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EXPERIMENTAL STUDY ON THE SINGLE-PHASE CONVECTIVE HEAT TRANSFER COEFFICIENT IN THE TUBE QUENCHING PROCESS

Túlio da Motta Corrêa
Leonardo Victor Silva Martins
Luiz Machado
Graduate Program in Mechanical Engineering, Federal Universityof Minas Gerais, Belo Horizonte, MG
tuliodamotta@gmail.com; leo.vsm@hotmail.com; luizm@demec.ufmg.br

Juan J. G. Pabon
Institute of Mechanical Engineering, Federal University of Itajubá, Itajubá, MG
langa_27@hotmail.com

Lis Nunes Soares
Vallourec Soluções Tubulares do Brasil S.A
lis.soares@vallourec.com

Abstract. *Immersion quenching is a thermal process frequently used to obtain required mechanical properties of steel products. Mathematical models allow to simulate different part geometries and cooling methods. However, a limitation of these models comes from the high uncertainty that exists in the value of the heat transfer coefficient (HTC) at the interface of the part surface and the quenchant. Knowledge and control of the HTC in various process conditions is necessary. An experimental work is carried out to study the HTC behavior during single-phase heat transfer by forced convection in the tube quenching process. Four different operating parameters are evaluated: tube surface temperature (50 - 100 °C), tube rotation speed (25 and 50 rpm), side jets water flow rate (0, 5 and 10 m³/h), and watjer temperature (25 and 35 °C). The proposed methodology has proved to be efficient in finding the HTC during the tube quenching process, presenting behaviors consistent with those indicated in the literature. The combination of side water jets and tube rotation results in a high heat removal capacity, showing significant HTC values still in the single-phase convection region.*

Keywords: *heat transfer coefficient, single-phase convection, tube quenching*

1. INTRODUCTION

The quenching process represents a fundamental step to ensure the quality of steel made products, used to obtain required mechanical properties (Buczek and Telejko, 2013; Apipe *et al.*, 2017). In order to achieve the desired specifications such as strength, toughness, sour resistance, etc., various cooling methods are developed having water as main cooling medium due its advantages in terms of cost and cooling rate (Sakamoto *et al.*, 2016). Despite being relatively well known from the operational point of view, the quenching technique involves some parameters that significantly influence the final quality of the tempered part, such as the velocity of water jets used in its cooling, time of contact of the part with water, etc. The study of cooling systems of tempered parts is expensive and time-consuming (Baleta *et al.*, 2017; Greif *et al.*, 2016; Ramezanzadeh *et al.*, 2017). Mathematical models based on finite element and/or finite volume methods allow to simulate different part geometries and to test different cooling situations and conditions of the quenching process. However, a limitation of these models comes from the high uncertainty that exists in the value of the heat transfer coefficient (HTC) between the tempered parts and the cooling fluid. The present work is focused on the study and determination of this coefficient in the convection region as a function of the following parameters of the quenching process: (1) tube rotation, (2) tube surface temperature, (3) water temperature and (4) water flow rate of jets incident on the tube.

1.1 Tube Quenching Process

The quenching technologies developed have to consider some difficulties during the tube heat treatment process. The asymmetry of cooling surface, non-uniformity of steel material characteristics in the circumferential direction and quench crack are some of these difficulties to be considered (Sakamoto *et al.*, 2016). Various cooling methods are developed to solve the above-mentioned problems in an industrial scale. The immersion cooling, shown in Fig. 1 is considered the simplest intense cooling method. In this process, a solid piece is heated at first to high temperatures, then it is immediately submerged in a subcooled liquid (Ramezanzadeh *et al.*, 2017). The method is very effective because it allows a good equanimity of the heat transfer coefficient between the tube and the water. With respect to the heat transfer between the water and the outer face of the tube, the heat transfer coefficient from the tube to the water is approximately uniform in the longitudinal direction because all parts of the tube are exposed to the water jets as the tube spins (uniformity of thermohydraulic conditions). The heat transfer mechanism between the inner face of the tube and the water flowing in its interior is more complex and will not be focus of this work.

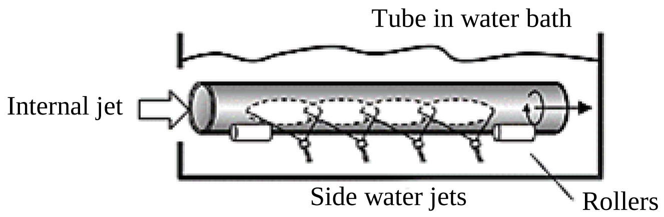

Figure 1. Immersion cooling method.

The cooling process during tube quenching is a non-linear phenomenon. According to Kobasko *et al.*, (2010), there are four modes of heat transfer that are typically encountered at the interface of the hot part and the quenchant: shock boiling, full film boiling, nucleate boiling and convective heat transfer (Fig. 2).

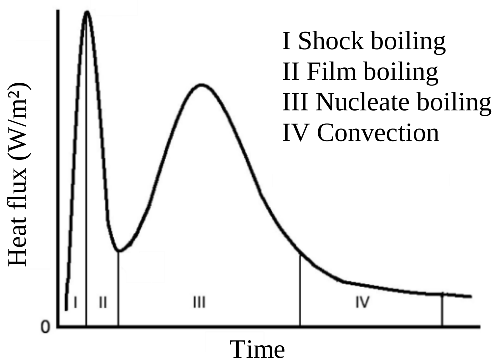

Figure 2. Heat transfer modes during tube quenching.

Shock boiling occurs over a very short period of time, immediately upon immersion of hot tube into water. In this mode, water flows along the tube surface while the tube is being immersed.

In the *film boiling*, the surface is fully covered by steam. This mode defines the critical heat flux and the tube experiences nonuniform heat transfer conditions.

The heat flux then becomes insufficient to support film boiling and the water begins contacting the surface directly. Changes in volume of the steel contribute to tube distortions during quenching process (Kobasko *et al.*, 2010). *Nucleate boiling* is characterized by a very high rate of heat extraction since the absence of the film allows the surface to be exposed to the liquid and tiny bubbles start to emerge.

After nucleate boiling ceases, the surface temperature decreases from slightly above the boiling temperature of the quenchant down to the bulk fluid temperature. *Convection cooling*, focus of this work, is characterized by a much slower cooling rate than either film boiling or the nucleate boiling. Convection cooling is also more uniform along the surface of the part and thus minimal distortion occurs during this heat transfer mode.

1.2 Convective Heat Transfer

The convection heat transfer process occurs between a surface of a solid body and a fluid, and takes place where particle movement is possible. There are two kinds of movement: natural and forced. Natural convection is due to a change of the fluid density with temperature in a gravitational field. Forced convection occurs under action of external devices, for example, water jets. The objective of this work is to determine convection coefficients for different operating conditions. In other words, we are looking for specific forms of the functions that can represent these coefficients. Using dimensionless similarity parameters, the forced convection coefficients may be correlated, by functions of the form (1): (Incropera *et al.*, 2007)

$$Nu = f(Re, Pr) \quad (1)$$

where *Nu*, *Re* and *Pr* are the Nusselt, Reynolds and Prandtl numbers defined by Eqs. (2), (3) and (4), respectively.

$$Nu = \frac{hD}{k} \tag{2}$$

$$Re = \frac{\rho V D}{\mu} \tag{3}$$

$$Pr = \frac{C_p \mu}{k} \tag{4}$$

where h is the convective heat transfer coefficient, D is the tube diameter, k is the fluid thermal conductivity, ρ is the fluid mass density, V is the fluid velocity, μ is the fluid viscosity, and C_p is the fluid specific heat.

In order to obtain these functions that could represent the phenomenon, two approaches can be used: one theoretical and the other experimental. The experimental or empirical approach (focus of this work) involves performing heat transfer measurements under controlled laboratory conditions and correlating the data in terms of appropriate dimensionless parameters.

There are two major advantages of working with these groups. The first is that the number of dimensional groups is smaller than the number of dimensional variables. Therefore, the number of tests conducted in the laboratory is significantly lower. The other advantage is that the use of dimensionless groups makes it possible to generalize the results. For example, results obtained from tests with a tube of a certain diameter may be applied to a tube of a different diameter, regardless of whether it is larger or smaller than the diameter of the tube tested in the laboratory. In addition, of course, the use of similarity allows testing on systems with better accessibility.

Figure 3 shows the heat transfer modes mentioned in the previous section that occur during the immersion quenching process of a tube in terms of the heat transfer coefficient and the metal surface temperature (Sakamoto *et al.*, 2016).

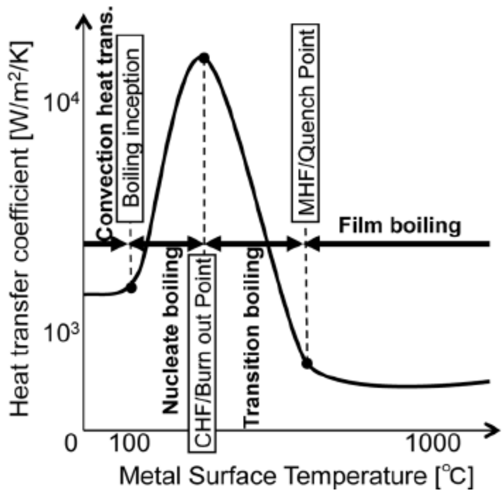

Figure 3. General HTC behavior during tube quenching process

When there is an external fluid flow over a tube, standard forced convection correlations can be used to estimate the heat transfer coefficient (Incropera *et al.*, 2007).

As the temperature of the tube surface increases, nucleate boiling will occur, causing the increase of the HTC. Due to the fact of phase change during nucleate boiling, heat transfer to the fluid can occur without influencing the fluid temperature. Large heat transfer rates can be achieved with small temperature differences. Three parameters are important when characterizing this region: the latent heat, surface tension at the liquid-vapor interface and the density difference between the two phases. Boiling heat transfer coefficients are generally much larger than those of single-phase convection because of the combined effects of the latent heat and the buoyancy-driven flow (Incropera *et al.*, 2007).

2. METHODOLOGY

2.1 Determination of the heat transfer coefficient

In order to determine the HTC, a test body shown in Fig. 4 is used. It consists of a steel tube of external diameter of 177,8 mm and length of 200 mm, which has a copper part internally filled with longitudinally disposed electric resistors. Two thermocouples are mounted in the tube in a radial direction with positions very close to the inner and outer faces. In addition to these thermocouples, another thermocouple is used to measure the temperature of the water inside the tank. Tube surface temperature is obtained by extrapolating temperatures T_1 and T_2, according to Eq. (5): (Incropera *et al.*, 2007)

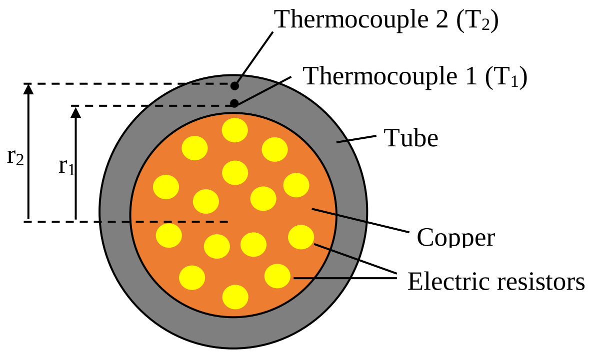

Figure 4. Cross-sectional view of test body.

$$T(r) = \frac{T_1 - T_2}{\ln(r_1/r_2)} ln\frac{r}{r_2} + T_2 \quad (5)$$

If a prescribed geometry, such as a tube, is heated electrically to maintain $T_s > T_\infty$, convection heat transfer occurs from the surface to the fluid. The convection coefficient h, which is the quantity of heat removed from a unit of a surface area per one degree (W/m² °C) and an average associated with the entire surface, can then be computed from Newton's law of cooling, Eq. (6):

$$q = h\, A_s(T_s - T_\infty) \quad (6)$$

where q is the total heat transfer rate which is equal to the electrical power, A_s is the surface area, T_s is the surface temperature and T_∞ is the fluid temperature.

The uncertainty in the calculation of the HTC depends on the uncertainty of the heat transfer rate q. Although the test body is insulated at the ends, there is a heat loss at these ends, and it can be expressive when the system operates at high power. In this present work, since the HTC study is focused on the convective single-phase heat transfer only, this heat loss can be considered minimum.

2.2 Experimental apparatus

Figure 5 shows an overview of the experimental apparatus built in the Refrigeration and Heating Laboratory (GREA) of the Federal University of Minas Gerais, consisted of three basic parts: the water tank, the test body and the hydraulic circuit. The hydraulic circuit is responsible for circulating the water through two side jets controlling the flow and keeping it within the desired temperature through an air cooler. The figure also shows the gearmotor responsible for the rotation of the test body.

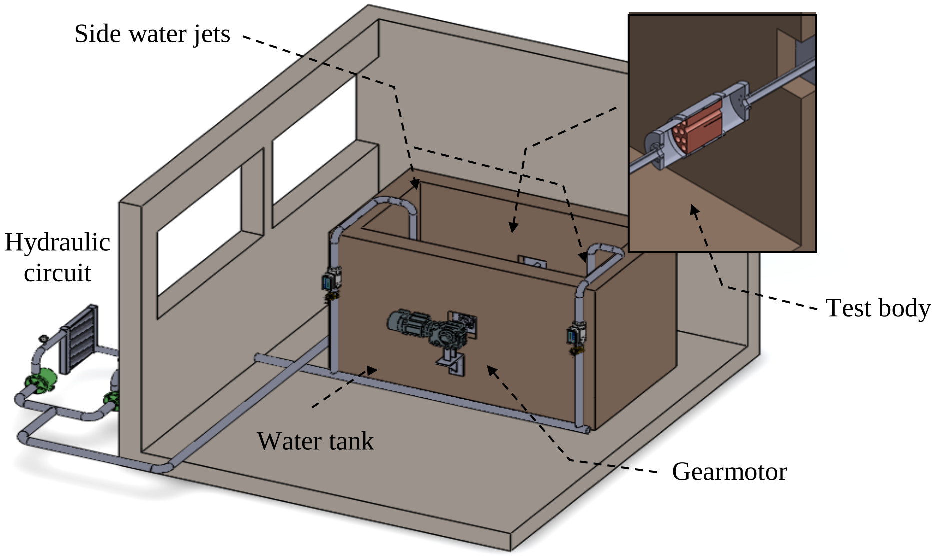

Figure 5. Experimental apparatus.

2.3 Operating conditions

In this work, several values for the HTC were determined for different thermohydraulic conditions. The tube surface and water tank temperatures, side jets water flow rates and tube rotations tested are shown in Tab. 1.

Table 1. Experimental conditions.

Tube rotation	25 and 50 rpm
Side jets water flow rate	0, 5 and 10 m^3/h (each jet)
Tube surface temperature	50 – 100 °C
Tank water temperature	25 and 35 °C

3. RESULTS AND DISCUSSION

This section presents the experimental results. A total of 56 experimental points were collected. The effects of different parameters over the heat transfer coefficient were investigated. All the heat transfer coefficient values presented here are in dimensionless form in order not to disclose confidential company information. Figures 6 and 7 present the results for the HTC in two different situations, where the water temperature of the tank is 25 and 35 °C, respectively. In both figures, letters (a) and (b) respectively show the results for 25 and 50 rpm rotations. For each of the figures, the results for the side jets water flow rates of 0, 5 and 10 m^3/h are represented.

3.1 Tube rotation effects

The effects of changing the tube rotation from 25 to 50 rpm on the heat transfer coefficient can be analyzed in Figs. 6 and 7.

As can be noticed, for the test conditions where the jets water flow rate were 0 and 5 m^3/h, the increase of the tube rotation resulted in a significant increase of the heat transfer coefficient. This behavior is expected due to the higher fluid flow velocity on the tube surface, and consequently a higher Reynolds value, one of the parameters that influence the HTC. For the conditions where the jets water flow rate was set to 10 m^3/h (maximum flow), the increase of the tube rotation seems to have caused no significant variation in the HTC.

3.2 Water jets effects

The effects of the side water jets operating with 0, 5 and 10 m^3/h on the heat transfer coefficient can be analyzed in Figs. 6 and 7.

In the tests which the tube rotation is lower, changing the jets flow rate from 0 to 5 m^3/h resulted in a higher heat removal capacity, increasing the heat transfer coefficient. When this rate is set to 10 m^3/h, the increase of the coefficient is even more significant, resulting in coefficients of at least twice the values when compared to the conditions operating with tube rotation only (water flow rate = 0 m^3/h).

At the higher tube rotation, the increase in the jets flow rate from 0 to 5 m^3/h showed little or no effect, resulting in heat transfer coefficients in the same order of magnitude. The highest HTC values were obtained when the side water jets flow rate was set to 10 m^3/h.

3.3 Cooling water temperature effects

Tests conducted in this work were performed at two different tank water temperatures: 25 and 35 °C. The effects of this temperature variation on the heat transfer coefficient can be verified in Figs. 6 and 7. It can be noticed that, the higher the tank water temperature, the higher HTC were verified.

3.4 Tube surface temperature effects

Figures 6 and 7 show the heat transfer coefficient behavior when the mean temperature of the tube surface is varied along the range of 50-100 °C.

The HTC seems to follow a trend of constant values for the test conditions in which the metal surface temperatures are lower. This is expected since the heat transfer coefficient in forced convection depends on parameters that do not involve the temperature difference between the heated surface and the fluid. This effect is better noticed in conditions operating with higher values of tube rotation and water jets flow rate.

As the mean tube surface temperature approaches the boiling temperature of the fluid, the HTC increases due to the combined effects of latent heat and the buoyancy-driven flow during the fluid phase change, as discussed in section 1.2.

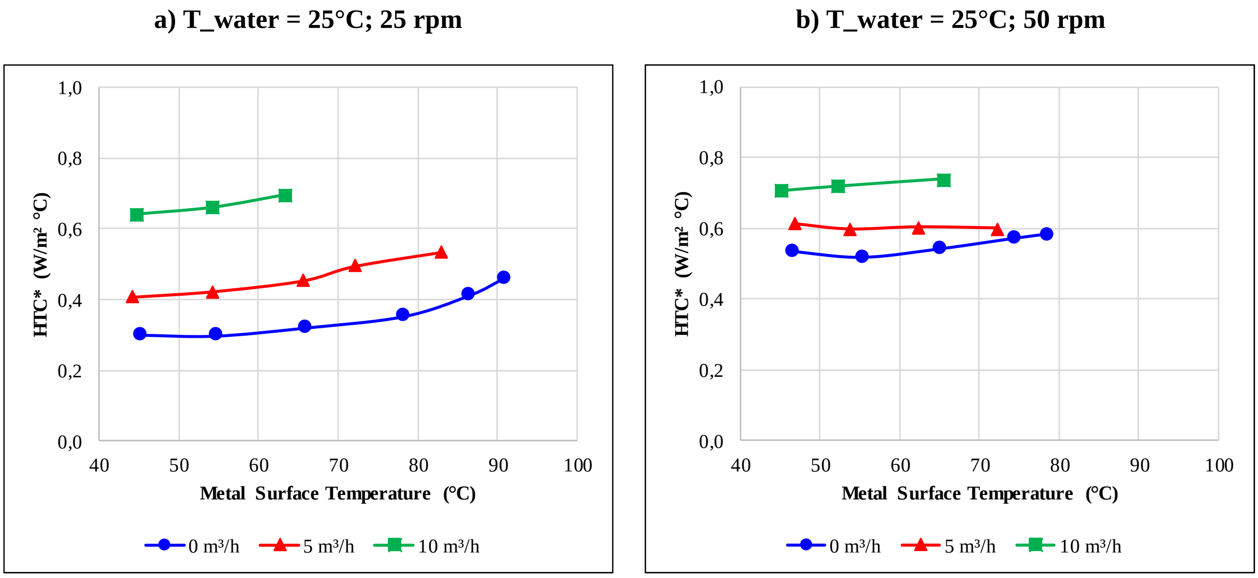

Figure 6. *Dimensionless HTC for two different rotations with cooling water at 25°C: a) 25 rpm; b) 50 rpm.

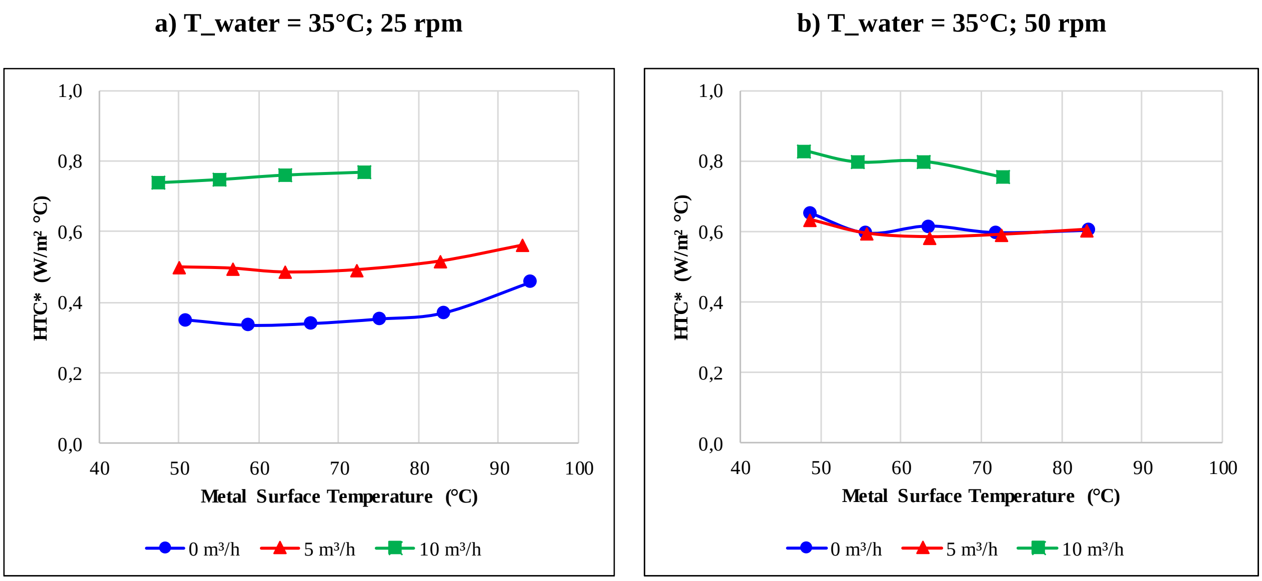

Figure 7. *Dimensionless HTC for two different rotations with cooling water at 35°C: a) 25 rpm; b) 50 rpm.

4. CONCLUSIONS

Studies related to the heat transfer coefficient in the cooling of steel parts in the quenching process is a great step towards the use of more safe and reliable computational simulations by steel technicians and scientists. The proposed methodology has proved to be efficient in finding the HTC during the tube quenching process, showing behaviors consistent with those indicated in the literature. Other conclusions that can be drawn from this work:

- Experimental results have shown that forced convection with tube rotation and side water jets is capable to remove higher heat fluxes. In other words, the higher the tube rotation and the jets water flow rate, the higher the convective heat transfer coefficient;
- The heat transfer coefficient in the conditions of forced convection may present significant values. The effects of the tube rotation on the HTC are very expressive, returning values much higher than those expected when the experimental apparatus was planned;
- The heat transfer coefficient tends to assume extremely high values in the phase change region. This makes experimental testing at this region of the quenching curve using electric resistors a difficult task since they would require a very high energy demand;
- The increase in tube rotation reduces the effect of the side water jets. At higher rotations, lower water flow rates tend to have little or no effect on the HTC;

- The tank water temperature variation from 25 to 35°C showed little influence on the HTC;
- In order to propose correlations that can provide the HTC during tube immersion quenching, more experimental points must be collected. Improvements and adaptations must be made in the experimental assembly so the HTC can be evaluated in the other regions of the tube quenching curve. Due to high heat flux values during the phase change in the boiling region, the second part of the curve of Fig. 3 should be obtained using a controlled cooling method.

5. ACKNOWLEDGEMENTS

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7. RESPONSIBILITY NOTICE

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