EVALUATION OF FILM, JACKET AND INTERNAL COOLING DURING FERMENTATION OF GRAPE JUICE

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SUPERVISOR

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I, Zaldeus Johannes Steenkamp submit this thesis in full fulfilment of the requirements for the degree of Master of Science in Engineering. I claim that it has not been submitted in this or any similar form for any degree at any University.

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ABSTRACT

The work described in this thesis deals with 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.

During the past decades questions were constantly raised concerning the thermal efficiency of equipment used in the above mentioned types of fermentation cooling. These questions were particularly asked when a wine cellar expanded operations, or after product damage had occurred.

The thesis presents the work that was carried out along the following aspects:

1. The building of models which will enable simulation of each type of fermentation cooling under different running conditions for various sizes of available equipment .

2. The experimental work on all three main types of fermentation cooling under different running conditions to confirm the validity of the mathematical models.

3. The 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.

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Honour and glory to my Creator and Saviour Jesus who enabled me to do this work.

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LIST OF SYMBOLS

Symbol	Description	Units
A	Area	m^2
C_p	Specific heat at constant pressure	J/kg°C
D	Diffusion coefficient	m^2/s
d	Diameter	m
de	Equivalent diameter	m
f	Friction factor	
g	Gravitational acceleration	m/s^2
H	Height	m
h	Heat transfer coefficient (surface to fluid interface)	W/m^2°C
h_d	Mass transfer coefficient	kg/m^2s
i	Enthalpy	J/kg
i_{fg}	Latent heat	J/kg
K	Pressure loss coefficient	
k	Thermal conductivity	W/m°C
L	Length	m
m	Mass flow rate	kg/s
NTU	Number of transfer units (UA/C_{min})	
P	Perimeter	m

p	Pressure	N/m^2
Q	Heat transfer rate	W
r	Radius	m
T	Temperature	°C or K
t	Thickness	m
U	Overall heat transfer coefficient	$W/m^2 °C$
V	Volume flow rate	m^3/s
v	Velocity	m/s
W	Width	m
w	Humidity ratio	
x	Distance	m
y	Distance	m
α	Thermal diffusivity, $k/\rho Cp$	m^2/s
β	Volume coefficient of expansion	$°C^{-1}$
Δ	Differential	
ϵ	Effectiveness	
μ	Absolute viscosity	kg/ms
ρ	Density	kg/m^3
τ	Time	s
ν	Kinematic viscosity (μ/ρ)	m^2/s

DIMENSIONLESS GROUPS

Gr	Grashof number, $g\rho^2 L^3 \beta(Tw-T)/\mu^2$ or $g\rho^2 d^3 \beta(Tw-T)/\mu^2$ for a tube
Nu	Nusselt number, Hl/k or hd/k for a tube
Pr	Prandtl number, $\mu C_p/k$
Re	Reynolds number, $\rho vL/\mu$ or $\rho vd/\mu$ for a tube
Sc	Schmidt number, $\mu/\rho D$

SUBSCRIPTS

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

CHAPTER 1
INTRODUCTION

1.1 BACKGROUND

Over the years the quality and variety of wines in this country 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.

Fermentation, in the context of wine making, is the process during which sugar is converted to alcohol. The rate of fermentation is among other things, a function of the temperature of the juice. The hotter the juice the higher the rate. Heat is generated during fermentation (*on average 544.28 kJ per kg sugar*) and must be constantly removed for better control of the process.

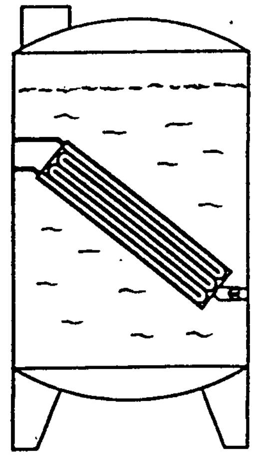
Most grapes arriving from the vineyard at the cellar are hot. In some districts it is quite common for cellars to receive grapes at temperatures of 37 °C or more.

After arriving, the berries are separated from the stalks. It is done by hitting the bunch hard with metal fingers attached to a rotating axle. Once the juice is set free from the berries, natural fermentation will start. Berries and juice (mash) are collected and pumped through a pre-cooler in order to cool the mash down to about 15 °C. Husks and pips are removed from the mash, the juice is pumped into a settling tank for the solids to settle on its bottom and then the juice is pumped into a fermentation tank.

Fermentation tanks must have sufficient cooling to remove the heat that is generated during fermentation. Film cooling, jacket cooling and internal heat exchangers, as shown overleaf, are used for this purpose.

Internal cooling

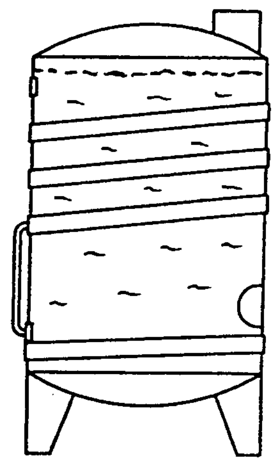
Internal heat exchangers such as the Weideman kind, are installed inside stainless steel or fibreglass tanks and are totally immersed in the juice. Cooled water is pumped through them and back to the refrigeration plant to be cooled again.



(a)

Jacket cooling

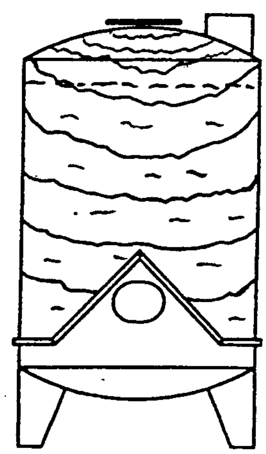
Jacket cooling is done by pumping cooled water through a rectangular stainless steel duct (jacket) welded to a tank. Today most jackets welded to tanks are in the form of a spiral. After its journey through the spiral conduit, the water gets pumped away to be cooled again.



(b)

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.



(c)

Figure 1.1 Wine fermentation tanks a) Tank with internal heat exchanger b) Jacket tank c) Film tank.

1.2 SYNOPSIS

Manufacturers of fermentation cooling equipment have relied on rules of thumb during the design and manufacturing stage. As a result, sufficiently reliable heat transfer data are either not readily available or non-existing. The consequence of this is that often the use of such equipment will affect the quality of wine produced, because the ability to cool the juice sufficiently and to monitor the fermentation temperature properly is the most important factor (besides the quality of grapes) that determines the wine quality. The release of appropriate technical data and design parameters would provide the wine industry with more efficient cooling systems.

CHAPTER 2
REVIEW ON COOLING FERMENTING GRAPE JUICE OVER THE PAST DECADES

2.1 AN HISTORICAL NOTE ON COOLING FERMENTING GRAPE JUICE

South African wine makers contributed greatly to paving the way for the effective cooling of grape juice before and during fermentation to produce high quality wines. Names like N.C. Krone, Pon van Zyl [1] and many more started as early as the nineteen thirties to experiment with fermentation cooling. They realised the value of cooling during the wine making process and today it is unthinkable to make delicate quality wines without it.

It appears that very little literature is available on the history and technology of fermentation cooling in the S.A. wine industry. Dates and facts given here, are as they were remembered by several entrepreneurs and contributors in this field, that were interviewed by the author. From the interviews it seems that the first form of cooling in the South African wine Industry, was when N.C. Krone of Twee Jonge Gezellen (in the middle 1930's) pumped water from a well through copper coils that were immersed in concrete wine tanks and wine barrels. In 1937 NC Krone pumped water from a channel through a Fuchs-cooler to precool must. The must was pumped through a large coil while the water from the channel was circulated around it. In 1940, while working at a cellar in Wellington, Krone experienced a lot of trouble controlling the fermentation process during the pressing season. With no cooling of any sort available he added matured wine of the previous season to the fermenting must and in this way controlled the fermentation process. As the years past by, he continued to experiment with cooling during the wine making process.

In 1959 a Freon refrigeration unit was installed at Twee Jonge Gezellen. The grape juice was precooled to a temperature between 8°C and 10°C using water which was cooled from the ice produced by the refrigeration unit. At the same time additional refrigeration was installed for the purpose of air conditioning the cellar. The temperature of the air inside the cellar was maintained at 10°C during the fermentation period, which is still the practice today at Twee Jonge Gezellen cellar. That year Twee Jonge Gezellen won 13 of the 14 potential prizes in the white wine category on the Cape Youth Wine Show.

According to Dick Vergunst of ERE, Thermotank (with whom he was the manager at the time) installed the first wine cooling unit in 1958/9 at the Co-op Riebeeck Kasteel wine cellar. About the same period another cooling plant was installed at Romansriver. The heat exchangers used were copper coils immersed in the must, contained in the concrete wine tanks. Before the installation of the cooling plant water from a cooling tower at a temperature of about 25°C was pumped through these copper coils. The copper coils used for precooling were later substituted with plate heat exchangers introduced by APV Hall (Ltd) to the South African market. These did not work well due to the narrow spacing between the plates and today large double tube (or tube in a tube) heat exchangers are being used for this purpose. Between 1965 and 1968 a lot of cellars moved away from concrete to stainless steel wine tanks. Cooling water was pumped over the tank in such a manner that a thin cascading water-film covered the total area of the wine tank. The water was collected in gutters at the floor around the tank, from where it was pumped back to the cooling unit.

Ray van Schalkwyk confirms that stainless steel wine tanks with water-film cooling were extensively used for fermentation cooling in 1967/8. The disadvantage of film-cooling was a constantly wet wine cellar during the pressing season. In 1969 Bob Brenell and Andries Weidemann manufactured plate heat exchangers for fermentation cooling. The plates were immersed in the wine tanks and cooled water or glycol was pumped through them. Today there are several heat exchangers of the immersed type used successfully during fermentation in almost every cellar in the Western Cape.

In 1970 Ray van Schalkwyk installed the first Jacket wine tank at Bovlei wine cellar. However, the cooling capacity of the Jacket heat exchanger was not sufficient to remove the fermentation heat and cold water was pumped over the tank to rescue the fermenting juice. In 1972 the first stainless steel gutters were welded to stainless steel water-film wine tanks at Ashton Co-op, replacing the gutter in the concrete floor. In 1982 Ray van Schalkwyk made another attempt to build a Jacket wine tank for his customer Jan Boland Coetzee of Vriesenhof. This time the Jacket had a cross section of 100 x 12 mm which worked well and it covered 40 % of the total tank area. Today Jacket wine tanks are a common feature in most wine cellars.

Mr Nico Mostert of Stellenbosch Refrigeration has been involved in fermentation cooling since the early 1950's. He recalls the first cooling unit to be installed in a wine cellar in 1951. Late Kowie Roux of the farm Verdun in Vlottenburg, approached Barlows who installed a Freon 12 cooling unit with a five horsepower electrical compressor in their estate wine cellar. The evaporator of the cooling unit was situated in a 1000 litre water tank. The water inside the tank was cooled down to just above 0°C. The grape juice was pumped through a copper coil installed inside the water tank around the pipes of the evaporator. The grape juice was precooled to 15°C. During fermentation the temperature rose and when it reached 18°C it was cooled again to 15°C. This cooling unit could handle approximately 3000 litres of grape juice per day successfully and consequently a second of its kind was installed in 1953.

In the early 1960's with the successful introduction of Lieberstein (a semi sweet wine) nearly every cellar started to produce semi sweet wines. In order to produce a semi sweet wine, sufficient cooling is required to stop the fermentation process while the sugar content of the grape juice is still relatively high. During the 60's, before wine makers moved from concrete to stainless steel wine tanks, mild steel wine tanks arrived on the market. These tanks were insulated with polyurethane on the outside, epoxy coated on the inside and only used for making high quality wines. Copper coils or stainless steel pipes were used as heat exchangers inside these special tanks.

2.2 COOLING AND THE WINE MAKING PROCESS

The wine making process actually begins at harvesting the grapes at the right time when the sugar and acid content are as desired by the wine maker.

As mentioned before, after the arrival of the grapes at the cellar the berries are separated from the stalks and the berries and juice (mash) pumped through a pre-cooler. It is cooled down to a temperature between 18 to 12 °C depending on what type of wine is being made [1], [2]. This decrease in temperature almost stops the fermentation process and allows further control. The mash is pumped from the pre-cooler into a separator where the husks and pips are removed from the mash. The juice is then pumped into a settling tank where the solids settle on the bottom of the tank over a period of 24 hours. The juice is then pumped into a fermentation tank where the wine maker converts it into wine. Heat is generated during fermentation and must be constantly removed to ensure control of the process. After fermentation stops, the juice is now deemed to be wine and is ready to be filtered and bottled.

Thus to sum up, pre-cooling is a rapid process where a large amount of mash is cooled in a short time to slow fermentation down to such a rate that the wine maker has full control over it. Fermentation cooling is a longer process from 9 to 22 days where all the heat generated during fermentation must be removed to enable the wine maker to follow his wine recipe.

2.3 FERMENTATION COOLING

Fermentation in the context of wine making is the process during which sugar is converted to alcohol. The rate of fermentation depends among other things, on the temperature of the juice. There is a direct relationship between the must temperature and the metabolic activity of the yeast [3]. The hotter the juice, the higher the rate of yeast growth and consequently more heat is generated. During fermentation on average heat is generated at a rate of 544.28 kJ for each kg of sugar that is converted to alcohol

[1],[2]. Apart from the heat that is generated during fermentation, heat may also be transferred from the environment to the fermenting juice. The amount of heat transferred from the environment to the juice depends mainly on the temperature inside the cellar and the type of fermentation cooling used. A rule of thumb that has been used through the years, is that the heat load from the environment is between 10 % [1] and 15 % [2] of the total cooling needed for fermentation. There is clearly a need for technical data required in calculating the heat gain from the environment. Many cellars have already invested a lot of money insulating and air conditioning their premises.

2.4 CALCULATION OF FERMENTATION COOLING

The fermentation period of a wine can be anything from 9 to 22 days depending on what type of wine is being made. As soon as the fermentation of a wine is completed, it is pumped from the fermentation wine tank to a storage tank and the fermentation tank becomes available for the next batch. Once the fermentation schedule and amount of fermentation tanks are determined, the cooling capacity needed for fermentation during the pressing season is calculated. Fermentation cooling is mainly a function of the amount of juice and the rate of fermentation. An example for fermentation cooling needed in a cellar is given in appendix A.

Tests done at Stellenbosch Farmers Winery [4] indicate that the fermentation load is increased by approximately 9% as a result of heat gain from the environment. These tests were performed in a well insulated cellar. In practice the heat gain from the environment is expressed as a percentage of the total cooling needed during the pressing season. During the same tests, it was found that the total heat gain in the refrigerated water distribution piping is approximately 20 % of the overall refrigeration needed. In general the figure of 15 % was being used following a paper by Vergunst [2] on refrigeration as applied in the wine industry, in Die Wynboer.

2.5 HEAT EXCHANGERS FOR FERMENTATION

Fermentation tanks must have sufficient cooling to remove the heat that is generated during fermentation. Water-film cooling, Jacket cooling and internal heat exchangers are used for this purpose.

Internal cooling

The internal heat exchangers must always cover the total height of a wine tank, to prevent a vertical temperature gradient which is not desirable at all when high quality wines are made. Advantages of the internal heat exchanger are that there is no heat gain from the environment to the cooling water running through them. The risk with this type of fermentation cooling is that if there is a leak in the heat exchanger, contamination of the wine will result. A few leaks were reported in the early years of refrigeration, but it became standard practice at most cellars to perform pressure tests on the heat exchangers prior to the pressing season and consequently no leaks during the pressing season were reported in recent years.

Jacket cooling

According to Rule [5] at least 25 % of the area of a stainless steel wine tank must be covered by a jacket for effective fermentation cooling. His concern about heat gain from the environment has prompted a suggestion that plastic wine tanks rather than stainless steel should be used with a jacket covering 50 % of the plastic tank area. The author's opinion is that heat gain from the environment to the cooling water in the jacket is not significant in most cellars, due to the fact that cellars are well insulated and almost draft-free. Furthermore, the size of a jacket should not be judged according to a percentage of a tank area, but rather according to an effective jacket length. In most cases Jacket heat exchangers reach an effectiveness of ± 80 % after the cooling water has been flowing over a distance of 25 to 30 meters in the jacket. At this stage the temperature gradient between the cooling water and the juice is relative small and quite a length of jacket will be required to obtain further significant increases in effectiveness.

Film cooling

Due to the fact that the cooling water is exposed to the environment over such a large area, Water-film cooling is the most vulnerable to heat gain from the environment. This is probably the reason why the Water-film is the only type of fermentation cooling on which analytical as well as experimental work was done to predict the heat gain from the environment under certain conditions. The work of Voigt [9] and Kröger [10] are examples where from a balance of shear and gravity forces the velocity and thickness of the water-film were calculated.

According to Voigt [9] the heat gain from the environment is mainly by natural convection and water vapour from the ambient air that condenses on the water-film. Kröger [10] agrees with Voigt, but suggests that radiation heat transfer must also be considered.

2.6 REFRIGERANTS USED FOR FERMENTATION COOLING

Through the years according to the Distilleries group [6], brines, Ammonia, Freon or water were used as coolants in the heat exchangers. The type of coolant is very important when it comes to fermentation cooling, because its bulk temperature and thermal characteristics determine the size and type of fermentation heat exchanger to be used. In systems using brine or water as a coolant, the coolant is cooled by a refrigeration unit and from there it is pumped to the fermenting juice where it enters a heat exchanger of some kind. At some cellars Ammonia and Freon are used in direct expansion systems where the refrigeration plant evaporator is placed in direct contact with the juice to be cooled. Handling problems occur with these systems as the coolants have to be transferred over considerable distances and the coolants are relatively expensive as well. The advantage of using these coolants is that it requires

40 to 60 % less heat exchanger surface area than brine coolants.

Today most cellars is South Africa use alcohol brines or cold water as coolants in secondary cooling systems where the coolant is first cooled by a refrigeration unit and then pumped to the fermenting juice. Alcohol brines are used rather than salt brines, because of the ease in preparing them, negligible corrosiveness and they do not block the pipes with salt precipitation.

CHAPTER 3
WEIDEMAN HEAT EXCHANGER

3.1 THE HEAT EXCHANGER FOR INTERNAL COOLING

The Weideman plate heat exchanger is a typical example of an internal cooling device and consists of two 0.9 mm thick, 316 stainless steel sheets. The sheets are placed on a jig and an elliptical duct is pressed into each. The sheets are then seam welded together. The heat exchanger now consists of a long elliptical channel with ± 10 mm metal strips linking six straights which are connected in series via U-turns. Circular 16 mm stainless steel pipes are welded at the inlet and outlet of the channel for water connections. Figure 3.1 shows a schematic diagram of a Weideman heat exchanger.

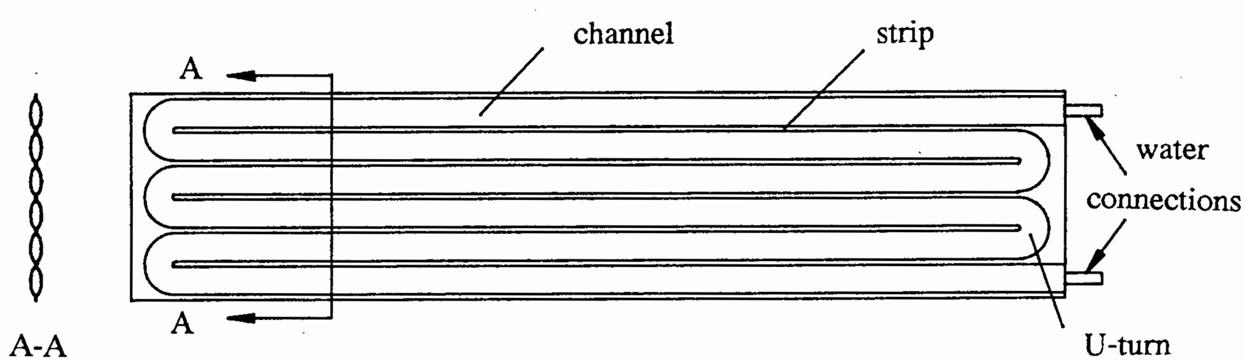


Figure 3.1 A Weideman heat exchanger

3.2 TECHNICAL DATA

The aim of the work done on the Weideman heat exchanger was to provide the manufacturers with technical data on the overall heat transfer coefficient as well as the effectiveness of the heat exchanger which in turn will assist the wine industry with the problem of selecting equipment for removing the heat created by the fermenting juice. As far as it can be determined from the literature on the subject, a prediction of the overall heat transfer coefficient has not been attempted in the past. The reason is perhaps that up to now there were no relationships available to enable the calculation of the free convection heat transfer coefficient between fermenting grape juice and the outside surface of this particular heat exchanger.

3.3 THEORETICAL MODEL

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

In order to simplify the analysis of the heat transfer process, the heat exchanger is modelled as being one-dimensional in nature (figure 3.2). Heat is transferred by natural convection on the outside of the channel, between the juice temperature (T_j) and the wall temperature (T_{wo}). The heat is conducted through the stainless steel between T_{wi} and the outside wall temperature (T_{wo}). Finally the heat is transferred by forced convection inside the channel, between the inside wall temperature (T_{wi}) and the mean bulk temperature of the cooling water (T_{wb}). The overall heat transfer coefficient will be between the bulk juice temperature to the bulk water temperature.

3.3.1 Energy balance

The cooling channel

The electrical analogy [11, pp. 34-36] indicated in figure 3.2 displays the resistances involved when heat flows from the juice to the cooling water through the cooling channel wall. (Part of the strips on the sides of the cooling channel are also displayed.)

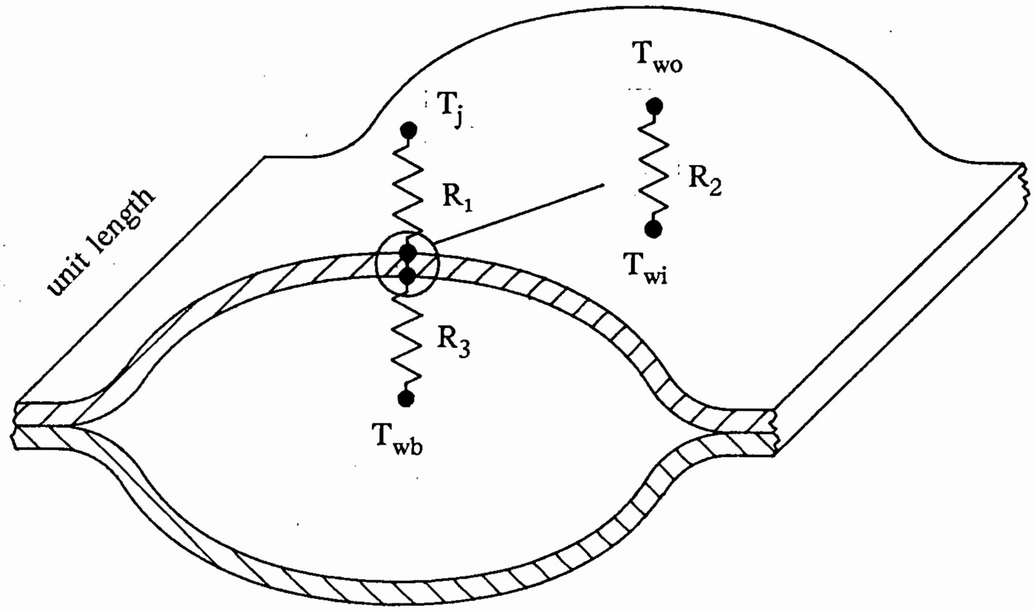


Figure 3.2 Electrical analogy of the one-dimensional heat transfer between the bulk juice temperature to the bulk water temperature.

i) resistance to heat transfer between the bulk juice temperature (T_j) and the channel outside surface temperature (T_{wo})

$$R_1 = \frac{1}{h_o A_{co}} \tag{3-1}$$

where h_o is natural or free convection and A_{co} is the outside channel area made up by the outside perimetry per unit length

ii) resistance to heat transfer through the channel wall

$$R_2 = \frac{t}{kA_c} \tag{3-2}$$

where k is the conduction through the channel wall with thickness t and A_c is the mean channel area (the average between the inside and outside area)

iii) resistance to heat transfer between the channel inside surface (T_{wi}) and the cooling water (T_{wb})

$$R_3 = \frac{1}{h_i A_{ci}} \tag{3-3}$$

where h_i is forced convection coefficient and A_{ci} is the inside channel area made up by the inside perimetry per unit length.

The heat transfer per unit length of channel, if the strips are neglected,

$$\frac{Q}{L} = \frac{\Delta T_{overall}}{\sum R_{th}} \tag{3-4}$$

$$\frac{Q}{L} = \frac{T_j - T_{wb}}{R_1 + R_2 + R_3} \tag{3-5}$$

where T_{wb} is the average bulk temperature of the cooling water entering and leaving the heat exchanger, [11, pp. 249-250]

$$T_{wb} = \frac{T_{cwi} + T_{cwo}}{2} \tag{3-6}$$

Due to the high conductance of stainless steel and the thin wall of the channel, R_2 is much smaller than R_1 or R_3 . This has the effect that the inside and outside wall temperatures differ very little and it is reasonable then to assumed that they have the same value (T_{wall}).

The strip

There are all together 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 3.3 displays one of these fins per unit channel length.

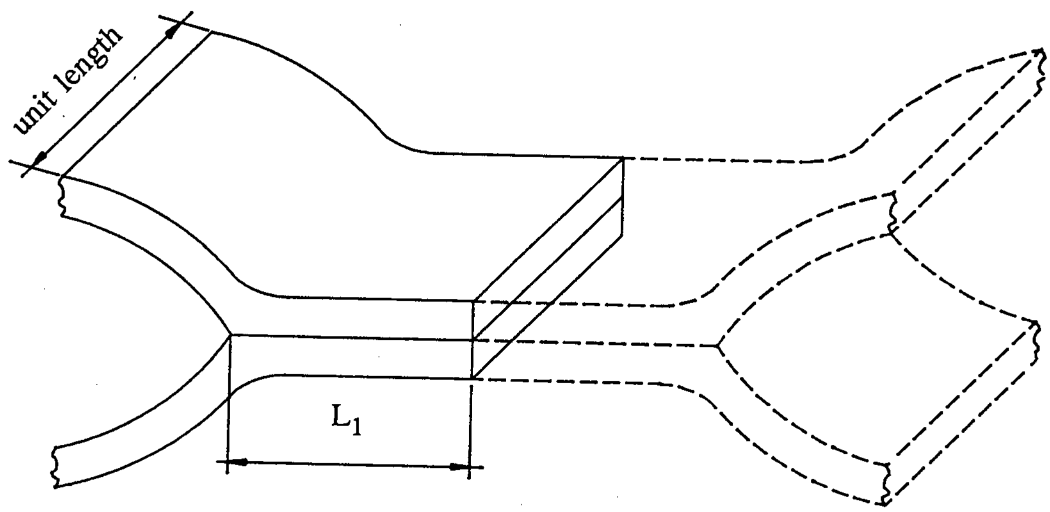


Figure 3.3 Half a strip between two cooling channels.

The heat transfer per unit channel length of these fins according to [11, pp.43-49] is calculated by

$$Q_{s1} = \sqrt{h_o P_{s1} k A_{s1}}\ (T_j - T_w) \tanh m_1 L_1 \tag{3-7}$$

where h_o is the outside convection coefficient, P_s the perimeter of the fin, k the thermal conductivity of the fin, A_s the cross sectional area of the fin, L_1 the length of the fin (half the strip's width), while m is defined as

$$m = \sqrt{\frac{h_o P_s}{k A_s}} \tag{3-8}$$

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. Figure 3.4 displays one of these fins per unit channel length.

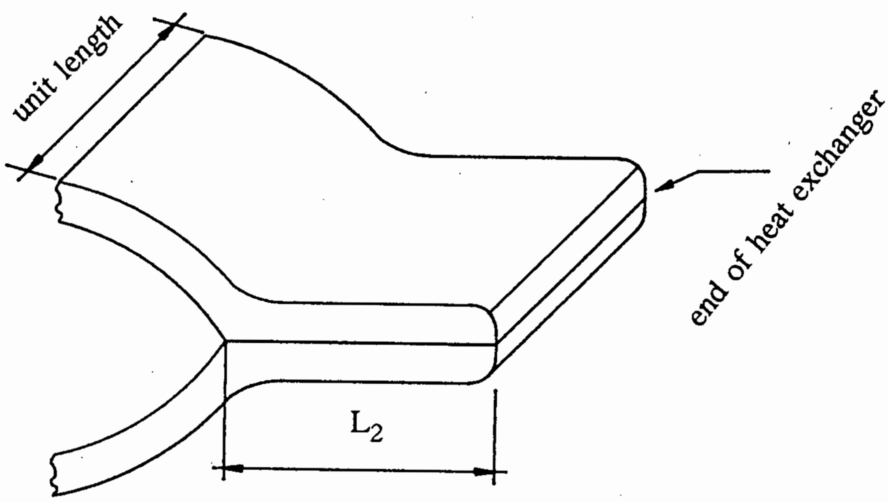


Figure 3.4 A strip on the end of the heat exchanger.

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

$$Q_{s2} = \sqrt{h_o P_{s2} k A_{s2}} \; (T_j - T_w) \; \frac{\sinh m_2 L_2 + (h_o/m_2 k)\cosh m_2 L_2}{\cosh m_2 L_2 + (h_o/m_2 k)\sinh m_2 L_2} \tag{3-9}$$

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

$$Q_s = 10 Q_{s1} + 2 Q_{s2} \tag{3-10}$$

Combination of the channel and strips

The electrical analogy for the heat transfer that takes place between the juice and the cooling water for the combination of the channel and strip is displayed in figure 3.5.

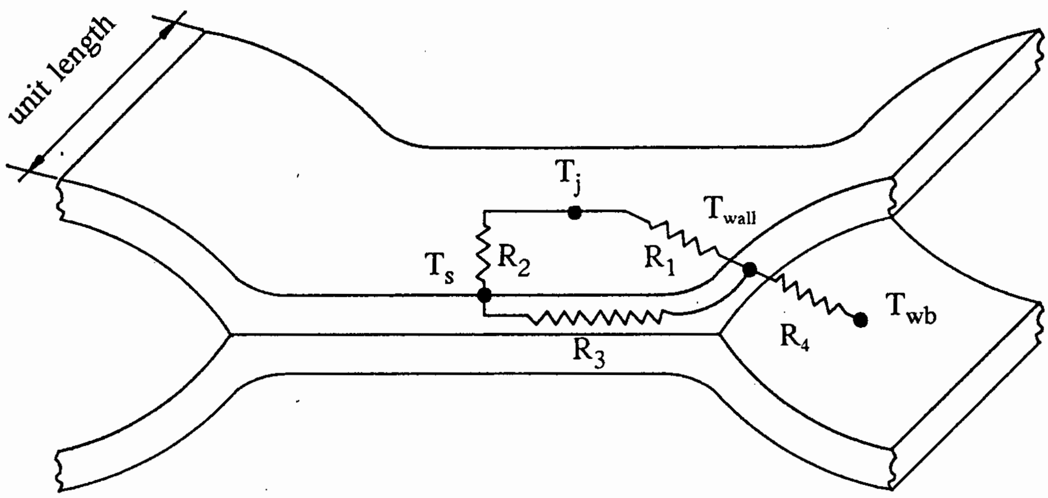


Figure 3.5 Electrical analogy of the one-dimensional heat transfer between the juice and the cooling water for the channel and its strips.

R_1 is the resistance between the bulk juice temperature (T_j) and the channel wall temperature (T_{wall}). R_2 is the resistance between the bulk juice temperature and the average surface temperature of the strips (T_s). R_3 is the resistance between T_s and T_{wall} which is the resistance of heat flow in the strip form the surface of the strips to the wall of the cooling channel. R_4 is the resistance of heat flow from the channel wall at T_{wall} to the cooling water at bulk temperature, (T_{wb}).

3.3.2 Calculations

The determination of the outside free convection coefficient

Data collected during the experimental phase of this study are used to calculate the heat transfer rates for the 18 different conditions at which each heat exchanger were tested. The heat transfer can be calculated from the temperature rise of the cooling water circulated through the heat exchanger, using the following equation

$$Q = m_w C_{pw} (T_{cwo} - T_{cwi}) \qquad (3\text{-}11)$$

Note: The assumption is made that 1 litre of water has a mass of 1 kg and the specific heat (C_{pw}) is taken as 4.197 kJ/kg °C at 10 °C

The results of the calculations are graphically displayed in the following figures.

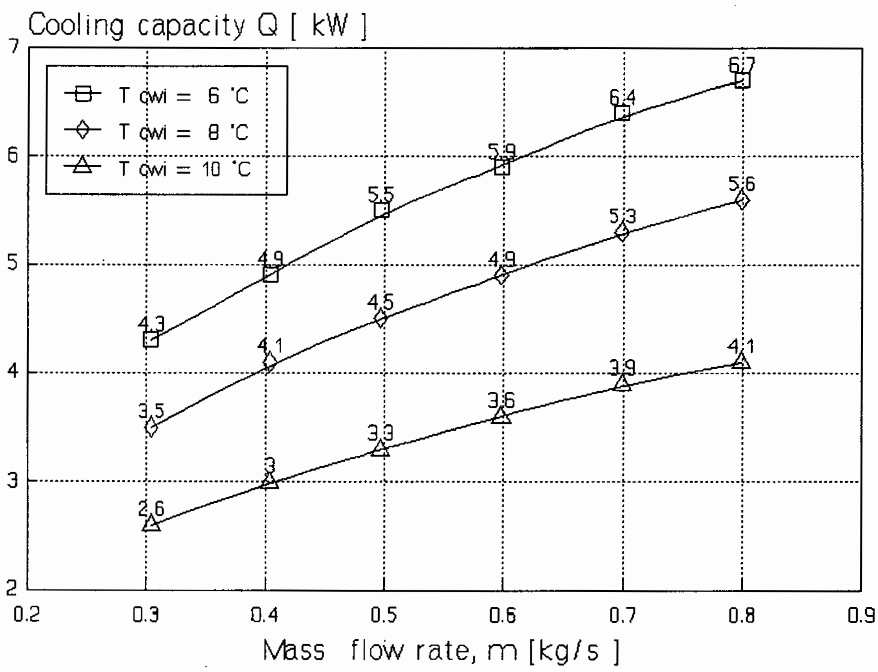


Figure 3.6 Cooling capacity of a 400x1820 mm Weideman heat exchanger.

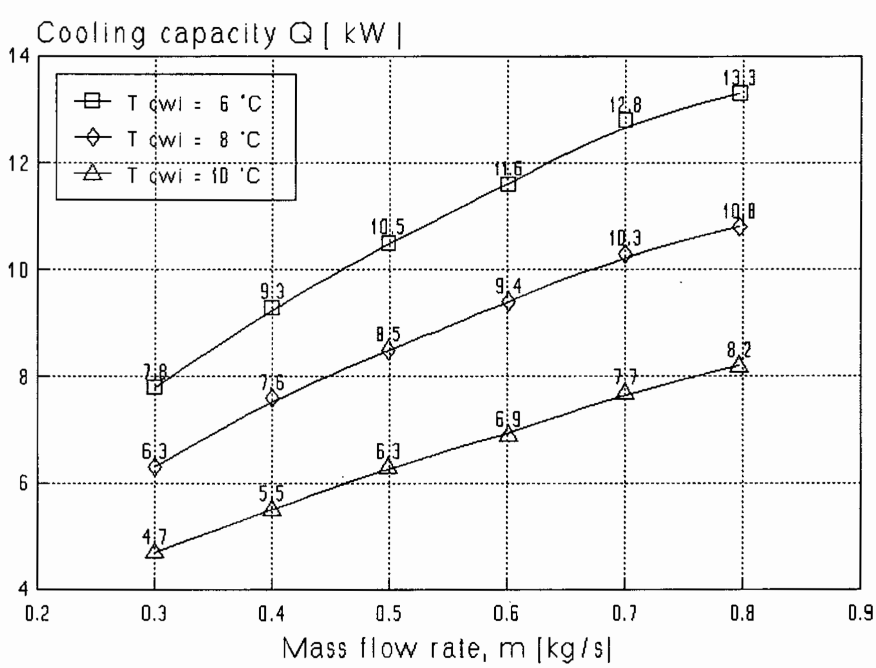


Figure 3.7 Cooling capacity of a 400x4550 mm Weideman heat exchanger.

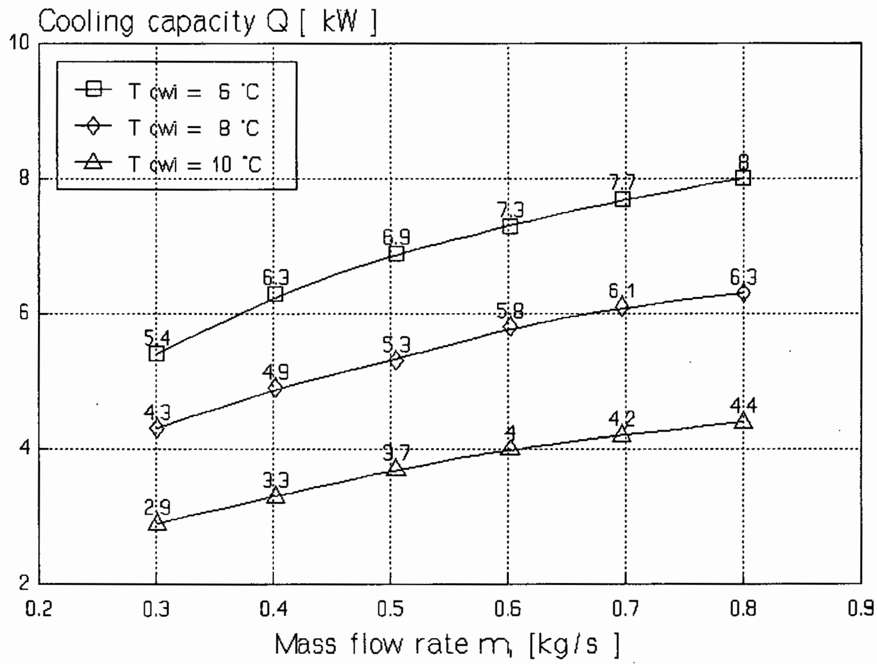


Figure 3.8 Cooling capacity of a 300x2490 mm Weideman heat exchanger.

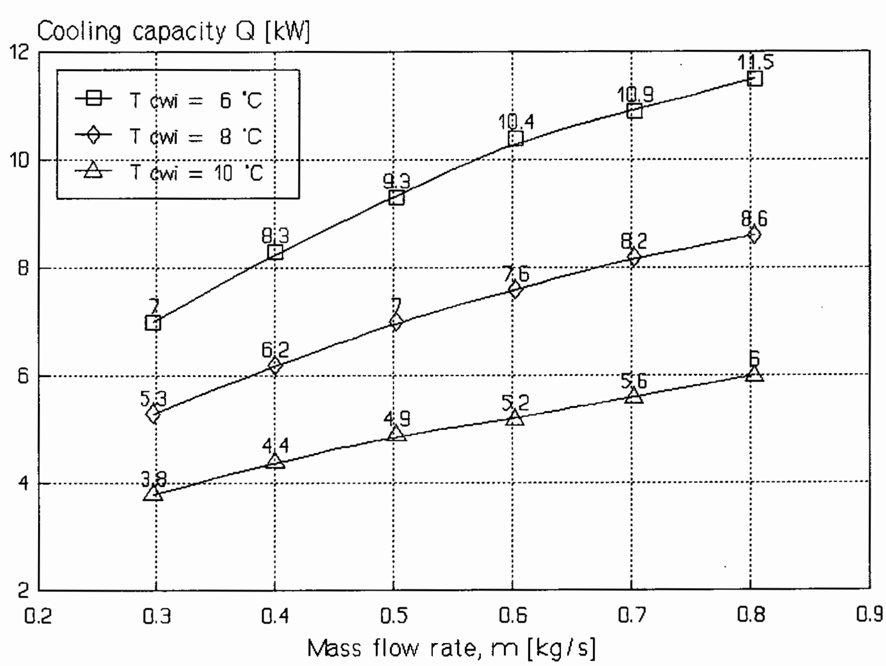


Figure 3.9 Cooling capacity of a 300x3485 mm Weideman heat exchanger.

The water bulk temperature is calculated, using equation (3-6) repeated here,

$$T_{wb} = \frac{T_{cwi} + T_{cwo}}{2}$$

Assuming a value for the temperature of the channel wall (T_{wall}), the film temperature between the wall and the cooling water is calculated in accordance with

$$T_f = \frac{T_{wall} + T_{wb}}{2} \tag{3-12}$$

The average Nusselt number inside the channel is calculated using the following equation which was developed by Petukhov [12, pp. 3.2.11] (viscosity variations are neglected).

$$Nu_i = \frac{\left[\frac{f}{8}\right] Re_w\, Pr_w}{1.07 + 12.7\sqrt{\frac{f}{8}}\left(Pr_w^{0.67} - 1\right)} \tag{3-13}$$

The inside forced convection heat transfer coefficient is then determined

$$h_i = \frac{Nu_i k_w}{de} \tag{3-14}$$

and finally the heat transfer is calculated from

$$Q = h_i A_i (T_{wall} - T_{wb}) \tag{3-15}$$

If the value of the heat transfer Q, determined by equation (3-11) does not correspond to the one calculated from the above equation then the value of T_{wall} is increased or decreased to better the situation and the calculations repeated until agreement is obtained. Once T_{wall} is determined the outside convection coefficient is determined iteratively from the following equation:

$$Q = h_o A_o (T_j - T_{wall}) + Q_s \tag{3-16}$$

where the first term is in accordance with the definition of natural convection heat transfer from a fluid to a surface, and the second term is in accordance with equation (3-10). The average outside convection heat transfer coefficient (h_o) is eventually needed to calculate the outside Nusselt number (Nu_o) for the whole heat exchanger. First of all the average surface temperature of the strip (T_s) is determined using the following relation

$$T_s = T_j - \frac{Q_s}{A_s h_o} \tag{3-17}$$

where A_s is the total strip area

The channel and strip temperatures and areas are now combined in the following equation to determine the average wall temperature ($\overline{T_{wall}}$) of the whole heat exchanger.

$$\overline{T_{wall}} = \frac{A_c T_{wall} + A_s T_s}{A_c + A_s} \tag{3-18}$$

The average outside convection heat transfer coefficient ($\overline{h_o}$) is now calculated from the following equation

$$\overline{h_o} = \frac{Q}{A_T(T_j - \overline{T_{wall}})} \tag{3-19}$$

where A_T is the total outside area of the heat exchanger (including channels and strips)

A new film temperature is calculated, using a similar equation as (3-12), which is used to evaluate the properties of the juice surrounding the outer surface of the heat exchanger.

$$T_f = \frac{\overline{T_{wall}} + T_j}{2}$$

The average outside Nusselt numbers were calculated

$$\overline{Nu_o} = \frac{\overline{h_o} W_h}{k_j} \tag{3-20}$$

where W_h is the width of the heat exchanger

The results are tabulated in tables B.20 to B.23 in appendix B.

The Prandtl numbers were calculated using the same film temperature to evaluate the fluid properties

$$Pr = \frac{C_p \mu}{k} \tag{3-21}$$

where C_p is the specific heat, μ is the absolute viscosity and k is the thermal conductivity of the juice.

Finally the corresponding Grashof numbers were calculated

$$Gr = \frac{g\beta (T_j - T_{wall}) W_h^3}{\nu^2} \tag{3-22}$$

where g is the acceleration of gravity, β is the volume coefficient of expansion, ν is the kinematic viscosity and W_h is the width of the heat exchanger.

The products of Grashof-Prandtl numbers ($GrPr$) are tabulated in tables B.24 to B.27 in appendix B.

The following power law relationship between the Grashof-Prandtl products and the average outside Nusselt numbers is assumed, with the constant C and exponent m to be determined from the data, contained in tables 3.1 to 3.8.

$$\overline{Nu_o} = C(Gr\,Pr)^m \tag{3-23}$$

The values of $\log(\overline{Nu_o})$ and $\log(GrPr)$ from all the heat exchangers' experimental data are plotted in figure 3.10.

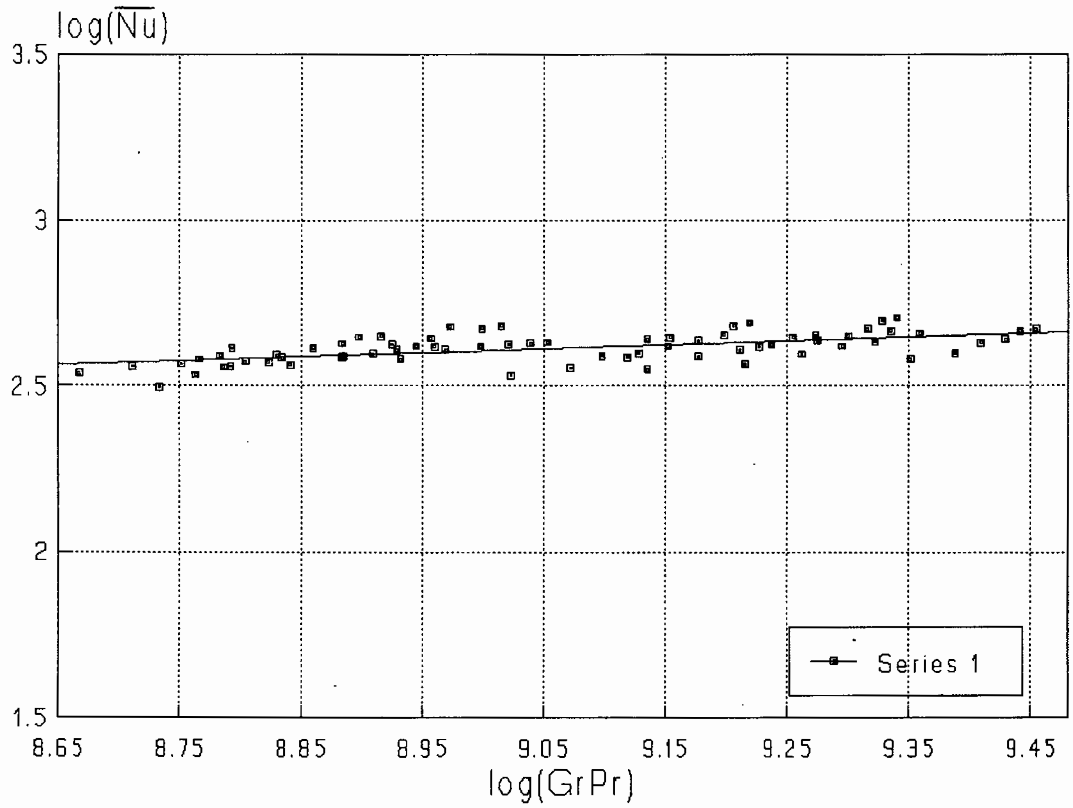


Figure 3.10 Free-convection heat transfer correlation between a cold surface of a given geometry immersed in fermenting grape juice .

A straight line drawn through the data, yielded values for the constant C=34.1 and exponent m=0.12. The resulting power law relationship is

$$\overline{Nu_o} = 34.1\,(Gr\,Pr)^{0.12} \qquad (3\text{-}24)$$

WHICH IS A NEW RESULT !

The overall heat transfer coefficient (U) and effectiveness (ϵ) of the heat exchanger

The calculated values for the inside forced convection coefficient h_i and the outside free convection coefficient $\overline{h}_o$ are now used to determine the overall heat transfer coefficient, U. As stated earlier due to the high conductance and the small wall thickness of the channel,, the conductance through the channel walls was neglected. Fouling can be neglected as well, because all the heat exchangers that were tested were brand new. Keeping in accordance with accepted practice the overall heat transfer coefficients based on the outside area of the heat exchangers were calculated using

$$U_o = \frac{1}{\frac{A_T}{A_i}\frac{1}{h_i} + \frac{1}{\overline{h}_o}} \tag{3-25}$$

The results are displayed in the following figures.

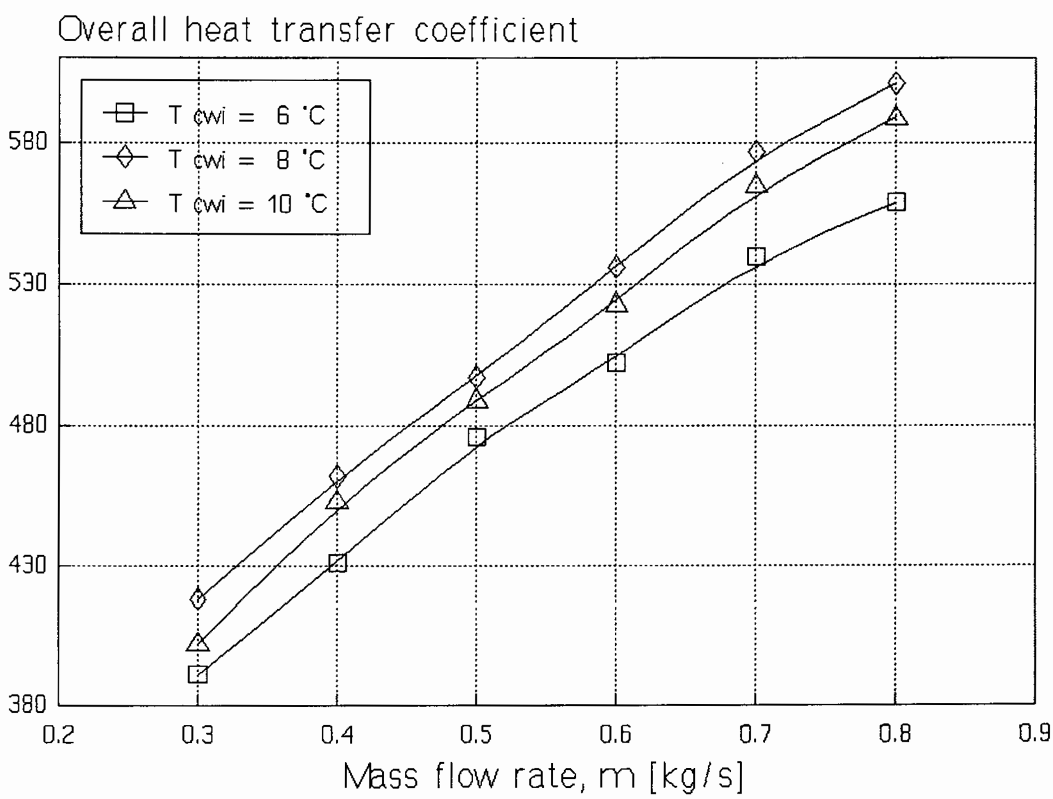


Figure 3.11 The overall heat transfer coefficient (U_o) for the 400x1820 mm Weideman heat exchanger.

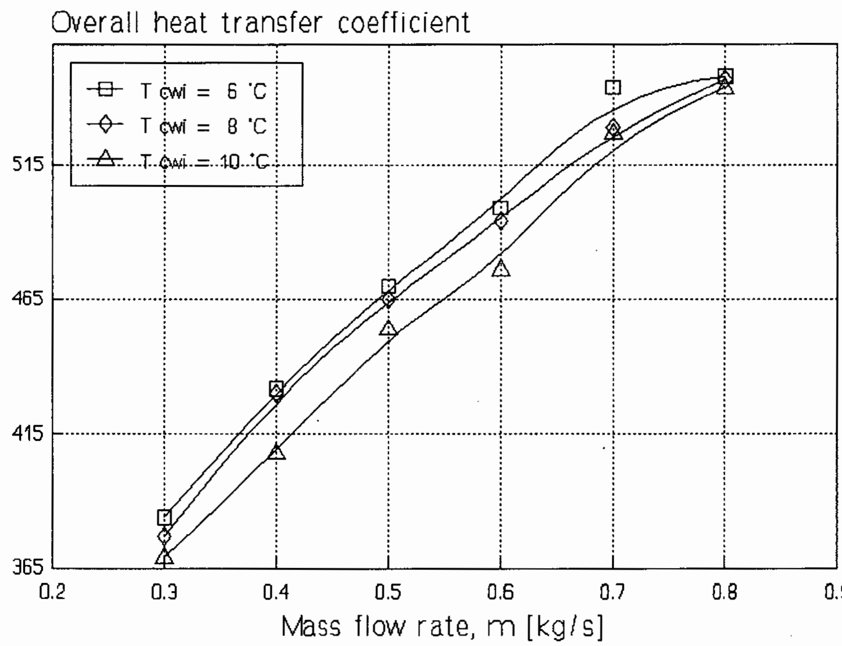


Figure 3.12 The overall heat transfer coefficient (U_o) for the 400x4550 mm Weideman heat exchanger.

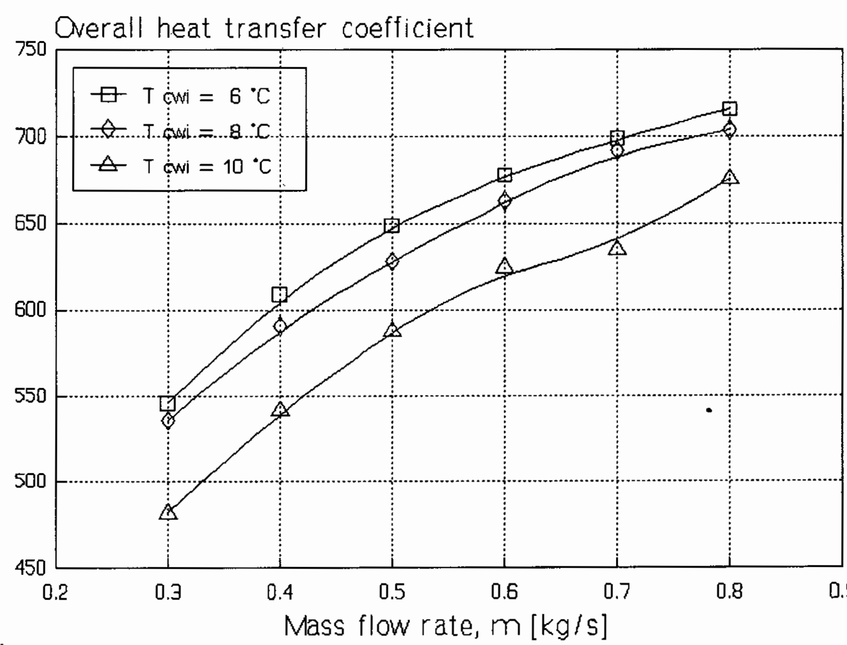


Figure 3.13 The overall heat transfer coefficient (U_o) for the 300x2490 mm Weideman heat exchanger.

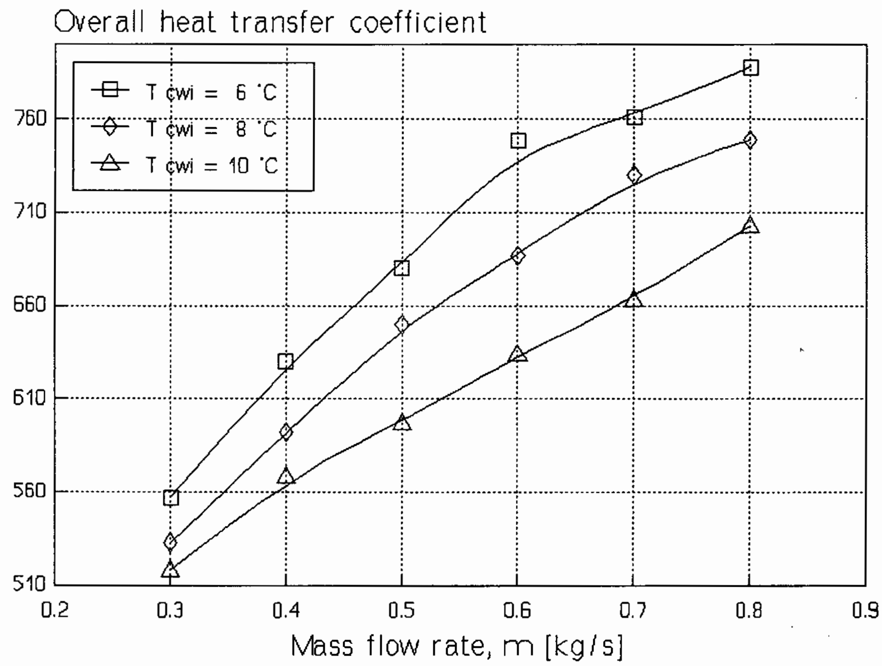


Figure 3.14 The overall heat transfer coefficient (U_o) for the 300x3485 mm Weideman heat exchanger.

The number of transfer units (*NTU*) were calculated from, [11, pp.547-559]

$$NTU = \frac{U_o A_o}{C_{\min}} \tag{3-26}$$

where C_{min} is by definition, the fluid with the minimum value of the product of mass flow rate and specific heat. However in this case since the capacity ratio is

$$\frac{C_{\min}}{C_{\max}} = 0$$

because the temperature of the juice does not change, the minimum fluid is the cooling water circulating through the heat exchanger. The *NTU* values calculated are plotted against the mass flow rate and are graphically displayed in figures 3.15 - 3.18.

Because fermenting juice acts as if it had infinite specific heat, the effectiveness of the heat exchanger is given by the following equation [11, pp.548]

$$\epsilon = 1 - e^{-NTU} \tag{3-27}$$

The effectiveness of the four Weideman heat exchangers tested, are plotted against the number of transfer units and the result is displayed in figures 3.15 to 3.18.

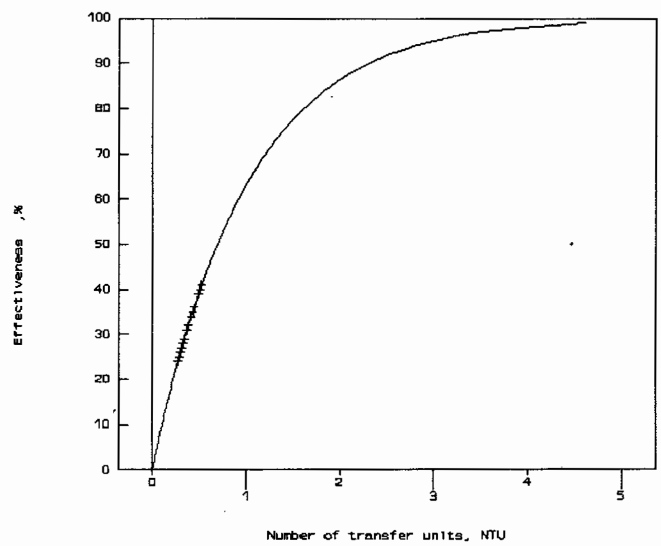


Figure 3.15 The effectiveness of the 400x1820 mm Weideman heat exchanger.

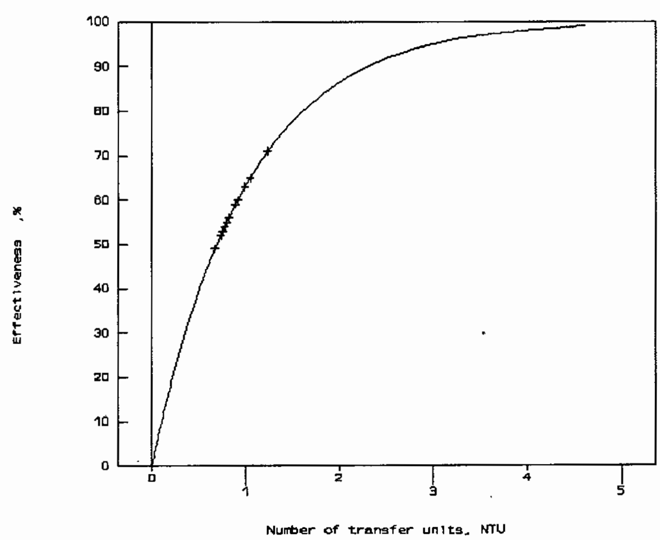


Figure 3.16 The effectiveness of the 400x4550 mm Weideman heat exchanger.

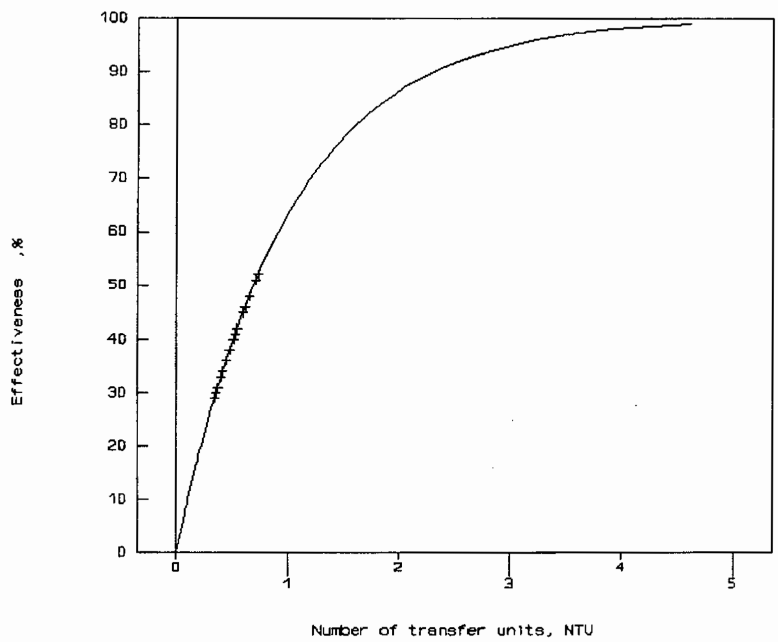


Figure 3.17 The effectiveness of the 300x2490 mm Weideman heat exchanger.

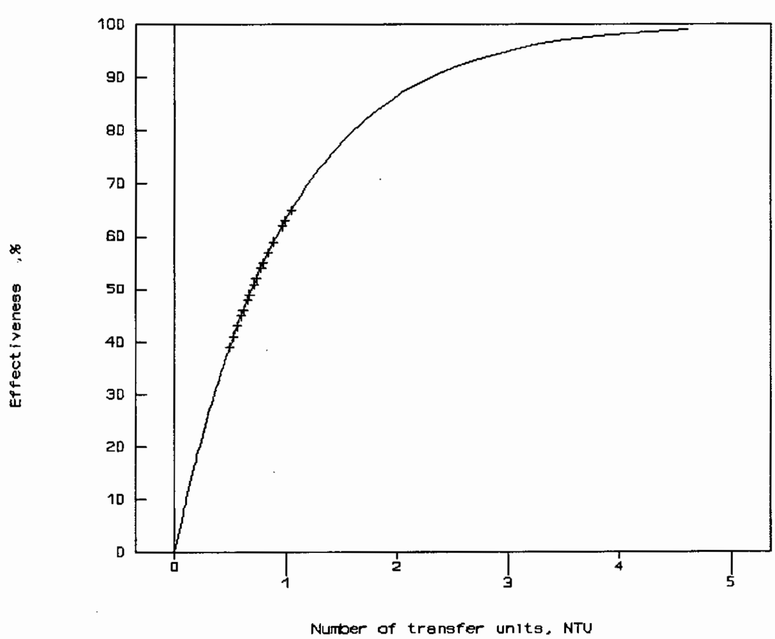


Figure 3.18 The effectiveness of the 300x3485 mm Weideman heat exchanger.

3.3.3 Pressure loss

The pressure loss through any of the Weideman heat exchangers that were tested can be theoretically predicted using the Darcy-Weisbach equation [14, pp. 196-197]

$$\Delta p = \rho_w \left[\left(f_{d1} \frac{L_c}{de} + \sum_{i=5}^{9} k_i \right) \frac{v_{w1}^{\ 2}}{2} + \left(f_{d2} \frac{L_p}{d_p} + \sum_{i=1}^{4} k_i \right) \frac{v_{w2}^{\ 2}}{2} \right] \tag{3-28}$$

where v_{w1}, f_{d1} *and* v_{w2}, f_{d2} denote the water velocity and friction factor in the cooling channel (station 1) and the circular inlet and outlet (station 2) respectively. The locations of high flow disturbance, which effect a significant contribution in pressure loss, are depicted in Figure 3.19. These contributions to the total pressure loss are represented in equation (3-28) by the appropriate pressure loss coefficients K_1 to K_9.

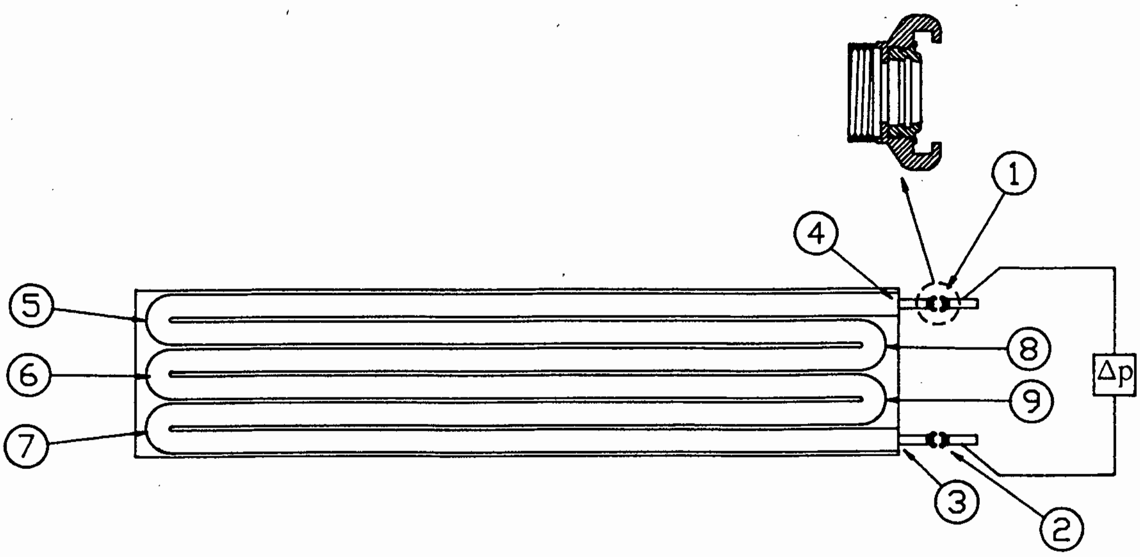


Figure 3.19 Pressure loss locations on Weideman heat exchanger with quick couplings on the inlet and outlet

Values for K_3 and K_4 were taken at 0.5 each [14, pp. 220], while K_5 to K_9 were taken at 2.2 each [14, pp. 227]. No data were available on the values of K_1 and K_2 which are the pressure loss coefficients through the quick coupling connections. *(In the quick coupling the water flows through a sharp edge metal ring of ± 20 mm, then through a thick rubber seal which decrease in diameter after the coupling is made and lastly through another sharp edge metal ring of ± 20 mm.)* It was assumed that the values of K_1 and K_2 are equal, making them the only unknowns if the total pressure loss is known. K_1 and K_2 were determined to be 0.86 using experimental data for two heat exchangers. (appendix B tables B.13 to B.16)

The general pressure drop equation for the Weideman heat exchanger and quick couplings as displayed in figure 3.19, is:

$$\Delta p = \rho_w \left[\left[f_{d1} \frac{L_c}{de} + 11 \right] \frac{v_{w1}^{\;2}}{2} + \left[f_{d2} \frac{L_p}{d_p} + 1.72 \right] \frac{v_{w2}^{\;2}}{2} \right] \tag{3-29}$$

A computer program was written in Turbo Pascal (see appendix D) and used to execute the calculations involved in the above equation.

3.4 EXPERIMENTAL PHASE

3.4.1 Apparatus

Electronic sensors were used for measuring temperatures, the water flow rate and differential pressures. The sensors were connected to a data logging system ("black box") which is controlled from a Personal-computer via a Turbo Pascal computer program.

Plate 3.1 The Personal-computer with program display and the "black box".

Temperatures were recorded by resistance temperature devices (RTD's) which are very stable and have an accuracy of ± 0.1 °C. As shown in plate 3.2 the RTD's were installed into T-pieces for direct contact with the cooling water circulating through the heat exchanger.

Plate 3.2 An installed RTD.

The water flow meter utilizes a turbine which generates electronic pulses through the action of a magnet on a Reed switch. The accuracy of the flowmeter was given by the manufacturers ± 2 % . It was installed in the water return line to the heat exchanger. Plate 3.3 displays a turbine flow meter installed in a water pipeline.

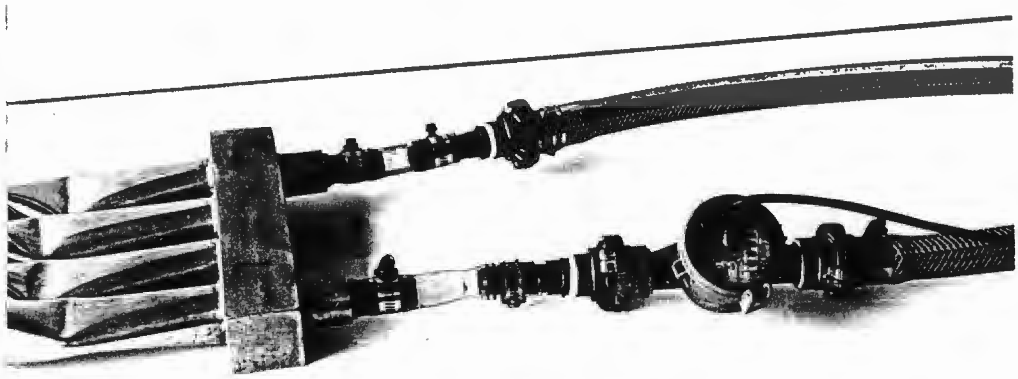


Plate 3.3 The turbine flow meter installed in the return cooling water pipeline.

The differential pressure transducer was connected across the heat exchanger and sensed the difference in pressure at the delivery and return pipes. Plastic pipes, filled with water, were used as connections between transducer and tapping off points. The delivery and return pressures are applied respectively on either side of a spring diaphragm inside the pressure transducer. When the two pressures differ, the diaphragm is deflected away from the higher pressure. Movement of the diaphragm is transmitted through low friction seals to activate a Hall effect sensor. General accuracy is $\pm$ 2 % full scale. Plate 3.4 displays the differential pressure transducer used.

Plate 3.4 Differential pressure transducer

The data logging system operated on commands from a Turbo Pascal program and it could accommodate 16 temperature sensors, three flow meters, one differential pressure transducer and one relative humidity sensor. The sampling frequency could be set from 20 seconds to 24 hours. Experiments could be monitored by watching the PC's video display screen for the values of experimental data as they were being sampled.

3.4.2 Experimental procedure

All heat exchangers were tested in a 581 hecto litre horizontal stainless steel wine tank. The heat exchangers were bolted to stands with their lengths parallel and their widths perpendicular to the horizontal. After the installation of a heat exchanger, the tank was filled with 32 cubic metres of water. Water in a reservoir at a cooling plant was cooled to the desired inlet water temperature (6°, 8°, 10°C). Once the required inlet water temperature was reached, the water was pumped from the reservoir through the heat exchanger and back to the reservoir. The flow rate through the heat exchanger was manually set by a valve in the water pipeline.

A perforated PVC pipe was installed on the bottom of the tank directly underneath the heat exchanger. Using a variable speed pump, water drawn from the tank was pumped through the perforated PVC pipe. This was done to simulate the stirring effect in grape juice during peak fermentation.

The water in the tank, which simulated the juice, was cooled or heated to 15°C before the start of any experiment. The heat exchanger inside the tank was used to cool the water, while 9 kW heating elements were used when heating was required. Once the water in the tank was at the desired temperature, the experiment started and no addition heating or cooling was effected to the water simulating the juice in the tank. Time wise the experiments were to short to remove sufficient heat during an experiment to make a significant change to the temperature of the 32 cubic metres of water in the tank as shown in the following simple calculations.

The largest Weideman heat exchanger was capable of 13.3 kW of cooling at maximum flow rate. (see figure 3.7). Once the cooling water in the reservoir had reached the temperature of (6°, 8° or 10°C) it took approximately half an hour to complete an experiment. Assuming that the tank is a system obeying the lump heat capacity model, one can calculate the expected temperature rise of the "juice" during the duration of the tests.

$$\Delta T = \frac{Q\tau}{\rho V_j C_{pj}}$$

$$= \frac{13300 \cdot 1800}{1 \cdot 32 \cdot 1000 \cdot 4197}$$

$$= 0.178°C$$

where τ is the time and V_j is the volume of the juice.

All instrumentation were calibrated before installation and checked before use. (appendix C) Water inlet and outlet temperatures were monitored for each heat exchanger. The differential pressure was also measured across the inlet and outlet, while the circulating cooling water flow rate was measured in the return pipe line. After completion of each experiment, the test was repeated with the temperature sensors at the inlet and outlet swopped around. This was done to eliminate any drift problems or faulty offsets. Experimental data were sampled at a frequency of twenty seconds.

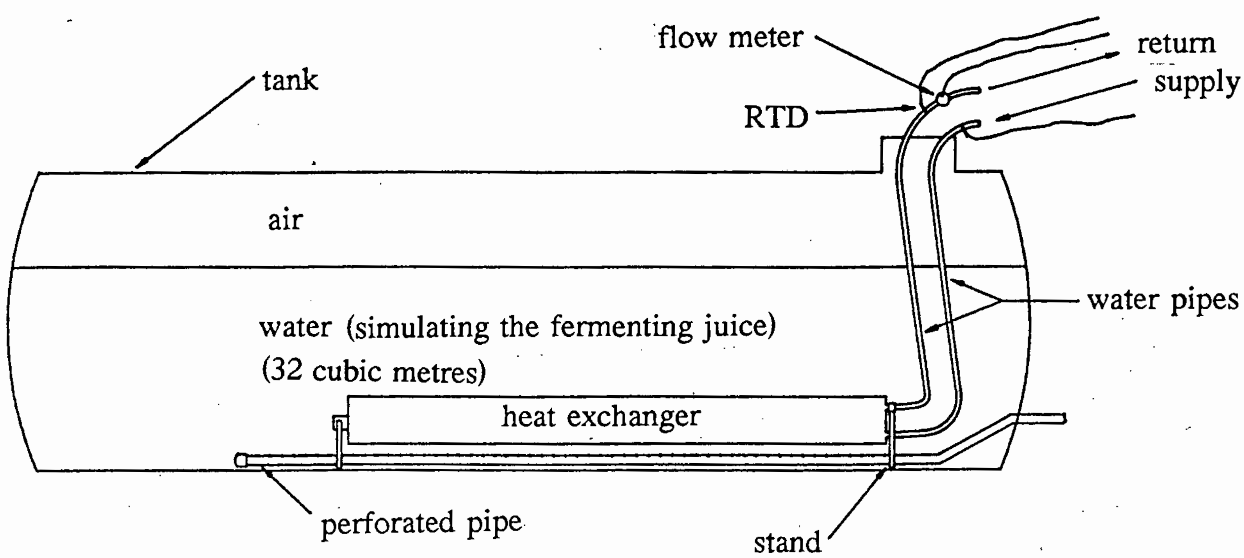


Figure 3.20 Experimental set up.

Experiments were performed at cooling water inlet temperatures of 6, 8 and 10 °C. During each of the experiments, data were sampled at 6 different flow rates. The flow rates were from 0.3 l/s to 0.8 l/s in steps of 0.1 l/s. All experiments were done with the water inside the tank at 15 °C, which is the required average juice temperature to be maintained during the wine making process at peak fermentation conditions.

Note: The water in the tank is referred to as the juice.

3.4.3 Experimental data

Four Weideman heat exchangers with the following sizes 400x1820, 400x4550, 300x2490, and a 300x3500 mm were tested. The experimental data are listed in appendix B: tables B.1 to B.12

3.4.4 Pressure loss

Differential pressure tests were done on the four Weideman heat exchangers. The results of pressure loss through the Weideman heat exchangers at different flow rates are shown in tables B.13 to B.16 in appendix B. The data is displayed graphically in figure 3.21.

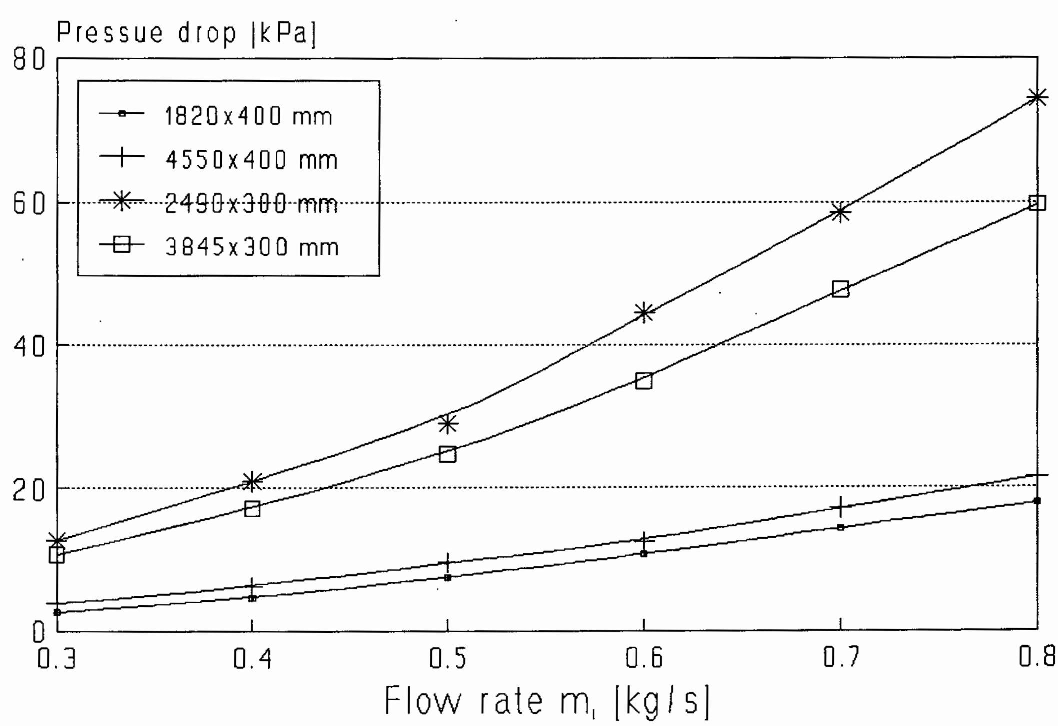


Figure 3.21 Pressure loss versus flow rate for the four Weideman heat exchangers

Flow rate versus line pressure

During the pressure loss tests the supply line pressures were also recorded. The flow rate through a heat exchanger is a function of the pressure in the water supply line. The water flow rates through the Weideman heat exchangers at different pressures in the water supply lines are shown in tables B.13 to B.16 in appendix B. The data are displayed graphically in figure 3.22.

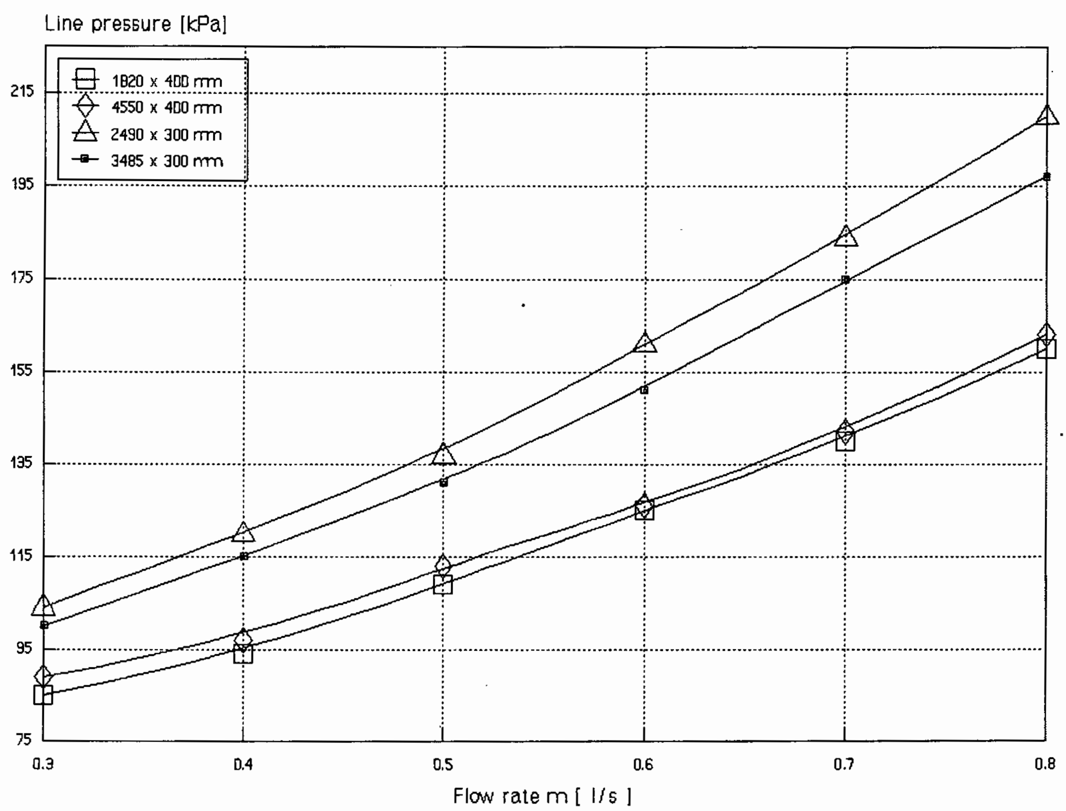


Figure 3.22 Line pressure versus flow rate for the four Weideman heat exchangers

3.5 FOULING

As mentioned earlier the experiments involving the Weideman heat exchangers were performed using clean surfaces. However it was recognized that for completeness it would be desirable to seek a heat exchanger which had been in use during the wine making season. It is a well known fact that heat exchanger surfaces get covered with tartar (a by-product of the fermenting process) and therefore in practice there is a "fouling factor" which must be taken into account when calculating the overall heat transfer coefficient, effectiveness of the heat exchanger, etc.

The author was fortunate to secure a 330x1850 mm Weideman heat exchanger which had just come out of service and was covered with a good layer of tartar ($\pm 450\ \mu m$).

3.5.1 Experimental results

A 300x1850 mm Weideman heat exchanger with its outside surface completely covered with tartar, was used to determine the fouling factor of tartar.

Experimental data were obtained on the fouled heat exchanger when immersed in water (in a 15 000 litre wine tank) at a temperature of $\pm$ 18°C while cooling water at a temperature of $\pm$ 5 °C was pumped through it, for varying flow rates from 0.25 l/s to 0.4 l/s in steps of 0.05 l/s.

Subsequently the heat exchanger was cleaned on the outside and tested under the same thermal conditions as when it was covered with tartar.

The experimental data are listed in table B.20 in appendix B. Equation (3-11) was used to calculate the cooling capacities. The results are displayed in the following figure. (Fig. 3.23).

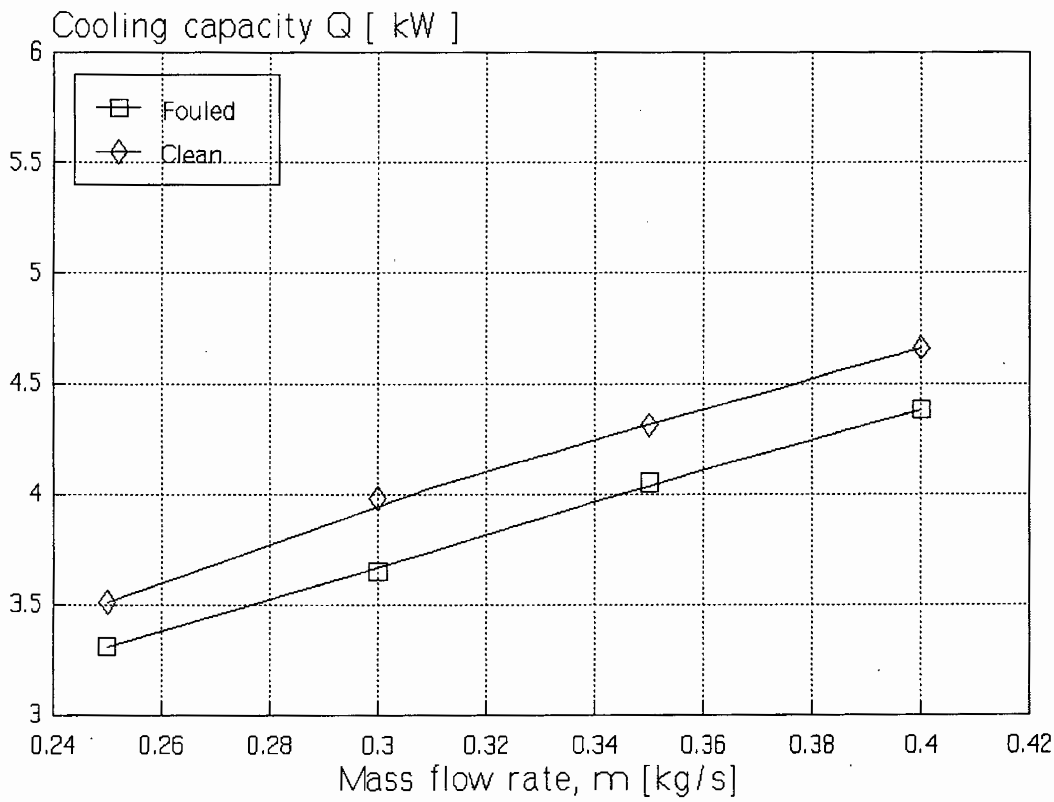


Figure 3.23 Cooling capacity of a 300x1850 mm Weideman heat exchanger, fouled and clean.

Figure 3.24 shows a trend graph of the percentage difference between cooling capacity at different flow rates of the heat exchanger when clean and when covered with tartar.

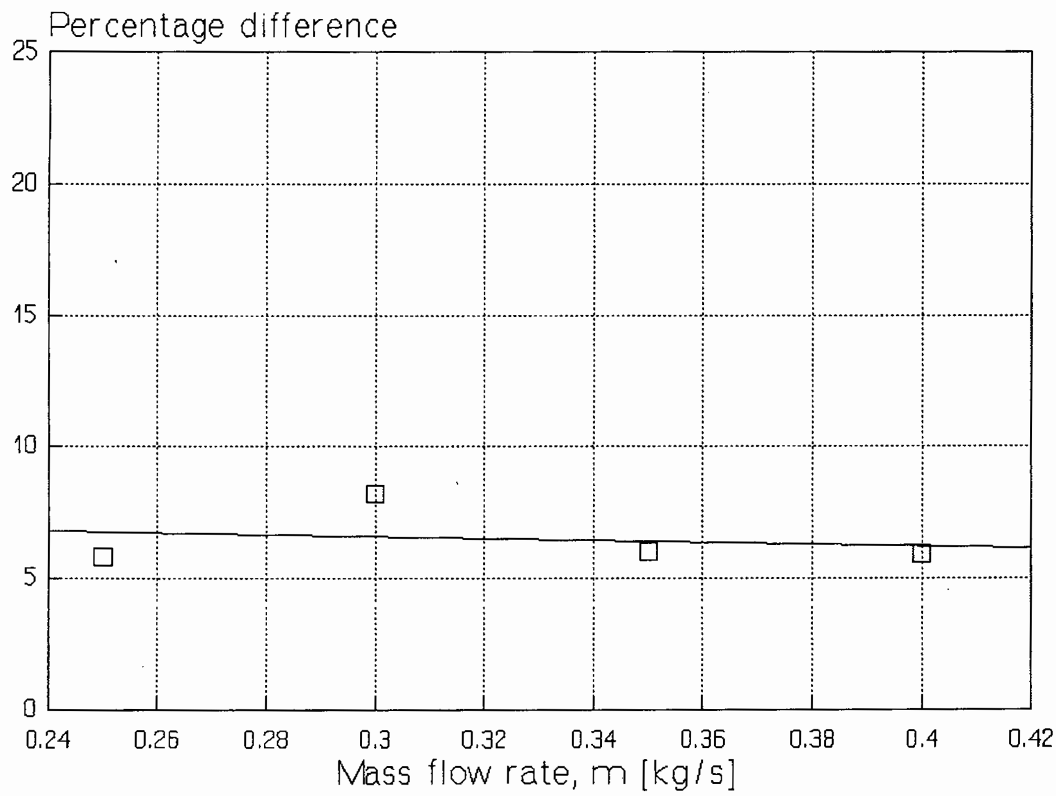


Figure 3.24 Percentage difference between cooling capacity of a 300x1850 mm Weideman heat exchanger when clean and when covered with tartar.

3.5.2 Calculation of the fouling factor [15, pp. 247-248]

The inside and outside convection coefficient as well as the overall heat transfer coefficients were calculated using the method described in section 3.2.2.

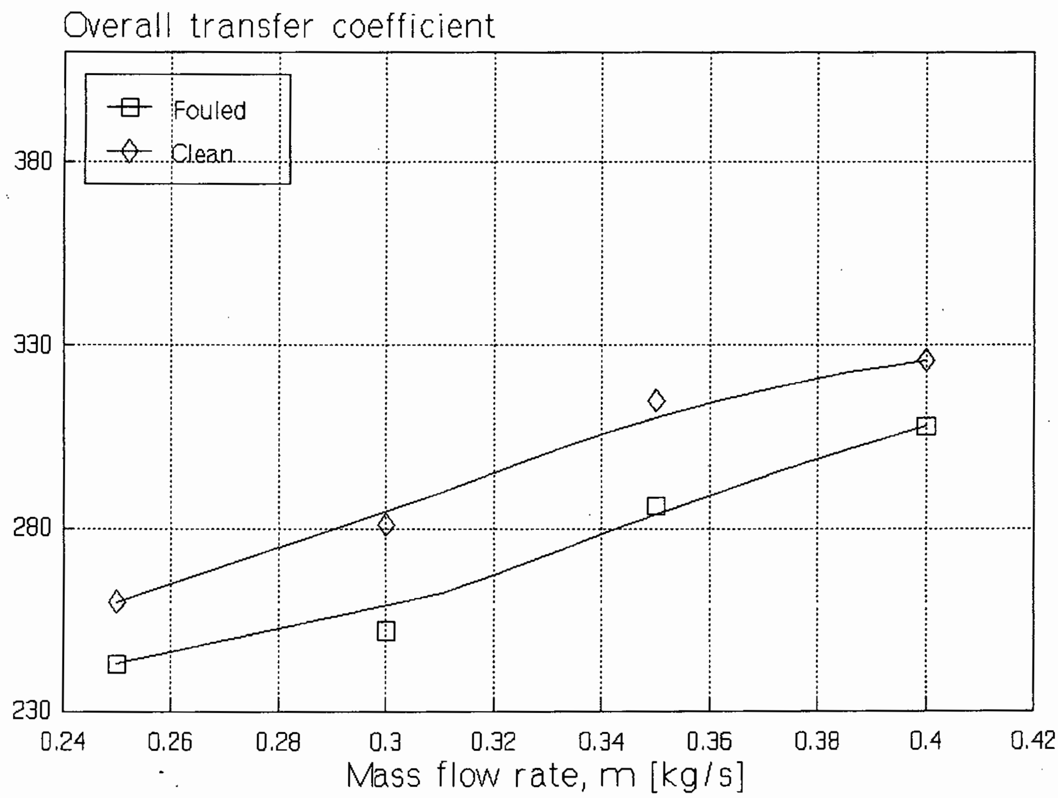


Figure 3.25 Overall heat transfer coefficient for a 300x1850 mm Weideman heat exchanger.

The overall heat transfer coefficient (U) of a fouled surface may be calculated from

$$U_{foul} = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o} + \frac{1}{h_{ff}}} \tag{3-30}$$

The author chose to calculate the fouling factor from

$$\frac{1}{h_{ff}} = \frac{1}{U_{foul}} - \frac{1}{U_{clean}}$$

The results are displayed in figure 3.25

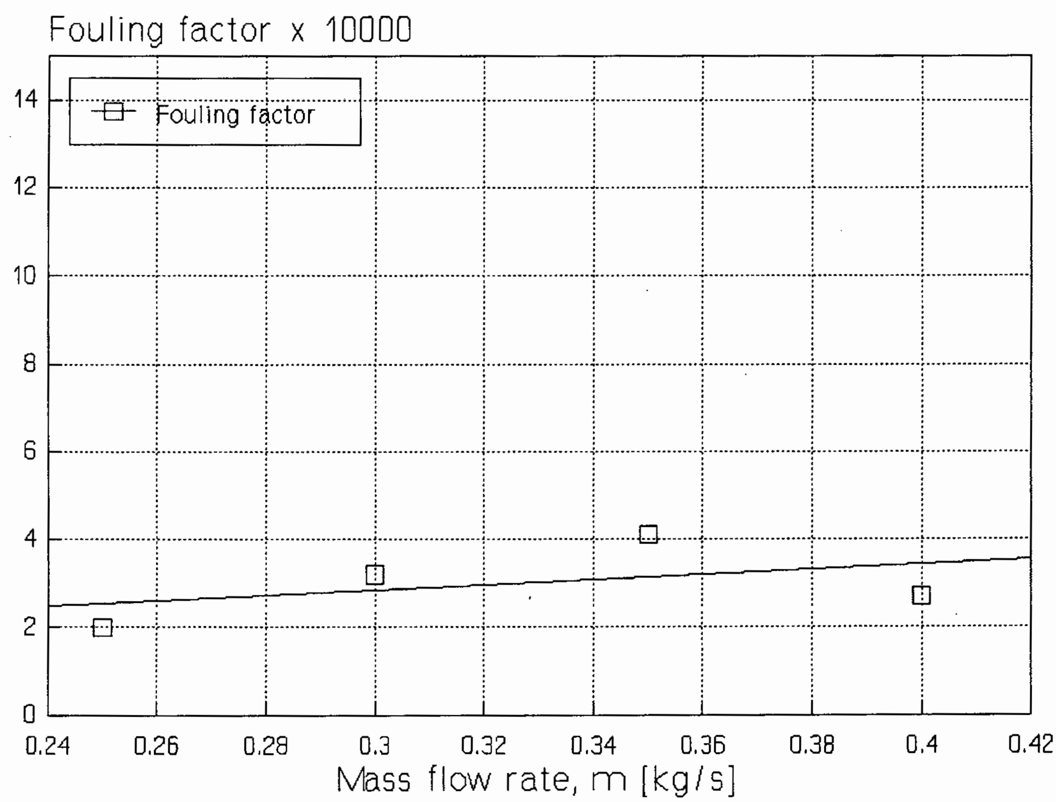


Figure 3.26 Fouling factor of tartar on the outside surface of a heat exchanger.

3.6 DISCUSSION

The preceding sections of the chapter dealt with the theoretical model, experimental procedure and results as they relate to the heat transfer and frictional losses for one particular type of internal heat exchanger often employed in the wine industry. The type of heat exchanger known as the Weideman heat exchanger was extensively tested both for its heat transfer and pressure loss parameters.

In attempting to predict analytically the heat transfer performance of this heat exchanger it was recognized that there were no data available in the heat transfer literature. In particular no predictions could be made of the heat transfer that takes place between the outer surface of this heat exchanger and the fluid that surrounds it.

The author felt it was reasonable to assume that the mode of heat transfer between the exterior surface of the Weideman heat exchanger and the fermenting juice, could be likened to natural convection between a vertical surface and a fluid, while there was no doubt about the mode of heat transfer, being forced convection within the internal passages of the heat exchanger.

In order to follow the accepted practice of presenting heat exchanger performance by the effectiveness or *NTU* method, overall heat transfer coefficients were calculated. In the process of calculating the overall heat transfer coefficients for the Weideman heat exchanger the calorimetric technique coupled with an iterative process was employed.

This approach yielded a power law relationship (Nu versus $C(GrPr)^m$) equation (3-24) for the external flow between the heat exchanger and the fermenting juice, **a new result**.

Experimental data show that tartar deposits on the surface of a heat exchanger reduced the cooling capacity approximately 8.2 %. The author suggests that an allowance of 10 % be made for fouling when cooling capacities are calculated in the wine industry.

Experimental data with regard to pressure drop and mass flow rates for given line pressures delivering cooling water to the Weideman heat exchanger completed the picture. It is now possible, to select or specify optimally a Weideman heat exchanger which will cope with the heat generated at maximum fermentation rate during the wine making process.

EXAMPLE

A wine-maker wants to ferment 41 000 litres of grape juice in a 3.5 m diameter horizontal wine tank with a length of 5 m at a rate of 2 degrees Balling per day, during peak fermentation. (°Balling is an industrial term that refers to a percentage of sugar in a product, on a mass basis).

The juice must be maintained at a temperature of 15°C, while the pressure in the water supply line at the tank is 120 kPa and the cooling water will be available at 9°C.

Fermentation heat

The heat which is generated during peak fermentation is calculated using equation (A-27) as described in appendix A.

$$Q_{ferm} = V_j fr \cdot 6.929 \cdot 10^{-5}$$

$$= 41000 \cdot 2 \cdot 6.929 \cdot 10^{-5}$$

$$= 5.68 \text{ kW}$$

To allow for fouling due to tartar deposit on the heat exchanger outer surface the cooling needed must be increased as recommended by the author by 10%.

$$Q_{ferm} = 5.68 \cdot 1.1$$
$$= 6.25 \text{ kW}$$

Perusing figures 3.6, 3.7, 3.8 and 3.9 the 400x4550 mm and 300x3485 mm Weideman heat exchangers are capable of handling a 6 kW cooling load with a water temperature (T_{cwi}) of 9°C entering the heat exchanger. In figure 3.22, at a supply water line pressure of 120 kPa, the flow rates of the 400x4550 mm and 300x3485 mm Weideman heat exchangers are determined as 0.56 and 0.43 kg/s respectively. Going back to figures 3.7 and 3.9 the expected cooling capacity of the 300x3485 and 400x4550 mm Weideman heat exchanger are ± 5.5 and 7.8 kW respectively. Therefore the 400x4550 mm Weideman heat exchanger is the obvious choice to do the job.

The effectiveness of this heat exchanger performing under the specified conditions can easily be calculated. First of all from figure 3.12 the overall heat transfer coefficient (U_o) is found to be ± 475 W/m^2 °C.

The number of transfer units are now calculated using equation (3-26)

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

$$= \frac{475 \cdot 3.74}{0.56 \cdot 4197}$$

$$= 0.76$$

The effectiveness can now be calculated with equation (3-27) or read from figure 3.16.

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

$$= 1 - e^{-0.76}$$

$$= 0.53 \text{ or } 53\ \%$$

CHAPTER 4
JACKET COOLING

4.1 THE HEAT EXCHANGER

Another popular configuration of wine fermentation cooling is the Jacket tank, where heat is removed from it's contents by pumping cooled water through a channel consisting of a rectangular duct (jacket) welded on the outside surface of the tank. Usually the jacket consists of four rings welded spirally around the tank. The norm is to weld three of the rings on the upper part of the tank and the fourth at the bottom. The top and bottom part of the channel are connected to each other through a circular pipe. When these tanks are used care should be taken to ensure that the water supply line does not exceed a pressure of 120 kPa. Experience has shown that at higher cooling water pressure, due to their geometry, the ducts often sustained permanent damage. Figure 4.1 displays a Jacket tank as well as a cross section of the jacket. The tank, duct and connection pipe are made of stainless steel.

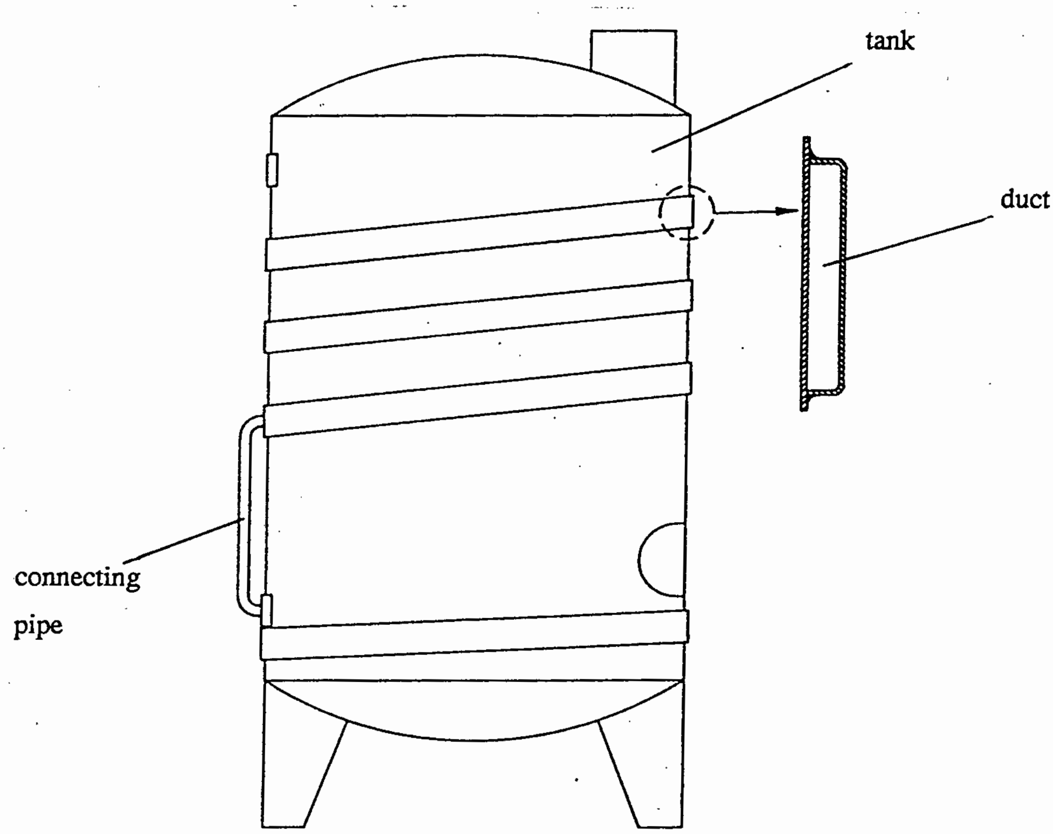


Figure 4.1 Jacket tank and a cross section of a Jacket heat exchanger channel

4.2 THEORETICAL MODEL

It is beyond the scopeof this thesis to analyse the Jacket heat exchanger in a rigorous manner. The approach that the author will take is to obtain the relevant experimental data on a Jacket heat exchanger while under operation during peak fermentation of juice inside the wine tank, and attempt, using relevant relationships in the literature, to approximate the same result. If this is successful the method would be packaged in a manner which would provide guidelines to manufacturers of "Jacket" wine heat exchangers.

Presently there is no standardisation or rigorous manufacturing procedure for such heat exchangers, rather construction of fermenting tanks as they are required in terms of volumetric capacity. Therefore "minimum" recommendations relating to the geometry in the construction of these tanks will be advanced.

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. Heat is transferred to the cooling water from the fermenting juice through the one side along the length of the channel, and provided the ambient temperature is higher than the temperature of the fermenting juice, heat is transferred back from the surroundings to the fermenting juice through the rest of the surface of the tank that is not jacketed. Heat is also transferred from the surroundings to the cooling water which is circulating in the jacket, by free convection between the ambient temperature (T_∞) and the water bulk temperature (T_{wb}). The outside jacket wall is assumed to be at the water-bulk temperature due to the much higher forced convection heat transfer coefficient (and fluid density) inside the jacket than on the outside Jacket surface which is in contact with the surrounding ambient air.

The heat transfer from the fermenting juice to the tank wall (forming the one side of the cooling channel) is by free convection between the juice temperature (T_j) and the tank wall temperature (T_{wall}). Due to the high conductance and small thickness of the wall, the resistance to heat flow through it is insignificant and will be ignored. Heat is transferred from the tank wall to the cooling water by forced convection between the tank wall temperature (T_{wall}) and the water bulk temperature (T_{wb}).

Although the heat transfer from the air and the juice sides to the water circulating in the jacket is by natural convection, the heat transferred from the air is so little in comparison that it has no significant effect on the outside wall temperature of the jacket, while this is not the case on the juice side of the jacket.

4.2.1 Energy balance

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.

It is evident that the portion of the tank which is exposed to the ambient will contribute heat to the fermenting juice provided of course that the ambient temperature is higher than the temperature of the juice required to be maintained. In nearly all cases the ambient temperature is expected to be higher than the fermentation temperature, if however it is not then there will obviously be heat loss from the juice to the environment. Regardless which case may be, this portion of the heat transfer, provided it could be reasonably estimated, could form part of the total heat load that the "Jacket" would have to handle. In other words the ambient's positive or negative contribution to the fermentation heat would be accommodated in the circulating water bulk temperature rise because it would be part of the total requirement.

The electrical analogy [11, pp. 34-36] indicated in figure 4.2 displays the resistances involved when heat is transferred from the surroundings to the cooling water on the one side, while heat is transferred from the fermenting juice through the tank wall to the cooling water on the opposite side of the cooling channel.

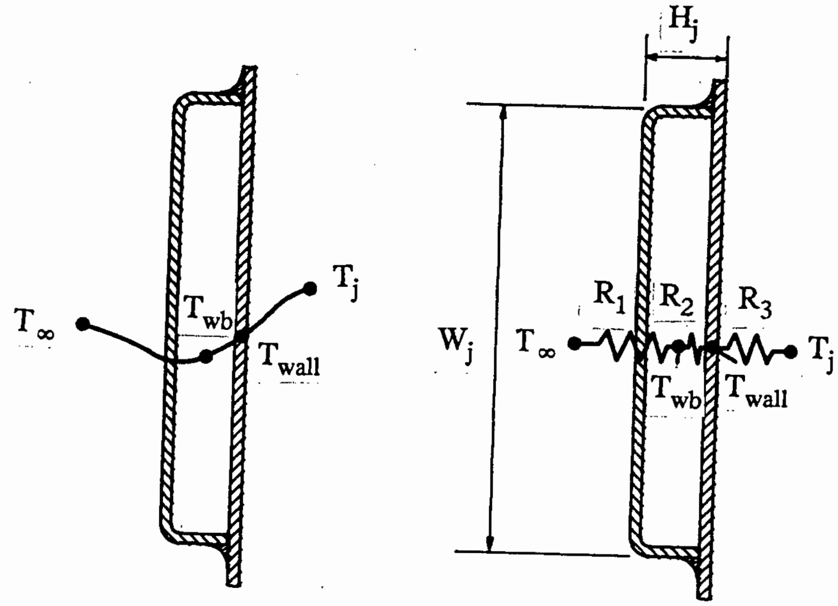


Figure 4.2 Electrical analogy and thermal circuit of the one-dimensional heat transfer between the ambient temperature and the juice temperature to the water bulk temperature.

i) resistance to heat transfer between the ambient temperature (T_∞) and the water balk temperature (T_{wb})

$$R_1 = \frac{1}{h_a A_{ja}} \tag{4-1}$$

where h_{air} is the free convection heat transfer coefficient and A_{ja} is the area of the Jacket that is exposed to the surroundings. This area is the product between the length of the Jacket (L_j) and the Jacket width plus twice its height ($W_j + 2H_j$).

ii) resistance to heat transfer between the bulk water temperature (T_{wb}) and the tank wall temperature (T_{wall}).

where A_{jt} is the tank area that forms the one side of the channel, and is made up by the product of the Jacket width (W_j) and its length (L_j), h_w is the forced convection heat

$$R_2 = \frac{1}{h_w A_{jt}} \tag{4-2}$$

transfer coefficient between the cooling water and the portion of the tank that makes up the jacket wall.

iii) resistance to heat transfer between the mean tank wall temperature (T_{wall}) and bulk juice temperature (T_j)

$$R_3 = \frac{1}{h_j A_{jt}} \tag{4-3}$$

where h_j is the natural or free convection heat transfer coefficient between the inside tank wall and the juice.

As mentioned before the resistance to heat transfer through the tank wall, is so small comparing to the other resistances involved that it can be neglected. Therefore the total heat transfer to the cooling water is given by

$$Q = \frac{T_\infty - T_{wb}}{R_1} + \frac{T_j - T_{wb}}{R_2 + R_3} \tag{4-4}$$

4.2.2 Calculations

Cooling capacity

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 (3-11) repeated here. [12, pp. Chap 3.5]

$$Q = m_w C_{pw} (T_{cwo} - T_{cwi})$$

Natural convection heat transfer coefficient on the air side of the Jacket

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:

$$Q_{air} = \frac{T_\infty - T_{wb}}{\dfrac{1}{h_\infty A_{jt}}} \tag{4-5}$$

In order to evaluate the free convection coefficient (h_a) we make use of the equation developed by Bayley [11, pp. 333]

$$Nu_\infty = 0.59(GrPr)^{\frac{1}{4}} \tag{4-6}$$

where Gr is the Grashof number and Pr is the Prandtl number.

The film temperature between the ambient air and the Jacket wall is calculated using equation (3-12) repeated here.

$$T_f = \frac{T_\infty + T_{wb}}{2}$$

The physical properties involved in the Grashof and Prandtl numbers are evaluated at the film temperature using the equations in appendix A.

The natural convection heat transfer coefficient between the ambient air and the bulk of the water is given by

$$h_\infty = \frac{Nu_\infty k_\infty}{W_j} \tag{4-7}$$

where k_∞ is the conductivity of the air and W_j is the width of the Jacket

Forced convection heat transfer coefficient inside the channel

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

$$Q_{juice} = \frac{T_j - T_{wb}}{\frac{1}{A_{jt}} \left(\frac{1}{h_w} + \frac{1}{h_j}\right)} \tag{4-8}$$

The Nusselt number on the water side of the tank wall, is calculated using the relationship for forced convection developed by Dittus and Boelter [12, pp. 3.2.10]

$$Nu_w = 0.0243\ Re^{0.8}\ Pr^{0.4} \tag{4-9}$$

The water bulk temperature is calculated using equation (3-6) repeated here

$$T_{wb} = \frac{T_{cwi} + T_{cwo}}{2}$$

The physical properties involved in the Nusselt, Reynolds and Prandtl numbers are evaluated at the water bulk temperature using the equations in appendix A.

The forced convection coefficient inside the channel is given by

$$h_w = \frac{Nu_w k_w}{de} \tag{4-10}$$

were k_w is the thermal conductivity of the water and de is the equivalent hydraulic diameter of the channel as calculated in Appendix A.

Natural heat transfer convection coefficient on the juice side of the Jacket

The Nusselt number on the juice side of the tank wall, is calculated using equation (4-6) repeated here,

$$Nu_j = 0.59\ (Gr\,Pr)^{1/4}$$

The film temperature between the tank wall and the juice is calculated using equation (3-12) repeated here.

$$T_f = \frac{T_{wall} + T_j}{2}$$

where T_{wall} is calculated as follows:

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

$$Q = \frac{T_\infty - T_{wb}}{\dfrac{1}{h_\infty A_{j\infty}}} + \frac{T_{wall} - T_{wb}}{\dfrac{1}{h_w A_{jt}}} \tag{4-11}$$

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

$$T_{wall} = \frac{Q - h_\infty A_{j\infty}(T_\infty - T_{wb})}{h_w A_{jt}} + T_{wb} \tag{4-12}$$

The physical properties involved in the Grashof and Prandtl numbers are evaluated at the film temperature using the equations in appendix A.

The natural convection heat transfer coefficient between the fermenting juice and the jacket wall is calculated from

$$h_j = \frac{Nu_j k_j}{W_j} \tag{4-13}$$

Finally the total heat transfer to the cooling water is calculated using equations (3-11) and (4.11). If the value of Q determined from equation (4.11) does not correspond to the value of Q determined from equation (3-11), then the value of the temperature of the cooling water outlet temperature (T_{cwo}) is increases or decreased to better the situation and the calculations are repeated until agreement is obtained. Once agreement is obtained, the cooling capacity of the Jacket heat exchanger is predicted.

The overall heat transfer coefficient (*U*)

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. Therefore one might say there are two overall heat transfer coefficients involved.

$$Q = U_\infty A_{j\infty} (T_\infty - T_{wb}) + U_j A_{jt} (T_j - T_{wb}) \tag{4-14}$$

where U_∞ is the overall heat transfer coefficient on the air side and U_j on the juice side of the Jacket.

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.

An approximate effectiveness (ϵ) of the Jacket heat exchanger

The effectiveness of any heat exchanger, traditionally, is given by [11, pp.545]

$$\epsilon = \frac{\textit{actual heat transfer}}{\textit{maximum possible heat transfer}} \tag{4-15}$$

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 from

$$\epsilon = \frac{\dfrac{A_{jt}(T_j - T_{wb})}{\dfrac{1}{h_w} + \dfrac{1}{h_j}}}{m_w C_{pw}(T_j - T_{cwi})} \tag{4-16}$$

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

$$T_{wb} = \frac{T_j + T_{cwi}}{2} \tag{4-17}$$

4.2.3 Pressure loss

Pressure loss through Jacket heat exchangers is simulated using the Darcy-Weisbach equation (3-28). [14, pp. 196-197]

$$\Delta p = \rho_w \left[\left(f_{d1} \frac{L_j}{de} + \sum_{i=1}^{n} K_i \right) \frac{v_{w1}^{\,2}}{2} + \left(f_{d2} \frac{L_p}{d_p} + \sum_{a=1}^{m} K_a \right) \frac{v_{w2}^{\,2}}{2} \right]$$

where v_{w1}, f_{d1} and v_{w2}, f_{d2} denote the water velocity and friction factor in the cooling channel (station 1) and the circular inlet, outlet and connection pipe (station 2) respectively. *(Plastic pipes of the pressure transducer are taped into sockets which are connected to the inlet and outlet of the heat exchanger via Giga couplings. This implies that during differential pressure tests the pressure loss through the jacket as well as* *through two giga coupling connections is measured which must be born in mind during the simulation calculations).* The locations of high flow disturbance, which effect significant pressure loss, are depicted in figure 4.3 and their contribution to the total pressure loss are related in the above equation by the appropriate loss coefficients K_1 to K_8.

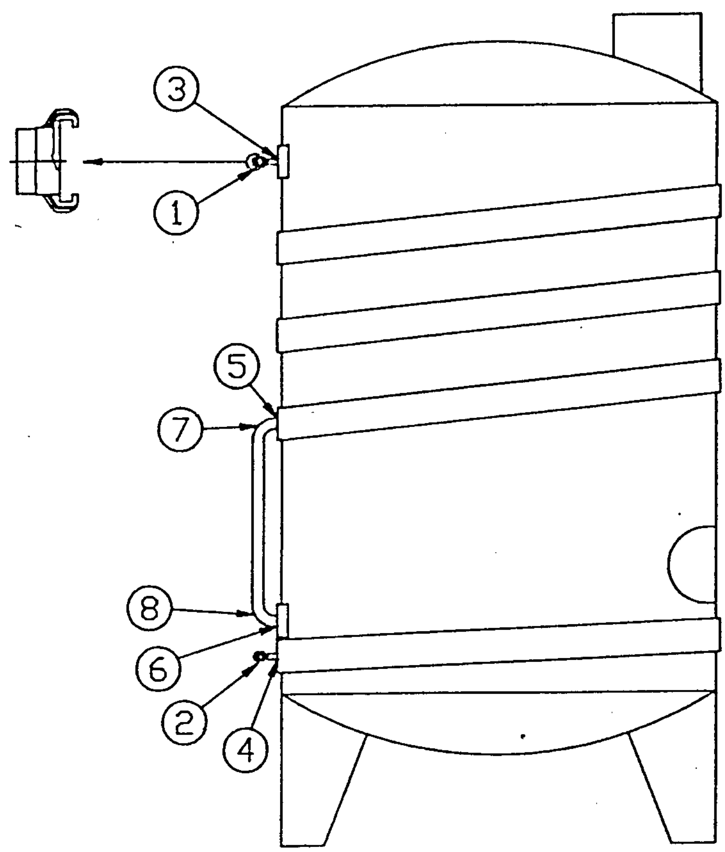


Figure 4.3 Pressure loss locations on a Jacket heat exchanger with Giga couplings.

Except for K_7 and K_8 [14, pp. 227] no applicable data were found in the literature for the other pressure loss coefficients. The experimental data were used and the sum of the pressure loss coefficients related to v_{w1} and v_{w2} were calculated to be 1 and 12.7 respectively.

The general pressure drop equation for the Jacket heat exchanger with the Giga couplings is:

$$\Delta p = \rho_w \left[\left[f_{d1} \frac{L_c}{de} + 1 \right] \frac{v_{w1}^{\ 2}}{2} + \left[f_{d2} \frac{L_p}{d_p} + 12.7 \right] \frac{v_{w2}^{\ 2}}{2} \right] \tag{4-18}$$

The results, using the above equation, are displayed in figure 4.4 section 4.3.3

4.3 EXPERIMENTAL PHASE

4.3.1 Apparatus

The apparatus (temperature sensors, differential pressure transducer, flow meter and data logging system) that were used during the experimental tests on the Weideman heat exchangers, were also used during the experiments done on the Jacket heat exchanger. Details of the apparatus are described in section 3.4.1 and the "Check and calibration" procedures are in appendix C.

4.3.2 Experimental procedures

Experiments were performed on a 26.4 metre long Jacket heat exchanger, during the wine making season with wine fermenting in the tank, therefore the only parameter that the author was permitted to manipulate during the tests was the flow rate of the cooling water. It was done by throttling a valve in the water pipeline, thus changing the water supply line pressure on the tank side of the water line. Two series of tests were performed. During the first, the temperature of the juice was ± 19.8 °C while the cooling water entered at ± 7.6 °C. During the second test series the juice temperature was ± 20.3 °C and the cooling water entered at ± 8.3 °C. The ambient temperature was approximately 18 °C during the tests. The mass flow rate of the cooling water through the heat exchanger was manually set by the valve situated in the cooling water pipeline. Mass flow rates were set from 0.3 kg/s to 0.6 kg/s at 0.1 kg/s steps.

All instrumentation were calibrated before installation as described in section 3.4.2. Water temperatures were monitored at the inlet and outlet of the Jacket heat exchanger. The differential pressures as well as the line pressure were measured across the inlet and outlet respectively, while the cooling water mass flow rate was measured in the cooling water supply.

Data were sampled for the different cooling water mass flow rates, mentioned above at time intervals of 40 seconds apart. All experiments were done during peak fermentation conditions when maximum cooling was needed.

4.3.3 Experimental data

Experimental data are listed in table B.17 in appendix B and equation (3-11) were used to calculate the cooling capacity of the Jacket heat exchanger. The results of the pressure loss at different flow rates are shown in table B.17 in appendix B. The experimental data together with predicted data calculated using equation (4-17) are displayed graphically in figure 4.4

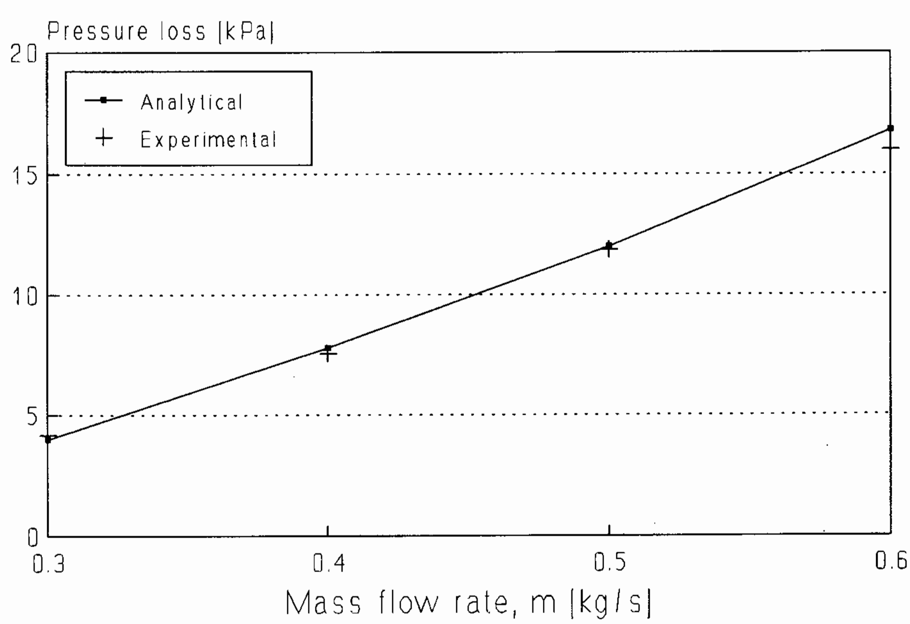


Figure 4.4 Pressure loss in the Jacket heat exchangers.

The pressure loss data are representative for almost every length of Jacket heat exchanger found in the wine industry which has similar inlet, outlet connecting pipe connections. The dimensions of the width (W_j) and height (H_j) of the Jacket heat exchanger are and should remain standard for the wine industry. It is assumed that the pressure loss due to the length of the Jacket is insignificant.

Flow rate versus line pressure.

Mass flow rates through the Jacket heat exchanger are plotted against the pressure in the water supply line in figure 4.5

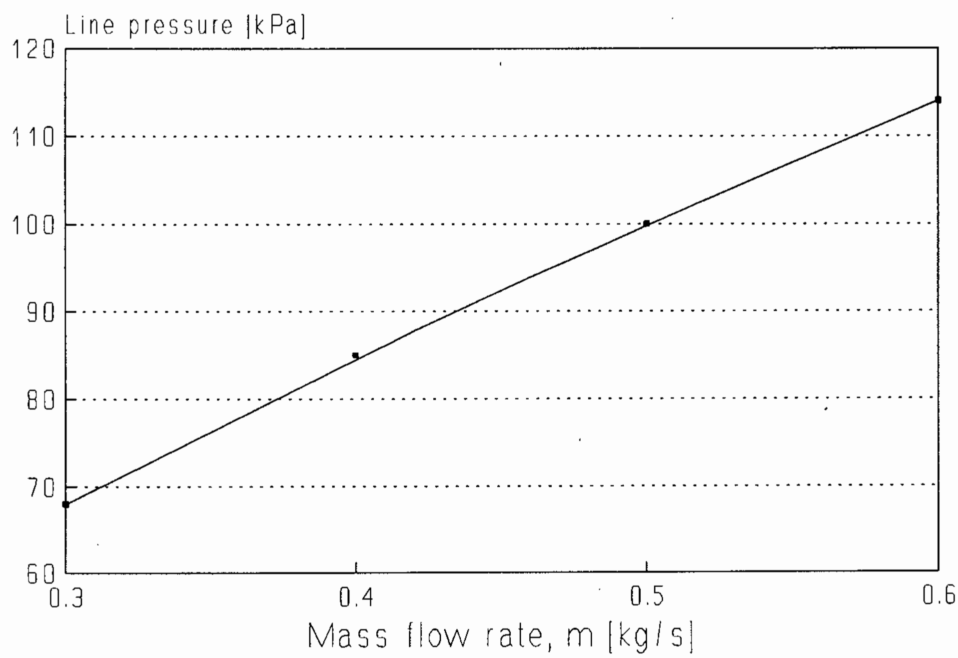


Figure 4.5 Line pressure versus flow rate for a 26.4 m long standard Jacket heat exchanger on a wine tank.

Typical line pressures in the wine industry supplying cooling water to Jacket heat exchangers range between 0.7 and 1.2 bar.

4.4 COMPUTER PROGRAM

The predicted results compare adequately with the experimental ones. Using the theoretical 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.

Due to the variety of sizes of these type of wine tanks used in the industry the length of the Jacket heat exchangers varies from tank to tank. In most cases the Jacket heat exchanger consists of four spiral conduits welded around the wine tank, thus the diameter of the tank determines the length of the Jacket heat exchanger. The majority have a length between 24 and 27 metres. Therefore it is sensible to base all calculations and predictions on a meter length, as a unit length, on the Jacket heat exchanger.

In a given situation where the requirements of a wine producer or a heat exchanger manufacturer require guidelines, the software developed requests the following input data:

* Tank size (m^3)
* Tank diameter (m)
* Maximum degrees balling fermented per day
* Ambient temperature (°C)
* Pressure (kPa) in the water supply line
* Water inlet temperature (°C)
* Juice temperature (°C)

The following results are produced:

* Jacket length (m) to complete the spirals
* Cooling needed (kW) to maintain a constant juice temperature
* Total heat transferred (kW) to the cooling water

* % heat gain or loss from ambient
* Effectiveness
* Pressure loss (kPa)

The program, listed in appendix D, was used to produce results which demonstrate the effect of different cooling water inlet temperature, juice and ambient temperatures on the cooling capacity and effectiveness of the Jacket heat exchanger considered in this chapter. For comparison purposes the experimental results obtained from data collected on the 26.4 m Jacket heat exchanger are shown on the same graphs.

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.

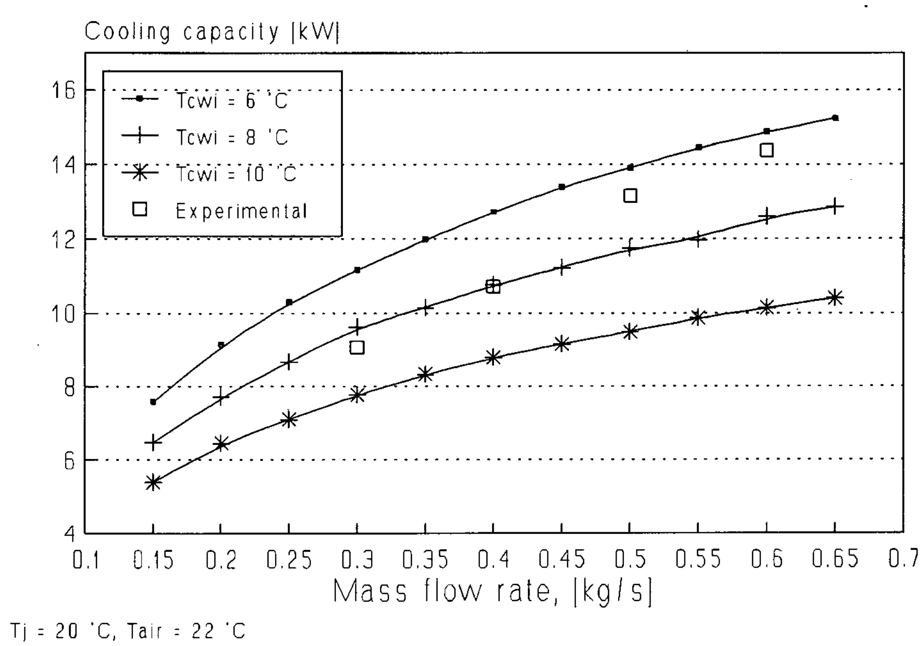


Figure 4.6 Cooling capacity of the Jacket heat exchanger with varying cooling water inlet temperature (T_{cwi}).

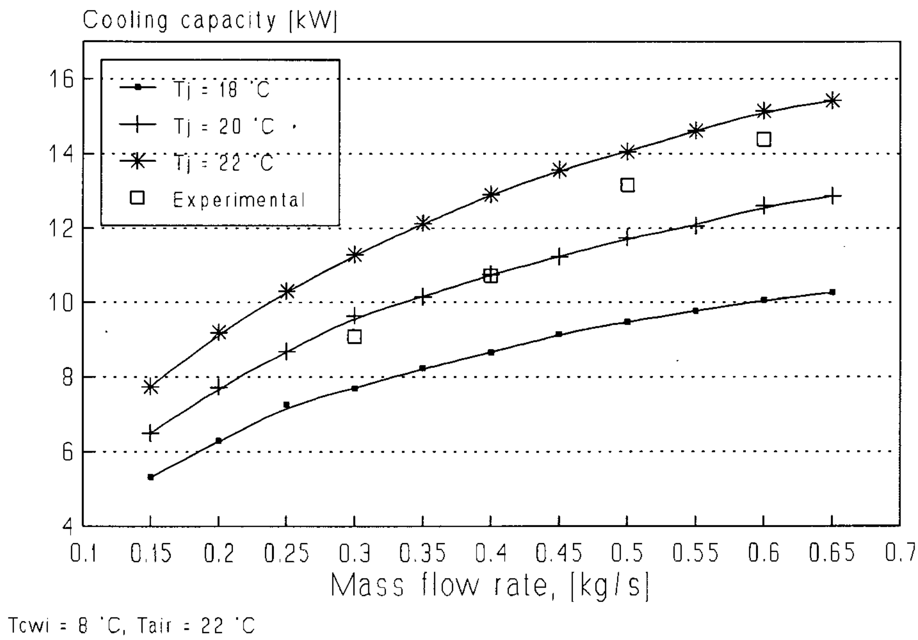


Figure 4.7 Cooling capacity of the Jacket heat exchanger with varying juice temperature (T_j).

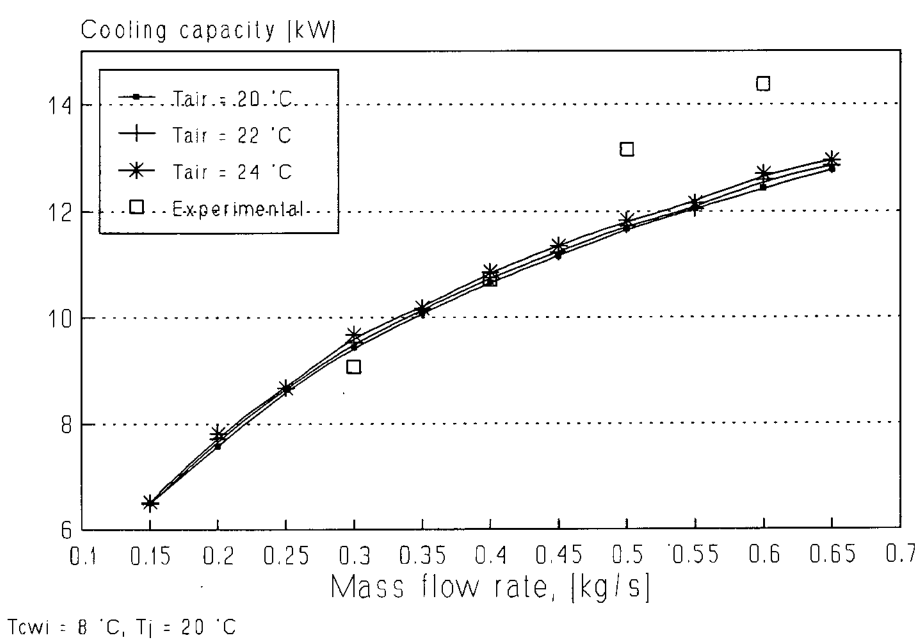


Figure 4.8 Cooling capacity of the Jacket heat exchanger with varying ambient temperature (T_∞).

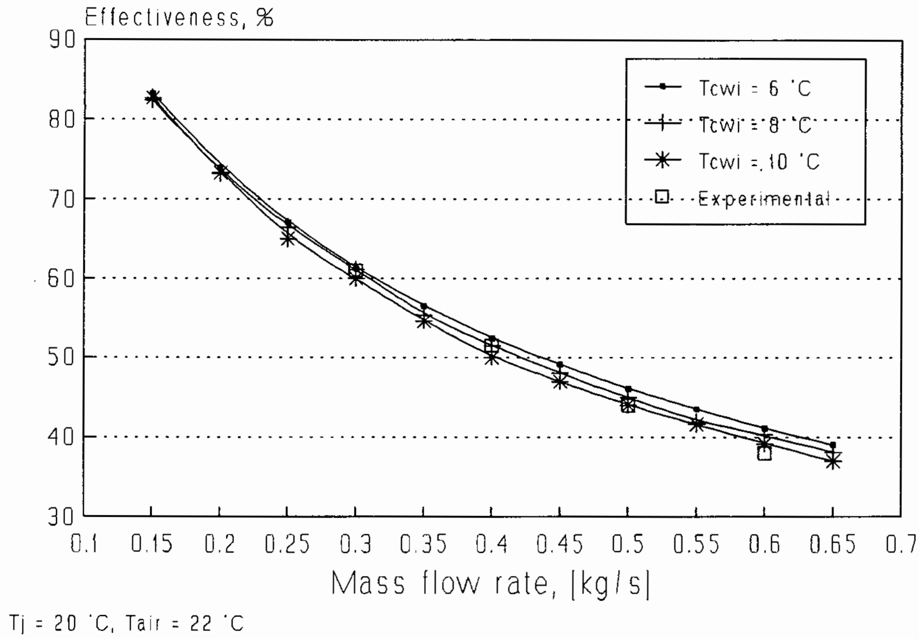


Figure 4.9 Effectiveness of the Jacket heat exchanger with varying cooling water inlet temperature (T_{cwi}).

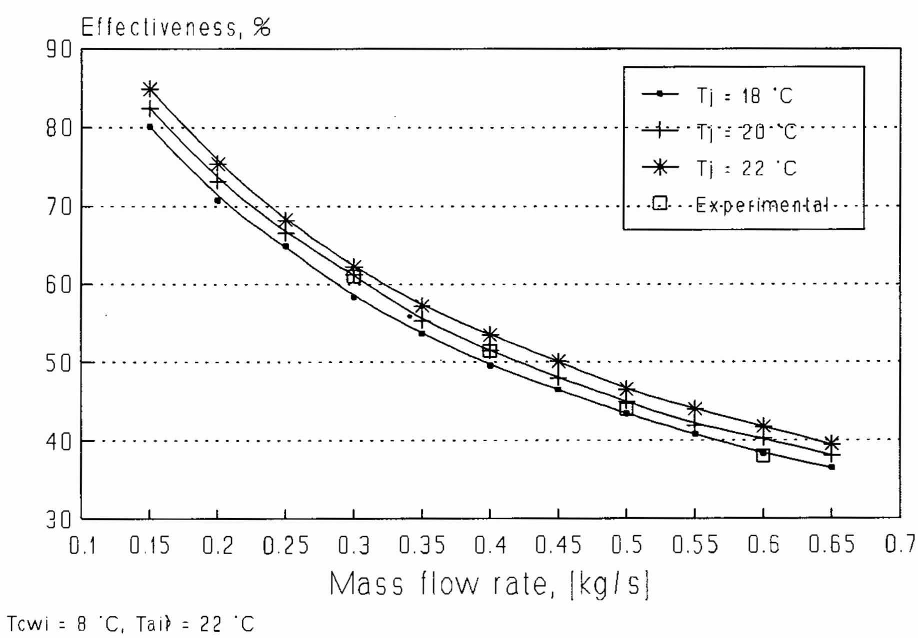


Figure 4.10 Effectiveness of the Jacket heat exchanger with varying juice temperature (T_j).

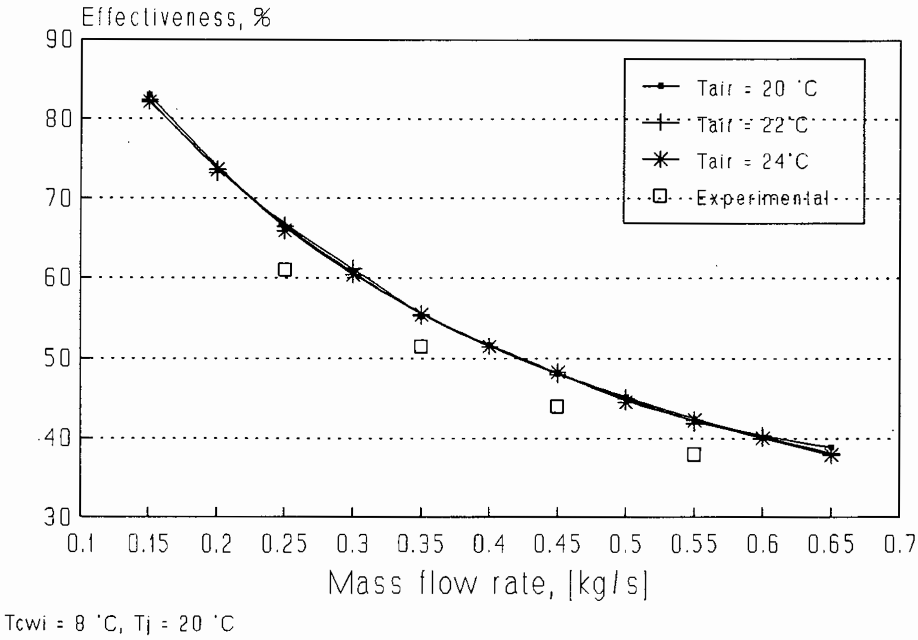


Figure 4.11 Effectiveness of the Jacket heat exchanger with varying ambient temperature (T_∞).

It is desirable 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 variation in the ambient temperature, the cooling capacity of the Jacket heat exchanger was less than one 1% change. 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

$$T_{eff} = 1 + 0.093(8 - T_{cwi}) + 0.0967(T_j - 20) \tag{4-19}$$

The results are displayed in figures (4.12) and (4.13).

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.

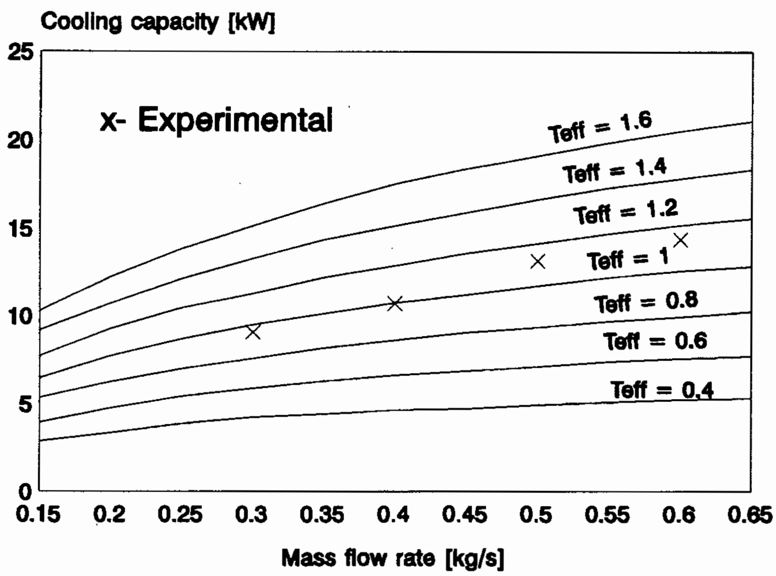


Figure 4.12 Cooling capacity of the Jacket heat exchanger with varying effective temperature (T_{eff}).

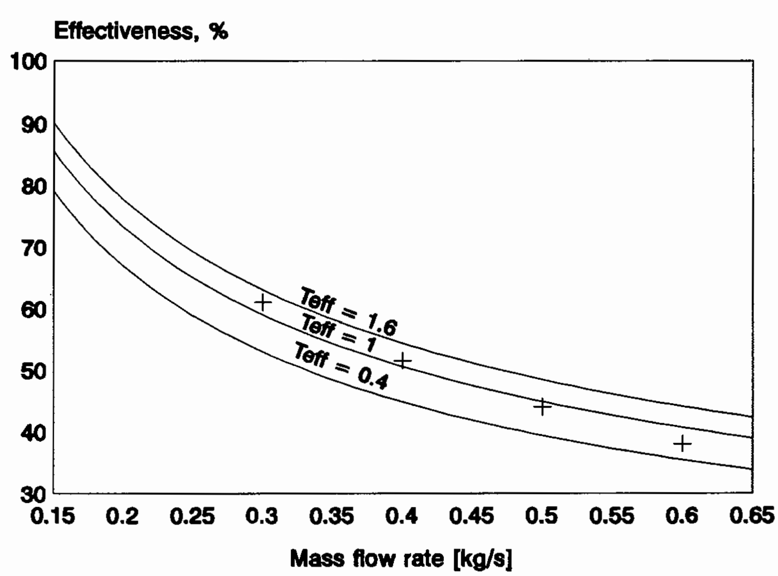


Figure 4.13 The effectiveness of the Jacket heat exchanger with varying effective temperature (T_{eff}).

4.5 DISCUSSION

This chapter attempted to predict the heat transfer and frictional losses of the Jacket heat exchanger that was tested, which was typical of the ones often employed in the wine industry.

The heat transfer performance was predicted using available power law relationships found in the literature and reasonable agreement was noted with the experimental results.

The graphs in figures 4.8 - 4.13 display how dominant the influence of each parameter is on the cooling capacity and effectiveness of the Jacket heat exchanger. It is apparent that a small change in the juice temperature has a marked effect on the cooling capacity and effectiveness of the Jacket heat exchanger. A change in the cooling water inlet temperature also has quite an effect on the cooling capacity and effectiveness of the Jacket heat exchanger, but to a lesser extent than a change in the juice temperature. A change in ambient temperature on the other hand, has little effect on cooling capacity and effectiveness of the Jacket heat exchanger.

Experimental data with regard to pressure drop and mass flow rates for given line pressures delivering cooling water to a Jacket heat exchanger completed the picture.

The computer program was written that for a given tank diameter it will calculate the length of the Jacket heat exchanger to complete four rings around the tank. Together with the tank diameter, the water supply line pressure, water inlet temperature, juice temperature and ambient temperature are required input data. These data will be used to calculate the length of the Jacket heat exchanger, the heat load capacity as a result of fermentation and ambient contributions, the cooling capacity of the Jacket heat exchanger, the percentage heat gain or loss from the ambient temperature and the effectiveness of the Jacket heat exchanger.

An example with detailed calculations is found in appendix A.

CHAPTER 5
WATER-FILM COOLING

5.1 THE WATER-FILM

Another, fairly widely used method of fermentation cooling in the wine industry is the one that uses water cascading on the outer surface of the tank containing the juice. Cooling water is pumped over a stainless steel wine tank through holes in a ring pipe above the tank and covers the outside tank area completely as a thin film. The water is collected in gutters at the bottom of the tank and is pumped back to the refrigeration equipment.

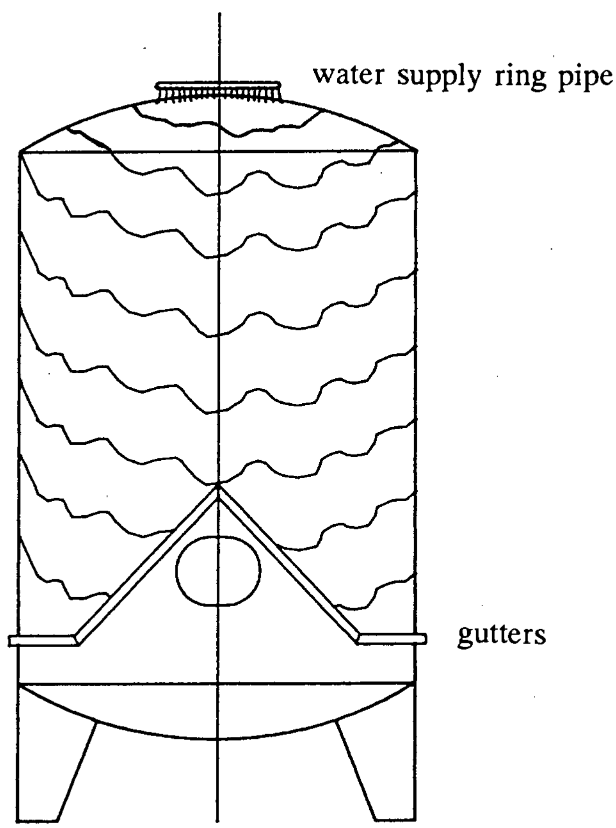


Figure 5.1 A wine tank with water-film cooling.

5.2 ANALYTICAL MODEL

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 is via radiation, convection and mass transfer [9],[10].

Heat transfer from the surroundings by radiation and convection takes place between the ambient temperature (T_∞) and the water-bulk temperature (T_{wb}) and by mass transfer due to the difference between the ambient vapour pressure ($P_{v\infty}$) and that of the saturated vapour of the water-film (P_{vw}) which is at water bulk temperature (T_{wb}). (It is assumed that the whole water-film is at the water bulk temperature (T_{wb}). It is assumed further that both sides of the tank wall are for practical purposes at the same temperature, due to the high conductance of the thin stainless steel wall). Heat is transferred from the tank wall to the water-film by forced convection between the tank wall temperature (T_{wall}) and the water bulk temperature (T_{wb}). This heat originates from the fermenting juice and is transferred by natural convection between the juice temperature (T_j) and the tank wall temperature (T_{wall}).

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 (9/10) of the tank 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 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}).

5.2.1 ANALYSIS

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

The flow of heat in any system is analogous to the flow of electrical current through a resistance and therefore the heat flow equation in general, can be expressed as [11, pp. 28]

$$Q_{total} = \frac{\Delta T_{overall}}{R_{total}} \tag{5-1}$$

where Q_{total} is the heat transfer that takes place across the overall temperature difference ($\Delta T_{overall}$) through the total resistance (R_{total}) that it encounters.

Equation (5-1) can now be expanded to describe the total heat transfer to the cascading water-film, once this film has reached the juice level

$$Q = \frac{T_{\infty} - T_{wb}}{\left[\dfrac{1}{R_1} + \dfrac{1}{R_2} + \dfrac{T_{wetb} - T_{wb}}{R_3\,(T_{\infty} - T_{wb})}\right]^{-1}} + \frac{T_j - T_{wb}}{R_4 + R_5} \tag{5-2}$$

where T_{∞} is the ambient temperature, T_{wetb} is the ambient wetbulb temperature, T_{wb} is the water bulk temperature, and T_j is the juice temperature.

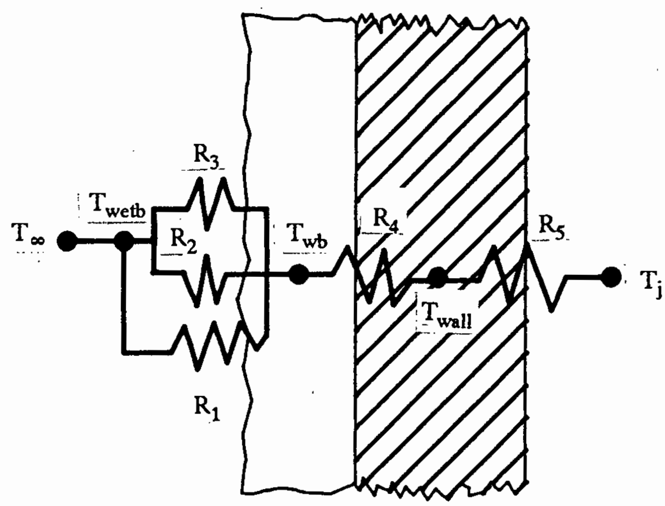


Figure 5.2 Electrical analogy of the heat transfer described in equation (5-2).

As mentioned above, the heat transfer between the ambient Temperature (T_∞) and the water bulk temperature (T_{wbi}) occurs via radiation, convection and mass transfer simultaneously or in parallel, and using the electrical analogy the following resistances are defined.

$$R_1 = \frac{1}{h_r A_{wtj}} \ , \ R_2 = \frac{1}{h_c A_{wtj}} \ , \ R_3 = \frac{1}{h_m A_{wtj}} \tag{5-3}$$

In the above analogous resistances, A_{wtj} is the tank area covered with water-film between the level of the juice and the gutters collecting the cooling water.

R_1 is the resistance due to radiation and h_r is the radiation heat transfer coefficient as defined by the following equation.[11, pp. 271].

$$h_r = \frac{F_{12}\epsilon\sigma(T_\infty^4 - T_{wb}^4)}{(T_\infty - T_{wb})} \tag{5-4}$$

where F_{12} is the shape factor which is unity in this case, ϵ is the emissivity of water, σ is the Stefan-Boltzmann constant, T_∞ is the ambient temperature and T_{wb} is the water bulk temperature which is the average between the water inlet temperature (T_{cwi}) and the water outlet temperature (T_{cwo}).

R_2 is the resistance to the heat flow encountered by natural convection where h_c is the natural heat transfer coefficient as calculated from [11, pp. 584-593]

$$h_c = 0.332 v_{max} \rho_\infty C_{p\infty} Re^{-\frac{1}{2}} Pr^{-\frac{2}{3}} \tag{5-5}$$

where v_{max} is the surface velocity of the water-film, *Re* is the Reynolds number and *Pr* is the Prandtl number of the flowing water.

Finally R_3 is the analogous resistance to the heat flow due to the mass transfer where h_m is the mass heat transfer coefficient defined as calculated from [9],[10],[12, Chap 8.2]

$$h_m = \frac{i_v\, h_D}{R_w\,(T_{wetb} - T_{wb})} \left[\frac{P_{v\infty}}{T_{wetb}} - \frac{P_{vw}}{T_{wb}}\right] \tag{5-6}$$

where i_v is the enthalpy of the vapour in the surrounding air, h_D is the mass transfer coefficient, P_{vw} is the vapour pressure of saturated vapour at the water bulk temperature (T_{wb}), $P_{v\infty}$ is the ambient vapour pressure, T_{wetb} is the ambient wetbulb temperature, R_w is the gas constant of water vapour.

Continuing with the second term of equation (5-2) the resistance to the heat transfer between the bulk water temperature (T_{wb}) and the tank wall temperature (T_{wall}) is

$$R_4 = \frac{1}{h_w A_{wt}} \tag{5-7}$$

where h_w is the forced convection coefficient which can be evaluated from [11, pp.260]

$$h_w = \frac{0.453\ Re^{\frac{1}{2}}\ Pr^{\frac{1}{3}} k_w}{H_{wt}} \tag{5-8}$$

where H_{wt} is the vertical distance of tank wall between the juice level and the gutters collecting the cooling water, k_w is the thermal conductivity of the water and Nu_w the Nusselt number on the water-film side of the tank wall.

The analogous resistance to heat transfer between the tank wall temperature (T_{wall}) and juice temperature (T_j) is

$$R_5 = \frac{1}{h_j A_{wt}} \tag{5-9}$$

where h_j is the natural or free convection coefficient between the juice and the tank wall calculated from [11, pp. 230]

$$h_j = \frac{0.1\ (GrPr)^{\frac{1}{3}} k_j}{H_{wt}} \tag{5-10}$$

where k_j is the thermal conductivity of the juice, Gr is the Grashof number, Pr the Prandtl number of fermenting juice and H_{wt} is the vertical distance of the tank wall between the juice level and the gutters collecting the cooling water.

The physical properties involved in the calculation of the heat transfer coefficients (h_w and h_j) on both sides of the tank wall are evaluated at the film temperature (T_f) using the equations in appendix A.

Initial heat transfer to the water

It was previously pointed out that there is an amount of heat transfer to the water-film from the moment it is formed on top of the tank till it reaches the level of the juice. This initial amount of heat transfer obviously changes the bulk water-film temperature to a higher value than the one it had as it left the ring pipe. It is only after the cascading water-film has reached the level of the juice that it will begin to cool the juice.

This initial heat transfer (Q_i) can be estimated using equation (5-2) suitably modified by dropping the last term referring to the juice side of the tank.

$$Q_i = \frac{A_{wta}(T_\infty - T_{wbi})}{\left[h_r + h_c + \frac{h_m(T_{wetb} - T_{wbi})}{T_\infty - T_{wbi}}\right]^{-1}} \tag{5-11}$$

where A_{wta} and T_{wbi} are the wetted surface area of the tank and the water bulk temperature respectively of the portion of the tank above the juice level.

5.2.2 Solution procedure

Initial heat transfer to the water

The initial heat transfer to the water-film is calculated 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}). [12, pp. Chap 3.5]

$$Q_i = m_w C_{pw}\left(T_{wjl} - T_{cwi}\right) \tag{5-12}$$

The heat transfer to the cooling water as calculated from the above equation, can also be evaluated by summing the initial heat transfer contributions by radiation (Q_{ri}), convection (Q_{hi}) and mass transfer (Q_{mi}), which resulted in equation (5-11) which is the special case of equation (5-2) as described on page 5.3.

The initial water bulk temperature (T_{wbi}) is the average between the cooling water inlet temperature (T_{cwi}) as it leaves the ring pipe, and the water temperature at juice level (T_{wjl}). [11, pp. 231]

$$T_{wbi} = \frac{T_{cwi} + T_{wjl}}{2} \tag{5-13}$$

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 (5-11) and (5-12), 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 (5-2) which is repeated here.

$$Q_{wtj} = \frac{A_{wta}(T_\infty - T_{wbj})}{\left[h_r + h_c + \dfrac{h_m(T_{wetb} - T_{wbj})}{T_\infty - T_{wbj}}\right]^{-1}} + \frac{A_{wtj}(T_j - T_{wbj})}{\dfrac{1}{h_w} + \dfrac{1}{h_j}}$$

The amount of heat gained by the water-film as described above can be determined using the energy equation (3-11) repeated here.

$$Q_{wtj} = m_w C_{pw}(T_{cwo} - T_{wjl})$$

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 (3-11). This amount of heat is compared with the amount calculated using equation (5-2) and when suitable agreement is reached the iterations are terminated. Note that in equation (5-2) the various heat transfer coefficients depend on known temperatures or temperatures that are calculated from known values, with the exception of h_j and h_w the heat transfer coefficients which depend on knowing the tank wall temperature. The value of the tank wall temperature is assumed for the purpose of the iterations, to be somewhere between the fermenting juice temperature and the water bulk temperature. The results obtained for the cooling capacity, using the analytical model and the previously described procedure, are shown in figures 5.3 to 5.5 together with the experimental results.

The overall heat transfer coefficient (*U*)

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 can not be calculated nor can the effectiveness of the heat exchanger be presented using the NTU method.

An approximate effectiveness (ϵ) of the Water-film

By definition the effectiveness of a heat exchanger is the actual heat transfer divided by the maximum possible heat transfer, that is [11, pp. 545]

$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\,transfer}$$

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 affectiveness for the cooling of the juice in this system as suggested below

$$\epsilon = \frac{\dfrac{A_{twj}(T_j - T_{wbj})}{\dfrac{1}{h_w} + \dfrac{1}{h_j}}}{m_w C_{pw}\ (T_j - T_{cwjl})} \tag{5-14}$$

The results of the effectiveness as calculated using equation (5-14) together with the experimental results are shown in figures 5.6 to 5.8.

5.3 EXPERIMENTAL PHASE

5.3.1 Apparatus

The apparatus used during the experimental work on the Water-film, was the same one used on the Weideman and Jacket heat exchangers. The temperatures were measured with temperature resistance devices (RTD's) . The mass flow rates of the cooling water were measured with a turbine flow meter. The analog signals from the different sensors were transmitted into a data logging system where they were amplified and converted to digital signals required by the software on the personal computer. Data were recorded every 40 seconds.

Further details of the apparatus is described in section 3.4.1 and the various sensor calibrations procedures can be found in appendix C.

5.3.2 Experimental procedures

Experiments were performed on a 80 000 litre Water-film wine tank while wine was fermenting in the tank. The experimental data were gathered during peak fermentation time when the juice generates heat at the fasted rate. As in the case of the Jacket heat exchanger the only parameter the wine-maker permitted the author to manipulate during the tests, was the flow rate of the cooling water. During the experiment the average juice temperature was ± 15°C and the cooling water temperature was ± 8.7°C. The ambient drybulb and wetbulb temperatures were on average 18.0°C and 16.5°C respectively.

The mass flow rate of the cooling water over the water-film tank was manually set by a valve situated in the cooling water supply line. Data were collected at mass flow rates of 0.3 kg/s to 0.8 kg/s in 0.1 kg/s steps.

The temperature of the cooling water were monitored as it left the ring above the tank and reached the gutters below the tank. The mass flow rate of the cooling water were measured in the water supply line.

All experimental data on the Water-film wine tank are listed in table B.18 and were used to calculate the cooling capacity.

5.4 COMPUTER PROGRAM

Based on the analytical model, software was written to perform the necessary calculations on Water-film wine tanks.

Given the parameters as input data:

* Tank size (m^3)
* Maximum degrees balling fermented per day
* Height of tank (m)
* Average height of gutters (m)
* Height of juice level (m)
* Diameter of tank (m)
* Diameter of ring above tank (m)
* Mass flow rate of the cooling water (kg/s)
* Water inlet temperature (°C)

* Ambient drybulb temperature (°C)
* Ambient wetbulb temperature (°C)
* Juice temperature (°C)

the following results can be obtained:

* Cooling needed (kW) to maintain a constant juice temperature.
* Total heat transferred (kW) to the cooling water.
* % heat gain from ambient.

The computer program, listed in appendix D, 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. The results using this program for the range of temperatures expected in practice are displayed in figures (5.3) to (5.8).

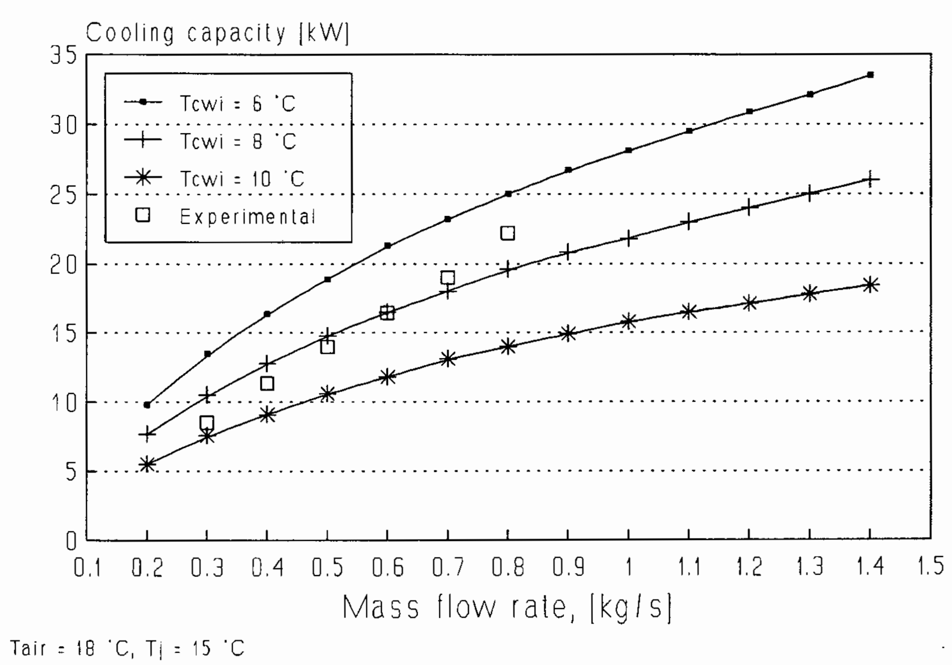


Figure 5.3 Cooling capacity of the Water-film with varying cooling water inlet temperature (T_{cwi}).

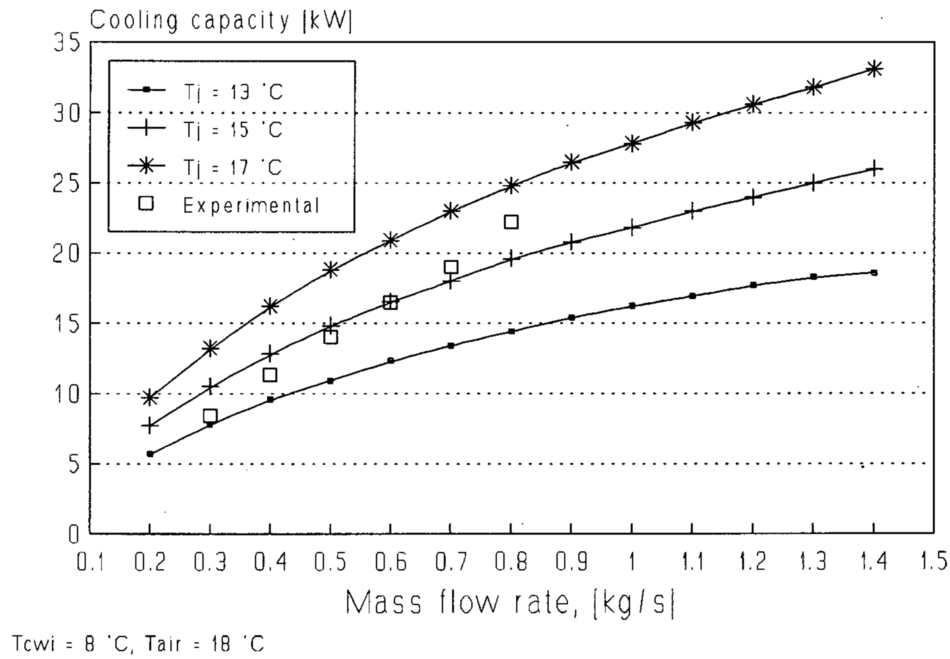


Figure 5.4 Cooling capacity of the Water-film with varying juice temperature (T_j).

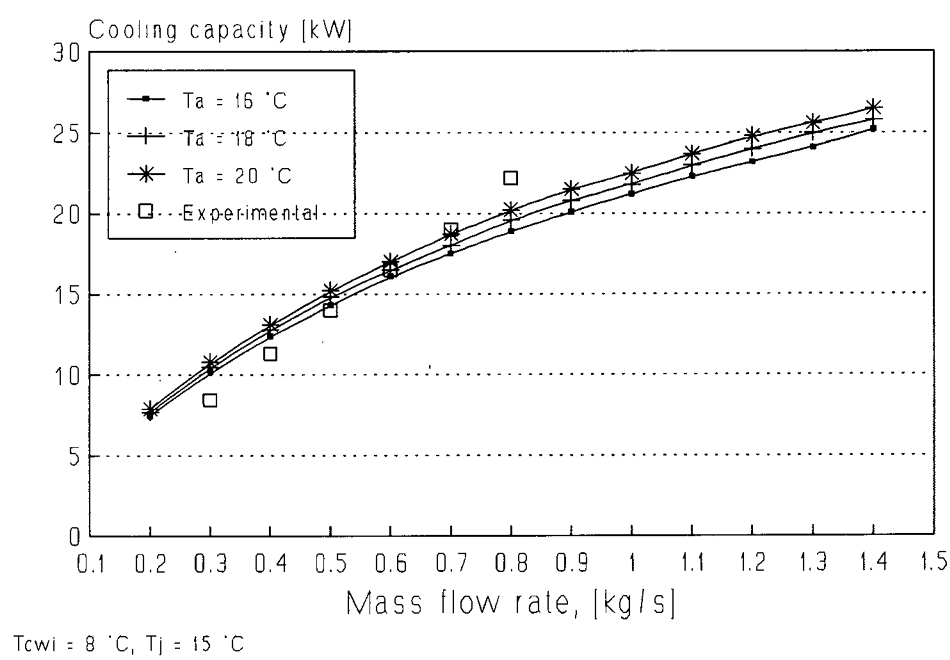


Figure 5.5 Cooling capacity of the Water-film with varying ambient temperature (T_{air}).

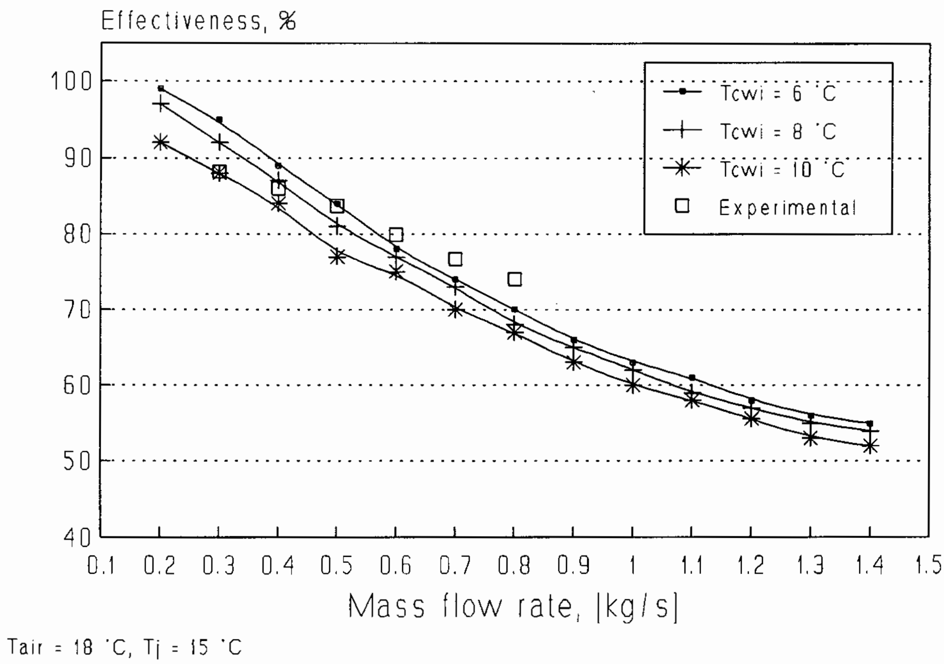


Figure 5.6 Effectiveness of the Water-film with varying cooling water inlet temperature (T_{cwi}).

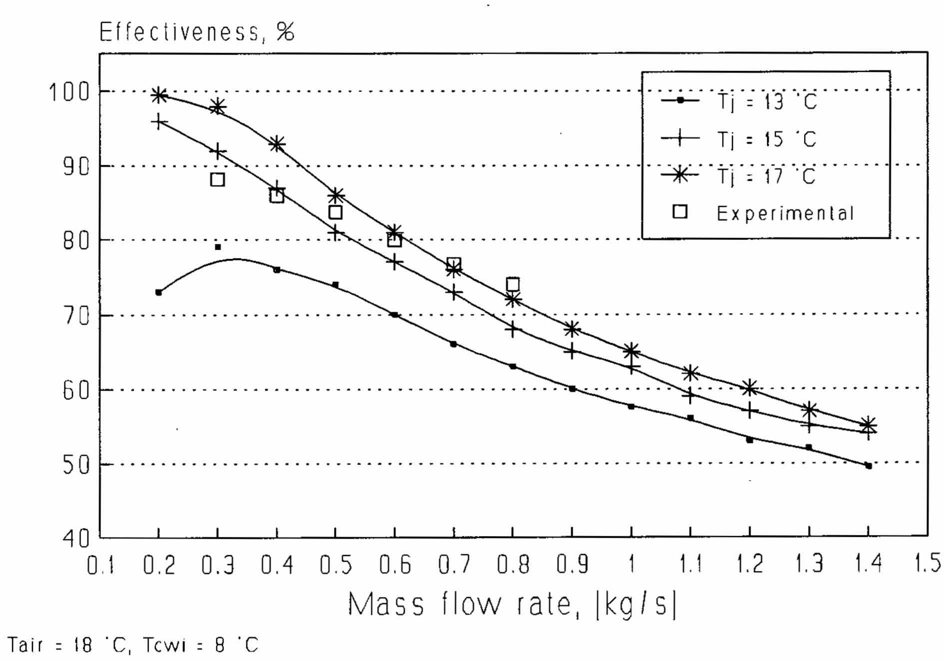


Figure 5.7 Effectiveness of the Water-film with varying juice temperature (T_j).

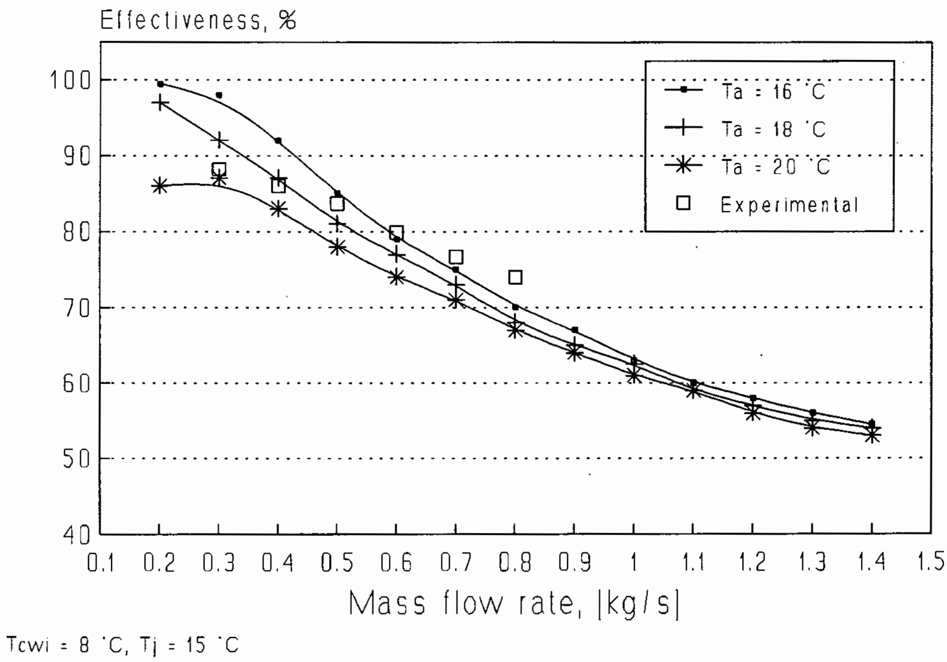


Figure 5.8 Effectiveness of the Water-film with varying ambient temperature (T_{air}).

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.

$$T_{eff} = 1 + 0.1415(8 - T_{cwi}) + 0.131(T_j - 15) + 0.015(T_\infty - 18) \qquad (5\text{-}15)$$

The results are displayed in figures (5.9) and (5.10).

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.

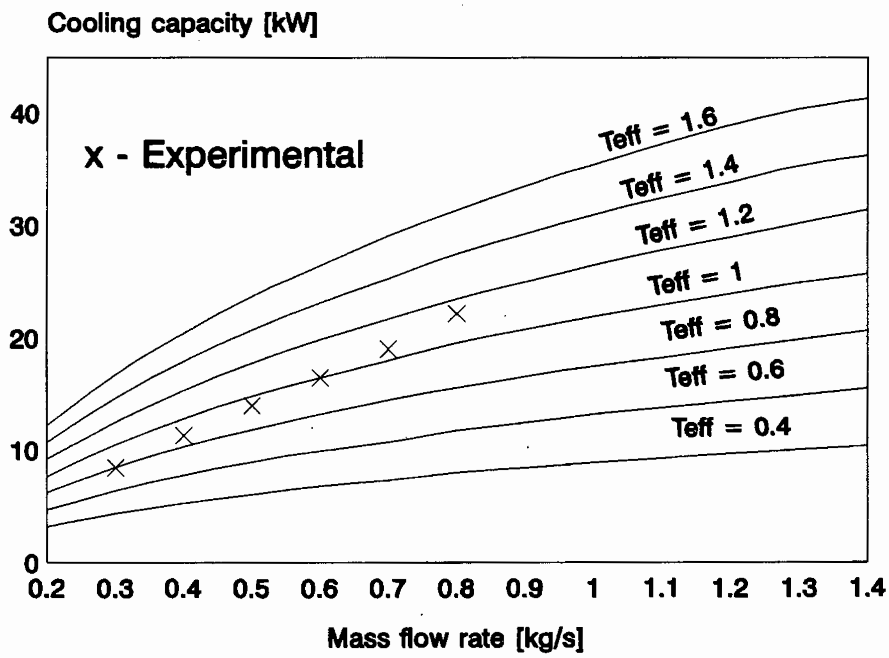


Figure 5.9 Cooling capacity of the Water-film for different values of the effective temperature (T_{eff}).

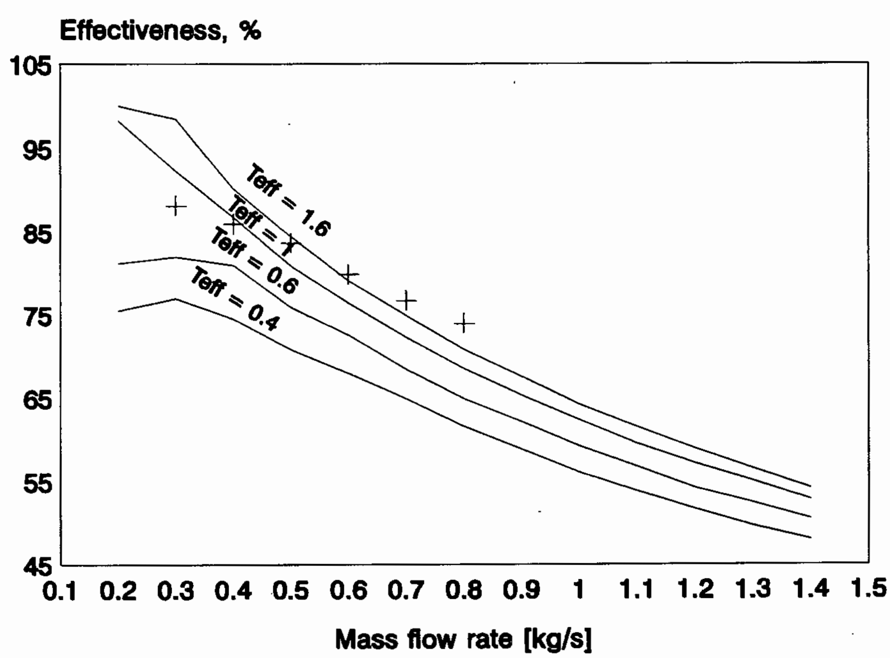


Figure 5.10 Effectiveness of the Water-film for different values of the effective temperature (T_{eff}).

5.5 DISCUSSION

The results for the cooling capacity and effectiveness in figures 5.3 to 5.8 indicate reasonable agreement between the analytical model and experimental data. In each figure the results depicted were obtained using the computer programme for the analytical model for a range of varying parameters such as cooling water inlet (figure 5.3), juice temperature (figure 5.4) etc. and for comparison purposes the experimental data have been included in these graphs. Comparing the analytical and experimental results in these figures, it appears that the analytical model can predict requirements for cooling capacity and thus provide the wine producer and water-film tank manufacturer with much needed data. The results also show to what extent, changes in each parameter affect the cooling capacity and effectiveness of the Water-film wine tank.

As expected the cooling capacity of the Water-film wine tank increases with an increase in juice temperature, a decrease in water inlet temperature, or a decrease in ambient temperature. In figures 5.3 and 5.4 we note that changes in the water inlet and juice temperatures has a greater effect on the cooling capacity than the ambient temperature. The experimental results were obtained with a cooling water inlet temperature of 8.7 °C and as predicted they lie between the results for cooling water inlet temperatures of 8 °C and 10 °C, and exhibit the same trend.

In figures 5.6 to 5.8 it is noted that an increase in effectiveness is obtained with a decrease in water inlet temperature, an increase in juice temperature and a decrease in ambient temperature.

The last two figures in this chapter depict the predicted results for the cooling capacity (figure 5.9) and the effectiveness (figure 5.10) of the Water-film heat exchanger using the effective temperature as described on page 5.16 and equation (5-15). For comparison purposes the experimental data have been included. Good agreement is noted.

An example with detail calculations is found in appendix A.

REFERENCES

1. SNYDERS, H.J., "Ingenieursaspekte van druif/mos verkoeling" The Directorate Regional Engineering, Elsenburg Wine Knowledge Notes, 1989

2. VERGUNST, M., "Regrigeration as applied in the Wine Industry" Die Wynboer, September, 1971

3. BOULTON, R., "The prediction of fermentation behavior by a Kinetic model" American Journal of Enology and Viticulture, vol. 31, no. 1, pp. 40-45, 1980

4. LE ROUX, J.M.W., PURCHAS, K., and NELL, B., "Regrigeration Requirements for Precooling and Fermentation Control in Wine Making" S. Afr. J. Enol. Vitic., vol. 7, no. 1, 1986

5. RULE, D., "Some tips on giving wine the big chill" Wines and Vines, November, pp. 67-69, 1988

6. DISTILLERIES GROUP, CSR., "Refrigeration and Cooling" The Australian Grapegrower & Winemaker, October, pp.28, 1988

7. DOWNS, M., "Refrigeration and Cooling" The Australian Grapegrower & Winemaker, Oktober, pp.30, 1988

8. MARAIS, N., "Gisting Per Rekenaar" Die Wynboer, Julie 1990

9. VOIGT, K.A., "'n Ondersoek na die eksterne hittelas op die kouewaterfilm soos gebruik by die verkoeling van groot tenke" Interne verslag, Dept. van Landbou-Ingenieurswese, Januarie 1972

10. KRöGER, D.G., LE ROUX, E.R., and PITOUT, P.A., "Temperature control in the wine industry" Heating, Air conditioning and Refrigeration, January, pp.29-43, 1974

11. HOLMAN J.P., Heat Transfer, Mc Graw Hill Book Co, Singapore, 1986

12. KRöGER D.G., Cooling Tower Performance Evaluation and Design, University of Stellenbosch, Stellenbosch, 1989

13. KERN D.Q., Process Heat Transfer, Mc Graw Hill Book Co, London, 1950

14. DAUGHERTY R.L. and FRANZINI J.B., Fluid Mechanics With Engineering Applications, McGraw Hill Kogakusha Ltd, Singapore, 1977

15 STOECKER W.F and JONES J.W.,Refrigertion & Air Conditioning, McGraw Hill International editions, Singapore, 1988

16 SPIEGEL M.R., Mathematical Handbook of Formulas and Tables, McGraw Hill International editions, Singapore, 1968

Appendix A

Calculations

CALCULATION OF FERMENTATION COOLING

Example from paper on fermentation cooling [2]

A cellar crushing a total of 10 000 tons of grapes over a period of two months (40 working days) experience a peak period during which 550 tons of grapes are crushed daily. Further data are as follow:

- maximum daily intake (M_g) of 550 tons
- 1 ton of grapes produces 720 litres (l_t) of juice
- sugar content of the juice is 22 degrees Balling (°B) (which means that on a mass basis 22 % of the juice is sugar.)
- density of the juice is 1.08 kg/l
- juice ferments at a rate of 2 °B (B_{max}) per 24 hours over a period of 11 days.
- heat of fermentation is 544,28 kJ/kg sugar

$$Mass\ of\ sugar\ (M_s) = M_g \times l_t \times \rho_j \times \frac{B_{max}}{100} \qquad \text{(A-1)}$$

$$= 550 \times 720 \times 1.08 \times \frac{2}{100}$$

$$= 8553.6 \quad kg\ sugar/day$$

$$Fermentation\ load\ (Q_f) = \frac{M_s \times \Delta H_s}{24 \times 3600} \tag{A-2}$$

$$= \frac{8553.6 \times 544.28}{24 \times 3600}$$

$$= 53.9 \quad kW$$

This is the amount of cooling capacity that must be available during the pressing season over and above the cooling that is needed to precool the grapes.

The cooling needed at a fermentation wine tank, is calculated in almost the exact manner. Say 82 000 litres of juice (l_j) are pumped into a fermentation tank. The mass of sugar is calculated

$$M_s = l_j \times \rho_j \times \frac{B_{max}}{100} \tag{A-3}$$

$$= 82000 \times 1.08 \times \frac{2}{100}$$

$$= 1771 \quad kg\ sugar/day$$

Equation (A-2) is now used to calculate the cooling needed at the tank

$$Q_f = \frac{M_s \times \Delta H_s}{24 \times 3600}$$

$$= \frac{1771 \times 544.28}{24 \times 3600}$$

$$= 11.2 \quad kW$$

CALCULATIONS ON A WEIDEMAN HEAT EXCHANGER

Calculation of the cooling capacity

The procedure as described in section 3.2.2 is followed for the prediction of the heat transfer from a 400x4585 mm Weideman heat exchanger, immersed in 14.78°C juice, while cooling water enters at 8°C at a rate of 0.5 l/s. Local heat generation is negligible.

Parameters which stay constant throughout the calculation are calculated first:

channel outside perimeter [16]

$$P_o = 2\pi \sqrt{\frac{\left[\frac{d_1}{2}\right]^2 + \left[\frac{d_2}{2}\right]^2}{2}} \tag{A-4}$$

$$= 2\pi \sqrt{\frac{\left[\frac{0.05167}{2}\right]^2 + \left[\frac{0.0223}{2}\right]^2}{2}}$$

$$= 0.1250 \ \text{m}$$

channel length

$$L_c = Z(L_h - 2d_1 - W_s) + \frac{\pi}{2}(Z - 1)(W_s + d_1 - 0.05) \tag{A-5}$$

$$= 6(4.585 - 2 \cdot 0.0517 - 0.01) + (6 - 1)\frac{\pi}{2}(0.01 + 0.0517 - 0.05)$$

$$= 26.928 \ \text{m}$$

See appendix A, page A-14 and A-15 for the derivation of equation (A-5).

channel outside area

$$A_{co} = L_c P_o \tag{A-6}$$

$$= 26.928 \cdot 0.1250$$

$$= 3.367 \quad m^2$$

channel inside perimeter

$$P_i = 2\pi \sqrt{\frac{\left[\frac{d_1}{2} - t\right]^2 + \left[\frac{d_2}{2} - t\right]^2}{2}} \tag{A-7}$$

$$= 2\pi \sqrt{\frac{\left[\frac{0.05167}{2} - 0.0009\right]^2 + \left[\frac{0.0223}{2} - 0.0009\right]^2}{2}}$$

$$= 0.120 \quad m$$

channel inside area

$$A_{ci} = L_c P_i \tag{A-8}$$

$$= 26.928 \cdot 0.12$$

$$= 3.225 \quad m^2$$

strip area

$$A_s = 2(W_h - Zd_1)L_c \tag{A-9}$$

$$= 2(0.39 - 6 \cdot 0.0517)4.585$$

$$= 0.732 \quad m^2$$

Calculation begins by using the experimental data to calculate the cooling capacity, using equation (3-11)

$$Q = m_w C_{pw}(T_{cwo} - T_{cwi})$$

$$= 0.5 \cdot 4197(12.06 - 8.0)$$

$$= 8521 \quad W$$

the water bulk temperature, using equation (3-6)

$$T_{wb} = 0.5(T_{cwo} + T_{cwi})$$

$$= 0.5(12.06 + 8.0)$$

$$= 10.03 \text{ °C}$$

The final value for T_{wall} is calculated iteratively (as described in section 3.3.2) to be 11.11 °C.

the film temperature, using equation (3-12)

$$T_f = \frac{T_{wall} + T_{wb}}{2}$$

$$= \frac{11.11 + 10.03}{2}$$

$$= 10.57 \text{ °C}$$

Forced convection inside the cooling channel

Prandlt number

$$Pr_w = \frac{C_{pw}\ \mu_w}{k_w} \tag{A-10}$$

$$= \frac{4196.2 \cdot 1.279 \cdot 10^{-3}}{0.58785}$$

$$= 9.13$$

Channel equivalent diameter

$$de = \frac{4A}{P} \tag{A-11}$$

$$= \frac{2\left[\frac{d_1}{2}\right]\left[\frac{d_2}{2}\right]}{\sqrt{0.5\left[\left[\frac{d_1}{2}\right]^2 + \left[\frac{d_2}{2}\right]^2\right]}}$$

$$= \frac{2\left[\frac{0.05167}{2}\right]\left[\frac{0.0223}{2}\right]}{\sqrt{0.5\left[\left[\frac{0.05167}{2}\right]^2 + \left[\frac{0.0223}{2}\right]^2\right]}}$$

$$= 0.028 \ \ \mathrm{m}$$

mean water velocity inside cooling channel (assume 1 m^3 = 1000 kg)

$$v_w = \frac{m_w \cdot 10^{-3}}{A_{ci}} \tag{A-12}$$

$$= \frac{m_w \cdot 10^{-3}}{\pi\left[\frac{d_1}{2} - t\right]\left[\frac{d_2}{2} - t\right]}$$

$$= \frac{0.5 \cdot 10^{-3}}{\pi\left[\frac{0.05167 \ - \ .0009}{2}\right]\left[\frac{0.0223 \ - \ 0.0009}{2}\right]}$$

$$= 0.586 \ \ \mathrm{m/s}$$

Reynolds number

$$Re_w = \frac{de\, v_w \rho_w}{\mu_w} \qquad \text{(A-13)}$$

$$= \frac{0.028 \cdot 0.586 \cdot 999.6}{1.279 \cdot 10^{-3}}$$

$$= 12824$$

friction factor

$$f = [1.82 \log(Re_w) - 1.64]^{-2} \qquad \text{(A-14)}$$

$$= [1.82 \log(12824) - 1.64]^{-2}$$

$$= 0.0294$$

Nusselt number, using equation (3-13)

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

$$= \frac{\left[\frac{0.0294}{8}\right] 12824 \cdot 9.13}{1.07 + 12.7 \sqrt{\frac{0.0294}{8}} \left(9.13^{0.67} - 1\right)}$$

$$= 116.5$$

forced convection heat transfer coefficient, using equation (3-14)

$$h_w = \frac{Nu\, k_w}{de}$$

$$= \frac{116.5 \cdot 0.58785}{0.028}$$

$$= 2448.2 \ \ \mathrm{W/m^2\,^\circ C}$$

heat transferred from the channel inside surface to the cooling water,using equation (3-15)

$$Q = h_i A_{ci}(T_{wall} - T_{wb})$$

$$= 2448.2 \cdot 3.225(11.11 - 10.03)$$

$$= 8528.1 \quad W$$

An initial value for h_o is chosen.

using equation (3-16), h_o is calculated iteratively

$$Q = h_o A_o (T_j - T_{wall}) + Q_s$$

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 with equation (3-10)

$$Q_s = 10Q_{s1} + 2Q_{s2}$$

which is the summation of equations (3-7) and (3-9)

If the value of the heat transfer Q, determined by equation (3-10) does not correspond to the one calculated from equation (3-15) then the value of h_o is increased or decreased and the calculations repeated until agreement is obtained. The final value for h_o was calculated iteratively to be 613.0 W/m²°C.

Heat transfer to the strips

heat transfer to the "fins" between the channels, using equation (3-7) with m as defined in equation (3-8)

$$m = \sqrt{\frac{h_o P_s}{kA_s}}$$

$$= \sqrt{\frac{613 \cdot 2(4.585 - 0.025)}{16.3 \cdot 0.0009 \cdot 2(4.585 - 0.025)}}$$

$$= 204.4$$

$$Q_{s1} = \sqrt{h_o P_{s1} k A_{s1}}\ (T_j - T_{wall}) \tanh mL_1$$

$$= \sqrt{613 \cdot 9.1 \cdot 16.3 \cdot 0.0082}\ (14.8 - 11.11) \tanh(204.4 \cdot 0.005)$$

$$= 77.2 \ \ W/\text{fin}$$

the heat transfer to the remaining two "fins" on either end along the heat exchanger is calculated, using equation (3-9)

$$Q_{s2} = \sqrt{h_o P_{s2} k A_{s2}}\ (T_j - T_{wall}) \frac{\sinh mL_2 + (h_o/mk)\cosh mL_2}{\cosh mL_2 + (h_o/mk)\sinh mL_2}$$

$$= \sqrt{613 \cdot 9.1 \cdot 16.3 \cdot 0.0082}\ (14.8 - 11.11) \frac{\sinh(1.53) + (613/3332)\cosh(1.53)}{\cosh(1.53) + (613/3332)\sinh(1.53)}$$

$$= 91.25 \ \ W$$

the total heat transfer to all the "fins", using equation (3-10)

$$Q_s = 10 Q_{s1} + 2 Q_{s2}$$

$$Q_s = 10 \cdot 77.2 + 2 \cdot 91.25$$

$$= 954.5 \ \ W$$

heat transfer to the outside surface of the heat exchanger, using equation (3-16)

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

$$= 613 \cdot 3.367(14.78 - 11.11) + 954.5$$

$$= 8528.1 \ \ \mathrm{W}$$

the average strip temperature, using equation (3-17)

$$T_s = T_j - \frac{Q_s}{A_s h_o}$$

$$= 14.78 - \frac{954.5}{0.728 \cdot 613}$$

$$= 12.64 \ {}^{\circ}\mathrm{C}$$

average wall temperature, using equation (3-18)

$$\overline{T_{wall}} = \frac{A_{co} T_{wall} + A_s T_s}{A_{co} + A_s}$$

$$= \frac{3.367 \cdot 11.11 + 0.728 \cdot 12.64}{3.367 + 0.728}$$

$$= 11.38 \ {}^{\circ}\mathrm{C}$$

average outside convection heat transfer coefficient, using equation (3-19)

$$\overline{h_o} = \frac{Q}{A_T(T_j - \overline{T_{wall}})}$$

$$= \frac{8521}{(3.367 + 0.728)(14.78 - 11.38)}$$

$$= 612.4 \ \ \mathrm{W/m^2 {}^{\circ}C}.$$

where A_T is the total outside area of the heat exchanger (*Due to the high conductance of the stainless steel, the size and the thickness of the strips, the average strip temperature does not differ much from the channel wall temperature. This has the effect that* $\overline{h_o}$ *does not differ much from* h_o *and a fair assumption would have been that* h_o *is the average natural convection coefficient over the total surface of the heat exchanger.*)

A new film temperature is calculated, using a similar equation as (3-12)

$$T_f = \frac{\overline{T_{wall}} + T_j}{2}$$

$$= \frac{11.38 + 14.78}{2}$$

$$= 13.08\ °\mathrm{C}$$

The average outside Nusselt number, using equation (3-20)

$$\overline{Nu_o} = \frac{\overline{h_o} W_h}{k_j}$$

$$= \frac{612.4 \cdot 0.39}{0.5878}$$

$$= 406.3$$

Prandtl number, using equation (3-21)

$$Pr_j = \frac{C_{pj} \mu_j}{k_j}$$

$$= \frac{3645 \cdot 1.2055 \cdot 10^{-3}}{0.5916}$$

$$= 7.43$$

Grashof number, using equation (3-22)

$$Gr = \frac{g\beta(T_j - T_{wo})W_h^3}{\nu^2}$$

$$= \frac{9.81 \cdot 1.538 \cdot 10^{-4}(14.78 - 11.11)0.39^3}{(1.1948 \cdot 10^{-6})^2}$$

$$= 2.30 \cdot 10^8$$

overall outside heat transfer coefficient, using equation (3-25)

$$U_o = \frac{1}{\frac{A_T}{A_i}\frac{1}{h_i} + \frac{1}{h_o}}$$

$$= \frac{1}{\frac{4.094}{3.225}\frac{1}{2448.2} + \frac{1}{612.4}}$$

$$= 465 \quad W/m^2{}^\circ C$$

number of transfer units (NTU), using equation (3-26)

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

$$= \frac{465 \cdot 4.094}{0.5 \cdot 4196.2}$$

$$= 0.91$$

where C_{min} is the juice in our case because it "remains" at constant temperature.

effectiveness (ϵ),using equation (3-27)

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

$$= 1 - e^{-0.91}$$

$$= 0.60$$

Pressure loss

The inlet and outlet of the Weideman heat exchanger were circular stainless steel pipes with a inside diameter of 22 mm and a length of 120 mm each.

water velocity inside the inlet and outlet pipes, using equation (A-12)

$$v_{w2} = \frac{m_w \cdot 10^{-3}}{A_{ci}}$$

$$= \frac{0.5 \cdot 10^{-3}}{\frac{\pi}{4}(0.022)^2}$$

$$= 1.315 \text{ m/s}$$

Reynolds number, using equation (A-13)

$$Re_p = \frac{v_{w2}\, d\rho_w}{u_w}$$

$$= \frac{1.315 \cdot 0.022 \cdot 999.6}{1.279 \cdot 10^{-3}}$$

$$= 22614$$

friction factor, using equation (A-14)

$$f_p = [1.82\log(Re_p) - 1.64]^{-2}$$

$$= [1.82\log(22614) - 1.64]^{-2}$$

$$= 0.0253$$

pressure loss, using equation (3-29)

$$\Delta p = \rho_w \left[\left(f_{d1}\frac{L_c}{de} + 11 \right) \frac{v_{w1}^{\ 2}}{2} + \left(f_{d2}\frac{L_p}{d_p} + 2.72 \right) \frac{v_{w2}^{\ 2}}{2} \right]$$

$$\Delta p = 999.6 \left[\left(0.0294\frac{26.928}{0.028} + 11 \right) \frac{0.586^2}{2} + \left(0.0253\frac{0.24}{0.022} + 2.72 \right) \frac{1.315^2}{2} \right]$$

$$= 9330 \ \text{Pa}$$

DERIVATION OF EQUATION (A-5)

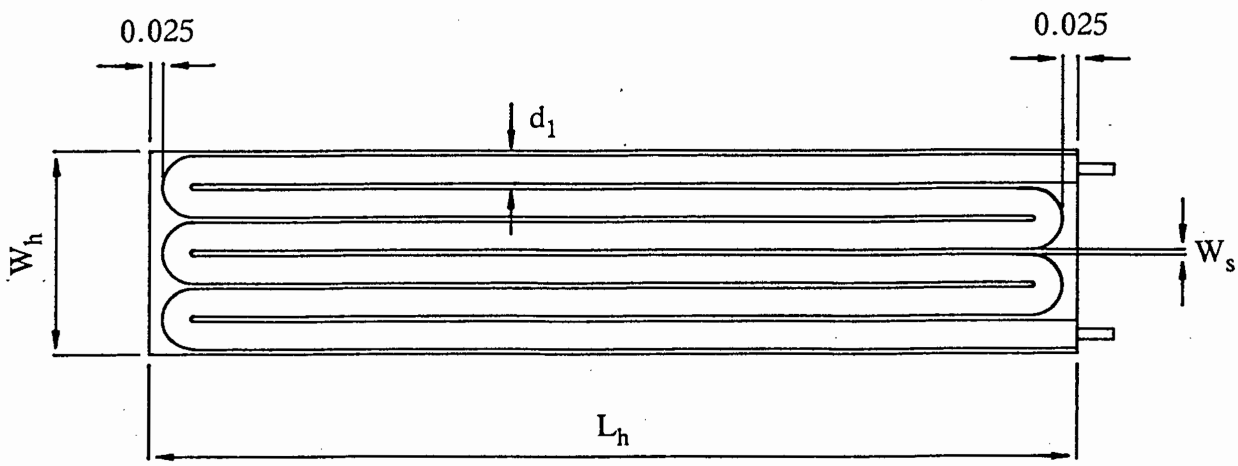


Figure A.1 A Weideman heat exchanger

Straights

$$L_c' = Z\left[L_h - 2\left[0.025 + d_1 + \frac{W_s}{2}\right]\right] + 2 \cdot 0.025 \tag{A-15}$$

$$= Z(L_h - 2d_1 - W_s) - 0.05(Z - 1)$$

U-turns

$$L_c'' = \frac{\pi}{2}\left[\frac{W_S}{2} + \left[d_1 + \frac{W_s}{2}\right]\right](Z - 1) \tag{A-16}$$

$$= \frac{\pi}{2}(W_s + d_1)(Z - 1)$$

Channel length

$$L_c = L_c' + L_c'' \tag{A-17}$$

$$= Z(L_h - 2d_1 - W_s) + \frac{\pi}{2}(Z - 1)(W_s + d_1 - 0.05)$$

THE THERMOPHYSICAL PROPERTIES OF SATURATED WATER LIQUID

The following equations is suitable to calculate thermophysical properties of water liquid that is at a temperature 273.15 K to 380 K.

Density [kg/m^3]

$$\rho_w = \frac{1}{a + bT + cT^2 + dT^6} \tag{A-18}$$

a = 1.49343×10^{-3}

b = -3.7164×10^{-6}

c = 7.09782×10^{-9}

d = -1.90321×10^{-20}

Specific heat [J/kgK]

$$c_{pw} = a + bT + cT^2 + dT^6 \tag{A-19}$$

a = 8.15599×10^3

b = -2.80627×10

c = 5.11283×10^{-2}

d = -2.17582×10^{-13}

Dynamic viscosity [kg/ms]

$$\mu_w = a10^{\frac{b}{(T-c)}} \tag{A-20}$$

a = 2.414×10^{-5}

b = 247.8

c = 140

Thermal conductivity [W/mK]

$$k_w = a + bT + cT^2 + dT^4 \tag{A-21}$$

a = -6.14255 x 10^{-1}

b = 6.9962 x 10^{-3}

c = -1.01075 x 10^{-5}

d = 4.74737 x 10^{-12}

Latent heat of vaporization [J/kg]

$$i_{fgw} = a + bT + cT^2 + dT^3 \tag{A-22}$$

a = 3.4831814 x 106

b = -5.8627703 x 103

c = 1.2139568 x 10

d = -1.40290431 x 10-2

Critical pressure [N/m^2]

$$P_{wc} = 22.09 \; x \; 10^6 \tag{A-23}$$

Volume coefficient of expansion [1/K]

$$\beta = a + bT + cT^2 + dT^3 \tag{A-24}$$

a = -3.3033487 x 10^{-5}

b = 1.7932227 x 10^{-5}

c = -2.6288421 x 10^{-7}

d = -1.7503647 x 10^{-9}

CALCULATION OF THE COOLING CAPACITY NEEDED DURING FERMENTATION

Example

volume of juice : 96 000 litres

Max fermentation rate : 2° Balling (in 24 hours)

heat released per kg of sugar fermented to alcohol = 544.284 kJ/kg

density of grape juice = 1.08 kg/l

$$m_j = V_j \rho_j \tag{A-25}$$

$$= 96000 \cdot 1.08$$

$$= 103680 \quad \text{kg}$$

where m_j is the mass of the juice, V_j is it's volume an ρ_j is it's density

$$q_{ferm} = 544.28 m_j fr \tag{A-26}$$

$$= 544.284 \cdot 103680 \cdot \frac{2}{100}$$

$$= 1128627.3 \quad \text{kJ} \quad (\textit{in 24 hours})$$

where q_{ferm} is the fermentation heat and fr is the fermentation rate per day

$$Q_{ferm} = \frac{q_{ferm}}{24 \cdot 3600} \tag{A-27}$$

$$= \frac{1128627.3}{24 \cdot 3600}$$

$$= 13.06 \quad \text{kW}$$

General equation for the calculation of fermentation heat

From the above equations the following equation was derived to calculate the heat rate generated during fermentation.

$$Q_{ferm} = V_j\, fr \cdot 6.804 \cdot 10^{-5} \tag{A-28}$$

CALCULATIONS ON A JACKET HEAT EXCHANGER

Cooling capacity

Following are the calculations for the heat transfer on a 26.4 m long jacket on a wine tank. The cooling water enters at a temperature 7.6 °C and a flow rate of 0.5 l/s, while the juice and ambient temperatures are 19.8 °C and 18.0 °C.

Parameters which stay constant throughout the calculation are calculated first:

tank area overlapping with the jacket

$$A_{jt} = L_j W_j \tag{A-29}$$

$$= 26.4 \cdot 0.23$$

$$= 6.072 \text{ m}^2$$

$$A_{j\infty} = L_j \; (W_j + 2H_j)$$

$$= 26.4 \; (0.23 + 2 \cdot 0.012)$$

$$= 6.706 \text{ m}^2$$

equivalent diameter of jacket

$$de = \frac{4A}{Pe} \tag{A-30}$$

$$= \frac{4(W_j - 2t)(H_j - t)}{2(W_j - 2t + H_j - t)}$$

$$= \frac{4(0.23 - 2\cdot 0.002)(0.012 - 0.002)}{2(0.23 - 2\cdot 0.002 + 0.012 - 0.002)}$$

$$= 1.915\cdot 10^{-2} \text{ m}$$

jacket cross section area

$$A_{j\ cross} = (W_j - 2t)(H_j - t) \qquad \text{(A-31)}$$

$$= (0.23 - 2\cdot 0.002)(0.012 - 0.002)$$

$$= 2.26\cdot 10^{-3} \text{ m}^2$$

Initial values for T_{wall} and T_{cwo} were chosen and final values were calculated as described in section 4.2.2 to be 12.21 °C and 13.6 °C.

Cooling capacity

The total heat transferred to the water, using equation (3-11)

$$Q = m_w C_{pw} (T_{cwo} - T_{cwi})$$

$$= 0.5\cdot 4196.7\ (13.6 - 7.6)$$

$$= 12588 \text{ W}$$

The water bulk temperature is calculated using equation (3-6).

$$T_{wb} = \frac{T_{cwo} + T_{cwi}}{2}$$

$$= 0.5\,(13.6 + 7.6)$$

$$= 10.6 \text{ °C}$$

Forced convection inside the jacket

water velocity, using equation (A-12)

$$v_w = \frac{m_w \cdot 10^{-3}}{A_{j\,cross}}$$

$$= \frac{0.5 \cdot 10^{-3}}{2.26 \cdot 10^{-3}}$$

$$= 0.2227 \quad \text{m/s}$$

Reynolds number, using equation (A-13)

$$Re_{de} = \frac{v_w de \rho_w}{\mu_w}$$

$$= \frac{0.2227 \cdot 1.915 \cdot 10^{-2} \cdot 999.6}{1.283 \cdot 10^{-3}}$$

$$= 3322$$

Prandtl number, using equation (A-10)

$$Pr = \frac{C_{pw}\mu_w}{k_w}$$

$$= \frac{4196.4 \cdot 1.283 \cdot 10^{-3}}{0.5873}$$

$$= 9.2$$

Nusselt number, using equation (4-9)

$$Nu_w = 0.0243 Re^{0.8} Pr^{0.4}$$

$$= 0.0243 \cdot 3322^{0.8} \cdot 9.2^{0.4}$$

$$= 38.7$$

forced convection coefficient, using equation (4-10)

$$h_w = \frac{Nu_w k_w}{de}$$

$$= \frac{38.7 \cdot 0.5873}{1.915 \cdot 10^{-2}}$$

$$= 1187.6 \text{ W/m}^2\text{°C}$$

Natural convection on the air side of the Jacket

film temperature, using equation (3-12)

$$T_f = \frac{T_\infty + T_{wb}}{2}$$

$$= \frac{18 + 10.6}{2}$$

$$= 14.3\text{°C}$$

Prandtl number, using equation (A-10)

$$Pr = \frac{C_{pw} \mu_w}{k_w}$$

$$= \frac{1005.6 \cdot 1.787 \cdot 10^{-5}}{0.0253}$$

$$= 0.711$$

Grashof number, using equation (3-22)

$$Gr = \frac{g\beta\,(T_{\infty} - T_{wb})\,W_j^3}{\nu^2}$$

$$= \frac{9.81\left[\frac{1}{291}\right](18 - 10.6)\,0.23^3}{(14.639\cdot 10^{-6})^2}$$

$$= 1.44\cdot 10^7$$

The Nusselt number using equation (4-6)

$$Nu_{\infty} = 0.59\,(Gr\;Pr)^{1/4}$$

$$= 0.59\,(1.44\cdot 10^7\cdot 0.711)^{1/4}$$

$$= 33.5$$

natural convection coefficient, using equation (4-7)

$$h_{\infty} = \frac{Nu\;k_{\infty}}{W_j}$$

$$= \frac{33.5\cdot 0.0252}{0.23}$$

$$= 3.7\ \mathrm{W/m^2\,^{\circ}C}$$

the wall temperature (T_{wall}), using equation (4-12)

$$T_{wall} = \frac{Q - h_{\infty}\,A_{j\infty}\,(T_{\infty} - T_{wb})}{h_w\,A_{jt}} + T_{wb}$$

$$= \frac{12588 - 3.7 \cdot 6.7\ (18 - 10.6)}{1187.6 \cdot 6.07} + 10.6$$

$$= 12.21°C$$

Convection on the inside of the tank

the film temperature, using equation (3-12)

$$T_{film} = \frac{T_j + T_{wall}}{2}$$

$$= \frac{(19.8 + 12.21)}{2}$$

$$= 16.0\ °C$$

volume coefficient of expansion at T_{film}, using equation (A-24)

$$\beta = 1.492 \cdot 10^{-4}\ \ °C^{-1}$$

Grashof number

$$Gr = \frac{9.81\beta(T_j - T_{wall})W_j^3}{\nu^2}$$

$$= \frac{9.8 \cdot 1.492 \cdot 10^{-4}(19.8 - 12.21)(0.23)^3}{(0.9327 \cdot 10^{-6})^2}$$

$$= 1.554 \cdot 10^8$$

Prandtl number, using equation (A-10)

$$Pr = \frac{C_{pj}\mu_j}{k_j}$$

$$= \frac{3645 \cdot 1.11 \cdot 10^{-3}}{0.596}$$

$$= 6.8$$

Nusselt number, using equation (4-6)

$$Nu_j = 0.59(Gr\,Pr)^{\frac{1}{4}}$$

$$= 0.59(1.554 \cdot 10^{8} \cdot 6.8)^{\frac{1}{4}}$$

$$= 106.2$$

convection coefficient, using equation (4-13)

$$h_j = \frac{Nu_j k_j}{W_j}$$

$$= \frac{106.2 \cdot 0.596}{0.23}$$

$$= 273.3 \ \ W/m^2{}^\circ C$$

total heat transfer to the cooling water, combing equations (4-1),(4-2) and (4-3) with equation (4-4)

$$Q = \frac{T_\infty - T_{wb}}{\frac{1}{h_\infty A_{j\infty}}} + \frac{T_j - T_{wb}}{\frac{1}{h_w A_{jt}} + \frac{1}{h_j A_{jt}}} \tag{A-32}$$

$$= \frac{18 - 10.6}{\frac{1}{3.7 \cdot 6.7}} + \frac{19.8 - 10.6}{\frac{1}{1187.6 \cdot 6.07} + \frac{1}{273.3 \cdot 6.07}}$$

$$= 12591 \text{ W}$$

This result hardly differs from the result produced by equation (3-11) and can thus be accepted as the predicted cooling capacity of this Jacket heat exchanger.

Pressure loss

friction factor inside the jacket, using equation (A-14)

$$f_p = (1.82\log(Re_{de}) - 1.64)^{-2}$$

$$= (1.82\log(3322) - 1.64)^{-2}$$

$$= 0.044$$

water velocity inside the circular pipe sections, using equation (A-12)

$$v_w = \frac{m_w \cdot 10^{-3}}{A_p}$$

$$= \frac{0.5 \cdot 10^{-3}}{\pi(0.011)^2}$$

$$= 1.315 \text{ m/s}$$

Reynolds number inside the circular pipe sections, using equation (A-13)

$$Re_d = \frac{v_w d_p \rho_w}{\mu_w}$$

$$= \frac{1.315 \cdot 0.022 \cdot 999.7}{1.29 \cdot 10^{-3}}$$

$$= 22425$$

friction factor inside the circular pipe sections, using equation (A-14)

$$f_p = (1.82 \log(Re_d) - 1.64)^{-2}$$

$$= (1.82 \log(22425) - 1.64)^{-2}$$

$$= 0.0253$$

pressure loss, using equation (3-28)

$$\Delta p = \rho_w \left[\left(f_{d1} \frac{L_c}{de} + 1 \right) \frac{v_{w1}^{\ 2}}{2} + \left(f_{d2} \frac{L_p}{d_p} + 10.9 \right) \frac{v_{w2}^{\ 2}}{2} \right]$$

$$= 999.7 \left[\left(0.044 \frac{26.4}{0.01915} + 1 \right) \frac{0.223^2}{2} + \left(0.0253 \frac{0.71}{0.022} + 10.9 \right) \frac{1.315^2}{2} \right]$$

$$= 11405 \ \ \mathrm{Pa}$$

CALCULATION OF THE WATER-FILM MEAN VELOCITY, SURFACE VELOCITY AND THICKNESS

Balance of the forces

$$\mu \frac{dv}{dy}\, dx = \rho g(\delta - y)\, dx \qquad \text{(A-33)}$$

$$dv = \rho g \frac{(\delta - y)}{\mu}\, dy$$

$$\int dv = \frac{\rho g}{\mu} \int (\rho - y)\, dy$$

$$v = \frac{\rho g}{\mu} \left[\delta y - \frac{y^2}{2} \right] \qquad \text{(A-34)}$$

where equation (A-34) is the general equation for the velocity of the water-film.

Mean velocity

$$v_{mean} = \frac{1}{\delta} \int_0^{\delta} v dy \qquad \text{(A-35)}$$

substituting equation (A-34) into equation (A-35)

$$v_{mean} = \frac{1}{\delta} \int_0^{\delta} \frac{\rho g}{\mu} \left[\delta y - \frac{y^2}{2} \right] dy$$

$$v_{mean} = \frac{\rho g \delta^2}{3\mu} \qquad \text{(A-36)}$$

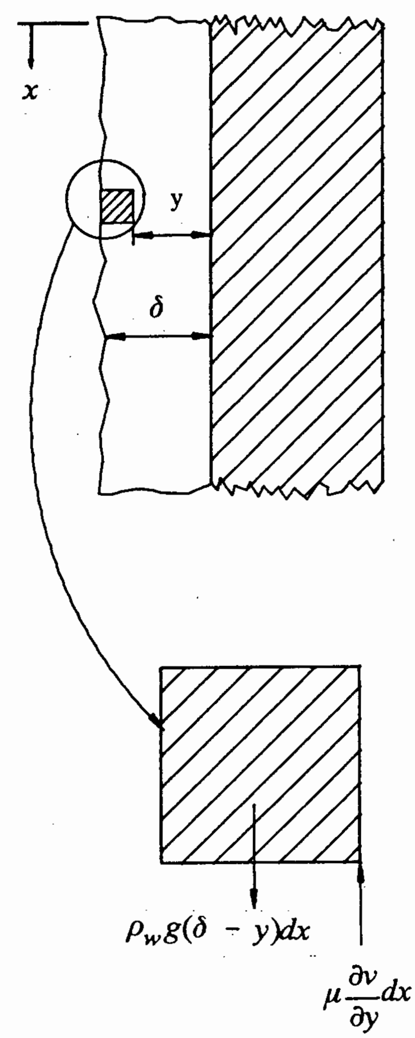


Figure A.2 Force analysis on a elemental volume in the water-film.

Maximum velocity where y = δ

$$v_{\max} = \frac{\rho_w g \delta^2}{2\mu} \tag{A-37}$$

The cross-sectional area of the water-film is calculated as follows

$$A = \frac{m_w \cdot 10^{-5}}{v_{mean}} \tag{A-38}$$

OR

$$A = \frac{\pi}{4}\left[(2\delta + D_t)^2 - D_t^2\right]$$

$$= \frac{\pi}{4}(4\delta^2 + 4\delta D_t)$$

and the term $4\delta^2$ does not have any significant contribution to the value of A, thus

$$A = \frac{\pi}{4}(4\delta D_t) \tag{A-39}$$

$$= \pi\delta D_t$$

set equation (A-38) equal to equation (A-39)

$$\pi\delta D_t = \frac{m_w \cdot 10^{-3}}{v_{mean}}$$

$$= \frac{3\mu\, m_w \cdot 10^{-3}}{\rho_w g \delta^2}$$

$$\Rightarrow \delta^3 = \frac{3\mu\, m_w \cdot 10^{-3}}{\rho_{wg} \pi D_t}$$

$$\delta = \left[\frac{3\mu m_w \cdot 10^{-3}}{\rho_w g \pi D_t}\right]^{\frac{1}{3}} \tag{A-40}$$

THE WATER-FILM ON TOP OF THE TANK

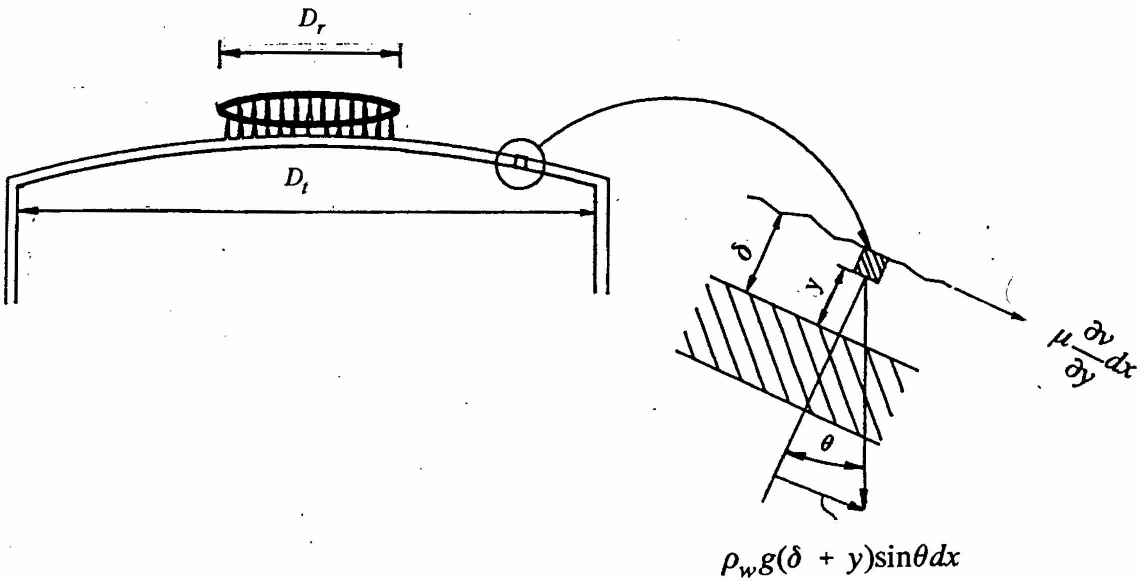


Figure A.3 Force analysis on a elemental volume in the water-film on top of the tank.

Velocity

equation (A-34) becomes

$$v = \rho_w g \left[\delta y - \frac{y^2}{2} \right] \sin\theta \tag{A-41}$$

Mean velocity

equation (A-36) becomes

$$v_{mean} = \frac{\rho_w g \delta^2 \sin\theta}{3\mu} \tag{A-42}$$

Maximum velocity

equation (A-37) becomes

$$v_{\max} = \frac{\rho_w g \delta^2 \sin\theta}{2\mu} \tag{A-43}$$

Water-film thickness

equation (A-39) becomes

$$A = \pi\delta \, \frac{D_t + 1.65}{2} \qquad \text{(A-44)}$$

substitute equation (A-42) in (A-38) and set (A-38) equal to (A-44)

$$\frac{\pi\delta(D_t + 1.65)}{2} = \frac{3\mu\, m_w \cdot 10^{-3}}{\rho_w g \delta^2 \sin\theta}$$

$$\Rightarrow \delta = \left[\frac{3\mu\, m_w \cdot 10^{-3}}{\rho_w g \pi\, (D_t + 1.65) \sin\theta}\right]^{\frac{1}{3}} \qquad \text{(A-45)}$$

CALCULATION ON A WINE TANK WITH WATER-FILM COOLING

Cooling capacity

Following are the heat transfer calculations on a 80 000 litre wine tank with water-film cooling. The cooling water enters at a temperature of 8.7 °C and a mass flow of 0.6 kg/s while the juice is at 15 °C. The ambient drybulb and wetbulb temperatures are 18 °C and 16.5 °C. The wine-maker wants to ferment a maximum of 2 degrees balling per day.

The tank dimensions are as follows:

*	Hight of tank (m)	= 7.570 m
*	Hight of juice level (m)	= 6.427 m
*	Average height of the gutters (m)	= 1.027 m
*	Diameter of tank (m)	= 3.800 m
*	Diameter of ring above tank (m)	= 1.650 m
*	Distance from the juice level to the top of the tank wall (m)	= 0.823 m

Parameters which stays constant throughout the calculations are calculated first.

On top of the wine tank

water-film thickness, using equation (A-45)

$$\delta = \left[\frac{3\mu\, m_w \cdot 10^{-3}}{\rho_w g \pi (D_t + 1.65) \sin\theta}\right]^{\frac{1}{3}}$$

$$= \left[\frac{3 \cdot 1.226 \cdot 10^{-3} \cdot 0.6 \cdot 10^{-3}}{999.4 \cdot 9.81 \cdot \pi (3.8 + 1.65) \sin (20°)}\right]^{\frac{1}{3}}$$

$$= 0.00034\ m$$

mean velocity, using equation (A-42)

$$v_{mean} = \frac{\rho_w g \delta^2 \sin\theta}{3\mu}$$

$$= \frac{999.4 \cdot 9.81(4.6 \cdot 10^{-4})^2 \cdot \sin(20°)}{3 \cdot 1.226 \cdot 10^{-3}}$$

$$= 0.104 \ m/s$$

maximum velocity, using equation (A-43)

$$v_{\max} = \frac{\rho_w g \delta^2 \sin\theta}{2\mu}$$

$$= \frac{999.4 \cdot 9.81(4.6 \cdot 10^{-4})^2 \sin(20°)}{2 \cdot 1.226 \cdot 10^{-3}}$$

$$= 0.156 \, m/s$$

initial water-film area on top of the tank

$$A_{it} = \frac{\pi (D_t^2 - 1.65^2)}{4\cos(\theta)} \qquad \text{(A-46)}$$

$$= \frac{\pi (3.8^2 - 1.65^2)}{4\cos(20)}$$

$$= 9.794 \, m^2$$

initial water-film area on the side of the tank

$$A_{is} = 0.823 \, \pi \, D_t \qquad \text{(A-47)}$$

$$= 0.823 \cdot \pi \cdot 3.8$$

$$= 9.825 \, m^2$$

total initial area

$$A_{wta} = A_{it} + A_{is} \tag{A-48}$$

$$= 9.794 + 9.824$$

$$= 19.619\, m^2$$

wetted tank area

(This is the tank area wetted by the water-film on the on side and fermenting juice on the other.)

$$A_{wtj} = 5.72\, \pi D_t \tag{A-49}$$

$$= 5.72\, \pi \cdot 3.8$$

$$= 68.286\, m^2$$

water-film thickness, using (A-40)

$$\delta = \left[\frac{3 \mu\, m_w \cdot 10^{-3}}{\rho_w g \pi D_t} \right]^{\frac{1}{3}}$$

$$= \left[\frac{3 \cdot 1.226 \cdot 10^{-3} \cdot 0.6 \cdot 10^{-3}}{999.4 \cdot 9.81\, \pi \cdot 3.8} \right]^{\frac{1}{3}}$$

$$= 2.67 \cdot 10^{-4}\, m$$

mean velocity, using equation (A-36)

$$v_{mean} = \frac{\rho_w g \delta^2}{3\mu}$$

$$= \frac{999.4 \cdot 9.81(2.6710^{-4})^2}{3 \cdot 1.226 \cdot 10^{-3}}$$

$$= 0.188\, m/s$$

maximum velocity, using equation (A-37)

$$v_{\max} = \frac{\rho_w g \delta^2}{2\mu}$$

$$= \frac{999.4 \cdot 9.81(2.67 \cdot 10^{-4})^2}{2 \cdot 1.226 \cdot 10^{-3}}$$

$$= 0.282\, m/s$$

Initial heat transfer to the water-film

First of all the heat transfer to the water-film on the initial water-film area is calculated. It is assumed that heat transferred from the still air inside the tank is insignificant. An initial value for T_{cwo} is chosen and the final value (for this area only) is calculated as described in section 5.2.2 to be 9.17°C.

The water bulk temperature is calculated using equation (3-6)

$$T_{wbi} = \frac{T_{cwo} + T_{cwi}}{2}$$

$$= \frac{9.17 + 8.7}{2}$$

$$= 8.935\ °C$$

Radiation

the radiation heat transfer coefficient, using equation (5-4)

$$h_r = \frac{\epsilon\sigma(T_\infty^4 - T_{wbi}^4)}{(T_\infty - T_{wbi})}$$

$$= \frac{0.95 \cdot 5.669 \cdot 10^{-8}(291.15^4 - 282.1^4)}{(291.15 - 282.1)}$$

$$= 5.074 \ W/m^2 {}^\circ C$$

radiation heat

$$Q_r = h_r A_{wta}(T_\infty - T_{wbi}) \tag{A-50}$$

$$= 5.074 \cdot 19.619(18 - 8.935)$$

$$= 902 \ \text{W}$$

Convection

film temperature, using equation (3-12)

$$T_f = \frac{T_\infty + T_{wb}}{2}$$

$$= \frac{18 + 8.935}{2}$$

$$= 13.47 \ {}^\circ C$$

Reynolds number, using equation (A-13) repeated here.

$$Re = \frac{v_{\max} x \rho_\infty}{\mu_\infty}$$

$$= \frac{0.282 \left[\frac{3.8 - 1.65}{2\cos(20)} + 0.823 \right] \cdot 1.221}{1.771 \cdot 10^{-5}}$$

$$= 38243$$

Prandtl number, using equation (3-21) repeated here

$$Pr = \frac{C_{p_\infty} \mu_\infty}{k_\infty}$$

$$= \frac{1017.7 \cdot 1.7716 \cdot 10^{-5}}{0.0251}$$

$$= 0.72$$

heat transfer coefficient, using equation (5-5)

$$h_c = 0.332\, v_{\max} \rho_\infty C_{p_\infty} Re^{-\frac{1}{2}} Pr^{-\frac{2}{3}}$$

$$= 0.332 \cdot 0.282 \cdot 1.221 \cdot 1017.7\, (38243)^{-\frac{1}{2}} (0.72)^{-\frac{2}{3}}$$

$$= 0.741 \;\; W/m^2\,°C$$

convective heat transfer

$$Q_h = h_c A_{wta} (T_\infty - T_{wbi}) \tag{A-51}$$

$$= 0.741 \cdot 19.619\; (18 - 8.935)$$

$$= 132 \;\; \mathrm{W}$$

Mass transfer

mass heat transfer coefficient, using equation (5-6)

$$h_m = \frac{i_v h_D}{R_w (T_{wetb} - T_{wbi})} \left[\frac{P_{vw}}{T_{wbi}} - \frac{P_{v\infty}}{T_{wetb}} \right]$$

$$= \frac{2465428 \cdot 0.7 \cdot 10^{-3}}{461.5(16.5 - 8.935)} \left[\frac{1780}{282} - \frac{1142}{289.5} \right]$$

$$= 1.09 \ W/m^2 \, °C$$

heat due to mass transfer

$$Q_m = h_m A_{wta} (T_{wetb} - T_{wbi}) \tag{A-52}$$

$$= 1.09 \cdot 16.619(16.5 - 8.935)$$

$$= 162 \quad W$$

Initial heat transfer to the water, using equation (5-11)

$$Q_i = \frac{A_{wta}(T_\infty - T_{wbi})}{\left[h_r + h_c + \frac{h_m (T_{wetb} - T_{wbi})}{T_\infty - T_{wbi}} \right]^{-1}}$$

$$= \frac{16.619(18 - 8.935)}{\left[5.074 + 0.741 + \frac{1.09(16.5 - 8.935)}{18 - 8.935} \right]^{-1}}$$

$$= 1196 \quad W$$

A new value for T_{cwo} is calculated by

$$T_{wjl} = \frac{Q_i}{m_w C_{pw}} + T_{cwi} \tag{A-53}$$

$$= \frac{1196}{0.6 \cdot 4198} + 8.7$$

$$= 9.1748 \text{ °}C$$

$$\sim 9.17 \text{ °}C$$

The heat transfer to the cooling water on the initial area is now predicted. (If the value of T_{wjl} calculated with equation (A-52) differs from the initial value of T_{cwo}, then the value of T_{cwo} is decreased or increased and the iteration continue until agreement is reached.)

Heat transfer from the juice level

(The heat transfer to the cooling water that will be calculated from here onwards, is transferred to the tank area that is wetted by the water-film on the one side and fermenting juice on the other.)

The value of T_{wjl} calculated in equation (A-52) is now used as the cold water inlet temperature for the area described in the paragraph above.

(A-54)

$$T_{cwi} = T_{wjl}$$

An initial value for T_{cwo} is chosen and the final value is calculated as described in section 5.2.2 to be 14.69 °C.

water bulk temperature, using equation (3-6) repeated here

$$T_{wbj} = \frac{T_{cwo} + T_{wjl}}{2}$$

$$= \frac{14.69 + 9.17}{2}$$

$$= 11.93\ °C.$$

Heat transfer from the surroundings

Radiation

the radiation heat transfer coefficient, using equation (5-4)

$$h_r = \epsilon\sigma(T_\infty^2 + T_{wbj}^2)(T_\infty + T_{wbj})$$

$$= 0.95 \cdot 5.669 \cdot 10^{-8}(291.15^2 + 285.1^2)(291.15 + 285.1)$$

$$= 5.153\ W/m^2\,°C$$

radiation heat

$$Q_r = h_r A_{wja}(T_\infty - T_{wbj}) \tag{A-55}$$

$$= 5.153 \cdot 68.286(18 - 11.93)$$

$$= 2136\ \ W$$

Convection

film temperature, using equation (3-12)

$$T_f = \frac{T_\infty + T_{wbj}}{2}$$

$$= \frac{18 + 11.93}{2}$$

$$= 14.97 \ °C$$

Reynolds number, using equation (A-13) repeated here.

$$Re = \frac{v_{max} x \rho_\infty}{\mu_\infty}$$

$$= \frac{0.282 \cdot 5.72 \cdot 1.216}{1.78 \cdot 10^{-5}}$$

$$= 110143$$

Prandtl number, using equation (3-21)

$$Pr = \frac{C_{p\infty} \mu_\infty}{k_\infty}$$

$$= \frac{1017.3 \cdot 1.78 \cdot 10^{-5}}{0.0252}$$

$$= 0.72$$

heat transfer coefficient, using equation (5-5) repeated here

$$h_c = 0.332 \, v_{max} \rho_\infty C_{p\infty} Re^{-\frac{1}{2}} Pr^{-\frac{2}{3}}$$

$$= 0.332 \cdot 0.282 \cdot 1.216 \cdot 1017.3 \, (110143)^{-\frac{1}{2}} (0.72)^{-\frac{2}{3}}$$

$$= 0.4344 \ W/m^2 \, °C$$

convective heat transfer using equation (A-51) repeated here

$$Q_h = h_c A_{wtj} (T_\infty - T_{wbj})$$

$$= 0.4344 \cdot 68.286\ (18 - 11.93)$$

$$= 180\ W$$

Mass transfer

mass heat transfer coefficient, using equation (5-6)

$$h_m = \frac{i_v h_D}{R_w\ (T_{wetb} - T_{wbj})} \left[\frac{P_{vw}}{T_{wbi}} - \frac{P_{v\infty}}{T_{wetb}} \right]$$

$$= \frac{2465428 \cdot 4.1 \cdot 10^{-4}}{461.5(16.5 - 11.93)} \left[\frac{1780}{285} - \frac{1396}{289.5} \right]$$

$$= 0.662\ W/m^2\ °C$$

heat due to mass transfer, using equation (A-52)

$$Q_m = h_m A_{wtj} (T_{wetb} - T_{wbj})$$

$$= 0.662 \cdot 68.286(16.5 - 11.93)$$

$$= 207\ W$$

An initial temperature for the tank wall (T_{wall}) is chosen and the final value is calculated as described in section 5.22 to be 13.76 °C.

Heat transferred between the water-film and the tank wall (due to fermentation)

(The water-film on the tank wall is treated as water flow over a flat plate.)

Reynolds number, using equation (A-13)

$$Re = \frac{v_{mean}\, x\, \rho_w}{\mu_w}$$

$$= \frac{0.188 \cdot 5.72 \cdot 999}{1.23 \cdot 10^{-3}}$$

$$= 873402$$

Prandtl number, using equation (3-21)

$$Pr = \frac{C_{p\infty}\mu_\infty}{k_\infty}$$

$$= \frac{4194 \cdot 0.00123}{0.590}$$

$$= 8.76$$

convection heat transfer coefficient, using equation (5-8) repeated here

$$h_w = \frac{0.453\, Re^{\frac{1}{2}}\, Pr^{\frac{1}{3}}\, k_w}{H_{wt}}$$

$$= \frac{0.453\ (873402)^{\frac{1}{2}}\ (8.76)^{\frac{1}{3}} \cdot 0.59}{5.72}$$

$$= 89.9\ W/m^2\ {}^\circ C$$

heat transfer between the tank wall and the water-film

$$Q_{wt} = h_w A_{wtj}\ (T_{wall} - T_{wbj}) \tag{A-56}$$

$$= 89.9 \cdot 68.286\ (13.76 - 11.93)$$

$$= 11258\ W$$

Heat transferred between the juice and the tank wall

film temperature, using equation (3-12)

$$T_f = \frac{T_j + T_{wall}}{2}$$

$$= \frac{15 + 13.76}{2}$$

$$= 14.38\ °C$$

Prandtl number, using equation (3-21)

$$Pr = \frac{C_{pj}\,\mu_j}{k_j}$$

$$= \frac{3645 \cdot 1.14 \cdot 10^{-3}}{0.595}$$

$$= 6.98$$

expansion coefficient, using equation (A-24)

$$\beta = 1.492 \cdot 10^{-4}$$

Grashof number, using equation (3-22)

$$Gr = \frac{9.81\beta\,(T_j - T_{wall})\,H_{wt}^3}{\nu^2}$$

$$= \frac{9.81 \cdot 1.492 \cdot 10^{-4} (15 - 13.76) \cdot 5.72^3}{(1.054 \cdot 10^{-6})^2}$$

$$= 3.058 \cdot 10^{11}$$

natural convection heat transfer coefficient, using equation (5-10)

$$h_j = \frac{0.1 \, (GrPr)^{\frac{1}{3}} \, k_j}{H_{wt}}$$

$$= \frac{0.1 \, (3.058 \cdot 10^{11} \cdot 6.98)^{\frac{1}{3}} \, 0.595}{5.72}$$

$$= 134.0 \; W/m^2 \, {}^\circ C$$

heat transfer between the tank wall and the fermenting juice

$$Q_{wj} = h_j A_{wtj} \, (T_j - T_{wall}) \tag{A-57}$$

$$= 134 \cdot 68.286 \, (15 - 13.76)$$

$$= 11310 \; W$$

Total heat transfer to the water-film from the juice level downwards, using equations (5-2)

$$Q_{wtj} = \frac{A_{wtj} (T_\infty - T_{wbj})}{\left[h_r + h_c + \frac{h_m (T_{wetb} - T_{wbj})}{T_\infty - T_{wbj}} \right]^{-1}} + \frac{A_{wtj} (T_j - T_{wbj})}{\frac{1}{h_w} + \frac{1}{h_j}}$$

$$= \frac{68.286 (18 - 11.93)}{\left[5.153 + 0.4344 + \frac{0.662 (16.5 - 11.93)}{18 - 11.93} \right]^{-1}} + \frac{68.286 (15 - 11.93)}{\frac{1}{89.9} + \frac{1}{134}}$$

$$= 13802 W$$

Total heat transfer to the water-film, using sum of the results calculated by equations (5-2) and (5-11) and comparing it with the result calculated by equation (3-11)

$$Q = Q_i + Q_{wj} \tag{A-58}$$

$$= 1196 + 13802$$

$$= 14998 \quad W$$

equation (3-11)

$$Q = m_w C_{pw} (T_{cwo} - T_{cwi})$$

$$= 0.6 \cdot 4198(14.69 - 8.7)$$

$$= 15088 \quad W$$

The different values calculated for the cooling capacity is at this stage within 0.6 % of one another. The computer can continue to decrease this difference until the operator is satisfied that the Water-film wine tank is theoretically simulated.

The contribution from the surroundings is as follows:

$$radiation = \frac{902 + 2136}{14998} \times 100$$

$$= 20.26\%$$

$$convection = \frac{132 + 180}{14998} \times 100$$

$$= 2.08\%$$

$$mass\ transfer = \frac{162 + 207}{14998} \times 100$$

$$= 2.48\%$$

which means that 24.8 % of the total heat gained, was from the surroundings and only 75.2 % from the fermenting juice. (Just for interest sake, if the dry bulb and wet bulb temperatures were each 3 °C higher, the heat contributed from the surrounding would be 34.5 % of the total heat gain.)

Appendix B

Experimental data

Table B.1 Experimental data of a 1820 x 400mm Weideman heat exchanger with cooling water entering at 6 °C

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
6.19	8.09	15.14	0.799
6.23	8.14	15.14	0.801
6.20	8.09	15.13	0.803
6.17	8.02	15.12	0.803
6.15	8.09	15.14	0.803
5.99	8.06	15.09	0.692
5.92	8.06	15.10	0.702
5.87	7.96	15.14	0.696
5.85	7.89	15.08	0.695
5.86	8.05	15.09	0.593
6.00	8.30	15.05	0.595
6.18	8.44	15.05	0.596
6.21	8.72	14.99	0.498
6.20	8.65	14.96	0.498
6.19	8.59	15.03	0.499
6.15	8.95	15.01	0.400
6.09	8.81	14.96	0.400
6.12	8.86	14.97	0.400
6.14	8.97	14.95	0.400
6.27	9.56	14.89	0.302
6.29	9.47	14.91	0.303
6.26	9.43	14.93	0.301

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
5.93	8.06	14.84	0.794
5.83	7.90	14.82	0.805
5.74	7.75	14.82	0.803
5.73	7.80	14.84	0.803
6.22	8.47	14.85	0.708
6.16	8.39	14.84	0.702
6.12	8.33	14.81	0.698
6.07	8.37	14.83	0.700
6.28	8.73	14.87	0.604
6.26	8.80	14.88	0.592
6.28	8.74	14.85	0.593
6.29	8.79	14.90	0.592
6.12	8.89	14.90	0.494
6.10	8.89	14.88	0.492
6.15	8.97	14.87	0.492
6.16	8.90	14.86	0.493
6.14	8.96	14.88	0.492
6.16	8.79	14.89	0.494
5.86	8.98	14.95	0.405
5.89	8.91	14.93	0.405
5.91	8.95	14.88	0.407
5.92	8.84	14.90	0.410
5.75	9.31	14.94	0.301
5.75	9.43	14.89	0.301
5.79	9.30	14.91	0.301
5.79	9.28	14.93	0.310

Table B.2 **Experimental data of a 1820 x 400mm Weideman heat exchanger with cooling water entering at 8 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
8.21	9.83	15.10	0.796
8.12	9.71	15.05	0.798
8.07	9.70	15.06	0.799
7.73	9.51	15.04	0.690
7.90	9.63	15.04	0.692
8.02	9.81	15.01	0.692
8.01	9.80	15.04	0.696
8.30	10.09	15.01	0.600
8.30	10.16	15.03	0.596
8.32	10.19	15.04	0.600
8.25	10.26	15.00	0.498
8.18	10.28	15.01	0.496
8.19	10.26	15.01	0.492
8.03	10.32	14.98	0.401
8.01	10.37	15.05	0.400
7.98	10.35	14.96	0.402
8.00	10.31	14.98	0.401
8.08	10.71	15.02	0.301
8.06	10.78	15.00	0.301
8.04	10.63	15.05	0.302
8.03	10.70	15.00	0.301

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
8.33	10.10	15.16	0.792
8.29	10.02	15.12	0.796
8.22	9.92	15.14	0.794
7.88	9.79	15.15	0.694
7.86	9.71	15.14	0.702
7.79	9.69	15.15	0.702
7.73	9.60	15.16	0.698
7.71	9.77	15.12	0.602
7.74	9.77	15.09	0.600
7.72	9.87	15.09	0.600
7.77	9.72	15.14	0.600
7.80	9.79	15.09	0.600
7.80	9.89	15.12	0.602
7.81	9.83	15.09	0.600
7.84	9.95	15.10	0.600
7.86	9.94	15.07	0.600
7.85	10.00	15.07	0.600
7.89	9.95	15.10	0.600
7.91	9.89	15.08	0.602
7.93	10.26	15.09	0.492
7.90	10.21	15.07	0.493
7.90	10.27	15.09	0.494
8.00	10.50	15.09	0.406
8.06	10.51	15.11	0.405
8.09	10.60	15.08	0.405
8.04	10.58	15.08	0.405
8.05	10.57	15.06	0.404
8.06	10.57	15.09	0.406
8.16	11.04	15.09	0.309
8.16	11.02	15.10	0.306
8.17	10.96	15.12	0.305
8.27	11.08	15.09	0.305

Table B.3 Experimental data of a 1820 x 400mm Weideman heat exchanger with cooling water entering at 10 °C

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
9.68	10.98	15.41	0.808
9.74	10.93	15.33	0.800
9.73	10.90	15.36	0.806
9.77	10.90	15.32	0.800
9.82	11.07	15.34	0.704
9.80	11.08	15.37	0.702
9.82	11.04	15.32	0.705
9.82	11.14	15.32	0.700
10.01	11.40	15.26	0.595
9.94	11.30	15.29	0.594
9.92	11.34	15.31	0.595
9.99	11.46	15.28	0.595
10.06	11.54	15.30	0.503
10.09	11.63	15.27	0.498
10.06	11.51	15.33	0.500
10.13	11.63	15.29	0.504
10.10	11.75	15.30	0.407
10.19	11.93	15.27	0.404
10.26	11.95	15.28	0.404
10.24	12.03	15.28	0.403
10.39	12.25	15.28	0.308
10.45	12.45	15.29	0.303
10.39	12.25	15.31	0.305
10.46	12.76	15.30	0.304

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
10.15	11.48	15.27	0.792
10.08	11.35	15.23	0.794
10.00	11.30	15.24	0.801
9.90	11.14	15.28	0.794
9.83	11.10	15.27	0.794
9.71	11.14	15.19	0.704
9.70	11.06	15.21	0.704
9.70	11.09	15.19	0.706
9.71	11.11	15.20	0.702
9.73	11.16	15.19	0.702
9.93	11.43	15.24	0.605
9.90	11.38	15.16	0.607
9.91	11.35	15.18	0.607
9.95	11.36	15.24	0.607
9.94	11.61	15.20	0.502
9.97	11.61	15.20	0.502
9.99	11.59	15.20	0.502
9.97	11.63	15.17	0.502
10.07	12.02	15.20	0.406
10.08	11.97	15.23	0.406
10.15	12.22	15.22	0.407
10.18	12.19	15.15	0.406
10.19	11.96	15.19	0.407
10.33	12.48	15.19	0.304
10.33	12.58	15.21	0.304
10.39	12.41	15.18	0.304
10.36	12.63	15.15	0.304
10.35	12.53	15.20	0.304

Table B.4 **Experimental data of a 4550 x 400mm Weideman heat exchanger with cooling water entering at 6 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
6.22	12.06	14.35	0.294
6.20	12.09	14.34	0.293
6.07	11.94	14.35	0.294
6.04	11.92	14.34	0.295
6.02	11.84	14.34	0.294
5.98	12.12	14.33	0.295
6.25	11.68	14.23	0.395
6.23	11.65	14.23	0.402
6.28	11.65	14.24	0.395
6.28	11.83	14.24	0.395
6.20	11.56	14.26	0.395
6.20	11.63	14.26	0.395
6.18	11.04	14.14	0.504
6.22	10.99	14.16	0.504
6.20	11.14	14.17	0.503
6.17	11.08	14.12	0.504
6.15	11.02	14.15	0.504
5.91	10.36	14.13	0.605
5.82	10.24	14.14	0.604
5.82	10.17	14.12	0.604
5.75	10.16	14.14	0.605
5.76	10.19	14.10	0.604
5.78	10.00	14.12	0.700
5.86	9.87	14.08	0.700
5.86	9.85	14.12	0.700
5.85	9.83	14.09	0.700
5.85	9.87	14.08	0.700
5.90	9.96	14.04	0.700
5.92	10.05	14.07	0.700
5.90	9.99	14.11	0.700
6.13	9.96	14.06	0.808
6.19	9.79	14.02	0.794
6.17	9.77	14.03	0.794
6.25	9.84	14.05	0.794
6.20	9.89	14.05	0.796
6.31	12.34	14.40	0.301
6.26	12.32	14.36	0.301
6.25	12.36	14.37	0.301
6.20	12.30	14.37	0.301
6.17	12.28	14.40	0.301
6.12	12.25	14.39	0.302
6.04	11.72	14.38	0.407
6.05	11.64	14.41	0.408
6.09	11.59	14.39	0.407
6.14	11.64	14.41	0.406
6.16	11.79	14.41	0.398
5.72	10.88	14.43	0.495
5.75	10.91	14.39	0.493
5.83	11.04	14.40	0.495
5.84	10.94	14.40	0.495
5.89	11.05	14.42	0.495
5.92	11.12	14.43	0.494
5.97	10.95	14.44	0.503
6.29	11.02	14.54	0.599
6.31	11.18	14.49	0.599
6.31	11.07	14.55	0.599
6.31	11.00	14.52	0.599
6.31	10.99	14.52	0.600
5.75	10.14	14.54	0.700
5.78	10.21	14.58	0.698
5.82	10.19	14.54	0.700
5.84	10.16	14.54	0.700
5.88	10.22	14.58	0.700
5.91	10.34	14.57	0.698
5.97	10.35	14.54	0.700
6.14	10.21	14.69	0.796
6.05	10.29	14.70	0.792
5.99	10.34	14.68	0.799

Table B.5 **Experimental data of a 4550 x 400mm Weideman heat exchanger with cooling water entering at 8 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)	T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
7.88	12.77	14.91	0.307	7.91	13.03	14.63	0.293
7.80	12.73	14.95	0.308	7.87	12.95	14.63	0.294
7.81	12.74	14.93	0.307	7.87	12.91	14.66	0.294
7.86	12.77	14.93	0.308	7.81	12.86	14.64	0.294
7.82	12.76	14.99	0.307	7.78	12.85	14.63	0.294
7.72	12.39	14.98	0.393	8.28	12.60	14.62	0.394
7.77	12.35	14.94	0.393	8.23	12.62	14.63	0.395
7.78	12.34	14.97	0.393	8.16	12.64	14.62	0.395
7.87	12.35	14.94	0.395	8.14	12.66	14.62	0.406
8.18	12.20	14.88	0.504	8.18	12.23	14.63	0.500
8.14	11.97	14.93	0.504	8.19	12.28	14.66	0.500
8.14	12.00	14.88	0.504	8.20	12.24	14.70	0.501
8.17	12.02	14.90	0.504	8.19	12.21	14.64	0.506
8.12	12.01	14.89	0.504	8.22	12.24	14.65	0.498
8.18	12.06	14.84	0.504				
8.24	11.93	14.87	0.602	7.99	11.84	14.67	0.594
8.18	11.76	14.91	0.602	7.99	11.91	14.67	0.605
8.27	11.81	14.89	0.604	8.02	11.87	14.70	0.603
8.24	11.81	14.91	0.602	8.02	11.79	14.68	0.605
7.99	11.37	14.86	0.704	7.80	11.44	14.71	0.700
7.93	11.25	14.85	0.702	7.84	11.49	14.68	0.697
7.82	11.23	14.85	0.702	7.84	11.53	14.67	0.695
7.70	11.19	14.84	0.704	7.86	11.50	14.67	0.691
				8.14	11.37	14.68	0.796
				8.15	11.35	14.65	0.796
				7.94	11.23	14.67	0.796
				7.85	11.02	14.68	0.796
				7.77	11.06	14.67	0.796
				7.71	10.94	14.71	0.796

Table B.6 Experimental data of a 4550 x 400mm Weideman heat exchanger with cooling water entering at 10 °C

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)	T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
10.30	13.97	15.15	0.306	10.27	14.03	15.16	0.298
10.29	14.14	15.12	0.303	10.25	13.83	15.12	0.299
10.16	13.89	15.15	0.304	10.23	13.89	15.11	0.297
10.12	13.81	15.11	0.304	10.17	13.79	15.16	0.297
				10.13	13.91	15.13	0.297
				10.10	14.01	15.14	0.297
				10.07	13.81	15.12	0.297
9.73	13.16	15.11	0.404	10.22	13.34	15.18	0.404
9.75	13.02	15.14	0.402	10.23	13.44	15.18	0.402
9.80	13.01	15.07	0.404	10.23	13.45	15.22	0.402
9.76	12.99	15.10	0.404	10.21	13.49	15.18	0.404
9.80	13.06	15.06	0.402	10.25	13.49	15.16	0.402
9.75	13.03	15.09	0.404	10.27	13.46	15.18	0.402
9.95	12.85	15.04	0.494	10.24	13.42	15.22	0.496
9.92	12.90	15.06	0.501	10.18	13.31	15.23	0.496
9.98	12.86	15.05	0.493	10.12	13.28	15.22	0.496
9.98	12.88	15.07	0.493	10.07	13.32	15.20	0.495
9.97	12.91	15.07	0.493				
10.03	12.91	15.06	0.496				
10.07	12.78	15.07	0.591	10.17	12.97	15.30	0.605
10.11	12.75	15.08	0.592	10.13	13.12	15.25	0.600
10.10	12.75	15.10	0.594	10.15	13.09	15.25	0.600
10.14	12.72	15.03	0.595	10.17	12.98	15.28	0.599
10.10	12.76	15.10	0.596	10.21	13.02	15.23	0.600
10.17	12.72	15.04	0.604	10.20	13.01	15.28	0.602
9.74	12.37	15.05	0.702	9.99	12.61	15.36	0.702
9.78	12.34	15.04	0.698	10.02	12.61	15.34	0.708
9.77	12.33	15.04	0.700	10.05	12.63	15.39	0.700
9.71	12.27	15.05	0.700	10.04	12.69	15.33	0.702
9.77	12.30	15.04	0.696				
9.76	12.24	15.01	0.708				
9.88	12.23	15.03	0.792	9.81	12.41	15.34	0.799
9.81	12.21	15.05	0.798	9.87	12.52	15.35	0.799
9.82	12.21	15.08	0.798	9.88	12.51	15.32	0.796
9.88	12.26	15.02	0.803				
9.88	12.25	15.03	0.796				
9.87	12.20	15.03	0.800				
9.88	12.21	15.05	0.798				
9.90	12.15	15.06	0.798				

Table B.7 **Experimental data of a 2500 x 300mm Weideman heat exchanger with cooling water entering at 6 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
6.30	8.61	14.79	0.803
6.20	8.40	14.81	0.803
6.14	8.37	14.81	0.800
6.09	8.35	14.82	0.794
6.03	8.38	14.79	0.792
6.15	8.72	14.77	0.695
6.25	8.86	14.79	0.698
6.23	8.88	14.77	0.698
6.24	8.78	14.74	0.700
6.29	9.18	14.64	0.594
6.27	9.12	14.62	0.595
6.23	9.05	14.69	0.594
6.18	9.11	14.62	0.595
5.94	9.09	14.64	0.509
5.95	9.07	14.62	0.506
5.92	9.09	14.67	0.506
5.91	9.07	14.62	0.507
5.86	9.54	14.62	0.409
5.77	9.40	14.63	0.408
5.74	9.31	14.61	0.408
5.74	9.38	14.63	0.406
5.81	9.82	14.59	0.309
5.77	9.88	14.61	0.308
5.79	9.92	14.57	0.308

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
5.41	7.88	14.53	0.792
5.35	7.77	14.54	0.796
5.28	7.68	14.57	0.796
5.21	7.61	14.55	0.794
5.17	7.66	14.59	0.796
5.11	7.49	14.59	0.794
5.05	7.47	14.51	0.796
5.03	7.49	14.53	0.794
5.01	7.34	14.53	0.796
6.29	8.99	14.54	0.693
6.29	8.99	14.53	0.691
6.30	8.89	14.54	0.691
6.31	8.97	14.57	0.692
6.32	8.97	14.57	0.692
6.30	8.97	14.55	0.693
6.18	9.15	14.56	0.593
6.19	9.21	14.54	0.591
6.18	9.17	14.61	0.592
6.20	9.11	14.60	0.591
6.23	9.21	14.54	0.607
6.05	9.33	14.57	0.509
6.08	9.37	14.58	0.509
6.08	9.42	14.60	0.510
6.11	9.47	14.58	0.508
6.13	9.39	14.59	0.506
6.17	9.42	14.54	0.506
5.77	9.64	14.55	0.401
5.79	9.72	14.58	0.401
5.82	9.65	14.59	0.401
5.90	9.61	14.58	0.402
5.93	9.67	14.56	0.402
5.77	10.36	14.66	0.291
5.73	10.27	14.67	0.292
5.72	10.31	14.63	0.292
5.93	10.37	14.64	0.291
5.88	10.30	14.61	0.293

Table B.8 **Experimental data of a 2500 x 300mm Weideman heat exchanger with cooling water emtering at 8 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
8.23	11.49	14.84	0.300
8.19	11.52	14.86	0.301
8.19	11.44	14.86	0.301
8.23	11.63	14.81	0.301
8.08	10.89	14.84	0.399
8.10	10.86	14.89	0.397
8.08	10.74	14.83	0.399
8.28	10.74	14.88	0.495
8.19	10.60	14.85	0.495
8.14	10.71	14.83	0.493
8.07	10.60	14.81	0.495
8.05	10.47	14.81	0.503
7.75	9.97	14.80	0.602
7.77	9.95	14.79	0.605
7.78	9.98	14.81	0.607
7.82	10.07	14.85	0.605
7.77	9.95	14.79	0.605
7.78	9.98	14.81	0.607
7.82	10.07	14.85	0.605
7.92	9.96	14.82	0.700
7.99	9.94	14.78	0.704
7.95	9.93	14.82	0.702
8.00	9.82	14.79	0.702
8.03	9.80	14.84	0.809
8.05	9.73	14.76	0.803
8.02	9.81	14.79	0.800
8.03	9.77	14.83	0.803
8.09	9.76	14.79	0.800
8.05	9.70	14.79	0.803
8.09	9.85	14.78	0.800

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
7.73	11.38	14.85	0.294
7.69	11.43	14.86	0.305
7.71	11.30	14.91	0.299
7.69	11.25	14.88	0.299
7.70	11.24	14.88	0.298
7.94	10.93	14.89	0.407
7.90	11.03	14.91	0.407
7.87	11.02	14.89	0.408
7.84	10.99	14.88	0.409
7.80	10.96	14.87	0.407
7.80	10.85	14.94	0.407
8.31	10.90	14.88	0.507
8.27	10.88	14.89	0.507
8.22	10.80	14.89	0.507
8.16	10.77	14.91	0.507
8.08	10.73	14.89	0.507
8.06	10.41	14.96	0.609
8.08	10.51	14.97	0.609
8.12	10.56	14.99	0.610
8.12	10.62	14.95	0.607
8.17	10.54	14.97	0.604
8.19	10.53	14.93	0.604
8.20	10.44	14.94	0.591
7.93	10.05	14.97	0.697
7.92	10.08	14.97	0.698
7.94	10.03	14.97	0.698
7.97	10.07	15.00	0.698
7.95	10.06	14.98	0.698
7.96	10.01	14.95	0.698
7.80	9.76	14.99	0.803
7.73	9.70	14.98	0.803
7.71	9.69	15.00	0.805
7.70	9.62	15.01	0.803
7.69	9.66	14.98	0.792
7.72	9.63	15.00	0.792

Table B.9 **Experimental data of a 2500 x 300mm Weideman heat exchanger with cooling water entering at 10 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
9.85	11.09	15.03	0.805
9.85	11.10	14.97	0.805
9.84	11.14	15.03	0.803
9.85	11.04	14.99	0.805
9.93	11.11	14.94	0.807
9.93	11.16	15.01	0.801
10.12	11.47	14.98	0.691
10.12	11.45	14.97	0.693
10.12	11.50	14.98	0.693
10.14	11.53	15.03	0.695
10.21	11.74	15.06	0.599
10.22	11.68	15.03	0.616
10.23	11.72	15.02	0.607
10.22	11.68	14.98	0.604
10.27	11.69	15.02	0.602
9.71	11.46	15.09	0.506
9.76	11.53	15.05	0.506
9.79	11.78	15.06	0.395
9.85	11.79	15.12	0.408
9.88	11.75	15.07	0.404
9.94	11.79	15.06	0.404
9.95	12.20	15.06	0.297
9.95	12.13	15.07	0.298
9.98	12.11	15.05	0.297
10.00	12.19	15.06	0.299
9.99	12.23	15.05	0.299

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
10.10	11.46	14.95	0.803
10.11	11.55	14.91	0.805
10.15	11.53	14.95	0.800
10.21	11.55	15.00	0.800
10.22	11.54	14.96	0.805
10.23	11.56	14.95	0.803
10.24	11.65	14.96	0.800
10.25	11.71	14.90	0.805
9.92	11.39	14.96	0.704
9.96	11.44	14.96	0.698
9.99	11.54	14.98	0.702
10.00	11.52	15.00	0.700
9.73	11.43	15.01	0.605
9.72	11.55	14.95	0.607
9.76	11.42	14.94	0.607
9.77	11.51	14.98	0.607
9.79	11.47	14.96	0.607
10.17	12.17	15.01	0.394
10.23	12.29	15.03	0.393
10.21	12.31	15.02	0.393
10.24	12.24	15.07	0.393
10.02	12.33	15.05	0.309
10.01	12.38	15.01	0.308
10.05	12.38	15.08	0.308
10.07	12.38	15.00	0.309

Table B.10 **Experimental data of a 3500 x 300mm Weideman heat exchanger with cooling water entering at 6 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
5.70	9.10	14.62	0.805
5.72	9.05	14.64	0.807
5.73	9.12	14.63	0.805
6.09	9.71	14.59	0.704
6.17	9.68	14.57	0.704
6.15	9.69	14.59	0.700
6.24	9.72	14.62	0.704
6.22	10.20	14.57	0.607
6.17	10.16	14.57	0.607
6.13	10.11	14.59	0.605
5.82	10.13	14.52	0.498
5.82	10.08	14.53	0.498
5.75	10.14	14.56	0.498
5.74	10.49	14.50	0.401
5.70	10.65	14.54	0.401
5.75	10.50	14.50	0.401
5.77	10.50	14.51	0.401
5.78	10.55	14.51	0.402
6.04	11.55	14.50	0.294
6.10	11.59	14.51	0.294
6.11	11.68	14.57	0.294
6.14	11.57	14.57	0.294
6.14	11.27	14.56	0.294
6.19	11.23	14.52	0.295

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
6.12	9.61	14.71	0.796
6.02	9.46	14.68	0.799
5.95	9.35	14.70	0.799
5.89	9.34	14.75	0.796
5.76	9.47	14.67	0.706
5.82	9.60	14.62	0.706
5.87	9.76	14.65	0.708
5.93	9.61	14.68	0.706
6.08	10.33	14.61	0.600
6.12	10.31	14.65	0.609
6.16	10.31	14.61	0.609
6.13	10.29	14.61	0.603
6.16	10.34	14.58	0.605
6.09	10.64	14.75	0.500
6.05	10.64	14.76	0.498
6.00	10.63	14.67	0.500
5.92	10.49	14.63	0.502
5.91	10.55	14.66	0.502
5.70	10.83	14.80	0.397
5.73	10.85	14.81	0.398
5.76	10.73	14.65	0.398
5.81	10.79	14.67	0.397
5.97	11.67	14.59	0.301
6.01	11.45	14.58	0.301
6.01	11.42	14.59	0.301
6.07	11.59	14.61	0.302
6.10	11.50	14.58	0.302
6.10	11.62	14.58	0.302

Table B.11 **Experimental data of a 3500 x 300mm Weideman heat exchanger with cooling water entering at 8 °C**

T wi (°C)	T wu (°C)	T sap (°C)	Vloeitempo (l/s)
8.24	12.30	14.86	0.299
8.27	12.24	14.87	0.299
8.16	12.19	14.88	0.299
8.03	11.68	14.84	0.392
8.12	11.73	14.82	0.408
8.14	11.75	14.86	0.409
8.27	11.49	14.80	0.500
8.26	11.44	14.81	0.502
8.29	11.51	14.84	0.500
8.22	11.21	14.77	0.597
8.18	11.21	14.77	0.597
8.11	11.15	14.79	0.597
7.74	10.53	14.75	0.700
7.76	10.63	14.72	0.698
7.82	10.64	14.75	0.698
7.95	10.46	14.72	0.807
8.02	10.50	14.74	0.805
7.99	10.54	14.71	0.805
8.02	10.54	14.77	0.805
8.00	10.61	14.71	0.808

T wi (°C)	T wu (°C)	T sap (°C)	Vloeitempo (l/s)
8.04	12.36	14.58	0.293
8.06	12.38	14.59	0.291
8.04	12.29	14.57	0.291
8.05	12.35	14.62	0.292
8.09	12.25	14.57	0.291
8.07	12.20	14.59	0.291
7.85	11.42	14.65	0.401
7.87	11.56	14.60	0.401
7.86	11.69	14.59	0.402
7.92	11.68	14.60	0.401
7.72	11.16	14.63	0.506
7.74	11.33	14.61	0.507
7.77	11.34	14.61	0.506
7.79	11.32	14.57	0.492
7.70	10.80	14.60	0.600
7.73	10.79	14.62	0.602
7.77	10.77	14.64	0.602
8.04	10.83	14.64	0.700
8.05	10.81	14.67	0.702
8.08	10.80	14.68	0.702
8.07	10.88	14.67	0.702
8.05	10.91	14.66	0.693
8.00	10.51	14.67	0.808
7.99	10.87	14.66	0.798
8.01	10.83	14.67	0.798

Table B.12 **Experimental data of a 3500 x 300mm Weideman heat exchanger with cooling water entering at 10 °C**

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
9.70	11.48	14.96	0.803
9.80	11.50	14.91	0.803
9.76	11.53	14.93	0.801
9.77	11.55	14.89	0.803
9.92	11.80	14.86	0.709
9.91	11.83	14.93	0.704
9.94	11.79	14.91	0.704
9.92	11.78	14.83	0.704
10.02	11.81	14.84	0.702
10.03	12.07	14.82	0.604
10.06	11.98	14.80	0.604
10.18	12.08	14.79	0.604
10.17	12.08	14.83	0.604
10.25	12.09	14.85	0.604
10.21	12.47	14.81	0.506
10.18	12.25	14.80	0.504
10.18	12.27	14.78	0.504
9.97	12.51	14.79	0.401
9.90	12.44	14.83	0.401
9.91	12.45	14.82	0.401
9.92	12.42	14.78	0.402
9.85	12.39	14.79	0.401
10.04	12.91	14.79	0.299
10.01	13.01	14.77	0.299
10.02	12.94	14.79	0.299
10.00	12.99	14.78	0.299
10.11	13.07	14.75	0.299
10.04	13.05	14.80	0.299
10.06	12.95	14.78	0.299

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow rate (l/s)
9.73	11.57	14.86	0.806
9.71	11.49	14.88	0.805
9.79	11.60	14.88	0.805
9.82	11.79	14.89	0.807
10.15	12.05	14.97	0.702
10.07	11.99	14.97	0.702
9.98	12.16	14.95	0.702
9.90	12.12	14.98	0.702
9.81	12.09	14.96	0.702
10.11	12.13	15.02	0.609
10.17	12.43	15.04	0.591
10.23	12.46	15.02	0.594
10.25	12.32	15.02	0.594
9.88	12.23	15.04	0.507
9.91	12.21	15.06	0.502
9.92	12.42	15.03	0.502
9.92	12.19	15.06	0.503
9.96	12.46	15.04	0.502
9.78	12.37	15.06	0.393
9.81	12.74	15.08	0.398
9.82	12.70	15.05	0.397
9.79	12.65	15.08	0.398
9.77	12.49	15.13	0.397
10.26	13.36	15.12	0.300
10.19	13.18	15.14	0.299
10.13	13.15	15.08	0.299
10.06	13.16	15.10	0.300
9.96	13.07	15.12	0.299
9.92	13.03	15.08	0.299
9.88	13.07	15.08	0.300

Table B.13 Experimental pressure loss data through a 1820 x 400mm Weideman heat exchanger for different water supply line pressures

Flow rate (l/s)	Pressure loss (kPa)	Line pressure (kPa)
0.252	1.70	84
0.297	2.60	85
0.344	3.50	88
0.393	4.65	94
0.448	6.10	102
0.502	7.50	109
0.553	8.90	115
0.607	10.75	125
0.658	12.70	135
0.707	14.40	140
0.750	16.40	150
0.792	17.90	160

Table B.14 Experimental pressure loss data through a 4550 x 400mm Weideman heat exchanger for different water supply line pressures

Flow rate (l/s)	Pressure drop (kPa)	Line pressure (kPa)
0.257	3.10	86
0.295	3.85	89
0.353	5.20	93
0.392	6.20	97
0.458	8.10	106
0.507	9.60	113
0.558	11.35	120
0.594	12.50	125
0.654	15.00	135
0.698	17.20	142
0.757	19.60	153
0.798	21.60	163

Table B.15 Experimental pressure loss data through a 2500 x 300mm Weideman heat exchanger for different water supply line pressures

Flow rate (l/s)	Pressure loss (kPa)	Line Pressure (kPa)
0.246	9.40	98
0.299	12.70	104
0.353	16.80	112
0.399	21.00	120
0.455	26.40	130
0.492	29.00	137
0.547	36.80	149
0.605	44.50	161
0.643	49.60	170
0.705	58.50	184
0.752	66.50	197
0.800	74.40	210

Table B.16 Experimental pressure loss data through a 3500 x 300mm Weideman heat exchanger for different water supply line pressures

Flow rate (l/s)	Pressure loss (kPa)	Line pressure (kPa)
0.245	7.00	94
0.305	10.70	100
0.355	14.20	108
0.402	17.20	115
0.444	20.40	124
0.491	24.70	131
0.556	31.10	144
0.597	35.00	151
0.647	41.30	162
0.702	47.80	175
0.757	53.50	185
0.799	59.70	197

Experimental data on a jacket tank

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)	Pressure drop (kPa)
8.56	14.39	20.40	0.595	16.9
8.55	14.32	20.53	0.594	17.0
8.58	14.30	20.55	0.595	17.0
8.52	14.30	20.55	0.594	16.9
8.51	14.27	20.36	0.596	17.1
8.56	14.28	20.45	0.595	17.0
8.55	14.25	20.40	0.595	16.9
8.49	14.29	20.49	0.595	16.9
8.49	14.23	20.44	0.595	17.2
8.49	14.25	20.41	0.595	17.0
8.37	14.54	20.39	0.509	12.1
8.35	14.56	20.37	0.503	12.0
8.35	14.54	20.34	0.505	12.0
8.35	14.56	20.30	0.503	12.0
8.32	14.59	20.28	0.504	12.0
8.33	14.59	20.21	0.399	7.3
8.28	14.63	20.21	0.398	7.2
8.30	14.72	20.30	0.399	7.3
8.28	14.75	20.33	0.399	7.4
8.24	14.79	20.31	0.398	7.4
8.21	15.28	20.25	0.295	3.8
8.20	15.36	20.25	0.295	3.8
8.19	15.48	20.24	0.295	3.7
8.17	15.55	20.21	0.295	3.9
8.16	15.64	20.22	0.295	3.8
8.14	15.69	20.24	0.295	3.7

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)	Pressure drop (kPa)
7.66	13.61	19.86	0.600	18.2
7.65	13.59	19.78	0.600	17.7
7.65	13.54	19.88	0.599	17.8
7.62	13.54	19.93	0.600	17.9
7.64	13.52	19.89	0.600	17.7
7.61	13.51	19.92	0.600	17.7
7.64	13.49	19.84	0.600	17.9
7.68	13.50	19.91	0.600	17.9
7.62	13.49	19.90	0.600	17.7
7.59	13.51	19.93	0.600	17.8
7.60	13.49	19.81	0.600	17.9
7.61	13.42	19.83	0.602	17.7
7.56	13.49	19.81	0.498	11.9
7.61	13.55	19.84	0.498	11.8
7.61	13.62	19.76	0.499	11.6
7.57	13.68	19.75	0.498	11.9
7.64	13.75	19.72	0.498	11.8
7.63	13.80	19.82	0.498	12.0
7.53	14.21	19.76	0.400	7.6
7.60	14.25	19.76	0.401	7.5
7.61	14.33	19.75	0.400	7.7
7.58	14.41	19.74	0.401	7.4
7.53	14.43	19.76	0.400	7.6
7.55	14.49	19.70	0.400	7.6
7.47	14.73	19.69	0.300	4.0
7.55	14.77	19.71	0.299	4.0
7.47	14.87	19.67	0.300	4.0
7.43	14.94	19.69	0.299	4.1
7.50	14.99	19.72	0.299	4.1
7.43	15.07	19.65	0.299	3.9
7.49	15.09	19.66	0.299	4.0

Table B.18 Experimental data on a film tank

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)	T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)
8.53	15.21	14.97	0.809	8.77	15.56	14.98	0.794
8.54	15.21	15.05	0.805	8.73	15.47	14.91	0.794
8.74	15.21	15.01	0.807	8.72	15.49	14.95	0.792
8.75	15.26	15.00	0.807	8.71	15.36	14.98	0.794
8.91	15.24	14.99	0.808	8.84	15.43	14.99	0.792
8.50	15.27	15.05	0.807	8.89	15.37	14.97	0.794
8.54	15.29	15.02	0.708	8.85	15.38	15.01	0.706
8.61	15.29	14.98	0.708	8.75	15.30	14.96	0.704
8.54	15.25	15.02	0.708	8.72	15.28	14.95	0.704
8.67	15.26	15.02	0.708	8.69	15.36	15.04	0.704
8.76	15.26	15.01	0.710	8.66	15.40	15.04	0.704
				8.77	15.35	15.06	0.600
				8.81	15.32	15.05	0.599
8.57	15.29	15.06	0.594	8.83	15.23	15.06	0.600
8.57	15.31	15.06	0.597	8.78	15.28	15.07	0.602
8.77	15.32	14.99	0.596	8.78	15.38	15.10	0.600
8.80	15.29	15.04	0.597	8.79	15.34	15.08	0.600
8.82	15.30	15.04	0.597	8.75	15.37	15.10	0.600
8.51	15.33	14.99	0.496				
8.72	15.29	15.03	0.499	8.94	15.33	15.07	0.506
8.67	15.34	14.98	0.499	8.88	15.39	15.10	0.507
8.63	15.33	15.05	0.499	8.87	15.24	15.02	0.506
8.59	15.35	15.00	0.498	8.85	15.40	15.10	0.506
8.63	15.33	15.01	0.499	8.83	15.26	15.03	0.506
8.86	15.33	15.04	0.404	8.87	15.35	15.07	0.394
8.69	15.38	15.03	0.402	8.93	15.26	15.05	0.396
8.63	15.39	15.02	0.404	8.92	15.42	15.10	0.396
8.70	15.37	15.04	0.402	8.91	15.41	15.08	0.395
8.49	15.38	15.04	0.404	8.83	15.28	15.07	0.394
8.81	15.39	15.02	0.304				
8.81	15.40	15.04	0.303				
8.69	15.37	15.04	0.302				
8.61	15.36	15.07	0.303	8.87	15.39	15.08	0.291
8.59	15.36	15.08	0.303	8.89	15.40	15.09	0.292
8.79	15.39	15.07	0.303	8.92	15.39	15.10	0.293
8.92	15.38	15.06	0.303	8.93	15.38	15.06	0.291
8.78	15.43	15.03	0.302	8.88	15.35	15.15	0.292
8.83	15.41	15.08	0.304	8.86	15.31	15.09	0.291

Table B.19 Experimental data on a fouled Weideman heat exchanger

T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)	T cwi (°C)	T cwo (°C)	T juice (°C)	Flow Rate (l/s)
4.88	8.14	18.41	0.245	4.76	8.17	18.29	0.250
4.91	8.04	18.34	0.251	4.95	8.21	18.37	0.250
4.92	8.15	18.33	0.243	4.92	8.28	18.30	0.250
4.95	8.21	18.38	0.245	4.89	8.28	18.38	0.251
4.88	8.02	18.40	0.250	4.92	8.23	18.29	0.250
4.41	7.32	18.53	0.300	4.41	7.62	18.40	0.300
4.31	7.17	18.56	0.300	4.45	7.62	18.44	0.299
4.46	7.31	18.55	0.316	4.45	7.66	18.32	0.300
4.40	7.31	18.54	0.300	4.41	7.47	18.35	0.300
4.44	7.60	18.50	0.300	4.45	7.61	18.39	0.300
4.98	7.74	18.58	0.350	5.05	7.80	18.38	0.356
5.01	7.78	18.57	0.350	5.05	7.88	18.40	0.353
4.95	7.70	18.54	0.351	4.93	7.96	18.47	0.347
4.98	7.74	18.58	0.350	5.01	8.03	18.42	0.348
5.05	7.80	18.53	0.351	5.01	8.04	18.53	0.347
4.73	7.34	18.54	0.399	4.41	7.21	18.40	0.397
4.61	7.22	18.58	0.399	4.57	7.45	18.46	0.392
4.53	7.16	18.57	0.394	4.58	7.47	18.47	0.387
4.68	7.34	18.55	0.392	4.74	7.54	18.42	0.392
4.59	7.38	18.58	0.381	4.76	7.57	18.43	0.390

Table B.20 The average outside Nusselt numbers for the 1820x400 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	Nu_o					
6	382	398	427	438	463	471
8	416	434	448	470	499	510
10	387	417	433	451	481	491

Table B.21 The average outside Nusselt numbers for the 4550x400 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	Nu_o					
6	368	395	416	431	464	456
8	355	389	407	420	444	450
10	341	359	390	398	437	444

Table B.22 The average outside Nusselt numbers for the 2500x300 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	Nu_o					
6	380	405	417	423	426	429
8	367	386	395	407	417	416
10	314	341	361	375	373	393

Table B.23 The average outside Nusselt numbers for the 3485x300 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	Nu_o					
6	389	422	441	478	472	580
8	363	385	412	425	444	447
10	345	362	367	381	392	411

Table B.24 The Grashof-Prandtl numbers (*GrPr*) on the outside of the 1820x400 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	$GrPr \times 10^{-9}$					
6	2.242	2.445	2.564	2.687	2.761	2.844
8	1.688	1.882	1.993	2.071	2.126	2.190
10	1.315	1.419	1.503	1.578	1.607	1.655

Table B.25 The Grashof-Prandtl numbers (*GrPr*) on the outside of the 4550x400 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	$GrPr \times 10^{-9}$					
6	1.642	1.827	1.972	2.097	2.163	2.287
8	1.363	1.502	1.626	1.727	1.799	1.875
10	1.055	1.180	1.253	1.342	1.363	1.426

Table B.26 The Grashof-Prandtl numbers *(GrPr)* on the outside of the 2490x300 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	$GrPr$ x 10^{-9}					
6	0.856	0.932	0.996	1.050	1.094	1.131
8	0.694	0.766	0.811	0.849	0.881	0.912
10	0.541	0.580	0.611	0.636	0.665	0.676

Table B.27 The Grashof-Prandtl numbers *(GrPr)* on the outside of the 3485x300 mm Weideman heat exchanger.

m_w [l/s]	0.3	0.4	0.5	0.6	0.7	0.8
T_{cwi} [°C]	$GrPr$ x 10^{-9}					
6	0.767	0.842	0.907	0.940	0.997	1.034
8	0.619	0.682	0.725	0.765	0.791	0.825
10	0.465	0.514	0.563	0.583	0.607	0.621

Appendix C

Calibration

CALIBRATION AND CHECK OF EQUIPMENT

The temperature resistance devices (RTD'S)

The RTD's together with the data logging system were calibrated using the following procedure prescribed by the developers of the system.

KALIBRASIE - PROSEDURE

1. Stel die rekenaar asook die WKTBS hardeware op waar getoets gaan word. Die temperatuur sensors (RTD's) moet nog nie gekoppel word nie.

2. Laai die program in TurboPascal en stel die TempOffset-matriks (lyn 25) na zero en die TempGain-matriks (lyn 41) na een. Laat loop nou die program.

3. Verwyder die hardeware kassie se deksel. Om sein kondisioneerder 1 te kalibreer, is dit nodig om 'n 100,2 ohm weerstand by poort 1 op die kassie in te prop en die "zero"-skroefie te verstel totdat 'n lesing van zero op die skerm vertoon word. Dit is egter die ideaal en prakties onbereikbaar aangesien die kondisioneerders so sensitief is. Vir ons toepassing is 'n toleransie van 0,1 akkuraat genoeg. Dan moet 'n 107,8 ohm weerstand in dieselfde poort geprop word en die "span"-skroefie verstel word totdat 'n lesing van 20 bereik word, ook met 'n toleransie van 0,1. Die "zero" en "span" beïnvloed mekaar egter en moet dus herhaaldelik om die beurt gestel word om so nader te kruip na die verlangde waardes. Dieselfde prosedure moet vir die ander drie kondisioneerders herhaal word, indien hulle gebruik gaan word.

4. Besluit tussen watter grense die RTD's gebruik gaan word.

5. Prop die RTD's in en plaas in water by of die boonste of die onderste grens soos besluit. Plaas ook die kontrole termometer se sensor in die water. Roer die water sodat 'n homogene temperatuur verkry word. Stip die tyd asook die kontrole temperatuur neer. Laat die rekenaar nou 'n paar (drie, maar hoe meer, hoe akkurater die gemiddeld) lesings neem. Stip weereens die tyd neer.

6. Hehaal die prosedure soos in stap 5 met water by die ander grens.

7. Doen nou 'n reglynige krommepassing vir elke sensor.
$(y = mx + c)$

8. Voer nou die konstantes m en c se waardes in die TempGain - en TempOffset-matriks onderskeidelik in.

9. Toets die sensor om te verseker dat die kalibrasie suksesvol was.

Die RTD's het 'n bereik van -250°C tot 1000°C maar hul uitset is nie reglynig nie. Die uitset kan egter as reglynig benader word tussen aanvaarbare grense. Hoe nader die grense egter aan mekaar, hoe akkurater sal die lesings gekry wees. Dit het tot gevolg dat indien 'n opstelling dit sou benodig, RTD's wat temperature van verskillende ordes gaan lees, apart gekalibreer kan word vir akkurate meting.

Tydens die eksperiment sal lesings outomaties elke 20 sekondes in die C:\WKTBS\DATA.DAT lêer gestoor word. Die lêer is van die ASCII formaat en word gewoonlik in LOTUS 123 ingelees vir verwerking.

Indien die lesings van so 'n aard is dat 'n temperatuur verskil, byvoorbeeld water in en water uit temperatuur by 'n hitteruiler, gemeet word, is dit noodsaaklik dat die sensors geruil word en die toets onder dieselfde omstandighede herhaal word. Die gemiddeld van die twee temperatuur veranderings word dan gebruik vir berekeninge.

The pressure transducer

Before the pressure tests were done, the pressure transducer was send to a agent. There it was calibrated and a graph of the calibration was given to us. A photocopy of the graph is shown on page C-6 of appendix C. An equation was formulated from the data and used in the software to interpret the signals read from the transducer. At the beginning of the pressure tests the mA readings in the pressure transducer were measured with a multimeter, the differential pressures were read from the computer screen and compared with the data on the graph.

The flow meter

Tests of mass collected per unit time were performed to check the accuracy of ± 2 % claimed by the supplies and to check the correct interpretation of hardware signals by the software.

A electronic scale (maximum 120 kg, tolerance 40 g), a stop watch and a 100 litre plastic bucket were used to perform the tests. The flow meter was installed in a water line and coupled to the data logging system. Due to constant pressure in the water line, the flow was constant. The bucket was filled with ± 60 kg of water and the time was recorded with stopwatch. The scale (which was zeroed with the bucket on it before the test) gave the mass of the water collected. The water temperature was also recorded to calculate the exact density.

Assumption: 1 kilogram = 1 litre of water

Table C.1 Mass flow test data

	Test 1	Test 2	Test 3
T_{water} [°C]	21.3	21.3	21.4
$Mass_{water}$ [kg]	61.41	61.13	60.84
Time [s]	200.8	116.3	76.8
Mass flow [kg/s]	0.310	0.532	0.787

Test 1

$$m_{water} = \frac{mass_{water}}{time}$$

$$= \frac{61.73 \cdot 10^{-3} \cdot 999.3}{200.8}$$

$$= 0.304 \text{ l/s}$$

$$accuracy = \frac{0.3040.31}{0.31} 100$$

$$= -1.9\ \%$$

Test 2

$$m_{water} = \frac{mass_{water}}{time}$$

$$= \frac{62.45 \cdot 10^{-3} \cdot 999.3}{116.3}$$

$$= 0.5365 \text{ l/s}$$

$$accuracy = \frac{0.5365 - 0.53}{0.53} 100$$

$$= 1.2 \%$$

Test 3

$$m_{water} = \frac{mass_{water}}{time}$$

$$= \frac{60.84 \cdot 10^{-3} \cdot 999.3}{76.8}$$

$$= 0.79 \text{ l/s}$$

$$accuracy = \frac{0.79 - 0.78}{0.78} 100$$

$$= 1.5 \%$$

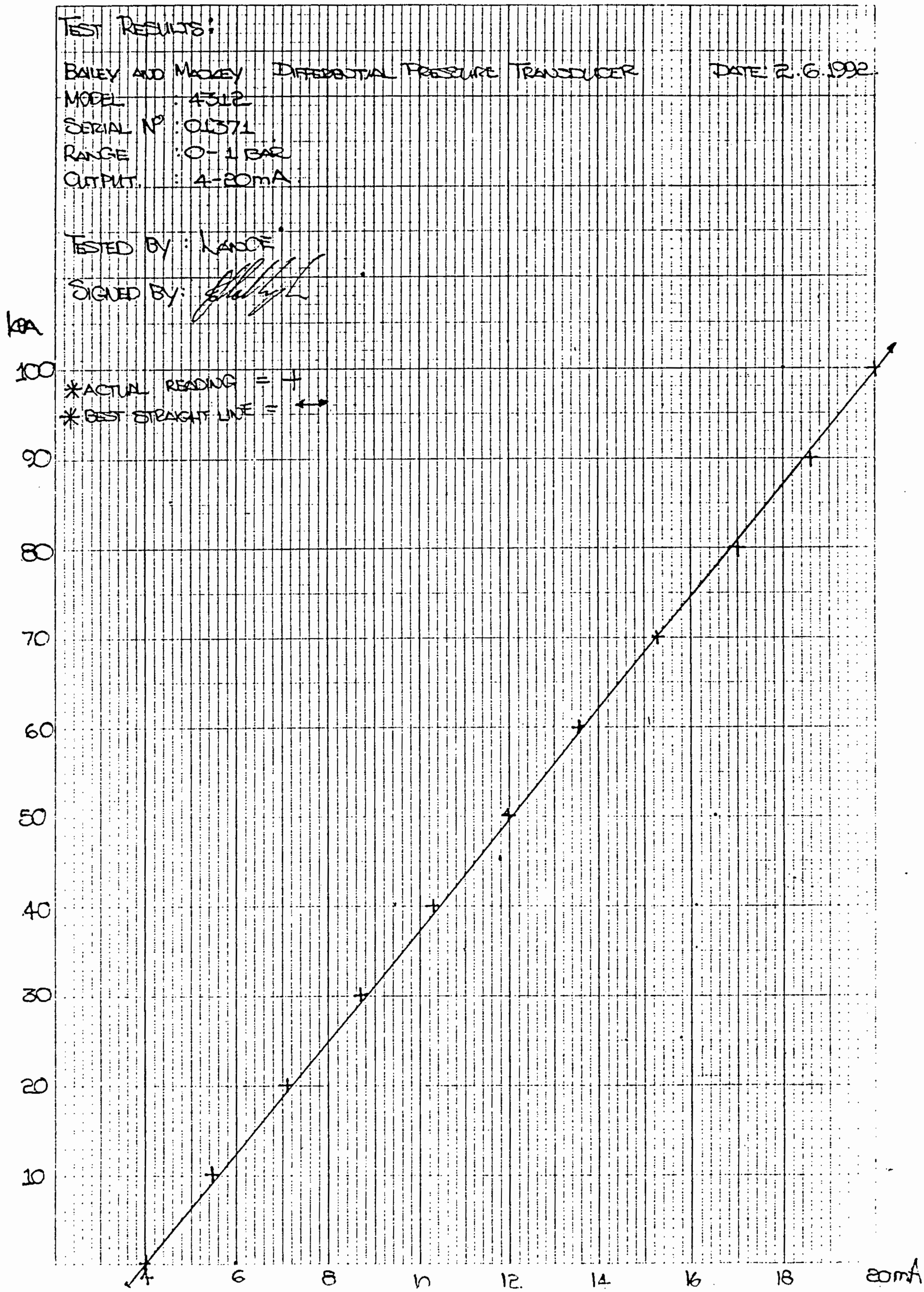


Figure C.1 Calibration curve of the pressure transducer done by Kudu instrumentation (PTY) LTD

Appendix D

Software

SOFTWARE ON THE WEIDEMAN HEAT EXCHANGER

```
Program WEIDEMAN
Uses Crt,ScrnCtrl,StdPar,FldProp,Utils,Printer;
Const
pi = 3.141592654;
m1 = ' Invallid Input ';
Var
Done                                               : Boolean;
Tw,Q3,Q2,Q1,hw,k,d1,d2,pl,pb,t,Tcwi,Tcwo,Theta     : real;
x,Re,fd,Qw,V,de,hs,Nu,e,g,F,Beta,Tf,Tj,Ra          : real;
mw,kan,Thold,StripWidth,ChannelWidth,Gr            : real;
Ak,Aki,Af,Afi,LMTD,Tfilm,ChannelPeri               : real;
Qf,Qfi,Q,delP,Two,Twi,ChannelPerii                 : real;
Q4,Fo,U,vp,pdiam,a1,a2,a3,a4,Gz                    : real;
Hold,hold3,Hold2,Rep,fdp,a5                        : real;
Rex,Pr,vj,nus,nuw,a6                               : real;
a,b                                                : double;
serie                                              : Integer;
c                                                  : array[0..20] of real;
{ Iteration Variables }
dt1,dt2,dt3 : Real;
{ control variables }
FileRead   : Boolean;
Normal     : Boolean;
foulFlag   : Boolean;
(***********************************************************************)
Function Power(Data:Real; Mag:Real):Real;
var
   r:Real;
   q:Real;
Begin
  r:= Frac(Data/2);
  q:= Frac(Mag/2);
  If Data <= -1 Then
  Begin
    IF (r <> 0) Then
    Begin
      If (Mag>0) Or (Mag<0) Then
      Begin
        If q <> 0 Then
        Power := -1 * exp(Mag*Ln(abs(data)))
        Else Power := exp(Mag*Ln(abs(data)));
      End
      Else
      If Mag = 0 Then
      Power := 1;
    End;
```

```
  End;
  If (Data<0) And (Data>-1) Then
  Begin
    If (Mag>0) Or (Mag<0) Then
    Begin
      If q < > 0 Then
      Power := -1 * exp(Mag*Ln(abs(data)))
      Else Power := exp(Mag*Ln(abs(data)));
    End
    Else
    If Mag = 0 Then
    Power := 1;
  End;
  If Data > 0 Then
  Power := exp(Mag*Ln(data));
  If Data=0 Then
  Begin
    If Mag = 0 Then Power:=1;
    If Mag < >0 Then Power:= 0;
  End;
End;
(*********************************************************************)
Procedure ReadDataFile;
Var
    Data : Text;
    FileName : String80;
Begin
  ClrScr;
  FileRead := False;
  DrawBoxD(2,1,79,25);
  PutStr(10,10,'Please Enter FileName');
  RestoreCurSor;
  TextColor(Black);
  TextBackground(White);
  PutStr(10,12,'            ');
  GotoXY(10,12);
  P := Readkey;
  Read_StringUC(10,12,12,FileName);
  TextColor(White);
  TextBackground(Blue);
  {$I-}
  Assign(Data,FileName);
  Reset(Data);
  If IOResult = 0 Then
  Begin
    Readln(Data,d1);        {channel width}
    ReadLn(Data,d2);        {channel hight}
    ReadLn(Data,pl);        {plate length}
```

```
    ReadLn(Data,pb);       {plaat width}
    ReadLn(Data,t);        {plaat dikte}
    ReadLn(Data,Tcwi);     {water in temp}
    Readln(Data,kan);      {number paralelle channels}
    ReadLn(Data,Theta);    {angle between plate and horisontal}
    ReadLn(Data,mw);       {water mass flow}
    ReadLn(Data,Tj);       {juice temperature}
    If IOResult < > 0 Then
    Begin
      DisplayError(5,5,'Cannot Read Datafile');
      Exit;
    End;
    Close(Data);
  End
  Else
  Begin
    Displayerror(0,0,'Invallid Filename Specified');
    Exit;
  End;
  {$I+}
  FileRead := True;
  Theta := Theta*pi/180;
End;
(**********************************************************************)
Procedure ShowStatus;
Begin
  PutStr(10,9,'Iteration Progress');
  PutStr(10,10,'Qw  :');
  PutStr(10,11,'Qs  :');
  PutStr(10,12,'Qf  :');
  PutStr(10,13,'Q1  :');
  PutStr(10,14,'Q2  :');
  PutStr(10,15,'Tcwo :');
  PutStr(10,16,'Tcwi :');
  PutStr(10,17,'hw  :');
  PutStr(10,18,'hs  :');
  PutStr(10,19,'Tf  :');
  GotoXY(20,10);
  WriteLn(Q4:8:2);
  GotoXY(20,11);
  WriteLn(Q3:8:2);
  GotoXY(20,12);
  WriteLn(Qf:8:2);
  GotoXY(20,13);
  WriteLn(Q1:8:2);
  GotoXY(20,14);
  WriteLn(Q2:8:2);
  GotoXY(20,15);
```

```
  WriteLn(Tcwo:8:2);
  GotoXY(20,16);
  WriteLn(Tcwi:8:2);
  GotoXY(20,17);
  WriteLn(hw:8:2);
  GotoXY(20,18);
  WriteLn(hs:8:2);
  GotoXY(20,19);
  WriteLn(Tf:8:2);
End;
(*********************************************************************)
Procedure JuiceConvection;
Begin
  Beta := 1.18253E-4;
  x := pb;
  Tfilm := (Tj + Two)/2;
  Gr  := 9.8*Beta*(Tj-Two)*Power(x,3)*Power(Wdensity(Tfilm+273.15),2)/
        Power(Wviscosity(Tfilm+273.15),2);
  Pr  := 3645*Wviscosity(Tfilm+273.15)/Wconductivity(Tfilm+273.15);
  x   := pb*(pl+pb*(sin(theta)/cos(theta)/cos(theta))/(pl+2*pb*sin(theta)/
         cos(theta)));
  If x >  pl then
  x := pl;
  vj := 0.67;
  Rex := x*vj*950/Wviscosity(Tfilm + 273.15);
  Nu := 0.453*sqrt(Rex)*Power(Pr,1/3);
  hs := Nu*Wconductivity(Tfilm+273.15)/x;
End;
(*********************************************************************)
Procedure WaterConvection;
Begin
  Tw := 0.5 * (Tcwi + Tcwo);
  a := 0.5 * (d1-t);
  b := 0.5 * (d2-t);
  de := 2 * a * b / sqrt(0.5 * (sqr(a) + sqr(b)));
  V := mw * 1E-3/(pi*(a)*(b));
  Pr:=WspesificHeat(Tw + 273.15)*Wviscosity(Tw+273.15)/
     Wconductivity(Tw + 273.15);
  Re := de*v*Wdensity(Tw + 273.15)/Wviscosity(Tw + 273.15);
  fd := Power((1.82*log(Re) - 1.64),-2);
  If Re < 20000 then
  Nu := 0.00165 * Power(Re,1.06) * Power(Pr,0.4)
  Else
  Nu := 0.0235*(Power(Re,0.8) - 230)*(1.8*Power(Pr,0.3) - 0.8)*
       (1 + power(de/ChannelWidth,0.67));
  hw := Nu * Wconductivity(Tw+273.15) / (de);
End;
(*********************************************************************)
```

```
Procedure ChannelCooling;
Begin
StripWidth := Pb - (d1*kan);
Af := StripWidth*pl*2;
Tf := 0.5*(Twi+Two);
JuiceConvection;
Qf := hs*Af*(Tj-(Tf + 0.2));
ShowStatus;
End;
(**********************************************************************)
Procedure ReporTStatus;
Begin
  ClrScr;
  DrawBoxD(2,1,79,25);
  PutStr(10,5,'Final Results After Complete Iteration');
  PutStr(10,6,'For Plate Cooling With Normal Conditions');
  PutStr(10,8,'Water Temperature Out (°C):');
  GotoXY(40,8);
  WriteLn(Tcwo:8:2);
  PutStr(10,9,'Total Cooling         (kW):');
  GotoXY(40,9);
  WriteLn((Q4)/1000:8:2);
  PutStr(10,10,'Channel Cooling        (W):');
  GotoXY(40,10);
  WriteLn((Qf):8:2);
  PutStr(10,11,'Fin Cooling            (%):');
  GotoXY(40,11);
  WriteLn(Qf*100/Q4:8:2);
  PutStr(10,12,'hs               (W/m^2°C):');
  GotoXY(40,12);
  WriteLn(hs:8:2);
  PutStr(10,13,'hw               (W/m^2°C):');
  GotoXY(40,13);
  WriteLn(hw:8:2);
  PutStr(10,14,'U                (W/m^2°C):');
  GotoXY(40,14);
  WriteLn(U:8:2);
  PutStr(10,15,'LMTD                  (°C):');
  GotoXY(40,15);
  WriteLn(lmtd:8:2);
  PutStr(10,16,'Q                     (kW):');
  GotoXY(40,16);
  WriteLn(Q/1000:8:2);
  PutStr(10,17,'q                 (kW/m^2):');
  GotoXY(40,17);
  WriteLn(U*LMTD/1000:8:3);
  PutStr(10,18,'delP                 (kPA):');
  GotoXY(40,18);
```

```
  WriteLn(delP/1000:8:3);
  PutStr(10,22,'Press Any Key To Continue');
  P := Readkey;
End;
(***********************************************************************)
Procedure InitScreen;
Begin
  TextColor(White);
  TextBackground(Blue);
  ClrScr;
  DrawboxD(2,1,79,25);
End;
(***********************************************************************)
Procedure Iteration;
Begin
  PutStr(10,4,'Iteration In Progress Please Wait');
  PutStr(10,6,'Press Any Key To Stop Iteration');
  Q4 := 0;
  Qf := 0;
  Q2 := 0;
  Q3 := 0;
  pdiam :=0.022;
  k   :=16.3;
  a   :=0.5 * d1;
  b   :=0.5 * d2;
  Tcwo :=Tcwi + 1;
  Two  :=Tj  - 2;
  Twi  :=0.5 * (Tcwi+Tcwo) + 0.1;
  Tw   :=0.5 * (Tcwi+Tcwo);
  ChannelCooling;
  ChannelPeri := 2*pi*Sqrt((sqr(a)+Sqr(b))/2);
  ChannelPerii := 2*pi*Sqrt((sqr(a-t)+Sqr(b-t))/2);
  dt3 := 1;
  Hold3 := 1;
  ChannelWidth := kan*(pl-2*(0.025+StripWidth/16+d1))+(kan-1)*pi/2*
               (StripWidth/16+d1) + 2*(StripWidth/16 + d1 + 0.025);
  Ak := ChannelPeri*ChannelWidth;
  Aki := ChannelPerii*ChannelWidth;
  Repeat
    If Hold3 > 0 Then
    Begin
      Tw := 0.5 * (Tcwo + Tcwi);
      Q4 := mw * WspesificHeat(Tw+273.15)*(Tcwo-Tcwi);
      dt3 := dt3/2;
      Tcwo := Tcwo - dt3;
    End
    Else Tcwo := Tcwo + dt3;
    Tw := 0.5 * (Tcwo + Tcwi);
```

```
 Q4 := mw * WspesificHeat(Tw+273.15)*(Tcwo-Tcwi);
 dt2 := 1;
 Two := Tj - 2;
 Repeat
   Tf := 0.0;
   ChannelCooling;
   JuiceConvection;
   Q3 := hs * Ak * (Tj-Two) + Qf;
   dt1 := 1;
   Twi := 0.5 * (Tcwi+Tcwo) + 0.1;
   WaterConvection;
   Tw := 0.5 * (Tcwo + Tcwi);
   Twi := (hs*Ak)/(hw*Aki)*(Tj-Two) + Tw;
   Q2 := 16.3/t*0.5*(Ak+Aki)*(Two - Twi) + Qf;
   ShowStatus;
   Hold2 := Q3 - Q2;
   If Hold2 > 0 Then
   Begin
     dt2 := dt2/2;
     Two := Two + dt2;
   End
   Else Two := Two - dt2;
 Until ABS(Hold2) < 15;
 If KeyPressed Then
 Begin
   P := Readkey;
   DisplayError(0,0,'Iteration Terminated By User');
   Exit;
 End;
 ShowStatus;
 Hold3 := Q4 - Q3;
 If KeyPressed Then
 Begin
   P := Readkey;
   DisplayError(0,0,'Iteration Terminated By User');
   Exit;
 End;
Until ABS(Hold3) < 15;
JuiceConvection;
U := 1/(1/hs + 1/hw + t/16.3 );
LMTD := (Tcwo - Tcwi)/ln((Tj - Tcwi)/(Tj - Tcwo));
Q := U*Ak*LMTD + Qf;
vp := mw/(1000*pi*sqr(pdiam)/4);
Rep := pdiam*vp*Wdensity(Tw + 273.15)/Wviscosity(Tw + 273.15);
fdp := Power((1.82*log(Rep) - 1.64),-2);
delP := Wdensity(Tw + 273.15)*((fD*ChannelWidth/De+11)*sqr(v)/2 +
      (fDp*0.24/pdiam + (2.52))*sqr(vp)/2);
ReportStatus;
```

```
End;
(***************************************************************************)
Procedure ReadData;
Begin
  ClrScr;
  DrawBoxD(2,1,79,25);
  RestoreCurSor;
  PutStr(10,4,'Please Enter The Following Data');
  PutStr(10,6,'Plate Length (pl)              :');
  ReadReal(42,6,2,3,pl,10,22,m1);
  PutStr(10,7,'Plate Width (pb)               :');
  ReadReal(42,7,2,3,pb,10,22,m1);
  PutStr(10,8,'Plate Thickness (t)            :');
  ReadReal(42,8,1,4,t,10,22,m1);
  PutStr(10,9,'Channel Width (d1)             :');
  ReadReal(42,9,2,4,d1,10,22,m1);
  PutStr(10,10,'Channel Height (d2)           :');
  ReadReal(42,10,2,4,d2,10,22,m1);
  PutStr(10,11,'Number Of Parrallel Channels  :');
  ReadReal(42,11,3,1,kan,10,22,m1);
  PutStr(10,12,'Angle Of Plate                :');
  ReadReal(42,12,3,2,Theta,10,22,m1);
End;
(***************************************************************************)
Procedure ReadDataIn;
Var DoneRead : Boolean;
    Key      : Char;
Begin
  Repeat
    ClrScr;
    HideCurSor;
    DoneRead := False;
    DrawBoxD(2,1,79,25);
    PutStr(10,3,' Calculation Of Heat Transfer For Plate Cooling ');
    PutStr(10,8,'Please Choose One Of The Following');
    PutStr(10,10,'(A) Read Data From Screen');
    PutStr(10,11,'(B) Read Data From Datafile');
    PutStr(10,13,'Press Esc To End');
    Repeat Key := Readkey;
    Until Upcase(key) In ['A','B',#27];
    Case Upcase(Key) Of
     'A' : Begin
             ReadData;
             DoneRead := True;
           End;
     'B' : Begin
             ReadDataFile;
           End;
```

```
    #27 : Begin
            ClrScr;
            RestoreCurSor;
            Halt;
          End;
    End;   { case }
  Until (DoneRead) Or (FileRead);
End;
(***********************************************************************)
Begin  { main program }
  hw := 0;
  hs := 0;
  Tf := 0;
  Repeat
    ReadDataFile;
    clrscr;
    InitScreen;
    Iteration;
    RestoreCurSor;
  Until Done;
End.
                        (*************************************)
```

SOFTWARE ON THE JACKET HEAT EXCHANGER

```
Program JACKET;
Uses Printer,ScrnCtrl,Dos,Crt,StdPar,utils,FldProp,Fermentation,Pressure;
Const pi = 3.141592654;
    m1 = 'Invalid Data Entered';
    Cpj = 3645;
    Rhoj= 1080;
    k = 16.3;
Var
Tw1                                        : Double;
dx1,dx2,dx3,dx4,dx5                        : Real;
Beta,Tfilm,Rex,f,Teff                      : Real;
Tcwo,Ta,fp,Rep,Lp,dp,vp,NTU,ef             : Real;
At,Ati,Aji,Ajo,k1,hs1,hs2,nu1,nu2          : Real;
Q111,Q211,Q311,gr1,gr2                     : Real;
Taw,T8,T7,h,gp,T5,Qt,Qj,Tj                 : Real;
g,x,z1,z2,Q,y,tt,yy,Af,Pf,havg,ty,yt       : Real;
kk,kw,hw,Twb,k2,de,v,vv,Re,fd,Pr,Lj,Nu     : Real;
A1,q3,q1,q2,Tw2,hs,e,t10,Gr,k4             : Real;
Fo,A,LMTD,U,Tj                             : Real;
d,Data                                     : Text;
Q4,ms,ha,q11,q21,q31                       : Real;
hold6,hold5,hold4,hold3,hold2,hold1,Hold   : Real;
temp,temp1,temp2,mw,Tcwi,delP              : Real;
{Iteration Variables}
   Positief : Boolean;
   OldSign  : Boolean;
   Positief1: Boolean;
   OldSign1 : Boolean;
   Positief2: Boolean;
   OldSign2 : Boolean;
   Positief3: Boolean;
   OldSign3 : Boolean;
   Positief4: Boolean;
   OldSign4 : Boolean;
{ control variables }
  FileRead  : Boolean;
  Normal    : Boolean;
  FowlFlag  : Boolean;
  Done      : Boolean;
(************************************************************************)
Procedure ReadDataScreen;
Begin
  RestoreCurSor;
  TextColor(White);
  TextBackground(Blue);
  ClrScr;
```

```
 DrawboxD(2,1,79,25);
 PutStr(15,2,'Please Supply The Following Information');
 PutStr(10,4,'Juice Temperature       : ');
 ReadReal(35,4,3,2,Tj,10,22,m1);
 PutStr(10,5,'Water Inlet Temperature: ');
 ReadReal(35,5,3,2,Tcwi,10,22,m1);
 PutStr(10,6,'Dry Bulb Temperature    : ');
 ReadReal(35,6,3,2,Taw,10,22,m1);
 PutStr(10,7,'Ambient Temperatuur     : ');
 ReadReal(35,7,3,2,Ta,10,22,m1);
 PutStr(10,9,'Mantel Width   (m)      : ');
 ReadReal(35,9,1,4,y,10,22,m1);
 PutStr(10,10,'Mantel Height (m)       : ');
 ReadReal(35,10,1,4,x,10,22,m1);
 PutStr(10,11,'Mantel Plate Thickness : ');
 ReadReal(35,11,1,4,z1,10,22,m1);
 PutStr(10,12,'Tank Plate Thickness   : ');
 ReadReal(35,12,1,4,z2,10,22,m1);
 PutStr(10,13,'Tank diameter          : ');
 ReadReal(35,13,1,4,Td,10,22,m1);
 PutStr(10,14,'Water supply pressure  : ');
 ReadReal(35,14,3,2,Pl,10,22,m1);
 If FowlFlag Then Begin
 PutStr(10,23,'Fowling Factor (Fo)           : ');
 ReadReal(39,23,1,7,Fo,10,22,m1);
End;
HideCursor;
ClrScr;
End;
(*************************************************************************)
Procedure ReadDataFile;
Var FileName : String80;
Begin
  ClrScr;
  FileRead := False;
  DrawBoxD(2,1,79,25);
  PutStr(10,10,'Please Enter FileName');   { vir voorbeeld TEST.DAT }
  RestoreCurSor;
  TextColor(Black);
  TextBackground(White);
  PutStr(10,12,'            ');
  GotoXY(10,12);
  P:=Readkey;
  Read_StringUC(10,12,30,FileName);
  TextColor(White);
  TextBackground(Blue);
  {$I-}
  Assign(D,FileName);
```

```
 Reset(D);
 {$I+}
 If IOResult = 0 Then Begin
 ReadLn(d,Tj);
 ReadLn(d,Tcwi);
 Readln(d,Ta);
 ReadLn(d,Taw);
 ReadLn(d,y);
 ReadLn(d,x);
 ReadLn(d,z1);
 Readln(d,z2);
 Readln(d,Td);
 ReadLn(d,Pl);
 If FowlFlag Then
 Readln(d,Fo);
 If IOResult <> 0 Then Begin
 Displayerror(0,0,'Unable To Read Data File');
 Exit;
End
Else FileRead := True;
Close(d);
End
Else Begin
  Displayerror(0,0,'Invallid Filename Specified');
End;
HideCurSor;
End;
(*********************************************************************)
Procedure GetOption;
Begin
  PutStr(10,11,'Normal Or Fermentation Condition [N/F] ?');
  Repeat
    P:=Readkey;
  Until Upcase(P) In ['F','N'];
  Case Upcase(P) Of
    'F' : Normal := False;
    'N' : Normal := True;
  End;   { case }
End;
(*********************************************************************)
Procedure GetFowl;
Begin
  ClrScr;
  HideCurSor;
  DrawboxD(2,1,79,25);
  PutStr(10,10,'Use Fowling Factor [Y/N] ?');
  Repeat
    P:=Readkey;
```

```
  Until Upcase(P) In ['Y','N'];
  Case Upcase(P) Of
    'Y' : FowlFlag := True;
    'N' : FowlFlag := False;
  End;  { case }
  If (Not FowlFlag) Then Fo := 0;
End;
(*********************************************************************)
Procedure ReadData;
Var DoneRead : Boolean;
    Key      : Char;
Begin
  Repeat
    ClrScr;
    HideCurSor;
    DoneRead := False;
    DrawBoxD(2,1,79,25);
    PutStr(10,5,'Calculation Of Cooling Capacity Of Type Mantel Cooling');
    PutStr(10,8,'Please Choose One Of The Following');
    PutStr(10,10,'(A) Read Data From Screen');
    PutStr(10,11,'(B) Read Data From Datafile');
    PutStr(10,13,'Press Esc To End');
    Repeat Key := Readkey;
    Until Upcase(key) In ['A','B',#27];
    Case Upcase(Key) Of
      'A' : Begin
      GetFowl;
      GetOption;
      DoneRead := True;
    End;
    'B' : Begin
      GetFowl;
      GetOption;
      ReadDataFile;
    End;
    #27 : Begin
      ClrScr;
      RestoreCurSor;
      Halt;
    End;
    End;  { case }
  Until (DoneRead) Or (FileRead);
End;
(*********************************************************************)
Procedure WaterKonVeksie;
var
   a,b,c,d,Nuwa,Nuwb   :real;
Begin
```

```
  a  := y-2*z1;
  b  := x-z1;
  de := 2*a*b/(a + b);
  V  := mw * 1E-03 /((x - z1) * ( y - z1*2) );
  Re := V * de * Wdensity(Twb+273)/Wviscosity(Twb+273);
  fd := Power((1.82 * log(Re) - 1.64),-2);
  c :=1/(Power(8/Re,10) + Power(Re/36500,20));
  d :=2.21*ln(Re/7);
  Pr:= WspesificHeat(Twb+273) * Wviscosity(Twb+273)/Wconductivity(Twb+273);
  Nu := 0.0243*power(Re,0.8)*power(Pr,0.4);
  hw := Nu * Wconductivity(Twb + 273.15)/de;
  End;
(***********************************************************************)
Procedure ALugKonveksie;
Var
   Pr,Gr,Nu:real;
Begin
  Pr := AirspecificHeat(Ta+273.15,Taw+273.15,100)*Airviscosity(Ta+273.15,
  Taw+273.15,100)/Airconductivity(Ta+273.15,Taw+273.15,100);
  temp1 := Airdensity(Ta+273.15,Taw+273.15,100);
  temp2 := Airviscosity(Ta+273.15,Taw+273.15,100);
  Gr := 9.81*(2{Ta-T2})*Power(0.8{y},3)*Sqr(temp1)/
       ((0.5*(Ta+T2)+273.15)*Sqr(temp2));
  Nu := 0.59 * Power((Gr*Pr),1/4);
  ha := Nu * Airconductivity(Ta+273.15,Taw+273.15,100)/0.8;
End;
(***********************************************************************)
Procedure BetaP;
var
  pe,pb,pc,pd :double;
Begin
   pe :=-3.303348739683579E-5;
   pb := 1.793222710602901E-5;
   pc := -2.628842077822704E-7;
   pd := -1.750364690130372E-9;
   Tfilm :=12.75;
  Beta := (pe + pb*(Tfilm) + pc*sqr(Tfilm) + pd*Power(Tfilm,3))
End;
(***********************************************************************)
Procedure SapKonveksie;
var
   Sigma,hr,h:real;
Begin
  Tfilm :=0.5*(Tj + Tw2);
  Pr := Cpj*Wviscosity(Tj+273)/Wconductivity(Tj+273.15);
  BetaP;
  Gr1 := (9.8*Beta)*(Tj-Tw2)*(Power(0.74,3))*Power(Rhoj,2)/
  Sqr(Wviscosity(Tj+273.15));
```

```
  Nu1 := 0.59 * Power((Gr1*Pr),0.25);
  hs1 := Nu1 * Wconductivity(Tj+273.15)/0.74;
  Gr2 := (9.8*Beta)*(Tj-Tw2)*(Power(y,3))*Power(Rhoj,2)/
  Sqr(Wviscosity(Tj+273.15));
  Nu2 := 0.59 * Power((Gr2*Pr),0.25);
  hs2 := Nu2 * Wconductivity(Tj+273.15)/y;
  hs := 0.763*hs1 + 0.237*hs2;
End;
(*********************************************************************)
Procedure FinalResults;
Begin
  TextColor(White);
  TextBackground(Blue);
  ClrScr;
  DrawboxD(2,1,79,25);
  PutStr(10,4,'Final Results After Iteration Of Mantel Cooling');
  If Normal Then PutStr(10,5,'Under Normal Conditions')
  Else
  PutStr(10,5,'Under Fermentation Conditions');
  If FowlFlag Then
  PutStr(10,6,'With Fowling Factor Used')
  Else
  PutStr(10,6,'Without Fowling Factor');
  GotoXY(10,8);
  WriteLn('Jacket lenght (m)                : ',Lj:8:2);
  GotoXY(10,10);
  WriteLn('Cooling needed (kW)              : ',Qj:4:2);
  GotoXY(10,12);
  WriteLn('Total cooling (kW)               : ',Q4:4:2);
  GotoXY(10,14);
  WriteLn('% heat loss/gain from ambient    : ',(q21)*100/Q4:8:2);
  GotoXY(10,16);
  WriteLn('Effectiveness, %                 : ',ef:8:2);
  GotoXY(10,18);
  WriteLn('Pressure loss (kPa)              : ',delP:8:2);
  PutStr(10,18,'Press Any Key To Continue');
  P:=Readkey;
End;
(*********************************************************************)
Procedure Pressureloss;
Begin
  dp :=0.022;
  Lp :=0.71;
  k1 :=1;
  k2 :=12.7;
  Vp  := mw * 1E-03*4 /(pi*sqr(dp));
  Rep := Vp * dp * Wdensity(Twb+273)/Wviscosity(Twb+273);
  fp := Power((1.82 * log(Rep) - 1.64),-2);
```

```
  delP :=Wdensity(Twb + 273.15)*((f*Lj/de + k1)*sqr(V)/2 +
  (fp*Lp/dp + k2)*sqr(Vp)/2)/1000;
End;
(*********************************************************************)
Procedure Iteration;
Const
    k    = 16.3;
Begin
  PutStr(10,4,'Iteration In Progress Please Wait');
  PutStr(10,6,'Press Any Key To Stop Iteration');
  Tcwo  := Tcwi + 1;
  Lj := Td*3*1.12;
  Ati  := Lj*y*1.1;
  Aji  := Lj*((y - 2*z1) + 2*(x - z1));
  Ajo  := Lj*(y + 2*x);
  Af   := Lj*z1;
  Pf   := 2*(Lj + z1);
  {heat transfered to water}
  dx5 := 1;
  ty :=0;
  yt :=0;
  Repeat
    Twb := 0.5*(Tcwi + Tcwo);
    mw := mw/(Wdensity(Twb+273.15)*1E-03);
    Q4 := mw * WspesificHeat(Twb+273.15) * (Tcwo - Tcwi);
    {heat transfered from surroundings}
    dx2 := 0.2;
    Tw1 :=Ta - 2;
    tt :=0;
    yy :=0;
    Repeat
      WaterKonveksie;
      Q31 := hw*Aji*(Tw1-Twb);
      ALugKonveksie;
      Q21 :=ha * Ajo*(Ta-Tw1);
      If KeyPressed Then
      Begin
        P:=ReadKey;
        DisPlayError(0,0,'Iteration Terminated By User');
        Exit;
      End;
      Hold1 := Q31 - Q21;
      If Hold1 < 0 Then
      Begin
        Tw1 := Tw1 + dx2;
        tt := tt + 1;
      End
      Else
```

```
  Begin
    Tw1 := Tw1 - dx2;
    yy := yy + 1;
  End;
  If (tt > 0) AND (yy > 0) Then
  Begin
    dx2 := dx2/2;
    tt := 0;
    yy := 0;
  End;
Until ABS(Hold1) < 70;
{heat transfered from juice}
dx4 := 1;
Tw2   := Tj - 1;
Repeat
  SapKonveksie;
  Q3  := hs * Ati * (Tj - Tw2) + Q21 ;
  WaterKonveksie;
  Q2 := hw*Ati*(Tw1-Twb) + hw*Aji*(Tw2-Twb);
  Hold2 := Q3 - Q2;
  Hold3 := Q3 - Q2;
  If Hold3 > 0 Then
  Begin
    Tw2 := Tw2 + dx4;
    tt := tt + 1;
  End
  Else
  Begin
    Tw2 := Tw2 - dx4;
    yy := yy + 1;
  End;
  If (tt > 0) AND (yy > 0) Then
  Begin
    dx4 := dx4/2;
    tt := 0;
    yy := 0;
  End;
Until ABS(Hold3) < 80;
hold4 :=Q4 - Q3;
A :=Lj*(y + 2*(x - z1));
If Hold4 < 0 Then
Begin
  Tcwo := Tcwo + dx5;
  ty := ty + 1;
End
Else
Begin
  Tcwo := Tcwo - dx5;
```

```
    yt := yt + 1;
   End;
   If (ty > 0) AND (yt > 0) Then
   Begin
    dx5 := dx5/2;
    ty := 0;
    yt := 0;
   End;
  Until ABS(Hold4) < 80;
  ef :=(Q3-Q21)/(mw * WspesificHeat(Twb+273.15) * (Tj - Tcwi));
  Teff :=1 + 0.0929*(8 - Tcwi) + 0.0967*(Tj - 20);
End;
(**********************************************************************)
Begin
  Done := False;
  Repeat
   Normal := True;
   FowlFlag := false;
   textcolor(White);
   TextBackground(Blue);
   hidecursor;
   ReadData;
   Fermentation;
   Pressure;
   ClrScr;
   Iteration;
   Pressureloss;
   RestoreCursor;
   FinalResults;
  Until Done;
End.
```

SOFTWARE ON THE WATER-FILM

```
Program WATER-FILM;
{Film Cooling Under Normal and fermentation Conditions}
Uses Crt,Printer,FLDPROP,SK,ScrnCtrl,StdPar,Utils,SaveScrn;
Const Pi = 3.141592654;
     m1 = 'Invallid Numerical Input';
     Rw = 461.5;
     k  = 16.3;
     Cpj = 3645;
     Rhoj = 1080;
     D  = 0.256E-04;
Var
   { program control variables }
   Hold,Hold1,Hold2,Hold3 : Double;
   dx1,dx2,dx3,dx4,dx5    : Double;
   tt,yy,ty,yt            : Double;
   Done                   : Boolean;
   Normal                 : Boolean;
   FileRead               : Boolean;
   FowlFlag               : Boolean;
   { program calculation variables }
   Two,Tw,Twoi,Twii,Shw  : Real;
   Qrx,Qhx,Qmx,Twj,Ai    : Real;
   Sh,Ht,ro,dout,Twb,At  : Real;
   Twall,Tfilm,Qi,Qmax   : Real;
   iv,y,v,v1,delta,GrPr  : Real;
   Q4,Q3,Q2,Q1,Qr,Qh,Qm  : Double;
   Qr1,Qr2,hw,hs,Rhos,df : Double;
   Re,Pr,hf,Gr,ha        : Double;
   Nu,x,Fi,Beta          : Double;
   Ato,Ati,Af,ann,ma,Tmax: Real;
   Vmax,Qm1,Qm2,Qh1,Qh2  : Double;
   Sc,hd1,hd2,rf         : Double;
   Fo,Ta,z,ef,NTU,U,hm,hr: Real;
   Tmax1,Tsize,Td,Rd,Hg  : Real;
   i                     : Integer;
   Tcwi,Tj,mw            : Real;
   {iteration variables}
   positief:boolean;
   oldsign:boolean;
   positief1:boolean;
   oldsign1:boolean;
   positief2:boolean;
   oldsign2:boolean;
(****************************************************************************)
Procedure ReadDataFile;
Var A : Text;
```

```
    FileName : String80;
  Begin
   ClrScr;
   FileRead := False;
   DrawBoxD(2,1,79,25);
   PutStr(10,5,'Read Data From Datafile');
   PutStr(10,10,'Please Enter FileName');
   RestoreCurSor;
   TextColor(Black);
   TextBackground(White);
   PutStr(10,12,'            ');
   GotoXY(10,12);
   P:=Readkey;
   Read_StringUC(10,12,12,FileName);
   TextColor(White);
   TextBackground(Blue);
   {$I-}
   Assign(A,FileName);
   Reset(A);
   If IOResult = 0 Then
   Begin
     ReadLn(A,Tsize);     { Tank size }
     ReadLn(A,B);         { max ferm rate }
     ReadLn(A,dout);      { Tank outside diameter }
     ReadLn(A,Ht);        { Tank height }
     ReadLn(A,Hg);        { Gutter height }
     ReadLn(A,Sh);        { Juice height }
     ReadLn(A,Td);        { Tank diameter }
     ReadLn(A,Rd);        { Ring diameter }
     ReadLn(A,Ta);        { Ambient temperature }
     Readln(A,Twb);       { Wetbulb temperature }
     ReadLn(A,z);         { Plate thickness }
     ReadLn(A,mw);        { Water flow rate }
     ReadLn(A,Tcwi);      { Water inlet temperature }
     ReadLn(A,Tj);        { Juice temperature }
     If FowlFlag Then Readln(A,Fo);
     If IOResult <> 0 Then
     Begin
       Displayerror(5,5,'*** Warning *** Cannot Read Datafile ');
       Exit;
     End;
     Close(A);
   End
   Else
   Begin
     DisplayError(10,10,'Invallid Filename Specified');
     Exit;
   End;
```

```
  {$I+}
  FileRead := True;
End;
(**********************************************************************)
Procedure Vloeisnelheid;
Var HoldV : Real;
Begin
  delta :=Power((3*mw*Wviscosity(Tw+273.15))/(pi*dout*9.81*
  1000*(Wdensity(Tw+273.15) - 1.18)),1/3);
  V :=9.81*(Wdensity(Tw+273.15))/(3*Wviscosity(Tw+273.15))*sqr(delta);
  Vmax :=9.81*(Wdensity(Tw+273.15))/(Wviscosity(Tw+273.15))*(sqr(delta)/2);
End;
(**********************************************************************)
Procedure WarmteOordrag;
var
   Tfilm : double;
Begin
  Tfilm :=0.5*(Ta + Tw);
  Re:=Vmax*x*Airdensity(Tfilm+273.15,Twb+273.15,101.325)/Airviscosity(Tfilm
      + 273.15,Twb + 273.15,101.325);
  Sc := Airviscosity(Tfilm + 273.15,Twb + 273.15,101.325)/D/Airdensity(Tfilm +
       273.15,Twb + 273.15,101.325);
  hd2 := 0.332 * Power(Re,-0.5)*Power(Sc,-2/3)*Vmax;
  Pr := Airspecificheat(Tfilm+273.15,Twb+273.15,101.325)*Airviscosity
       (Tfilm+273.15,Twb+273.15,101.325)/Airconductivity(Tfilm+273.15,
       Twb + 273.15,101.325);
  hf := hd2 * Airdensity(Tfilm+273.15,Twb + 273.15,101.325)*
       Airspecificheat(Tfilm+273.15,Twb+273.15,101.325)*Power(Sc/Pr,2/3);
  Qh := hf *  Ato * (Ta - Tw);
End;
(**********************************************************************)
Procedure Entalphy;
var
   Tdewp,ke,kb,kc,kd :double;
Begin
   ke := 2501700;
   kb := -2446.66667;
   kc := 6;
   kd := -0.133333333;
   Tdewp:=15.2;
  iv := (ke + kb*(15.2) + kc*sqr(15.2) + kd*Power(15.2,3));
End;
(**********************************************************************)
Procedure MassaOordrag;
Begin
  Entalphy;
  DrybulbWetbulb(Ta,Twb);
  Qm :=iv*hd2*Ato*(Vaporpressure(Tw+273.15)*1000/(Tw+273.15)-Pv/
```

```
     (Twb+273.15))/(Rw);
  hm :=Qm/(Ato*(Tw-Twb));
End;
(**********************************************************************)
Procedure Radiasie;
Begin
  Qr := 0.95*5.669E-08*Ato*(Power(Ta+273.15,4)-Power(Tw+273.15,4));
  hr :=Qr/(Ato*(Ta-Tw));
End;
(**********************************************************************)
Procedure WaterKonveksie;
Begin
  Re :=V * Sh * Wdensity(Tw + 273.15) / Wviscosity(Tw + 273.15);
  Pr := Wspesificheat(Tw+273.15)*Wviscosity(Tw+273.15)/
       Wconductivity(Tw+273.15);
  Nu := 0.453*Power(Re,0.5)*Power(Pr,1/3);
  hw := Nu * Wconductivity(Tw + 273.15) / Sh;
End;
(**********************************************************************)
Procedure BetaP;
var
   pe,pb,pc,pd :double;
Begin
  pe :=-3.303348739683579E-5;
  pb := 1.793222710602901E-5;
  pc := -2.628842077822704E-7;
  pd := -1.750364690130372E-9;
  Tfilm :=12.75;
  Beta := (pe + pb*(Tfilm) + pc*sqr(Tfilm) + pd*Power(Tfilm,3));
End;
(**********************************************************************)
Procedure SapKonveksie;
Begin
  Tfilm :=0.5*(Tj + Twall);
  Pr := Cpj*Wviscosity(Tj+273)/Wconductivity(Tj+273.15);
  BetaP;
  Gr := (9.8*Beta)*(Tj-Twall)*(Power(Shw,3))*Power(Rhoj,2)/
       Sqr(Wviscosity(Tj+273.15));
  GrPr :=Gr*Pr;
  Nu := 0.1 * Power((Gr*Pr),1/3);
  hs := Nu * Wconductivity(Tj+273.15)/Shw;
End;
(**********************************************************************)
Procedure ALugKonveksie;
Var
   Pra,Gra,Nua        : Real;
   temp1,temp2,temp3 : Real;
Begin
```

```
  Pra := AirspecificHeat(Ta+273.15,Twb+273.15,100)*Airviscosity
        (Ta+273.15,Twb+273.15,100)/Airconductivity
        (Ta+273.15,Twb+273.15,100);
  temp1 := Airdensity(Ta+273.15,Twb+273.15,100);
  temp2 := Airviscosity(Ta+273.15,Twb+273.15,100);
  Gra:=9.81*(Ta-Tcwi+2)*Power(2.8,3)*Sqr(temp1)/((0.5*(Ta+Tcwi+2)+273.15)
        *Sqr(temp2));
  Nua := 0.59 * Power((Gra*Pra),1/4)*5;
  ha := Nua * {Airconductivity(Ta+273.15,Twb+273.15,100)}0.0258/2.8;
End;
(**********************************************************************)
Procedure InitialHeatTransfer;
var
   Tw1,dummy,Q5 :double;
Begin
  Twoi := Tcwi;
  dummy := Tcwi;
  Tw :=0.5*(Tcwi+Twoi);
  Repeat
    dummy := Twoi;
    Tw := 0.5*(Twoi + Twii);
    Ato  := (dout + 2*delta) * pi * (0.1135*Ht) + pi * ((dout/2) +
    (Rd/2)) * (Ht - Hs);
    Ai :=Ato;
    Warmteoordrag;
    Radiasie;
    Massaoordrag;
    Twoi :=(Qr + Qh - Qm)/(mw*Wspesificheat(Twoi + 273.15)) + Tcwi;
  Until abs(Twoi - dummy) < 0.005;
  Qi :=mw*Wspesificheat(Tw + 273.15)*(Twoi - Twii);
  ann := -7;
End;
(**********************************************************************)
Procedure Iteration;
Begin
  Twii := Tcwi;
  Two := Tcwi + 0.1;
  Twall := Tj - 0.01;
  dx4 := 0.5;
  Shw :=Sh - 0.707;
  positief2:=false;
  oldsign2:=false;
  Vloeisnelheid;
  InitialHeatTransfer;
  Ato := dout * pi * Shw;
  At :=Ato + Ai;
  Ati := (dout-z) * pi * Shw;
  Tcwi := Twoi;
```

```
Tw := 0.5 * (Tcwi + Two);
ty :=0;
yt :=0;
Repeat
  Tw := 0.5 * (Tcwi + Two);
  Twall := Tw + 0.003;
  Q4 := mw * Wspesificheat(Tw+273.15) * (Two - Twii);
  If hold2 >= 0 Then
  Begin
    Two := Two - dx4;
    ty := ty + 1;
  End
  Else
  Begin
    Two := Two + dx4;
    yt := yt + 1;
  End;
  If (ty > 0) AND (yt > 0) Then
  Begin
    dx4 := dx4/2;
    ty := 0;
    yt := 0;
  End;
  dx2 := 1.0;
  positief1:=false;
  oldsign1:=false;
  tt :=0;
  yy :=0;
  Repeat
    Radiasie;
    Warmteoordrag;
    Massaoordrag;
    WaterKonveksie;
    Q3 := hw*Ato*(Twall - Tw) + Qr + Qh - Qm + Qi;
    SapKonveksie;
    Q2 := hs * Ati * (Tj - Twall) + Qr + Qh - Qm + Qi;
    Hold := Q2 - Q3;
    If (Hold) >= 0 Then
    Begin
      Twall := Twall + dx2;
      tt := tt + 1;
    End
    Else
    Begin
      Twall := Twall - dx2;
      yy := yy + 1;
    End;
    If (tt > 0) AND (yy > 0) Then
```

```
    Begin
      dx2 := dx2/2;
      tt := 0;
      yy := 0;
    End;
   Until Abs(hold) < 150;
   hold2 := Q4 - Q3;
  Until Abs(hold2) < 150;
  Twj :=0.5*(Twii+Tj);
  Qmax := mw * Wspesificheat(Tw+273.15) * (Tj - Twii);
  Qhx := hf *  At * (Ta - Twj);
  Qmx :=iv*hd2*At*(Vaporpressure(Twj+273.15)*1000/(Twj+273.15)-
        Pv/(Twb+273.15))/(Rw);
  Qrx := 0.95*5.669E-08*At*(Power(Ta+273.15,4)-Power(Twj+273.15,4));
  ef :=(hs * Ati * (Tj - Twall))/Qmax;
End;
(*********************************************************************)
Procedure InitScreen;
Begin
  TextColor(White);
  TextBackground(Blue);
  ClrScr;
  DrawboxD(2,1,79,25);
End;
(*********************************************************************)
Procedure GetFowling;
Begin
  ClrScr;
  DrawBoxD(2,1,79,25);
  PutStr(10,5,'Please Enter The Following Fowling Factors');
  PutStr(10,10,'Please Enter Fowling Factor OutSide    :');
  ReadReal(50,10,1,4,Fo,0,0,'Invallid Fowling Factor');
  PutStr(10,10,'Please Enter Fowling Factor OutSide    :');
End;
(*********************************************************************)
Procedure GetOption;
Begin
  PutStr(10,11,'Normal Or Fermentation Condition [N/F] ?');
  Repeat
    P:=Readkey;
  Until Upcase(P) In ['F','N'];
  Case Upcase(P) Of
    'F' : Normal := False;
    'N' : Normal := True;
  End;  { case }
End;
(*********************************************************************)
Procedure GetFowl;
```

```
Begin
  ClrScr;
  HideCurSor;
  DrawboxD(2,1,79,25);
  PutStr(10,10,'Use Fowling Factor [Y/N] ?');
  Repeat
    P:=Readkey;
  Until Upcase(P) In ['Y','N'];
  Case Upcase(P) Of
    'Y' : FowlFlag := True;
    'N' : FowlFlag := False;
  End;  { case }
  If (Not FowlFlag) Then Fo := 0;
End;
(*********************************************************************)
Procedure ReadDataIn;
Var DoneRead : Boolean;
    Key      : Char;
Begin
  Repeat
    ClrScr;
    HideCurSor;
    DoneRead := False;
    DrawBoxD(2,1,79,25);
    PutStr(15,3,' Calculation Of Heat Transfer For Film Cooling ');
    PutStr(10,8,'Please Choose One Of The Following');
    PutStr(10,10,'(A) Read Data From Screen');
    PutStr(10,11,'(B) Read Data From Datafile');
    PutStr(10,13,'Press Esc To End');
    Repeat Key := Readkey;
    Until Upcase(key) In ['A','B',#27];
    Case Upcase(Key) Of
      'A' : Begin
        GetFowl;
        GetOption;
        DoneRead := True;
      End;
      'B' : Begin
        GetFowl;
        GetOption;
        ReadDataFile;
      End;
      #27 : Begin
        ClrScr;
        RestoreCurSor;
        Halt;
      End;
    End;  { case }
```

```
  Until (DoneRead) Or (FileRead);
End;
(**********************************************************************)
Begin  { main program }
  Done := False;
  Normal := True;
  FowlFlag := False;
  InitScreen;
  Repeat
  ReadDataFile;
    DrybulbWetbulb(Ta,Twb);
    HideCurSor;
    InitScreen;
    clrscr;
    Iteration;
    RestoreCurSor;
  Until Done;
End.
                         (***************************)
```