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Simulation of a cooling processes of wort using a immersion chiller

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Abstract. *Brewing beer involves many processes that require tight control of temperature. One of the most important occurs right after the boiling of the wort, when rapid cooling is needed to avoid contamination by bacteria and give the beer the right taste. The most common method for cooling the wort used by home-brewers is the pipe coil, since it is cheap, easy to use, and requires minimal maintenance. The pipe coil is immersed in the hot wort, which is contained in a vessel, and cold water flows through the pipe. This paper investigates the heat transfer in such a type of system to predict the water consumption and the time required to cool the wort to the desired temperature, necessary for a better design the cooling system. Three heat transfer processes were taken into account: natural convection between vessel and air, cooling due to wort evaporation, and cooling promoted by the pipe coil. In the last case, they are considered forced internal convection due to the water flow inside the pipe and natural convection between the pipe external surface and the wort. These processes are coupled and were numerically solved using a computer code written in Phyton. The wort temperature changes very slowly as compared to the temperature variation of the water flowing along the pipe. Therefore, in each time step of the simulation, the conditions of the internal flow were assumed to be steady. For the numerical simulation, the pipe can be discretized into several segments along its axial direction. When only a single segment is assumed for the entire pipe, the solution equals the approximated solution presented in the literature for internal forced flow. The result shows that, as expected, a higher flow of water leads to higher heat transfer thus decreasing the time required for cooling with the drawback of more water consumption. More results and factors (such as the cost of the system, for example) need to be obtained to optimize the system design. Finally, it is worth to observe that this type of cooling system is employed in different processes, and therefore the obtained results are expected to be helpful in applications diverse of the considered brewing beer.*
Keywords: *Cooling process, Heat Transfer, Numerical Simulation.*

1. INTRODUCTION

Some industrial processing in food and beverage need a tight temperature control to achieve the desired results. It is the case for beer, during processes of cooling beer wort. For predicting heat and mass in beer manufacturing, numerical methods and computational simulations can be employed, since they can can lead to conclusions regarding to complex phenomena in a relatively ease manner. Toapanta-Ramos *et al.* (2020) and Klembt *et al.* (2021) applied numerical methods for the wort cooling, and by Dame *et al.* (2020), for the fermentation process.

In the industrial level, the most common process to cool wort, according to Tijerino (2020), uses a plate heat exchanger. After passing through this plate, the cooling water is heated to about 90°C and can be used to other applications which need hot water.

In home beer production, on the other hand, the most common cooling system is a pipe coil, because it is cheaper, easy to use and require virtually no maintenance. However, its drawback is the high water consumption. In this system, the pipe coil is submerged into the hot wort and cold water flows inside it. The vessel which contains the wort is usually open on top, exposing the wort to the ambient air, which evaporates. Therefore, the wort looses heat on top by natural convection and evaporation, and the vessel losses heat by its lateral wall and bottom walls. In this work the cooling of the wort was numerically solved considering the heat transfer due to the pipe, convection heat transfer in the vessel lateral wall and heat transfer due to evaporation and heat convection in the wort-air interface. The heat transfer between the bottom of the vessel and air was disregarded, since it was verified that does not significantly contribute to the overall heat transfer.

2. MATHEMATICAL MODELING

The system consists of a vessel full of hot wort with an initial temperature of 100°C. The vessel has a cylindrical shape, with the top part open to the ambient air. Submerged into the hot wort there is a helical pipe, inside which flows cold water. The following characteristics are assumed for the system:

- The wort is uniform and has the properties of water.
- The vessel and coiled tube heat exchanger are made of aluminum.
- Ambient air properties remain constant during the simulation.
- The flow rate of water through the coiled tube is constant.
- Inlet water temperature is constant throughout the process.
- The system is in a quasi-static regime.

Three heat rates are computed and coupled for the complete solution:

1. The heat transfer between cold water and wort promoted by the heat exchanger;
2. The heat transfer through the lateral vessel wall;
3. The heat loss by the wort liquid surface due to convection to the air and evaporation.

The total heat transfer is the sum of the components, as given by Eq.(1)

$$q_{total} = q_{ex} + q_{wa} + q_l \tag{1}$$

q_{ex} is the heat rate of for the heat exchanger, q_{wa} is the heat rate of for the interface between wort and air and q_l is the heat rate of for in the lateral vessel wall.

The variation of the wort temperature with time was numerically calculated considering an energy balance in a control volume encompassing the system. Thus, for each time step, a new wort temperature is obtained according to Eq.(2).

$$T_{wort_{t+1}} = T_{wort_t} - \frac{q_{total}}{c_p \cdot m_{wort}} \cdot dt \tag{2}$$

Where was used an explicit scheme, being T_{wort_t} is the wort temperature in the current time step and $T_{wort_{t+1}}$ the wort temperature in the next time step, c_p is the specific heat of the wort, m_{wort} is the mass of wort, dt is the time interval between each interaction.

2.1 Heat exchanger

The heat exchanger is a coiled tube. For numerical solution it was discretized into segments along the axial direction. For each small time step, heat rates were computed with steady-state equations and the water inlet temperature of each tube segment was assumed as the outlet temperature of the previous segment, except for the entrance of the tube, where the water inlet temperature is known. They were considered convection between wort and the external surface of the tube, conduction in the aluminium tube wall and convection between the internal surface of the tube and the flowing water. Employed equations are presented as follows.

The heat rate was computed for each tube segment, and the total rate of heat removal by the heat exchanger is the summation:

$$q_{ex} = \sum_{i=0}^{n} q_{ex_i} \tag{3}$$

where the subscript i refers to the tube segment.

The q_{ex_i} are calculated according to Bergman *et al.* (2011), for a tube of any length with internal and external convection, which is given by Eq.(4).

$$q_{ex_i} = UA_i \cdot \Delta T_{lm} \tag{4}$$

ΔT_{lm} is the logarithmic difference of temperature according to Eq.(5) and UA_i is the product of the global heat transfer coefficient and reference area considered for it's obtention , which is given by according to Eq.(6).

$$\Delta T_{lm} = \frac{(T_{wort} - T_{m_{out}}) - (T_{wort} - T_{m_{in}})}{\ln\left[(T_{wort} - T_{m_{out}})/(T_{wort} - T_{m_{in}})\right]} \tag{5}$$

where T_m is the median temperature of the water flow and the subscript in refers to the inlet plane and the subscript out the outlet plane.

$$UA_i = \left[\frac{1}{U_{in_i} \cdot A_{in_i}} + \frac{1}{U_{out_i} \cdot A_{out_i}} + \frac{1}{U_c}\right]^{-1} \tag{6}$$

where U_{out_i} is the external convection coefficient,A_{out_i} is the external area of the tube section i, U_{in_i} is the internal convection coefficient, A_{in_i} is the internal area of the tube section i,and U_c is the resistance of the tube wall.

Convection coefficients and tube resistance were calculated in the subsequent sections.

The difference between the temperature difference between the two fluids in the inlet region and outlet region is given by the Eq.(7)

$$\frac{T_{wort} - T_{m_{out}}}{T_{wort} - T_{m_{in}}} = \exp\left(-\frac{UA_i}{\dot{m} \cdot c_p}\right) \tag{7}$$

where $\dot{m}$ is the water flow inside of the tube.

Isolation the outlet temperature in the equation above, the outlet water temperature for each section of the tube is calculated by the Eq.(8).

$$T_{m_{out}} = T_{wort} - (T_{wort} - T_{m_{in}}) \cdot \exp\left(-\frac{UA_i}{\dot{m} \cdot c_p}\right) \tag{8}$$

As the water pass for each section the temperature change is significant, the water proprieties such as viscosity μ, Prandt Number (Pr), heat transfer coefficient (k), thermal diffusivity α and thermal expansion β whore evaluated for each section at a temperature T_m , those propriety's whore calculated using the python library of the IAWPS (International Association for the Properties of Water and Steam). For the internal convection was at the mean temperature of the water for section give by the Eq.(9) and for the external convention was at the mean temperature of the boundary layer between the wort and the outer tube surface calculated according to the Eq.(10).

$$T_m = \frac{\bar{T}_{m_{in}} + \bar{T}_{m_{out}}}{2} \tag{9}$$

$$T_m = \frac{T_{wort} + T_{s_{out}}}{2} \tag{10}$$

where $T_{s_{out}}$ is the average temperature of the external tube surface, according by Eq.(12).

The average temperatures of the internal and external surfaces were calculated by solving the thermal circuit, for the heat exchange in which the temperatures of the internal and external surfaces were calculated according to the Eq.(11) and Eq.(12).

$$T_{s_{in}} = T_{m_{in}} + \frac{q_{ex_i}}{h_{in_i} \cdot A_{in}} \tag{11}$$

where $T_{s_{in}}$ is the average temperature of the internal tube surface.

$$T_{s_{out}} = T_{wort} - \frac{q_{ex_i}}{h_{out_i} \cdot A_{out}} \tag{12}$$

As those equation to solve the temperature are couple, the wore calculated using the method a numerical approach, using a Python algorithm combine with library's such numpy, matplotlib, and IAWPS.

2.1.1 External convection

The heat rate at the external surface of the tube was computed assuming natural convection in a horizontal tube, which is computed with the the empirical correlations proposed by Churchil (1975). The effects of the plume generated by the other coils was not considerate for the considerations of the heat transfer coefficient.

$$Ra = \frac{g\beta(T_{s_{out}} - T_{wort})D^3}{v \cdot \alpha} \tag{13}$$

where g is the gravitational acceleration and D is the characteristic length.

$$\bar{Nu} = \left\{0.6 + \frac{0.387Ra^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2 \tag{14}$$

This equation is valid for Rayleigh number to 10^12, while the values occurring in the present simulation range from $2 \cdot 10^4$ and 10^6. The heat transfer coefficient is then:

$$U = \frac{Nu \cdot k}{D} \tag{15}$$

where D is the outer diameter of the tube.

2.1.2 Internal convection

Internal convection was computed for each section of the tube, since the fluid temperature varies from the entrance to the exit. The effect of the helical format of the tube was neglected in turbulent flow because the induction of turbulence in the flow due to the shape is minimal,thus do not increasing turbulence. The flow was assumed as fully developed, since the tube is long, (L/D>=20). The internal convection coefficient was calculated similarly to the one described in the Eq.(15) but the Nusselt number conforms to the using of different correlation both for turbulent flow in Eq.(17) and for laminar flow in Eq.(19a). The Reynolds number was calculated for the internal flow to determine whether the flow is turbulent or laminar through the Eq.(16), as in Bejan (2013).

$$Re = \frac{4\dot{m}}{\mu\pi \cdot D} \tag{16}$$

D is the diameter is the internal diameter of the pipe.

Where evaluated both cases for a turbulent flow and for laminar flow so the Reynolds number was considered for with correlation to used to calculated the heat transfer coefficient. For turbulent flow the Nusselt number was calculated using the correlations made by Dittus and Boelter (1985), for the case of turbulent heating it is given by the equation (17)

$$Nu = 0.023Re^{0.8} \cdot Pr^{0.4} \tag{17}$$

In the case of laminar flow, the correlations for helical tubes recommended by Kakaç *et al.* (1987) were used, according to the Eq.(19a), where C is the diameter the coils of the heat exchange and Re_{dc} the critical Reynolds number due to the increase in turbulence by the helical geometry and μ_s is the water viscosity evaluated at the internal surface temperature, is given by the Eq.(18)

$$Re_{dc} = Re \cdot (1 + 12(D/C)^{1/2}) \tag{18}$$

$$Nu = \left[\left(3.66 + \frac{4,343}{a}\right)^3 + 1,158 \cdot \left(\frac{Re_{dc} \cdot (D/C)^{1/2}}{b}\right)^{3/2}\right]^{1/3} \cdot \left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{19a}$$

$$a = 1 + \frac{957 \cdot (D/C)}{Re_d c^2 \cdot Pr} \qquad b = 1 + \frac{0.477}{Pr} \tag{19b}$$

The convection transfer coefficient was calculated according to the Eq. (15)

2.1.3 Conduction through tube wall

The conduction was assumed as one-dimensional in the radial direction, and inside each time step the heat rate can be obtained with steady-stated equations. Therefore, conduction resistance computed with Eq.(20) could be used.

$$U_c = \frac{2\pi k \cdot L_i}{\ln(D_{out}/D_{in})} \tag{20}$$

where k is the aluminium thermal conductivity, L_i is the length of each section, D_{out} and, D_{in} are respectively the external and internal diameters of the tube.

2.2 Interface between wort and air

Two regimes of heat transfer occur at the interface between wort and air, convection and heat removal by evaporation. Convection was modeled as that occurring in the upper surface of a hot plate. For evaporation, first it was computed the diffusion of the wort water into air and second, the heat lost by the the transition from liquid to vapor. The same assumption is made by Bejan (2013) to solve problems involving natural convection and mass transfer at liquid/gas interfaces. Such an approximation solution is sufficiently accurate for the considered problem, since the contribution of this interface to the overall heat transfer is small as compared to that due to the heat exchanger.

$$q_s = q_e + q_c \tag{21}$$

In which the heat transfer by convection is given by Eq. (22):

$$q_c = \bar{h} \cdot A \cdot (T_{\text{inf}} - T_s) \tag{22}$$

where the surface area is the area of the interface between liquid and air. For this case, the Grashof number can be calculated ,accordingly Bergman *et al.* (2011) by Eq. (23)

$$Gr = \frac{g(\rho_{\text{inf}} - \rho_s) \cdot L^3}{\bar{\rho} \cdot \nu^2} \tag{23}$$

where L is the characteristic length of the geometry defined by Eq.(24) and $\bar{\rho}$ is the arithmetic mean of the densities by Eq.(25).

$$L = \frac{As}{P} \tag{24}$$

$$\bar{\rho} = \frac{\rho_{\text{inf}} + \rho_s}{2} \tag{25}$$

With the Grashof number given by Eq. (23), the Rayleigh number can be computed as Eq.(26).

$$Ra = Gr \cdot Pr \tag{26}$$

The Nusselt number is given by the correlation proposed by Lloyd and Moran (1974) for the superior surface of a hot plate:

$$Nu_c = 0.54 \cdot (Ra)^{1/4} \tag{27}$$

Therefore, the convective heat transfer coefficient is:

$$h_c = Nu_c \cdot k_{air}/L \tag{28}$$

where L is the characteristic length, and k_{air} is the heat transfer coefficient of air.

For each interaction the propriety's of air and water wore evaluated at the median boundary layer temperature. The heat transfer due to evaporation is given by Eq.(29)

$$q_e = \eta_A \cdot h_{fg} \tag{29}$$

where h_{fg} is the enthalpy of phase change of the water from liquid to vapor, and the rate of evaporation η_A is given by Eq.(30).

$$\eta_A = \bar{h_m} \cdot A_s(\rho_{a_s} - \rho_{a_{\text{inf}}}) \tag{30}$$

where the coefficient of mass transfer h_m is given by Eq. (31)

$$\bar{h_m} = Sh \cdot D_{ab}/L \tag{31}$$

where D_{ab} is the diffusion rate between water and air.

Using the heat and mass transfer correlations, we obtain that Sherwood number for a flat plate with $T_s > T_{\text{inf}}$ is given by Eq. (32), using the correlations found by Lloyd and Moran (1974).

$$Sh = 0.54 \cdot (Ra)^{1/4} \tag{32}$$

where Ra is the Rayleigh number of the transfer of mass given by Eq. (33).

$$Ra = Gr \cdot Sc \tag{33}$$

where Sc is the Schmidt number given by Eq.(34).

$$Sc = \nu_{air}/D_{AB} \tag{34}$$

where ν_{air} is the kinematic viscosity of air.

2.3 Lateral Vessel wall

It is the heat transfer between wort and air through the lateral wall of the vessel and it was calculated for each time step using the global heat transfer coefficient, U:

$$q_l = U \cdot A \cdot (T_{\inf} - T_{wort}) \tag{35}$$

Once again Eq. (26) was employed to determined the global heat transfer coefficient. However the thermal resistances were calculated using convection over a flat wall for both internal and external convection and the thermal resistance due the conduction in radial direction.

2.3.1 Internal convection

Internal convection was modeled as the natural convection of wort over a flat wall. The properties of the wort were evaluated as that of water at the average temperature of the boundary layer, the average temperature between the inner wall and the wort. According to the correlations found by Churchill and Chu (1975) it is possible to calculate Nu for a Rayleigh < 10^9, calculated according to Eq.(26), with characteristic length equal to the height of the vessel, thus Nu was determined by the Eq.(36).

$$Nu = 0.68 + \frac{0.67Ra^{1/4}}{[1 + (0.492/Pr)^{9/16}]^{4/9}} \tag{36}$$

The overall heat transfer coefficient for internal convection was calculated analogously to the Eq. (15), where the surface area is the lateral area of the container.

2.3.2 External convection

External convection was modeled as the natural convection of ambient air on the outer wall of the container. The properties of the air were evaluated at the mean temperature of the boundary layer, the temperature between the air and the outer wall of the container and was calculated according to the Eq.(26), with characteristic length equal to the height of the wall. The Nusselt number was calculated according to the Eq.(36), and the global heat transfer coefficient analogous to Eq.(15), where the surface area is the lateral area of the container.

3. Results and Discussions

The results show the influence of water flow parameters and their effect on water consumption and the cooling time up to a temperature of 40°C. Also, it is presented a comparison between results for the heat exchanger obtained directly from correlations presented in literature with results in which the tube of the heat exchanger is discretized.

Table 1 presents unchanged parameter.

Table 1 : General simulation parameters.

Parameter	Value
Wort Mass	20 kg
Vessel size	280 Ø mm x 230 mm
Ambient Temperature	25°C
Ambient Relative Humidity	60%
Water Inlet Temperature	25°C
Heat Exchange internal diameter	10 mm
Heat Exchange external diameter	12 mm
Heat Exchange length	10 m

The water flow was simulated in the range from 25 g/s to 100 g/s, and a Reynolds number between 12000 and 44000. Fig.1 shows the heat rate in function of time by the vessel wall, interface between wort and air and heat exchanger, as well as wort and heat exchanger outlet temperatures in function of time.

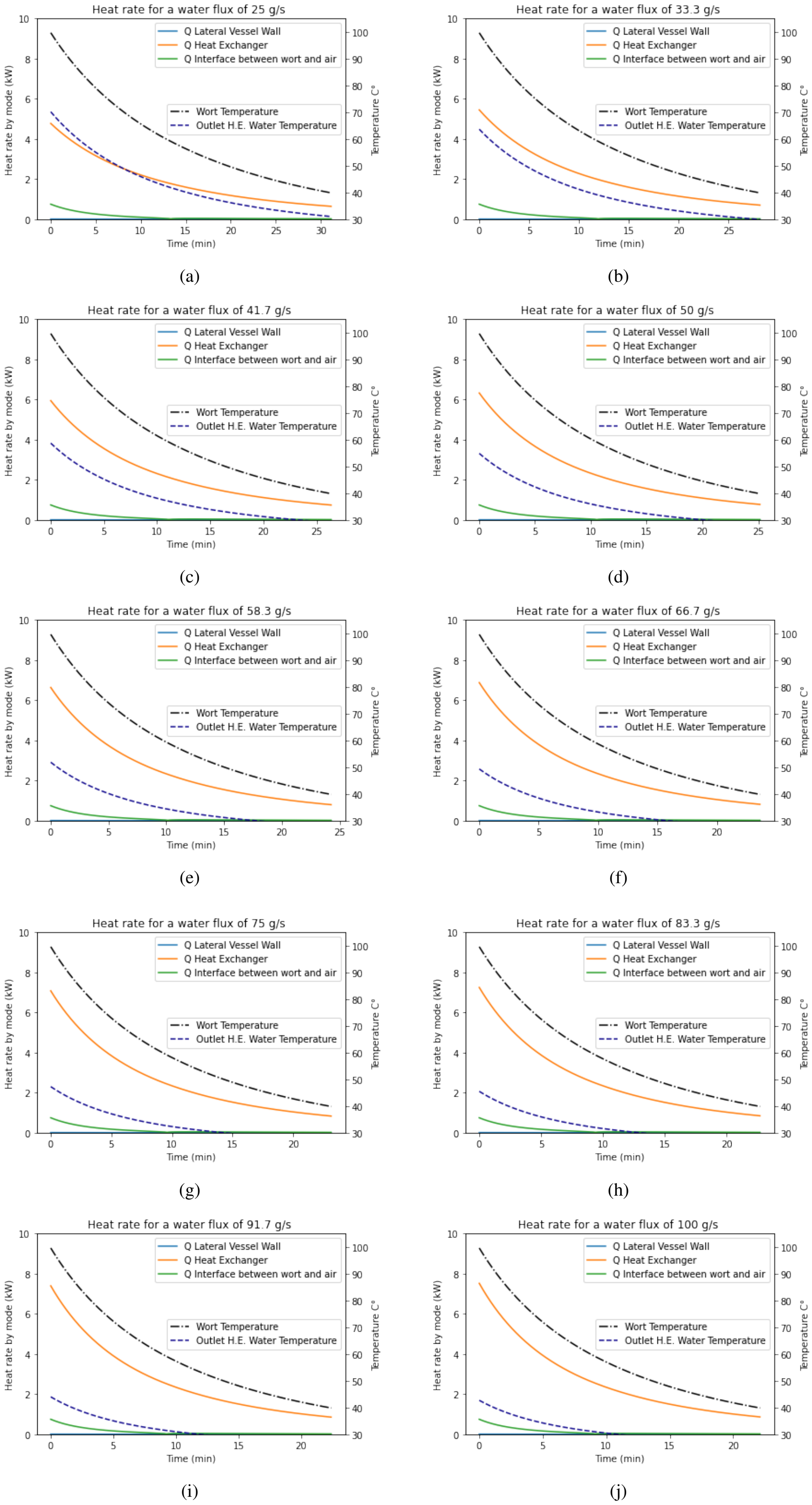

Figure 1. Time dependent heat rate in function of time by the vessel wall, interface between wort and air and heat exchanger, and wort and heat exchanger outlet temperatures for water flows from 25 g/s to 100 g/s

Fig. 2 shows the time necessary to coll the wort until the temperature of 40^o C in function of mass water flow.

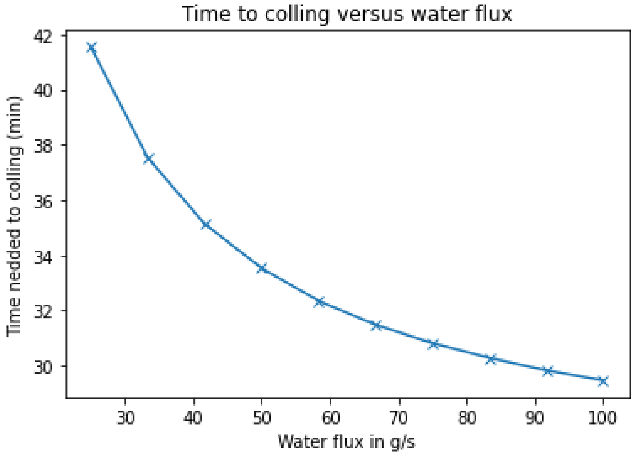

Figure 2. Time necessary to coll the wort until the temperature of 40^o C in function of mass water flow.

The greatest contributor to the cooling process is the heat exchanger, second is the interface between wort and air, especially due to evaporation. The heat transfer between vessel walls and air is less significant.

As expected, the cooling time is correlated with the flow of water in the heat exchanger. However, it does not decrease linearly with the increase of water flow. After 25 minutes of cooling time, increase in water flow has small influence on the cooling time decrease.

Fig.3 shows the effect of the heat exchanger tube. Three cases can be seen. In the first case, the tube is not discretized and the results are equivalent to that obtained analytically using correlations found in the literature Bergman *et al.* (2011). In the others two cases, the tube was divided into 10 and 100 pieces. The parameters used in these simulations are the same as those presented previously, with a water flow of 25 g/s. The results were normalized based on that for a single tube.

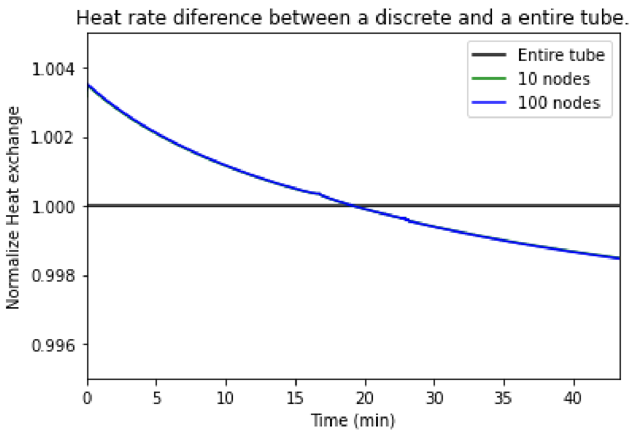

Figure 3. Discritizaction impact of the tube in the heat transfer

With 10 divisions, the results can be considered numerically converged, almost perfectly matching with those for 100 divisions. Regarding the analytical solution, difference was minimal, as showed in Fig.3. When compared to the more precise discrete cases, the maximum difference was 0.4%. Thus, the simplest approach which considers only a single tube can be used without entails significant errors. Nevertheless, in other circumstances the discretization of the tube can be desirable, for example when temperature gradients are stronger.

Some assumptions to simplify the solution could result in errors. One possible source of error is that the convective heat transfer coefficient at the external tube wall was assumed equal to that for natural convection in a horizontal tube. However, this is not the case. Since the tube is coiled, a plume which forms in tube segments located at the upper region of the vessel may have an influence on segments closer to the bottom. Therefore, in order to evaluate how significant can be the effect of such assumptions on the overall results, measurements of temperature of water contained in the vessel were taken at different times of the cooling process to be compared with the simulated results. Water was used in place of the wort because in the simulations properties of wort was assumed as the same of the water.

The mass of the water in the vessel was 15 kg and the water flow through the tube was 40g/s. Outers conditions during the measurements were close to that presented in Table 1. Comparison between the measured temperature and that obtained with simulation are shown in fig.4.

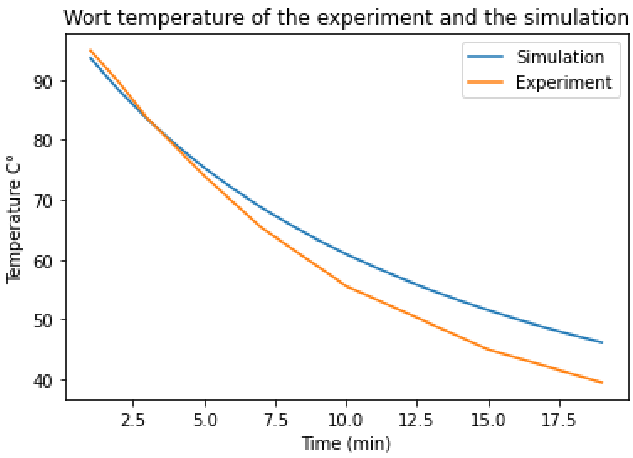

Figure 4. Wort temperature measured and of the simulation

The averaged temperature difference between simulation and measurements was 3.85 ^{o} C. This difference presented a growth with cooling time, since initial conditions adopted for measurements and simulation was the same. Thus the maximum temperature difference occurred at 19 minutes, which was the time of the last measurement, where the simulated was about 25% higher then the measured temperature. The cooling was considerably fester in reality than in the simulation. In spite of these quantitative differences, the simulation is still useful to evaluate the cooling process and the relation between water consumption and time of cooling.

4. Conclusions

The use of numerical methods for the simulation of cooling processes was verified as very useful, since it can be helpful to plain the use of resources in a more effective way. This can be applied to the food and beverage industry, where consumption of water and energy is high.

In this specific case of home brewer process the simulation provides an approximated result between water flow and cooling time, which can be used to improve the process, leading to save money and preserve water, an important natural resource.

The results obtained in this study allowed to determine the order of importance of different cooling processes involved in the overall cooling. The higher heat rate is due to the coiled tube, followed by the heat transfer at the interface between wort and air, where evaporation of wort contributes to the heat transfer. It was also verified that the heat transfer through vessel walls is not very significant.

Simple procedures presented in text books for the solution of the internal flow in the tube were verified to be very accurate in comparison with the accurate solution for the internal flow numerically obtained by dividing the tube in small segments.

A comparison between measured and simulated temperatures, which depend on the overall heat transfer show that the simulated results present significant differences in relation to the measurements. The simulation predicts slower cooling in relation to reality. Thus the current algorithm cannot be used to obtain accurate quantitative results. However, it can be employed to identify trends which can be helpful to plain more efficient processes. Further investigations must be accomplished in order to identify the sources of the differences between measurements and differences.

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