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Commissioning of an Experimental Bench to Determine the Internal Heat Transfer Coefficient in Rotating Tubes During the Quenching Process

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Abstract. *The objective of this work is the presentation and commissioning of a test bench for experimental determination of the heat transfer coefficient between the internal surface of a steel pipe and water jets as typically in the quenching process. In general, this process is carried out by immersing the heated rotating tube in a water tank in association with an internal water jet. Mathematical models can be used to simulate various tube cooling conditions during the quenching, but their reliability depends significantly on the uncertainties related to heat transfer coefficients. The relevance of the present study comes from the lack of correlations in the literature to calculate these coefficients. For effects of commissioning, the tests were performed in a tube with a rotation range of 0-100 RPM that operated internally, sometimes with water other with vapor. The results were compared with theoretical correlations available in the literature. For the static tube, in which the correlations presented in the literature are more established, the experimental and theoretical results showed good agreement. MARD of 14% for vapor and 28% for liquid state was obtained, indicating that, the experimental bench and the methodology used may be extended for studies with rotating tubes and subjected to two-phase internal flow.*

Keywords: *Quenching, Rotating tubes, Heat transfer coefficient, Experimental study, Commissioning, Bench test.*

1. INTRODUCTION

Over the centuries, steel has been used in the most diverse fields of engineering, with applicability in several segments, such as, for example, in the fields of oil exploration, the automotive industry and civil construction. Many of these applications require specific mechanical properties of steel, which are achieved through heat treatments. Among these treatments, one of the most important and known is quenching. In summary the quenching process aims to transform the molecular structure of the steel into a martensitic structure, giving it high hardness and high tensile strength. This transformation is achieved by heating the steel to its austenitization temperature, which is between a range of 815^{o}C and 900^{o}C, followed by rapid cooling (Callister and Retwisch, 2012).

Among the various existing quenching processes, cooling in a water tank is one of the most used, due to its simplicity, low cost and high cooling capacity (Corrêa, 2019). Sakamoto *et al.* (2016) describe some techniques to promote a more uniform heat transfer in a steel tube immersed in water, such as tube rotation and the presence of internal and external jets to the tube.

According to Kobasko *et al.* (2010) the heat transfer mechanisms between the part and the cooling medium during quenching are complex. If this medium is water, the heat transfer between the workpiece surface and the fluid occurs differently depending on the temperature range during cooling. Small variations in the cooling rate can cause significant differences in the desired mechanical properties and/or cause the formation of internal stresses and/or deformations in the part. A quenching process may be optimized by applying an appropriate cooling rate, which, in turn, can be determined experimentally or by mathematical models. Experimental studies on industrial quenching systems are costly and time-consuming.

According to Baleta *et al.* (2018) numerical methods still play an insufficient role in the development of the production process. In their work Baleta *et al.* (2018) performed numerical simulation of the spray quenching process to determine the validity of the mathematical models implemented within the CFD commercial code, especially droplet evaporation/condensation and the wall/droplet heat transfer model. The simulation results indicated the transfer model is not able to reproduce the heat transfer for a dense spray of water, thus, the model was improved with the implementation of experimental correlation for heat transfer coefficient during spray quenching.

Ramezanzadeh *et al.* (2017) numerically studied the forced convection quenching process of hot parts in subcooled oil. The CFD code developed by the authors was validated showing good agreement with the analytical solution and existing experimental data.

Greif *et al.* (2017) used the finite element method to simulate the quenching process of aluminum billets under different quenching conditions. The cooling curve of aluminum parts was obtained experimentally, while the heat transfer coefficient was obtained through the inverse method. The finite element model was used to predict residual stresses resulting from the quenching process and was validated using the X-ray diffraction method.

In particular, in the case of heat transfer inside a tube, Seghir-Ouali *et al.* (2006) alert about the difficulty of experimentally establishing a heat transfer correlation involving the influences of the tube rotation and the water flow inside it. One of these difficulties resides in the heat exchange asymmetry involving the internal and external surfaces of these tubes, implying decoupled methods for the experimental determinations of the heat tranfer coefficients in these two surfaces. Seghir-Ouali *et al.* (2006) carried out a study to experimentally identify the heat transfer coefficient by convection inside a rotating cylinder with air flow. The heat transfer coefficients were identified by three methods: inverse method; cylinder wall treated as a thin isothermal layer; analytical method. According to the authors, the results showed the existence of two heat transfer regimes. For low rotational speeds, heat transfer increased with rotational and axial Reynolds numbers. For larger rotations, the values of the Nusselt number tended to values almost independent of the axial Reynolds number. The authors also used a mathematical criterion to separate the two regimes, and correlations were proposed to predict the coefficient in both.

According to Wagner *et al.* (1989) some researchers studied certain rotation conditions and presented results that indicated a large increase in heat transfer, while others found a large reduction, or even results that indicated that rotation did not have significant effects on heat transfer. For the author, this disparity in the results is indicative of differences in the measurement techniques and models used in the experiments, as well as the non-uniformity of the test conditions.

Furthermore, the gigantic nature of the industrial process of quenching steel tubes makes experimental research on the subject rare, with the work of Sakamoto *et al.* (2016) an exception. In their work Sakamoto *et al.* (2016) used the inverse method to determine the cooling curve and the heat transfer coefficient between the part and the water during the quenching process by immersion in a water tank.

Mathematical models are more versatile than experimental studies because they allow you to simulate different part geometries and different cooling conditions related to the quenching process. However, the reliability of the results generated by these models is strongly linked to the accuracy of the heat transfer coefficient between the part and the fluid, a model parameter that is difficult to know with good accuracy. The objective of the present work was the commissioning of an experimental apparatus to determine the heat transfer coefficient between the inner wall of a steel tube and the cooling water under conditions similar to the quenching process. To validate the test bench, the trials were limited to single-phase and internal water flow along the tube.

2. METHODOLOGY

2.1 Experimental methodology

The heat transfer coefficient between the inner surface of the tube can be obtained by classical technique, equating the rate of heat generation on the outside of the tube with the rate of heat transfer on the inner surface of the tube (Incropera *et al.*, 2011). The Figure 1 illustrates the method described above for obtaining an internal convection heat transfer correlation in a tube in which a fluid flows. The tube, wound with an external long resistive wire, is heated by passing an electric current through this element. The tube temperature is kept constant with the heat generated by Joule effect in the resistive element. This heat was absorbed by the tube and integrally recovered by the fluid flowing inside the tube, once this system was insulated and wasn't heat dissipation for another source. The transfer of heat from the tube to the fluid takes place by convection. Thus, by measuring the temperature of the inner surface of the tube and the inlet and outlet temperatures of the fluid, as well as the electrical power supply of the resistive element, the average convection coefficient between the tube wall and the fluid can be calculated from Newton's law of cooling, defined by Eq. 1.

$$Q = h \cdot A \cdot (T_s - T_m) \tag{1}$$

where: Q is the rate of heat transfer from the tube surface measured by wattmeter, A is the heat transfer area of the tube, T_s is the surface temperature and T_m is the average water temperature.

In addition to the methodology described above, a rotation system was mounted next to the tube, allowing the evaluation of the effects of tube rotation on heat transfer. To make the results valid for a wide range of Reynolds and Prandtl,

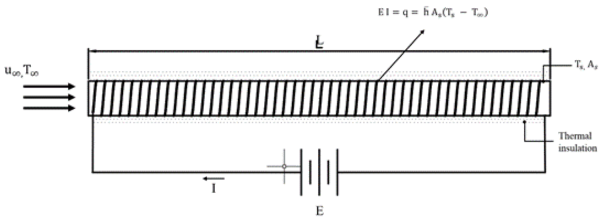

Figure 1. Scheme for measuring the average convection heat transfer coefficient
SOURCE: Adapted from Incropera *et al.* (2011)

the tube rotation, velocity, temperature and nature of the flow was varied. The equation adopted to evaluated the Reynolds number for the stationary condition, no pipe rotation, is described by Eq.2. For the situation were the pipe rotation was considered, the Reynolds number is defined by Eq.3.

$$Re_a = \frac{4\dot{m}}{\mu \pi D} \tag{2}$$

$$Re_r = \frac{\pi \cdot \rho \cdot n \cdot D^2}{60 \cdot \mu} \tag{3}$$

where: $\dot{m}$ is the mass flow rate, n is the pipe rotation, μ is the dynamic viscosity, D is the internal diameter of test section and ρ is the density.

The number of Nusselt was calculated by means Eq.4

$$N_u = \frac{h \cdot D}{k} \tag{4}$$

in that h is the heat transfer coefficient determined by Eq. 1 and k is thermal conductivity of fluid.

All fluids properties was evaluated in the average fluid temperature by Eq.5.

$$T_m = \frac{T_i + T_o}{2} \tag{5}$$

whose parameter, T_i and T_o, are respectively the inlet and outlet fluid temperature.

2.2 Experimental apparatus

The experimental test bench, shown in Figure 2, was developed at the Lab-GREA (Laboratory of the Refrigeration and Heating Group at UFMG). The bench is composed of a test section, a heating and cooling system for the water that circulates in the test section, data acquisition and control system.

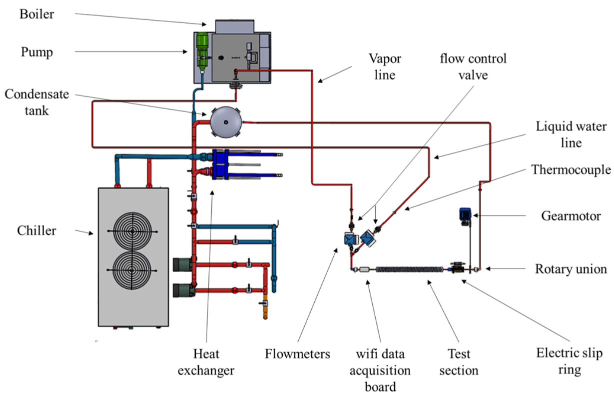

Figure 2. Test bench to determine the internal heat transfer coefficient.

2.2.1 Testing section

The test body of the experimental bench, schematically showed in Figure 3, is a component destined to determine the heat transfer coefficient between the water and the wall of a tube. Basically, the test section is a cylindrical aluminum billet ,length of 1.0 m and external diameter of 76.4 mm, with a channel of 12.7 mm in the center through which water flows. Electrical resistors were regularly wrapped around the outside wall of the billet in order to impose heating in the internal water flow. Upstream of the test section there is a straight pipe section with the same internal diameter of the test section and approximately 40 times this diameter in length, whose function is to ensure a fully developed flow of water from a hydrodynamic point of view. Along the test section, five thermocouples were installed with the tips located approximately 0.2mm of the inner wall. The thermocouple sheaths were inserted into mini-channels and attached into staggered holes, positioned diagonally along of the billet, which were drilled using a long, thin drill. This way, the average temperature of the inner wall of the channel can be determined with good accuracy without interference from electrical resistances. To ensure that virtually all the heat generated in the heaters is dissipated into the water, the set is lined with rock wool insulation above electric resistance. The fluid temperature was measured by thermocouples assembled in the inlet and outlet of test section.

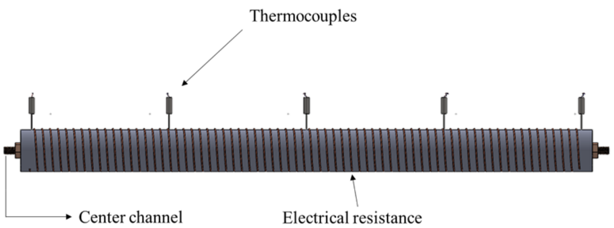

Figure 3. Test body for determining the internal heat transfer coefficient

2.2.2 Rotation system

The rotation system allows evaluating the effects of rotation on the heat transfer coefficient. The activation of this system is done by a geared motor. The rotation transmission to the billet is done through a pulley and belt system. The system rotation control is done by a frequency inverter. This inverter allows the easily management of control the rotation, since the frequency of the device is proportional to the rotation. To know the relationship between these variables, a calibration curve was constructed using a tachometer with an accuracy of $\pm$ 1 RPM. Five measurements were taken for six rotations within the desired operating range. Figure 4 shows the geared motor calibration curve, in which the linearity between rotation and frequency can be observed. The trend line obtained in the calibration (in blue) is between the measurement uncertainty lines (dashed) over the entire interval. Thus, the speed of the gearmotor can be determined using the inverter frequency with an accuracy of $\pm$ 1 RPM.

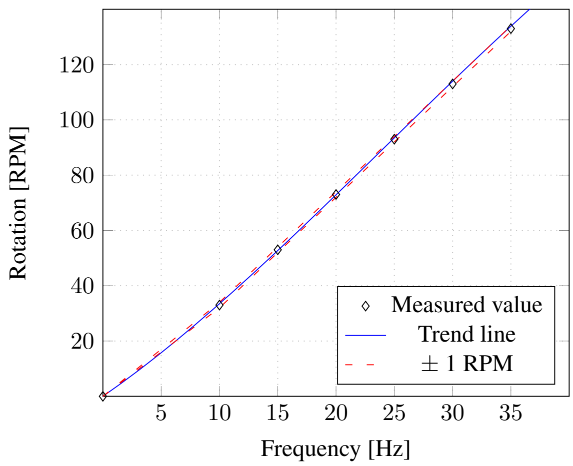

Figure 4. Gear motor rotation calibration curve

The connection between the test section and the piping was carried out using rotary joints. A rotating joint provides a good mechanical seal between the stationary and the rotating tube, allowing water to flow through the inner channel without leaking at the ends. For the transmission of electrical power, a slip ring was used. It is a set of electrified rings designed to supply power for the rotating part from a stationary source. Finally, the signal transmission from the thermocouples was carried out through a WIFI data acquisition board.

2.2.3 Heating system

The heating system for the water used in the test bench consists of an electric boiler, a condensate tank and a feed pump. This system is responsible for delivering liquid water at room temperature or heated; saturated vapor or a mixture of both for the test section. The boiler, built especially for this project, has a vapor outlet at the top and a liquid water outlet at the bottom.

When the system operates only with liquid water, it is possible to control the temperature of the fluid through a thermostat that turns the boiler's electrical resistances in the on/off mode. When the system operates with steam, the boiler is controlled by a pressure switch. When you want to produce a two-phase flow of saturated vapor and liquid, the liquid and vapor outlets are opened, allowing the two flows to feed a mixer, which, upstream, has valves to control the flow rates of vapor and liquid water. Thus, and with the help of energy and mass balance equations, it is possible to impose a quality value (ratio between vapor flow and total flow, recorded in flow meters) for the two-phase flow at the mixer outlet. This test bench operation mode is not the focus of this work, so it will not be covered in detail. The condensate tank has the function of receiving liquid water or condensed vapor from the test section and feeding liquid water to the system pump, which, in turn, feeds back the boiler.

2.2.4 Cooling system

Every time the water comes through the testing section, the water comes out heated. As the system operates in cycle, the continuous operation of the system would also imply a continuous increase in the water temperature. To keep the water temperature stable within a certain range, there is a cooling system connected in series with the test section and boiler. This system is composed of a parallel plate exchanger and a chiller. The heat exchanger has the function of removing the heat supplied by the test section, while the chiller has the function of producing cold water that is used as secondary fluid by the exchanger.

2.2.5 Data acquisition and control system

Temperature measurement is performed by T-type thermocouples, liquid water and saturated vapor flow measurement is performed by turbine and vortex flowmeters respectively. Electrical power is measured by a true rms wattmeter and the rotation measurement is based on the inverter frequency as described above. To acquire the measured data, an acquisition system and a supervisory program developed for this experiment were used.

2.3 Uncertainty Analysis

The uncertainties of experimentally measured parameters were determined using manuals, technical specifications and calibration curves provided by the manufacturers. To calculate the uncertainties of the calculated variables, the method presented by Taylor and Kuyatt (1994) was used, given by Eq. 6.

$$U_y = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial y}{\partial x_i}\right)^2 \cdot U_{x_i}^2} \tag{6}$$

2.4 Error Calculation

To validate the methodology and the test bench used to determine the internal heat transfer coefficient, the data obtained experimentally were compared with correlations proposed in the literature. The precision of the heat transfer coefficient was evaluated through the absolute percentage relative error (MARD), which is defined by Eq. 7.

$$MARD = \frac{1}{n} \cdot \sum_{1}^{n} \left| \frac{y_{i_{cal}} - y_{i_{exp}}}{y_{i_{exp}}} \right| \tag{7}$$

2.5 Correlations for turbulent flow in circular tubes

According to Incropera *et al.* (2011), the analysis of internal heat transfer in tubes is considerably more complex in turbulent flows. Thus, most authors use empirical methods to determine correlations for the heat transfer coefficient. Table1 shows some of these correlations, these correlations were used in the validation process of this work.

Table 1. Nusselt numbers for turbulent flow fully developed in smooth tubes

Author	**Correlation**	**Application range**
Dittus and Boelter (1930)	$Nu = 0,023 \cdot Re^{0,8} \cdot Pr^{n}$	$10^4 \leq Re \leq 1,2 \cdot 10^5$ $0,7 \leq Pr \leq 120$ $L/D \geq 60$
Colburn (1933)	$Nu = 0,023 \cdot Re^{0,8} \cdot Pr^{\frac{1}{3}}$	$10^4 \leq Re \leq 1,2 \cdot 10^5$ $0,7 \leq Pr \leq 120$ $L/D \geq 60$
McAdams (1961)	$Nu = 0,021 \cdot Re^{0,8} \cdot Pr^{0,4}$	$10^4 \leq Re \leq 10^6$ $0.7 \leq Pr \leq 10^4$
Gnielinski (1976)	$Nu = \frac{\frac{f}{8} \cdot (Re-1000) \cdot Pr}{1+12,7 \cdot \left(\frac{f}{8}\right)^{\frac{1}{2}} \cdot \left(Pr^{\frac{2}{3}}-1\right)}$	$2300 \leq Re \leq 5 \cdot 10^6$ $0,5 \leq Pr \leq 2000$

SOURCE:Rohsenow *et al.* (1998)

3. RESULTS

3.1 Validation of the experimental method

The validation of the experimental method was carried out with the test section stationary (without rotation) and with a flow of water in the liquid state and vapor in the axial direction of the aluminum billet. This configuration was adopted in view of the existence of several works and correlations on the heat transfer by convection inside a stationary tube. The graph in Figure 5 reveals that the Nusselt number as a function of the product $Re^{0.8} \cdot Pr^{0.4}$ it develops almost linearly, a result expected and widely discussed in the specialized literature, indicating coherence of the results obtained experimentally with those provided by the literature. Although the correlation of Dittus and Boelter (1930) is valid for turbulent flows ($Re > 10^4$), the points in the Figure 5 with $Re < 10^4$ were closer that correlation than the points with $Re > 10^4$. Tube roughness effects may have affected the heat transfer coefficient, so small adjustments in the values of the exponents of the Reynolds and Prandtl numbers present in the abscissa of the graph may reduce the deviations between the experimental and theoretical results.

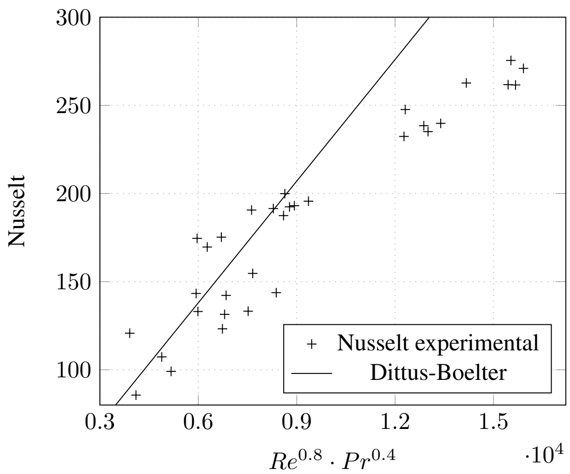

Figure 5. Nusselt number as a function of Reynolds and Prandtl

Figure 6 shows the behavior of the heat transfer coefficient as a function of the Reynolds number. As it can be observed the increase in flow causes an increase in heat exchange, this increase occurs more pronounced for water in the liquid state, however the behavior for both states is consistent with the literature.

Figure 7 show the values of the heat transfer coefficients obtained experimentally compared with values theoretically obtained from correlations available in the literature for water in the vapor and liquid state, respectively. The results obtained experimentally had good agreement with the theoretical results obtained from correlations. For water in the vapor state, 95% of the experimental points showed errors equal to or less than ± 30% in relation to the theoretical results, with a MARD of approximately 18%. For liquid water, 75% of the points had an error below or equal to ± 30%, with a MARD of 28%. It is also observed that the experimental values, for the most part, are below the theoretical values, in other words, the correlations tend to overestimate the heat transfer coefficient. This behavior occurs more prominently for the correlation of Gnielinski (1976).

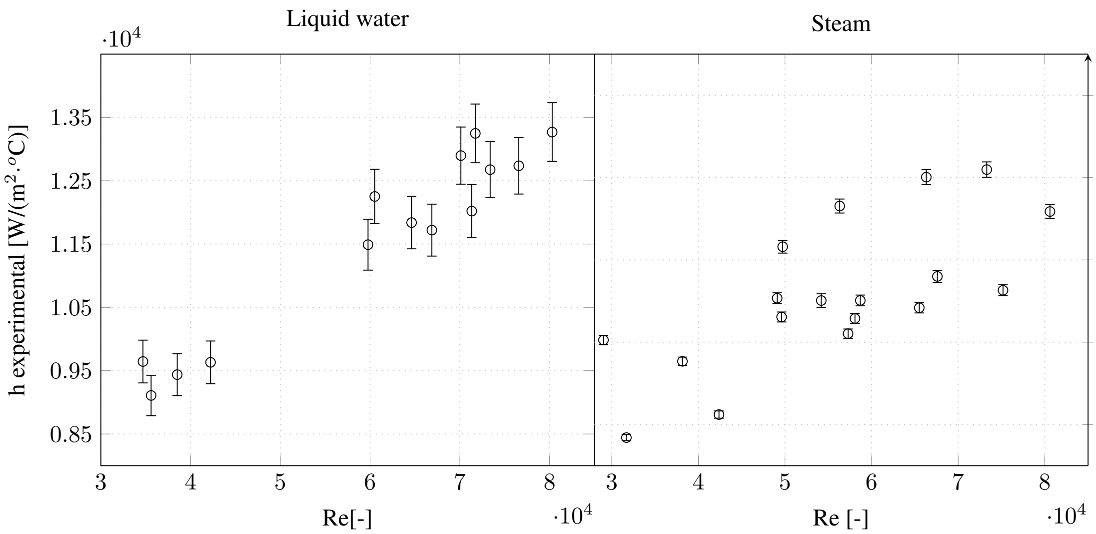

Figure 6. Convective heat transfer coefficient as a function of the axial Reynolds number

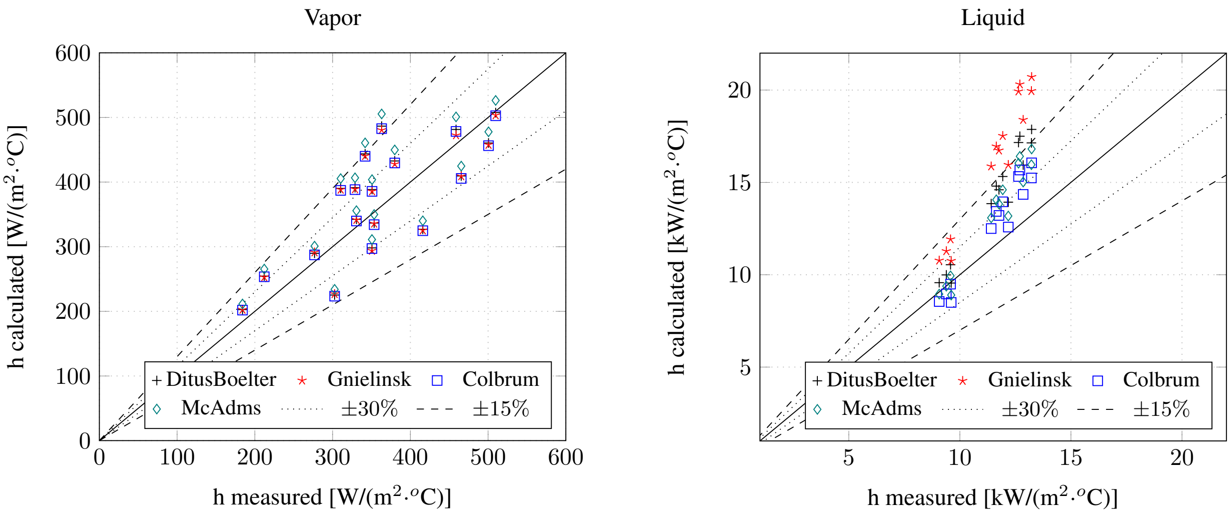

Figure 7. Comparison of experimental data with theoretical values for water (a) in the vapor state and (b) in the liquid state

Figure 8 compares the experimental values obtained for water in the vapor state with the correlations of Dittus and Boelter (1930) and Gnielinski (1976) and their respective uncertainties. Figure 9 shows the results for liquid water. It is observed that, for vapor, the two correlations present similar behaviors and with good assertiveness, presenting MARD values around 18%. On the other hand, for the liquid state, the correlations showed a significant difference with MARD around 14% for the Dittus and Boelter (1930) and 28% for the Gnielinski (1976). Another aspect in evidence in the case of liquid water is that the correlations presented values systematically higher than the values of the experimental heat transfer coefficient.

When analyzing the graphs in Figures 8 and Figure 9, it is evident that the correlations showed better agreement for water in the vapor state than in the liquid state. This fact can be explained by the effect of the roughness of the surface. A similar behavior was observed by Nunner (1955), who, when using air (Pr = 0.7) as heat tranfer fluid, roughness had little influence on heat transfer. As water vapor has a low Prandtl number (Pr = 1.0), close of the air, the same behavior could have occurred in the present work. A fact that reinforces this argument is that the correlation of Dittus and Boelter (1930) was more assertive than the correlation of Gnielinski (1976), with the second taking into account the friction factor, while the first does not. Furthermore, the internal surface of the test section billet may have become subject to what Numrich (1991) called “sand” type irregularities caused by corrosion and/or incrustation. As the test section of the present work is difficult to disassemble, it has never been cleaned since its installation in the test bench. Thus, the friction factor on the inside wall of the billet would be significantly high at the time of testing.

The fact that the correlations have shown higher values than those obtained experimentally for liquid water can be explained based on what was observed by Dipprey and Sabersky (1963) in the transition region between the "smooth" surface and the "rough" surface for high numbers of Prandtl (5.94). Under these circumstances, roughness negatively affects heat transfer. As the tests with water present a Prandtl number around 5, the lower experimental coefficients can be explained by the roughness of the billet surface being in the transition between “smooth” tube and “rough” tube. This also reinforces the hypothesis that, with use, the surface of the test section billet has acquired a “sand” type roughness.

The analysis of figures 8 and 9 shows that the uncertainty of the experimental results has the same order of magnitude as the theoretical results and that, in general, the Gnielinski (1976) correlation has greater uncertainty than the Dittus and Boelter (1930) correlation.

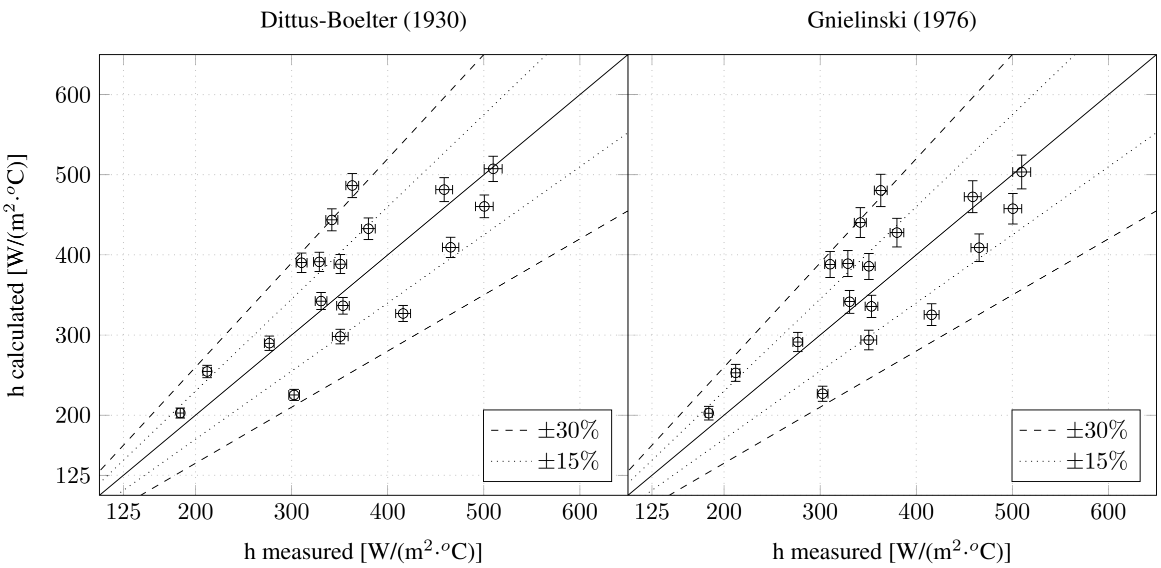

Figure 8. Comparison of uncertainties of experimental data and theoretical values for water in the vapor state

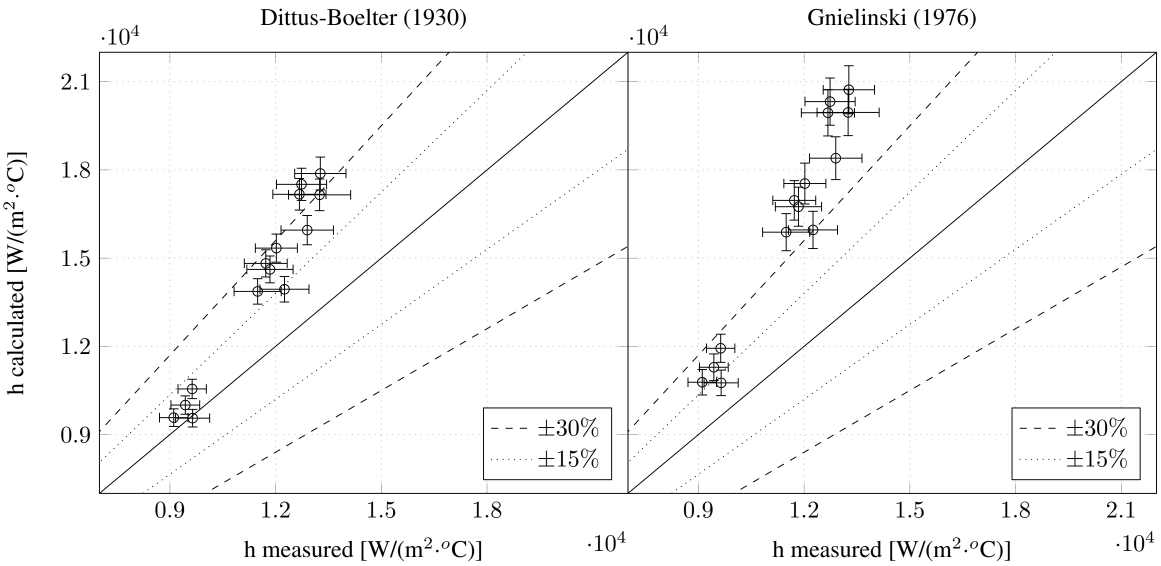

Figure 9. Comparison of experimental data uncertainties and theoretical values for liquid water

3.2 Effects of rotation on heat exchange

The effects of rotation (0, 25, 50 and 100 RPM) were studied using water only in the vapor state. Figure 10a shows the influence of rotation on the Nusselt number for a fixed flow. It can be observed that despite the Nusselt number having oscillated in the intermediate rotations in relation to the value of the stationary tube, there was a decrease in the value of the Nusselt number for the highest rotation. The trend that heat transfer deteriorated with rotation resonates with studies found in the literature. According to the studies by Reich and Beer (1989) and Reich *et al.* (1992), rotation can stabilize the flow, which would change from turbulent to laminar, thus reducing the heat transfer.

Figure 10b shows the influence of increasing vapor flow for a fixed rotation. It is observed that, despite the oscillations

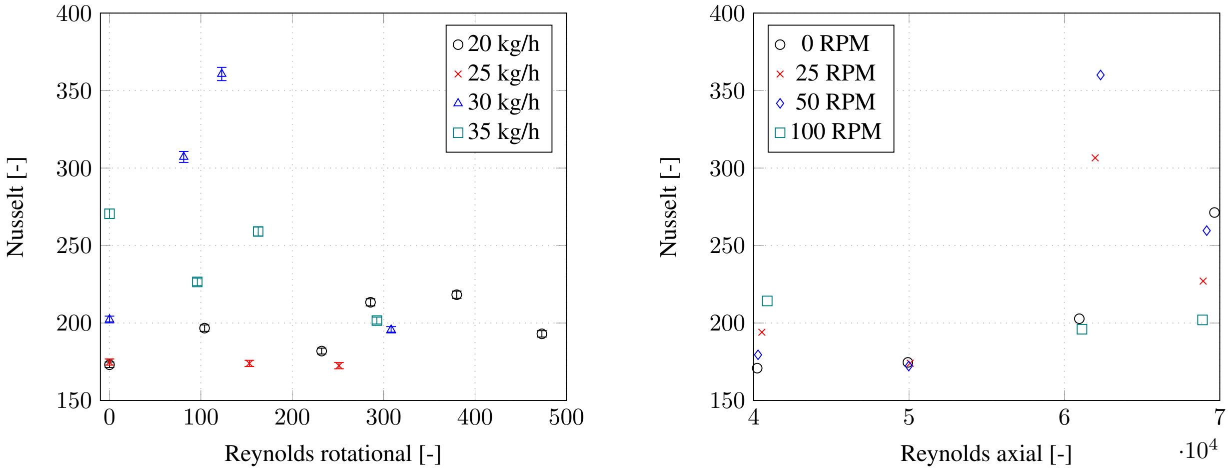

Figure 10. Influence of rotation on the Nusselt number (a) for a fixed mass flow rate (b) for a fixed rotational speed

shown in the figure, in all rotations, the Nusselt number relative to the largest axial Reynolds (higher flow) was smaller than that relative to the stationary tube, implying degradation of heat transfer.

Figure 11 shows the behavior of the theoretical Nusselt number, obtained with the correlation proposed by Seghir-Ouali *et al.* (2006), using the values obtained experimentally in the present work. It is important to emphasize that these values are outside the valid range for that correlation, so the values presented should only be used for an analysis of the trend of heat transfer coefficient behavior as a function of tube rotation and fluid flow rates in the tube. In general, the behavior of the experimental results was coherent with those obtained from the correlations. It is observed that the theoretical values showed the same oscillatory trend at intermediate rotations and with a reduction in relation to the stationary value for higher rotations. On the other hand, both the oscillation and the decrease in the Nusselt number were smoother than those observed experimentally. For some flows and within the margin of experimental uncertainties, it is observed that the rotation did not influence the heat transfer.

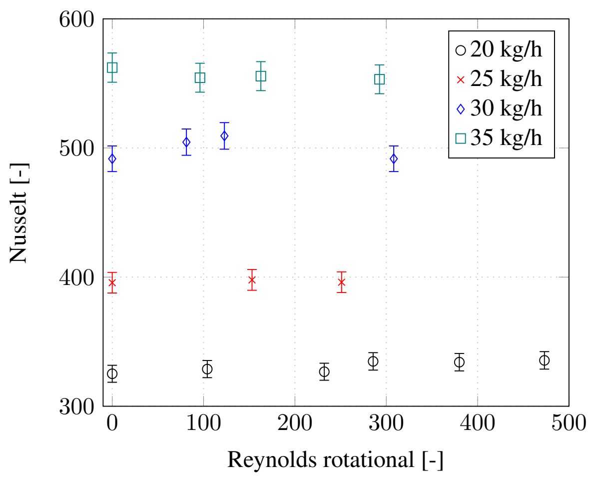

Figure 11. Theoretical Nusselt number as a function of the rotational Reynolds number for a fixed steam flow

4. CONCLUSION

Validation tests were performed for water in liquid and vapor state. 18 experimental points were obtained for steam and 14 for liquid water and a stationary tube. The experimental results of the heat transfer coefficient were compared with theoretical values calculated from correlations available in the literature. Tests with water in the vapor state showed good agreement with the literature with 95% of the points showing maximum deviations of $\pm$ 30% and with a MARD value of 14%. The tests with liquid water showed a lower agreement with the literature, with 75% of the points showing maximum deviations of $\pm$ 30% and MARD of 28%. The tests with water in the liquid state also showed a trend towards smaller experimental coefficients than those obtained theoretically. This tendency can be explained by the roughness of the surface of the test section of the experimental apparatus being in the transition region between “smooth” and “rough”. As roughness does not have much influence for lower Prandtl numbers, in the case of water in the vapor state, this trend was observed only for liquid water.

Tests to evaluate the effects of rotation on heat transfer were conducted for four different rotations, totaling 17 experimental points. In the intermediate rotation ranges, a random behavior of the Nusselt number was observed. For a greater rotation, a tendency was observed for the value of the Nusselt number to be lower in relation to that of a stationary tube. This reduction in the Nusselt number agrees with most studies available in the literature, which indicate a decrease in heat transfer with rotation. Experimental data were also compared with theoretical values, which, although outside the range of validity of the correlation, showed the same behavior profile.

Experimental tests, both with stationary and rotating tubes, presented values consistent with the literature. Thus, the methodology and experimental test bench are validated and can be applied to other situations, such as for two-phase water flows inside a stationary or rotating tube, typical of the quenching process.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Baleta, J., Qi, F., Zivic, M. and Lovrenic-Jugovic, M., 2018. "Numerical investigation of the transient spray cooling process for quenching applications". *Thermal Science*, Vol. 22, pp. 261–261.

Callister, W.D. and Retwisch, D.G., 2012. *Ciência e engenharia de materiais uma introdução*. LTC.

Colburn, A., 1933. "Heat transfer in automobile radiator of the tubular type". *Trans. AICHE*, Vol. 29, pp. 174–210.

Corrêa, T.D.M., 2019. *Estudo experimental em regime permanente para determinação do coeficiente de transferência de calor externo durante o processo de têmpera por imersão em água de tubos de aço*. Dissertação de mestrado, Universidade Federal de Minas Gerais.

Dipprey, D. and Sabersky, R., 1963. "Heat and momentum transfer in smooth and rough tubes at various prandtl numbers". *International Journal of Heat and Mass Transfer*, Vol. 6, No. 5, pp. 329 – 353.

Dittus, F. and Boelter, L., 1930. "Heat transfer in automobile radiator of the tubular type". *University of California at Berkley Publ. Eng.*, Vol. 2, pp. 443 – 461.

Gnielinski, V., 1976. "New equations for heat and mass transfer in turbulent pipe and channel flow". *Int. Chem. Eng.*, Vol. 16, No. 2, pp. 359–368.

Greif, D., Kopun, R., Kosir, N. and Zhang, D., 2017. "Numerical simulation approach for immersion quenching of aluminum and steel components". *International Journal of Automotive Engineering*, Vol. 8, pp. 45–49.

Incropera, F.P., Lavine, A.S., Bergman, T.L. and Dewitt, D.P., 2011. *Introduction to Heat Transfer*. John Wiley & Sons, 6th edition.

Kobasko, N., Aronov, M. and Powell, J., 2010. *Intensive Quenching Systems: Engineering and Design*. ASTM International. ISBN 978-0-8031-7019-3.

McAdams, L., 1961. "Transmission de la chaleur". *Dunod Paris*.

Numrich, R., 1991. "Heat transfer in rough tubes". *Chemical Engineering and Processing: Process Intensification*, Vol. 29, No. 2, pp. 117 – 119.

Nunner, I.W., 1955. *Dissertation*. Master's thesis, Tech. Univ.. Hanover.

Ramezanzadeh, H., Ramiar, A. and Yousefifard, M., 2017. "Numerical investigation into coolant liquid velocity effect on forced convection quenching process". *Applied Thermal Engineering*, Vol. 122.

Reich, G. and Beer, H., 1989. "Fluid flow and heat transfer in an axially rotating pipe—i. effect of rotation on turbulent pipe flow". *International Journal of Heat and Mass Transfer*, Vol. 32, No. 3, pp. 551–562.

Reich, G., Weigand, B. and Beer, H., 1992. "Fluid flow and heat transfer in an axially rotating pipe subjected to external convection". *International Journal of Heat and Mass Transfer*, Vol. 35, No. 7, pp. 1803–1809.

Rohsenow, W.M., Hartnett, J.P. and Cho, Y.I., 1998. *Handbook of Heat Transfer*. MCcGraw-Hill, 3rd edition.

Sakamoto, A., Okamura, K., Serizawa, Y., Yamamoto, K. and Yamamoto, K., 2016. "Water cooling technologies for steel pipe production processes". Technical report, NIPPON STEEL & SUMITOMO.

Seghir-Ouali, S., Saury, D., Harmand, S., Phillipart, O. and Laloy, D., 2006. "Convective heat transfer inside a rotating cylinder with an axial air flow". *International Journal of Thermal Sciences*, Vol. 45, No. 12, pp. 1166 – 1178.

Taylor, B.N. and Kuyatt, C.E., 1994. *Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results*. National Institute of Standards and Technology, 1st edition.

Wagner, J.H., Johnson, B.V. and Hajek, 1989. "Heat transfer in rotating passages with smooth walls and radial outward how". *The American Society of Mechanical Engineers*, Vol. 1, No. 89-GT-272.

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