THERMAL ANALYSIS OF DIRECT COOLING SYSTEMS IN CYLINDROCONICAL FERMENTERS

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Abstract: *Alcoholic fermentation, brewery industry's central process, is a process that liberates a great amount of heat. Therefore, the fermentation containers should be equipped with cooling installations for correct temperature control. The present research aims to analyze the heat exchange in cylindroconical fermenters endowed with a half-pipe coil direct cooling system. To achieve this objective, the elaboration of a safe calculation route based on equations and experiences found in renowned references was necessary. The validation of the results was accomplished from the values obtained through the calculation program now used in one of the largest supplying companies of this kind of equipment for the brewer market, Dedini Indústrias de Base. It was verified that the flow of ammonia for the cooling system obtained by the itinerary introduced in the present article was larger than the one calculated in the program, and it can be concluded that the differences and cooling difficulties found in similar equipments supplied to different customers can have origin in the amount of ammonia used in the cooling system. The values for the overall heat transfer coefficient do not depend on the calculation itinerary followed, because there is a maximum variation of 3.5% in the results for the calculation of the coefficient. The same is verified for the mass flows of requested ammonia, where this variation is still smaller (about 3.0%).*

Keywords: *Fermenters. Cooling. Half-Pipe Coils.*

1. INTRODUCTION

In the production of quality beers, the vertical cylindroconical tanks with direct expansion of cooling fluid, broadly accepted during the last 40 years, were object of countless studies seeking their improvement. One of these studies refers to the dimensioning criterion of the cooling system to reduce the time requested by the process (Unterstein, 2006).

Dedini Indústrias de Base, a large company in the sector of consumer goods located in Piracicaba-SP (Brazil), supplied two customers, with identical processes, similar tanks that presented different cooling times. This fact stimulates the development of a calculation itinerary to allow confronting the results obtained by the standard software of the company, with two important points for analysis: the overall heat transfer coefficient and mass flow of cooling fluid.

The main objective of this paper is to develop and to standardize a safe calculation itinerary for the heat transfer surface design of cylindroconical fermenters endowed with direct cooling system through half-pipe coil and to confront the results with those obtained through the calculation program now used in one of the largest supplying companies of this equipment type for the brewer market.

2. FERMENTATION AND MATURATION

Yeast can be defined as an aqueous solution of sugars, the food for the fungus which accomplishes fermentation, brewer's industry central process, creating alcohol.

The fermentation stage usually happens in vertical cylindroconical tanks built in stainless steel (Fig. 1).

Figure 1: Cylindroconical fermenters still in assembly phase (without thermal isolation). DEDINI (2008)

The capacity of the cylindroconical vessels (CCV's) ranges from 500 to 13000 hL (50 to 1300 m³), and their height can reach 22 meters (the composition of the fermentation products is affected by the height of the yeast) and their diameter varies from 2 to 8 meters. The empty space above the yeast varies from 8 to 25% of the total volume (due to the enormous foam volume generated by the emission of CO_2). The cone shaped lower part has an angle of 60 to 75° and due to strong convection of the yeast in fermentation, hardly any differences exist in temperature, pH, extract reduction and number of ferment cells suspended during the main fermentation (Cervesia, 2003).

At the end of fermentation there is a great amount of microorganisms and undesirable substances mixed to the beer. In order to separate them, the maturation process is performed, where the beer rests at a zero degrees (or less), during a period from 15 to 60 days.

It is calculated that during fermentation 586.6 kJ (140 kcal or 0.16 kWh, Tg) are produced per kg of extract (E). In reality only about 2/3 of extract is fermented (real attenuation of 65% in extract reduction).

The greatest cooling requirement occurs in most cases during cooling down of the beer in 24 to 48 hours after primary fermentation.

To remove the heat generated by fermentation, various factors must be analyzed, some with particular interest: the cooling agent employed, the arrangement of the CCV cooling, the cooling zones and the heat insulation of the CCV.

The indicator for the state of beer maturation is diacetyl removal. It can be assumed that if almost all the diacetyl has been removed the other green beer aromas will also have disappeared.

It is a fact that all fermentation processes proceed faster at higher temperatures. Thus if the pitching is performed at 8 ºC and the temperature allowed to rise to 12 to 14 ºC (Fig. 2), much more diacetyl is formed but it is also more rapidly and more completely removed. Only after diacetyl removal, the beer is cooled to lagering temperature (KUNZE, 1999).

This is the process of the fermenter vessel in analysis, however with some differences as for the values of time and cooling temperature.

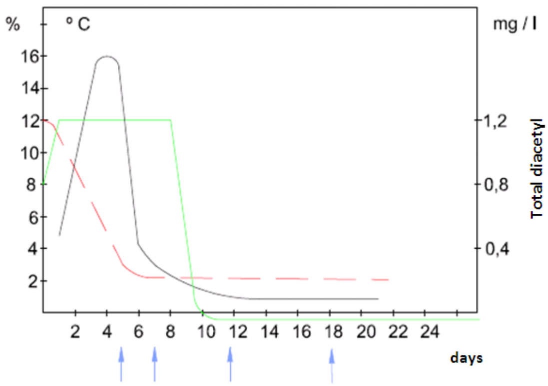

Figure 2: Warm fermentation without pressure – cold maturation (Kunze, 1999)

3. CYLINDROCONICAL FERMENTER VESSELS (CCV's)

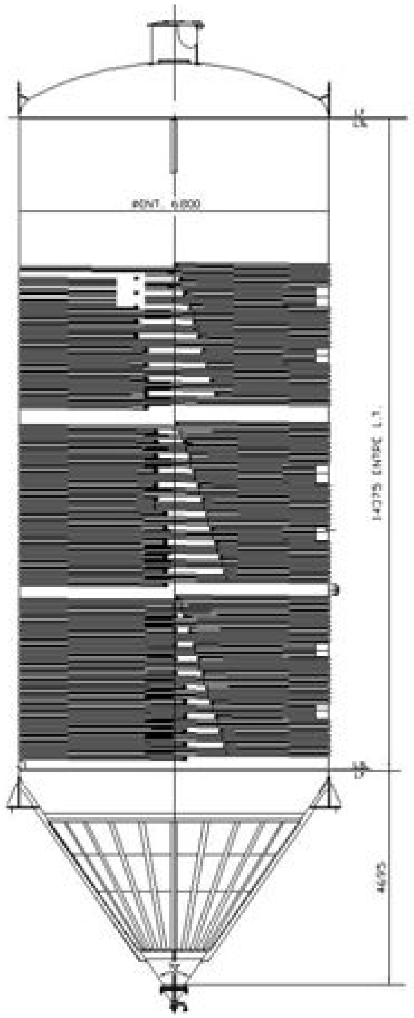

Figure 3: Cylindroconical fermenter

Cylindroconical vessels are built with a cylindrical upper part and a cone shaped lower part. As a result of this shape the yeast collects at the bottom and so can be easily and completely removed. Emptying and cleaning is also made easier.

In the fermenter studied (Fig. 3), the arrangement of the cooling coils is distributed in areas with several entrances and exits of the cooling agent.

The actual fermenter tanks are designed to direct cooling, because it has several advantages over indirect cooling (with glycol): smaller pumps are required; it is possible to work with compressors at higher temperatures; the glycol stage is unnecessary; and others.

In the case of direct evaporation, the liquid ammonia is introduced into a distribution pipe device from above and evaporates while it passes downwards and is led away. The cooling distribution pipes may be arranged horizontally or vertically. In the case of horizontal distribution (Fig. 3) there are 4 to 6 pipe coils in a cooling zone assembled into one section (train). Many distribution pipe units contain 12 to 15 entries of NH_3 per m².

The temperature is not uniformly distributed inside a CCV. In the intensive fermentation phase considerable movement occurs – especially as a result of CO_2 evolution. Convection (resulting in position exchange) also occurs in the tank whereby the colder beer sinks downwards while the warmer beer flows upwards, due the density difference (Fig. 4).

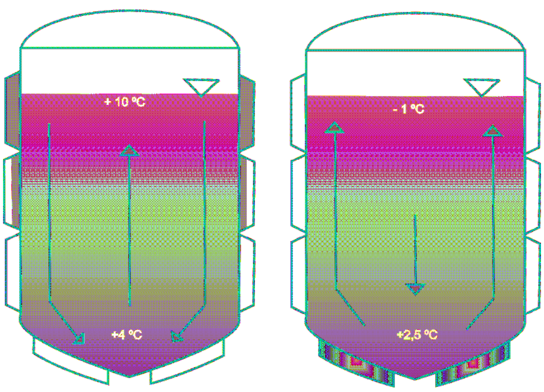

Figure 4: On the left, convection inside a CCV;
on the right, cold storage of the beer (Kunze, 1999)

In general, the beer is most dense at about +2.5 °C. Beers with higher extract content are most dense at about +1.0 °C; beers with lower content at +3.0 °C.

For the cooling down and cold lagering of the beer to 0 to -2 °C it is essential that CCV has cone cooling, otherwise it is impossible to cool this region to these temperatures (Fig. 3 and 4).

4. HEAT TRANSFER IN CYLINDROCONICAL FERMENTERS

Jacketing a process vessel provides excellent heat transfer in terms of efficiency, control, and product quality. A half-pipe coil jacket (shown in Fig. 3 and 5) consists of a continuous channel welded to the vessel wall.

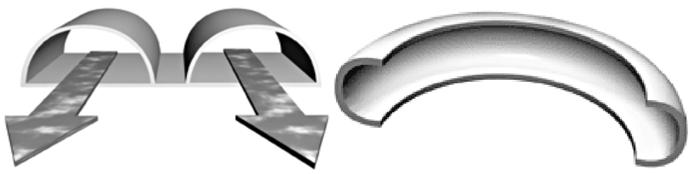

Figure 5: On the left schematic representation of the flow through the half-pipe coil; on the right, calendered profile of a half-pipe coil

Because there are no limitations to the number and location of inlet and outlet connections, the half-pipe coil jacket can be divided into multiple zones (as shown in Fig. 3) for maximum flexibility and efficiency. Multiple zoning reduces the pressure drop of the heat-transfer medium in the jacket (McKetta, 1992).

Typical flow rates in a half-pipe coil are in the range of 0.77 to 1.52 m/s. Heat transfer coefficients can be calculated by assuming various velocities in this range (McKetta, 1992). For the actual calculation the average velocity in the appropriate equations is used.

As the curvature ratio d_i/D increases, where d_i is the tube's inner diameter (m) and D the average diameter of coil curvature (m), the change-over from laminar to turbulent flow shifts towards higher *Reynolds* number than those in a straight tube (VDI, 1993).

Critical *Reynolds* number for flow through a coil or spiral tube is given by Eq. (1) - (Schmidt, 1967):

$$\mathrm{Re}_{crit} = 2300 \cdot \left[1 + 8{,}6 \cdot \left(\frac{d_i}{D} \right)^{0{,}45} \right] \quad (1)$$

The average diameter of a spiral, D, with n turns and a pitch h (m) formed from a tube of length l (m), for very large fermenter tanks, can be the outer diameter of the tank (d_{ro}, m), because the tube is not considerably bent and the pitch (h) is not large.

A transition zone between Re_{crit} and $N_{Re} = 2.2 \cdot 10^4$ in the *Nusselt* number *Nu* curve was discovered (Schmidt, 1967). The measured values were determined on air ($N_{Pr} = 0.7$) and water ($2 < N_{Pr} < 5$). They conform with a deviation of ± 15% to the Eq. (2) if $N_{Re} > 2.2 \cdot 10^4$ (Gnielinski ,1986):

$$Nu = \frac{\xi / 8 \cdot N_{\mathrm{Re}} \cdot N_{\mathrm{Pr}}}{1 + 12{,}7 \cdot \sqrt{\xi / 8} \cdot \left(N_{\mathrm{Pr}}^{2/3} - 1 \right)} \cdot \left(\frac{N_{\mathrm{Pr}}}{N_{\mathrm{Pr}\,w}} \right)^{0{,}14} \quad (2)$$

where

$$\xi = \frac{0{,}3164}{N_{\mathrm{Re}}^{0{,}25}} + 0{,}03 \cdot \left(\frac{d}{D} \right)^{0{,}5} \quad (3)$$

The *Reynolds* and *Prandtl* numbers are given by Eq. (4) and (5), respectively:

$$N_{\mathrm{Re}} = \frac{V \cdot L \cdot \rho}{\mu} \quad (4)$$

$$N_{\mathrm{Pr}} = \frac{c_p \cdot \mu}{k} \quad (5)$$

where V is the fluid average velocity (m/s), L is the channel diameter (characteristic length, m), ρ and μ the density (kg/m³) and the viscosity (kg/m.s), respectively, and c_p is the fluid specific heat to the constant pressure (J/kg.K); $\partial_m = (\partial_i + \partial_o)/2$ is the mean fluid temperature (ºC) to which the physical properties are referred.

The properties at the wall temperature ∂_M apply for the *Prandtl* number N_{Prw}.

The equations submitted before for spiral tubes (Eq. 1 and 2) can be made to apply to coiled welded semicircular tubes by substituting the thermal diameter $d_{th}=(\pi/2)d_i$ for the tube's inner diameter (m) in the expressions for the *Reynolds* and *Nusselt* numbers. The diameter ratio d_i/D in the equation for the critical *Reynolds* number should be replaced by $d_s/2d_{ro}$, where d_S is the inner diameter (m) of the semicircular tubes (Stein and Schmidt, 1986).

Eq. (2) and (3) for spiral tubes can be made to apply to coiled welded semicircular tubes by substituting the thermal diameter $d_{th}=(\pi/2)d_i$ for the tube's inner diameter in the expressions for the *Reynolds* and *Nusselt* numbers. The diameter ratio d_i/D for the critical *Reynolds* number should be replaced by $d_s/2d_{ro}$, where d_S is the inner diameter of the semicircular tubes (m).

To compute the total heat flow of the tank, all that is needed is a simple addition, following Eq. (6):

$$Q = Qa + Q_g + Q_V \tag{6}$$

where Q is total heat flow (W), Qa is heat flow to be transferred (W), Qg is fermentation heat flow (W) and Q_V is outside heat flow (W)

For the current fermenter the thermal transmission by conduction (thickness too small in comparison to tank dimensions) and by radiation (Q_V) are disregarded, the latter because of total thermal isolation. These values are very small when compared with the heat transfered by convection.

The heat transfer in process vessels follows Eq. (7), for the circumstances of stationary heat transfer and of plane heat exchange between the surfaces.

$$Q = U \cdot A \cdot \Delta\partial_{LM} \tag{7}$$

where ∂_{LM} is the mean logarithmic temperature differential (ºC).

The overall heat transfer coefficient, U (W/m².K), is calculated by Eq. (8):

$$\frac{1}{U} = \frac{1}{h_i} + ff_i + \frac{x}{k} + ff_j + \frac{1}{h_j} \tag{8}$$

where h_i and h_j are the heat transfer coefficients inside the vessel and the half-pipe coil (W/m².K), respectively.

For the jacket or internal coils, an appropriate ff_j (m².K/W) can be selected of sources as of Tubular Exchangers Manufacturers Association (TEMA). For the ff_i, (m².K/W) the selection of the appropriate value is much more difficult and it is usually based on past experiences in similar processes (Swarbrick and Boylan; 2002). For beer, the adopted value is that found for the water due to the similar characteristics for heat transfer calculations.

The mean logarithmic temperature differential ∂_{LM} (º C) is given by Eq. (9):

$$\Delta\partial_{LM} = \frac{(\partial_{wi} - \partial_i) - (\partial_{wo} - \partial_o)}{\ln\left(\dfrac{\partial_{wi} - \partial_i}{\partial_{wo} - \partial_o}\right)} \tag{9}$$

where ∂_i and ∂_o are the inlet and outlet temperature of the cooling medium (ºC) and ∂_{wi} e ∂_{wo} are the starting and the final temperatures of the medium (ºC).

For the wet part (yeast / beer), room temperature will be the temperature of the cooling medium (the temperature of - 6 ºC of the NH_3 refrigerant).

The heat transfer area (A, m²) is computed by Eq. (10):

$$A = d_{ro} \cdot h \cdot z \cdot n \cdot x \cdot t_k \tag{10}$$

where h is pitch of the coils (m), z is number of cooling zones, n is average number of turns per duct, x is number of ducts per cooling zone and t_k is pitch of the cooling zones (m)

The average dimensioning of the convection heat transmission coefficient for the laminar flow range of $Ra = 10^{-1}$ to $Ra = 10^{12}$ (the product $Gr.\ N_{Pr}$ is also referred to as *Rayleigh* number *Ra*), is defined by Eq. (11):

$$Nu_C = \left\{ 0{,}825 + \frac{0{,}387 \cdot (Ra)^{1/6}}{\left[1 + \left(\frac{0{,}492}{N_{\Pr c}} \right)^{9/16} \right]^{8/27}} \right\}^2 \qquad 11$$

where N_{PrC} is the beer *Prandtl* number (adimensional) and *Ra* is the *Rayleigh* number (adimensional).

The *Grashof* number is defined by Eq. (12):

$$Gr = \frac{g \cdot \beta' \cdot L^3 \cdot \Delta T \cdot \rho_c^{\ 2}}{\mu_c^{\ 2}} \qquad 12$$

where *g* is the gravity acceleration (m/s²), ΔT the difference between the hot fluid (1/ºC) and the room temperature, ρ_c the fluid density (kg/m³), μ_c the fluid dynamic viscosity (kg/m.s) and β' the thermal coefficient of expansion (1/ºC).

The theoretical NH_3 (m_{NH3}, kg/h) quantity is computed from the heat flow to be dissipated, using Eq. (13):

$$mNH_3 = \frac{Q}{h_{lv}} \qquad 13$$

where h_{lv} is the evaporation enthalpy of ammonia (kJ/kg).

The circulating NH_3 ($m_{NH3.uml}$, kg/h) quantity is determined from empirical data. This value can be estimated from Eq. (14) - (Gross, 1998):

$$mNH_3uml = 4 \cdot mNH_3 \qquad 14$$

The velocity (V, m/s) inside the cooling coil can be computed from the circulating NH_3, using Eq. (15):

$$V = \frac{mNH_3uml \cdot va}{z \cdot x \cdot A_S} \qquad 15$$

The pressure loss (ΔP, bar), which is of major importance in the cooling zones, is computed using Eq. (16):

$$\Delta P = \left[\varsigma_{ui} + \varsigma_{uo} + \left(\varsigma \cdot \frac{L_S}{d_h} \right) \right] \cdot \frac{V^2 \cdot \rho}{2 \cdot 10^5} \qquad 16$$

The coil drag coefficient, ς (adimensional), is calculated from Eq. (17) - (VDI, 1993):

$$\varsigma = 0{,}0054 + \frac{0{,}3964}{(N_{\mathrm{Re}})^{0{,}3}} \qquad 17$$

The drag coefficient at the entry into and exit from the cooling profile, ς_{ui} and ς_{uo}, are given by *Crane* (1999), and, in a generic way, they are equal to 0.5 and 1.0 (adimensional), respectively.

The software used is standard in Dedini Industrias de Base (Dedini-Schmidding) and is certifiedly effective in cylindroconical fermenter calculations.

The methods used are in accordance to items 7 and 8, which describe a safety itinerary based on previous experiences and traditional and empiric equations that will be used as base for the thermal dimensioning of the actual cylindroconical fermenter.

5. RESULTS

The calculations regarding the thermal calculation of the cylindroconical fermenter follow three different itineraries (Fig. 6 and Tab. 1).

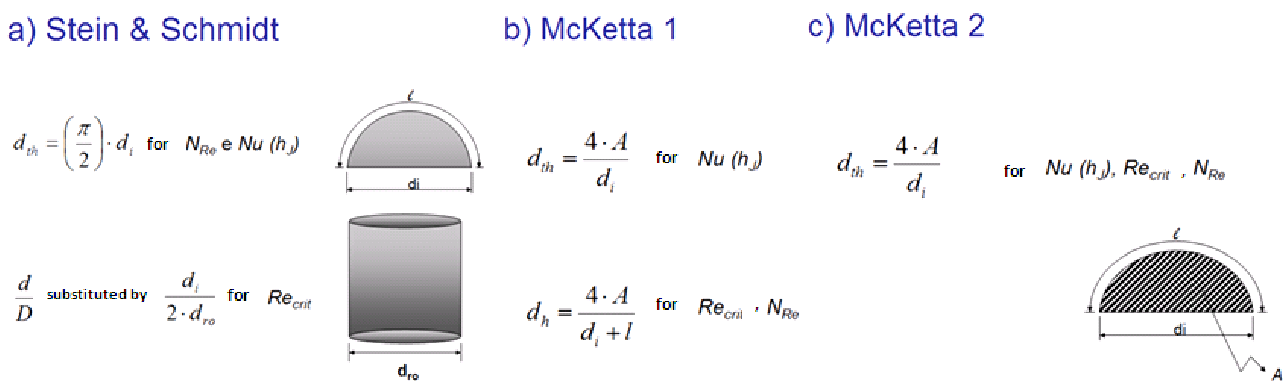

Figure 6: Simplified scheme of the main considerations of the itineraries followed

The first itinerary uses the equations given by Stein & Schmidt (1986) - item 7 - for the thermal diameter used in the calculation of the film coefficient of the refrigerant fluid (hj) as well as in the calculation of the adimensional numbers (N_{Re} e Nu). For the Re_{crit} calculation, half coil inner diameter in the ratio d / D is used (Fig. 6 - item a).

The second itinerary uses the considerations given by McKetta (1992) for the thermal diameter used in the calculation of the film coefficient of the refrigerant fluid (hj) as well as in the calculation of the Nu adimensional. The dimensioning, according to Fig. 6 (b), does not use thermal diameter in the calculation of the adimensional numbers which refer to fluid flow (Re_{crit} and N_{Re}).

Finally, the third itinerary uses all the considerations given by McKetta (1992) for the thermal diameter (Fig. 6 - item c) used in the calculation of the film coefficient of the refrigerant fluid (hj) as well as in the calculation of the adimensional numbers (Re_{crit}, N_{Re} and Nu).

Table 1. Results obtained in the thermal dimensioning of the fermenter

	Unit	Stein & Schmidt	McKetta 1	McKetta 2
dh	m	0.020	0.020	0.020
dth	m	0.118	0.042	0.042
Re_{crit}	-	4204	3723	4309
N_{Re}	-	482000	80340	173000
ε	-	0.016	0.020	0.018
N_{Pr}	-	1.444	1.444	1.444
N_{Prw}	-	1.444	1.444	1.444
Nu	-	1119	251	479
hj	W/m²K	5884	3433	6541
Gr	-	1.981x10E13	1.981x10E13	1.981x10E13
Ra	-	2.032x10E14	2.032x10E14	2.032x10E14
Nu_C	-	8126	8126	8126
hi	W/m².K	288	288	288
U	W/m².K	232	225	233
E	kg/hl.h	0.125	0.125	0.125
Tg	kcal/h	85750	85750	85750
Qg	Kw	65	65	65
$\Delta\partial_{LM}$	ºC	12	12	12
A	m²	243	243	243
Qa	W	654700	636800	657300
Q	kW	720	702	722
q	kW/m²	2.963	2.889	2.973
mNH_3	kg/h	2.020	1.970	2.030
$mNH_{3(uml)}$	kg/s	2.245	2.189	2.253
VNH_3	m³/h	12	12	13
v	m/s	0.133	0.129	0.133
ζ	-	0.013	0.019	0.016
Δp	bar	0.002	0.003	0.003

Considering the maximum value obtained in the necessary amount of ammonia through the software calculation (Tab.2), the difference as for the obtained by the itineraries (Tab.1) is of:

- Itinerary 1 (*Stein & Schmidt*): 17.0 % smaller.
- Itinerary 2 (*McKetta 1*): 14.1 % smaller
- Itinerary 3 (*McKetta 2*): 17.6 % smaller

Despite the great differences in values found in the heat transfer calculation of the cooling fluid and the *Reynolds* number in Tab.1, due to the different applications of the hydraulic and thermal diameters, it is verified that the estimates for overall heat transfer coefficient do not depend on the itinerary followed, because there is a maximum variation of 3.5% in the results for the calculation of this coefficient. The same is verified for the requested ammonia mass flow, where this variation is even smaller (about 3.0%).

Table 2. CCV cooling calculation by software Dedini – *Schmidding*

Ti (° C)	Tf (° C)	T_{NH3}(° C)	Q (kW)	Flow (kg/h)
14	13	-6	620.5	1726.6
13	12	-6	559.4	1556.6
12	11	-6	500.9	1393.8
11	10	-6	445.1	1238.5
10	9	-6	392.2	1091.3
9	8	-6	341.6	950.5
8	7	-6	292.3	813.2
7	6	-6	243.6	677.7
6	5	-6	195.7	544.5
5	4	-6	148.9	414.2
4	3	-6	93.1	259.0
3	2	-6	70.9	97.2
2	1	-6	84.3	235.7
1	0	-6	94.2	262.2

The overall heat transfer coefficient regarding the maximum mass flow (1726.6 kg/h) calculated by the software (Tab.2) was 201 W/m².K. Therefore, the maximum variation in the calculation for the manual method in relation to the software is around 16% larger, this value being responsible for the larger flow of ammonia found by the itineraries (directly proportional).

For approximately one decade, Dedini Indústrias de Base considers the amount of NH_3 as being double the calculated theoretical value, in other words, half the considered by *Gross* (1998). Before that, the company also considered this multiplication number (four times the calculated flow), and this number was abolished due to the technological improvements along this decade.

5. CONCLUSIONS

- The values for the overall heat transfer coefficient do not depend on the itinerary followed, with a maximum variation of 3.5% among the results of the different itineraries. For requested ammonia mass flow this variation is even smaller (maximum of 3.0%).
- The maximum variation in the overall heat transfer coefficient regarding the maximum mass flow, for the manual method in relation to the software, is around 16% larger. It can be concluded starting from item 7, that this variation is in agreement with the use of the heat transfer equations in the turbulent regime which provides a deviation of ± 15% if $N_{Re} > 2.2 \, . \, 10^4$.
- The requested ammonia flow for the cooling system of the proposed cylindroconical process equipment is directly proportional to the overall heat transfer coefficient.
- The amount of theoretical flow of circulating ammonia, described by Gross (1998) can be reduced in half, without damaging the cooling system.

6. REFERENCES

CERVESIA. Tecnologia Cervejeira, <http://www.cervesia.com.br>.

CRANE VALVES NORTH AMERICA. Flow of Fluids Through Valves, Fittings and Pipe. Technical Paper nº410M, Metric Edition1999.

Gnielinski, V. Proc. 8th Int. Heat Transf. Conf. San Francisco: Hemisphere, v.6, 1986.

Gross, P. Programme for Dimensioning the Cooling Areas at Cylindroconical Tanks. Brauwelt International, Technical Feature. Steiβlingen, 1998/II.

Kunze, W; Mieth, H.O. Technologie Brauer und Mälzer. Berlin: VLB, v.2, 1999.

McKetta, J.J. Heat Transfer Design Methods. Texas: CRC Press, 1992.

Schmidt, E.F.: Chemie Ing. – Techn.1967

STANDARDS OF THE TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION . 8th Edition , New York, 1999.

Swarbrick, J.; Boylan, J.C. Encyclopedia of pharmaceutical technology. 2nd ed. v.7, New York; Basel: Marcel Dekker, Inc., 2002

Unterstein, K. Cylindroconical tanks: considerations relating to tank and cooling surface sizing. Brauwelt International, Knowledge. Munich, v. 24, 2006.

VDI HEAT ATLAS / Ed. Verein Deutscher Ingenieure. VDI– Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (GVC). Düsseldorf: VDI – Verl., 1993.

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