λ does not depend on this variable and can be obtained using a simple formula, depending only on the relative roughness.

Dipprey and Sabersky (1962), conducted a study to evaluate the relationship between heat transfer and roughness in smooth and rough tubes. The experiments were conducted with distilled water flowing through 0.4 inch diameter tubes that were heated by the passage of alternating electric current through the walls of the tube. Three rough tubes were used. The authors observed that for any Re and Prandtl (Pr), the convective coefficient of heat transfer by convection (h) progressively increases to higher values of roughness, and, correspondingly, to higher coefficients of friction. In addition, it was observed that for a given Pr, the general trend is that h increases with Re in the transition region.

Zhang et al. (2014) investigated the effects of spray pressure and surface roughness on the heat flow (q”) and heat transfer coefficient (h) at the interface between an aluminum surface. The experimental apparatus consisted of a water spray cooling system and a sample and data acquisition system. With the tests, the authors found that a higher spray pressure tends to produce a greater flow of interfacial heat and a convection heat transfer coefficient.

Corrêa (2019) carried out an experimental study to determine the heat transfer coefficient between the outer surface of a steel tube and water. For this, a bench and tests capable of representing the conditions of a real quenching process were used. During the tests, a steel tube heated by means of electrical resistors disposed inside was used. Tests were carried out in a steady state with the tube surface temperature varying between 50°C and 115°C and tube rotation speeds between 0 and 40 rpm. The author analyzed that for higher surface temperatures and higher rotations, higher heat exchange coefficients were evaluated.

This work aims to study the influence of the surface finish of seamless steel tubes on the heat transfer coefficient during the heat treatment of immersion quenching.

2. METHODOLOGY

The methodology applied to analyze the influence of surface roughness on heat transfer during immersion quenching consisted of an experimental procedure for characterizing the surface; performing an experimental procedure that simulates the quench cooling curve by immersion; performing uncertainties analysis; experimentally determining the heat transfer coefficient and validating the obtained equations.

The characterization of the surface of the horizontal tubes was carried out through the daily measurement of the surface roughness, for this purpose the portable roughness meter Surtronic S128, from the manufacturer Taylor Hobson was used. Ten measurements were made in each of the test tube quadrants, to subsequently obtain the average of the indications. The main parameter evaluated was the arithmetic mean of the deviations from the surface (R_a).

As the roughness between the tests did not vary equally, two parameters were defined that made it possible to compare the results. Increase in roughness (b), “Eq. (1)”, is the relative percentage difference between the surface roughness values of two of the tests performed.

$$b = \frac{R_{a,1} - R_{a,2}}{R_{a,1}} \, 100 \tag{1}$$

Average increase (a), “Eq. (2)”, is the average of the relative percentage difference, for each temperature, between the heat transfer coefficient of two of the tests performed.

$$a = \sum_i^n \frac{h_{Temperature\ i_{Test1}} - h_{Temperature\ i_{Test2}}}{h_{Temperature\ i_{Test1}}} \, 100 \tag{2}$$

The experimental procedure was carried out in the laboratory of the Refrigeration and Heating Group (GREA) of the Federal University of Minas Gerais (UFMG), which is located in the city of Belo Horizonte, in Pampulha region, in this region the water boiling temperature is approximately 97 ºC. The experimental procedure consisted of reproducing in a punctual and stationary way a range of temperatures that occurs in the real tempering process, this simulation allowed the study of the heat transfer in the studied temperature range and thus to predict the convection heat transfer coefficient that is equivalent to that of the quenching process, in this same temperature range. In total, 78 trials were carried out between 07/16/2019 and 08/28/2019. The bench used consists of a test tank, with a water-cooling system, in which a specimen, heated by electrical resistances, is rotated by means of mechanical actuation. “Figure 1” shows the components schematically.

The specimen used in the tests, "Fig. 2", consists of a 7" (177.8mm) steel tube with a length of 200 mm. The heating of the specimen during the tests came from a copper billet located inside the steel tube. In this billet, there are 18 electric longitudinal resistors with 2 kW of power, so that the maximum operating power is 36 kW, considering the eighteen resistors.

The temperature measurement of the specimen was carried out by a set of thermocouples, located in strategic positions. The central thermocouple (T_c) is responsible for measuring the temperature in the center of the copper billet and its function is to act as an instrument to control the maximum temperature reached by the specimen. The temperature of the steel tube was measured by means of 8 thermocouples, two positioned in each quadrant of the tube, at half the length. These thermocouples were positioned radially close to the inner and outer surface of the tube. Figure 2 also shows the cross section of the tube and the positioning of the thermocouples.

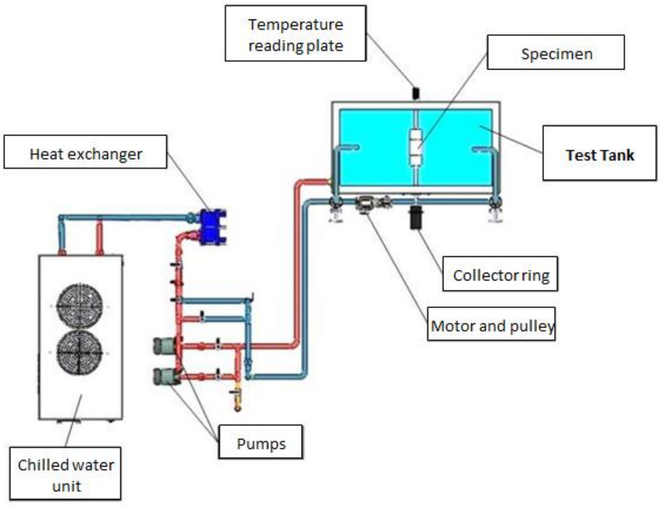

Figure 1. Schematic of the experimental bench.
Available from: Corrêa (2019)

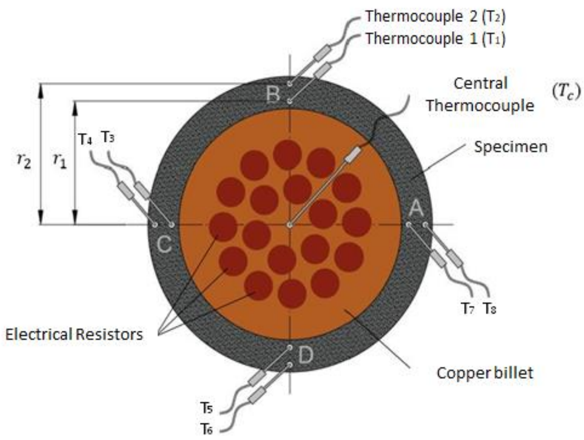

Figure 2. Specimen and Scheme of the cross section.
Available from: Adapted from Corrêa (2019)

The test tank has dimensions of length, width and height equal to 2 m, 1 m and 1.5 m, respectively, which means that are used around 3,000 liters of water in the tests. The test tank has two mobile thermocouples for measuring water temperature. During the tests, they were positioned at the same height as the specimen, at a distance of 30 cm. This position is recommended by Corrêa (2019), who carried out preliminary bench tests.

The activation of the mechanical assembly was carried out by means of a bevel geared motor model KT67 of 2 CV from the manufacturer SEW-Eurodrive, with a reduction factor of 7.28 that allows the maximum rotation of 235 rpm at the output.

The measurement of the temperature of the specimen and the water contained in the tank was performed using T-type thermocouples (positive copper and negative constantan) from the manufacturer Ecil. The working range of this device is between -270ºC and 370ºC, so it is suitable to use it in the tests. In addition, this thermocouple is recommended in oxidizing atmospheres, which makes it resistant to corrosion in humid environments. The thermocouples used are encapsulated with mineral insulation and have a reduced diameter of 1.5 mm and a sheath length of 15 cm. As the specimen during the tests was placed in rotational movements, the temperature signals were transmitted via wi-fi, through a National Instruments plate.

The hydraulic circuit is responsible for maintaining the circulation of water in the system, as well as for carrying water for cooling. The specimen remains heated throughout the test period, so the water temperature in the vicinity of the specimen is increased due to heat transfer processes. However, it is necessary to keep the water temperature constant during the tests, for this purpose a gasketed plate heat exchanger was used. The water-cooling system of the experimental bench consists of a model T5-MFG exchanger from the manufacturer Alfa Laval, with a thermal capacity of 55 kW.

During the tests, the rotation of the tube and the surface temperature of the specimen were varied, while the water temperature was kept constant at 25ºC. The experimental conditions of these parameters are shown in bands in Table 1.

Table 1. Experimental Conditions

Parameter	Working Range
Tube rotation (rpm)	10, 30, 50 e 60
Tube surface temperature range T_s (ºC)	90 ºC – 115 ºC
Tank water temperature T_w (ºC)	25 ºC

The test procedure performed, Fig. 3, was in accordance with that proposed by Corrêa (2019).

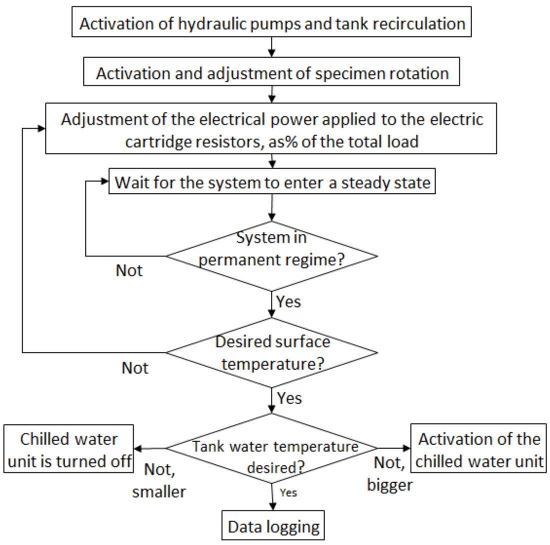

Figure 3. Test procedure.
Available from: Corrêa (2019)

2.1 Analysis of Uncertainties

The variables uncertainties measured experimentally were obtained through data provided by the equipment manufacturers according to their manuals and/or technical specifications. Such uncertainties are highlighted in Table 2.

To calculate the measurement of the variables uncertainty calculated: Power, Surface area, temperature variation and heat transfer coefficient, an uncertainty propagation method was used for more than one independent variable, “Eq. (3)”.

$$u_x^2 = \left(\frac{\partial x}{da}\right)^2 u_a^2 + \left(\frac{\partial x}{db}\right)^2 u_b^2 + \left(\frac{\partial x}{dc}\right)^2 u_c^2 + \cdots \tag{3}$$

In “Eq. (3)”, u_x, the evaluated variable uncertainty, is the sum of the square of the partial derivative of each parameter multiplied by the square of their uncertainty.

Table 2. Uncertainties of the measured variables.

Name	Instrument Name	Uncertainty
Temperature	T-type thermocouple	± 0,5 °C
Current	SCT-013 sensor	± 3 % of the value read
Tension	MVB DSO Oscilloscope	± 3 % of the value read
Tube dimensions	Mitutoyo 530-104B Caliper	± 0,5 mm
Position of internal thermocouples	Lathe / Milling Machine	± 0,3 mm
Roughness	Rugosimeter	5%

2.2 Evaluation of Correlations

The experimental results were evaluated according to correlations obtained in the literature. In order to verify the accuracy of these correlations, the relative percentage error of each point (E_{rel}) and the average relative percentage error of each correlation (E_{med}) were calculated, according to "Eq. (4) and (5)".

$$E_{rel,n} = \left(\frac{|Y_{Exp,n} - Y_{Cal,n}|}{Y_{Exp,n}}\right) 100 \tag{4}$$

$$E_{med} = \left(\frac{\sum_1^n E_{rel,n}}{n}\right) 100 \tag{5}$$

Where n is the number of data and subscripts exp and cal are the experimental and calculated values, respectively.

2.3 Experimental determination of the heat transfer coefficient and validation of the obtained equations

The experimental determination of the heat transfer coefficient was given by the thermal exchange between the steel tube, the surface was kept at a constant temperature (T_s) and the water was kept at a constant temperature (T_w = 25 ºC). Heat transfer occurred by convection and the heat transfer coefficient (h) is the amount of heat transferred from one unit of surface area per unit of temperature, and was calculated by Newton's law of cooling, "Eq. (6)".

$$h = \frac{q''}{T_s - T_w} \tag{6}$$

The characterization of convection during the tests was carried out using the dimensionless parameter Nusselt (Nu). The calculation of the experimental Nu can be performed with the "Eq. (7)". The experimental values were compared with values obtained through correlations proposed in the literature.

$$Nu = (h.D/k) \tag{7}$$

Where h is the convection heat transfer coefficient, D is the diameter and k is the thermal conductivity of the fluid.

For the forced convection, correlations proposed by Becker (1963) and Corrêa (2019) were used. The correlation proposed by Becker (1963) is presented in "Eq. (8)".

$$Nu = 0{,}133\, Re^{2/3}\, Pr^{1/3} \tag{8}$$

The correlation proposed by Corrêa (2019) is evidenced in "Eq. (9)".

$$Nu = 0{,}235\, Re^{2/3}\, Pr^{1/3} \tag{9}$$

For natural convection, correlations proposed by Morgan (1975) and Churchill and Chu (1975) were used. The correlation proposed by Morgan (1975) is shown in "Eq. (10)". Where Ra represents Rayleigh's Dimensional Number.

$$Nu = 0{,}125\, Ra^{0{,}333} \tag{10}$$

The correlation proposed by Churchill and Chu (1975), "Eq. (11), is valid for $Ra \leq 10^{12}$.

$$\mathrm{Nu} = \left\{0{,}60 + \frac{0{,}387\,\mathrm{Ra}^{1/6}}{\left[1+\left(\frac{0{,}559}{\mathrm{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 \tag{11}$$

The mixed convection was evaluated according to "Eq. (12)", with parameter n = 4. In addition, the correlation proposed by Corrêa (2019), "Eq. (13)", in which Gr represents Grashof's Dimensional Number.

$$Nu^n = Nu_f^n \pm Nu_n^n \tag{12}$$

According to geometry, Nu_f^n and Nu_n^n numbers are determined according to the proposed correlations for forced and natural convection, respectively. The positive sign indicates that the flow is parallel or transverse, while the negative sign indicates the opposite to natural convection flow.

$$\mathrm{Nu} = 0{,}04634\,(1{,}959\,\mathrm{Re}^2 + \mathrm{Gr})^{0{,}4031} \tag{13}$$

3. RESULTS

During the tests and during their interval, the specimen was kept either immersed in water, inside the tank, or kept in contact with the atmosphere. The changes in the surface layer of the specimen were due to oxidation, caused by the humid environment in which it was found. Table 3 presents a summary of the values for the average roughness (R_a) obtained in the specimen on the day that each test was performed, values that were used in the development of the correlations.

Table 3. Specimen roughness (µm).

Test (-) / Velocity (rpm)	1	2	3
10	11.65 ± 0.97	14.02 ± 1.06	13.98 ± 0.91
30	13.68 ± 0.95	12.94 ± 0.84	12.81 ± 0.84
50	12.66 ± 1.32	13.13 ± 0.79	12.27 ± 0.92
60	12.66 ± 1.32	12.51 ± 0.98	12.27 ± 0.92

According to the data in Tab. 3, it can be seen that the roughness of the specimen did not vary significantly between the tests, that is, the oxidation mechanism caused mild changes in its surface characteristics. This situation is similar to what happens in an industrial heat treatment plant. Products subjected to heat treatment come from the same manufacturing process and are therefore expected to have similar surface characteristics or with minimal changes.

The results of the tests conducted were expressed as three curves heat transfer coefficient (h, W/(m² ºC)) versus Temperature (T, ºC), for rotations 10, 30, 50 and 60 rpm, being that to lift these curves 78 experimental points were measured. The curves for the 10 and 60 rpm rotations are shown in Fig. 4, and through this, it is possible to verify the influence of the rotation speed, temperature and surface on the heat transfer, through the values obtained for the coefficient h. The main conclusions when analyzing the curves are:

i) The heat transfer coefficient increases with increasing tube temperature;

ii) The heat transfer coefficient increases with increasing tube rotation;

iii) The rougher the surface, the higher the values observed for the heat transfer coefficient.

Regarding the first finding about the increase in the heat transfer coefficient with the increase in the surface temperature, this is because h is calculated as a function of the numbers of Grashof, Reynolds and Prandtl, these parameters, in turn, are calculated as a function of the film temperature, being directly proportional to it, that is, the average between the surface and water temperature, so this increase is expected. In relation to the second finding, the increase in the heat coefficient with the increase in rotation occurs because higher rotations imply higher Reynolds numbers.

Table 4 presents information on the influence of roughness on the heat transfer coeficiente.

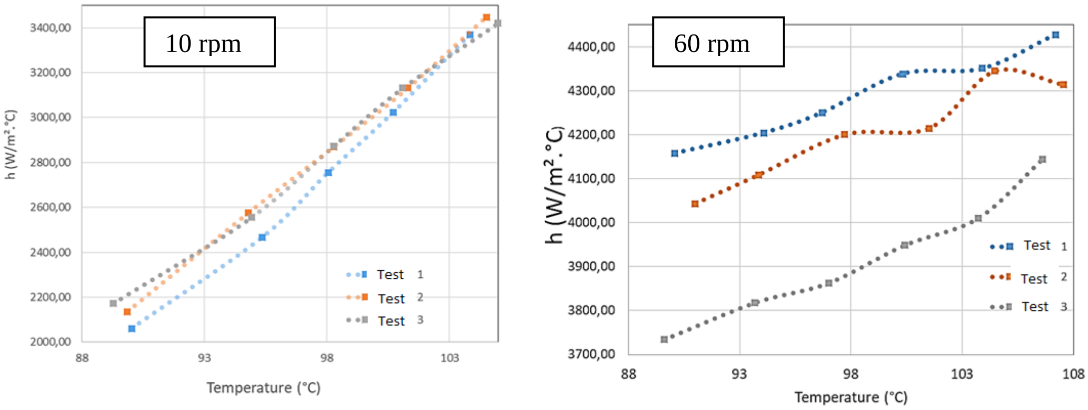

Figure 4. Test results for 10 and 60 rpm speeds.
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Table 4 . Influence of roughness on heat transfer.

Rotation (rpm)	10	30	50	60
Roughness Increase (b,%)	20.0	5.7	7.0	3.0
Temperature (ºC)	Increase in heat transfer coefficient (%)			
90	3.66	9.58	10.23	11.35
95	4.43	7.10	8.95	10.13
98	4.16	7.02	9.53	10.03
101	3.53	7.19	7.64	9.88
104	2.33	6.19	9.45	8.49
107	-	7.62	6.40	6.84
109	-	5.76	5.01	-
112	-	5.49	-	-
Average increase (a,%)	3.62	6.99	8.17	9.45
Ratio (a/b)	0.18	1.23	1.17	3.2

The data shown in Tab. 4 allows us to draw some conclusions about the influence of roughness on heat transfer. Note, initially, that the increase in roughness implies a more efficient thermal exchange, however, with the increase in temperature this influence was reduced for all rotations.

Using parameters "a" and "b", the ratio between the average increase in the heat transfer coefficient and the increase in roughness was calculated for each of the rotations. Through this reason (a/b) it was possible to notice that the influence of the roughness increased with the increase in the speed of rotation of the test tube, this can be explained by the effects of turbulence, which increases with the rough surface. It is known that there is a limit, that is, from a certain roughness, this parameter no longer influences the thermal exchange, but this limit was not reached during the tests. The increase in the influence of roughness had an approximately linear characteristic, except for the rotation of 50 rpm, which obtained a ratio value of 1.17, very close to the value obtained by the rotation of 30 rpm, of 1.23, which may mean a region of stagnation from the influence of roughness.

3.1 Evaluated uncertainties

The relative uncertainties of power (Q), surface area (s), temperature variation (ΔT) and heat transfer coefficient (h) are shown in Table 5. The variables uncertainties measured experimentally were obtained through data equipment manufacturers according to their manuals and/or technical specifications.

Table 5. Relative uncertainties of the calculated variables.

Relative Uncertainty	u(Q)/Q	u(s)/s	u(ΔT)/ΔT	u(h)/h
-	0.04	0.01	0.02	0.05
%	4.24	0.81	1.85	4.70
influence (%)	81.48	2.97	15.55	100

As can be seen from Table 5, the variable that most influences the uncertainty of the heat transfer coefficient is power, being responsible for approximately 80% of this uncertainty. The indirect measurement of the surface temperature is another factor that influences the uncertainty of the h, with an influence of approximately 15%.

The improvement in the measurement of these variables, especially in the measurement of power, is the factor that will contribute to obtain values with less uncertainty for the heat transfer coefficient in the tests.

3.2 Evaluation of results and comparative analysis of correlations for mixed convection

In order to carry out the evaluation of the experimental data through correlations obtained in the literature, the data was divided according to the type of heat transfer that occurred during the tests: forced, natural or mixed convection. The form used to verify the type of convection was extracted from the Gr/Re^2 ratio for each of the experimental points.

It was verified that the average value of the Gr/Re^2 ratio for the experimental points referring to the rotation of 10 rpm was around 9, which means that the thermal exchange occurred through the natural convection regime. The experimental points referring to the rotations of 30, 50 and 60 rpm presented average values for the relation already mentioned equals to 0.95, 0.32 and 0.22, respectively. These values are between 0 and 1, therefore, the heat transfer process took place through mixed convection.

For the validation of the experimental data concerning the 10 rpm rotation and for the natural convection portion of the experimental data relating to the 30, 50 and 60 rotations, the correlations of Morgan (1975), "Eq. (10)", and Churchill and Chu (1975), "Eq. (11)". To assess the effects of forced convection, the equations proposed by Becker (1963), "Eq. (8)", and Corrêa (2019), "Eq. (9)". Finally, a correlation proposed by Corrêa (2019), "Eq. (13)", to evaluate the combined effects of natural and forced convection, verified for speeds of 30, 50 and 60 rpm.

As previously mentioned, the heat transfer obtained at the experimental points related to the 10 rpm rotation was characterized as transfer by natural convection, this section being used to analyze these points through the correlation obtained by Morgan (1975). It can be seen that the values for Nu achieved experimentally, are higher than those calculated by the correlation of Morgan (1975). This behavior was verified in the three tests performed for the rotation of 10 rpm, with an average relative error of 51.34%. The justification for this error permeates the fact that the Morgan equation is valid for cases in which the heat transfer occurs only in the radial direction, thus, these results suggest that during the tests the low rotation occurred significant heat losses in the direction. The same occurs for the values for Nu calculated by the correlation of Churchill and Chu (1975). The average percentage error assessed by this correlation was 38%.

Regarding the assertiveness of the correlations used in the validation of experimental data for mixed convection. Figure 5 shows the Nu x Re curves, referring to Test 1 for the 50 rpm rotation. In the Figure it is possible to see the curve referring to the experimental points and also the curves obtained by means of theoretical correlations.

It is possible to verify that the correlations that presented results closer to the values obtained experimentally were: the characterization of the mixed convection by the combination of the equations of Churchill and Chu (1975) and Corrêa (2019), the characterization of the mixed convection by the equation proposed by Corrêa (2019) and the characterization of mixed convection by combining the Morgan (1975) and Corrêa (2019) equations.

These correlations showed an average percentage relative error equal to approximately 9.00%, 9.18% and 9.57%, respectively. In addition, the maximum percentage relative error of these correlations was 25.52%, 19.80% and 27.15%, respectively.

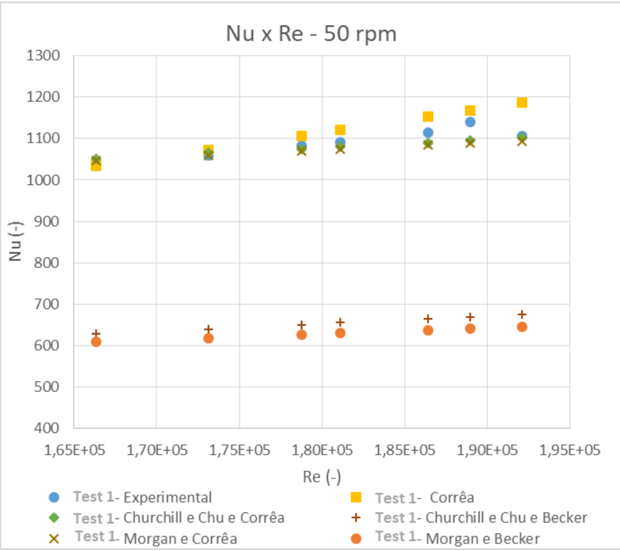

Figure 4. Comparison between the correlations used for mixed convection.
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3.3 Evaluation of results and comparative analysis of correlations for mixed convection

The results achieved in section 3.2 served as an initial approach for the treatment of data obtained experimentally. Two experimental correlations have been proposed for the mixed convection that best fit the experimental data. In order to obtain more accurate results, the influence of surface characterization was considered in these equations, through the average value of surface roughness.

Equation (14) shows the first equation developed for Nu, in this case, Nu was evaluated as a function of surface roughness (R_a), outer diameter of the tube (D_e), and dimensionless numbers Re and Gr. This equation presented E_{med} equal to 3.8% and the maximum E_{rel} found was 9.4%.

$$Nu = \left(\frac{R_a\ 10^{-6}}{D_e}\right)^{-0,05672} \left(0,7409\ Re^2 + Gr^{0,265}\right) \tag{14}$$

Equation (15) exposes the second equation developed for Nu. The Nu number was evaluated according to the surface roughness (R_a), the outside diameter of the tube (D_e), the dimensionless Re and Gr and the constants F, F_1 and F_2 depending on the rotation of the tube and obtained through the graph presented in Fig. 5. This equation presented E_{med} equals to 2.3% and the maximum E_{rel} found was 6.5%.

$$Nu = \left(\frac{R_a\ x\ 10^{-6}}{D_e}\right)^{F} \left(F_1\ Re^2 + Gr^{F_2}\right) \tag{15}$$

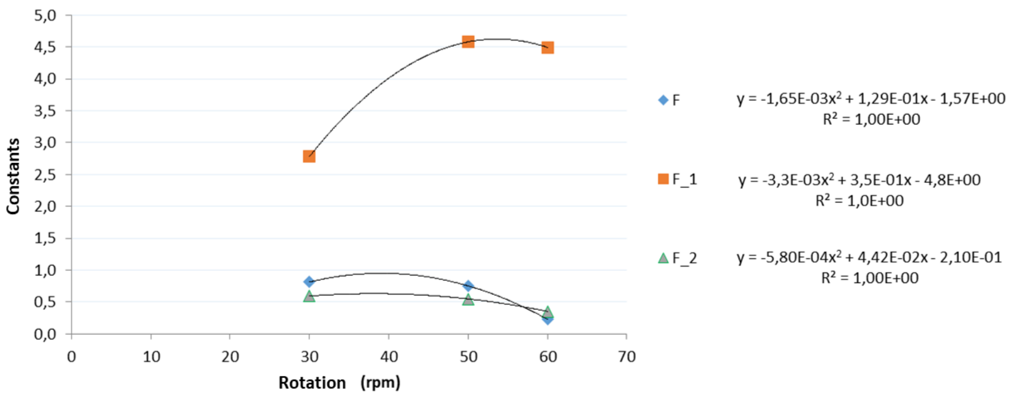

Figure 5. Constants F, F_1 and F_2.
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These correlations, "Eq. (14)" and "Eq. (15)", were obtained using the EES software, as well as the optimization tool provided by the software. Correlations were estimated from experimental data with Re ranging between 100,610 - 229,116 and Gr ranging between $7.2x10^9$ – $1.63x10^{10}$.

4. CONCLUSION

From the study of the "Influence of surface roughness on the heat transfer of steel tubes submitted to quenching by immersion in water", the following conclusions can be expressed:

i) The changes caused in the surface layer of the test body resulting from oxidation, caused by the humid environment in which it was found were subtle;

ii) The convection heat transfer coefficient increased with increasing temperature;

iii) The heat transfer coefficient increased with increasing rotation;

iv) The increase in roughness implied a more efficient thermal exchange;

v) The increase in temperature reduced the influence of the surface finish on the thermal exchange for all rotations;

vi) The influence of roughness increased with the speed of rotation of the test tube;

vii) The increase in the influence of roughness had an approximately linear characteristic, except for the rotation of 50 rpm, which obtained a ratio value equal to 1.17, very close to the value obtained by the rotation of 30 rpm equal to 1.23, which can signify a region of stagnation from the influence of roughness;

viii) The improvement in the surface temperature measurement and mainly in the power measurement is a factor that will contribute to obtain values with less uncertainty for the convection heat transfer coefficient;

ix) For the tests performed with a rotation equal to 10 rpm, the correlations provided in the literature, "Eq. (10)" and "Eq. (11)", were not very assertive, with E_{med} of 51.34% and 38%, respectively. This suggests significant heat losses in the longitudinal direction in tests performed at low speed;

x) For the tests performed with rotations equal to 30, 50, and 60 rpm, that the heat transfer was through the process of mixed convection, the correlations that showed the best characterization of the mixed convection were: combination of the “Eq. (11)” and “Eq. (9)”, the “Eq. (13)” and a combination of the “Eq. (10)” and “Eq. (9)”. These correlations showed E_{med} of approximately 9.00%, 9.18% and 9.57%, respectively. In addition, the E_{rel} of these correlations were 25.52%, 19.80% and 27.15%, respectively.

xi) From the preliminary evaluation of the experimental data, it was possible to propose two correlations for the mixed convection. These equations presented E_{med} of 3.8% and 2.3%. The correlations were estimated from the experimental data with Re varying between 100610 - 229116 and Gr varying between $7{,}2x10^9 – 1{,}63x10^{10}$.

5. ACKNOWLEDGEMENTS

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