Evaluation of film, jacket and internal cooling during fermentation of grape juice

J. Gryzagoridis[1] and Z.J. Steenkamp
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Nomenclature

A	Area	m^2
C_p	Specific heat at constant pressure	J/kg°C
de	Equivalent diameter	m
f	Friction factor	
F	Shape factor	
g	Gravitational acceleration	m/s^2
H	Height	m
h	Heat transfer coefficient	$W/m^2°C$
h_d	Mass transfer coefficient	kg/m^2s
i	Enthalpy	J/kg
k	Thermal conductivity	W/m°C
L	Length	m
m	Mass flow rate	kg/s
NTU	Number of transfer units	
P	Perimeter	m
p	Pressure	N/m^2
Q	Heat transfer rate	W
R	Gas constant	J/kg°C
T	Temperature	°C or K
U	Overall heat transfer coefficient	$W/m^2°C$
v	Velocity	m/s
W	Width	m
β	Volume coefficient of expansion	K^{-1}
ϵ	Effectiveness or emissivity	
μ	Absolute viscosity	kg/ms
ς	Density	kg/m^3
ν	Kinematic viscosity (μ/ς)	m^2/s
σ	Stefan-Boltzmann constant	$W/m^2°K$

Analytical and experimental aspects of water-film cooling, jacket cooling and internal cooling, which are the three predominant types of fermentation cooling found in the South African wine industry, are dealt with. The work focused on the following aspects: formulation of models which will enable simulation of each type of fermentation cooling under different running conditions for various sizes of available equipment; experimental work on all three main types of fermentation cooling under different running conditions to confirm the validity of the mathematical models; and comparison of analytical or predicted results of the performance (cooling capacity, pressure drop and effectiveness of the three types of fermentation cooling) with the experimental or measured values.

[1]Department of Mechanical Engineering, University of Cape Town, Private Bag, Rondebosch, 7700 South Africa

Dimensionless groups

Gr	Grashof number, $g\varsigma^2L^3\beta(T_w - T)/\mu^2$ or
Nu	Nusselt number, hL/k or hd/k for a tube
Pr	Prandtl number, $\mu C_p/k$
Re	Reynolds number, $\varsigma\nu L/\mu$ *or* $\varsigma vd/\mu$ for a tube

Subscripts

a	air
c	channel or convection
ci	channel inside
co	channel outside
cwi	cooled water in
cwo	cooled water out
eff	effective
f	film
h	heat exchanger
i	inside, initial
j	juice
l	level
m	mass
o	outside
p	constant pressure or pipe
r	radiation
s	strip
t	tank
T	total
v	vapour
w	water
wb	water bulk
wetb	wetbulb
wi	inside wall
wo	outside wall
∞	free-stream conditions

Introduction

Over the years the quality and variety of wines in South Africa has improved and increased, respectively. This is a direct result of growing better quality grapes and better thermal control during the wine making process. In countries with hot weather during harvest time, sufficient cooling is needed to ensure that a good wine recipe can be followed.

In other words, tanks holding juice must have sufficient cooling to remove the heat that is generated during fermentation. Film cooling, jacket cooling and internal heat exchangers, as shown in Figures 1(a), 1(b) and 1(c),

respectively, are the devices most commonly used for this purpose.

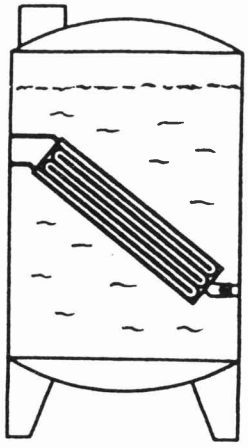


Figure 1(a) Tank with internal heat exchanger

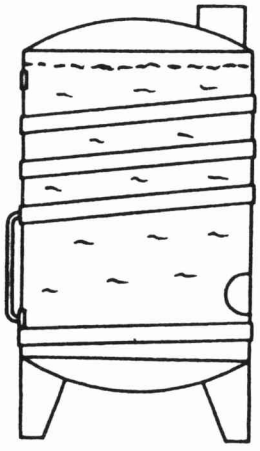


Figure 1(b) Jacket tank

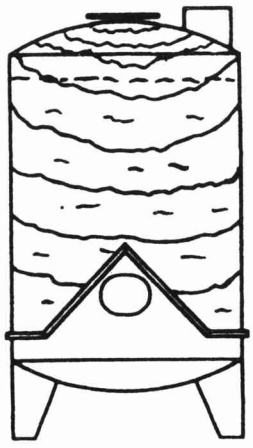


Figure 1(c) Film tank

Internal cooling

Internal heat exchangers are installed inside stainless steel or fibreglass tanks and are totally immersed in the juice. Cooled water is pumped through them and back to the refrigeration plant to be cooled again.

Jacket cooling

Jacket cooling is done by pumping cooled water through a rectangular stainless steel duct (jacket) welded to a tank. Today most jackets welded to tanks are in the form of a spiral. After its journey through the spiral conduit, the water gets pumped away to be cooled again.

Film cooling

Film cooling is done by pumping cold water over a stainless steel tank. Water flows from holes in a ring pipe above the tank and basically covers the outside tank area completely. The water flows into gutters and gets pumped away to be cooled again.

Internal cooling

The Weideman heat exchanger, is the most common internal heat exchanger used in wine cellars in the Western Cape. It consists of a long elliptical channel with metal strips linking six straights which are connected in series via U-turns. (See Figure 2.)

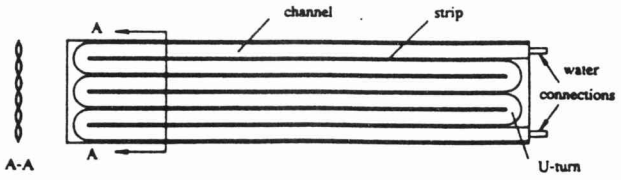


Figure 2 The Weideman heat exchanger

Analytical model

Consider the heat exchanger submersed, in a large volume of fermenting juice which is to be maintained at a constant temperature.

There are altogether seven strips on the heat exchanger. Five of them are between the cooling channels, where it can be assumed that the strip's end next to the channel is at T_{wall} and that no heat flows across the centre of the strip. That is, if one splits each of these five strips in the centre, each strip can be considered as two fins with their origin at T_{wall} and their other end insulated. Figure 3 displays one of these fins per unit channel length.

The heat transfer per unit channel length of these fins according to [3] is calculated by

$$Q_{s1} = \sqrt{h_o P_{s1} k A_{s1}}\,(T_j - T_{\text{wall}}) \tanh m_1 L_1 \qquad (1)$$

where m is defined as

$$m = \sqrt{\frac{h_o P_s}{k A_s}} \qquad (2)$$

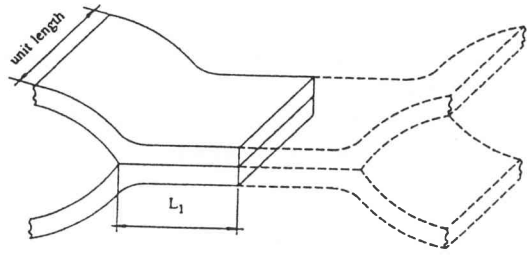


Figure 3 Half a strip between two cooling channels

The remaining two strips are at either side of the heat exchanger. They can be considered as fins with a finite length (total width of the strip) that lose heat by convection.

The heat transfer per unit channel length of these fins is calculated by

$$Q_{s2} = \sqrt{h_o P_{s2} k A_{s2}}\,(T_j - T_{wall})\,f(mL) \qquad (3)$$

where

$$f(mL) = \frac{\sinh m_2 L_2 + (h_o/m_2 k)\cosh m_2 L_2}{\cosh m_2 L_2 + (h_o/m_2 k)\sinh m_2 L_2}$$

the total heat transfer to the strips (Q_s) per unit channel length is

$$Q_s = 10Q_{s1} + 2Q_{s2} \qquad (4)$$

The heat transfer can also be calculated from the temperature rise of the cooling water circulated through the heat exchanger, using the following equation

$$Q = m_w C_{pw}\,(T_{cwo} - T_{cwi}) \qquad (5)$$

Inside the cooling channels heat transfer takes place by forced convection, therefore the Nusselt number can be calculated using the following equation which was developed by Petukhov (viscosity variations were neglected).[4]

$$Nu_i = \frac{\left[\frac{f}{8}\right] Re_w \Pr_w}{1.07 + 12.7\sqrt{\frac{f}{8}}\,(\Pr_w^{0.67} - 1)} \qquad (6)$$

and the heat transfer is calculated from

$$Q = h_i A_i\,(T_{wall} - T_{wb}) \qquad (7)$$

An iteration procedure involving equations 5,6 and 7 is adopted in order to determine T_{wall}. Once T_{wall} is determined, the outside convection coefficient (between the heat exchanger and the juice) can be evaluated adopting a further iterative solution using the following equation:

$$Q = h_o A_{co}\,(T_j - T_{wall}) + Q_s \qquad (8)$$

where the first term is in accordance with the definition of natural convection heat transfer from a surface to a fluid, and the second term is in accordance with equation (4).

Grashof, Prandtl and Nusselt numbers were calculated as shown in Figure 4 and the following power law relationship of the product of the Grashof-Prandtl numbers versus the average Nusselt numbers was obtained. Equation (9), derived by the authors, is later shown to provide an excellent model of the heat transfer processes and has hitherto not appeared in the literature.

$$\overline{Nu_o} = 34.1\,(Gr\,\Pr)^{0.12} \qquad (9)$$

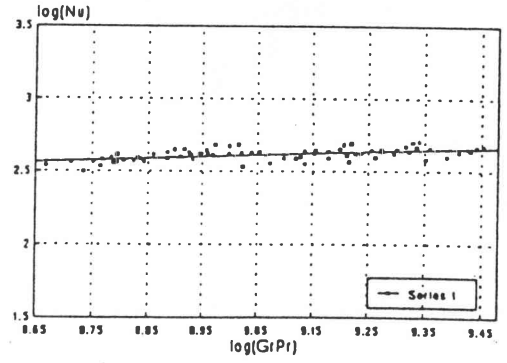


Figure 4 Free-convection heat transfer correlation between a cold surface of a given geometry (Weideman heat exchanger profile) immersed in fermenting grape juice

The overall heat transfer coefficient (U) and effectiveness (ϵ)

The calculated values for the inside forced convection coefficient h_i and the outside free convection coefficient h_o were now used to determine the overall heat transfer coefficient, U. Fouling was neglected, because all the heat exchangers tested were new. The overall heat transfer coefficients based on the outside area of the heat exchangers were calculated using

$$U_o = \frac{1}{\frac{A_T}{A_i}\frac{1}{h_i}} \qquad (10)$$

The results of a 1 820 × 400 mm Weideman heat exchanger are displayed in Figure 5.

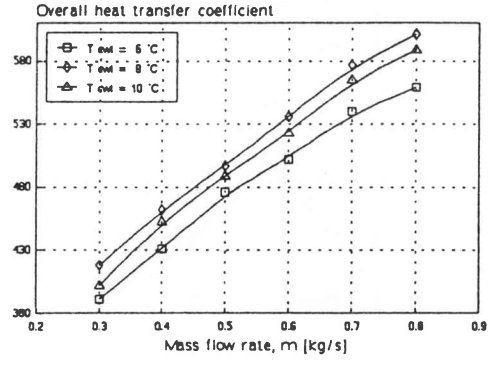


Figure 5 The overall heat transfer coefficient (U_o) for the 1 820 × 400 mm Weideman heat exchanger

The number of transfer units (NTU) were calculated from, [3]

$$NTU = \frac{U_o A_o}{C_{min}} \tag{11}$$

The effectiveness of the heat exchanger is approximately given by equation (3)

$$\epsilon = 1 - e^{-NTU} \tag{12}$$

The effectiveness of the tested heat exchangers are plotted against the number of transfer units and the results of a 1 820 × 400 mm Weideman heat exchanger are displayed in Figure 6.

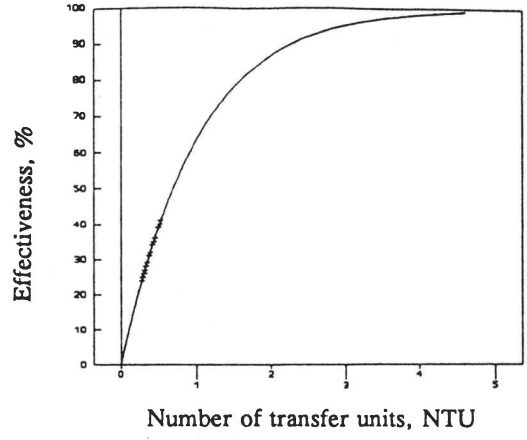


Figure 6 The effectiveness of the 1 820 × 400 mm Weideman heat exchanger

Jacket cooling

Analytical model

Experimental data were obtained on a jacket heat exchanger while under operation during peak fermentation of juice inside the wine tank. Using relevant relationships found in the literature, the results obtained approximate the experimental data and therefore may be used to provide guidelines to manufacturers of 'Jacket' wine heat exchangers.

Consider the tank filled with fermenting juice while cooling water is supplied and pumped through the jacket to remove the heat generated by the fermenting juice. For the purpose of this analysis, the energy balance will be performed on the jacket portion of the fermentation tank, because in fact that is the heat exchanger proper.

Solution procedure

A value for the cooling water outlet temperature (T_{cwo}) is assumed and the cooling capacity for a given flow rate (m_w) and water inlet temperature (T_{cwi}) is calculated, using equation (5).

The contribution of the heat transfer from the ambient, to the total heat transfer, as measured by the temperature increase of the circulating water, can be calculated knowing the ambient temperature (T_∞) from

$$Q_{air} = \frac{T_\infty - T_{wb}}{\frac{1}{h_a A_{jt}}} \tag{13}$$

In order to evaluate the free convection coefficient (h_a) we make use of the equation developed by Bayley [3]

$$h_a = \frac{0.59\,(Gr\,\mathrm{Pr})^{\frac{1}{4}}\,k_a}{W_j} \tag{14}$$

The heat transfer contribution from the fermenting juice to the circulating water can be calculated for a known juice temperature (T_j) from

$$Q_{juice} = \frac{T_j - T_{wb}}{\frac{1}{A_{jt}}\left[\frac{1}{h_w} + \frac{1}{h_j}\right]} \tag{15}$$

The forced convection heat transfer coefficient inside the channel is calculated using the relationship for forced convection developed by Dittus & Boelter [4]

$$h_w = \frac{0.0243\,Re^{0.8}\,\mathrm{Pr}^{0.4}\,k_w}{de} \tag{16}$$

The natural heat transfer convection coefficient on the juice side of the Jacket is calculated using the following relation

$$h_j = \frac{0.59\,(Gr\,\mathrm{Pr})^{\frac{1}{4}}\,k_j}{W_j} \tag{17}$$

The heat transfer from the air together with the heat transfer from the juice, to the circulating water, can be calculated from

$$Q = \frac{T_\infty - T_{wb}}{\frac{1}{h_a A_{ja}}} + \frac{T_{wall} - T_{wb}}{\frac{1}{h_w A_{jt}}} \tag{18}$$

where T_{wall} is the only unknown and can be found by rearranging the above equation

$$T_{wall} = \frac{Q - h_a A_{ja}(T_\infty - T_{wb})}{h_w A_{jt}} + T_{wb} \tag{19}$$

Finally the total heat transfer to the cooling water is calculated using equations (5) and (18). If the value of Q determined from equation (18) does not correspond to the value of Q determined from equation (5), then the value of the temperature of the cooling water outlet temperature (T_{cwo}) is varied to better the situation and the calculations are repeated until agreement is obtained, which signifies that the cooling capacity of the Jacket heat exchanger is predicted.

The overall heat transfer coefficient (U) and approx. effectiveness (ϵ)

In this specific case it is not possible to calculate a single overall heat transfer coefficient and relate it with a single temperature difference, because there are two temperature differences involved which are independent of each other.

The water bulk temperature (T_{wb}) is influenced by the ambient temperature (T_∞) and the juice temperature (T_j), but the ambient temperature and the juice temperature are totally independent of each other.

It is thus not possible to use the traditional *NTU* method to calculate the effectiveness (ϵ) of the Jacket heat exchanger as it was done in the case of the Weideman heat exchanger.

The only purpose of the Jacket heat exchanger as used in the wine industry is to remove heat generated by the fermenting juice. This will only happen as long as the temperature of the cooling water is below the juice temperature. It can thus be assumed that when the cooling water outlet temperature (T_{cwo}) is equal to the juice temperature (T_j) the maximum possible heat has been transferred from the juice to the cooling water. The effectiveness of the Jacket heat exchanger therefore can be defined as the actual heat that is removed from the juice against the maximum possible heat that could have been removed. The effectiveness therefore can be calculated by

$$\epsilon = \frac{A_{jt}\,(T_j - T_{wb})}{\left[\frac{1}{h_w} + \frac{1}{h_j}\right] m_w C_{pw}\,(T_j - T_{cwi})} \qquad (20)$$

where the water bulk temperature (T_{wb}) for this application will be calculated by

$$T_{wb} = \frac{T_j + T_{cwi}}{2} \qquad (21)$$

Computer program

The predicted results compare adequately with the experimental ones. Using the analytical model, graphs are produced where the cooling capacity and the effectiveness of a Jacket heat exchanger can be determined for different mass flow rates, cooling water inlet temperatures, juice temperature and ambient temperatures. Some of the results are displayed in the following graphs. (During experiments $T_{cwi} = 8°C$; $T_j = 20°C$ and $T_\infty = 22°C$.)

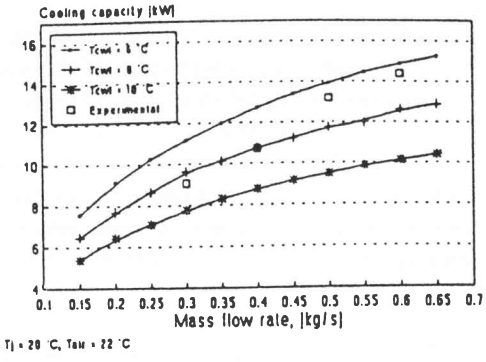


Figure 7 Cooling capacity of the Jacket heat exchanger with varying cooling water inlet temperature (T_{cwi})

However these results were obtained by varying one parameter and keeping the others constant or at a given value; useful results, in that they give an indication of the effect of the particular parameter. It is desirable however to get an overall effect on the basis of a cross mix of values for the cold water inlet temperature (T_{cwi}), required temperature of the juice (T_j) and ambient temperature (T_∞). A representative temperature (T_{eff}), which combines the overall effect of any combination of the above-mentioned temperatures, on the cooling capacity and effectiveness of the Jacket heat exchanger was arrived at after trial and error calculation and intuition. It was found that for a realistic large variation in the ambient temperature, the cooling capacity of the Jacket heat exchanger was less than 1%. Therefore the representative temperature (T_{eff}) will only be expressed as a function of the cold water inlet temperature (T_{cwi}) and the juice temperature (T_j). The end result is a relationship which produces a family of curves of (T_{eff}) an effective temperature calculated by

$$T_{eff} = 1 + 0.093\,(8 - T_{cwi}) + 0.0967\,(T_j - 20) \qquad (22)$$

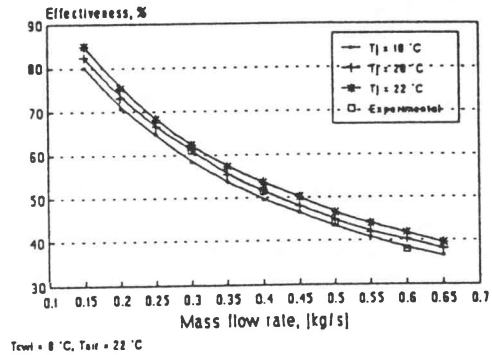


Figure 8 Effectiveness of the Jacket heat exchanger with varying juice temperature (T_j)

Typically, the cooling capacity is affected by 9.3% for every change of one degree Celsius of the cooling water inlet temperature, or 9.7% change for each degree change of juice temperature. The results are displayed in Figures 9 and 10.

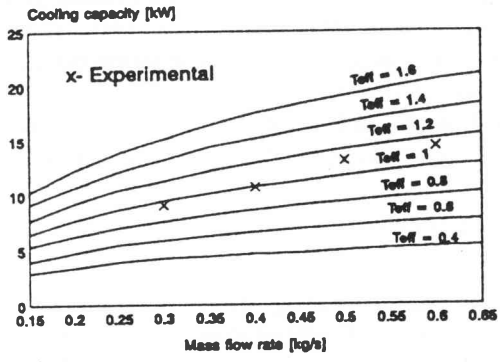


Figure 9 Cooling capacity of the Jacket heat exchanger with varying effective temperature (T_{eff})

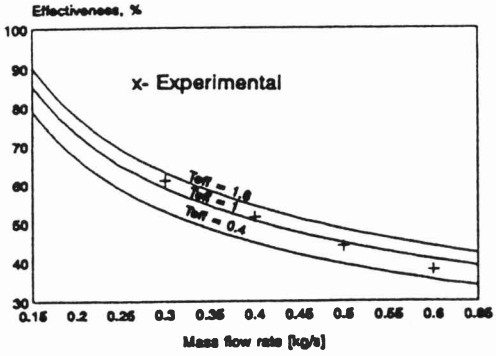


Figure 10 The effectiveness of the Jacket heat exchanger with varying effective temperature (T_{eff})

Water-film cooling

Consider the tank filled with fermenting juice. Heat is transferred to the water-film from the fermenting juice as well as from the surroundings. Heat transferred from the surroundings according to [1] and [2] is via radiation, convection, and mass transfer.

Note that there is an area of the tank (the top part) that is covered with a water-film on the outside and with air on the inside, due to the fact that these tanks are only filled ±85% of full capacity during fermentation, approximately $\frac{9}{10}$ of the tank's height. An amount of heat is transferred to the water-film at this stage, mainly from the surroundings, before any heat is removed from the fermenting juice. During this stage it is assumed that the heat transferred from the air inside the tank to the water-film is insignificant due to the low natural convection between the enclosed air and the tank wall.

As the cold water flows from the ring pipe, it forms a water-film on top of the tank which flows radially towards and then down the vertical walls of the tank. The amount of heat which is transferred to the water-film, till it reaches the level of the juice, will be referred to from now on as the **initial heat transfer** (Q_{i}). This initial amount of heat transfer obviously increases the water-film temperature. It is only after the cascading water-film has reached the level of the juice, that heat will be transferred from the fermenting juice. The total heat that is transferred to the water-film below the level of the juice will from now on be referred to as **heat transfer from the juice level** (Q_{wtj}).

Total heat transfer from the juice level to the cascading water-film

The total heat transfer to the cascading water-film, once this film has reached the juice level, is calculated from

$$Q_{\text{wtj}} = \frac{A_{\text{wta}}\left(T_{\infty} - T_{\text{wbj}}\right)}{\left[h_{\text{r}} + h_{\text{c}} + \frac{h_{\text{m}}\left(T_{\text{wetb}} - T_{\text{wbj}}\right)}{T_{\infty} - T_{\text{wbj}}}\right]^{-1}} + \frac{A_{\text{wtj}}\left(T_{\text{j}} - T_{\text{wbj}}\right)}{\frac{1}{h_{\text{w}}} + \frac{1}{h_{\text{j}}}} \tag{23}$$

h_{r} is the radiation heat transfer coefficient as defined by the following equation.[3]

$$h_{\text{r}} = \epsilon\sigma F_{12}\left(T_{\infty}^2 + T_{\text{wb}}^2\right)\left(T_{\infty} + T_{\text{wb}}\right) \tag{24}$$

$h_{\text{c}}$is the natural heat transfer coefficient as calculated from (3).

$$h_{\text{c}} = 0.332\nu_{\max}\varsigma_{\infty}C_{\text{p}\infty}Re^{-\frac{1}{2}}Pr^{-\frac{2}{3}} \tag{25}$$

h_{m} is the mass heat transfer coefficient defined as calculated from (1), (2), (4).

$$h_{\text{m}} = \frac{i_{\text{v}}h_D\left[\frac{P_{\text{v}\infty}}{T_{\text{wetb}}} - \frac{P_{\text{vw}}}{T_{\text{wb}}}\right]}{R_{\text{w}}\left(T_{\text{wetb}} - T_{\text{wb}}\right)} \tag{26}$$

h_{w} is the forced convection coefficient which can be evaluated from (3).

$$h_{\text{w}} = \frac{0.453\,Re^{\frac{1}{2}}Pr^{\frac{1}{3}}k_{\text{w}}}{H_{\text{wt}}} \tag{27}$$

h_{j} is the natural or free convection coefficient between the juice and the tank wall, calculated from (3).

$$h_{\text{j}} = \frac{0.1\left(GrPr^{\frac{1}{3}}k_{\text{j}}\right)}{H_{\text{wt}}} \tag{28}$$

Initial heat transfer to the water

The initial heat transfer (Q_{i}) can be estimated using equation (23) suitably modified by dropping the last term referring to the juice side of the tank.

Solution procedure

Initial heat transfer to the water

The initial heat transfer to the water-film is calculated, using equation (5), for a given cooling water mass flow rate (m_{w}) and water inlet temperature (T_{cwi}), by assuming the cooling water outlet temperature at the level of the juice (T_{wjl}).

The water temperature at the juice level (T_{wjl}) is increased or decreased as the calculations are repeated until the values of the initial heat transfer (Q_{i}), calculated from equations (5) and (29), agree with each other. In this manner the temperature of the cascading water-film at the level of the juice is determined for a given mass flow rate.

Heat transfer from the level of the juice downwards

As we saw earlier, the calculations to determine the initial heat transfer (Q_{i}) were necessary because they yielded the bulk temperature of the cascading water-film as it reached the level of the juice. Thereafter the fermenting juice contributes a significant amount of heat which is added to the heat gained from the surroundings. The total heat gained by the cascading water-film starting at the level of the

juice until it reaches the bottom of the tank can be determined using equation (23). The amount of heat gained by the water-film can also be determined by using the energy equation (5).

The following trial and error procedure is adapted in order determine the total heat gained by the cascading film:

The temperature of the water-film as it has reached the bottom of the tank is assumed and a quick calculation of the total heat is made using the energy equation (5). This amount of heat is compared with the amount calculated using equation (23) and when suitable agreement is reached the iterations are terminated.

The overall heat transfer coefficient (U) and approx. effectiveness (ϵ)

Here we have a similar situation as in the case of the Jacket heat exchanger where heat transfer to the cooling water takes place from two different sources which are at different temperatures independent of one another. Thus in the traditional sense an overall heat transfer coefficient cannot be calculated nor can the effectiveness of the heat exchanger be presented using the NTU method.

In order for the cascading water to continue cooling the system on the side of the juice the water outlet temperature (T_{cwo}) should remain below the juice temperature. Therefore the flow rate of the cooling water must be such to accommodate the contribution or input of heat from the juice side as well as from the surrounding.

If we take the case that the surrounding environment temperature (T_∞) is greater than (T_{j}) the juice temperature, it is obvious that the maximum value that we should allow the water outlet temperature to reach, would be the temperature of the juice. Therefore one could possibly formulate an effectiveness for the cooling of the juice in this system as suggested below:

$$\epsilon = \frac{A_{\text{twj}}\,(T_{\text{j}} - T_{\text{wbj}})}{\left[\frac{1}{h_{\text{w}}} + \frac{1}{h_{\text{j}}}\right] m_{\text{w}} C_{\text{pw}}\,(T_{\text{j}} - T_{\text{cwjl}})} \qquad (29)$$

Computer program

Based on the analytical model, software was written to perform calculations on Water-film wine tanks. The computer program may be used to predict the cooling capacity and effectiveness of the Water-film wine tank for different water inlet temperatures, juice temperatures and ambient temperatures. Some of the results using this program for the range of temperatures expected in practice are displayed in the following figures. (During experiments $T_{\text{cwi}} = 8°\text{C}$, $T_{\text{j}} = 15°\text{C}$ and $T_\infty = 18°\text{C}$.)

As in the case of the Jacket heat exchanger, it is desirable to get a representative temperature (T_{eff}) combining the overall effect of any combination of the cold water inlet temperature (T_{cwi}), required temperature of the juice (T_{j}) and ambient temperature (T_∞) on the cooling capacity and effectiveness of the Water-film heat exchanger. The end result is a relationship which produces a family of curves of (T_{eff}) an effective temperature calculated as follows:

$$T_{\text{eff}} = 1 + 0.1415\,(8 - T_{\text{cwi}}) + 0.131\,(T_{\text{j}} - 15) + 0.015\,(T_\infty - 18) \qquad (30)$$

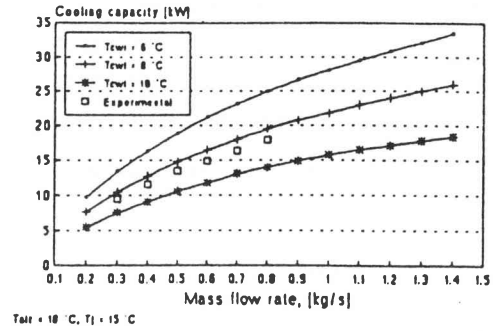


Figure 11 Cooling capacity of the Water-film with varying cooling water inlet temperature (T_{cwi})

The results are displayed in Figures 12 and 13.

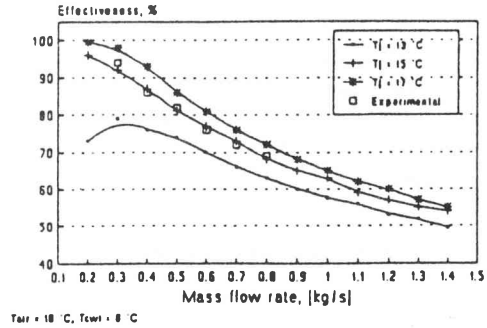


Figure 12 Effectiveness of the Water-film with varying juice temperature (T_{j})

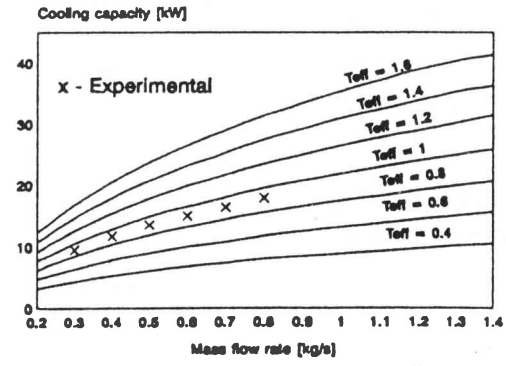


Figure 13 Cooling capacity of the Water-film for different values (T_{eff})

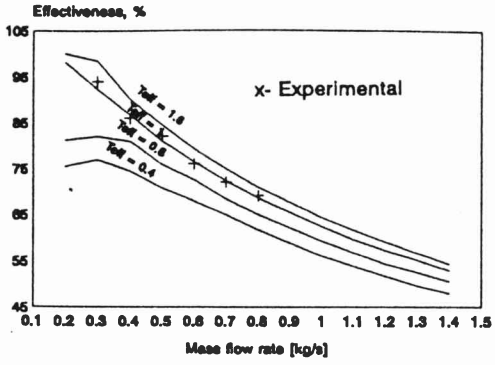


Figure 14 Effectiveness of the Water-film for different values of the (T_{eff})

Typically, the cooling capacity is affected by 14% for every change of one degree Celsius of the cooling water inlet temperature, or 13% change for each degree change of juice temperature, or 1,5% change for each degree of ambient temperature.

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