ANALYSIS OF EVAPORATIVE COOLERS AND CONDENSERS

by

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Declaration

I, the undersigned, hereby declare that the work contained in this thesis is my own original work and has not previously, in its entirety or in part, been submitted at any university for a degree.

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ABSTRACT

In this report various mathematical models for the thermal evaluation of evaporative coolers and condensers are presented. These models range from the exact model based on the work by Poppe [84PO1] to the simplified logarithmic models based on the work of McAdams [54Mc1] and Mizushina et al. [67MI1], [68MI1].

Various computer programs were written to perform rating and selection calculations on cross-flow and counterflow evaporative coolers and condensers.

Experimental tests were conducted on a cross-flow evaporative cooler to determine the governing heat and mass transfer coefficients. The experimentally determined coefficients were correlated and these correlations are compared to the existing correlations. The two-phase pressure drop across the tube bundle was also measured and a correlation for two-phase pressure drop across a tube bundle is presented.

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NOMENCLATURE

A	Area,	[m^2]
a	Effective suface area of tubes per unit volume,	[m^2/m^3]
B	Constant defined in section 4.3,	[-]
b	Slope of the air saturation enthalpy curve,	[J/kgK]
C	Concentration,	[kg/m^3]
$c_1, c_2, ..$	Coefficients,	[-]
c_p	Specific heat at constant pressure,	[J/kgK]
D	Diffusivity,	[m^2/s]
d	Diameter (characteristic length),	[m]
E	Coefficient defined by equation 2.5,	[-]
EP	Dimensionless enthalpy potential defined by equation 4.2.21,	[-]
F	LMTD correction factor or force,	[-] or [N]
f	Friction factor,	[-]
f_a	Arrangement factor (Appendix D),	[-]
f_n	Correction factor for small number of tube rows (Appendix D),	[-]
f_z	Correction factor for non-isothermal flow (Appendix D),	[-]
f_{zn}	Correction factor for non-isothermal flow and small number of tube rows (Appendix D),	[-]
G	Mass velocity,	[kg/m^2s]
g	Gravitational acceleration,	[m/s^2]
h	Heat transfer coefficient,	[W/m^2K]
h_D	Mass transfer coefficient based on T_w,	[kg/m^2s]
h_{Di}	Mass transfer coefficient based on T_i,	[kg/m^2s]
h_{Do}	Overall mass transfer coefficient,	[kg/m^2s]
h_{Dp}	Mass transfer coefficient based on partial pressure driving potential and T_w,	[s/m]
h_{Dpi}	Mass transfer coefficient based on partial pressure driving potential and T_i,	[s/m]
i	Enthalpy,	[J/kg]
i_{fg}	Latent heat of evaporation,	[J/kg]
i'_{fg}	Corrected latent heat of evaporation,	[J/kg]
i_{vo}	Latent heat of evaporation at 0°C,	[J/kg]
Δi	"Entalpy potential" defined by equation 4.2.21,	[J/kg]
K_{wb}	"Wet bulb K" defined by equation 2.1,	[-]
K	Loss coefficient,	[-]
K_g	Coefficient defined by equation C.4,	[s/m]
k	Thermal conductivity,	[W/mK]
k_g	Coefficient defined by equation C.5,	[s/m]
k_l	Coefficient defined by equation C.6,	[m/s]
L	Length,	[m]
LMED	Log mean enthalpy difference,	[J/kg]
LMTD	Log mean temperature difference,	[C]
LVF	Liquid void fraction,	[-]
m	Massflow rate,	[kg/s]
N	Constant defined by equation 2.6,	[-]
n	Number	[-]
NTU	Number of transfer units,	[-]
P	Pitch,	[m]
p	Pressure,	[N/m^2]
Δp	Pressure drop,	[N/m^2]

Δp^*	Pressure drop based on massflow rate of both phases and the properties of one of the phases,	[N/m^2]
q	Heat transfer rate,	[W]
q"	Heat flux,	[W/m^2]
R	Universal gas constant,	[J/kgK]
Re^*	Reynolds number based on the massflow rate of both phases and the properties of one of the phases,	[-]
R_R	Ratio defined by equation 4.2.19,	[-]
Ry	Characteristic flow parameter,	[m^{-1}]
r	Ratio defined by equation B.21,	[-]
T	Temperature,	[°C]
t	Thickness,	[m]
U	Overall heat transfer coefficient,	[W/m^{2}K]
v	Velocity,	[m/s]
W	Width,	[m]
w	Humidity ratio,	[kg water/kg dry air]
X	Mole fraction,	[-]
x	Vapour quality,	[-]
y	Ratio defined by equation 4.3.10,	[-]
z	Height,thickness,	[m]
α	Thermal diffusivity, $k/\rho c_p$,	[m^2/s]
Γ	Recirculating water massflow rate per side per unit length of tube,	[kg/ms]
Γ_G	Ratio defined by equation 4.3.14,	[-]
δ	Film thickness,	[m]
δ_c	Concentration boundary layer thickness,	[m]
δ_m	Momentum boundary layer thickness,	[m]
δ_t	Thermal boundary layer thickness,	[m]
ε	Heat exchanger effectiveness	[-]
ζ	Parameter defined by equations 4.3.2 and 4.3.12,	[-]
η	Parameter defined by equation 6.17,	[-]
θ	Angle,	[ ]
μ	Dynamic viscosity,	[kg/ms]
ν	Kinematic viscosity,	[m^2/s]
ρ	Density,	[kg/m^3]
ϕ	Parameter defined by equations 4.3.18 and 4.3.22,	[-]
ψ	Parameter defined by equation 4.3.6,	[-]
X_{tt}	Martinelli parameter defined by equation 4.3.21,	[-]

Dimensionless Groups

Le	Lewis number, α/D, Sc/Pr
Nu	Nusselt number, hd/k
Pr	Prandtl number, $c_p\mu/k$
Re	Reynolds number, $\rho vd/\mu$
Sc	Schmidt number, ν/D

Abbreviations

BTF	Back-To-Front
BTT	Bottom-To-Top
FTB	Front-To-Back
TTB	Top-To-Bottom

Subscripts

a	Air
atm	Atmospheric
as	Saturated air
asi	Air saturated at air/water interface temperature
asp	Air saturated at process fluid temperature
asw	Air saturated at bulk recirculating water temperature
c	Convective or convection or condensate
ct	Cooling tower
crit	Critical
d	Diagonal or downstream
db	Dry bulb
de	Drift eliminator
e	Equivalant or effective
eb	Equivalent (tube-) bundle
ec	Equivalent constriction
ff	Film cooler
fr	Frontal
g	Gas
go	Gas only
he	Heat exchanger
hor	Horizontal
i	Inlet or inside or interface
il	In-line
l	Longitudinal or liquid
lo	Liquid only
lsl	Laminar sublayer
m	Mean or moist
max	Maximum
min	Minimum
o	Outlet or outside
obl	Oblique
p	Process fluid (water)
r	Refrigerant
rows	Rows
rest	Restrictions
st	Staggered
t	Tube or transverse
tp	Two phase
theo	Theoretical
v	Vapour
ver	Vertical
w	Recirculating (spray) water
wb	Wet bulb
∞	Free stream

CHAPTER 1

INTRODUCTION

The phenomenon of cooling by evaporation is well-known and it has found many applications. The ancient Egyptians used porous clay containers to keep water cool thousands of year ago.

Today evaporative cooling is used extensively in industry, ranging from the cooling of power generating plants to the cooling of condensers in air-conditioning systems.

In evaporative cooling, the medium which is being cooled can theoretically reach the air wet bulb temperature whereas the minimum temperature which can be reached in dry cooling would be the air dry bulb temperature. The use of evaporative cooling can lead to major cost savings and improvements in thermal efficiency because of the lower temperatures which can be reached.

In a conventional direct contact cooling tower (see figure 1.1) the water to be cooled flows through the cooling tower where it is cooled by counterflow or cross-flow airstream. The cooled water is then passed through a heat exchanger or a condenser to cool a process fluid or condense a vapour. This requires two separate units, i.e. the cooling tower and the heat exchanger or condenser.

An evaporative cooler or condenser combines the heat exchanger or condenser and the cooling tower in one unit with the evaporative cooler or condenser tubes replacing the packing of the cooling tower. Figure 1.2 shows a schematic layout of a counterflow evaporative cooler.

The operation of an evaporative cooler or condenser can be described as follows: Recirculating water is sprayed onto a bank of horizontal tubes containing a hot process fluid or a vapour which is to be condensed while air is drawn across the wet tube bank. The recirculating water is heated by the hot process fluid or the condensing vapour inside the tubes while it is cooled from the airside by a combined heat and mass transfer process.

The airflow through the evaporative cooler or condenser may be horizontal, in which case the unit is referred to as a cross-flow evaporative cooler or condenser or vertically upwards through the tube bundle where it is known as a counterflow evaporative cooler or condenser. Various other configurations of evaporative coolers or condensers have been proposed in the literature, but these are not commonly used.

In this report analytical models for the evaluation of cross-flow and counterflow evaporative coolers and condensers are presented. The models range from a comprehensive model which requires numerical integration and successive calculations to a simplified model which allows easy and quick sizing and rating calculations. Computer programs have been written to analyse cross-flow and counterflow evaporative coolers and condensers.

Since correlations or data for heat and mass transfer coefficients for cross-flow evaporative coolers are practically non-existent, a series of tests were performed on such a unit in order to determine the required coefficients experimentally. The two phase pressure drop across the wet tube bundle was also measured and compared with existing correlations.

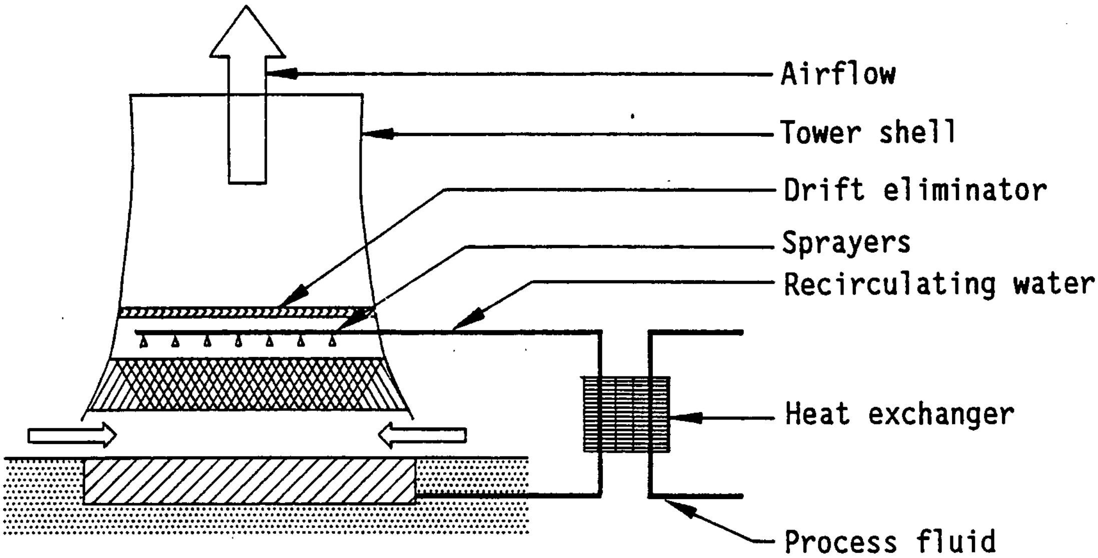

Figure 1.1 Conventional direct contact counterflow cooling tower layout.

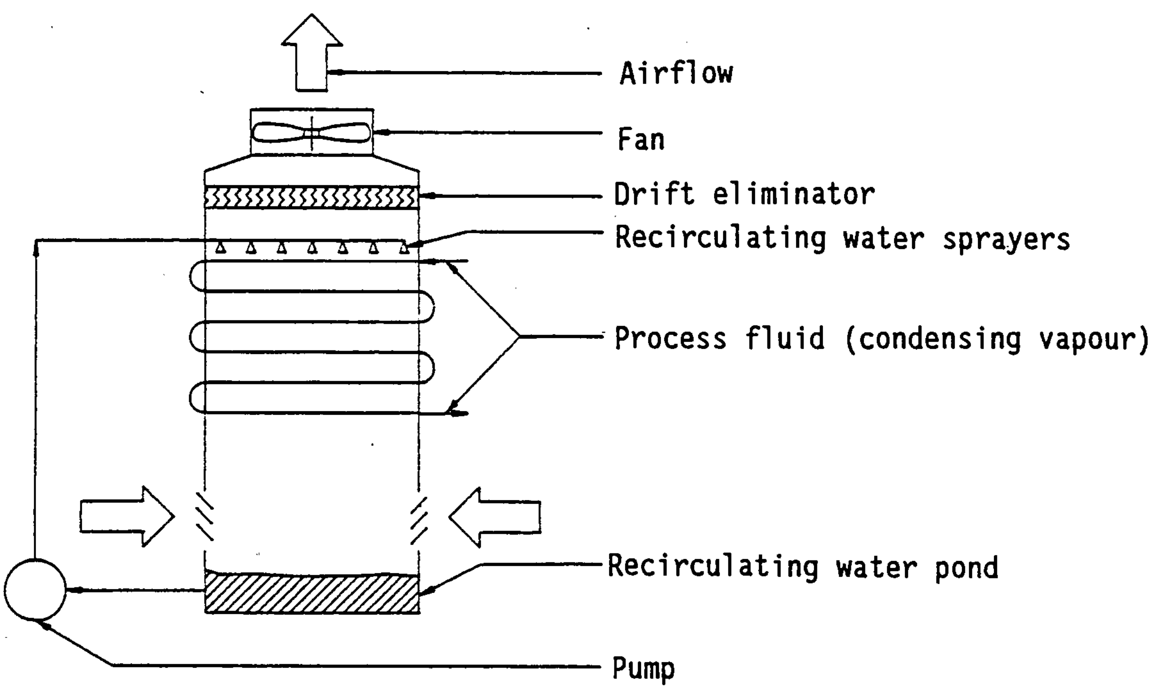

Figure 1.2 Counterflow evaporative cooler (or condenser).

CHAPTER 2

LITERATURE SURVEY

The mathematical modelling of an evaporative cooler or condenser is complicated by the fact that three fluids, sometimes flowing in different directions, interact with heat and/or mass transfer processes taking place.

Numerous modelling procedures each with varying degrees of approximation, can be found in the literature. The older models often assumed the recirculating water temperature to be constant throughout the cooler and most of the models used a one-dimensional modelling procedure.

Several authors have studied other types of evaporative coolers or condensers where the air and the recirculating water flow inside the tubes or through narrow slots between closely spaced plates. These studies are also of interest since the same interface phenomena occur in these coolers or condensers as those considered in the current study.

Scott [29SC1] conducted a series of simple tests on a single tube evaporative condenser to determine the coefficients involved in the heat and mass transfer process.

The apparatus used consisted of a vertical tube with steam condensing on the outside of the tube, while the recirculating water flowed as a thin film on the inside of the tube concurrent with the air stream. A sample design procedure for a single tube evaporative condenser was also presented.The recirculating water temperature was assumed to be constant throughout the tube.

One of the first attempts to evaluate a conventional horizontal tube evaporative condenser was made by James [37JA1]. He described an evaporative condenser in which the water was splashed up into the airstream from a sump by a revolving perforated drum. The water was then carried to the horizontal tube bundle by the air which flowed through the tube bundle.

Since the air reached the coil almost saturated, he assumed that the driving force for the heat transfer was the difference between the condensing temperature and the air wet bulb temperature. Mathematically his method stated

$$q = K_{wb} \left(T_r - T_{awb} \right) A \tag{2.1}$$

where K_{wb} was called the "wet bulb K" and was defined as:

$$K_{wb} = \left[\frac{d_o}{d_i\ h_r} + \frac{t}{k_t} + \frac{1}{h_w} + \frac{1}{h_c} \right]^{-1} \tag{2.2}$$

James noted that the tube to water and water to air coefficients, h_w and h_c, would be the controlling coefficients, and that a significant improvement in performance of the condenser could be achieved if these coefficients could be improved. A simple graphical design procedure was also presented.

<u>Goodman</u> [38GO1] and [38GO2] gave the first useful procedure to rate or to design counterflow evaporative condensers. In his analysis of the process he uses the enthalpy potential, as first derived by Merkel [26ME1], as the driving force for heat transfer from the recirculating water to air.

He used the difference between the condensing temperature and the recirculating water temperature as the driving force for the heat transfer from the refrigerant.

The assumption of a constant recirculating water temperature throughout the condenser was justified as follows:

" In as much as the spray water wets the outside surface of the coil, the heat is transferred through the wall of the coil to the water on its outer surface. But, the water as fast as it receives this heat trans=fers it in turn to the air flowing over the coil. As the water is neither heated nor cooled while it is circulated, it must attain an equilibrium temperature, but remains constant as long as the operating conditions remain unchanged".

Although this is not strictly correct, it is still in fairly good approxi=mation in the case of a counterflow evaporative condenser.

The design method relied on a graph which was used to determine the recirculating water temperature if the condensing temperature and the entering air wet bulb temperature was known.

The capacity of the condenser could then be determined from

$$q = U_o \left(T_r - T_w \right) A \qquad (2.3)$$

or from

$$q = m_a \left(i_{asw} - i_{ai} \right) E \qquad (2.4)$$

where

$$E = 1 - e^{-N} \qquad (2.5)$$

and

$$N = \frac{h_c A_o}{c_{pm} m_a} \qquad (2.6)$$

Note that if the Lewis relation holds we have that

$$\frac{h_c}{h_D c_{pm}} = 1 \quad => \quad h_D = \frac{h_c}{c_{pm}} \qquad (2.7)$$

By substituting this into the relation for N it follows that

$$N = \frac{h_D A_o}{m_a} \qquad (2.8)$$

This is similar to an ε - NTU_a approach with the one fluid at a constant temperature ($C_{max}/C_{min} \rightarrow \infty$) and $N = NTU_a$

No correlations for any of the coefficients were given.

Thomson [39TH1] studied the heat and mass transfer processes in an evaporative condenser and conducted a series of tests to determine the heat transfer coefficient and the rate of evaporation from the water film to the air.

A single horizontal tube (at a slight angle) in a horizontal airstream was studied. It was found that the amount of water evaporated from the tube was dependant on the film thickness (more water evaporates off a thinner film) and that the total amount of water evaporated was always less than 8%.

Thomsen [46TH1] proposed a graphical design procedure for simple evaporative condenser design calculations. He assumed that the recirculating water temperature stays constant throughout the condenser and used the method given by Goodman [38GO1] to determine the temperature of the recirculating water.

He formulated the concept of a single resistance for latent and sensible heat transfer by assuming that the driving force is the difference between the spray water (recirculating water) temperature and the average air wet bulb temperature. This approach is similar to that of James [37JA1].

Wile [50WI1] studied the operation of evaporative condensers proving that the recirculating water temperature is not constant throughout the condenser. For calculation purposes he assumed that the recirculating temperature can be represented by a single equivalent temperature which would yield the same result as a varying temperature profile. He proposed a method by which the performance of an evaporative condenser at any operating condition could be determined by using the results of a few tests over the normal operating range. The representative test data can be converted into rating tables or curves that would apply over a wide temperature range.

Wile [58WI1] discussed the operation of an evaporative condenser, covering subjects like bleeding, scale deposits, winter control methods, desuper= heating coils and general system performance. No design method was given.

Parker and Treybal [61PA1] gave the first accurate design procedure for the evaluation and design of vertical counterflow evaporative coolers. The model was kept simple by employing the following assumptions:

i) The air-water heat and mass transfer can be described using the "Merkel" type equation

$$dq = h_D \left(i_{asw} - i_a \right) dA, \tag{2.9}$$

ii) The Lewis factor is equal to unity

$$\frac{h_c}{h_D \, c_{pm}} = 1, \tag{2.10}$$

iii) The amount of water evaporated from the cooler was considered negligible,

iv) Air saturation enthalpy is a linear function of temperature.

In solving the model Parker and Treybal realized that the recirculating water temperature could not be constant, but since the variation in recirculating water temperature is small the error introduced by assuming a linear relation between the air saturation enthalpy and the recirculating water temperature is negligible.

After manipulation of the governing equations the set of three differential equations could be solved analytically. The resulting three equations could now be employed in the rating or selection of evaporative coolers. The model as given by Parker and Treybal is not explicit, since the coefficients in the design equations have to be found by simultaneous solution of these equations.

Parker and Treybal also conducted a series of tests on a vertical airflow evaporative cooler to determine the required mass and heat transfer coefficients. Correlations for these coefficients were determined from the test data.

N.A. Harris [62HA1] and [64HA1] described the operation of a new type of cooler which he called an "air-evaporative cooler."

According to Harris the definition of such a cooler is given by the following:

"Air-evaporative cooler units have all or part of their heat transfer surface as finned tubing so it can operate as straight air coolers when the air temperature is low enough. When the air temperature is not sufficiently low to produce the desired process temperature, a water spray can be turned on to provide evaporative cooling."

He gave no design method of such a cooler. Various configurations were proposed and simple cost comparisons were made between the air-evaporative cooler and conventional dry coolers.

Mizushina et al. [67MI1] described an experimental study performed on a counterflow evaporative cooler. The controlling transfer coefficients: *

i) between the process fluid and the tube wall,

ii) between the tube wall and the recirculating film and

iii) between the recirculating water and the air were determined.

The experiments were conducted using three different tube sizes:

i) d_o = 12,7 mm, d_i = 10,7 mm,

ii) d_o = 19,05 mm, d_i = 16,05 mm and

iii) d_o = 40,0 mm, d_i = 38 mm

The tubes were spaced in a 2 x d_o pitch triangular array in eight or twelve tube rows. Mizushina et al. [67MI1] measured the recirculating water temperature at various places inside the cooler and they observed a temperature variation in the region of 2°C. They used a simplified model originally proposed by McAdams [54MC1] to determine an approximate average recirculating water temperature. Using this approximate average recirculating temperature the controlling transfer coefficients were found by employing logarithmic type equations describing the heat and mass transfer through the whole cooler.

Correlations for the required transfer coefficients were derived for the numerical evaluation of evaporative coolers.

Mizushina et al.[68MI1] described the thermal design of vertical airflow evaporative coolers. The one-dimensional model used was derived in detail by evaluating the energy and mass balances of a single small element.

The main assumptions made were:

i) No change in recirculating water massflow (evaporation neglected),

ii) Lewis factor = 1,

iii) The saturation enthalpy of air is a linear relation of temperature in the applicable temperature range.

Two design methods were proposed. The first method is based on the method given by McAdams [54Mc1]. Assuming recirculating water temperature to be constant, the governing equations can be integrated analytically into a single equation which can be used iteratively for rating or sizing calculations.

For the second design method the cooler is divided into a number of vertical elements and the three governing differential equations are integrated (using a numerical method) for every element. By a method of successive calculation the whole cooler is then evaluated.

The paper gives a numerical example of each of the two design methods.

<u>Finlay and McMillan</u> [70FI1] derived an analytical model to evaluate the performance of a mist cooler, the mist cooler consisted of a horizontal tube bank with horizontal airflow across the tubes. Small amounts of water spray was added to the air flow in order to wet the tubes. The analytical model which is based on the work of Berman [61BE1] represents the heat and mass transfer process in terms of five differential equations. Separate equations were derived to describe the transfer process when the air has become saturated.

By numerical integration of the controlling equations the local air properties, the cooling water and process fluid temperature could be determined for every position in the cooler. This method was used for the evaluation of a typical cooler and the effects of varying spray water inlet temperatures and varying air velocity were determined.

The required coefficients for heat and mass transfer were calculated from dry tube data, employing the Lewis relation and by using Elperins' [61EL1] equation for two-phase heat transfer.

Two-phase pressure drop measurements were made and the data compared favourably with the simple theoretical model cited.

N.A.

Anastasov [67AN1] tested a vertical tube evaporative condenser where the vapour condensed on the outside of the vertical tubes while both air and recirculating water flowed downwards through the inside of the tubes. The test results were discussed and guide values for the size and capacity of vertical tube condensers were given.

N.A.

Kals [71KA1] described an evaporative cooler where the air enters from the top and flows downwards over a tube bundle concurrent with a gravity flow of recirculating water. The airstream is then turned upwards again before it is discharged. The concurrent flow of the air and the water prevents the breakup of the water blanket which covers the tubes. Changing the direction of the airstream after it has passed through the tube bundle forces all the entrained water droplets to leave the airstream. A simple graphical design procedure was provided.

Tasnadi [72TA1] was the first author to describe the operation of a cross-flow evaporative cooler. His model employed the following assumptions:

i) Lewis factor = $\dfrac{h_c}{h_D \, c_{pm}} = 1$,

ii) The air at the air/water boundary is saturated at the bulk recirculating water temperature and

iii) the water film flow is so turbulent that the temperature of the film can be taken as the bulk water temperature $(T_w = T_i)$.

Using these assumptions he derived a model in which the heat transfer from the process liquid to the water film was given by

$$dq = U_o \left(T_p - T_w \right) dA \tag{2.11}$$

and the heat transfer from the film to the air could be described by

$$dq = h_D \left(i_{asw} - i_a \right) dA \tag{2.12}$$

By transforming the heat transfer driving force between the process fluid and the recirculating water to an enthalpy driving force he could then write the complete heat/mass transfer process as

$$dq = h_{Do} \left(i_{asp} - i_a \right) dA \tag{2.13}$$

where

$$h_{Do} = \left[\frac{1}{h_D} + \frac{b}{U_o} \right]^{-1} \tag{2.14}$$

Tasnadi gave no indication of how the heat and mass transfer coefficients were determined, and no numerical solution or example was given.

<u>Tezuka et al</u>.[72TE1] modelled the operation of an evaporative cooler in terms of an overall mass transfer coefficient and a Merkel type enthalpy difference.

The overall heat/mass transfer coefficient approach used is similar to that used by Tasnadi [72TA1] in that the governing equation for the heat transfer from the process fluid to the air is written in terms of an enthalpy driving force.

Tezuka also determined simple dimensional correlations for the overall transfer coefficient and for the pressure drop across the coil.

<u>Tezuka</u> [72TE2] continued his previous work on evaporative coolers by conducting series of experiments on a counterflow evaporative cooler to determine the film heat transfer coefficient which governs the heat transfer from the tube wall to the recirculating water film. A dimensional empirical correlation is given for the film heat transfer coefficient.

Finlay and Grant [72FI1] formulated a comprehensive thermal design model for evaporative coolers. This model did not assume the Lewis factor to be unity and the evaporation of the recirculating water was also taken into consideration. The cooler evaluation was performed by numerically integrating the controlling differential equations through the whole cooler.

A simplified model was also obtained by assuming the Lewis factor to be equal to unity and by ignoring the evaporation of recirculating water. This simplified model gave three controlling differential equations which still had to be solved numerically.

A fairly comprehensive literature study summarized most of the important contributions for the determination of the controlling heat- and mass transfer coefficients.

An example of a typical rating solution was also given, evaluating the effects of varying heat and mass transfer coefficients. It was noted that although the correlations of Mizushina et al. and Parker and Treybal vary by up to 30 %, the opposing effect (h_w predicted lower by Parker and Treybal, while they predicted a higher mass transfer coefficient) of these differences cancel and the overall agreement in performance prediction between these two methods are good.

Finlay and Grant [72FI1] found that in the presence of fins on the outside of the tubes the heat transfer coefficient from the tube to the recirculating water film was reduced, but the mass transfer coefficient between the recirculating water and the air was considerably enhanced. The lower film heat transfer coefficient was attributed to the water held up in between the fins by surface tension. It was consequently proposed that the airflow should be arranged downwards to flow concurrently with the recirculating water to assist in the transport of the recirculating water through the tube bank.

Finlay and Grant [74FI1] compared the accuracy of various design procedures for evaporative coolers. As a reference the accurate model which was introduced in a previous paper by the same authors [72FI1] was used. The mass and heat transfer coefficients required were obtained from the correlations of Mizushina et al. [67MI1].

It was found that the simplified method of Parker and Treybal [61PA1] was "in good agreement for most engineering purposes" to the accurate method.

According to Finlay and Grant the "usual logarithmic temperature driving force does not apply" because of the recirculating water profile that exists. The methods of James [37JA1], Goodman [38GO1] [38GO2], Thomsen [38TH1] and Wile [50WI1] were consequently not used for comparison since these methods employed a single mean recirculating water temperature.

The rating method of Tezuka et al. [76TE1] was shown to differ quite significantly from the accurate solution.

They concluded that for tube banks of less than seven rows and a small cooling range the assumption of constant recirculating water temperature would be reasonably valid as long as a close approach to air wet bulb temperature is not required.

Mizushina et al.[74MI1] presented a simple design procedure for the design of evaporative coolers or condensers. This model is similar to the simplified model given by Mizushina et al. [68MI1]. Flow charts of the calculation procedure were also provided.

Tezuka et al.[76TE1] experimentally evaluated five different evaporative cooling cores to determine correlations for the pressure drop across wet tube bundles and the overall transfer coefficient as defined in a previous paper [72TE1].

Correlations for pressure drop and the overall transfer coefficients were presented for each of the five evaporative cooler coils. These correlations were subsequently written in terms of dimensionless groups and a single relation for the overall transfer coefficient was then derived to unite the existing five correlations.

Kreid, Johnson and Faletti [78KR1] used a similar approach to Tasnadi [72TA1] and Tezuka et al.[72TE1] to give the governing equations for the operation of a wet surface finned heat exchanger in terms of an enthalpy difference and an overall transfer coefficient.

The governing equations for a wet surface heat exchanger was shown to have the same form as the corresponding dry surface equations, which then gave the governing equation for a wet surface cooler (finned or unfinned) as

$$q = F h_{Do} \left(i_{asp} - i_a \right) A \tag{2.15}$$

where F is the conventional correction factor used in the LMTD approach of heat exchanger design.

The design method for wet surface heat exchangers was then also extended to the heat exchanger effectiveness form [ε - NTU form].

The wet heat and mass transfer coefficients were obtained from the analogy between heat and mass transfer.

Kreid, Hauser and Johnson [81KR1] continued the previous work of Kreid et al. [78KR1] by experimentally evaluating the unknown wet fin heat transfer coefficient. This coefficient could not be determined from either first principles or from existing empirical correlations.

Threlkeld [70TH1] discussed the operation of wet surface finned tube heat exchangers. A similar approach to that of Kreid et al. [78KR1] and [81KR1] was used in that a fictitious saturation enthalpy at the process fluid temperature was defined. The heat transfer from the process fluid to the air was then described in terms of an overall mass transfer coefficient and a mean logarithmic enthalpy difference.

Leidenfrost and Korenic [79LE1] analyzed the operation of a counterflow evaporative condenser. The analytical model derived was based on earlier graphical method by Bosjnakovic [60BO1]. The only significant simplifying assumption made in the analytical model was the assumption that the Lewis factor is equal to unity.

The solution of this model involves a rather complicated integration procedure, involving so-called "pulling points" which could be graphically illustrated on a Mollier i_a - w_a chart. The model takes partial dryness of certain tubes into consideration by a rather crude dryness factor which has to be specified. The condenser to be evaluated is divided into elemental modules. By a successive numerical evaluation of each module in the condenser the operating point of the condenser can be found.

All the required coefficients were discussed in detail except the mass transfer coefficient. The mass transfer coefficient is calculated from the analogy between mass and heat transfer.

The results of evaporative condenser simulations show that evaporative condensers can still operate at ambient dry bulb temperatures higher than the condensing temperature and that close fin spacing is not required. The amount of water evaporated is said to be about 1% of the recirculating water flow.

In a later paper Leidenfrost and Korenic [82LE1] used the same model as derived previously [79LE1] to evaluate finned counterflow evaporative condensers. This paper discussed experimental work which was done on evaporative condensers in order to verify the computer model which had been set up.

In the computer model the Lewis factor was not assumed to be unity, but it was calculated from a relation given by Bosjnakovic [60BO1].

The findings of the tests were described in detail and an empirical relation was given for the film heat transfer coefficient. This correlation gives values which fall between the values predicted by similar correlations given by Mizushina et al. [67MI1] and Parker and Treybal [61PA1].

A graphical representation of the measured pressure drop across the wetted coil was presented. This showed an increase of up to 40 % in pressure drop across the wet coil compared to the dry operation of the same coil.

It was experimentally shown that the amount of recirculating water needed for complete wetting of the coil was sufficient to ensure maximum performance of the evaporative condenser. Increasing the air flow rate increased the capacity of the condenser until up to a point where the airflow caused the water film to break up.

<u>Fisher, Leidenfrost and Li</u> [83FI1] described the modelling and operation of a vertical tube evaporative condenser. In the cooler described the air flows upwards inside the vertical tube while the recirculating water flows downward as a thin film inside the tube. Vapour is condensed on the outside of the tubes. The condenser is similar to units described by Anastasov [67AM1] and Perez-Blanco [82PE1] and [84PE1].

An experimental study was conducted to determine the controlling coefficients used by the computer simulation program. The program used was a modified version of the original program compiled by Leidenfrost and Korenic [82LE1] for the evaluation of finned counterflow evaporative condensers.

<u>Perez-Blanco and Bird</u> [82PE1] and [84PE2] studied the heat and mass transfer process that occurs in a vertical tube evaporative cooler where the air and the cooling water film flow countercurrently inside the tube. An analytical model based on existing heat and mass transfer correlations was developed. These transfer coefficients were then experimentally varified.

Perez-Blanco and Linkous [83PE1] studied a similar vertical evaporative cooler to Perez-Blanco and Bird [82PE1]. They noted that the common drawback in existing procedures to evaluate evaporative coolers lies in the fact that the driving forces for heat and mass transfer differ.

They defined a fictitious air saturation enthalpy (at the process water temperature) to formulate a single overall coefficient. According to their model the capacity is given by

$$q = h_{Do} \, A \, LMED \tag{2.16}$$

where

$$\frac{1}{h_{Do}} = \frac{b}{U_o} + \frac{1}{h_D} \tag{2.17}$$

and

$$LMED = \frac{\left(i_{aspi} - i_{ai} \right) - \left(i_{aspo} - i_{ao} \right)}{\ln \left(\dfrac{i_{aspi} - i_{ai}}{i_{aspo} - i_{ao}} \right)} \tag{2.18}$$

The formulation of a single transfer coefficient allowed the identification of the controlling resistance in the transfer process. They identified the controlling resistance to heat and mass transfer as being concentrated at the air/water interface.

When this model was experimentally verified it was found that the LMED formulation could only be used for evaporative condensers or when the cooling/recirculating water temperature change was small, otherwise a stepwise evaluation would be necessary.

Perez-Blanco and Webb [84PE1] noted from the work of Perez-Blanco and Linkous [83PE1] that the controlling resistance at the air/water interface has to be lowered in order to enhance the performance of a vertical tube evaporative cooler. They studied the effect of coiled wire turbulence promoters inside the vertical tube as an alternative to extended surfaces. The turbulence promoters were placed away from the tube wall in order to mix the air boundary layer and not the water film. Experimental work showed a marked increase in cooler performance. The spacing between the promoter and the tube wall was found to be of critical importance.

Peterson [84PE3] studied the operation of a counterflow evaporative condenser and modelled the heat and mass transfer processes at the air-water interface very thoroughly. The complete model given by Peterson required a set of eight differential equations to be solved, which would require a numerical integration procedure. This model was then significantly simplified to give a model very similar to the model of Parker and Treybal [61PA1] for an evaporative cooler. The major simplifications were

i) The Lewis factor was taken as unity,

ii) The evaporation of recirculating water was ignored and

iii) Air saturation enthalpy was taken as a linear function of temperature for the operating temperature range.

Peterson obtained values for the controlling mass transfer and film heat transfer coefficients after a series of experiments on an industrial evaporative condenser, condensing Freon-22.

The correlation for the mass transfer coefficient agrees very well with that obtained by Parker and Treybal [61PA1] but she could not correlate the film coefficient because of the scatter of the experimental readings. Criticism could however be raised against the assumption of Peterson that the condensation heat transfer coefficient for the Freon-22 condensing on the inside of the tubes is constant at 8000 W/m^2K. In the condensation of Freon the heat transfer coefficients are normally found to be in the region of 1500 W/m^2K because of the low thermal conductivity of liquid Freon. The low condensation heat transfer coefficient would be the governing resistance to heat transfer from the condensing Freon to the water film on the outside of the tubes.

The fact that Peterson could not find a correlation for the film coefficient after measuring the overall heat transfer coefficient could be ascribed to this incorrect assumption.

Webb and Villacres [84WE1] and [84WE2] made the following assumptions to simplify the evaluation of cooling towers, evaporative coolers and evaporative condensers:

i) The total heat flux can be written in terms of the enthalpy difference of moist air, the so-called "Merkel equation",

ii) The loss of water through evaporation, entrainment and blowdown could be ignored,

iii) The saturation enthalpy of the air at the air/water interface can be calculated at the bulk recirculating water temperature rather than at the recirculating water interface temperature,

iv) Uniform and complete wetting of the packing or tubes and

v) Heat and mass transfer coefficients are constant through the whole process.

The controlling heat and mass transfer coefficients governing the operation of the cooling tower, evaporative cooler and evaporative condenser units are discussed. The modelling procedures for evaporative coolers and evaporative condensers assume a constant recirculating water temperature.

This approximation makes it possible to integrate the controlling equations and to give a single equation governing the operation of an evaporative cooler or condenser. Mizushina et al. [67MI1], [68MI1] used a similar approach to evaluate evaporative coolers.

Rating and selection procedures for all three types of cooler units are described for both the simplified approach and the successive calculation methods.

In a later paper Webb and Villacres [84WE2] used the methods described in their previous work [84WE1] to set up computer programs for the rating of any of the three units as described on the first article [84WE1] at off-design conditions.

The heat and mass transfer characteristic for the cooler or condenser to be rated is determined from the rating data at the design point. These programs allow the user to evaluate the effects of various off-design conditions on the cooler. The programs were able to predict the rating data given by the manufacturers within 3% for the coolers evaluated. Various other papers [84WE3],[84WE1] and [85WE1] described the same work as given in this article.

The complete Fortran program codes for all three the rating programs were included in the paper.

Wassel et al. [84WA1] and [87WA1] modelled a countercurrent falling film evaporative condenser consisting of closely spaced vertical metal plates. On the air side of each plate a water film flows downwards while the airstream flows upwards while vapour condenses on the other side of the plate. The model developed does not assume a constant recirculating water temperature through the cooler and it takes into account the cooling of the recirculating water after it leaves the bottom of the plates until it reaches the water sump. They found that the cooling of the recirculating water between the sprayers and the top of the plates is negligible.

Rana et al. [86RA1], [87RA1] tested various counterflow single and multi-tube evaporative coolers to evaluate the mass transfer from the recirculating water to the air.

They compared their data to the theoretical mass transfer prediction obtained by employing the Chilton Colburn heat/mass transfer analogy with a Lewis factor of 0,92. It was found that the mass transfer from a single tube evaporative cooler was between 200 % and 500 % higher than for a multi-tube evaporative cooler.

The correlations given by Rana et al. for design purposes gives the correction factor which should be used together with the heat/mass transfer analogy to determine the mass transfer coefficient. The correlations given include a term ($\Delta i/i_{fg}$) where Δi is a function of the air inlet and outlet conditions. The fact that the outlet air enthalpy is required to determine the mass transfer coefficient presents a complication if the equations are to be used for cooler rating, since the outlet conditions are not known in the rating calculations.

Erens [87ER1] used the principles of the design method of Mizushina [68MI1] to build a computer model for rating and sizing of evaporative cooler units.

Block diagrams were presented to show the calculating procedure for the rating and sizing calculations. The counterflow cooler was divided into a number of elementary units; for each unit the controlling differential equations were solved to obtain the inlet/outlet conditions for the next element. By a method of successive calculation the whole cooler could then be evaluated.

Since it is known that the inlet and outlet recirculating water streams must have the same temperature, the solution procedure assumes a value of the outlet recirculating water temperature and by the successive calculating procedure the inlet recirculating water temperature is found.

The correct choice of inlet recirculating water temperature will give an outlet recirculating water temperature which is equal to the chosen inlet temperature.

Examples of the temperature profiles along the flow path were given as well as numerical examples of the rating and the selection programs.

Erens [88ER1] realized that conventional Munters type cooling tower fill could be used together with the bare coil of the evaporative cooler to enhance the performance of the unit.

The packing has the effect of enlarging the mass/heat transfer area and consequently the average recirculating water temperature is lowered resulting in an improved cooler capacity. Two different variations were compared to the bare tube cooler by employing modified versions of the bare cooler rating program. The so-called "integral cooler" combined the coils and the packing while the second layout consisted of a conventional bare coil section with the packing placed underneath the coil.

A typical comparative calculation gave the following results:

Cooler	Capacity
Bare tube cooler	147,1 kW
Integral fill cooler	199,2 kW
Bare tube + fill cooler	206,6 kW

Erens noted that by using fill together with the tubes it was possible to use a considerable number of tube rows less than would be required for a bare tube cooler of the same capacity.

Erens and Dreyer [88ER2] used the more accurate modelling procedure of Poppe [84PO1] and Bourillot [83BO1] to evaluate a typical element of an evaporative cooler. This model did not include the Merkel assumptions of a Lewis factor equal to unity and negligible water loss as result of evaporation.

Five controlling differential equations were given for the evaluation of a typical element when the air is not saturated. If the air entering an element is saturated, the mass transfer driving potential changes and separate controlling equations were derived for this case.

The modelling procedure is similar to that of Leidenfrost and Korenic in the sense that each element (module) was considered to be an imaginary block around a length of tube.

By using a fourth order Runge-Kutta integration process with successive calculations the whole cooler could be evaluated. Both cross-flow and counterflow evaporative coolers were evaluated. Typical temperature profiles were presented for both types of cooler units and it was shown that the temperature variation of the recirculating water at the outlet side was negligible in the case of a counterflow cooler, thus a one dimensional analysis model would be sufficient.

The non-existance of correlations for heat and mass transfer coefficients for cross-flow evaporative coolers was stated as the reason for the application of the counterflow correlations for these coefficients. The Lewis factor was calculated using the relation given by Bosjnakovic [60BO1].

In conventional cooling tower theory the Merkel type model has become the accepted model for the analysis of a direct contact cooling tower.

Since 1970 various investigators, including Yadigaroglu and Pastor [74YA1] Bourillot [83BO1], Majumdar et al.[83MA1], Sutherland [83SU1] and Poppe [84PO1], have proposed more accurate models for the analysis of conventional wet cooling towers.

Webb [88WE1] gave a critical evaluation of current cooling tower practice. The assumptions made in the different models were clearly shown, and the effect of the different assumptions were discussed in detail. *

Although the conventional cooling tower theory is not directly applicable to evaporative coolers or condensers the fundamentals of the heat and mass transfer from the water to the air at the interface are similar.

The various modelling procedures for evaluating evaporative coolers and condensers, given in the literature vary significantly in accuracy and complexity of use. In many of the earlier models, the basic equations were not explicitly stated which resulted in some dubious design models. The first accurate mathematical model was presented by Parker and Treybal [61PA1]. Various accurate numerical integration models have since then been published in the literature. In many of the articles the mass and heat transfer coefficients are not adequately defined and sometimes certain coefficients are not defined at all. None of the models presented in the literature has yet been established as the accepted model for the analysis of evaporative coolers and condensers.

CHAPTER 3

MATHEMATICAL MODELLING OF EVAPORATIVE COOLERS AND CONDENSERS

In the theoretical analysis of cross-flow and counterflow evaporative coolers and condensers the following assumptions are made to obtain the analytical model:

i) the system is in a steady state,

ii) radiative heat transfer can be ignored,

iii) low mass transfer rates (At high mass transfer rates the heat transfer coefficient would be influenced by the mass transfer rate; refer to Apppendix I),

iv) even distribution of recirculating water along each tube and complete wetting of the tube surface,

v) the water film temperature at the air/water interface is approximately equal to the bulk film temperature (see Appendix H for a discussion of this assumption),

vi) the temperature rise of the recirculating water because of pump work is negligible,

vii) the air/water interface area is approximately the same as the outer surface of the tube bundle, i.e. the water films on the tubes are very thin, and

viii) the heat transfer to the surroundings from the U-bends outside the cooler or condenser can be assumed to be negligible.

By employing these assumptions the analytical models for both evaporative coolers and condensers can now be derived from basic principles.

The exact analytical method presented uses the same basic approach as Poppe [84P01] and Bourillot [83B01] to describe the transfer processes between the air and the recirculating water in a conventional cooling tower.

The more commonly used Merkel model can easily be found from the controlling equations of the exact analytical model.

3.1 Basic theory for evaporative coolers

3.1.1 Exact analysis (Poppe model)

Consider a typical element of an evaporative cooler. The inlet and outlet conditions of the cross-flow and counterflow elements are shown in figure 3.1)a) and figure 3.1)b) respectively. This choice of inlet and outlet conditions results in the same sign convention for cross-flow and counterflow units, and consequently the controlling equations would have the same signs for both cross-flow and counterflow models.

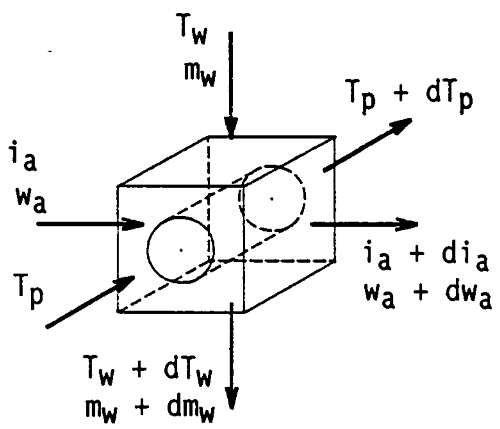

Figure 3.1)a) Control volume for cross-flow evaporative cooler cooler analysis.

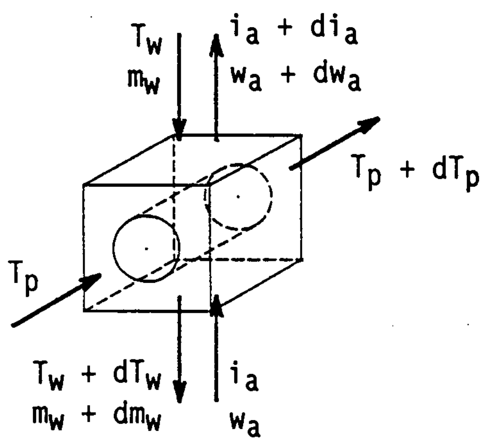

Figure 3.1)b) Control volume for counterflow evaporative analysis.

The mass balance of the control volume gives

$$m_a \, dw_a + dm_w = 0$$

$$\therefore dw_a = - \frac{dm_w}{m_a} \qquad (3.1.1)$$

The energy balance of the control volume gives

$$m_a i_a + m_w c_{pw} T_w + m_p c_{pp} T_p =$$

$$m_a \left(i_a + di_a \right) + \left(m_w + dm_w \right) c_{pw} \left(T_w + dT_w \right) + m_p \, c_{pp} \left(T_p + dT_p \right)$$

After simplification and by ignoring the second order terms the energy balance gives

$$m_a di_a + m_w c_{pw} dT_w + c_{pw} T_w dm_w + m_p c_{pp} dT_p = 0$$

$$\therefore dT_w = \frac{1}{m_w \, c_{pw}} \left[- m_a di_a - c_{pw} T_w dm_w - m_p c_{pp} dT_p \right] \qquad (3.1.2)$$

The controlling equation governing the heat and mass transfer from the water film to the air is dependant on whether or not the air is over-saturated (mist).

CASE 1 - Non-saturated moist air

The massflow of recirculating water evaporating from a typical element into non-saturated air is given as

$$dm_w = -h_D \left(w_{asw} - w_a \right) dA_o \qquad (3.1.3)$$

At the water/air interface simultaneous heat and mass transfer takes place as given by

$$dq = -i_v dm_w + h_c \left(T_w - T_a \right) dA_o \qquad (3.1.4)$$

By using equation (3.1.3) and noting that $dq = m_a di_a$ this becomes

$$m_a di_a = h_D \left(w_{asw} - w_a \right) i_v dA_o + h_c \left(T_w - T_a \right) dA_o \qquad (3.1.5)$$

The following supplementary equations can now be used to simplify the equation above:

i) $c_{pm} = c_{pa} + w_a c_{pv}$ (3.1.6)

ii) $i_v = i_{vo} + c_{pv} T_w$ (3.1.7)

iii) $i_a = \left(c_{pa} + w_a c_{pv} \right) T_a + w_a i_{vo}$ (3.1.8)

iv) $i_{asw} = \left(c_{pa} + w_{asw} c_{pv} \right) T_w + w_{asw} i_{vo}$ (3.1.9)

Rewriting equation (3.1.5) and employing equation (3.1.6) gives

$$m_a di_a = h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D c_{pm}} \right) \left(c_{pa} + w_a c_{pv} \right) \left(T_w - T_a \right) \right]$$

$$= h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D c_{pm}} \right) \left[\left(c_{pa} + w_a c_{pv} \right) T_w - \left(c_{pa} + w_a c_{pv} \right) T_a \right] \right]$$

By rewriting equation (3.1.8) as $\left(c_{pa} + w_a c_{pv} \right) T_a = i_a - w_a i_{vo}$ and substituting it into the relation above, it follows that

$$m_a di_a = h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D c_{pm}} \right) \left[\left(c_{pa} + w_a c_{pv} + w_{asw} c_{pv} - w_{asw} c_{pv} \right) T_w - i_a + w_a i_{vo} \right] \right]$$

$$= h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \left[\left(c_{pa} + w_{asw}\, c_{pv} \right) T_w - \left(w_{asw} - w_a \right) c_{pv}\, T_w - i_a + w_a i_{vo} \right] \right]$$

By rewriting equation (3.1.9) as

$$\left(c_{pa} + w_{asw}\, c_{pv} \right) T_w = i_{asw} - w_{asw}\, i_{vo}$$

and substituting it in the relation above, gives

$$\therefore m_a di_a = h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \left[i_{asw} - w_{asw} i_{vo} - \left(w_{asw} - w_a \right) c_{pv} T_w - i_a + w_a i_{vo} \right] \right]$$

$$= h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) \left(i_{vo} + c_{pv} T_w \right) \right] \right]$$

By noting from equation (3.1.7) that $i_{vo} + c_{pv} T_w = i_v$, this becomes

$$m_a di_a = h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right] \right]$$

$$= h_D dA_o \left[\left(w_{asw} - w_a \right) i_v + \left[\left(\frac{h_c}{h_D\, c_{pm}} \right) - 1 \right] \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right] + \left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right]$$

$$= h_D dA_o \left[\left(i_{asw} - i_a \right) + \left[\left(\frac{h_c}{h_D\, c_{pm}} \right) - 1 \right] \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right] \right]$$

$$\therefore di_a = \frac{h_D\, dA_o}{m_a} \left[\left(i_{asw} - i_a \right) + \left[\left(\frac{h_c}{h_D c_{pm}} \right) - 1 \right] \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right] \right] \qquad (3.1.10)$$

CASE 2 - Saturated air

The massflow rate of recirculating water evaporating from the tube surface of a typical element into saturated air is controlled by the following equation

$$dm_w = -h_D \left(w_{asw} - w_{as} \right) dA_o \qquad (3.1.11)$$

At the air/water interface the simultaneous heat and mass transfer can be described by

$$dq = -i_v dm_w + h_c \left(T_w - T_a \right) dA_o$$

By employing equation (3.1.11) and noting that $dq = m_a di_a$ it follows that

$$m_a di_a = h_D \left(w_{asw} - w_{as} \right) i_v \, dA_o + h_c \left(T_w - T_a \right) dA_o \qquad (3.1.12)$$

The following supplementary equations can be used to simplify the equation above:

i) $c_{pm} = c_{pa} + w_{as} \, c_{pv} + \left(w_a - w_{as} \right) c_{pw}$ (3.1.13)

ii) $i_v = i_{vo} + c_{pv} T_w$ (3.1.14)

iii) $i_a = \left(c_{pa} + w_{as} c_{pv} \right) T_a + w_{as} \, i_{vo} + \left(w_a - w_{as} \right) c_{pw} T_a$ (3.1.15)

iv) $i_{asw} = \left(c_{pa} + w_{asw} c_{pv} \right) T_w + w_{asw} \, i_{vo}$ (3.1.16)

The last term in each of equations (3.1.13) and (3.1.15) constitutes a correction to take into account the amount of water in the form of mist in the saturated air.

Rewriting equation (3.1.12) and substituting (3.1.13) into it gives

$$m_a di_a = h_D dA_o \left[\left(w_{asw} - w_{as} \right) i_v + \left(\frac{h_c}{h_D \, c_{pm}} \right) \left\{ c_{pa} + w_{as} c_{pv} + \left(w_a - w_{as} \right) c_{pw} \right\} \left(T_w - T_a \right) \right]$$

By noting that from equation (3.1.15) that

$$\left[c_{pa} + w_{as}c_{pv} + \left(w_a - w_{as} \right) c_{pw} \right] T_a = i_a - w_{as} i_{vo}$$

and from equation (3.1.16) that

$$c_{pa} T_w = i_{asw} - w_{asw}\, c_{pv}\, T_w - w_{asw}\, i_{vo}$$

this can be rewritten as

$$m_a di_a = h_D dA_o \Bigg[\left(w_{asw} - w_{as} \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \Big(i_{asw} - w_{asw}c_{pv}\, T_w - w_{asw} i_{vo} + w_{as}c_{pv}\, T_w + \left(w_a - w_{as} \right) c_{pw}T_w - i_a + w_{as} i_{vo} \Big) \Bigg]$$

$$= h_D dA_o \Bigg[\left(w_{asw} - w_{as} \right) i_v + \left(\frac{h_c}{h_D\, c_{pm}} \right) \Big[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_{as} \right)\left(i_{vo} + c_{pv}T_w \right) + \left(w_a - w_{as} \right) c_{pw}T_w \Big] \Bigg]$$

$$= h_D dA_o \Bigg[\left(w_{asw} - w_{as} \right) i_v + \left[\left(\frac{h_c}{h_D\, c_{pm}} \right) - 1 \right] \Big[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_{as} \right) i_v + \left(w_a - w_{as} \right) c_{pw}\, T_w \Big] + \left(i_{asw} - i_a \right) - \left(w_{asw} - w_{as} \right) i_v + \left(w_a - w_{as} \right) c_{pw}\, T_w \Bigg]$$

$$= h_D dA_o \left[\left(i_{asw} - i_a \right) + \left[\left(\frac{h_c}{h_D\ c_{pm}} \right) - 1 \right] \left[\left(i_{asw} - i_a \right) \right.\right.$$

$$\left.\left. - \left(w_{asw} - w_{as} \right) i_v + \left(w_a - w_{as} \right) c_{pw}\ T_w \right] + \left(w_a - w_{as} \right) c_{pw} T_w \right]$$

$$\therefore\ di_a = \frac{h_D dA_o}{m_a} \left[\left(i_{asw} - i_a \right) + \left[\left(\frac{h_c}{h_D\ c_{pm}} \right) - 1 \right] \left[\left(i_{asw} - i_a \right) \right.\right.$$

$$\left.\left. - \left(w_{asw} - w_{as} \right) i_v \right] + \left(\frac{h_c}{h_D c_{pm}} \right) \left(w_a - w_{as} \right) c_{pw} T_w \right] \tag{3.1.17}$$

The heat transfer from the process fluid to the recirculating water is expressed by

$$dq = U_o \left(T_p - T_w \right) dA_o$$

where

$$\frac{1}{U_o} = \left[\left(\frac{1}{h_p} + \frac{1}{h_{fi}} \right) \frac{d_o}{d_i} + \frac{d_o \ln \left(d_o / d_i \right)}{2\ k_t} + \frac{1}{h_{fo}} + \frac{1}{h_w} \right]$$

The change in process water temperature can now be expressed as follows by noting that $dq = -m_p\ c_{pp}\ dT_p$

$$dT_p = - \frac{U_o}{m_p\ c_{pp}} \left(T_p - T_w \right) dA_o \tag{3.1.18}$$

The full system of five differential equations describing the operation of the given element in the case of non-saturated air is now given as equations (3.1.1), (3.1.2), (3.1.3), (3.1.10) and (3.1.18) and the five controlling equations for the case of saturated air are given by equations (3.1.1), (3.1.2), (3.1.11), (3.1.17) and (3.1.18).

3.1.2 Merkel analysis

The main assumptions made in the Merkel type analysis are
i) the evaporation of recirculating water is negligible and
ii) the Lewis factor is equal to unity.

The only variables employed in the Merkel analysis are T_w, T_p and i_a. Note that the Merkel analysis does not involve separate handling of the saturated air case.

By applying the two assumptions the controlling differential equations for the Merkel analysis are found from equations (3.1.2), (3.1.10) and (3.1.18) as

$$dT_w = \frac{1}{m_w c_{pw}} \left(- m_a di_a - m_p c_{pp} dT_p \right) \tag{3.1.19}$$

$$di_a = \frac{h_D}{m_a} \left(i_{asw} - i_a \right) dA_o \tag{3.1.20}$$

$$dT_p = - \frac{U_o}{m_p c_{pp}} \left(T_p - T_w \right) dA_o \tag{3.1.21}$$

3.1.3 Improved Merkel analysis

Singham [83SI1] described an extra equation to use with the Merkel equations to describe the air conditions more closely.

According to Singham the change in absolute humidity of the air in each element can be given as

$$dw_a = \left[\frac{w_{asw} - w_a}{i_{asw} - i_a} \right] \left[\frac{1}{1 - w_{asw}} \right] di_a$$

By substituting equation (3.1.20) into the equation above, it can be reduced to

$$dw_a = \frac{h_D \, dA_o}{m_a} \left[\frac{w_{asw} - w_a}{1 - w_{asw}} \right] \qquad (3.1.22)$$

The Merkel model is considerably improved by using equation (3.1.22) in conjunction with equations (3.1.19), (3.1.20) and (3.1.21). If the Singham equation is used with the three Merkel equations two air properties, humidity and enthalpy, are known at every position in the cooler. If two properties of air are known then any other property can be determined uniquely. This is of considerable importance in natural draft coolers and condensers.

If the Singham equation is not employed with the Merkel method the outlet air density has to be calculated after assuming that the air leaving the evaporative coil is saturated.

3.1.4 Simplified model

Consider the following two different evaporative cooler layouts.

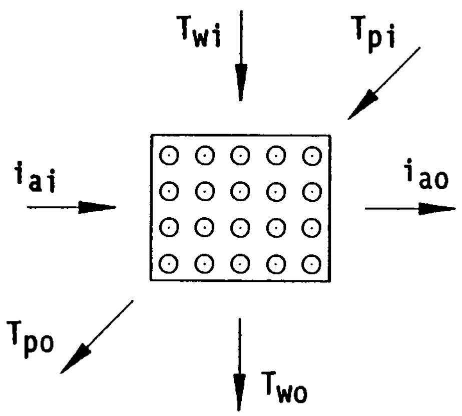

Figure 3.2)a) Cross-flow evaporative cooler layout.

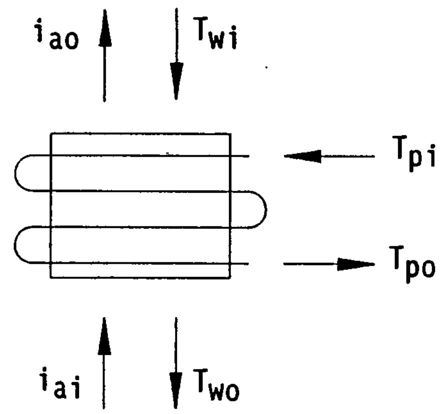

Figure 3.2)b) Counterflow evaporative cooler layout.

Assuming that the recirculating water temperature is constant throughout the cooler the cooler performance can be fully described (using the Merkel assumptions) by

$$m_p c_{pp} dT_p = U_o \left(T_p - T_{wm} \right) dA_o \qquad (3.1.23)$$

$$m_a di_a = h_D \left(i_{asw} - i_a \right) dA_o \qquad (3.1.24)$$

Rewriting equation (3.1.23) and integrating between T_{pi} and T_{po} gives

$$dA_o = \frac{m_p\ c_{pp}}{U_o} \left[\frac{dT_p}{T_p - T_{wm}} \right]$$

$$\therefore A_o = \frac{m_p\ c_{pp}}{U_o} \left[\ln \left(T_p - T_{wm} \right) \right] \Big|_{T_{po}}^{T_{pi}}$$

$$\therefore A_o = \frac{m_p\ c_{pp}}{U_o} \ln \left[\frac{T_{pi} - T_{wm}}{T_{po} - T_{wm}} \right] \qquad (3.1.25)$$

Rewriting equation (3.1.24) gives

$$dA_o = \frac{m_a}{h_D} \left[\frac{di_a}{\left(i_{asw} - i_a \right)} \right]$$

Integration between i_{ai} and i_{ao} gives

$$A_o = \frac{m_a}{h_D} \ln \left[\frac{i_{asw} - i_{ai}}{i_{asw} - i_{ao}} \right] \qquad (3.1.26)$$

From equations (3.1.25) and (3.1.26) it follows that

$$\frac{m_p c_{pp}}{U_o} \ln \left[\frac{T_{pi} - T_{wm}}{T_{po} - T_{wm}} \right] = \frac{m_a}{h_D} \ln \left[\frac{i_{asw} - i_{ai}}{i_{\cancel{aw}\,asw} - i_{ao}} \right] \qquad (3.1.27)$$

By solving equation (3.1.27) iteratively for T_{wm} [note that $i_{asw} = i_{as}$ (T_{wm})] and then using this value of T_{wm} that satisfies equation (3.1.27) in either equations (3.1.25) of (3.1.26), the required cooler surface area can be found.

This iterative procedure could be used for the rating of evaporative coolers as well, but the rating procedure is greatly simplified by rewriting the controlling equations as follows:

From equation (3.1.25) we have

$$T_{po} = T_{wm} + \left(T_{pi} - T_{wm} \right) e^{-NTU_p}$$

$$\text{with } NTU_p = \frac{A_o U_o}{m_p c_{pp}} \qquad (3.1.28)$$

and from equation (3.1.26) we have

$$i_{ao} = i_{asw} - \left(i_{asw} - i_{ai} \right) e^{-NTU_a}$$

$$\text{with } NTU_a = \frac{A_o\, h_D}{m_a} \tag{3.1.29}$$

For the whole cooler we have

$$q = m_a \left(i_{ao} - i_{ai} \right) = m_p\, c_{pp} \left(T_{pi} - T_{po} \right)$$

By substitution of the NTU relations into this equation we have

$$m_a \left(i_{asw} - \left(i_{asw} - i_{ai} \right) e^{-NTU_a} - i_{ai} \right) = m_p\, c_{pp} \left(T_{pi} - T_{wm} - \left(T_{pi} - T_{wm} \right) e^{-NTU_p} \right)$$

$$\therefore m_a \left(\left(i_{asw} - i_{ai} \right) \left(1 - e^{-NTU_a} \right) \right) = m_p\, c_{pp} \left[\left(T_{pi} - T_{wm} \right) \left(1 - e^{-NTU_p} \right) \right]$$

$$\therefore T_{wm} = T_{pi} - \frac{m_a \left(i_{asw} - i_{ai} \right) \left(1 - e^{-NTU_a} \right)}{m_p\, c_{pp} \left(1 - e^{-NTU_p} \right)} \tag{3.1.30}$$

Rating of a coil can now easily be done using equation (3.1.30) as follows; A value of T_{wm} is chosen and by using equation (3.1.30) the value of T_{wm} is corrected until T_{wm} converges to a fixed value. By now employing equations (3.1.28) and (3.1.29) the outlet conditions of the cooler can be found.

3.2 Basic theory for evaporative condensers

3.2.1 Exact analysis (Poppe model)

Consider the typical elements from a typical cross-flow and counterflow evaporative condenser in figures 3.3)a) and 3.3)b) respectively. As in section 3.1, the sign convention used results in the same equations for both the cross-flow and counterflow models.

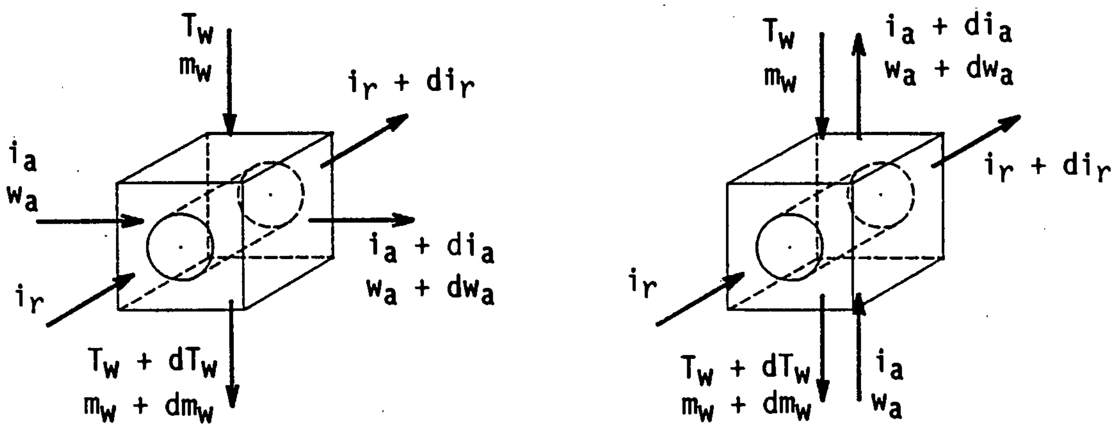

Figure 3.3)a) Control volume for a cross-flow evaporative condenser.

Figure 3.3)b) Control volume for a counterflow evaporative condenser.

The mass balance of the control volume gives

$$\left(1 + w_a\right) m_a + m_w + m_r = \left(1 + w_a + dw_a\right) m_a + m_w + dm_w + m_r$$

$$m_a dw_a + dm_w = 0$$

$$\therefore dw_a = - \frac{dm_w}{m_a} \qquad (3.2.1)$$

The energy balance for the control volume gives

$$m_a i_a + m_w c_{pw} T_w + m_r i_r =$$

$$\left(m_w + dm_w \right) c_{pw} \left(T_w + dT_w \right) + m_a \left(i_a + di_a \right) + m_r \left(i_r + di_r \right)$$

After simplification

$$m_a di_a + m_w c_{pw} dT_w + c_{pw} T_w dm_w + m_r di_r = 0$$

$$\therefore \quad dT_w = \frac{1}{m_w c_{pw}} \left(- m_a di_a - c_{pw} T_w \, dm_w - m_r \, di_r \right) \qquad (3.2.2)$$

Depending on whether the air is saturated or not, the controlling equation for the heat and mass transfer from the water film to the air is given by case 1 and case 2 respectively.

CASE 1 - Non-saturated moist air

The mass transfer from the water into the air is given by

$$dm_w = -h_D \left(w_{asw} - w_a \right) dA_o \qquad (3.2.3)$$

From the exact analysis given in section 3.1.1 the change of air enthalpy is given by

$$di_a = \frac{h_D \, dA_o}{m_a} \left[\left(i_{asw} - i_a \right) + \left(\frac{h_c}{h_D \, c_{pm}} - 1 \right) \cdot \left[\left(i_{asw} - i_a \right) - \left(w_{asw} - w_a \right) i_v \right] \right] \qquad (3.2.4)$$

The heat transfer from the condensing refrigerant to the recirculating water is given by

$$dq = U_o \left(T_r - T_w \right) dA_o \tag{3.2.5}$$

where

$$\frac{1}{U_o} = \left[\left(\frac{1}{h_r} + \frac{1}{h_{fi}} \right) \frac{d_o}{d_i} + \frac{d_o \ln \left(d_o / d_i \right)}{2\, k_t} + \frac{1}{h_{fo}} + \frac{1}{h_w} \right] \tag{3.2.6}$$

The change of refrigerant enthalpy can now be written as

$$di_r = - \frac{dq}{m_r}$$

By substituting equation (3.2.5) into the relation above, it follows that we have

$$di_r = - \frac{U_o}{m_r} \left(T_r - T_w \right) dA_o \tag{3.2.7}$$

The five equations (3.2.1), (3.2.2), (3.2.3), (3.2.4), and (3.2.7) fully describe the processes that take place in a single element of an evaporative condenser if the air is not saturated.

<u>CASE 2 - Saturated air</u>

The mass transfer from the water to the saturated air is given by

$$dm_w = -h_D \left(w_{asw} - w_{as} \right) dA_o \tag{3.2.8}$$

From section (3.1.1) the Poppe-type analysis for the case of saturated air results in

$$di_a = \frac{h_D\, dA}{m_a} \left[\left(i_{asw} - i_a \right) + \left(\frac{h_c}{h_D\, c_{pm}} - 1 \right) \left[\left(i_{asw} - i_a \right) \right.\right.$$

$$- \left(w_{asw} - w_{as} \right) i_v \Big] + \frac{h_c}{h_D\, c_{pm}} \left(w_a - w_{as} \right) c_{pw}\, T_w \Big] \qquad (3.2.9)$$

The complete system for the case of saturated inlet air is now given by equations (3.2.1), (3.2.2), (3.2.7), (3.2.8) and (3.2.9).

3.2.2 Merkel analysis

The main assumptions that need to be made to reduce the exact analysis to the Merkel analysis are

i) the evaporation of the recirculating water is negligible and

ii) the Lewis factor is equal to unity.

The Merkel-type analysis does not involve separate equations for the case of saturated air.

The governing equations of the Merkel analysis are given as

$$di_a = \frac{h_D\, dA_o}{m_a} \left(i_{asw} - i_a \right) \qquad (3.2.10)$$

$$dT_w = \frac{1}{m_w\, c_{pw}} \left(-m_a di_a - m_r di_r \right) \qquad (3.2.11)$$

$$di_r = - \frac{U_o}{m_r} \left(T_r - T_w \right) dA_o \qquad (3.2.12)$$

3.2.3 Improved Merkel analysis

As seen in section (3.1.3) a supplementary equation for describing the air conditions in a control volume has been proposed by Singham [83SI1]. This equation (3.1.22) holds without alteration for the application in an evaporative condenser, together with the three Merkel equations (3.2.10), (3.2.11) and (3.1.12).

3.2.4 Simplified Model

The simplified modelling approach considers the evaporative condensers as a single unit, using only the inlet and outlet values of the unit for the analysis. The layout of a typical evaporative condenser is shown in Figures 3.4)a) and 3.4)b).

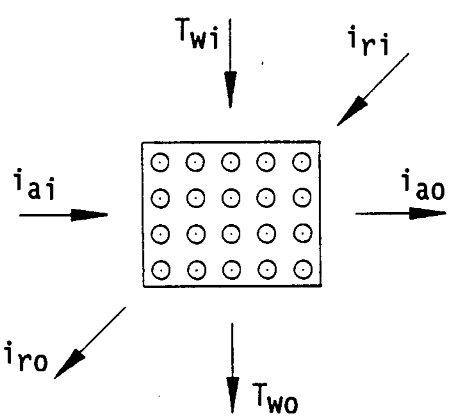

Figure 3.4)a) Cross-flow evaporative condenser layout.

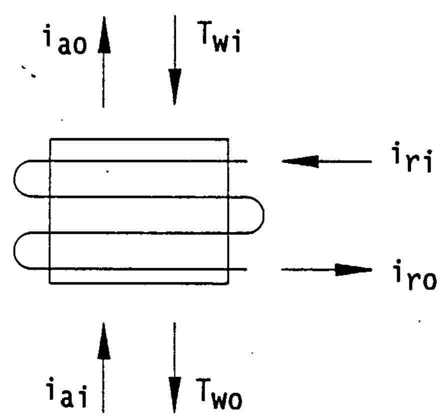

Figure 3.4)b) Counterflow evaporative condenser layout.

The load on the evaporative condenser will typically be specified as a given vapour massflow, an inlet vapour quality, x_i, and an outlet vapour quality, x_o.

$$q = m_a \left(i_{ao} - i_{ai} \right) = m_r \left(x_i - x_o \right) i_{fg}$$

If saturated vapour enter the condenser and the refrigerant leaves the condenser as a saturated liquid the condenser load is

$$q = m_r i_{fg}$$

By making the Merkel assumptions, the operation of an evaporative condenser can be described by the following two relations

$$m_a di_a = U_o \left(T_r - T_w \right) dA_o \tag{3.2.13}$$

$$m_a di_a = h_D \left(i_{asw} - i_a \right) dA_o \tag{3.2.14}$$

Assuming that the recirculating water is constant through the condenser, integration of equation (3.2.13) between the condenser inlet and outlet sides gives

$$A_o = \frac{m_a}{U_o} \left[\frac{i_{ao} - i_{ai}}{T_r - T_{wm}} \right] \tag{3.2.15}$$

and integration of equation (3.2.14) between the inlet and outlet sides result in

$$\ln \left[\frac{i_{asw} - i_{ai}}{i_{asw} - i_{ao}} \right] = \frac{h_D}{m_a} A_o \tag{3.2.16}$$

Substituting equation (3.2.15) into equation (3.2.16) gives

$$\ln \left[\frac{i_{asw} - i_{ai}}{i_{asw} - i_{ao}} \right] = \frac{h_D}{U_o} \left[\frac{i_{ao} - i_{ai}}{T_r - T_{wm}} \right] \tag{3.2.17}$$

By solving equation (3.2.17) iteratively for T_{wm} and then using the value of T_{wm} which satisfies equation (3.2.17) in equation (3.2.15), the required condenser area for the given load can be determined.

Equation (3.2.17) could be used for condenser rating by employing a complicated iterative procedure. A simpler approach for condenser rating can be found by rewriting equation (3.2.16) as follows

$$\left[\frac{i_{asw} - i_{ai}}{i_{asw} - i_{ao}} \right] = e^{-NTU_a}$$

where

$$NTU_a = \frac{h_D A_o}{m_a}$$

$$\therefore i_{ao} = i_{asw} - \left(i_{asw} - i_{ai} \right) e^{-NTU_a} \qquad (3.2.18)$$

From the energy balance of the condenser we have

$$q = m_a \left(i_{ao} - i_{ai} \right) = U_o \left(T_r - T_{wm} \right) A_o$$

$$\therefore i_{ao} = i_{ai} + \frac{U_o A_o}{m_a} \left(T_r - T_{wm} \right) \qquad (3.2.19)$$

By equating equations (3.2.18) and (3.2.19) we have

$$i_{asw} - \left(i_{asw} - i_{ai} \right) e^{-NTU_a} = i_{ai} + \frac{U_o A_o}{m_a} \left(T_r - T_{wm} \right)$$

$$\therefore \left(i_{asw} - i_{ai} \right) \left(1 - e^{-NTU_a} \right) = \frac{U_o A_o}{m_a} \left(T_r - T_{wm} \right)$$

$$\therefore T_{wm} = T_r - \frac{m_a \left(i_{asw} - i_{ai} \right) \left(1 - e^{-NTU_a} \right)}{U_o A_o} \qquad (3.2.20)$$

Rating of a condenser coil can now easily be done by choosing an initial value for T_{wm} and then utilizing equation (3.2.20) to correct the previous choice of T_{wm} until T_{wm} converges. The outlet enthalpy of the air can now be found by employing equations (3.2.18) or (3.2.19).

From the energy balance of the condenser the massflow of condensate condensed would be given by

$$m_r = \frac{m_a \left(i_{ao} - i_{ai} \right)}{i_{fg}} \tag{3.3.21}$$

It is interesting to note that equation (3.2.20) could easily be found using the ε-NTU design approach used for the rating of conventional heat exchangers. For heat exchanger with one fluid at a constant temperature the efficiency ε is given by

$$\varepsilon = 1 - e^{-NTU} \tag{3.2.22}$$

CHAPTER 4

HEAT/MASS TRANSFER AND PRESSURE DROP CORRELATIONS

Various correlations for the governing heat and mass transfer coefficients and for pressure drop across tube bundles were found in the literature. The majority of these correlations were based on experimental results, but a few analytical models were also proposed. In this chapter the various correlations for the required coefficients and pressure drops, which are relevant in the evaluation of evaporative coolers and condensers are summarised and graphically compared.

4.1 Film heat transfer coefficient

The heat transfer between the cooler or condenser tube and the recirculating water film is governed by the film heat transfer coefficient. Various investigators have determined this coefficient experimentally and analytically for vertical and horizontal tubes in evaporative coolers or condensers and in so called "film"-, "trickle"- or "trombone" coolers.

In a "film" cooler there is no airflow through the cooler to cool the water film flowing over the tubes as is the case in an evaporative cooler or condenser.

Parker and Treybal [61PA1] studied horizontal tube counterflow evaporative coolers and horizontal tube falling film coolers. According to Parker and Treybal the film heat transfer coefficient in an evaporative cooler or condenser is approximately 20 % less than in a film cooler, expressed mathematically as

$$\therefore h_w = 0{,}8\, h_{ff} \qquad (4.1.1)$$

Parker and Treybal correlated the film heat transfer coefficient in an evaporative cooler with 19 mm O.D. tubes as

heat transfer coef between tube and water

$$h_w = 704 \quad (1,3936 + 0,02214\ T_w) \left(\frac{\Gamma}{d_o} \right)^{1/3} \tag{4.1.2}$$

for

$$15,6°C < T_w < 71°C$$

and

$$1,36 < \frac{\Gamma}{d_o} < 3 \quad [\ kg/m^2s\ ]$$

<u>McAdams</u> [54Mc1] determined the following correlation for the film coefficient in a horizontal tube film cooler as,

$$h_{ff} = 3334,6 \left(\frac{\Gamma}{d_o} \right)^{1/3} \tag{4.1.3}$$

if

$$\frac{4\Gamma}{\mu_w} < 2100$$

By employing the conversion given by Parker and Treybal the correlation for an evaporative cooler or condenser would be

$$h_w = 2667,7 \left(\frac{\Gamma}{d_o} \right)^{1/3} \tag{4.1.4}$$

Mizushina et al.[67MI1] found the following correlation for the film heat transfer coefficient in a counterflow horizontal tube evaporative cooler

$$h_w = 2102{,}9 \left[\frac{\Gamma}{d_o} \right]^{1/3} \qquad (4.1.5)$$

with

$$0{,}195 < \frac{\Gamma}{d_o} < 5{,}556 \quad [kg/m^2s]$$

The correlation given by Mizushina et al. was obtained from test data using tube diameters of 12,7 mm, 19,05 mm, and 40,00 mm.

Conti [78CO1] and Owens [78WO1] evaluated the heat transfer to an evaporating ammonia film flow over horizontal tubes. They correlated the data with the following empirical relation

$$h_w = 2.2 \left[\frac{p_v}{d_o} \right]^{0,1} \left[\frac{g\ k_w}{\gamma_w} \right]^{1/3} \cdot \left[\frac{4\Gamma}{\mu_w} \right]^{-1/3} \qquad (4.1.6)$$

$$= f\left(d_o \right) \left[\frac{\Gamma}{d_o} \right]^{-1/3}$$

The exponent of the term (Γ/d_o) in the relation above differs considerably from the exponents found for this term in the other film coefficient correlations. This discrepancy might be attributed to the influence of surface tension which would play a much larger role in an ammonia film than it would in an water film.

Nakoryakov et al.[79NA1] conducted a series of experiments to determine the heat transfer coefficients between horizontal tubes and a falling film. Nakoryakov et al. correlated the data with the following relation.

$$Nu_{ff} = 1{,}06 \left[Re_w Pr_w \frac{2\,\delta}{\rho\, d_o} \right]^{1/3} \tag{4.1.7}$$

with

$$1{,}5 < \left[\frac{Re_w Pr_w \quad \delta}{d_o} \right]$$

By employing the definition of the film Reynolds number and the long established Nusselt equation for film thickness flowing down a vertical surface the correlation above can be rewritten as follows:

$$Re_w = \frac{\Gamma}{\mu_w}, \quad Pr_w = \frac{c_{pw}\,\mu_w}{k_w}, \quad \delta = \left[\frac{3\,\Gamma\,\mu_w}{g\,\rho_w^2} \right]^{1/3}$$

$$h_{ff} = 0{,}912 \quad k_w \;\delta^{-0{,}67} \left[\frac{Pr_w}{\mu_w} \right]^{1/3} \left[\frac{\Gamma}{d_o} \right]^{1/3} \tag{4.1.8}$$

This relation was derived for a film cooler, but by using the factor proposed by Parker and Treybal [61PA1] this relation can be rewritten for use in an evaporative cooler or condenser as follows

$$h_w = 0{,}735 \quad k_w \;\delta^{-0{,}67} \left[\frac{Pr_w}{\mu_w} \right]^{1/3} \left[\frac{\Gamma}{d_o} \right]^{1/3} \tag{4.1.9}$$

Rogers [81RO1] studied the flow and heat transfer characteristics of laminar falling films flowing over horizontal tubes using an analytical approach.

According to him the transition from laminar to turbulent flow of falling films occurs in the film Reynolds number range between 1000 and 2000. Rogers divided the flow regions over the tube in two distinct regions e.a. the development region and the developed region. By employing an integral method he determined local heat transfer coefficients with the following form

$$h_w = f\left[\frac{k_w}{d_o},\ \left(\frac{c_p\,\Gamma}{k_w}\right),\ \left(\frac{g\,\rho^2\,d^3}{\Gamma\,\mu_f}\right)\right] \qquad (4.1.10)$$

for both the developed and the developing regions. The actual determination of the mean film heat transfer coefficient is rather complicated since it requires a numerical integration procedure to determine the required coefficients for the two regions. The mean film heat transfer coefficient consists of a combination of the film coefficients of the two regions.

A graphical example given by Rogers shows that the results obtained using the analytical approach compares well with the simple empirical correlation given by McAdams [54Mc1].

Ganic and Mastanaiah [82GA1] gives an extensive survey of the literature on the hydrodynamics and heat transfer in falling films up to 1981. The subjects discussed include descriptions of the flow regimes, correlations for mean film thickness and conditions for the onset of turbulence and wavy flow. Correlations for the heat transfer to subcooled and saturated film flowing over horizontal and vertical tubes are compared with experimental data.

Leidenfrost and Korenic [82LE1] conducted a series of tests on an in-line horizontal tube evaporative condenser to determine the film heat transfer coefficient. They correlated their test data with the following relation

$$h_w = 2064,3 \left[\frac{\Gamma}{d_o} \right]^{0,252} \tag{4.1.11}$$

where

$$2 < \frac{\Gamma}{d_o} < 5,6 \quad [kg/m^2s]$$

Leidenfrost and Korenic used a tube diameter of 15,9 mm for all their tests.

<u>Dorokhov et al</u>.[83DO1] proposed a similar correlation as Nakoryakov et al. [79NA1] for the film heat transfer coefficient based on a series of experiments with a water and Li-Br mixture flowing as a film over horizontal tubes.

The correlation given by Dorokhov et al. states

$$Nu_{ff} = 1,03 \left[\left(\frac{\Gamma}{\mu_w} \right) Pr_w \left(\frac{2\,\delta}{\pi\, d_o} \right) \right]^{0,46} \tag{4.1.12}$$

with

$$1,6 < Re_w\, Pr_w \left(\frac{\delta}{d_o} \right) < 32$$

After simplification and using equation (4.1.1) this gives the film heat transfer coefficient in an evaporative cooler or condenser as

$$h_w = 0,67\; k_w\; \delta^{-0,54} \left[\frac{P_{rw}}{\mu_w} \right]^{0,46} \left[\frac{\Gamma}{d_o} \right]^{0,46} \tag{4.1.13}$$

where

$$\delta = \left[\frac{3 \, \Gamma \, \mu_w}{g \, \rho_w^2} \right]^{1/3} \tag{4.1.14}$$

Peterson [84PE3] used a similar approach to that of Parker and Treybal [61PA1] to determine the controlling coefficients in the operation of a counterflow evaporative condenser. It was found that the film heat transfer coefficient correlation given by Parker and Treybal fitted the new test data very well in the following extended range,

$$1,3 < \left[\frac{\Gamma}{d_o} \right] < 3,4 \quad [kg/m^2s]$$

Chyn and Bergles [87CH1] proposed a model for calculating the film heat transfer coefficient between a saturated water film and a horizontal tube. The model is based on three definite heat transfer regions: the jet impingement region, the thermal developing region and the fully developed region. The exponent of the term (Γ/d_o) in the proposed model is about -0,22 in the wavy-laminar flow region.

This model correlated experimental data well when the liquid flowed from one tube to the next as a sheet but not if the liquid feeds in columns and droplets, in which case the model underpredicts the heat transfer coefficient.

Discussion of film coefficient correlations

The correlations of Parker and Treybal [61PA1], Mizushina et al.[67MI1] and Leidenfrost and Korenic [82LE1] were all determined for evaporative coolers or condensers. Parker and Treybal [61PA1] however determined their correlation for the film coefficient by operating the evaporative

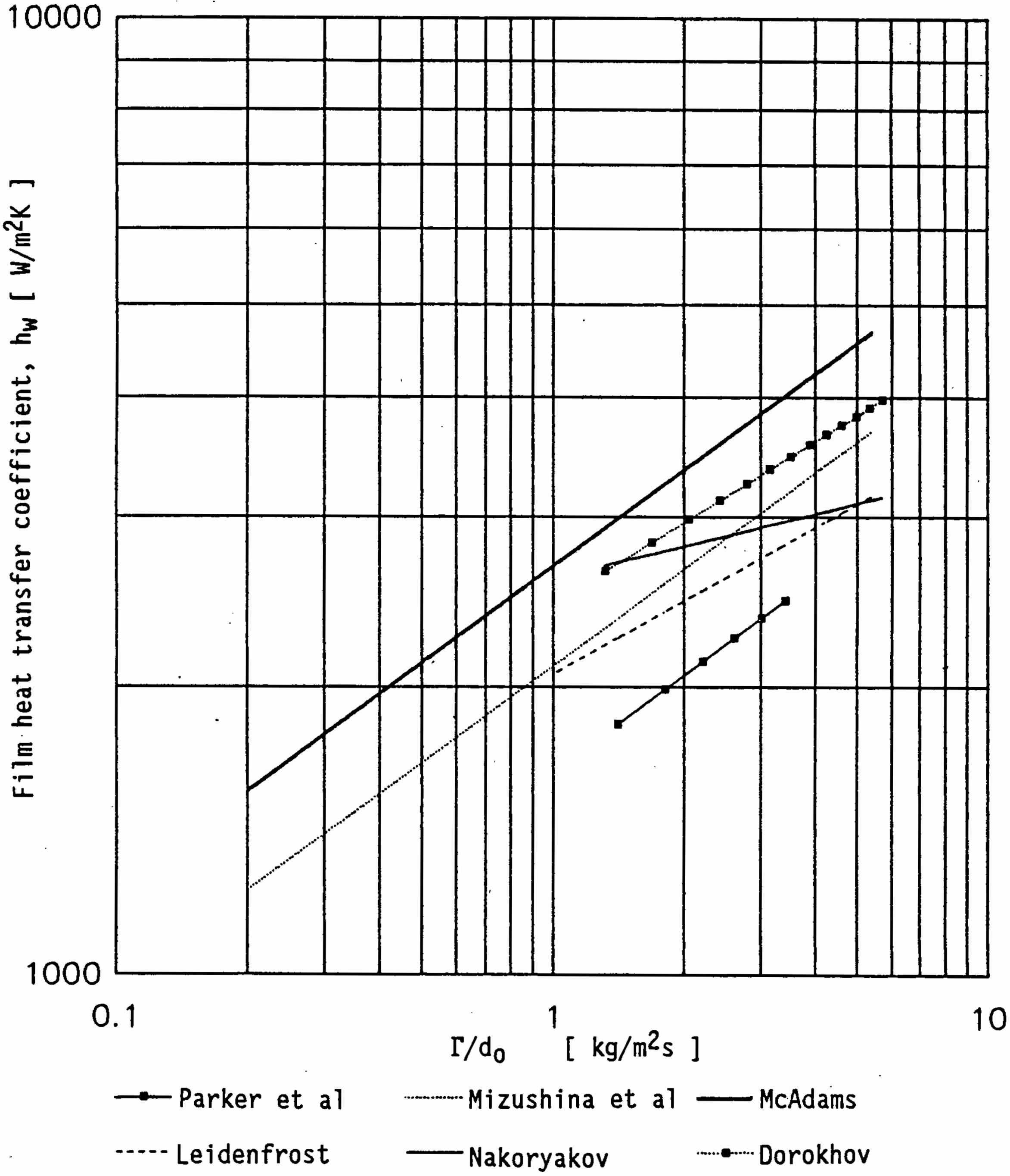

Figure 4.1 Correlations for the film heat transfer coefficient.

cooler as a "film" cooler i.e. without airflow. All three of these correlations express the film heat transfer as being a function of Γ^n where the exponent n has a positive value. The correlations by McAdams [54Mc1], Nakoryakov et al. [79NA1], Rogers [81RO1] and Dorokhov et al. [83DO1] also use a positive value for the exponent n. Since the film thickness increases with higher water massflow rates (higher Γ) the film coefficient should decrease with increasing water massflow, which implies that the exponent, n, must be negative.
This approach however neglects the effects of evaporation of the film and the entrance effects as the water strikes the tube from above. Chyn and Bergles [87CH1] found that their analytical model correlated experimental data well when the water flowed from the one tube to the next as a sheet but not when the water flows from one tube to the next in columns and drops. In evaporative coolers and condensers the water flow from one tube to the next is almost always as columns or drops which together with the fair degree of splashing and entrainment which occurs can explain the difference found between exponent n used by the various correlations.
In the evaluation of evaporative coolers and condensers it is advisable to use one of the correlations which was determined specifically for an evaporative cooler or condenser.

The film coefficient correlations are compared graphically in figure 4.1. The fluid properties needed for the graphical comparison was determined at 40°C and a tube diameter of 38,1 mm was assumed.

4.2 Mass transfer coefficient

Following the approach of Tasnadi [72TA1], Tezuka et al. [72TE1] and Perez-Blanco et al.[82PE1] the heat transfer from the process fluid or the condensing refrigerant to the airstream can be expressed as the product of an overall heat transfer coefficient and an enthalpy driving potential.
By evaluating the different terms in the overall heat transfer coefficient relation it is possible to determine the relative

contributions of each of the resistances to the flow of heat from the tubes to the airstream. The mass transfer coefficient term in the overall heat transfer relation contributes the largest resistance according to Perez-Blanco [82PE1].

Very little information on the mass transfer from tube bundles is available in the literature. Several investigators have determined the heat transfer coefficient governing the heat transfer from tube bundles to the fluid flowing through the bundle. By employing the analogy between heat- and mass transfer it is possible to estimate the mass transfer coefficient for a given geometrical layout if the heat transfer coefficient for the same geometrical layout is known.

The analogy usually gives good results for well defined layouts such as the heat- and mass transfer from a flat plate, but in the case of the flow of air through a wet surface tube bank the interfacial area between the air and the water film would not be the same as the outside surface of the tubes, because of the falling films and drops. The heat- and mass transfer analogy would thus fail in the case of the wet surface tube bundle, because of the non-similarity of the wet- and dry surface areas.

Parker and Treybal [61PA1] conducted a series tests on a counterflow evaporative cooler. The cooler tested consisted of a bank of sixty 19,05 mm O.D. tubes on a 2 x d_0 triangular pitch, 6 tubes wide and 10 tube rows deep in the direction of the airflow. Based on the two film theory of Treybal [55TR1] the mass transfer coefficient was assumed to be of the following form

$$h_D = \left[\frac{1}{h_{Di}} + \frac{b}{h_L} \right]^{-1} \qquad (4.2.1)$$

where

$$b = \left(\frac{di_{asw}}{dT_w} \right) \tag{4.2.2}$$

The coefficient h_L is the heat transfer coefficient between the recirculating water and the air/water interface. Parker and Treybal approximated this coefficient by

$$h_L = 11\ 360 \quad [W/m^2\ K]$$

The mass transfer coefficient h_{Di} was correlated as

$$h_{Di} = 49,35 \times 10^{-3} \left[\left(1 + w_a \right) G_{max} \right]^{0,905} \tag{4.2.3}$$

The value of the slope b can be determined through differentiation of the curve for air saturation enthalpy as a function of temperature given by Stoecker and Jones [84ST1] or graphically from a psychrometric chart.

According to Stoecker and Jones the air saturation enthalpy at sea level is given by

$$i_{as} = \left(4,7926 + 2,568\ T - 0,029834\ T^2 + 0,0016657\ T^3 \right) \times 10^3 \tag{4.2.4}$$

Through differentation of equation (4.2.4) it follows that

$$\frac{di_{asw}}{dT_w} = b = \left(2,568 - 0,059668\ T_w + 0,0049971\ T_w^2 \right) \times 10^3 \tag{4.2.5}$$

The mass transfer coefficient correlated by Parker and Treybal can now be given as

$$h_D = \left(\frac{1}{49,35 \times 10^{-3} \left(\left(1 + w_a \right) G_{max} \right)^{0,905}} + \frac{b}{11\ 360} \right)^{-1} \tag{4.2.6}$$

in the ranges $0{,}68 < G_{max} < 5{,}02$ [kg/m²s]

$1{,}36 < \Gamma/d_o < 3$ [kg/m²s]

Peterson [84PE3] followed the work of Parker and Treybal [61PA1] closely in determining the mass transfer coefficient in a counterflow evaporative condenser. The condenser coil studied by Peterson consisted of a bundle of tubes, six rows deep and 33 rows wide. The tubes were spaced in a triangular array with the following dimensions:

d_i = 27 [mm]
d_o = 24,5 [mm]
P_l = 64 [mm]
P_t = 81,6 [mm]

The mass transfer coefficient determined by Peterson was correlated as

$$h_D = \left\{ \frac{1}{44{,}3 \times 10^{-3} \left[\left(1 + w_a \right) G_{max} \right]^{0{,}905}} + \frac{b}{11360} \right\}^{-1} \qquad (4.2.7)$$

Peterson assumed throughout that the air saturation enthalpy at a level of 1 700 m above sea level is given by

$$i_{as} = 4593 T_w - 2956{,}3 \quad \text{[J/kg]}$$

$$\therefore b = 4\,593 \quad \text{[J/kgK]} \qquad (4.2.8)$$

The correlation given by Peterson holds for the following flow ranges

$6{,}3 < G_{max} < 9{,}6$ [kg/m²s]

$1{,}3 < \frac{\Gamma}{d_o} < 3{,}4$ [kg/m²s]

Peterson found that the correlation for the mass transfer coefficient given by Parker and Treybal [61PA1] holds with good approximation in the range,

$$0,68 < G_{max} < 9,6 \quad [kg/m^2s]$$

Mizushina et al.[67MI1] conducted a series of tests on a counterflow evaporative cooler using three different tube bundles with tube diameters of 12,7 [mm], 19,05 [mm] and 40 [mm] respectively. The tubes were spaced in a $2 \times d_o$ triangular array with either eight or twelve tube rows along the path of the airflow. The following volumetric correlation for the mass transfer coefficient was determined by Mizushina et al. to fit the test data.

$$h_D\, a = 5,0278 \times 10^{-8} \left(Re_a \right)^{0,9} \left(Re_w \right)^{0,15} \left(d_o \right)^{-2,6} \qquad (4.2.9)$$

where

$$Re_a = \frac{\rho_a}{\mu_a} G_{max}$$

$$Re_w = \frac{4\,\Gamma}{\mu_w}$$

with

$$1\,500 < Re_a < 8\,000$$

$$0,195 < \frac{\Gamma}{d_o} < 5,6 \text{ kg/m}^2\text{s}$$

The interfacial area per unit volume of a tube bundle in a 2 x do array can be expressed as

$$a = \frac{\pi\, d_o}{\left(2\, d_o \right) \left(\sqrt{3}\, d_o \right)} = \frac{\pi}{2 \sqrt{3}\, d_o} = \frac{0,9069}{d_o} \qquad (4.2.10)$$

The mass transfer coefficient correlation can be rewritten by employing the relation above, as

$$h_D = 5{,}544 \times 10^{-8} \left(Re_a \right)^{0{,}9} \left(Re_w \right)^{0{,}15} \left(d_o \right)^{-1{,}6} \qquad (4.2.11)$$

<u>Tezuka et al</u>.[76TE1] determined a correlation for the overall heat transfer coefficient in a counterflow evaporative cooler. The overall transfer coefficient approach states that

$$dq = h_{Do} \left(i_{asp} - i_a \right) dA \qquad (4.2.12)$$

where

$$h_{Do} = \left[\frac{1}{h_D} + \frac{b}{U_o} \right]^{-1}$$

and

$$b = \frac{di_{as}}{dT_p}$$

Tezuka et al. evaluated five different test sections with different diameters and tube configurations. By defining an effective diameter as

$$d_e = \frac{4 \times \text{flow area}}{\text{wetted perimeter}}$$

$$= \frac{2}{\pi d_o} \left(2 P_l \right) \left(P_t \right) - d_o$$

$$= \frac{4 P_t P_l}{\pi d_o} - d_o \qquad (4.2.13)$$

the overall transfer coefficient data can be expressed by one single correlation

$$h_{Do}a = 1,55 \left(d_e \; d_i \right)^{-0,3} \left(\frac{m_a}{A_{fr}} \right)^{0,45} \left(\frac{m_p}{A_{fr}} \right)^{0,3} \cdot \left(\frac{m_w}{A_{fr}} \right)^{0,25} \left(T_p \right)^{-0,75} \qquad (4.2.14)$$

For a tube bundle with tubes spaced in a 2 x d_0 triangular array the mass transfer coefficient can be expressed as

$$h_D = \left(\frac{1}{h_{Do}} - \frac{b}{U_o} \right)^{-1} \qquad (4.2.15)$$

where

$$U_o = \left[\left(\frac{1}{h_p} + \frac{1}{h_{fi}} \right) \frac{d_o}{d_i} + \frac{d_o \ln \left(d_o / d_i \right)}{2 \, k_t} + \frac{1}{h_{fo}} + \frac{1}{h_w} \right]^{-1} \qquad (4.2.16)$$

and

$$h_{Do} = 1,1828 \left(\frac{d_o^{0,7}}{d_i^{0,3}} \right) \left(\frac{m_a}{A_{fr}} \right)^{0,45} \left(\frac{m_p}{A_{fr}} \right)^{0,3} \left(\frac{m_w}{A_{fr}} \right)^{0,25} T_p^{-0,75} \qquad (4.2.17)$$

in the ranges

$$1 < \left(\frac{m_p}{A_{fr}} \right) < 2,22 \; [kg/m^2s]$$

$$1 < \left(\frac{m_p}{A_{fr}} \right) < 4,2 \quad [kg/m^2s]$$

$$1 < \left(\frac{m_w}{A_{fr}} \right) < 4,2 \quad [kg/m^2s]$$

and $30° < T_p < 50°C$.

Rana et al.[81RA1], [86RA1] and [87RA1] experimentally investigated the mass transfer coefficient which governs the heat transfer in a counterflow evaporative cooler or condenser. By using the heat and mass transfer analogy and a Lewis factor of 0,92 they determined the theoretical mass transfer coefficient from

$$h_{D,theo} = \frac{h_c}{0,92\ c_{pm}} \tag{4.2.18}$$

The predicted theoretical mass transfer coefficients were found to vary quite significantly from the experimentally determined values. In order to obtain a useful correlation for mass transfer coefficient Rana defined the following ratio,

$$R_R = \frac{h_D}{h_{D,\ theo}} \tag{4.2.19}$$

Various studies were conducted by Rana et al.[81RA1], [86RA1] and [87RA1] to find correlations for the ratio R_R. The experimental work was carried out on various counterflow evaporative cooler layouts, including a single tube unit. According to Rana [87RA1] single tube correlations developed by Rana [86RA1] overpredicts the mass transfer coefficients by between 200 and 500 %.
Rana et al.[81RA1] determined the following correlation for a full coil test unit

$$R_R = 1,76 \left(Re_p \right)^{0,22} \left(Re_w \right)^{0,39} \left(Re_a \right)^{-0,75} \tag{4.2.20}$$

where

$$Re_p = \frac{\rho_p \, v_p \, d_i}{\mu_p} ,$$

$$Re_w = \frac{4 \, \Gamma}{\mu_w} ,$$

$$Re_a = \frac{\rho_a \, v_{a,max} \, d_o}{\mu_a}$$

For a row of tubes in a counterflow evaporative cooler Rana [87RA1] correlated the ratio of experimental to theoretical mass transfer, R_R, as

$$R_R = 1{,}7838 \left(EP \right)^{0{,}3985} \left(Re_w \right)^{0{,}3765} \left(Re_a \right)^{-0{,}4114} \qquad (4.2.21)$$

where

$$EP = \frac{\Delta i}{i_{fg}}$$

and

$$\Delta i = \frac{\left(i_{as,wall,\,i} - i_{ai} \right) - \left(i_{as,wall,o} - i_{ao} \right)}{\ln \left(\dfrac{i_{as,wall,i} - i_{ai}}{i_{as,wall,o} - i_{ao}} \right)}$$

in the ranges

$0{,}0544 \le EP \le 0{,}1971$
$41{,}9 \le Re_w \le 294{,}3$
$692 \le Re_a \le 2764$

Rana [87RA1] used the following correlation to determine the convective heat transfer coefficient from the dry tube bundle

$$Nu_c = \frac{h_c\, d_o}{k_a} = 0{,}242 \left(Re_a \right)^{0,628} \tag{4.2.22}$$

Various other investigators determined correlations for the convective heat transfer coefficient in dry tube bundles e.g. Zukauskas [74ZH1] or Grimison [37GR1].

Discussion of mass transfer coefficient correlations

The mass transfer coefficient correlations given by Parker and Treybal [61PA1], Mizushina et al.[67MI1], Tezuka et al.[76TE1], Peterson [84PE3] and Rana et al. [87RA1] are compared graphically in figures 4.2 and 4.3 for tube diameters of 19,05 mm and 38,1 mm respectively. The fluid properties were evaluated at 35°C and a water flow rate of Γ = 300/3600 kg/ms was assumed.

The correlations of Parker and Treybal [61PA1], Mizushina et al.[67MI1] and Peterson [84PE3] compare well for a tube diameter of 19,05 mm. At the larger tube size the correlation by Mizushina et al. [67MI1] predicts values which are much lower than the predictions by Parker and Treybal [61PA1] and Peterson [84PE3]. It was noted by Finlay and McMillan [74FI1] that the mass transfer correlation given by Parker and Treybal [61PA1] gives mass transfer coefficients which are higher than those predicted by the correlation of Mizushina et al.[67MI1] but that the film heat transfer coefficients given by the Parker and Treybal [61PA1] correlation are lower than those found by Mizushina et al. [67MI1]. Finlay and McMillan [74FI1] found that the models of Parker and Treybal [61PA1] and Mizushina et al.[67MI1] are in good agreement if each model uses its own correlations for the film heat transfer coefficient and the mass transfer coefficients.

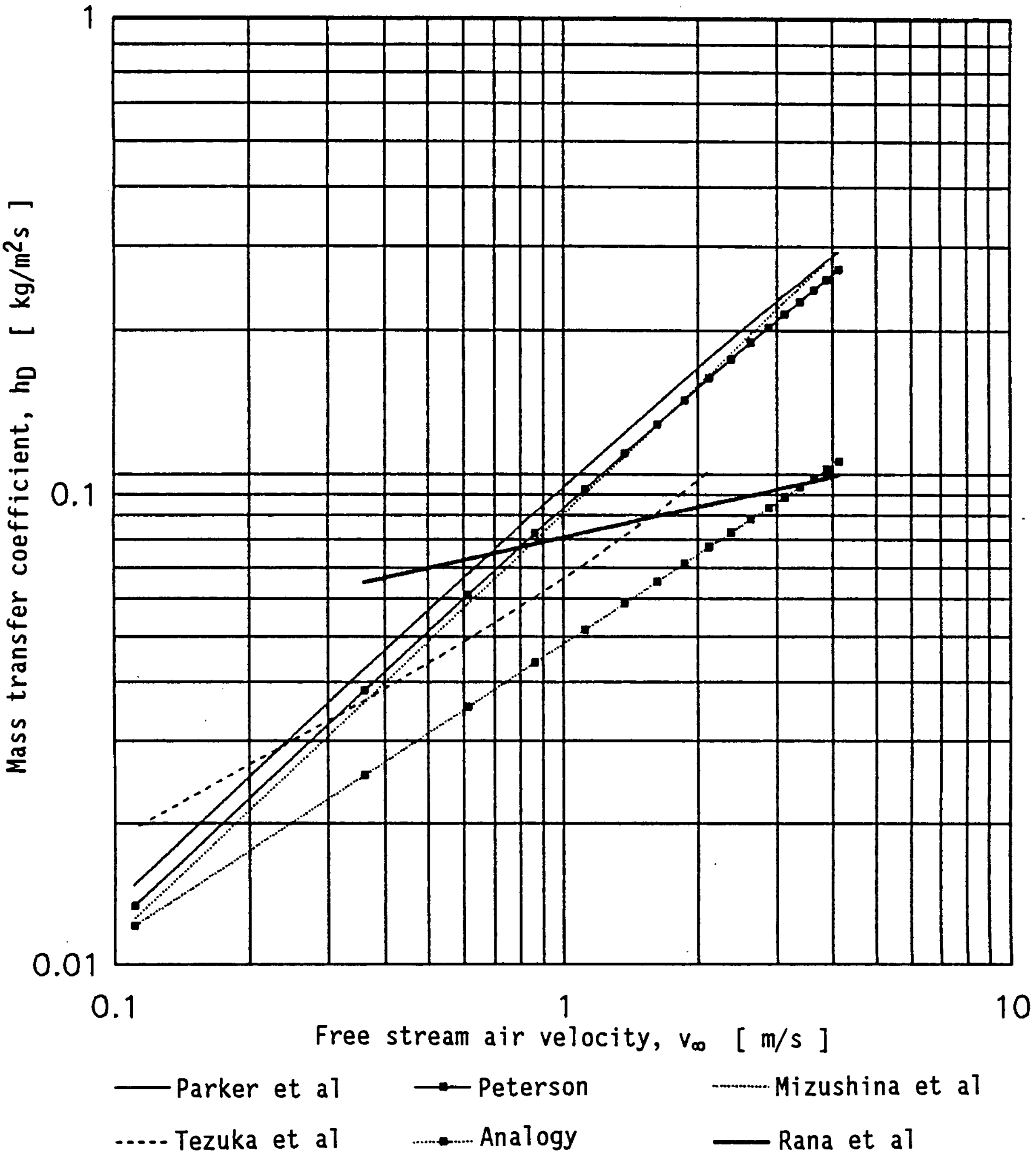

Figure 4.2 Correlations for the mass transfer coefficient, d_o = 19,05 mm.

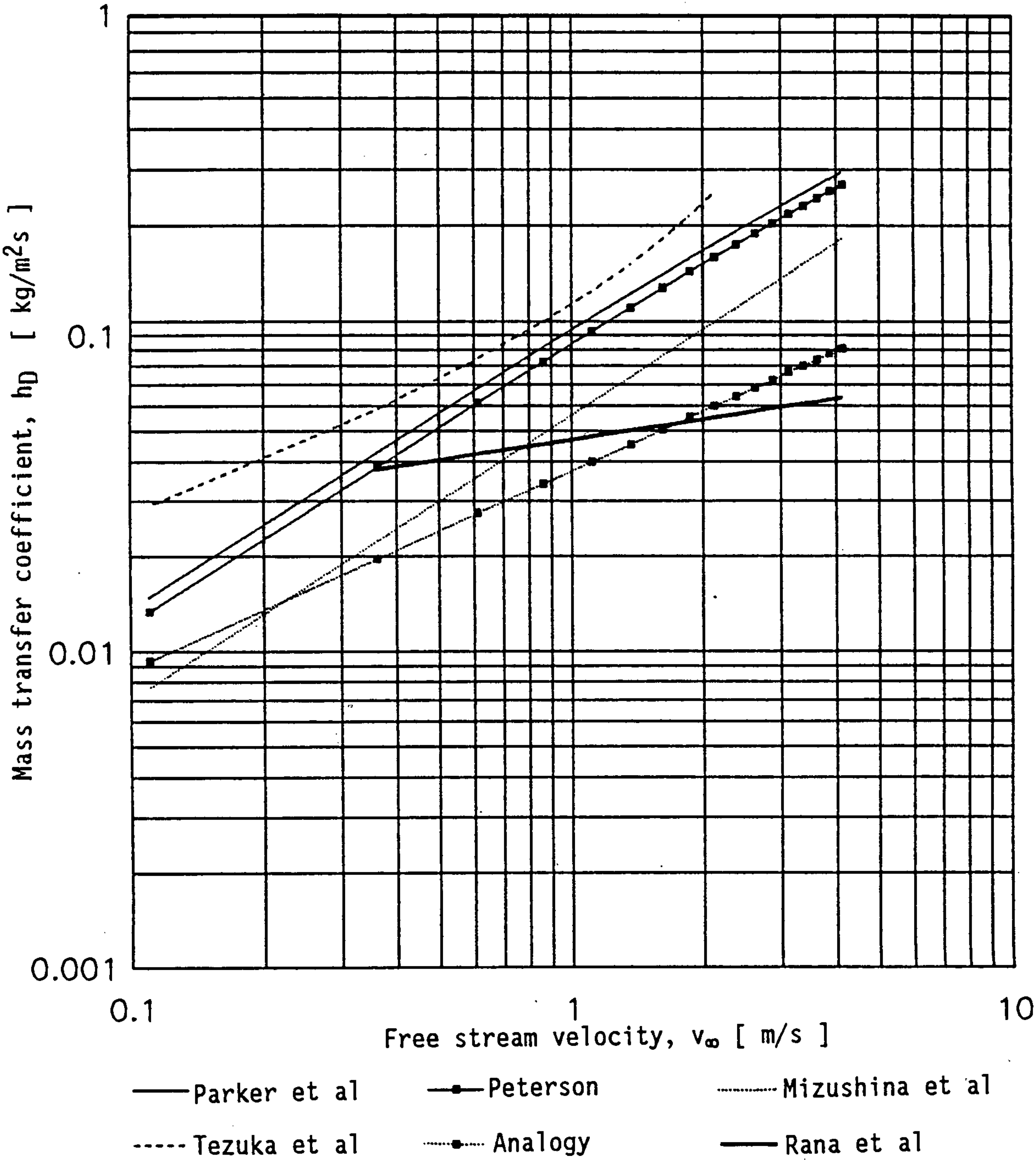

Figure 4.3 Correlations for the mass transfer coefficient, d_o = 38,1 mm.

The use of the correlations by Mizushina et al. [67MI1] are advisable since the mass transfer coefficient correlation by Mizushina et al. [67MI1] covers a much wider range of conditions and the effect of recirculating water and tube diameter are taken into account.

The correlation of Tezuka et al. [76TE1] is rather dubious since it does not give results which compares well with the other correlations. It is also illogical that the overall mass transfer should depend on the inner diameter of the tube. Similar criticism can be raised against the correlation for the mass transfer coefficient given by Rana et al.[81RA1] since this correlation gives the mass transfer coefficient, amongst other parameters, as a function of the process fluid Reynolds number.

The mass transfer coefficient correlation given by Rana et al. [87RA1] is rather cumbersome to use for cooler (or condenser) rating since the correlation requires the outlet conditions of the cooler to be known in order to determine the so called "enthalpy potential". Even if this correlation is to be used for design with known outlet conditions the tube wall temperatures has to be determined in order to evaluate the mass transfer coefficient.

4.3 Pressure drop across horizontal tube bundles in cross-flow and counterflow

The pressure drop across the tube bundle of an evaporative cooler or condenser of importance in the estimation of the required fan size in the case of a mechanical draft cooler or condenser and it is of major importance in the determination of the air massflow rate through the cooler or condenser in a natural draft application.

Various researchers have studied the single phase pressure drop across tube bundles. Appendix D gives an overview of the available correlations for single phase pressure drop across a tube bundle.

In an evaporative cooler or condenser the pressure drop calculations are complicated by the presence of the recirculating water flow. The airflow in an evaporative cooler and condenser can either be horizontal (cross-flow) or vertical upwards (counterflow) or vertically downwards (concurrent). The recirculating water normally flows downwards under the influence of gravity only.

The pressure drop characteristics of a wet surface evaporative cooler or condenser depends on various factors including the air massflow rate, recirculating water massflow rate, average temperature of recirculating water, tube array configuration etc.

Diehl [57DI1] proposed a method to calculate the two phase pressure drop across a tube bundle with the air and the recirculating water flowing concurrent by downwards through the tube bundle (downflow) and he proposed two graphical correlations for pressure drop across in-line tube banks and across staggered tube banks.

Diehl and Unruh [58DI1] tested various tube bundles to determine two-phase pressure drop correlations for different tube layouts. Graphical correlations were presented for staggered tubes with a 45° layout and 60° layout as well as for in-line tubes. They found that the correlations for the in-line tube bundle and the staggered bundle with the 60° triangular layout were the same. The pressure drop for the tube bundle with the 45° staggered layout was found to higher than that for the other two layouts.

Simple regression analysis of the graphical correlations yielded the following simple-to-use equations to determine the two phase pressure drop across a horizontal tube bundle.
For cross-flow across banks of tubes spaced in a 60° staggered layout or an in-line configuration the correlations are

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = 52{,}167\ \zeta^3 - 26{,}677\ \zeta^2 + 2{,}788\ \zeta + 1{,}00985 \qquad (4.3.1)$$

where

$$\zeta = LVF \frac{\rho_w}{\rho_a} = \frac{m_w}{m_a + m_w (\rho_a/\rho_w)} \qquad (4.3.2)$$

in the range $0 \leq \zeta \leq 0,25$ and

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = \frac{1,33\ 035}{1 + \zeta} + 0,020016\ \zeta\ -0,257908 \qquad (4.3.3)$$

for the range $0,25 \leq \zeta \leq 10$.

For cross-flow across banks of tubes spaced in a staggered 45° layout the following regression curves were found

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = 152,961\ \zeta^3 - 67,9895\ \zeta^2 + 7,274\ \zeta + 1,02375 \qquad (4.3.4)$$

where $0 \leq \zeta \leq 0,25$ and

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = \left(\frac{1,327}{1 + \zeta}\right) - 0,0795 + 0,002888\ \zeta \qquad (4.3.5)$$

in the range $0,25 \leq \zeta \leq 10$

The counterflow pressure drop across staggered tube bundles were correlated by Diehl and Unruh using the following parameter,

$$\psi = \frac{LVF}{\left(\rho_a/\rho_w\right)\left(Re_a^*\right)^{0,5}}$$

$$= \frac{m_w}{\left[m_a + m_w\left(\rho_a/\rho_w\right)\right]\left(Re_a^*\right)^{0,5}} \qquad (4.3.6)$$

If $0 < \psi < 0{,}007$ the correlation gives

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = 1{,}370042 + 44591{,}59\ \psi - \left[\frac{0{,}0000369198}{0{,}0001 + \psi} \right]$$

$$- 103378{,}776 \log_{10} \left[1 + \psi \right] \qquad (4.3.7)$$

and if $0{,}007 < \psi < 1{,}0$ the data was correlated by

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = - 0{,}00376946 + 0{,}0087965111\ \psi + \left[\frac{0{,}00261965}{0{,}001 + \psi} \right]$$

$$- 0{,}0052407713\ \psi^{2} \qquad (4.3.8)$$

Wallis [69WA1] presented a simple theoretical equation based on the homogeneous flow model to determine the two phase pressure drop for horizontal cross-flow through a tube bank with a staggered tube layout.

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = y + \left[1 - y \right] \left[\frac{\rho_a}{\rho_w} \right] \qquad (4.3.9)$$

where

$$y = \frac{m_a}{m_a + m_w} \qquad (4.3.10)$$

Collier [72CO1] rewrote the model of Wallis [69WA1] and compared the result with the data given by Diehl and Unruh [57DI1], [58DI1]. Collier adapted the Wallis model as follows

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = \frac{1}{1 + \zeta\ (1 - (\rho_a/\rho_w))} \approx \frac{1}{1+\zeta} \tag{4.3.11}$$

where

$$\zeta = \mathrm{LVF}\ \frac{\rho_w}{\rho_a} = \frac{m_w}{m_a + m_w\ (\rho_a/\rho_w)} \tag{4.3.12}$$

Grant and Chisholm [79GR1] conducted a study on the two phase pressure drop through the shell side of a segmentally baffled shell and tube heat exchanger. The correlation presented is of the following form

$$\frac{\Delta p_{tp}}{\Delta p_w^*} = 1 + \left(\Gamma_G^2 - 1 \right) \left[B\ y^{(2-n)}\ (\ 1-y\ )^{(2-n)/2} + y^{2-n} \right] \tag{4.3.13}$$

where

$$\Gamma_G = \left(\frac{\Delta p_a^*}{\Delta p_w^*} \right)^{1/2} \tag{4.3.14}$$

The coefficient n is the exponent in the Blasius type single phase fluid friction equation as given by

$$f = \frac{c_1}{Re^n} \tag{4.3.15}$$

Grant and Chisholm uses the value of $n = 0{,}46$ in the cross-flow pressure drop correlation. The correlation can consequently be simplified as

$$\frac{\Delta p_{tp}}{\Delta p_w^*} = 1 + \left[\left(\frac{\Delta p_a^*}{\Delta p_w^*} \right) - 1 \right] \left(B\ y^{0,77}\ (1-y)^{0,77} + y^{1,54} \right) \tag{4.3.16}$$

For the flow regimes usually encountered in evaporative coolers or condensers the following values of B are proposed:

B = 1 for vertical up-and-down flow and
B = 0,75 for horizontal side-to-side flow.

Grant and Chisholm reports that the correlation matches the data of Diehl and Unruh [57DI1] and [58DI1] to within 2 percent for $y \leq 0,6$ and

$$B = 0,75 + 3,5\, y^{10} \tag{4.3.17}$$

<u>Schrage et al</u>.[87SC1] and [88SC1] measured void fractions and pressure drop in two phase vertical cross-flow in a horizontal tube bundle. From the experimental data correlations for the void fraction and two phase friction multiplier were developed. Ishihara et al.[77IS1] first proposed the use of a Martinelli type multiplier to determine the two phase pressure drop across a horizontal tube bundle. For spray liquid flows as encountered in an evaporative cooler or condenser Schrage et al. proposed the following correlation,

$$\Delta p_{tp} = \phi_w^2\, \Delta p_w \tag{4.3.18}$$

where

$$\phi_w^2 = 1 + \frac{C}{X_{tt}} + \frac{0,205}{X_{tt}^2} \tag{4.3.19}$$

with

$$C = 1180\, G_{max}^{-1,5} \ln X_{tt} + 3,87\, G_{max}^{0,207} \tag{4.3.20}$$

and

$$X_{tt}^2 = \left[\frac{1-y}{y}\right]^{1,8} \frac{\rho_a}{\rho_w} \left[\frac{\mu_w}{\mu_a}\right]^{0,2} \tag{4.3.21}$$

This correlation holds only for $G_{max} \geq 43$ kg/m^{2}s. Schrage suggested that the correlation of Ishihara et al.[77IS1] be used if $G_{max} < 43$ kg/m^{2}s. The Ishihara correlation gives the two phase pressure drop as

$$\Delta p_{tp} = \phi_w^2 \, \Delta p_w \tag{4.3.22}$$

where

$$\phi_w^2 = 1 + \frac{8}{X_{tt}} + \frac{1}{X_{tt}^2} \tag{4.3.23}$$

Very little pressure drop data measured on an actual evaporative cooler or condenser have been supplied in the literature. Two investigators reported pressure drop data for counterflow evaporative coolers or condensers while no data has been found on horizontal cross-flow pressure drop across an evaporative cooler or condenser.

Tezuka et al.[76TE1] correlated the pressure drop across five different counterflow evaporative cooler coils using a correlation of the form

$$\frac{\Delta p_{tp}}{z} = 66,034 \times 10^6 \, C_1 \left[\frac{m_w}{A_{fr}}\right]^{0,32} \left[\frac{m_a}{A_{fr}}\right]^{1,6} \tag{4.3.24}$$

A different C_1 value was proposed for each of the five coils tested.

The following table gives the C_1 value for each of the coils tested:

Coil	C_1	d_o[mm]	P_l/d_o	P_t/d_o
A	$1,1 \times 10^{-7}$	27,2	1,65	2,30
B	$1,97 \times 10^{-7}$	34	1,44	2,18
C	$1,91 \times 10^{-7}$	42,7	1,17	2,25
D	$0,84 \times 10^{-7}$	42,7	1,17	2,93
E	$1,15 \times 10^{-7}$	42,7	1,12	2,08

Leidenfrost and Korenic [82LE1] reported that for the in-line evaporative condenser tested the pressure drop increase due to the recirculating water at the lowest air massflow was between 24% and 62% when compared to the dry operation of the coil. At the maximum air massflow rate the pressure drop increase due to the recirculating water flow was only between 12% and 18% more than the corresponding pressure drop across the dry tube bundle.

Discussion of pressure drop correlations

Many of the two phase pressure drop correlations require calculation of single phase pressure drop in order to use the two phase correlation. Single phase pressure drop across bundle of horizontal tubes have been extensively studied by many authors. Refer to Appendix D for a survey of the available single phase pressure drop correlations.

The correlation by Gaddis and Gnielinski [85GA1] is very comprehensive but its complexity does not allow fast calculations. The correlations of Gunter and Shaw [45GU1] and Jakób [38JA1] are easy to use and they are normally accurate enough for design purposes.

The available correlations for two phase pressure drop across a horizontal tube bank in cross-flow and counterflow are shown graphically in figures 4.4 and 4.5.

None of these correlations except that of Tezuka et al. [76TE1] was developed from tests on evaporative coolers or condensers and it is therefore advisable to use the more conservative correlations for design purposes.

It can also be seen from figure 4.5 that the two phase pressure drop prediction by Diehl and Unruh [58DI1] drops below the dry (air only) pressure drop. The correlation by Diehl and Unruh [58DI1] holds for concurrent flows which explains the low pressure drop at high water loading.

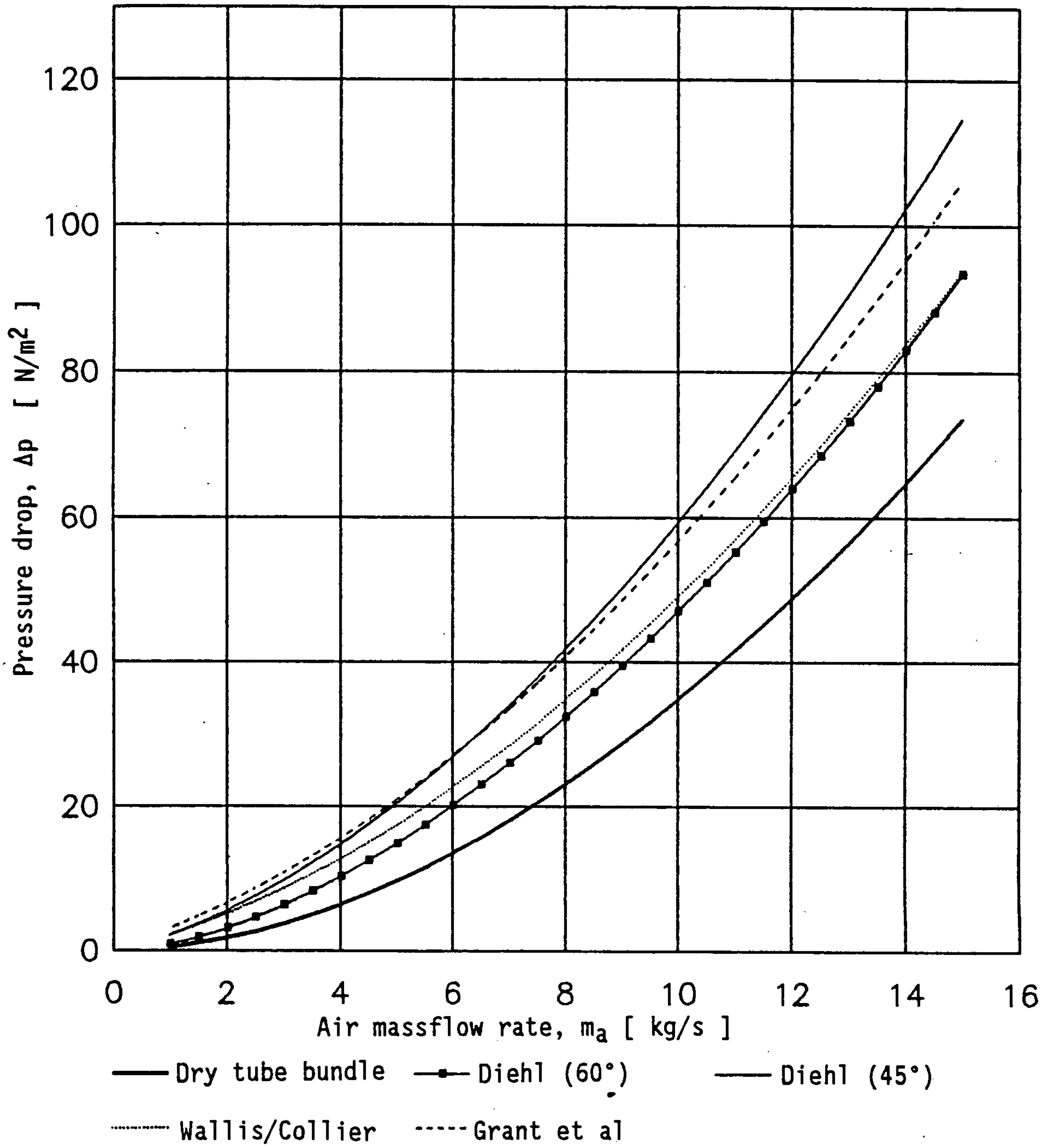

Figure 4.4 Pressure drop across a tube bundle in cross-flow with a recirculating water massflow rate of 5 kg/s.

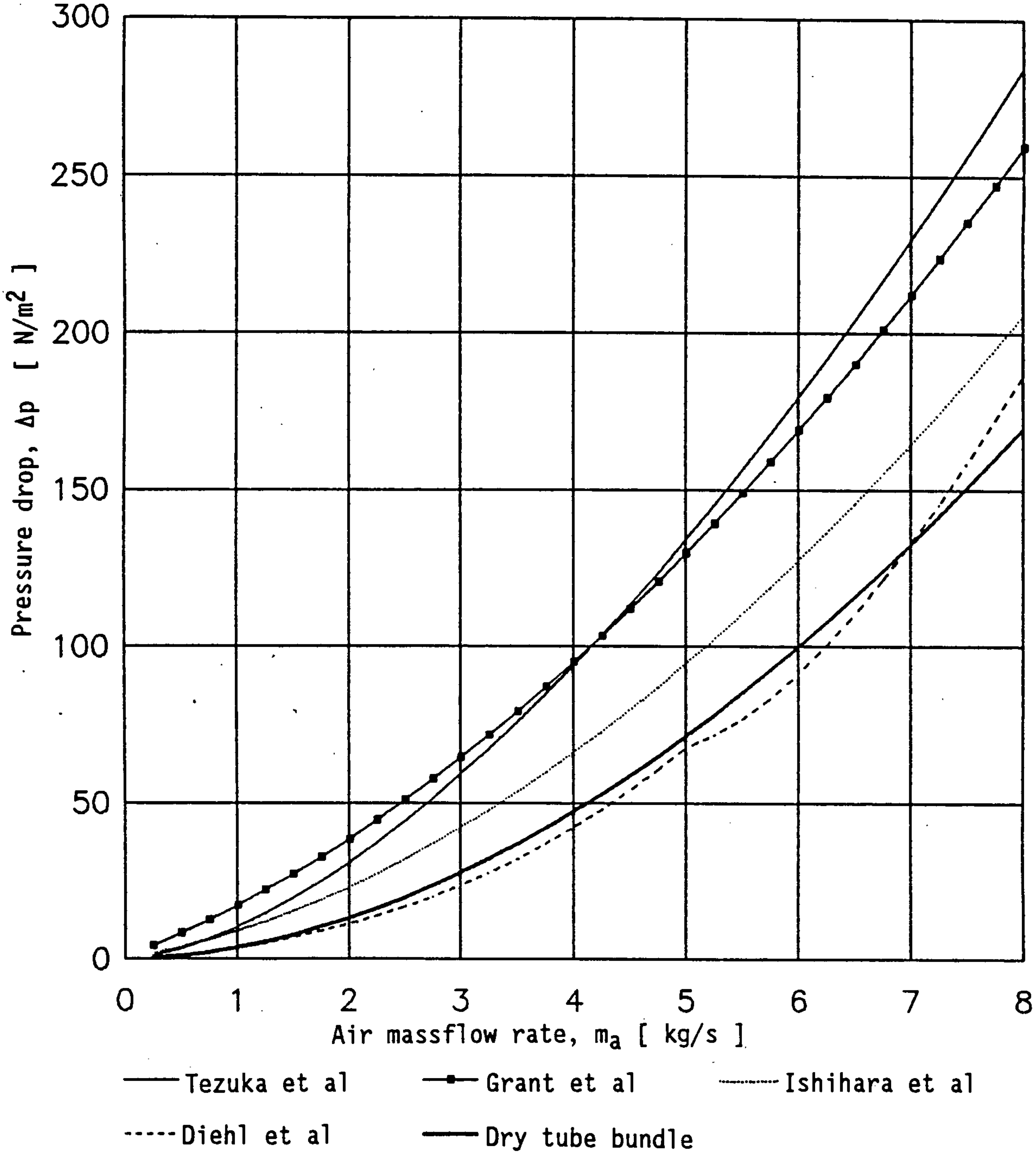

Figure 4.5 Pressure drop across a tube bundle in a counterflow layout with a recirculating water massflow rate of 4 kg/s.

CHAPTER 5

COMPUTER SIMULATION

Various computer programmes have been written in Fortran 77 source code on a Digital VAX 785, to simulate the operation of evaporative coolers and condensers.

The numerical models employed in these programs include the simplified approach and successive calculation models with varying degrees of approximation.

The following table lists the various programs and the solution method used in each program.

Name	Description	Model	Rating or Selection
CROSS	Cross-flow evaporative cooler	E,IM,M	R
COUNTER	Counterflow evaporative cooler	E,IM,M	R
COMBINE	Counterflow evaporative cooler with packing	M	R + S
SCROSS	Cross-flow evaporative cooler	M*	R + S
SCOUNT	Counterflow evaporative cooler	M*	R + S
CSCROSS	Cross-flow evaporative condenser	M*	R + S
CSCOUNT	Counterflow evaporative condenser	M*	R + S
TOWER	Natural draft cooling tower employing cross-flow evaporative cooling units	E,IM,M	R

E - Exact (Poppe) model

IM - Improved Merkel model

M - Merkel model

M* - Simplified (Merkel) model

R - Rating

S - Selection

5.1 Determination of coefficients

All the programs assume a 2 x d_o triangular tube spacing as shown in figures 5.1)a) and 5.1)b) for cross-flow and counterflow respectively.

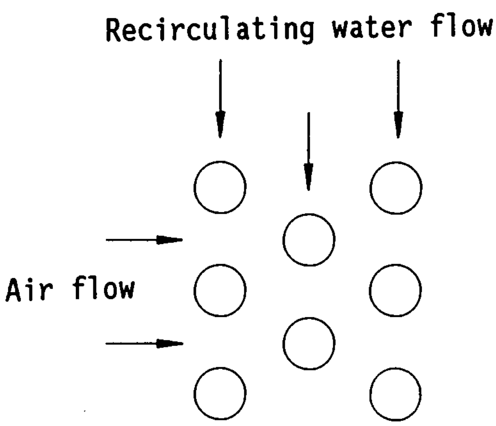

Figure 5.1) a) Tube layout for cross-flow evaporative cooler or condenser.

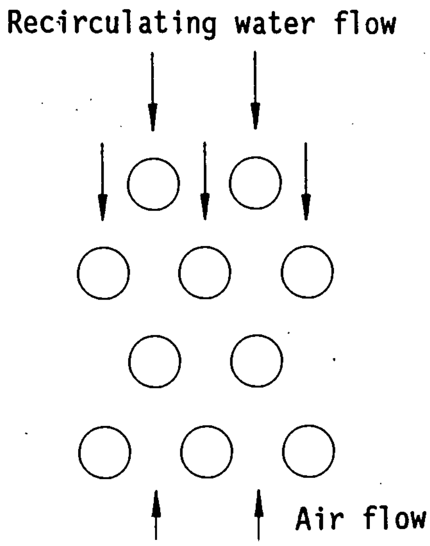

Figure 5.1) b) Tube layout for counterflow evaporative cooler or condenser.

The massflow rate of recirculating water is usually defined in the literature by the massflow rate of recirculating water flowing down one side of a tube per unit length. According to this definition the recirculating water massflow rate in a cross-flow evaporative cooler or condenser with triangular spacing is defined as

$$m_w = 2\,\Gamma\,L\,n_{hor} \tag{5.1.1}$$

or

$$\Gamma = \frac{m_w}{2\,L\,n_{hor}} \tag{5.1.2}$$

and for a counterflow evaporative cooler or condenser the recirculating water massflow rate is defined as

$$m_w = 4\ \Gamma\ L\ n_{hor} \tag{5.1.3}$$

or

$$\Gamma = \frac{m_w}{4\ L\ n_{hor}} \tag{5.1.4}$$

The mass transfer coefficient correlation given by Mizushina et al. [67MI1] (see Chapter 4.2) was used to determine the mass transfer coefficient for both cross-flow and counterflow evaporative coolers or condensers. The mass transfer coefficient correlation given by Mizushina et al. [67MI1] was determined for a counterflow evaporative cooler, but because of the lack of more suitable data this correlation was also used for the cross-flow coolers and condensers.

The film heat transfer coefficients used in all the programs are determined with the correlation presented by Mizushina et al. [67MI1]. Refer to Chapter 4.1 for a description of this film heat transfer coefficient correlation.

The heat transfer coefficients on the inside of the tubes in the case of an evaporative cooler are calculated from the correlations by Gnielinski [75GN1] and Kays et al. [55KA1] for turbulent and laminar flows respectively.

In the case of an evaporative condenser the correlations by Shah [79SH1] and Chato [62CH1] were used to determine the condensation coefficient inside the tubes. The correlation given by Chato [62CH1] was used when the vapour Reynolds number at the tube inlet was below 35000, otherwise the Shah [79SH1] correlation was used.

The Shah-correlation seems to predict very conservative condensation heat transfer coefficients if fluorocarbons (Freon's) are used as refrigerant, because of the low thermal conductivity of Freon liquid. Refer to Appendix G for the heat transfer coefficient correlations and condensation coefficient correlations.

5.2 Successive calculation models

The successive calculation models for the evaluation of evaporative coolers employ a numerical integration procedure for the evaluation of the governing differential equations. The cooler which is to be evaluated is subdivided into imaginary blocks (control volumes) with each block surrounding a length of tube.

By employing a fourth order Runge-Kutta integration procedure the outlet conditions (temperature, enthalpy etc.) of a given element can be determined if the inlet conditions of the element are known. The governing coefficients for heat and mass transfer are calculated for every block. Various flow geometries are possible for the process fluid flow, each requiring a different calculation algorithm. Four different process fluid patterns were considered in the case of cross-flow evaporative coolers, i.e.

i) Single pass (Straight through),
ii) Top-to-bottom (TTB),
iii) Front-to-back (FTB) and
iv) Back-to-front (BTF)

Only two process fluid flow patterns were considered as options for counterflow evaporative coolers, i.e.

i) Top-to-bottom (TTB) and
ii) Bottom-to-top (BTT)

The different process fluid flow patterns for cross-flow and

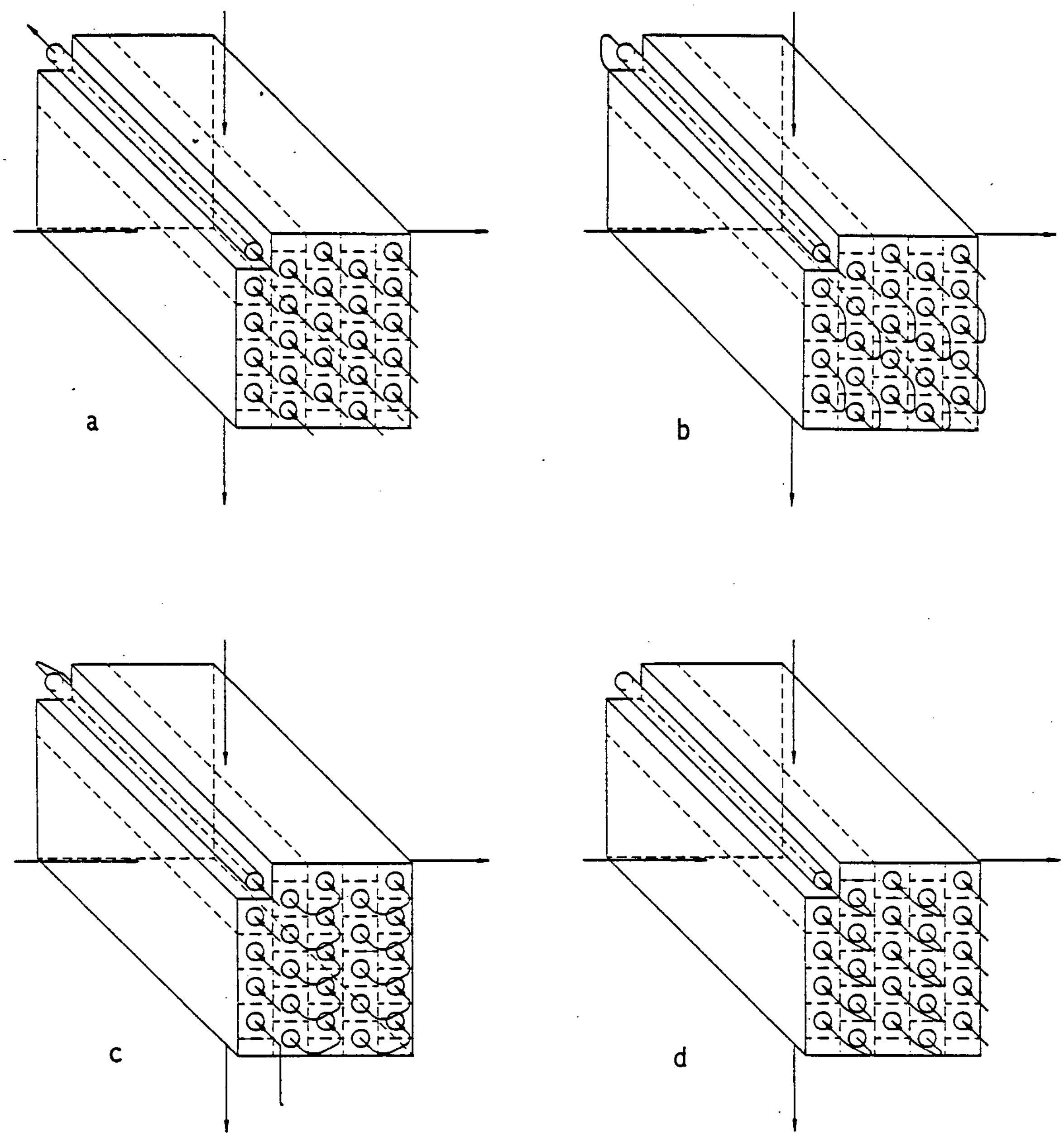

Figure 5.2 Serpentining arrangements for cross-flow evaporative coolers:
a) Straight through (single pass), b) top-to-bottom,
c) front-to-back and d) back-to-front.

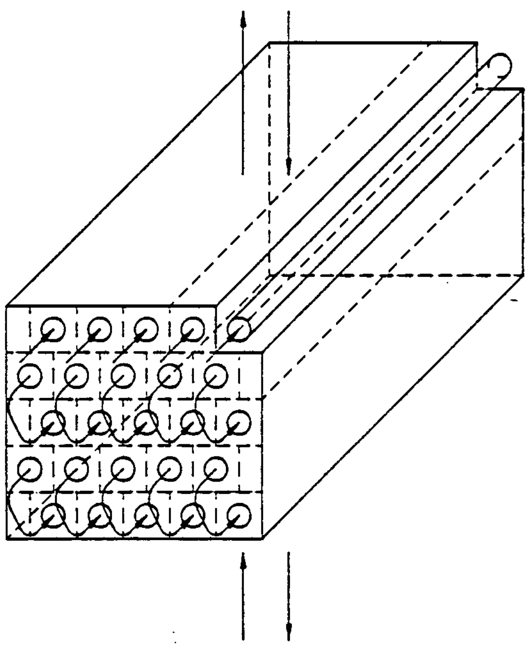

Figure 5.3 Top-to-bottom serpentining arrangement as used in a counterflow evaporative cooler.

counterflow evaporative coolers are shown in figures 5.2 and 5.3 respectively.

If the water flowing over the tubes is recirculated the inlet and outlet recirculating water temperature should be the same, as soon as the cooler is operating in a steady state. If the water flowing over the tubes is not recirculated, but fresh water is sprayed over the tubes, then the inlet spray water temperature has to be specified in order to evaluate the cooler performance.

5.2.1 Cross-flow evaporative cooler simulation

The solution procedure for the coolers with single-pass, top-to-bottom and front-to-back process fluid flow patterns are similar. Execution proceeds from the top front corner element where all the inlet conditions are known.

If the cooling water flowing over the outer surface of the tubes is recirculated, a viable inlet recirculating water temperature is chosen (the recirculating temperature at the inlet will always be larger than the air inlet wet bulb temperature and smaller than the process fluid inlet temperature).

By using a fourth order Runge-Kutta integration procedure the outlet conditions for the first element is computed. The outlet conditions of the first block is then used as inlet conditions for the surrounding blocks eg. the outlet process fluid temperature of the first block is used as the inlet process fluid temperature for the next block in the top row facing the airstream. By continuing the calculations, all the blocks along the top tube in the first row are evaluated. The evaluation of the next tube in the first row proceeds in a similar fashion until all the tubes in the first row have been evaluated. The next row of tubes can now be evaluated using the outlet air conditions of the previous row as the inlet conditions for the current row. Since the tubes are packed in a staggered array the inlet air conditions of a given block are taken as the average of the two blocks immediately in front of the block under evaluation.

As soon as all the blocks have been evaluated the average outlet cooling water temperature can be determined. If the cooling water is recirculated and the chosen inlet temperature of the recirculating water differs from the outlet temperature, a new value of inlet temperature of the recirculating water is chosen and the whole calculation is repeated until the inlet and outlet recirculating water temperatures are the same, giving the operating point of the cooler.

Interval halving could be used to determine the correct recirculating water inlet temperature, but it was found that the number of iterations could be cut dramatically by using a modified interval halving technique.

The modified technique is compared to the standard interval halving technique in figure 5.4.

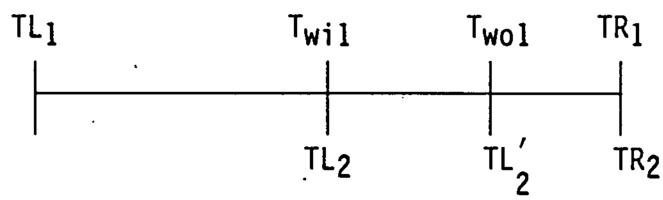

Figure 5.4 Graphical representation of the modified interval halving procedure.

TL signifies a left boundary and TR a right-hand boundary. By using the conventional interval halving procedure the first chosen inlet temperature is T_{wi1} which results in an outlet recirculating water temperature of T_{wo1}. The conventional interval halving method uses TL_2 as the new left-hand boundary while the modified interval halving procedure uses T_{wo1}, as the new left-hand boundary. The same holds for the right-hand boundary. It has been found that the outlet temperature of the recirculating water after the first choice of inlet temperature is very close to the final value which can be obtained by more iterations. Since a low choice of inlet temperature would result in a low outlet temperature and vice versa the operating point will always be between the left-hand and right-hand boundaries even if the modified interval halving method is used.

The modified interval halving method typically requires less than half the number of iterations that would be required by the conventional interval halving technique.

A simple flowchart showing the calculation procedure for the single pass, top-to-bottom and front-to-back process fluid flow patterns is shown in figure 5.5.

The evaluation of a cross-flow evaporative cooler with a back-to-front process fluid flow pattern is slightly more complicated than that of the other three patterns since there is no element of which all the inlet conditions are known even after an initial choice of recirculating water inlet temperature.

The solution is obtained by choosing the process fluid outlet temperature for each of the elements in the first row facing the airstream. By following a similar solution procedure as described for the front-to-back flow pattern the process fluid inlet temperature can be calculated. If the calculated average process fluid inlet temperature differs from the specified inlet temperature or if the calculated inlet temperature values very significantly from the average calculated value, the chosen outlet temperature of the process fluid of each horizontal row is changed by half the difference between the specified and the calculated process fluid inlet temperature in the given horizontal row.

Once the calculated and specified process fluid inlet temperatures are equal the average inlet and outlet recirculating water temperatures are compared and the inlet recirculating water temperature is adjusted

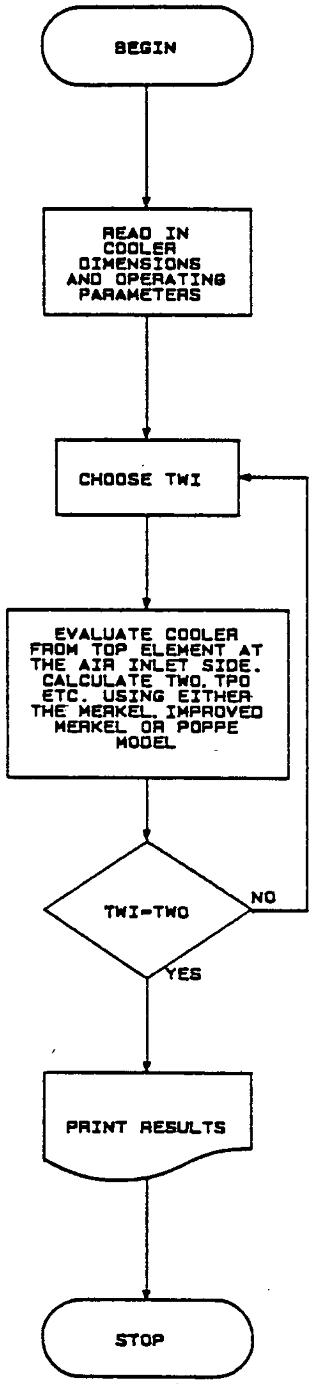

Figure 5.5 Program logic used in the determination of the operating point of a cross-flow evaporative cooler with either a straight through, top-to-bottom or a front-to-back process fluid flow arrangement.

accordingly as discussed above. The two iterative procedures for recirculating water inlet temperature and the process fluid outlet temperature are repeated until the solution is found.

The modified interval halving temperature is again employed in the determination of the recirculating water inlet temperature.

Figure 5.6 shows the calculation procedure for an evaporative cooler with a back-to-front process fluid flow pattern.

If the cooling water flowing over the tubes is not recirculated, the cooling water inlet temperature has to be specified and no iterative solution method would be needed in determining the operating point except in the case of back to front process fluid flow where the iterative solution method for determining the process fluid outlet temperature would still be needed.

Typical temperature profiles as determined with the program CROSS for the different arrangements are shown in figures 5.7)a) to 5.7)c).

The performance of a given cross-flow evaporative cooler using a front-to-back process fluid flow pattern would be very similar to the performance of the same cooler with a back-to-front process fluid flow pattern. The additional computer time needed for the evaluation of the back-to-front flow case is often not justified by the improvement in accuracy obtained by using the back-to-front algorithm instead of the simpler front-to-back algorithm.

Apppendix K shows the results of the program CROSS for a few example calculations to compare the different flow patterns and the analytical models.

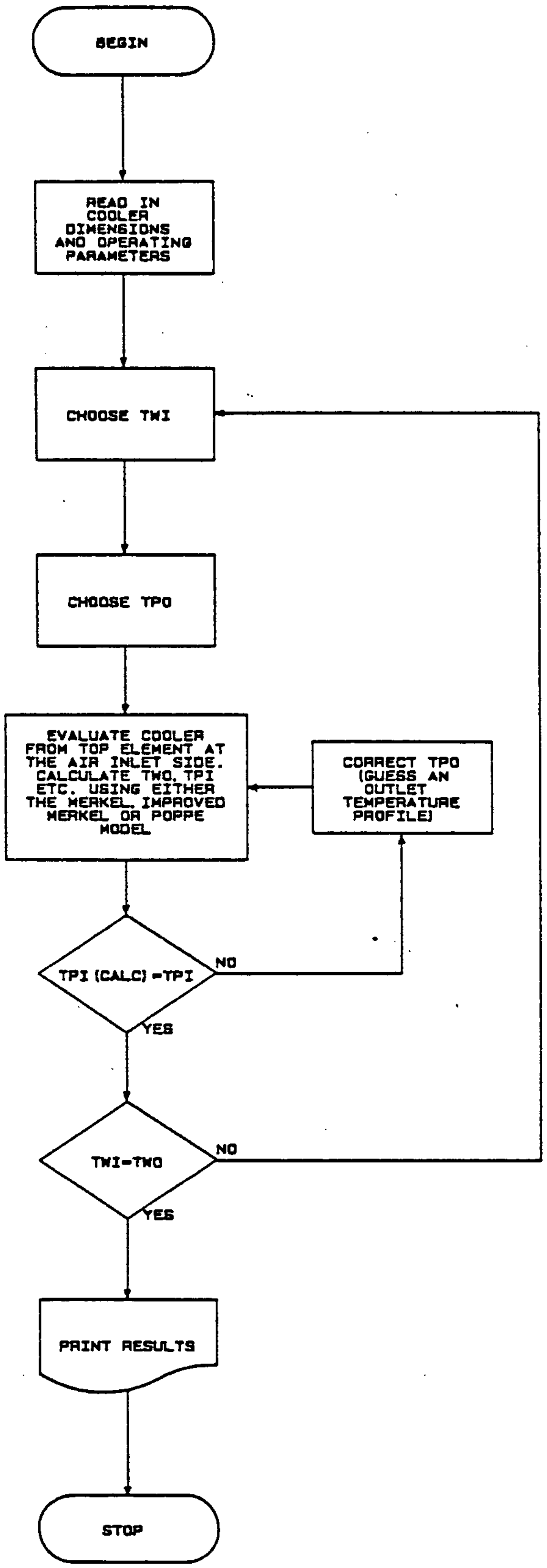

Figure 5.6 Program logic used in the determination of the operating point of a cross-flow evaporative cooler with a back-to-front process fluid flow arrangement.

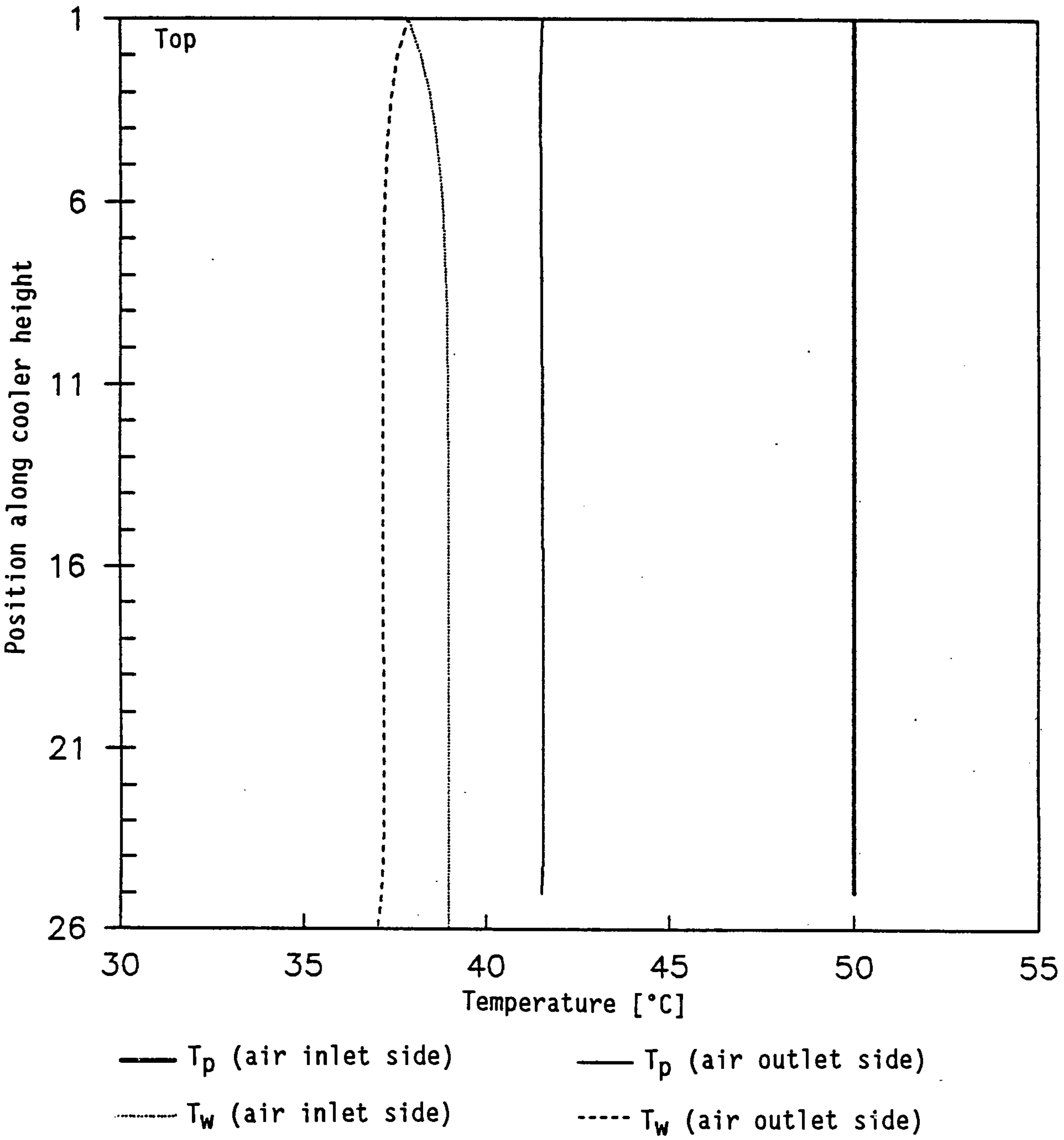

Figure 5.7 a) Temperature profiles along the height of a cross-flow evaporative cooler with a front-to-back process fluid flow arrangement.

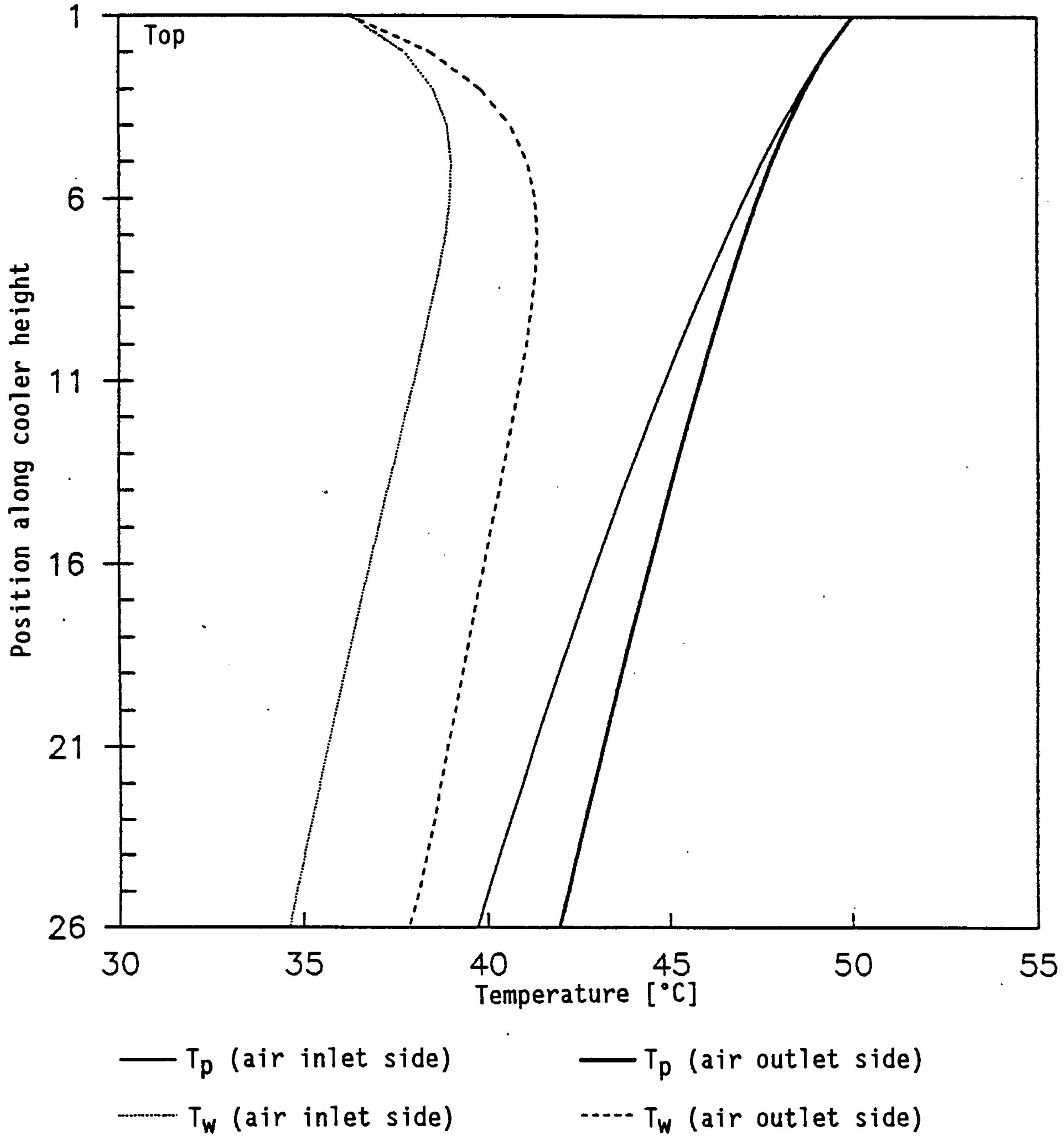

Figure 5.7 b) Temperature profiles along the height of a cross-flow evaporative cooler with a top-to-bottom process fluid flow arrangement.

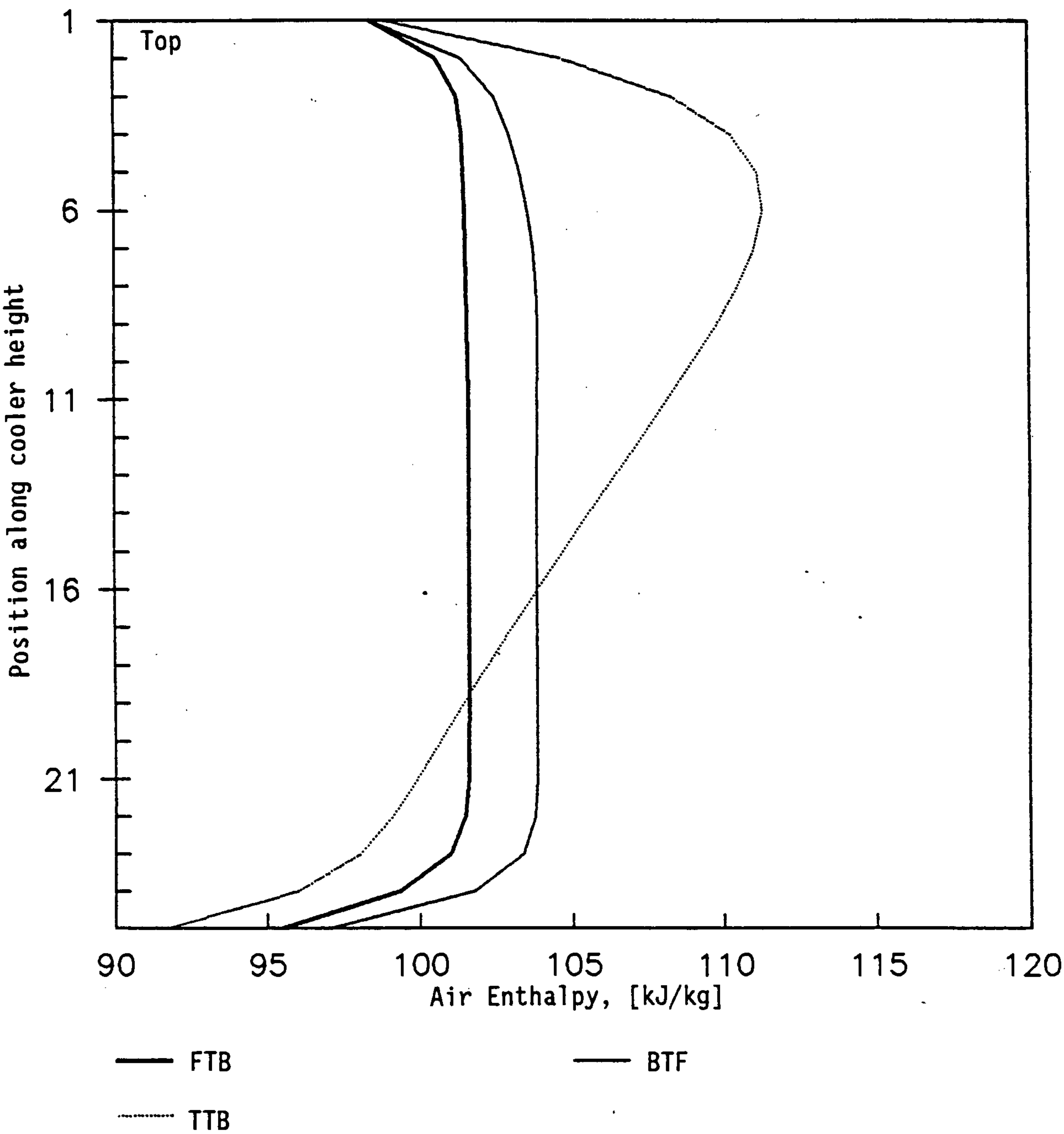

Figure 5.7 c) Outlet air enthalpy profiles along the height of a cross-flow evaporative cooler.

5.2.2 Counterflow evaporative cooler simulation

In the solution of a counterflow evaporative cooler a two dimensional model is used. The actual cooler consists of a number of similar vertical elements alongside each other. It is only necessary to analyse one of these vertical rows. The integration commences at the bottom of the cooler since the air properties are fully defined at the bottom of the cooler.

The solution of a counterflow evaporative cooler with a bottom-to-top process fluid flow pattern starts by choosing an average outlet recirculating water temperature and a value for the outlet recirculating water massflow rate (a given fraction of the recirculating water evaporate into the airstream).
Through an iterative numerical integration procedure the recirculating water outlet temperature and the recirculating water outlet massflow rate are determined when the inlet conditions are satisfied.

If more than one block is chosen along the length of the cooler the solution is further complicated by the fact that a different recirculating water outlet temperature has to be selected for each block to ensure that the calculated recirculating water inlet temperatures are constant along the top tube.

The solution of a counterflow evaporative cooler is very sensitive to the choice of outlet recirculating water temperature and double precision variables are essential to obtain a solution.

If the counterflow evaporative cooler uses a top-to-bottom process fluid flow pattern the solution would be even further complicated by the fact that the outlet process fluid temperature has to be selected and corrected after every integration through the cooler to ensure that the calculated inlet temperature of the process fluid corresponds to the specified value at the operating point of the cooler. The iteration procedure for the evaluation of counterflow evaporative coolers is shown in figures 5.8 and 5.9.

If the cooling water is not recirculated the solution of the cooler is still fairly complicated since the variation in the outlet temperature of the cooling water along the tube length must still be such that a uniform cooling water inlet temperature corresponding to the specified value is obtained at the top of the cooler.

The results of a few sample calculations using the counterflow evaporative cooler simulation program COUNTER are presented in Appendix K.

Various temperature and enthalpy profiles as determined with COUNTER are shown in figures 5.11a) to 5.11)c). It can be noted that the variation of recirculating water temperature along the outlet of the cooler is so insignificant that a one-dimensional model could be used without the loss of accuracy.

5.2.3 Combination cooler

A combination cooler is a counterflow evaporative cooler which employs a section of conventional Munters-type cooling tower packing either above or below the bare tube coil. The packing provides a large surface for mass transfer and this results in a lower average recirculating water temperature.

The program COMBINE uses sections of the COUNTER program but several simplifications have been introduced to allow the practical use of the program on a personal computer.
The simplifications include the following:
i) Only the Merkel solution method can be used,
ii) The model is one-dimensional and
iii) Only recirculating cooling water can be used.

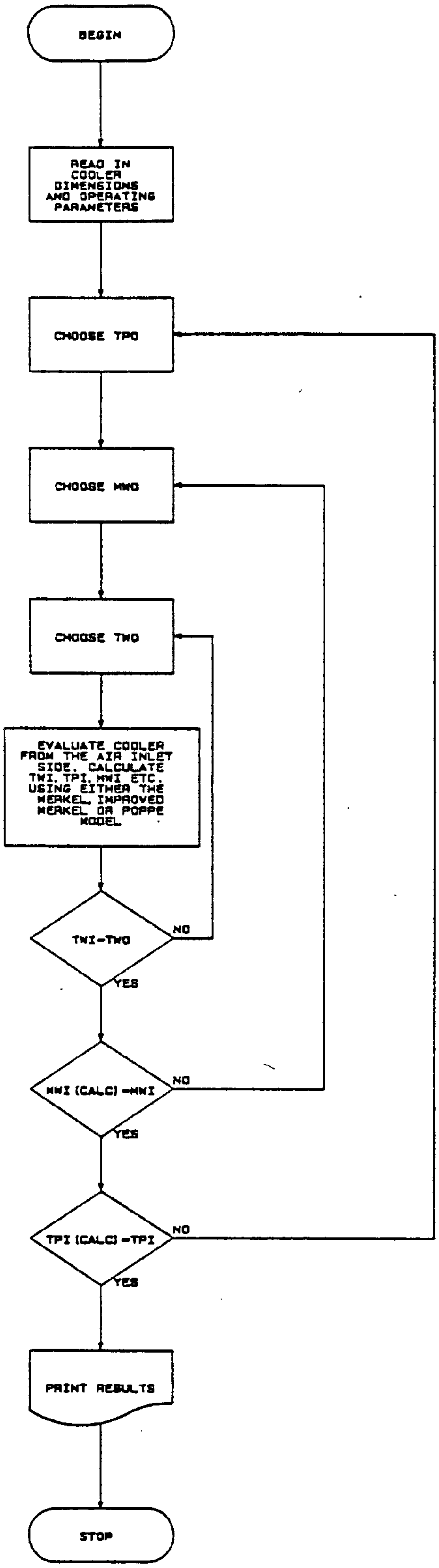

Figure 5.8 Program logic used in the determination of the operating point of a counterflow evaporative cooler with a top-to-bottom (TTB) process fluid flow arrangement.

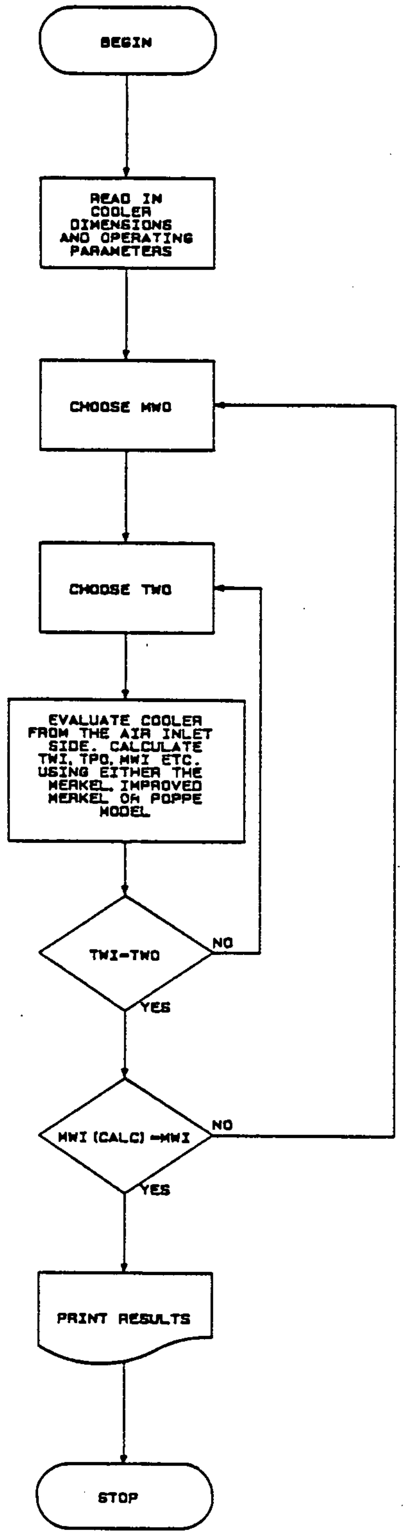

Figure 5.9 Program logic used in the determination of the operating point of a counterflow evaporative cooler with a bottom-to-top (BTT) process fluid flow arrangement.

The simplifications had to be made to lower program size, storage needs and execution time.

The following features have been introduced in the program to simplify counterflow evaporative cooler calculations:

i) Cooler selection or rating calculations can be performed,
ii) The cooler can be evaluate as a bare tube unit,
iii) Conventional fill can be used above or below the bare tube coil,
iv) Bottom-to-top process fluid flow layout calculations can be performed as a first approximation (Bottom-to-top calculations are much faster than the top-to-bottom calculations.)

The evaluation of the cooler proceeds in a very similar fashion to that of the COUNTER program discussed under section 3.2.2 except that the recirculating water cooling through the packing has to be considered. If the packing is placed above the tube section the temperature of the water falling on the packing has to be equal to the temperature of the water leaving the tubes. If the packing is placed below the coil the recirculating water entering from above the coil must have the same temperature as the water leaving the packing. Figure 5.10 shows the evaluation algorithm used to evaluate a combination cooler. The integration procedure through the packing is described in Appendix J.

It has been found that the addition of a section of fill material can lead to a significant decrease in the number of tube rows required to exchange a given amount of heat. Typical numerical examples are shown in Appendix K.

Figures 5.11)a) to 5.11)c) show the temperature and enthalpy profiles through a typical combination cooler.

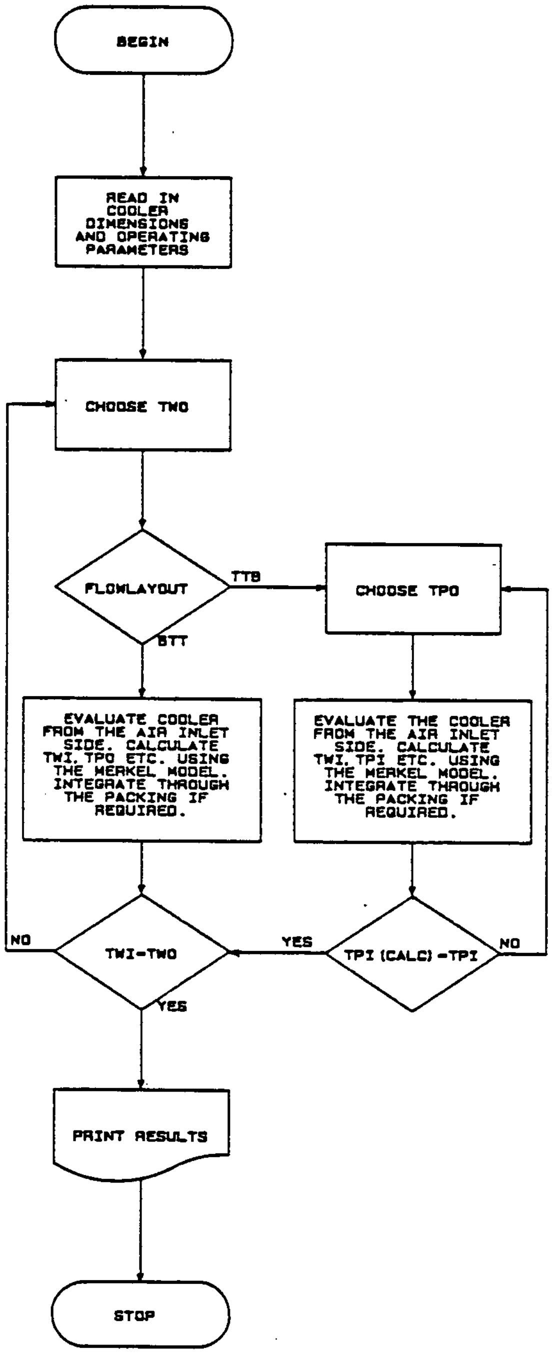

Figure 5.10 Program logic used in the determination of the operating point of a counterflow evaporative cooler with conventional cooling tower fill placed above or below the tubes.

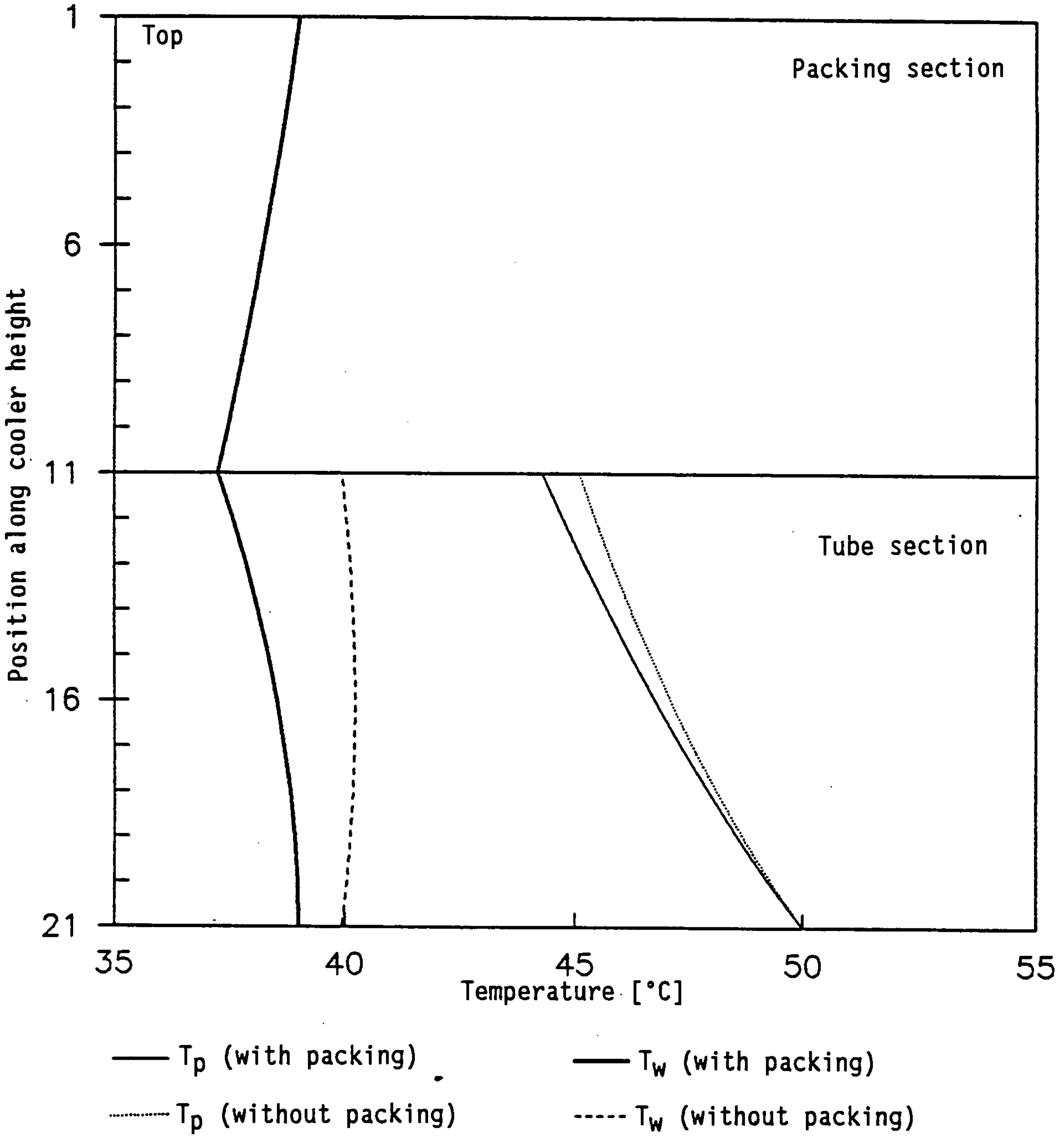

Figure 5.11 a) Temperature profiles along the height of a counterflow evaporative cooler with a BTT process fluid flow arrangement and conventional cooling tower packing placed above the tubes.

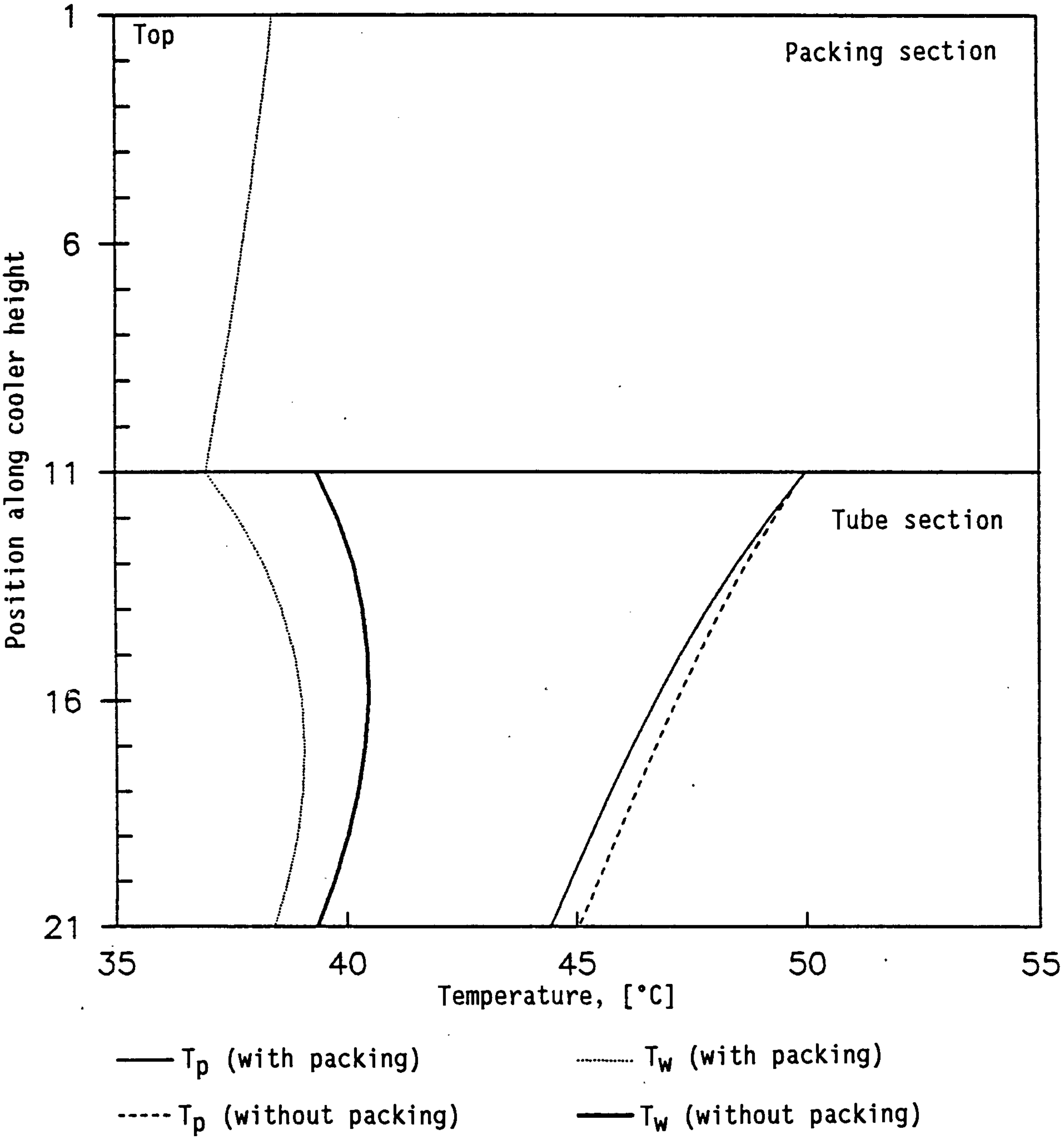

Figure 5.11 b) Temperature profiles along the height of a counterflow evaporative cooler with a TTB process fluid flow arrangement and conventional cooling tower packing placed above the tubes.

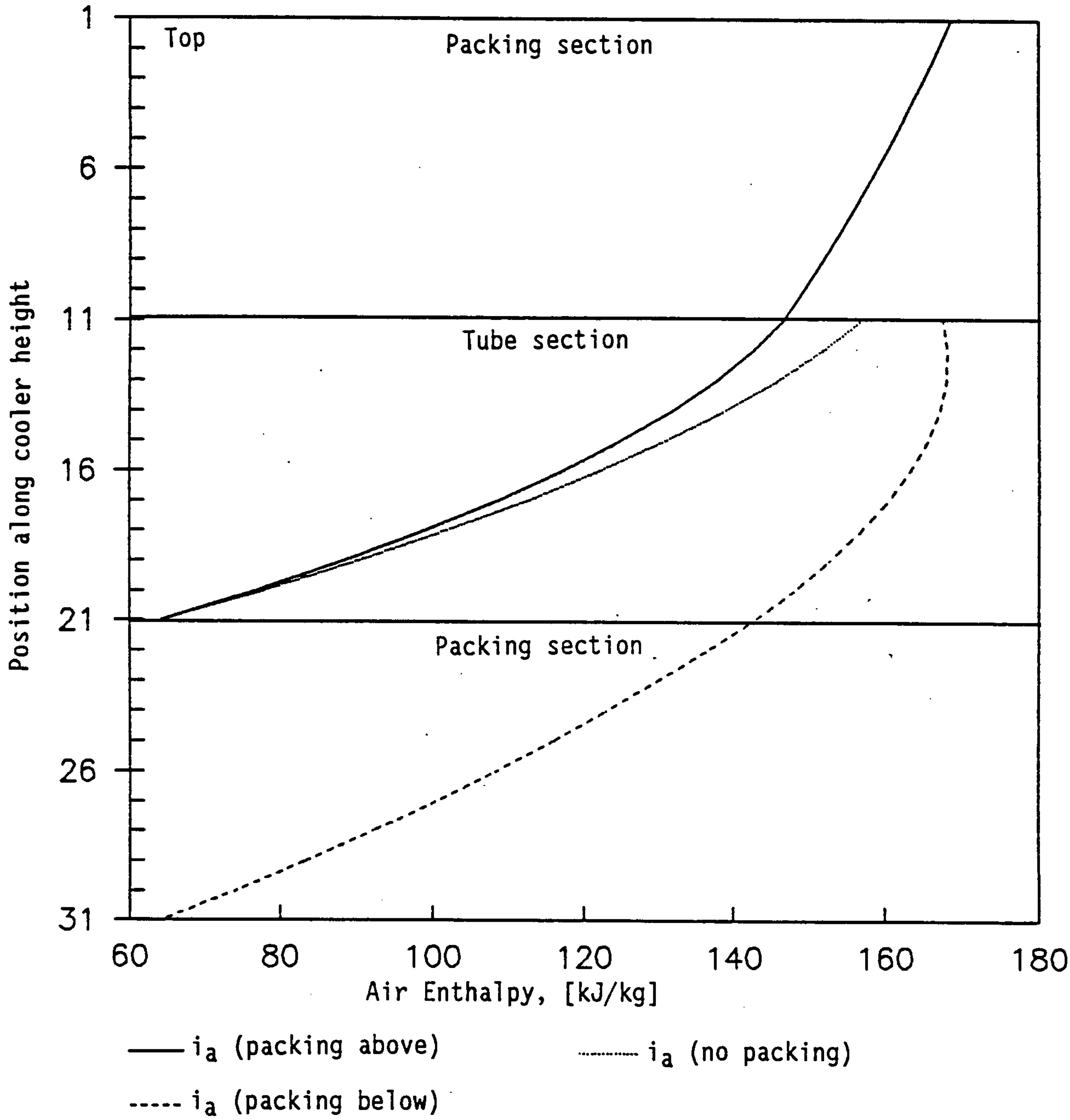

Figure 5.11 c) Enthalpy profiles along the height of a counterflow evaporative cooler with a TTB process fluid flow arrangement.

5.3 Simplified models

Four programs, SCOUNT, SCROSS, CSCOUNT and CSROSS using the simplified analytical modelling procedure for evaluating evaporative coolers or condensers have been written.

All four of these programs can be used for cooler or condenser rating and sizing calculations, and the relative fast execution time allows for the easy adaptation of these programs for execution on a personal computer.

The iterative procedures for the rating and selection of evaporative coolers are discussed in detail in section 3.1.4 and the rating and selection procedures for evaporative condensers are discussed in detail in section 3.2.4.

In the case of counterflow evaporative coolers the simplified model yields results which are within 1% of the results obtained with the two dimensional successive calculation numerical integration procedure.

The simplified model gives results which agree fairly well with the results obtained with numerical integration model in cross-flow evaporative coolers with relative short tube lengths. The discrepancy in results at longer tube lengths is due to the three dimensional nature of the recirculating water temperature profile, which cannot be represented well enough by a single representative temperature.

The four different process fluid flow patterns considered in the numerical integration analysis of cross-flow evaporative coolers can be compared to the results of the simplified method if the correct process fluid velocity for the chosen process fluid flow pattern is used in the calculation of the heat transfer coefficient on the inside of the tubes.

The simplified methods allow the easy evaluation of the effect of serpentining on both cross-flow and counterflow evaporative coolers since no complicated new integration procedure has to be used for higher order serpentining. In order to evaluate a given cooler layout with second order serpentining only the flow velocity of the process fluid has to be doubled when calculating the heat transfer coefficient on the inside of the tubes.

Refer to Appendix K for a comparison between the results obtained with the simplified and accurate models.

The iterative selection and rating procedures for evaporative condensers are discussed in detail in section 3.2.4. The simplified method is expected to yield very good results in the counterflow model since the recirculating water temperature in an evaporative condenser would be almost constant because of the constant condensing temperature.

It is expected that the simplified cross-flow evaporative condenser simulation would yield fairly accurate results because of the relative flat recirculating water temperature profile which would be prevailing in an evaporative condenser with a constant condensing temperature.

Refer to Appendix K for some typical results which have been obtained with the evaporative condenser simulation programs.

5.4 Natural draft cooling tower

The accurate numerical integration routines used in the program CROSS for the simulation of a cross-flow evaporative cooler have been linked to the natural draft equation for a cross-flow tower for the evaluation of a cross-flow evaporative cooling tower.

The proposed tower (shown in figure E.1) consists of large cross-flow evaporative cooler modules placed around the outer perimeter of a cooling tower shell. The cross-flow evaporative cooler modules may be arranged in an A-frame configuration to obtain a larger surface area without enlarging the tower base diameter too much.

In order to keep the process fluid velocity within allowable limits the front-to-back or the back-to-front process fluid flow patterns are normally employed. As mentioned before the relatively long execution time of the back-to-front process fluid flow pattern compared to that of the front-to-back flow pattern does not justify its use since there is very little difference in the cooler capacities obtained with these two flow patterns.

The draft equation for a typical cross-flow cooling tower is derived in detail in Appendix E. The pressure drop coefficients are also discussed in Appendix E, except for the pressure drop across the wet tube bundle which is calculated from the correlation given by Collier [79CO1]. The correlation of Collier is discussed in Chapter 4.

In analyzing a natural draft cooling tower an air massflow rate is chosen, the cooler units evaluated, the total pressure drop through the tower is computed and the available pressure difference determined.

If the initial air massflow rate was chosen correctly the pressure drop through the tower would be exactly balanced by the available draft, but if the pressure drop is not matched by the available draft a new air massflow rate has to be selected and the whole calculation process must be repeated.

The calculating procedure is shown in figure 5.12.

Typical results of the natural draft tower simulation program is presented in Appendix J. The air leaving the cooler units around the

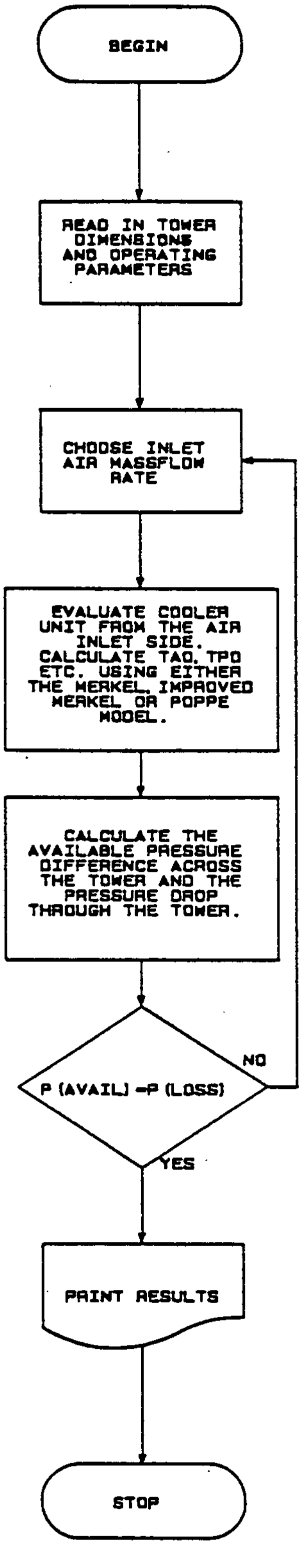

Figure 5.12 Program logic used in the determination of the operating point of a natural draft cooling tower with cross-flow evaporative cooling units placed around the outer perimeter of the tower.

base has been found to be almost saturated, if not fully saturated, and consequently the assumption of saturated outlet air to determine the outlet air density when employing the Merkel type analytical model is normally a good assumption. The exact method does not employ this assumption since the air properties are fixed at every part of the cooler, therefore it is generally expected that the exact model will yield more accurate results.

CHAPTER 6

EXPERIMENTAL DETERMINATION OF THE HEAT AND MASS TRANSFER COEFFICIENTS IN A CROSS-FLOW EVAPORATIVE COOLER

No accurate data or correlations for the determination of the heat and mass transfer coefficients in a cross-flow evaporative cooler or condenser could be found in the literature. These coefficients can only be found experimentally since the heat/mass transfer analogy cannot be applied to an evaporative cooler because of the uncertainty about the actual air/water interface area. The analogy fails because of the geometrical dissimilarity of a dry tube bundle and a wet tube bundle.

Various factors influence the transfer coefficients in an evaporative cooler including process fluid temperature, air massflow rate, recirculating water massflow rate, process water massflow rate, and inlet air conditions, and tube geometry.

A test tunnel was erected at the Department of Mechanical Engineering of the University of Stellenbosch in order to conduct a series of tests on a cross-flow evaporative cooler.

6.1 Description of test tunnel and apparatus

A horizontal tunnel with a 2 x 2 m cross section was built in order to test wet heat exchanger coils and evaporative coolers.

The tunnel shown in figure 6.1 consists of an inlet section, a test section, a mixing/measurement section and an induced draft fan. The tunnel walls downstream of the test section are insulated to minimize any change in the air temperature between the test section and the air sampling station. The inlet air temperatures are measured in the inlet section of the tunnel and the outlet air temperatures are

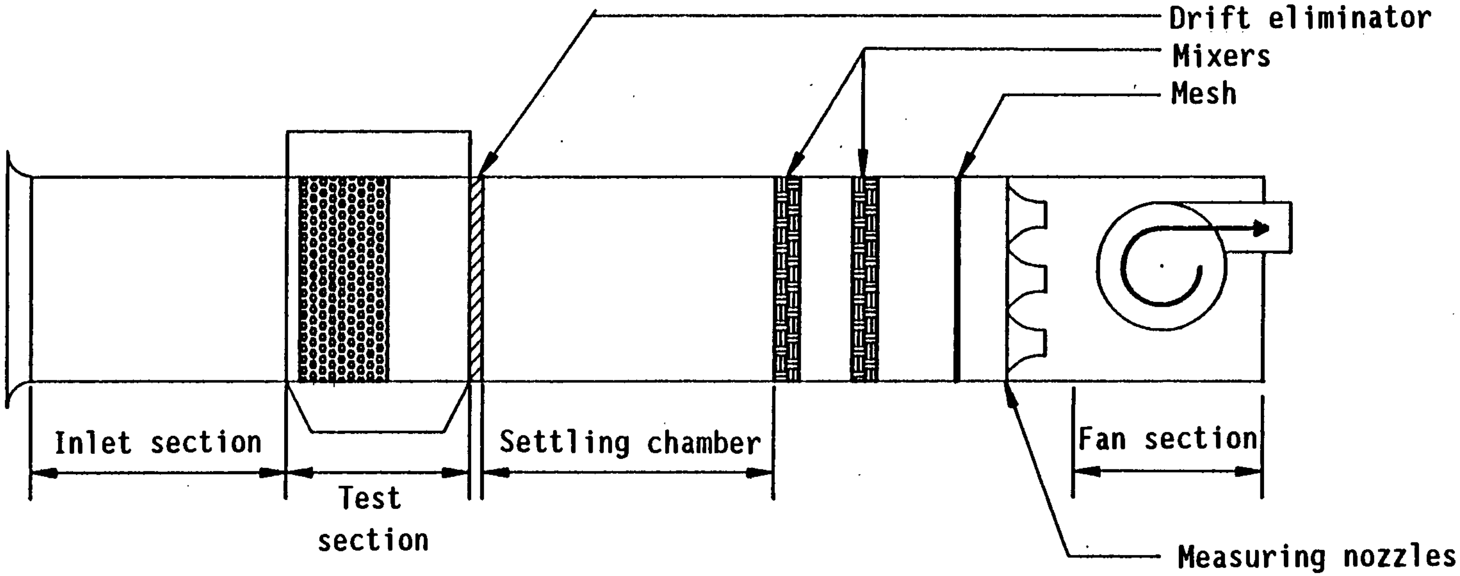

Figure 6.1 Layout of the experimental tunnel for the cross-flow evaporative cooler tests.

measured downstream of two sets of mixers to ensure good mixed air temperatures.

The air massflow rate through the tunnel is infinitely variable through the use of a stepless electronic speed control device which governs the speed of the centrifugal fan motor. Air massflow rates of up to 15 kg/s through the tunnel can be obtained depending on the flow resistance of the particular test section installed.

Two cross-flow evaporative cooler test sections were built. The first test section (see figure 6.2) consisted of 250 galvanized steel tubes, 38,1 mm OD and 34,9 mm ID, spaced in a 2 x d_0 triangular array of ten vertical rows. The sides of the test section ~~was~~ were made of a 13 mm thick transparent Perspex plate to allow observation of the test section. Incomplete wetting of the lower tubes facing the airstream was observed when testing the upright test section at high air velocities and low recirculating water flow rates.

The second test section was suspended in a frame which pivoted around the middle of the test section as seen in figure 6.3, this allowed the test section to be rotated by up to 18,75° from the vertical. Only 22 vertical rows of tubes could be fitted in the rotating test section in a 2 x d_0 triangular array. The same 38,1 mm OD and 34,9 mm ID galvanized steel tubes were used for the inclined and the upright test sections.

The tubes were connected with flexible rubber hoses in a top-to-bottom serpentining arrangement. Drift eliminators were installed downstream of the test section to prevent entrained water droplets from travelling down the tunnel in the airstream.

The water in a 40 m^3 underground tank was heated to the required temperature by means of a two-pass oil burning boiler. The hot water was then pumped from the surface of the tank to the inlet header of the test section. After flowing through the test section the cooled

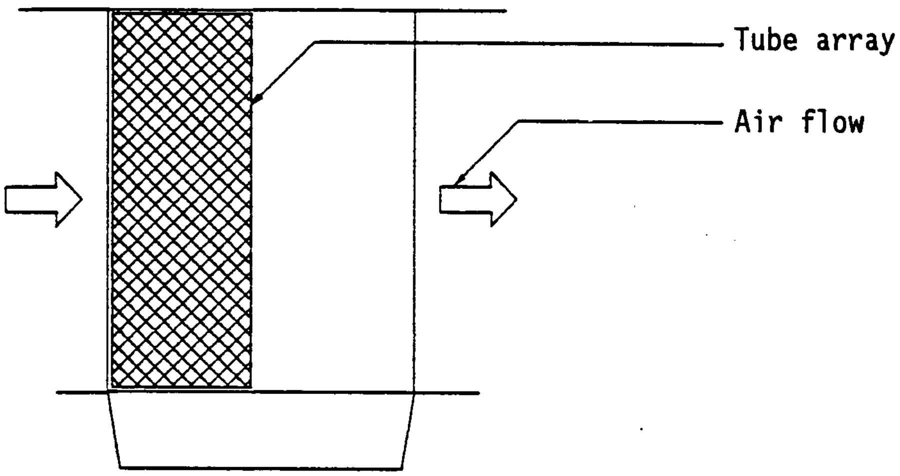

Figure 6.2 Upright test section layout.

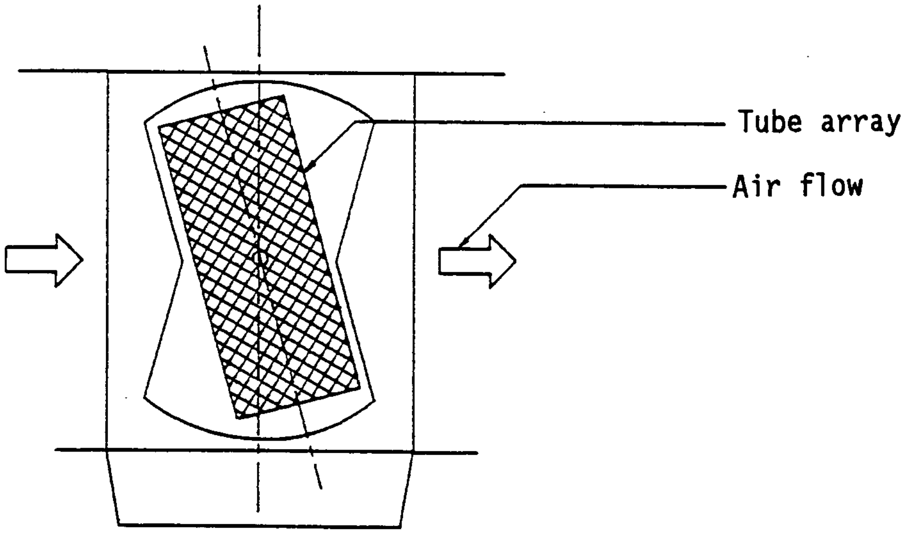

Figure 6.3 Inclined test section layout.

process water flowed back into the bottom of the water storage tank, ensuring a stable process water inlet temperature at the test section.

The recirculating water was pumped from the sump underneath the test section to the spray tubes located above the test section. Each of the tubes in the top row of the test section has a spray tube directly above it to ensure an even distribution of recirculating water. The layout of a spray tube is shown in figure 6.4. A spray tube consists of a horizontal copper tube which has small diameter holes drilled into the top of the tube along its length. The copper tube is enclosed in a larger diameter plastic tube. The plastic tube has a narrow slot machined at the bottom of the tube along its length. The recirculating water which is pumped from the sump underneath the test section is fed into a header which distributes the water to the copper tubes of each spray tube. Since the pressure inside the copper tubes is high, an equal amount of recirculating water is sprayed out of each hole at the top of the copper tubes. The spray water strikes the inside of the larger diameter plastic tube and flows downwards and out through the slit in the bottom of the plastic tube and onto the top tubes in the test section.

Special care was taken to prevent the air stream from short circuiting the test section. Galvanized plates were suspended underneath the bottom row of tubes in the test section. The ends of the plates hung in the water in the recirculating water sump effectively stopping the air from short circuiting underneath the test section.
Flat galvanized steel plates were placed on top of the spray tubes to prevent short circuiting of the air through the gaps between the spray tubes.
In the evaporative cooler tests the following quantities had to be measured:

i) Process water - massflow rate, inlet and outlet temperature

ii) Recirculating water - massflow rate, inlet and outlet temperatures

iii) Air - massflow rate, inlet and outlet temperatures (wet bulb and dry bulb)

iv) Other - Atmospheric pressure

a) Massflow measurements

The process water massflow rate was measured using an orifice plate placed in the process water supply line between the hot water tank and the test section. The orifice plate was made and installed according to the BS-1042 standard with pressure tappings at a distance equal to one tube diameter upstream of the orifice plate and half a tube diameter downstream of the orifice plate.

The pressure difference across the orifice plate was recorded with two Foxboro differential pressure transducers. These two transducers covered different pressure ranges and this allowed a wide massflow range to be measured without having to change the orifice plate. The 4 - 20 mA signal delivered by the Foxboro pressure transducers were converted to a voltage signal (between 1 and 5 V) by passing the current through a high precision 250 ohm resistor. The pressure transducers were calibrated by using a zero differential pressure signal as the low range calibration point and a known pressure difference near the pressure transducer full scale position as the high range calibration point. The calibration of the transducers were checked using a weighing drum and a stopwatch.

Two instruments were installed to measure the recirculating water massflow rate i.e. a rotameter for measuring the low massflow rates and an orifice plate for measuring the higher recirculating water massflow rates. The recirculating water orifice plate was made using the same BS-1042 standard as for the process water orifice plates. The differential pressure across the recirculating water orifice plate was also obtained with a calibrated Foxboro pressure transducer.

Since the recirculating water orifice plate was installed at a distance of about 20 diameters from the pump it was deemed necessary to install a flow straightener immediately downstream of the pump. The straightener was also made according to the BS-1042 standard.

The rotameter which was installed can measure a water massflow rate of up to 3,33 kg/s on a linear scale from 0 to 25. The rotameter was consequently calibrated by using a stopwatch and a weighing drum and a simple second order polynomial curve was fitted to the data and this curve was then used as the calibration curve for the rotameter.

The air massflow rate was determined from the differential pressure measured across the air measuring nozzles in the test tunnel (see figure 6.1). The five elliptical nozzles were made according to the ASHRAE 51 - 75 standard. The differential pressure readings across the nozzles were taken with a calibrated low pressure Foxboro transducer. As in the use of the other pressure transducers the current signal of the transducer was converted to a voltage reading through the use of a precision resistor. At low air massflow rates one or more of the nozzles were closed up to give higher differential pressure readings to ensure more accurate massflow determination.

The difference in pressure between the atmosphere and the pressure inside the tunnel upstream of the nozzles was recorded for every test and this value was subtracted from the atmospheric pressure in calculating the density of the air entering the nozzles.

b) <u>Temperature readings</u>

The temperature readings were all made with calibrated copper-constantan thermocouples. The thermocouples were calibrated by determining the thermocouple readings at ice melting point at water boiling point at atmospheric pressure. The calibration values were then used to correct every temperature reading taken with each thermocouple.

The inlet and outlet temperatures of the process water was measured with two calibrated thermocouples in both the inlet and outlet process water manifolds.

The inlet recirculating water temperature was measured with two calibrated thermocouples placed in the recirculating water inlet header. Special thermocouple probes were made to measure the bulk temperature of the water film flowing over a tube. By using these probes the average temperature of the recirculating water leaving the coil could be determined.
The probes were made from a short piece of cylindrical Perspex with holes drilled axially and radially into it as shown by figure 6.5. A 3 mm thermocouple fitted snugly into the axial hole with the tip of the thermocouple just visible through the radial holes. The required dimensions of the radial holes were determined by a trial and error method. If a temperature probe is held under a tube with the Perspex just touching the water film flowing over the tube, the surface tension draws the water into the larger of the radial holes at the top of the probe. The water drops out of the bottom hole continuously wetting the tip of the thermocouple. Ten thermocouples were fitted with these probes and installed under every second tube in the bottom tube layer on either side of the test section.

For energy balance calculations it is necessary to determine the average temperature of recirculating water leaving the bottom for of the cooler. The average of the ten film temperature probe readings could be used as the average outlet recirculating water temperature, but since the possibility of unequal recirculating water distribution among the ten vertical tube rows exists, a temperature measuring trough was installed under the tube bank. The trough collects the water leaving all ten vertical rows and the mixed temperature in this trough could be taken taken as the average recirculating water outlet temperature.

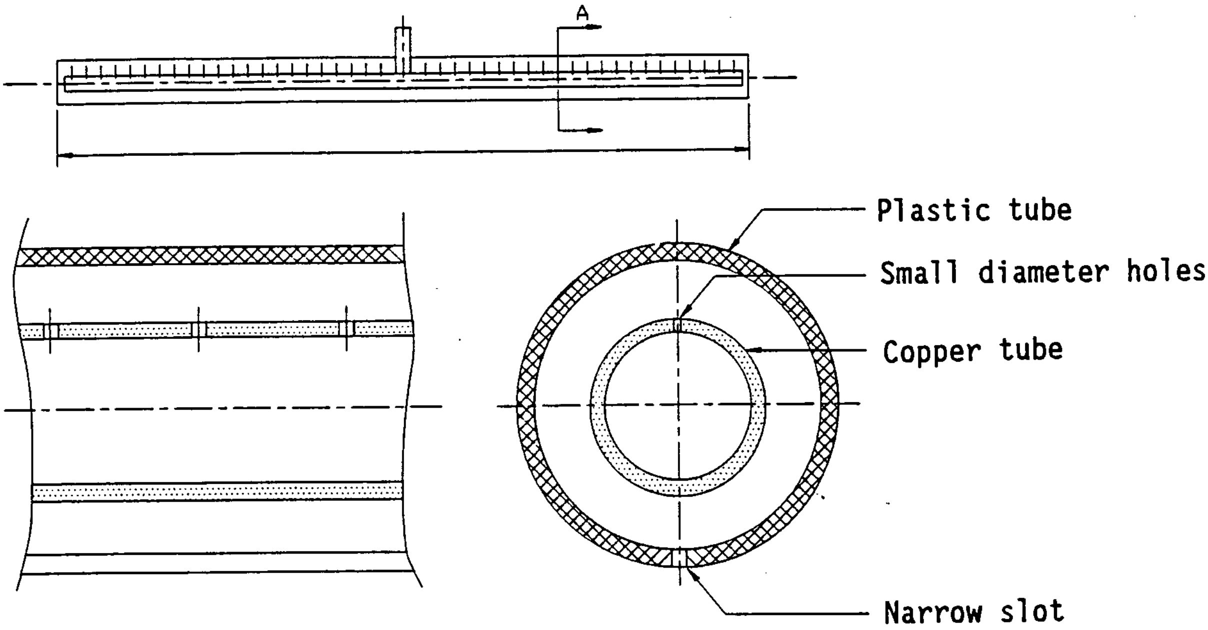

Figure 6.4 Spray tube layout.

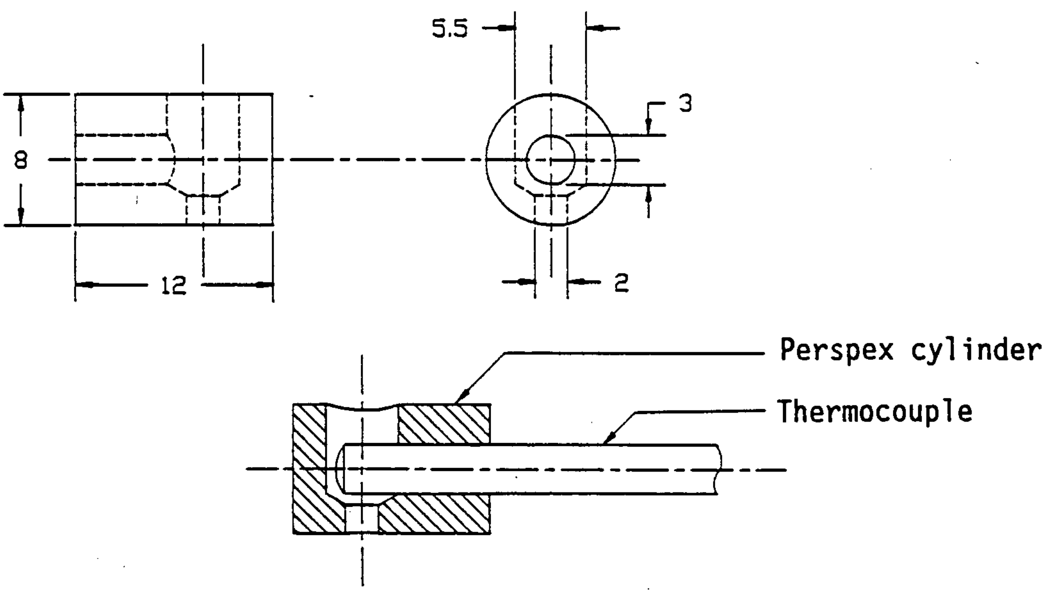

Figure 6.5 Recirculating water bulk temperature measuring probes.

The air wet bulb and dry bulb temperature were measured upstream and downstream of the test section. The wet bulb temperature readings were taken using a simple sampling tube as shown in figure 6.6. The wet bulb thermocouples were kept wet by a small cotton sleeve which was pulled over the tip of the thermocouple while the other end of the sleeve was suspended in a small water reservoir to keep it wet. To ensure that the correct wet bulb temperature would be read with the wetted sleeve thermocouple, a small fan was installed to draw the air through the air sampling tube at between 3 and 5 m/s. In order to read the average air temperature, five wet bulb and five dry bulb thermocouples were installed at each air sampling point.

c) Other measurements

The barometric pressure was recorded with a mercury column barometer before every test. The two phase pressure drop across the wet tube bundle was measured with a Betz manometer. At the upstream side of the test section the walls of the tunnel always remained dry and conventional pressure tappings could be installed. Downstream of the test section the walls of the tunnel were wet because of splashing and drop entrainment.

Special pressure tappings were needed to measure the static pressure inside the tunnel in the presence of water on the inside wall, since any water trapped in the pressure lines would result in faulty pressure measurements. The downstream pressure tappings were constructed from a copper tube with a relatively large diameter installed flush with the inside wall of the tunnel as shown in figure 6.7. The copper tube enters the wall at 90° but it is then bent upwards to prevent any water from flowing down the pressure lines. The relative large diameter of the copper tube ensured that water drops do not close off the whole cross section of the tube as would be the case with a small diameter tube.

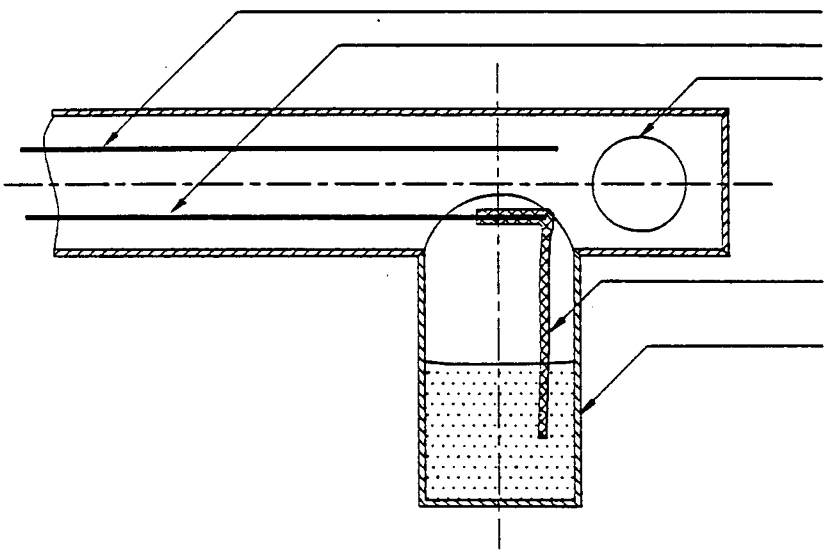

Figure 6.6 Air sampling probe.

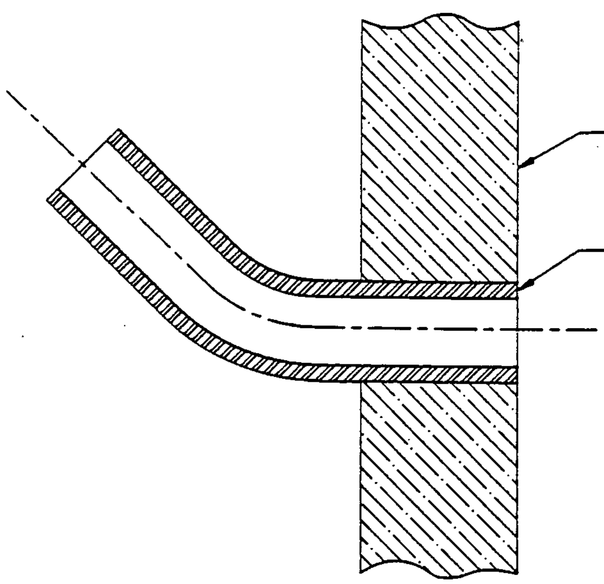

Figure 6.7 Layout of pressure tappings for static pressure readings on a wet wall.

6.2 Data logging and energy balance calculations

The data logging was performed using a Kayes Digilink 4 data logger linked to an Olivetti M21 personal computer. The data logging system layout is shown in figure 6.8.

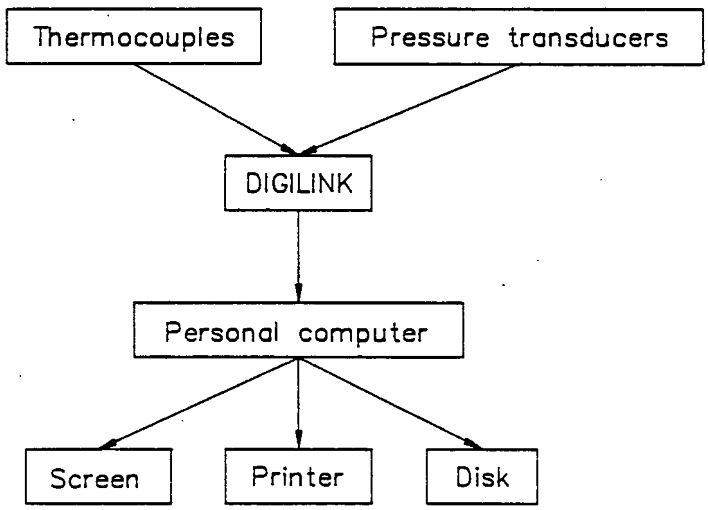

Figure 6.8 Block diagram showing the data logging system layout.

The thermocouples were all directly connected to the Digilink, the internal electronic ice point of the Digilink eliminating the need for an ice bath. The Digilink was programmed to convert all the temperature readings to degrees Centigrade before transferring them to the computer.

The pressure transducer signals were all converted to voltage signals which could be measured with the Digilink. The Digilink was programmed to convert the voltage signal of each pressure transducer into a pressure reading in Pascal through the use of the transducer calibration data. The personal computer connected to the Digilink could then read all the required temperatures and pressures directly in degrees Centrigrade and Pascal respectively.

A computer program was written in TurboBasic to read all the data from the Digilink and to perform the necessary energy balance calculations on the data. The flow chart for the data logging program is shown in figure 6.9.

The temperature readings were corrected using the thermocouple calibration data. The massflow rates were then computed from the measured pressure differentials across the orifice plates and from the rotameter reading if the rotameter was used for the recirculating water massflow measurement. The energy balances were computed with the following equation:

$$\text{Energy balance} = \frac{\Delta q_p + \Delta q_w + \Delta q_a}{\Delta q_p} \times 100\ \% \qquad (6.1)$$

where

$$\Delta q_p = m_p \left(c_{ppi}\, T_{pi} - c_{ppo}\, T_{po} \right) \qquad (6.2)$$

$$\Delta q_w = m_{wi}\, c_{pwi}\, T_{wi} - \left(m_{wi} - m_a \left(w_{ao} - w_{ai} \right) \right) c_{pwo}\, T_{wo} \qquad (6.3)$$

$$\Delta q_a = m_a \left(i_{ai} - i_{ao} \right) \qquad (6.4)$$

The immediate processing of the data made it possible to continue the tests until a completely steady state was reached. After each data set was taken graphical displays of temperature and massflow rate versus time could be displayed to show any fluctuations and variations.

6.3 Experimental procedure

The following variable parameters have an influence on the performance of the experimental evaporative cooler:

i) air massflow rate
ii) process water massflow rate
iii) recirculating water massflow rate
iv) inlet temperature of process water
v) inlet air conditions and
vi) the swing angle, θ, of the inclined test section.

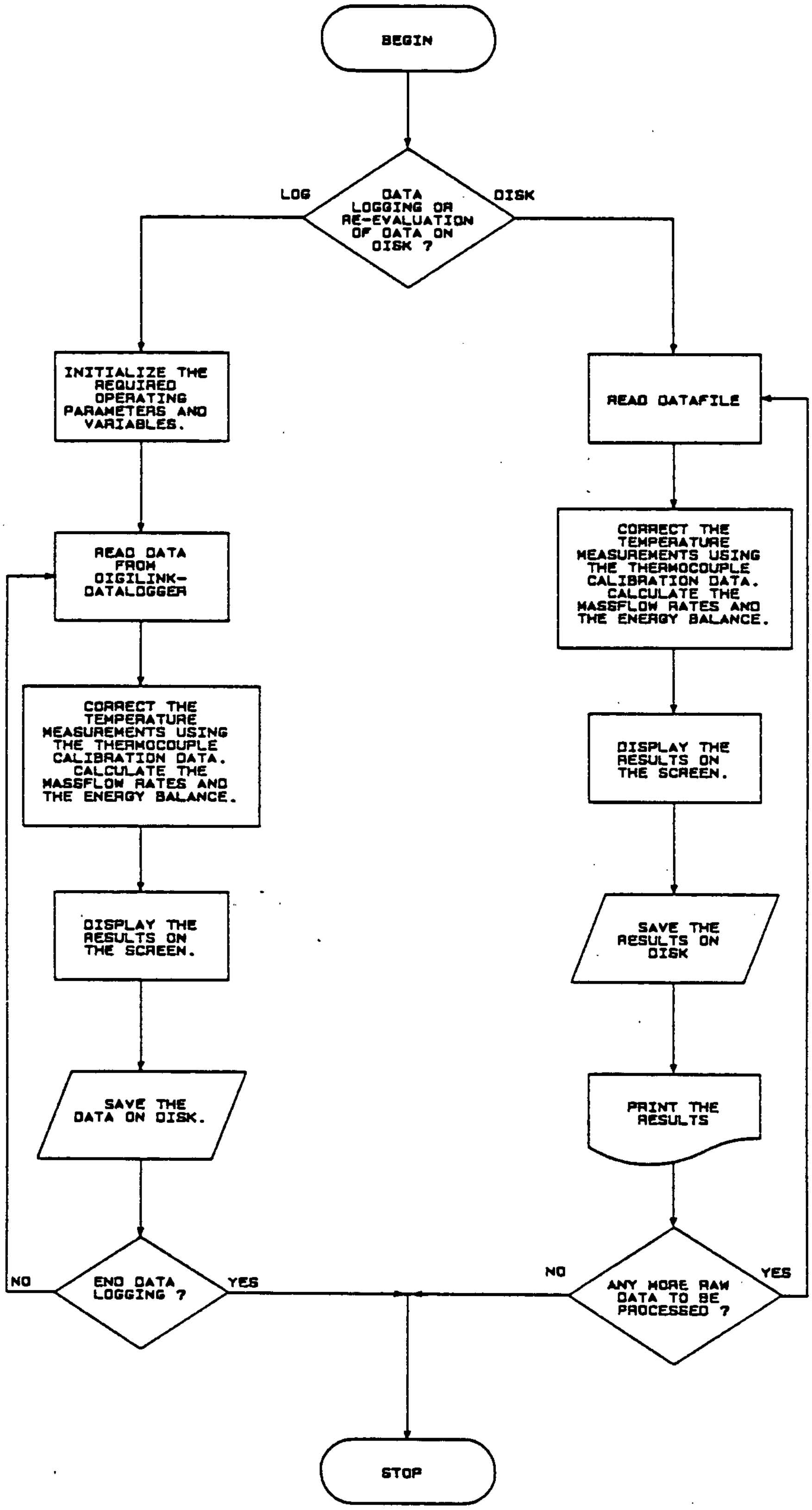

Figure 6.9 Program logic used in the data logging program.

The inlet process water temperatures used for all the tests lie between 38°C and 55°C which would be the normal operating temperatures for evaporative coolers. Since the tunnel draws in fresh atmospheric air the tests were all conducted without any control over the inlet air conditions.

The swing angle of the inclined test section was always set to ensure complete wetting of all the tube rows. The water distribution on the tubes is dependant on the air and the recirculating water massflow rates which implies that the optimum swing angle is a function of the air and recirculating water massflow rates.

The only parameters which were freely variable were the massflow rates of the air, process water and the recirculating water. The process water massflow rate could be varied between 5 and 16 kg/s. For the upright test section the following massflow ranges were possible, $1 \leq m_w \leq 4$ kg/s and $1 \leq m_a \leq 6$ kg/s and for the inclined test section the following massflow ranges were covered: $1 < m_w < 7$ kg/s and $1 < m_a < 12$ kg/s.

Upon starting a new test the hot process water was allowed to circulate through the test section and back to the hot water reservoir to warm the piping and the test section. A low air massflow rate through the tunnel ensured that the tunnel walls were sufficiently warm in order to shorten the time needed to reach a steady operating condition when the cooler operates as an evaporative cooler. After about five minutes the recirculating water was started and the mass flow set to the required flow rate. The air massflow rate was then increased to the required flow rate and the make-up line to the recirculating water sump was closed off.

The tests were run until the following stabilization criteria were met

i) an energy balance of better than 5 % and

ii) the difference between the inlet and outlet recirculating water temperatures stabilized.

6.4 Observations and results

As expected it was found that the recirculating water flow was dragged downstream by the cross-flow air at high air velocities which meant that the bottom tubes facing airstream started to form dry patches.

At free stream air speeds of up to 1,25 m/s ($m_a \approx 6$ kg/s) the distribution of recirculating water among the tube rows in the upright test section was still good. As the air velocity increased the first few tube tows received less and less recirculating water until they ran completely dry. The obvious solution to this problem was to swing the test section through a small angle in order to align each horizontal tube below and slightly downstream of the previous tube above it. It was observed that the recirculating water flowed from one tube to the next in the form of evenly spaced columns or droplets. It was also noted that a recirculating water column falling from a tube would adhere to the tube below if it only touched the lower tube. If the airspeed was just slightly higher the deflected column would miss the lower tube completely and it would be swept away. This phenomena is graphically illustrated in figure 6.10.

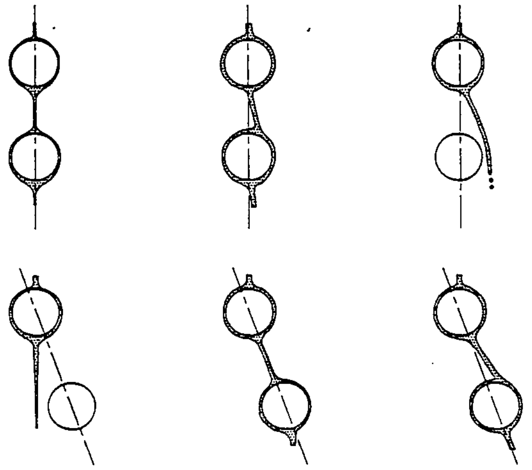

Figure 6.10 Column deflection by cross-flow airstream
a) $v_\infty = 0$ m/s b) $v_\infty \approx 1$ m/s c) $v_\infty > 1,5$ m/s

If the lower tube was placed slightly downstream of the upper tube the water column which would just be swept away in an upright test section, would strike the lower tube and a good distribution of recirculating water would still be obtained.

Yung et al. [80YU1] studied the problem of liquid entrainment by a cross-flow airstream where the liquid falls from one tube to the next in either a droplet or a column mode. They presented a criterion for the onset of the column formation and they derived equations for the calculation of droplet and column deflection due to the gas cross-flow.

Based on the work of Yung et al. [80YU1] and the current test section dimensions ($P_t = 2 \times d_o$, $P_d = 2 \times d_o$), two graphs were plotted to determine the deflection of the recirculating water flowing in droplet and column modes respectively. According to Yung et al. the liquid flow in the droplet mode consists of a primary drop and four or five smaller secondary drops. The smallest drops are obviously swept away first by the cross-flowing air stream.

From figure 6.11 it can be seen that the smallest drops are swept away from the lower tube in the upright test section at a free stream velocity of about 2 m/s, if the test section is inclined at 18° the smallest drops would be swept away only at a free stream velocity of 3 m/s.

Figure 6.12 shows the deflection of the water flow in the column flow mode. According to the criterion given by Yung et al. the column mode starts at $\Gamma \approx 95$ kg/m/hr for water at 40°C. It can be seen from figure 6.12 that the maximum allowable free stream velocity (before the water column is swept away) increases dramatically by inclining the test section through relatively small angles from the vertical.

As the maximum obtainable free stream air velocity in the tunnel is 3

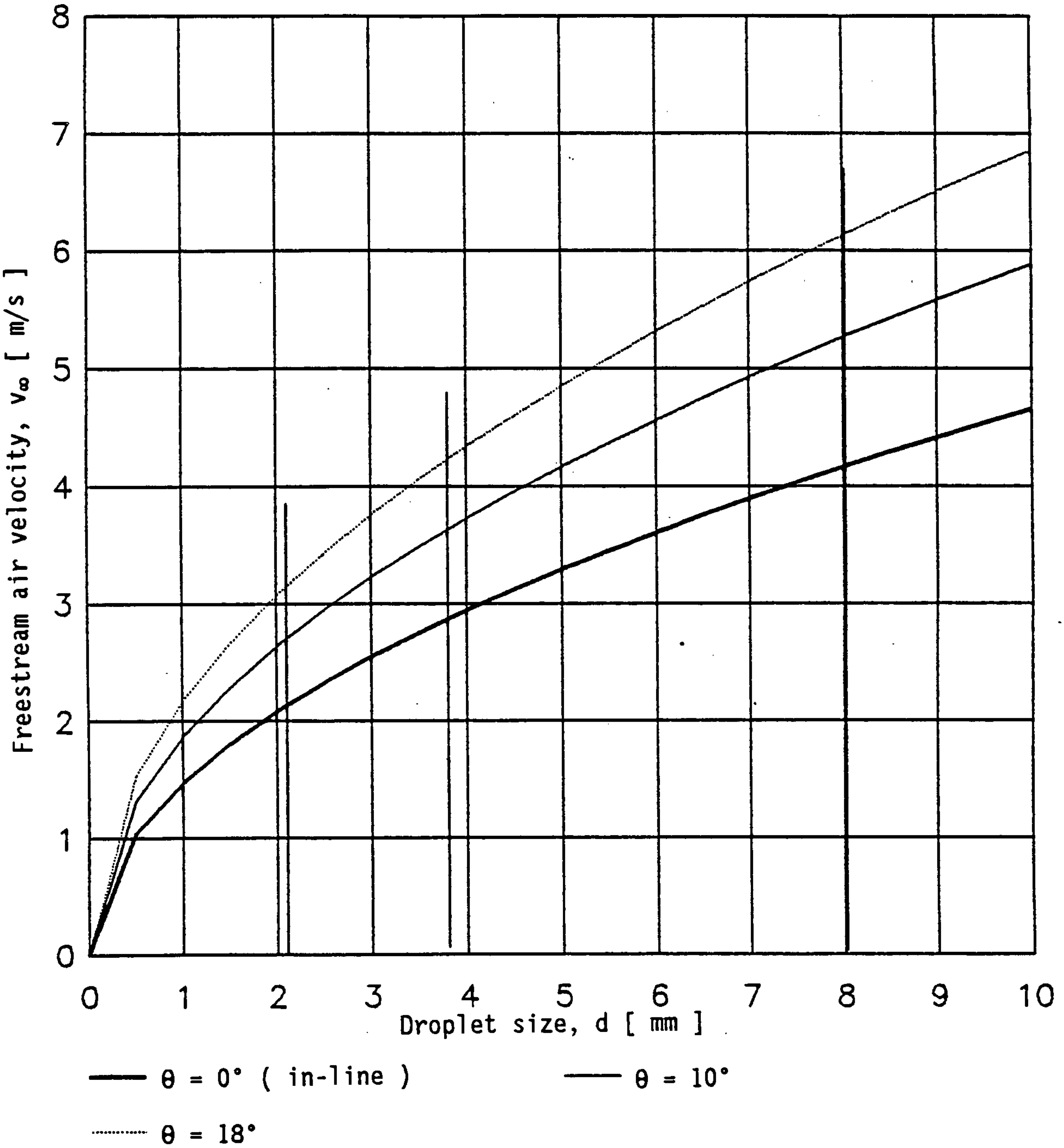

Figure 6.11 Maximum allowable freestream air velocity versus droplet size for droplet deflection for in-line and inclined tubes.

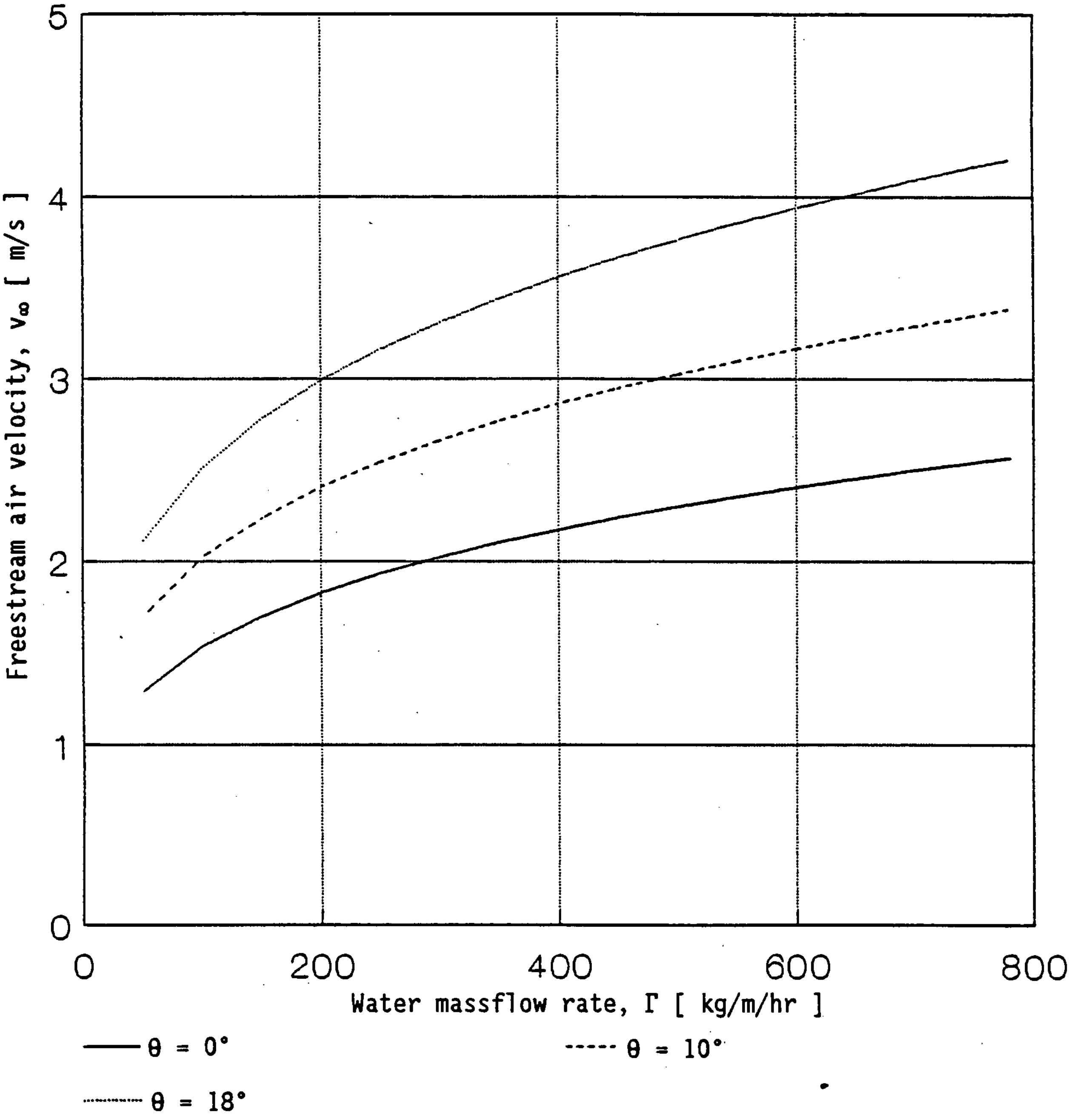

Figure 6.12 Maximum allowable freestream air velocity versus water massflow rate for column deflection for in-line and inclined tubes.

m/s it can be deduced from figures 6.11 and 6.12 that a swing angle of 18,75° should be sufficient to ensure very little water entrainment by the airstream.

Since there was some uncertainty as to what size and how many of the small spray holes should be drilled along the top edge of copper tubes in the spray tubes, it was decided to drill 40 holes of 1,5 mm diameter along the top of each copper tube. These spray tubes were then used in the tests of the upright test section and this limited the recirculating water massflow rate to 4 kg/s.

The second test section used tubes spaced in a 2 x d_0 triangular array but the whole tube bundle could be swung through an angle of up to 18,75° from the vertical. Much higher air velocities were possible without influencing the recirculating water distribution when using the inclined test section than was possible with the upright test section.

The holes in the top of the copper tubes inside the spray tubes were increased to 2,25 mm in diameter and this allowed recirculating water massflow rates of up to 7 kg/s.

The recirculating water outlet temperatures were measured using the film bulk temperature probes and the averaging temperature measuring trough installed under the bottom row of tubes along the middle of the test section. It was found that the recirculating water temperatures measured with the trough was always higher than the reading taken with the film temperature measuring probes. This was probably due to insufficient mixing of the water in the trough. The temperature which was measured in the trough was not used in further calculations.

The results of the tests are presented in table 6.1. Tests 908.1 to 1808.3 were conducted on the upright test section and tests 2610.1 to 411.8 were conducted on the inclined test section.

Table 6.1 Experimental results of the tests conducted on the inclined and upright test sections.

Test	P_{atm}	T_{pi}	T_{po}	T_{wi}	T_{wo}	T_{aidb}	T_{aiwb}	m_p	m_a	m_w
908.1	101.690	48.900	45.024	43.411	43.720	15.602	10.777	14.285	2.309	3.905
908.2	101.690	46.746	43.394	41.778	42.110	15.223	10.701	14.285	2.323	2.996
908.3	101.690	44.982	42.099	40.396	40.814	16.583	11.550	14.245	2.377	1.603
1508.1	101.098	47.054	40.160	38.562	38.581	19.064	14.760	10.298	4.997	3.897
1508.2	101.098	45.507	39.292	37.645	37.675	20.064	14.671	10.373	5.013	2.996
1508.3	101.098	44.361	38.873	36.904	37.167	20.131	14.728	10.384	5.021	1.603
1508.4	100.584	47.276	42.204	40.254	40.493	19.340	15.348	14.861	4.960	3.902
1508.5	100.584	46.193	41.441	38.959	39.746	18.631	15.040	14.836	4.968	2.996
1608.1	100.566	49.998	42.186	40.406	40.380	19.266	14.628	10.134	4.800	3.916
1608.2	100.566	47.915	40.896	39.047	39.170	19.362	14.485	10.131	4.847	2.996
1608.3	100.566	45.980	39.888	37.442	38.060	18.910	14.273	10.089	4.904	1.603
1608.4	100.566	43.999	39.719	38.584	38.619	18.281	14.104	9.783	2.448	3.858
1608.5	100.566	43.160	39.166	37.894	38.013	18.258	13.932	9.808	2.487	2.996
1608.6	100.566	42.054	38.480	37.022	37.315	17.963	13.745	9.774	2.512	1.603
1708.1	100.792	50.024	41.334	39.385	39.328	18.588	14.052	10.470	6.021	3.837
1708.2	100.792	47.709	39.940	37.876	38.002	17.964	13.936	10.483	6.063	2.996
1708.3	100.792	45.573	38.908	36.231	36.879	17.553	13.679	10.457	6.135	1.603
1808.1	100.524	52.734	44.467	42.757	42.747	14.959	11.519	9.366	3.482	3.824
1808.2	100.524	50.276	42.900	40.985	41.220	14.992	11.381	9.402	3.553	2.996
1808.3	100.524	47.700	41.391	39.120	39.630	14.701	11.226	9.469	3.691	1.603
2610.1	100.919	47.582	39.508	36.747	37.515	23.563	14.967	12.420	8.011	6.861
2610.2	100.919	41.847	35.680	33.213	34.021	22.971	14.724	12.358	8.160	5.824
2610.3	100.919	40.266	34.861	32.466	33.186	22.423	14.579	12.289	8.224	4.653
2610.4	100.919	37.563	33.157	30.612	31.524	21.791	14.345	12.100	8.334	2.996
2610.5	100.581	53.087	39.720	35.863	36.975	11.773	9.773	10.204	10.002	6.927
2610.6	100.581	43.936	35.806	32.424	33.494	12.032	9.724	12.169	10.211	5.807
2610.7	100.581	40.691	33.882	30.641	31.533	11.274	9.355	12.467	10.305	4.828
2610.8	100.583	37.444	32.043	28.557	29.495	10.549	8.911	13.027	10.443	2.996

Table 6.1 (cont.) Experimental results of the tests conducted on the inclined and upright test sections.

Test	P_{atm}	T_{pi}	T_{po}	T_{wi}	T_{wo}	T_{aidb}	T_{aiwb}	m_p	m_a	m_w
2710.1	99.645	49.943	42.116	38.422	39.939	20.219	14.670	14.177	7.495	6.865
2710.2	99.645	47.739	40.660	37.083	38.559	19.265	14.232	14.150	7.580	6.078
2710.3	99.645	45.517	39.043	35.372	36.840	19.160	13.870	14.155	8.330	4.614
2710.4	99.645	42.530	37.159	33.472	34.795	17.465	12.755	13.911	8.344	2.996
2710.5	99.915	48.139	39.095	35.506	36.639	13.813	11.450	13.218	9.434	6.790
2710.6	99.915	46.476	38.211	34.591	35.851	14.160	11.544	13.233	9.505	5.916
2710.7	99.915	45.115	37.680	33.880	35.159	13.783	11.426	13.239	9.585	4.336
2710.8	99.915	42.943	36.608	32.474	33.961	13.722	11.553	13.246	9.688	2.996
2810.1	100.179	47.897	39.173	35.701	36.838	17.470	11.992	13.308	9.146	6.806
2810.2	100.179	45.588	37.962	34.456	35.772	17.432	11.945	13.330	8.938	5.784
2810.3	100.179	43.633	36.912	33.388	34.684	17.754	11.969	13.332	9.129	4.493
2810.4	100.179	42.355	36.488	32.624	33.893	17.985	11.590	13.319	8.945	2.996
2810.5	100.179	49.554	40.047	36.011	37.630	17.330	11.984	13.264	9.116	6.764
2910.1	101.020	52.764	43.028	38.679	40.270	15.283	10.074	15.062	9.039	6.778
2910.2	101.020	49.949	41.601	37.327	39.123	14.907	10.009	15.084	8.712	6.029
2910.3	101.020	47.744	40.552	36.249	38.109	15.087	10.314	15.032	8.600	4.614
2910.4	101.020	46.053	39.872	35.220	36.856	14.991	10.347	15.033	8.567	2.996
2910.5	101.020	43.083	36.993	33.079	34.686	13.659	9.674	13.820	8.591	3.609
3010.1	100.438	53.889	43.559	39.227	40.617	18.372	12.891	14.852	9.700	6.769
3010.2	100.438	50.774	41.966	37.780	39.344	18.234	13.022	14.806	9.330	5.908
3010.3	100.438	48.785	40.916	36.710	38.249	17.672	12.773	14.801	9.418	4.742
3010.4	100.438	45.843	39.650	35.300	36.775	17.523	12.836	14.816	9.012	2.996
3010.5	100.438	44.344	38.326	34.291	35.927	16.924	12.536	14.804	9.014	3.610
3110.1	100.290	55.714	44.766	41.385	42.467	18.002	15.140	13.177	8.098	6.737
3110.2	100.290	53.014	43.358	39.748	40.894	17.698	14.857	13.186	8.226	5.798
3110.3	100.290	50.257	41.663	38.149	39.284	17.414	14.404	13.164	8.329	5.143
3110.4	100.290	47.725	40.152	36.815	37.861	17.420	14.767	13.183	8.405	4.458
3110.5	100.290	45.858	39.103	35.688	36.785	16.519	14.219	13.178	8.424	3.610

Table 6.1 (cont.) Experimental results of the tests conducted on the inclined and upright test sections.

Test	P_{atm}	T_{pi}	T_{po}	T_{wi}	T_{wo}	T_{aidb}	T_{aiwb}	m_p	m_a	m_w
3110.6	100.290	44.720	38.688	34.895	36.036	17.643	14.697	13.174	8.482	2.485
111.1	100.262	52.173	44.284	42.061	42.330	25.168	15.669	13.633	5.898	6.782
111.2	100.262	50.291	43.173	40.914	41.316	25.481	15.588	13.650	5.969	5.970
111.3	100.262	48.443	42.064	39.725	40.151	24.476	15.149	13.674	6.069	4.952
111.4	100.262	46.364	40.730	38.401	38.854	24.377	15.179	13.674	6.079	3.935
111.5	100.262	44.069	39.061	36.900	37.274	23.021	14.703	13.630	6.141	3.610
111.6	100.262	42.560	37.492	35.768	36.050	23.167	14.823	13.638	6.092	5.957
111.7	100.262	41.775	36.850	35.267	35.524	23.039	14.886	13.659	6.170	6.643
211.1	100.189	46.369	39.095	37.120	37.275	12.652	10.376	12.925	6.926	6.596
211.2	100.189	44.156	37.644	35.730	35.870	12.532	10.484	12.910	7.005	5.956
211.3	100.189	43.026	36.907	34.934	35.174	12.395	10.413	12.900	7.013	5.310
211.4	100.189	41.779	36.091	34.099	34.377	12.612	10.572	12.922	7.051	4.906
311.1	100.874	52.728	44.045	41.575	41.930	16.847	11.558	13.852	6.531	6.795
311.2	100.874	50.625	42.843	40.297	40.695	16.566	10.940	13.806	6.609	5.530
311.3	100.874	47.377	40.937	38.306	38.757	15.870	11.186	13.801	6.747	4.005
311.4	100.874	45.163	39.361	36.810	37.223	16.918	11.537	13.784	6.801	3.610
411.1	100.710	52.419	43.927	41.375	41.614	16.437	11.777	14.970	7.144	6.774
411.2	100.710	50.419	42.711	40.236	40.439	16.495	11.461	14.957	7.282	5.795
411.3	100.710	48.072	41.141	38.587	38.959	17.169	11.870	14.926	7.365	5.087
411.4	100.710	47.141	40.754	38.121	38.425	17.347	11.417	14.945	7.457	3.974
411.5	100.710	45.961	40.225	37.375	37.739	17.859	12.122	14.864	7.537	2.996
411.6	100.710	44.146	37.969	36.053	36.082	17.757	11.913	14.853	7.567	6.793
411.7	100.710	42.613	37.095	35.122	35.250	18.317	12.273	14.737	7.633	5.574
411.8	100.710	40.559	35.829	33.748	33.964	19.093	12.803	14.472	7.757	4.192

The critical cross-flow velocity of the air was determined experimentally at various inclination angles of the test section. The experimental values are compared to the theoretical values based on the model by by Yung et al. [80YU1] in table 6.2 It can be seen that the theoretical maximum allowable cross-flow velocity is up to two times as as high as the experimental maximum. It should however be noted that it was difficult to determine accurately when the water falling from one tube to the next would just miss the lower tube since the water columns oscillated back and forth quite significantly.

Table 6.2 Experimentally determined critical cross-flow air velocity compared with theoretical values.

Angle	m_w [kg/s]	v_{crit} based on droplet deflection [m/s]	v_{crit} based on column deflection [m/s]	v_{crit} observed experimentally	$\frac{v_{crit,theo}}{v_{crit,exp}}$
0°	1,0	2,08	1,50	0,80	1,9
	4,0	2,08	2,12	1,00	2,1
	7,0	2,08	2,44	1,10	1,9
5°	1,0	2,36	1,73	1,31	1,3
	4,0	2,36	2,45	1,48	1,6
	7,0	2,36	2,81	1,53	1,5
10°	1,0	2,63	1,97	1,46	1,4
	4,0	2,63	2,79	1,56	1,7
	7,0	2,63	3,21	1,70	1,5
15°	1,0	2,90	2,23	1,70	1,3
	4,0	2,90	3,16	1,80	1,6
	7,0	2,90	3,63	1,95	1,5
18,75°	1,0	3,11	2,45	2,04	1,2
	4,0	3,11	3,47	2,16	1,4
	7,0	3,11	3,99	2,40	1,3

The single phase pressure drop was measured across the tube bank and the results are tabulated in table 6.3 The pressure drop measurements were all taken across the movable test section. The upstream pressure tappings were placed in a section of the tunnel where the cross section area was $4m^2$, but the downstream readings were taken behind the test section where the cross section area of the tunnel was $3{,}429m^2$. The measured pressure drop values were corrected to take the contraction of the flow into account.

Table 6.3 Measured pressure drop across the dry tube bundle.

m_a	dP_a	$\Delta p_a(corr)$
9.55	41.00	39.82
9.43	41.50	40.35
9.51	41.50	40.33
8.86	37.00	35.98
8.94	37.00	35.96
8.96	37.50	36.46
8.45	33.50	32.58
8.40	33.50	32.59
7.77	28.50	27.72
7.76	28.80	28.02
7.12	24.80	24.14
7.18	25.00	24.33
6.72	21.40	20.81
6.62	21.00	20.43
6.59	21.00	20.44
5.94	18.00	17.54
5.46	15.00	14.61
4.81	12.00	11.70
4.82	12.20	11.90
4.16	9.50	9.28
4.22	10.00	9.77
3.60	7.20	7.03
3.57	7.00	6.83
3.04	5.20	5.08
3.01	5.20	5.08
2.41	3.90	3.82
2.46	3.80	3.72

The two phase pressure drop across the test section was measured for various combinations of air and recirculating water massflow rates. In all the pressure drop readings it was ensured that the recirculating water distribution through the tube bundle was uniform.

Table 6.4 Measured pressure drop across the wet tube bundle.

m_w	m_a	$\Delta p_{tp}(corr)$
5.71	8.22	76.13
5.23	8.26	72.12
4.47	8.38	70.09
4.48	8.41	68.08
6.58	7.44	67.28
5.88	7.48	64.28
3.61	8.36	62.10
3.61	8.46	62.07
4.94	7.50	61.27
5.45	7.23	58.32
4.43	7.42	56.79
6.64	6.98	55.87
4.74	7.24	55.32
5.89	6.89	53.89
5.91	6.94	53.38
6.76	6.55	52.94
6.90	6.41	51.97
6.87	6.35	51.48
5.38	6.92	51.38
5.35	7.01	51.36
3.64	7.33	51.30
6.98	6.31	50.48
6.85	6.31	49.98
3.00	7.45	49.28
4.85	6.96	48.87
4.84	7.10	48.85
3.00	7.24	48.32
6.09	6.51	47.45
6.12	6.42	47.27
6.63	6.08	46.02
6.61	6.07	45.52
2.00	7.53	45.27
5.31	6.40	44.97
2.00	7.37	44.30
6.87	5.93	44.05
4.00	6.63	43.93
3.61	6.76	43.91
5.97	6.18	43.01
6.82	5.83	42.56

Table 6.4 (cont.) Measured pressure drop across the wet tube bundle.

m_w	m_a	Δp_{tp}(corr)
5.91	6.11	42.02
4.82	6.38	41.97
4.61	6.50	41.95
4.63	6.44	41.86
5.62	6.26	41.49
6.90	5.82	41.06
6.91	5.84	41.06
5.93	5.93	40.55
4.20	6.34	39.48
4.02	6.38	39.47
5.92	5.92	38.55
5.96	5.93	38.54
4.98	6.01	38.53
4.98	6.02	38.53
4.97	5.98	38.04
4.33	6.30	37.99
4.37	6.32	37.98
3.61	6.36	37.48
4.01	6.12	36.02
3.87	6.08	35.77
3.94	6.06	35.52
3.00	6.31	35.48
5.07	5.87	35.05
3.61	6.10	34.52
3.61	6.11	34.52
3.61	6.13	34.51
2.00	6.11	29.52
3.00	5.82	29.46
4.26	5.34	27.63
4.18	5.38	27.63
1.10	6.01	26.03
2.00	5.77	25.77
3.24	5.44	25.62
1.10	5.96	25.54
3.28	5.40	25.37
6.87	4.35	23.36
6.81	4.38	23.25
6.00	4.46	22.24
5.96	4.44	21.95

Table 6.4 (cont.) Measured pressure drop across the wet tube bundle.

m_w	m_a	$\Delta p_{tp}(corr)$
1.10	5.40	21.62
5.00	4.44	21.25
1.10	4.44	21.25
5.32	4.46	21.24
5.06	4.41	20.95
1.10	5.32	20.63
6.77	4.06	19.79
6.72	4.01	19.29
4.13	4.46	19.24
4.21	4.32	18.96
5.96	4.10	18.78
3.61	4.42	18.25
5.36	4.01	17.59
4.91	4.06	17.29
3.00	4.43	17.25
4.83	4.03	17.09
4.80	4.00	16.89
4.54	4.00	16.69
2.00	4.49	15.49
3.61	4.03	15.29
1.10	4.52	14.74
3.00	4.05	14.54
2.00	4.13	13.28
2.00	3.99	12.79
1.10	4.16	12.58
1.10	4.09	12.28
1.10	4.10	12.28
6.86	3.08	10.88
6.90	3.06	10.63
6.83	3.03	10.38
6.51	3.06	10.38
6.28	3.01	10.08
6.12	3.10	9.88
5.04	3.06	9.38
4.34	3.10	8.88
3.61	3.10	8.38
3.00	3.10	7.88
2.00	3.10	6.88
1.10	3.10	6.38
1.10	3.12	6.37

Table 6.4 shows the measured two phase pressure drop across the tube bundle at various combinations of air and recirculating water massflow rates.

6.5 Determination of coefficients and correlations

A Fortran program, called COEFFS, was developed to calculate the required heat and mass transfer coefficients from the experimental data. This program uses the routines of the cross-flow evaporative cooler rating program iteratively to determine the coefficients from the known experimental temperatures and massflow rates. The program logic for COEFFS is shown in figure 6.13.

All three analytical models available in the rating program have been incorporated into COEFFS, i.e. the Merkel, the Improved Merkel and the Poppe models.

As discussed in Chapter 3 and Appendixes C and H, the basic relations for heat and mass transfer between the water film and the tube and between the water film and the air can be based on the bulk film temperature

$$dq = h_w \left(T_{wall} - T_w \right) dA \tag{6.5}$$

$$dm_w = h_D \left(i_{asw} - i_a \right) dA \tag{6.6}$$

or it can be based on film/air interface temperature as follows

$$dq = h_{wi} \left(T_{wall} - T_i \right) dA \tag{6.7}$$

$$dm_w = h_{Di} \left(i_{asi} - i_a \right) dA \tag{6.8}$$

The program, COEFFS, calculates h_D and h_w values using the Merkel model by default but the user can calculate h_D and h_D values

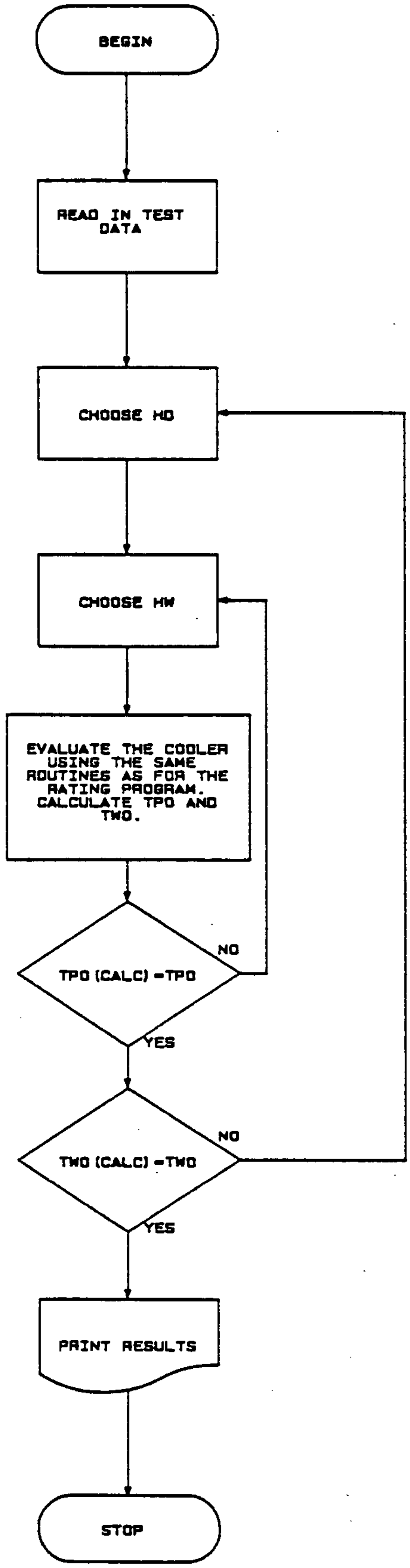

Figure 6.13 Flow chart for program COEFFS, showing program logic for the iterative determination of the required coefficients from the test data.

using either the Poppe or the Improved Merkel models as well.
The program also allows the calculation of h_{wi} and h_{Di} by employing the theory described in Appendix H to calculate the interface temperature.
The program was used to calculate the coefficients h_D and h_w based on the bulk recirculating water temperature using both the Merkel and the Poppe models. The coefficients h_{Di} and h_{wi} were also calculated using the Merkel model. Table 6.5 summarizes the calculated coefficients for each of the tests.

The following correlations were obtained through the use of Lotus 123

$$h_{D,Merkel} = 6,72238 \times 10^{-5} \; Re_a^{0,62} \; Re_w^{0,20} \tag{6.9}$$

$$h_{D,Poppe} = 7,36673 \times 10^{-5} \; Re_a^{0,61} \; Re_w^{0,21} \tag{6.10}$$

$$h_{Di,Merkel} = 5,07155 \times 10^{-5} \; Re_a^{0,66} \; Re_w^{0,20} \tag{6.11}$$

where the Reynolds numbers are defined as

$$Re_a = \frac{\rho_a}{\mu_a} \left(\frac{m_a}{A_{min}} \right) \tag{6.12}$$

and

$$Re_w = \frac{4\Gamma}{\mu_w} \tag{6.13}$$

The correlations for the mass transfer coefficient holds in the following ranges

$2500 < Re_a < 13500$

$230 < Re_w < 1100$

Table 6.5 Calculated film heat transfer coefficients and mass transfer coefficients.

Test	Γ	Re_a	Re_w	h_D(Merkel)	h_w(Merkel)	h_D(Poppe)	h_w(Poppe)	h_{Di}(Merkel)	h_{wi}(Merkel)
908.1	351.45	2546.7	638.97	0.02984	3701.57	0.03237	3738.40	0.03139	2358.26
908.2	269.64	2565.0	475.67	0.02781	3279.79	0.03005	3309.86	0.02922	2085.23
908.3	144.27	2616.5	247.98	0.02536	2958.66	0.02730	2990.27	0.02657	1882.92
1508.1	350.73	5485.6	582.05	0.05059	4144.73	0.05397	4211.76	0.05360	2673.18
1508.2	269.64	5484.8	439.57	0.04812	3718.14	0.05127	3776.73	0.05107	2389.72
1508.3	144.27	5492.9	231.79	0.04238	3202.19	0.04508	3256.89	0.04502	2048.53
1508.4	351.18	5446.2	602.00	0.04795	3843.58	0.05135	3891.41	0.05123	2465.37
1508.5	269.64	5464.5	450.92	0.04526	3660.53	0.04838	3705.04	0.04830	2340.62
1608.1	352.44	5265.2	605.91	0.04942	3856.67	0.05293	3917.75	0.05284	2495.13
1608.2	269.64	5313.8	451.69	0.04690	3844.75	0.05011	3910.86	0.04985	2474.64
1608.3	144.27	5382.2	234.26	0.04167	3166.47	0.04439	3221.39	0.04441	2031.56
1608.4	347.22	2691.4	576.47	0.03362	3329.44	0.03595	3358.52	0.03527	2132.26
1608.5	269.64	2733.7	441.71	0.03156	3013.84	0.03372	3047.29	0.03316	1927.19
1608.6	144.27	2762.9	232.33	0.02793	2909.93	0.02977	2950.46	0.02918	1860.75
1708.1	345.33	6612.2	582.25	0.05830	4045.10	0.06227	4116.66	0.06262	2619.84
1708.2	269.64	6670.4	441.56	0.05601	3888.85	0.05964	3958.03	0.05995	2507.91
1708.3	144.27	6755.7	228.72	0.04942	3204.03	0.05249	3260.43	0.05303	2056.25
1808.1	344.16	4367.2	618.25	0.04648	5428.06	0.05018	5505.13	0.04887	3530.10
1808.2	269.64	3930.7	468.65	0.03911	3890.13	0.04203	3958.96	0.04136	2505.29
1808.3	144.27	4086.3	242.01	0.03558	3307.30	0.03808	3365.16	0.03764	2124.41
2610.1	617.49	9828.9	989.02	0.08264	4494.71	0.08829	4524.31	0.08976	2942.95
2610.2	524.16	10026.3	781.74	0.07995	4265.03	0.08482	4284.49	0.08597	2774.54
2610.3	418.77	10120.0	614.97	0.07337	3738.80	0.07769	3752.91	0.07894	2415.25
2610.4	269.64	10271.9	380.82	0.06497	3179.74	0.06855	3188.51	0.06983	2040.57
2610.5	623.43	12642.5	981.15	0.09782	4093.38	0.10414	4133.51	0.10854	2715.03
2610.6	522.63	12894.9	766.82	0.08988	3709.77	0.09526	3730.41	0.09853	2420.50
2610.7	434.52	13039.0	614.07	0.08858	3150.60	0.09363	3165.21	0.09762	2047.77
2610.8	269.64	13236.8	364.33	0.08364	2466.87	0.08818	2473.79	0.09325	1588.54

Table 6.5 (cont.) Calculated film heat transfer coefficients and mass transfer coefficients.

Test	Γ	Re_a	Re_w	h_D(Merkel)	h_w(Merkel)	h_D(Poppe)	h_w(Poppe)	h_{Di}(Merkel)	h_{wi}(Merkel)
2710.1	617.85	9291.3	1022.58	0.07294	4294.84	0.07816	4319.65	0.07983	2806.67
2710.2	547.02	9418.4	881.98	0.07151	4122.43	0.07641	4143.90	0.07796	2685.04
2710.3	415.26	10347.4	647.14	0.07235	3757.88	0.07706	3776.44	0.07907	2434.31
2710.4	269.64	10402.3	404.29	0.06654	2882.88	0.07064	2893.07	0.07328	1855.04
2710.5	611.10	11876.2	954.90	0.09283	4181.99	0.09880	4209.57	0.10239	2739.34
2710.6	532.44	11953.8	816.77	0.08832	4099.57	0.09387	4124.41	0.09688	2677.03
2710.7	390.24	12067.0	590.03	0.08229	3540.67	0.08734	3560.42	0.09069	2296.25
2710.8	269.64	12201.5	396.04	0.07550	2872.04	0.07996	2883.95	0.08389	1853.50
2810.1	612.54	11387.3	960.89	0.09031	4325.63	0.09633	4353.78	0.09913	2831.03
2810.2	520.56	11128.8	796.36	0.08358	4143.25	0.08898	4166.89	0.09114	2700.23
2810.3	404.37	11355.3	605.26	0.07883	3677.37	0.08379	3694.75	0.08606	2385.14
2810.4	269.64	11111.6	397.27	0.07217	2700.67	0.07662	2711.54	0.08025	1739.77
2810.5	608.76	11354.9	960.90	0.09087	4513.10	0.09704	4546.70	0.09985	2962.38
2910.1	610.02	11299.8	1014.65	0.08730	4682.09	0.09368	4722.33	0.09636	3062.32
2910.2	542.61	10903.3	879.07	0.07917	4553.10	0.08478	4587.53	0.08647	2964.65
2910.3	415.26	10761.6	658.58	0.07227	4064.34	0.07723	4093.91	0.07879	2632.89
2910.4	269.64	10724.2	418.92	0.06859	2719.29	0.07315	2731.76	0.07701	1749.46
2910.5	324.81	10790.3	483.09	0.06984	3364.13	0.07426	3379.53	0.07607	2168.78
3010.1	609.21	12058.8	1024.06	0.09335	4545.70	0.10010	4588.45	0.10418	2982.27
3010.2	531.72	11606.4	869.10	0.08551	4464.02	0.09149	4499.26	0.09429	2912.19
3010.3	426.78	11732.4	683.06	0.08139	3975.33	0.08690	4003.95	0.08991	2581.95
3010.4	269.64	11233.1	419.60	0.07130	2863.86	0.07592	2876.61	0.07990	1844.77
3010.5	324.90	11251.9	495.38	0.07224	3410.69	0.07685	3427.28	0.07928	2200.74
3110.1	606.33	10117.7	1061.79	0.08143	6394.40	0.09017	5576.01	0.09168	3618.45
3110.2	521.82	10282.6	885.95	0.07925	4854.96	0.08484	4902.00	0.08662	3171.81
3110.3	462.87	10412.8	762.03	0.07765	4505.11	0.08290	4541.70	0.08473	2931.35
3110.4	401.22	10514.1	643.49	0.07649	4187.04	0.08140	4216.47	0.08341	2713.28
3110.5	324.90	10558.7	509.54	0.07209	3682.75	0.07658	3705.29	0.07873	2381.59
3110.6	223.65	10601.5	345.20	0.06862	2773.41	0.07279	2784.53	0.07652	1782.76

Table 6.5 (cont.) Calculated film heat transfer coefficients and mass transfer coefficients.

Test	Γ	Re_a	Re_w	h_D(Merkel)	h_w(Merkel)	h_D(Poppe)	h_w(Poppe)	h_{Di}(Merkel)	h_{wi}(Merkel)
111.1	610.38	7209.5	1082.45	0.06858	5509.24	0.07406	5557.38	0.07365	3579.95
111.2	537.30	7288.1	932.62	0.06477	5345.74	0.06979	5388.56	0.06922	3465.03
111.3	445.68	7427.9	756.35	0.06152	4575.65	0.06610	4606.08	0.06599	2953.15
111.4	354.15	7442.9	585.90	0.05894	4202.60	0.06314	4228.18	0.06315	2702.66
111.5	324.90	7544.9	521.96	0.05784	3908.75	0.06176	3926.28	0.06190	2505.19
111.6	536.13	7482.8	842.16	0.06719	4868.84	0.07163	4888.91	0.07123	3139.77
111.7	597.87	7582.5	929.75	0.06850	5160.67	0.07293	5180.31	0.07234	3330.10
211.1	593.64	8737.5	957.84	0.07464	5264.20	0.07964	5301.91	0.07949	3413.92
211.2	536.04	8842.0	841.38	0.07312	4807.32	0.07780	4836.53	0.07788	3109.10
211.3	477.90	8855.2	738.21	0.07082	4710.36	0.07527	4736.94	0.07528	3039.09
211.4	441.54	8899.0	670.59	0.06948	4394.84	0.07374	4416.11	0.07394	2830.78
311.1	611.55	8141.5	1074.74	0.07195	5796.51	0.07752	5854.45	0.07712	3771.94
311.2	497.70	8238.5	853.87	0.06756	5097.19	0.07262	5142.31	0.07250	3301.38
311.3	360.45	8432.6	595.22	0.06233	4100.14	0.06670	4126.33	0.06716	2635.77
311.4	324.90	8475.9	521.03	0.06273	3730.59	0.06698	3750.53	0.06769	2398.35
411.1	609.66	8920.7	1067.42	0.07802	5486.96	0.08395	5537.88	0.08439	3564.67
411.2	521.55	9086.9	893.75	0.07493	5172.01	0.08045	5216.12	0.08083	3347.98
411.3	457.83	9176.3	760.16	0.07370	4843.60	0.07891	4878.82	0.07941	3126.66
411.4	357.66	9279.1	588.50	0.06884	4144.68	0.07366	4172.00	0.07458	2664.01
411.5	269.64	9373.4	437.25	0.06432	3360.34	0.06870	3377.03	0.07037	2154.23
411.6	611.37	9410.8	965.82	0.08382	4772.11	0.08939	4797.22	0.09017	3083.72
411.7	501.66	9481.1	777.86	0.07725	4370.83	0.08224	4391.06	0.08292	2820.29
411.8	377.28	9619.3	568.90	0.07044	3577.96	0.07478	3589.92	0.07596	2296.45

The mass transfer coefficient data and correlations are shown in figures 6.14, 6.15 and 6.16.
The film heat transfer coefficient was correlated through the use of Lotus 123 as

$$h_{w,Merkel} = 2946,494 \left(\frac{\Gamma}{d_o} \right)^{0,32} \tag{6.14}$$

$$h_{w,Poppe} = 2937,132 \left(\frac{\Gamma}{d_o} \right)^{0,33} \tag{6.15}$$

$$h_{wi,Merkel} = 1843,035 \left(\frac{\Gamma}{d_o} \right)^{0,35} \tag{6.16}$$

The correlations for the film coefficient holds in the following range

$140 < \Gamma < 650$ [kg/m/hr]

Tests 1808.1, 2610.4 - 2610.8, 2710.5, 2710.8, 2810.4, 2910.4 3010.4, 3110.1, 3110.6 and 0411.5 were not considered in the film coefficient correlation since they were either conducted with non-uniform recirculating water distribution or the tests did not stabilize due to limited hot water tank size.
At the higher air velocities the cooling capacity provided by the test section was so large that the hot water tank temperature cooled down to fast to ensure a completely stable test. This did not seem to influence the mass transfer coefficients significantly.

The film heat transfer coefficient data and correlations are shown graphically in figures 6.17, 6.18 and 6.19.

The single phase pressure drop measurements are shown graphically in figure 6.20 together with the single phase predictions by Jakob [38JA1] and Gaddis and Gnielinski [85GA1].

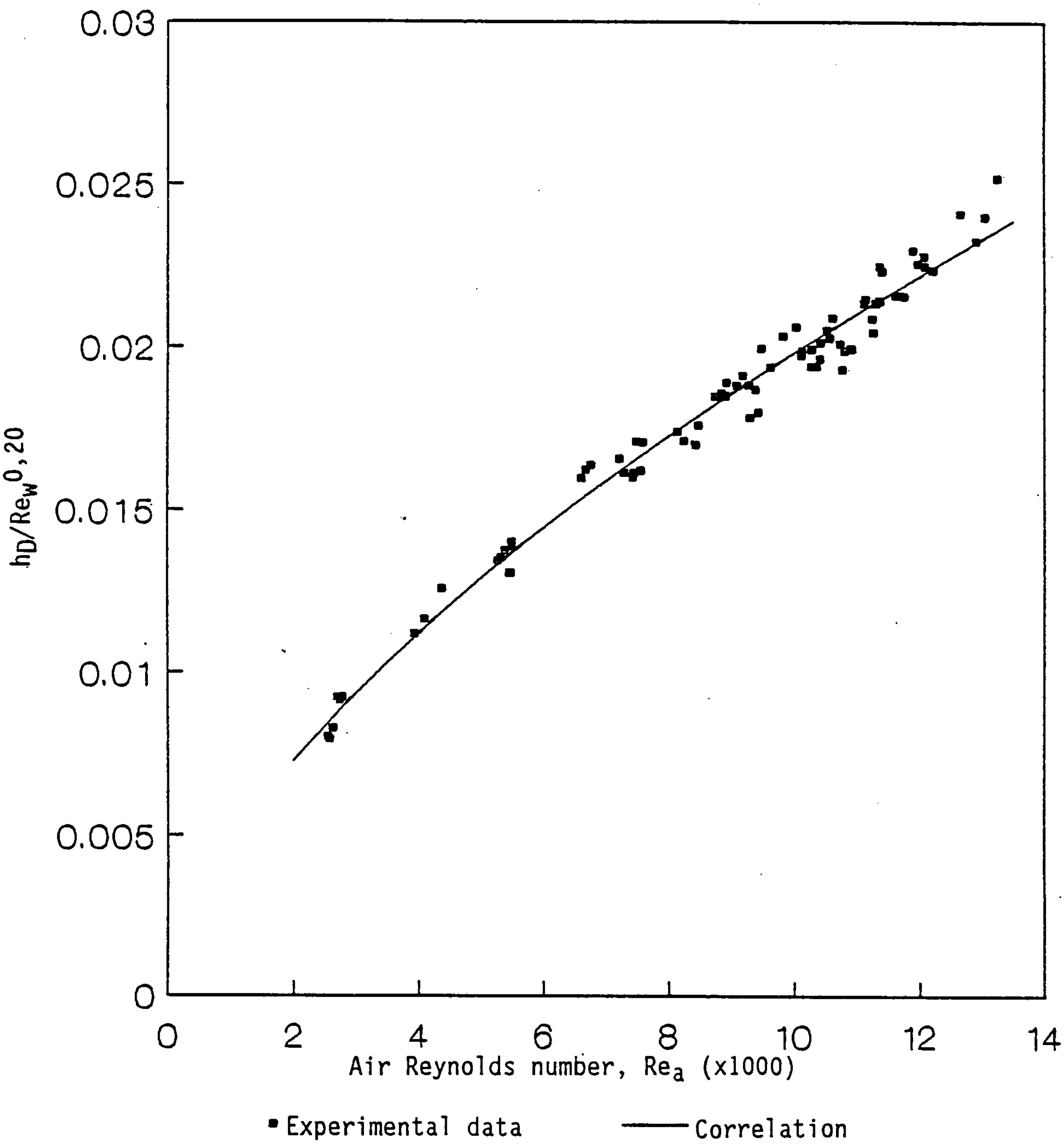

Figure 6.14 Experimentally determined mass transfer coefficients based on the Merkel model and the bulk recirculating water temperature, T_w.

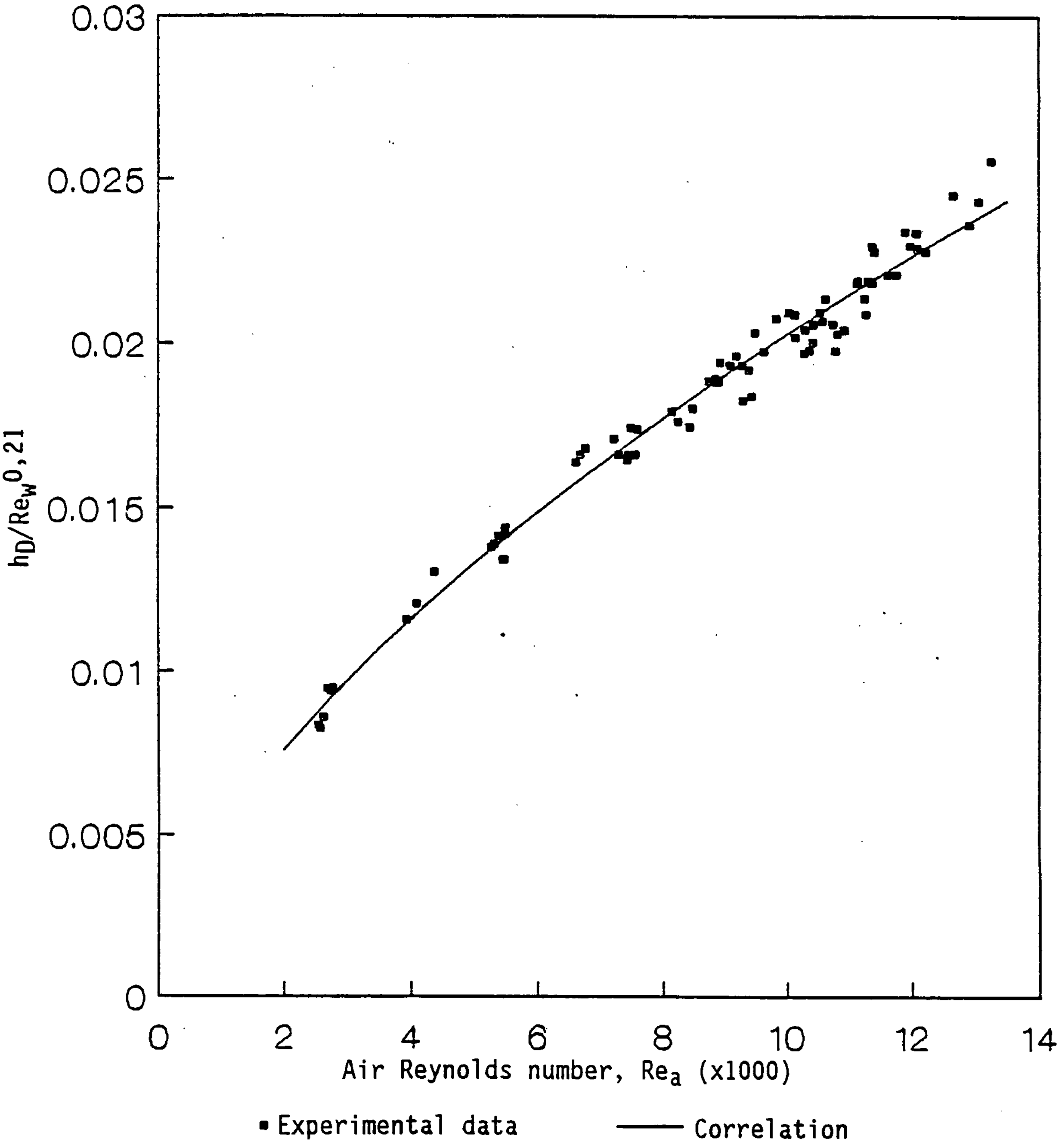

Figure 6.15 Experimentally determined mass transfer coefficients based on the Poppe model and the bulk recirculating water temperature, T_w.

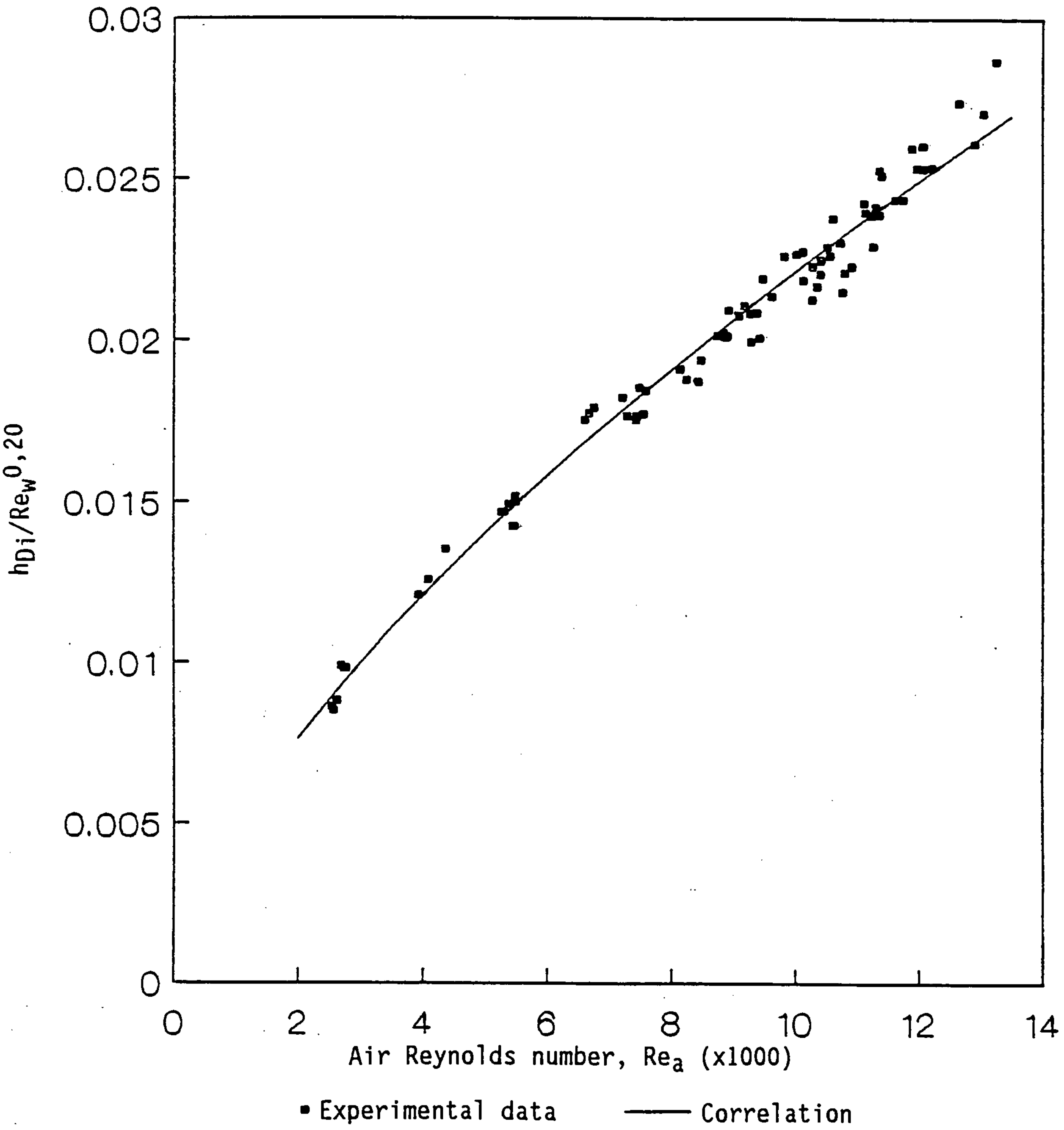

Figure 6.16 Experimentally determined mass transfer coefficients based on the Merkel model and the air/water interface temperature, T_i.

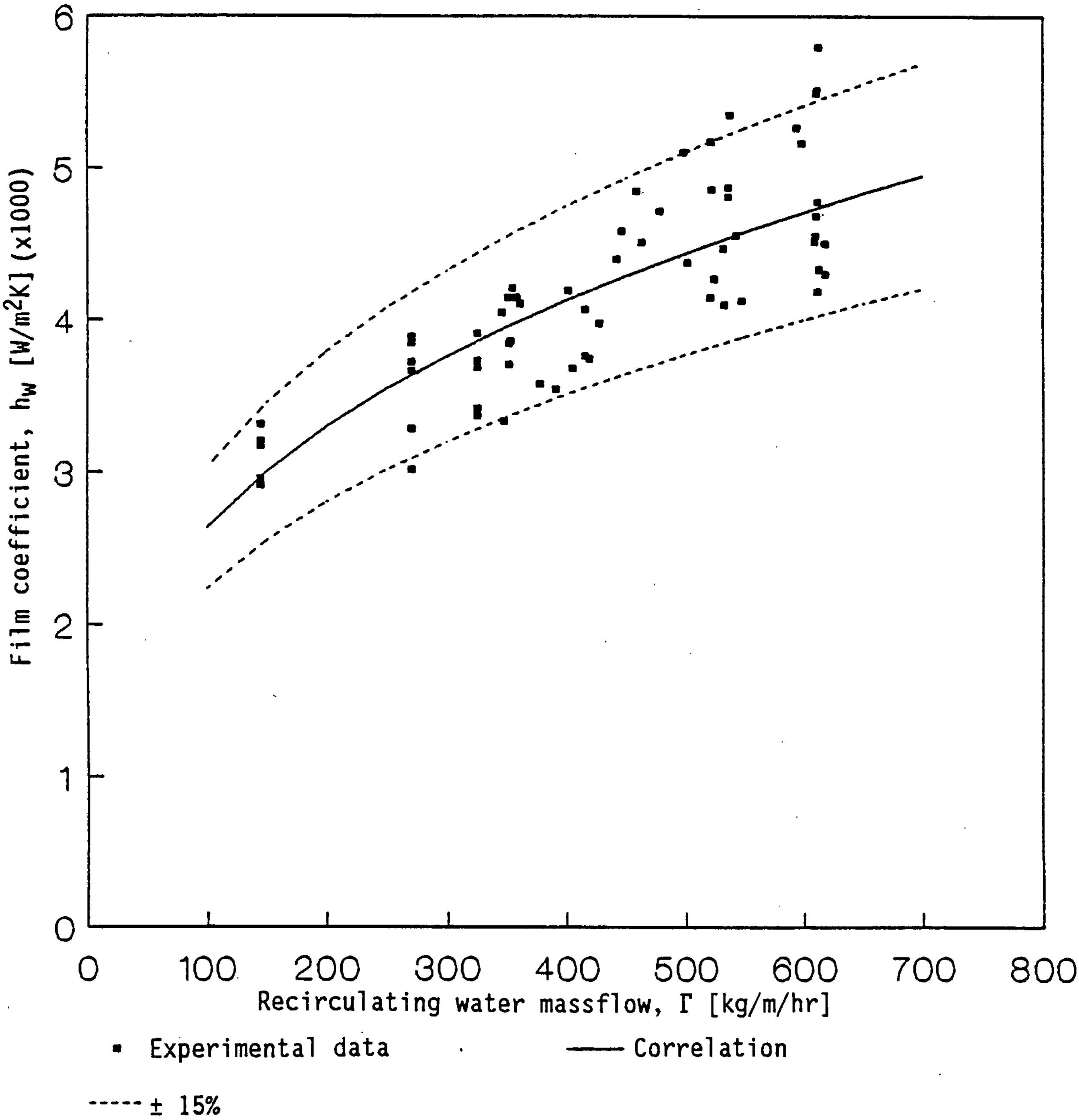

Figure 6.17 Experimentally determined film heat transfer coefficients based on the Merkel model and the bulk recirculating water temperature, T_w.

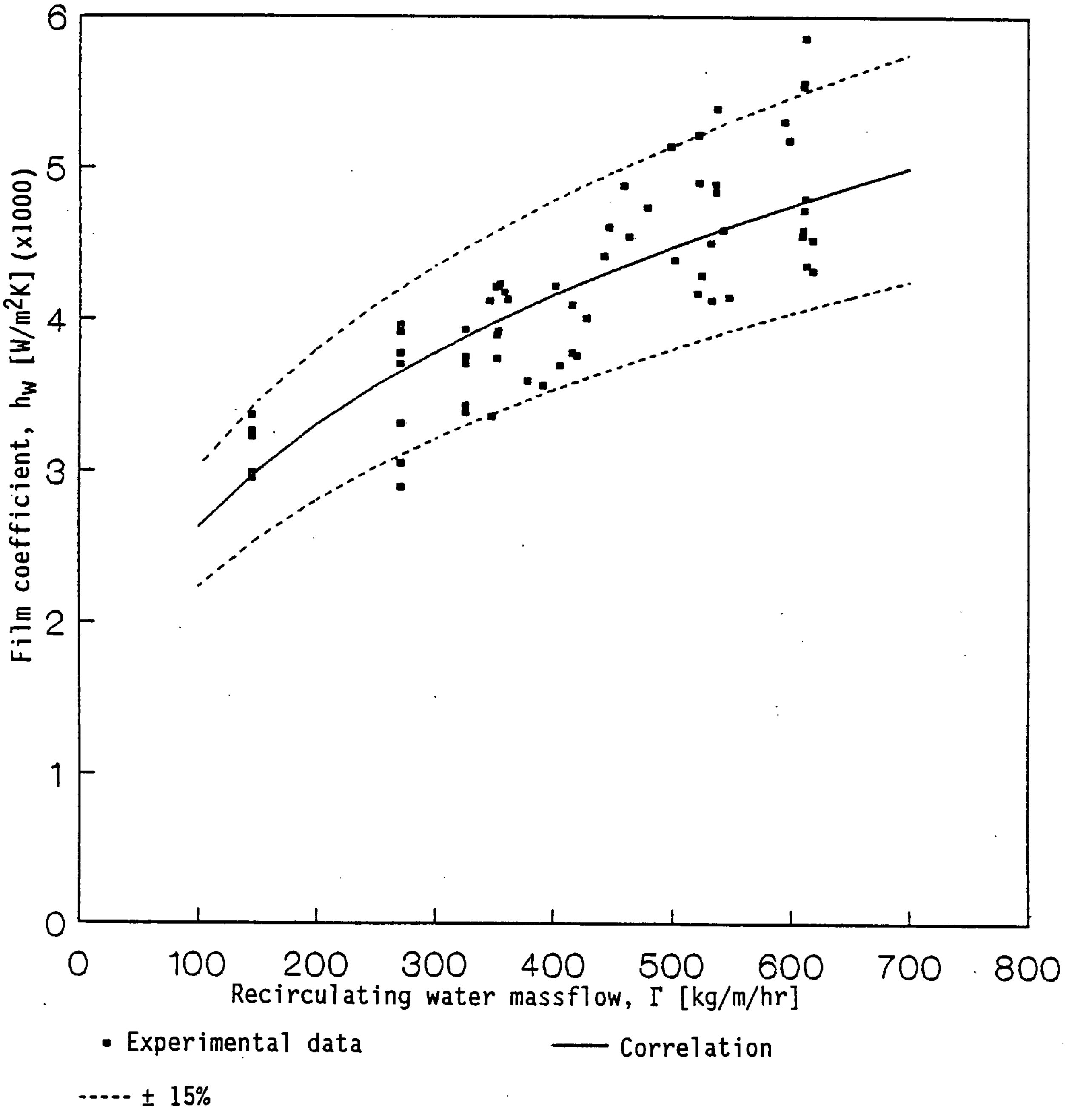

Figure 6.18 Experimentally determined film heat transfer coefficients based on the Poppe model and the bulk recirculating water temperature, T_w.

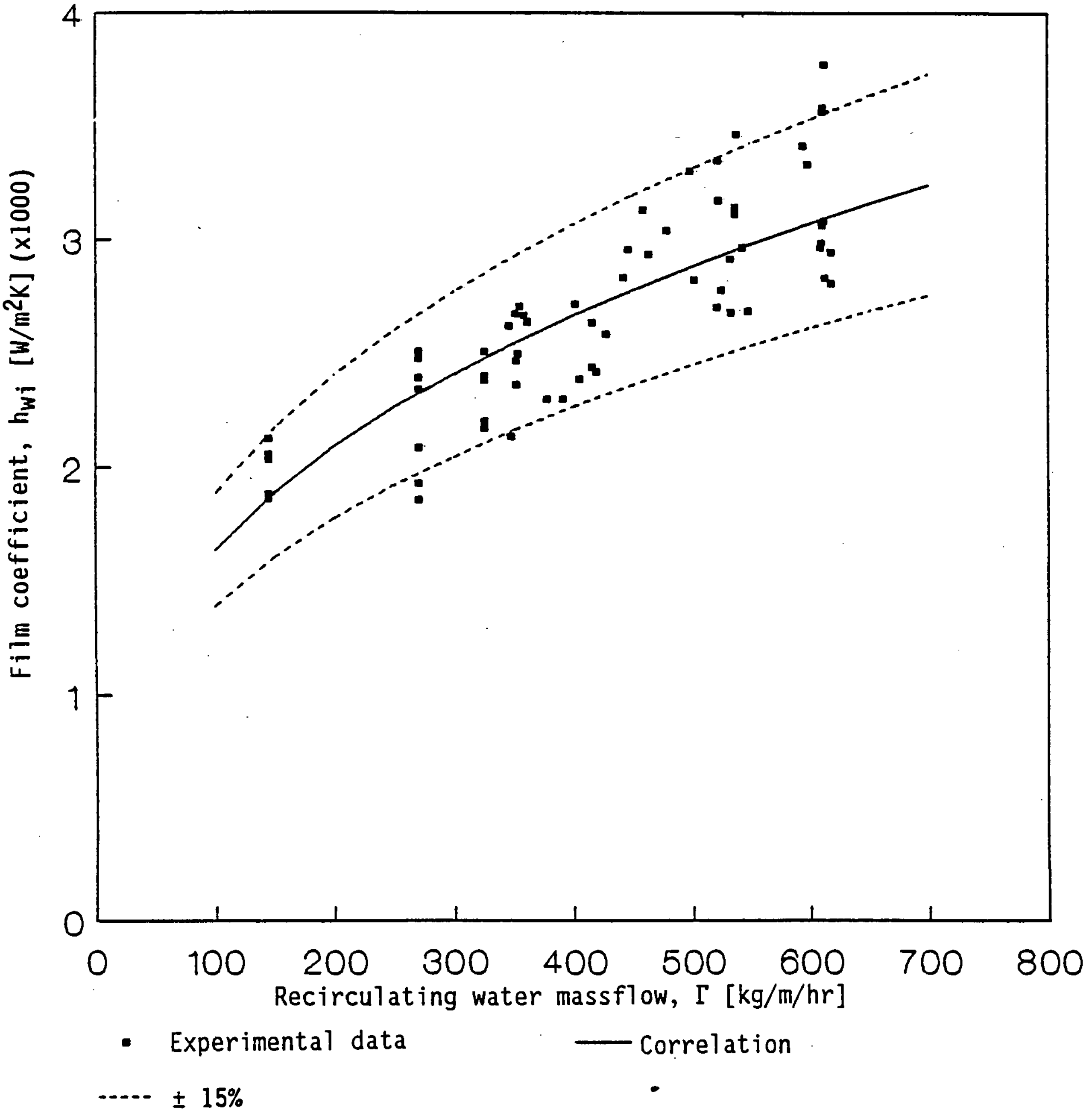

Figure 6.19 Experimentally determined film heat transfer coefficients based on the Merkel model and the air/water interface temperature, T_i.

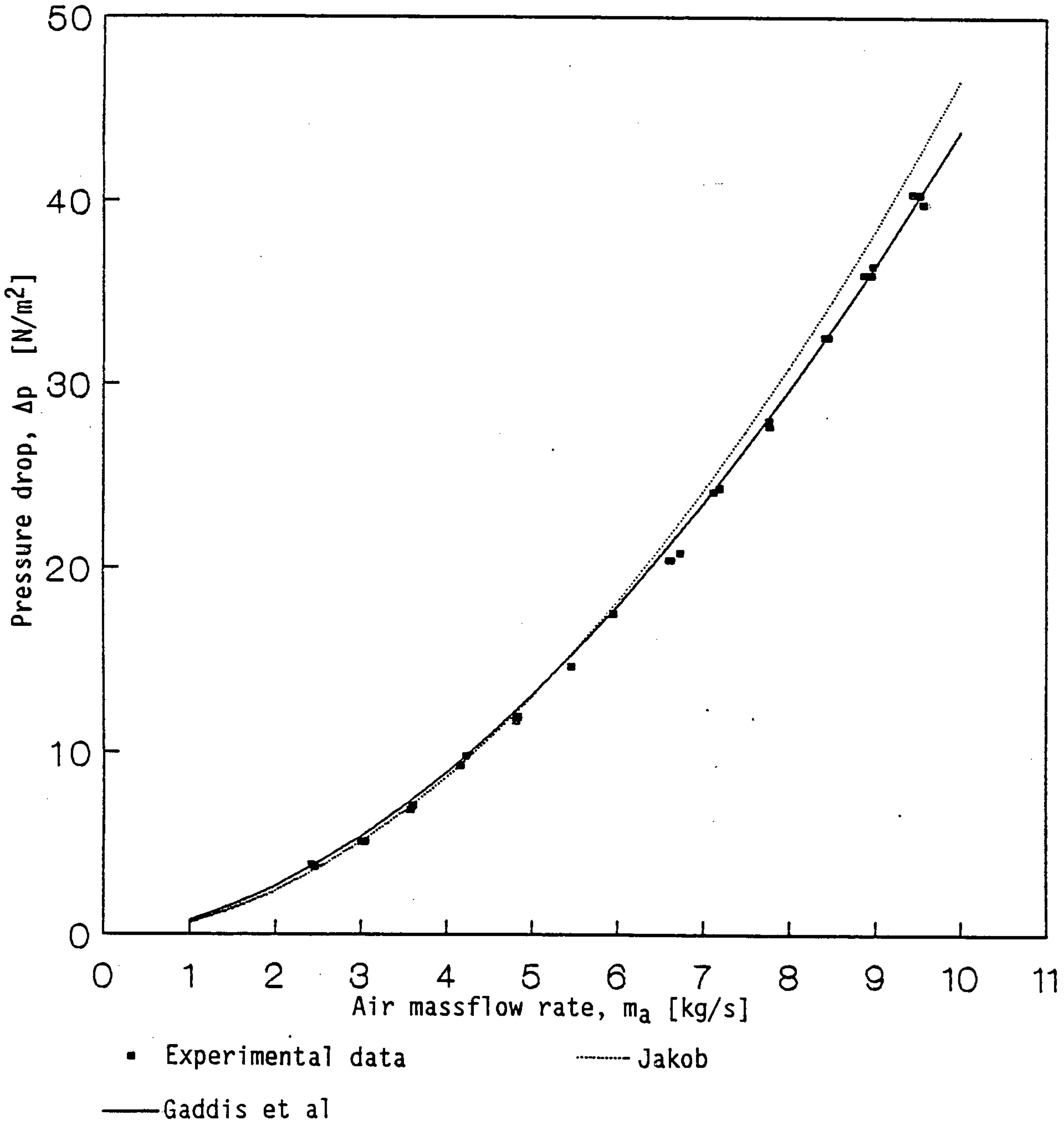

Figure 6.20 Single phase pressure drop values measured across the dry tube bundle compared to existing correlations for single phase pressure drop.

The measured two phase pressure drop was correlated by defining the following parameter

$$\eta = \frac{m_w}{\left[m_a + m_w \ (\rho_a/\rho_w) \right] Re_a^*} \tag{6.17}$$

where $Re_a^* = \left[\frac{m_a + m_w}{A_{fr,min}} \right] \frac{d_o}{\mu_a}$

By simple regression analysis, using Lotus 123, the following correlation for two phase pressure drop across a wet tube bundle could be found

$$\frac{\Delta p_{tp}}{\Delta p_a^*} = \frac{1,5482 \times 10^{-4}}{\eta + 9,25 \times 10^{-5}} - 0,32773 \tag{6.18}$$

in the ranges

$$2,15 \times 10^{-5} < \eta < 19 \times 10^{-5}$$

$$0,85 < \frac{m_a}{A_{fr}} < 2,5 \quad [kg/m^2/s]$$

$$100 < \Gamma < 630 \ [kg/m/hr]$$

The measured two phase pressure drop data and the pressure drop correlation are shown graphically in figure 6.21.

6.6 Discussion of results

The correlations for the mass transfer coefficient presented in this report has the same form as the correlations presented by Mizushina et al. [67MI1]. The exponent of the air Reynolds number in the

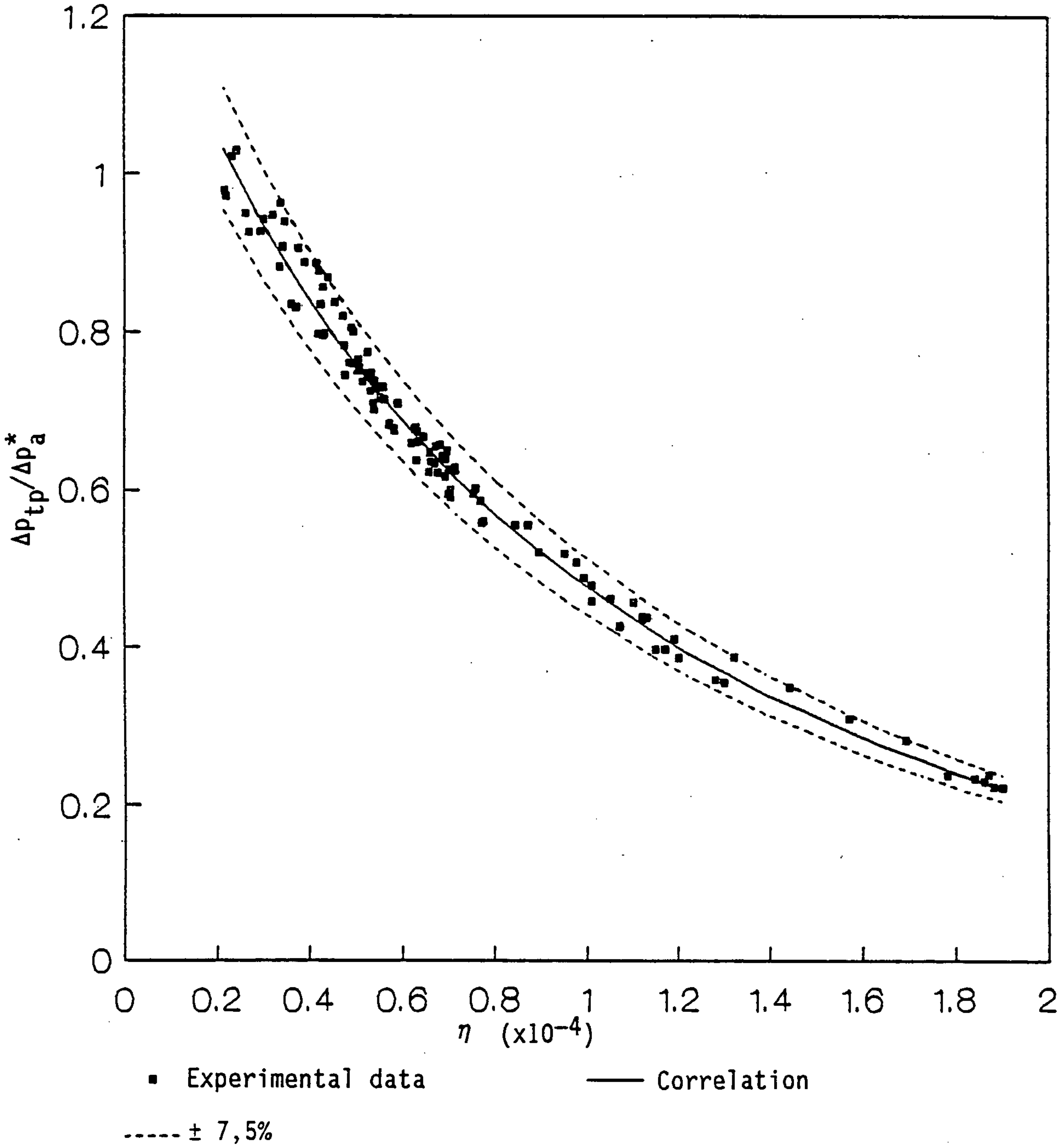

Figure 6.21 Two phase pressure drop values measured across the test section.

correlation by Mizushina et al. is 0,9. Parker and Treybal [61PA1] and Peterson [84PE3] found the exponent of the air Reynolds number to be 0,905 in their correlations for the mass transfer coefficient. In the correlations for the mass transfer coefficient presented in this report the exponent of the air Reynolds number was found to be 0,62. It should however be noted that the correlations by Parker and Treybal [61PA1], Mizushina et al. [67MI1] and Peterson [84PE3] were determined for counterflow evaporative coolers and condensers while the current study was done on a cross-flow evaporative cooler.
The heat transfer correlations for heat transfer from a dry tube bundle with staggered tubes spaced in a 2 x d_o triangular array, typically gives heat transfer coefficient as

$$h_c = C\, Re_a^{0,6} \tag{6.19}$$

If the heat/mass transfer analogy is used to determine the mass transfer coefficient governing the mass transfer from a wet tube bundle the exponent of the air Reynolds number in this correlation would be 0,6. The exponent of Re_a in the mass transfer coefficient correlation found in this study as 0,62 would then seem realistic.

Parker and Treybal [61PA1] and Peterson [84PE1] found the mass transfer coefficient to be independent of the recirculating water flow while Mizushina et al. [67MI1] found that the mass transfer coefficient was dependant on the recirculating water Reynolds number to the power of 0,15 which is in agreement with the findings of this study where the exponent of the recirculating water Reynolds number was found to be 0,2.

The mass transfer coefficient correlations for the mass transfer coefficients based on the Poppe theory and the Merkel theory, using the interface temperature rather than the bulk recirculating water temperature, shows similar dependencies on the air and water Reynolds numbers as the correlation for the mass transfer coefficient based on the Merkel model and the bulk recirculating water temperature.

The three correlations determined in this study are graphically compared to the correlation by Mizushina et al. [67MI1] and the heat- and mass transfer analogy in figure 6.22. The heat transfer correlation for forced convection from a tube bundle by Grimison [37GR1] was used to calculate the mass transfer coefficient in the analogy approach. From figure 6.22 it can be seen that the current correlations fall between the predictions by Mizushina et al. [67MI1] and the heat/mass transfer analogy. The correlation given by Mizushina was derived for a counterflow evaporative cooler where the air/water interaction is expected to be more pronounced than in the current study which was performed on a cross-flow evaporative cooler. As expected the mass transfer coefficients in a counterflow evaporative cooler are higher than for a cross-flow evaporative cooler at the same air Reynolds number. The mass transfer coefficients calculated from the analogy between heat transfer and mass transfer does not take into account the air/water interaction and this consequently results in lower mass transfer coefficients than the coefficients determined experimentally for the cross-flow evaporative cooler.

The experimentally determined film heat transfer coefficient shows a fair amount of scatter. The scatter can be attributed to the fact that the film heat transfer coefficient represents a relatively small part of the overall thermal resistance and it is consequently very sensitive to small variations in the recirculating water temperature.
A small variation of 0,2°C in the outlet recirculating water temperature resulted in a variation of the film coefficient of up to 20%.
The correlations fitted on the experimental data represents the data fairly well with only about 12,5% of the data points falling outside the 15% variation zone around the correlation as shown in figures 6.17, 6.18 and 6.19.

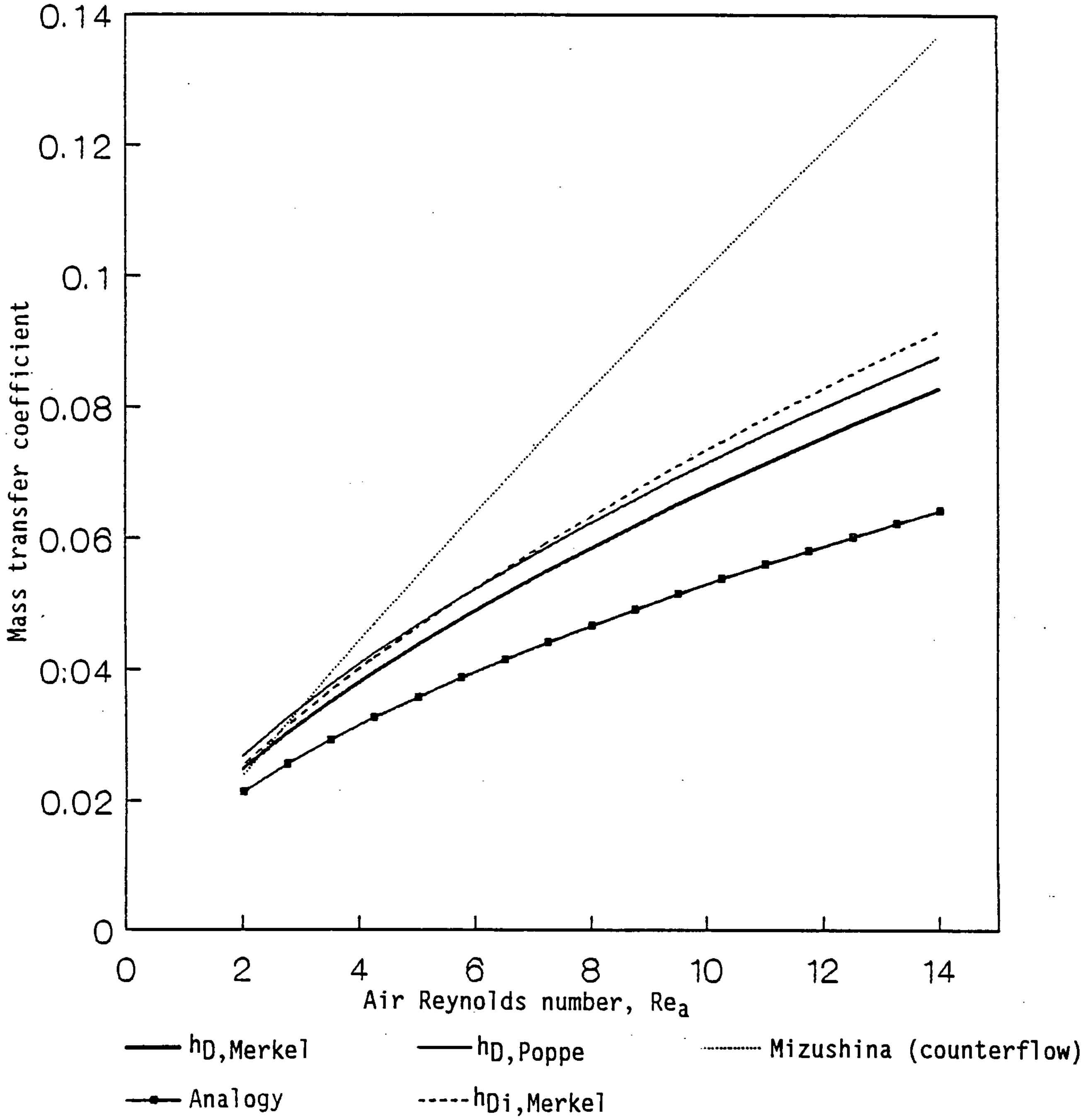

Figure 6.22 Graphical comparison of the present mass transfer coefficient correlations with existing correlations at Re_w = 400.

The correlations fitted to the data show the film heat transfer coefficient to be dependant on the recirculating water massflow rate per unit length (Γ) to the power of about 0,33, which is similar to the findings of other investigators including McAdams [54Mc1], Parker and Treybal [61PA1] and Mizushina et al. [67MI1].
The correlations for film coefficient determined in this study is graphically compared to the correlations of McAdams [54Mc1] and Mizushina et al. [67MI1] in figure 6.23. The correlation by McAdams [54MI1] was determined for a film cooler (without airflow) while the correlation given by Mizushina et al. [67MI1] was determined for a counterflow evaporative cooler. The new correlation for h_w based on the Merkel model corresponds closely to the correlation for h_w based on the Poppe model as expected since they are both based on the same driving force (T_{wall} - T_w).
The correlation for h_{wi} gives film coefficients which are lower than those based on the bulk water temperature since the driving force for the film coefficient based on the interface temperature is larger than the driving force based on the bulk recirculating water temperature at the same heat flux.

The single phase pressure drop measured across the tube bundle corresponds very well to the correlations by Jakob [37JA1] and Gaddis and Gnielinski [85GA1] as seen in figure 6.20. The two phase pressure drop across the tube bundle was correlated by a parameter η given by

$$\eta = \frac{m_w}{\left(m_a + m_w \, (\rho_a/\rho_w) \right) Re_a^*}$$

The correlation fitted through the data correlates the data very well with only 1 of 117 points differing from the correlation by more than 7,5% as seen in figure 6.21.

The correlation is compared to the other available cross-flow correlations (see Chapter 4) in figure 6.24 for a bank of tubes spaced

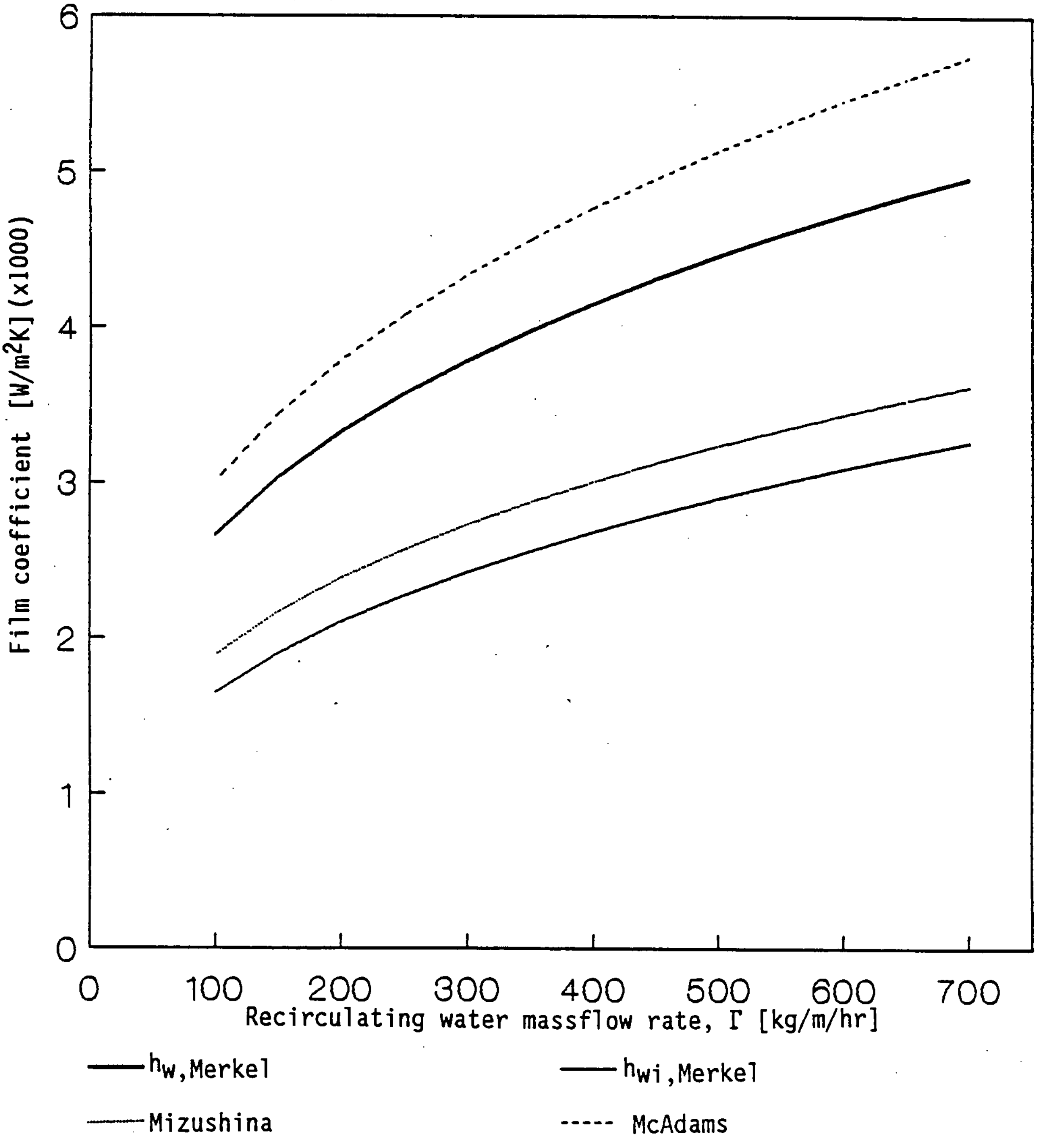

Figure 6.23 Graphical comparison of the present film coefficient correlations with existing correlations.

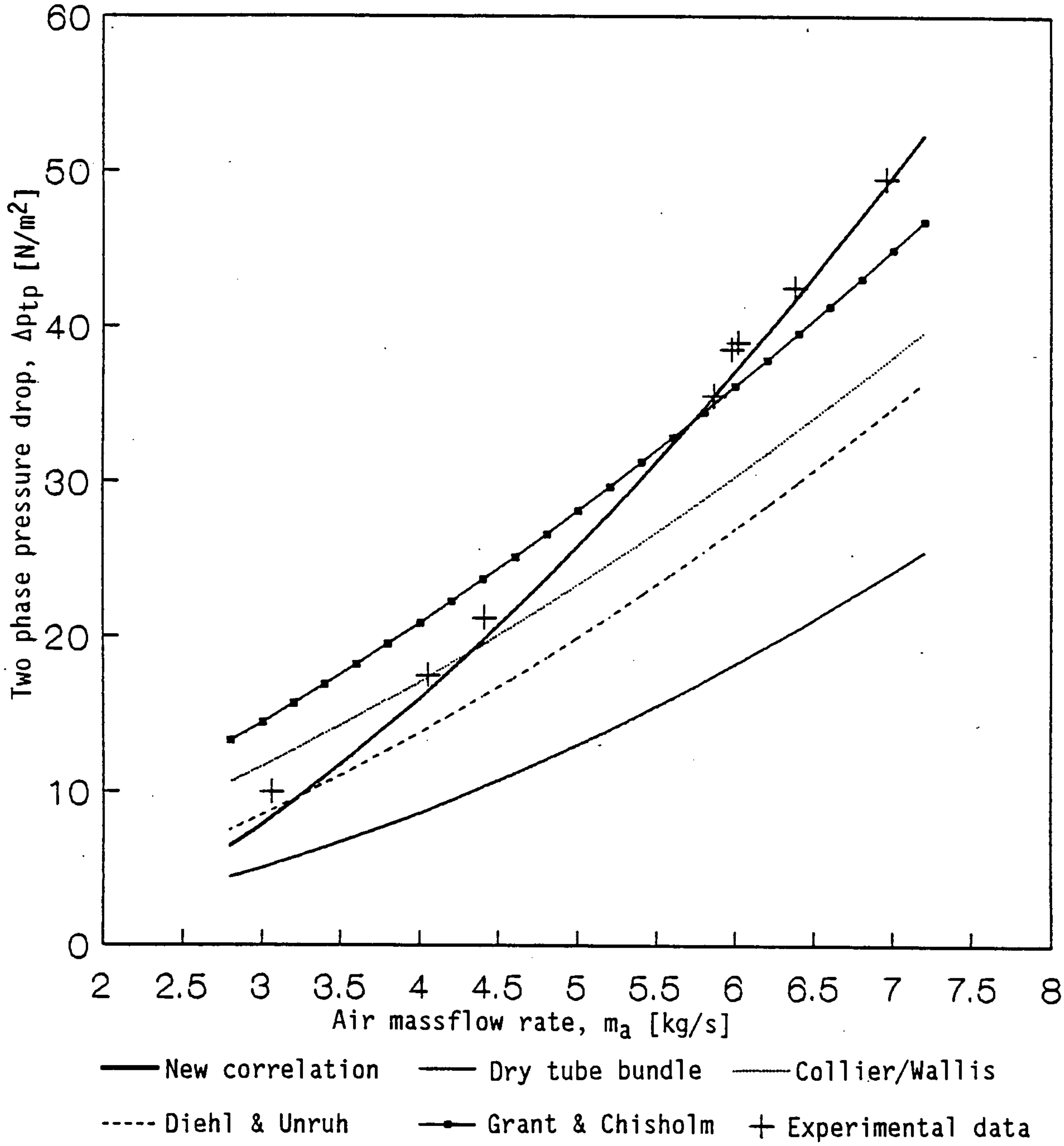

Figure 6.24 Graphical comparison of the present pressure drop correlation with existing correlations at a recirculating water massflow rate of 5 kg/s.

in a 2 x d_o triangular array with d_o = 0,0381 m and a frontal area of 3,429m^2. From figure 6.24 it can be seen that the new correlation gives pressure drops which are significantly higher than those predicted by the models of Collier [72CO1], Wallis [69WA1] and Grant and Chisholm [79GR1] at high air massflow rates while at the new correlation predicts lower values than the other correlations at low air massflow rates.

The significant difference in the slope of the new two phase correlation when compared to the existing correlations can be due to the fact that the new correlation was derived for a set-up where the water and the air does not flow in the same direction (as was the case for the other correlations) but the water flow across the airstream.

CHAPTER 7

CONCLUSION

In this report complete theoretical models for the analysis of evaporative coolers and condensers have been derived. These models range from a very simple model, which allows for fast rating and sizing calculations, to more accurate models requiring numerical integration and successive calculating procedures.
Computer programs to analyse different cross-flow and counterflow evaporative coolers and condensers have been compiled. It was found that the simplified models usually gives results which are accurate enough for simple engineering sizing and rating calculations.
The simplified model for the analysis of evaporative condensers is expected to be very accurate since the assumption of constant recirculating water temperature is a good approximation because of the constant condensing temperature.

The simplified model can also be used in the analysis of cross-flow evaporative coolers with a fair degree of accuracy. The simplified model cannot be expected to yield accurate results in analysing cross-flow evaporative coolers with relatively long tubes since the three-dimensional recirculating water temperature profile becomes to complex to describe with a single representative recirculating water temperature.

A survey of the available data for the heat- and mass transfer coefficients was conducted and all the relevant correlations were summarized and compared. The relevant correlations for two phase pressure drop across a tube bundle, which could be found in the literature are also presented.

An experimental study was conducted on a cross-flow evaporative cooler to determine the governing heat- and mass transfer coefficients and the two phase pressure drop across the tube bundle. The correlations which were fitted to the experimental data are compared to the other available correlations.

The use of Mizushina's [67MI1] correlations for the heat- and mass transfer coefficients, are recommended for the analysis of counterflow evaporative coolers and condensers. The correlations presented here should be used in the analysis of cross-flow evaporative coolers and condensers.

The effect of tube diameter and tube spacing on the heat and mass transfer coefficient could be the subject of further investigations.

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APPENDICES

APPENDIX A

PROPERTIES OF FLUIDS

A.1 The thermophysical properties of dry air from 220 K to 380 K

Density

$$\rho_a = p_a/RT, \ \text{kg/m}^3 \qquad \text{(A.1.1)}$$

where R = 287.08 J/kgK

Specific heat [82AN1]

$$c_{pa} = a + bT + cT^2 + dT^3, \quad \text{J/kgK} \qquad \text{(A.1.2)}$$

$a = 1.045356\text{x}10^{3}$
$b = -3.161783\text{x}10^{-1}$
$c = 7.083814\text{x}10^{-4}$
$d = -2.705209\text{x}10^{-7}$

Dynamic viscosity [82AN1]

$$\mu_a = a + bT + cT^2 + dT^3, \quad \text{kg/sm} \qquad \text{(A.1.3)}$$

$a = 2.287973\text{x}10^{-6}$
$b = 6.259793\text{x}10^{-8}$
$c = -3.131956\text{x}10^{-11}$
$d = 8.150380\text{x}10^{-15}$

Thermal conductivity

$$k_a = a + bT + cT^2 + dT^3, \quad \text{W/mK} \qquad \text{(A.1.4)}$$

$a = -4.937787\text{x}10^{-4}$
$b = 1.018087\text{x}10^{-4}$
$c = -4.627937\text{x}10^{-8}$
$d = 1.250603\text{x}10^{-11}$

Table A.1: The thermophysical properties of dry air at standard atmospheric pressure (101325 N/m^2)

T K	ρ_a kg/m^3	c_{pa} J/kgK	μ_a kg/ms $\times 10^5$	k_a W/mK	α_a $\times 10^5$	Pr_a
220	1.60432	1007.20	1.46304	0.0197973	1.22518	0.744330
225	1.56866	1006.99	1.48797	0.0202127	1.27957	0.741309
230	1.53456	1006.81	1.51278	0.0206262	1.33500	0.738428
235	1.50191	1006.66	1.53746	0.0210378	1.39145	0.735680
240	1.47062	1006.53	1.56201	0.0214475	1.44892	0.733056
245	1.44061	1006.43	1.58643	0.0218553	1.50738	0.730550
250	1.41180	1006.35	1.61073	0.0222613	1.56684	0.728156
255	1.38411	1006.30	1.63490	0.0226655	1.62727	0.725867
260	1.35750	1006.28	1.65894	0.0230678	1.68867	0.723678
265	1.33188	1006.28	1.68286	0.0234683	1.75103	0.721585
270	1.30722	1006.30	1.70666	0.0238669	1.81433	0.719581
275	1.28345	1006.35	1.73033	0.0242638	1.87857	0.717663
280	1.26053	1006.42	1.75388	0.0246589	1.94373	0.715828
285	1.23842	1006.52	1.77731	0.0250521	2.00980	0.714070
290	1.21707	1006.64	1.80061	0.0254436	2.07677	0.712387
295	1.19644	1006.78	1.82380	0.0258334	2.14463	0.710776
300	1.17650	1006.95	1.84686	0.0262213	2.21336	0.709233
305	1.15721	1007.14	1.86980	0.0266075	2.28297	0.707755
310	1.13854	1007.35	1.89263	0.0269920	2.35342	0.706340
315	1.12047	1007.59	1.91533	0.0273747	2.42472	0.704985
320	1.10297	1007.85	1.93792	0.0277558	2.49685	0.703688
325	1.08600	1008.13	1.96039	0.0281351	2.56980	0.702446
330	1.06954	1008.43	1.98274	0.0285127	2.64356	0.701258
335	1.05358	1008.76	2.00498	0.0288886	2.71811	0.700122
340	1.03808	1009.11	2.02710	0.0292628	2.79345	0.699035
345	1.02304	1009.48	2.04911	0.0296353	2.86957	0.697997
350	1.00842	1009.87	2.07100	0.0300062	2.94645	0.697004
355	0.99422	1010.28	2.09278	0.0303754	3.02408	0.696056
360	0.98041	1010.71	2.11444	0.0307430	3.10246	0.695151
365	0.96698	1011.17	2.13599	0.0311089	3.18156	0.694288
370	0.95392	1011.64	2.15743	0.0314732	3.26138	0.693465
375	0.94120	1012.13	2.17876	0.0318359	3.34191	0.692681
380	0.92881	1012.65	2.19998	0.0321970	3.42313	0.691935

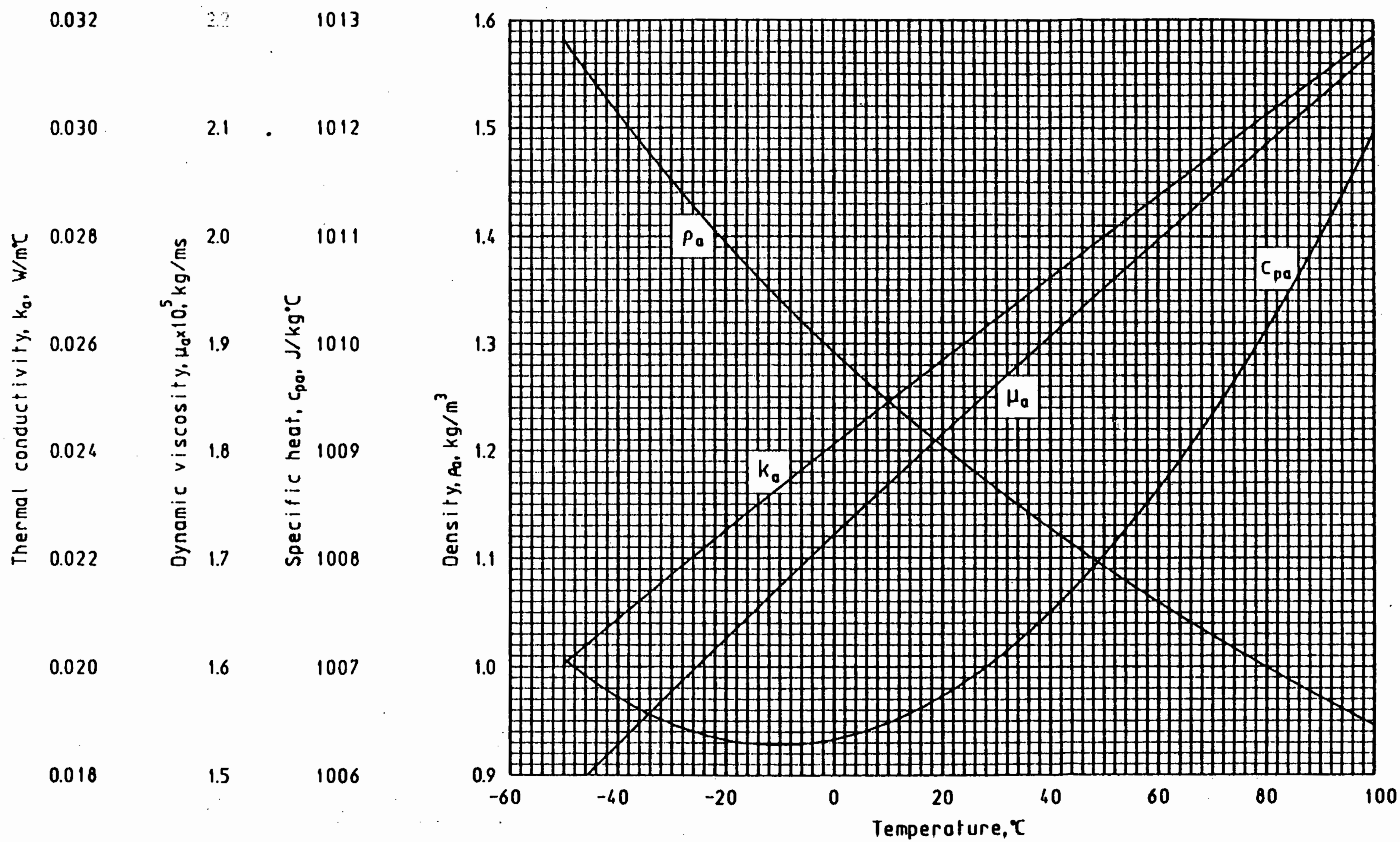

Figure A.1: The thermophysical properties of dry air at standard atmospheric pressure (101325 N/m^2)

A.2 The thermophysical properties of saturated water vapor from 273.15 K to 380 K

Vapor pressure [46G01]

$$p_v = 10^z, \quad N/m^2 \tag{A.2.1}$$

$$z = a(1-x) + b\log_{10}(x) + c\left[1-10^{d\{(1/x)-1\}}\right] + e(10^{f(1-x)}-1) + g$$

$$x = 273.16/T$$

$a = 1.079586x10$
$b = 5.028080$
$c = 1.504740x10^{-4}$
$d = -8.296920$
$e = 4.287300x10^{-4}$
$f = 4.769550$
$g = 2.786118312$

Specific heat

$$c_{pv} = a + bT + cT^5 + dT^6, \quad J/kgK \tag{A.2.2}$$

$a = 1.3605x10^3$
$b = 2.31334$
$c = -2.46784x10^{-10}$
$d = 5.91332x10^{-13}$

Dynamic viscosity

$$\mu_v = a + bT + cT^2 + dT^3, \quad kg/ms \tag{A.2.3}$$

$a = 2.562435x10^{-6}$
$b = 1.816683x10^{-8}$
$c = 2.579066x10^{-11}$
$d = -1.067299x10^{-14}$

Thermal conductivity [82AN1]

$$k_v = a + bT + cT^2 + dT^3, \quad W/mK \tag{A.2.4}$$

a = 1.3046×10^{-2}
b = -3.756191×10^{-5}
c = 2.217964×10^{-7}
d = $-1.111562 \times 10^{-10}$

Vapor density [70UK1]

$$\rho_v = a + bT + cT^2 + dT^3 + eT^4 + fT^5, \quad kg/m^3 \tag{A.2.5}$$

a = -4.062329056
b = 0.10277044
c = $-9.76300388 \times 10^{-4}$
d = $4.475240795 \times 10^{-6}$
e = $-1.004596894 \times 10^{-8}$
f = $8.9154895 \times 10^{-12}$

Table A.2: The thermophysical properties of saturated water vapor

T K	p_v N/m^2	ρ_v kg/m^3	c_{pv} J/kgK	μ_v kg/ms x10^6	k_v W/mK	α_v x10^5	Pr_v
275	697.820	0.00550	1864.29	9.28676	0.0171781	167.602	1.00786
280	990.897	0.00767	1868.46	9.4368	0.0174774	121.992	1.00887
285	1387.70	0.01056	1872.66	9.58775	0.0177831	90.0091	1.00964
290	1918.11	0.01436	1876.92	9.73950	0.0180951	67.2667	1.01023
295	2618.61	0.01928	1881.31	9.89208	0.0184134	50.8805	1.01068
300	3533.19	0.02557	1885.89	10.0454	0.0187378	38.9260	1.01103
305	4714.45	0.03355	1890.74	10.1996	0.0190684	30.1011	1.01135
310	6224.58	0.04360	1895.92	10.3546	0.0194049	23.5132	1.01168
315	8136.44	0.05611	1901.52	10.5104	0.0197474	18.5427	1.01207
320	10534.7	0.07155	1907.63	10.6670	0.0200957	14.7547	1.01259
325	13516.9	0.09045	1914.35	10.8244	0.0204498	11.8400	1.01329
330	17194.7	0.11341	1921.79	10.9825	0.0208095	9.57698	1.01425
335	21694.5	0.14108	1930.04	11.1414	0.0211749	7.80452	1.01551
340	27158.9	0.17418	1939.25	11.3010	0.0215457	6.40488	1.01716
345	33747.7	0.21352	1949.63	11.4614	0.0219219	5.29095	1.01927
350	41638.4	0.26000	1961.03	11.6225	0.0223035	4.39779	1.02191
355	51027.6	0.31455	1973.90	11.7844	0.0226904	3.67653	1.02516
360	62131.3	0.37821	1988.29	11.9470	0.0230824	3.09016	1.02910
365	75186.3	0.45213	2004.37	12.1102	0.0234795	2.61037	1.03382
370	90450.0	0.53750	2022.33	12.2742	0.0238816	2.21538	1.03940
375	108201	0.63568	2042.35	12.4389	0.0242886	1.888304	1.04595
380	128743	0.74799	2064.63	12.6043	0.0247005	1.615964	1.05355

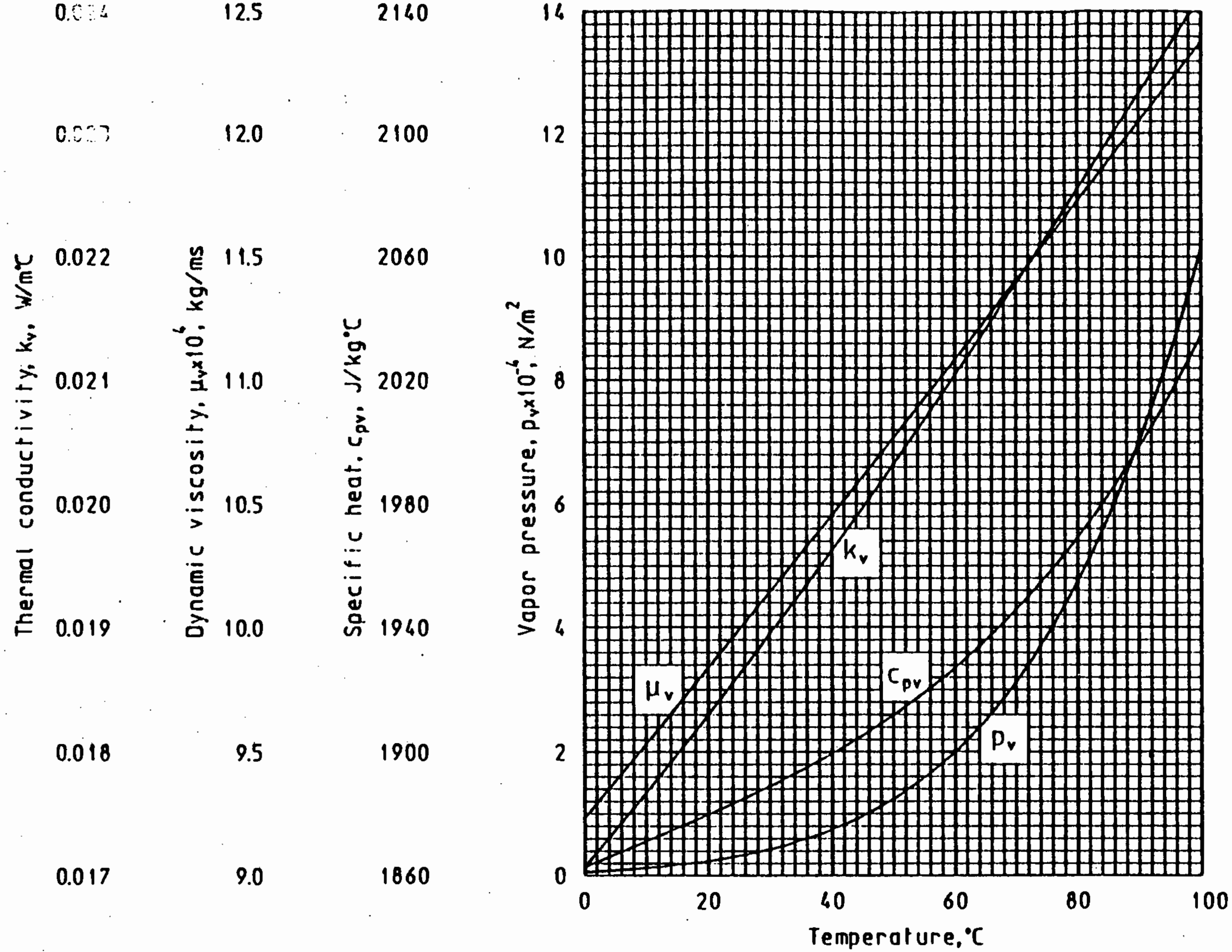

Figure A.2: The thermophysical properties of saturated water vapor

A.3 The thermophysical properties of mixtures of dry air and water vapor

Density [72AS1]

$$\rho_{av} = (1+w)\left[1-w/(w+0.62198)\right](p_{abs}/RT),\ \text{kg/m}^3 \qquad \text{(A.3.1)}$$

where R = 287.08 J/kgK

Specific heat [78FA1]

$$c_{pav} = (c_{pa} + wc_{pv})/(1+w),\quad \text{J/kgK} \qquad \text{(A.3.2)}$$

Dynamic viscosity [54GO1]

$$\mu_{av} = (X_a\, \mu_a M_a^{0.5} + X_v \mu_v M_v^{0.5})/(X_a M_a^{0.5} + X_v M_v^{0.5}),\ \text{kg/ms} \qquad \text{(A.3.3)}$$

Thermal conductivity [57LE1]

$$k_{av} = (X_a k_a M_a^{0.33} + X_v k_v M_v^{0.33})/(X_a M_a^{0.33} + X_v M_v^{0.33}),\ \text{W/mK} \qquad \text{(A.3.4)}$$

where

M_a = 28.97 kg/mole
M_v = 18.016 kg/mole
X_a = 1/(1+1.608w)
X_v = w/(w+0.622)

Humidity ratio

$$w = \left(\frac{2501.6 - 2.3263(T_{wb} - 273.15)}{2501.6 + 1.8577(T_{db} - 273.15) - 4.184(T_{wb} - 273.15)}\right) \times \left(\frac{0.62509 p_{vwb}}{p_{abs} - 1.005 p_{vwb}}\right) - \left(\frac{1.00416(T_{db} - T_{wb})}{2501.6 + 1.8577(T_{db} - 273.15) - 4.184(T_{wb} - 273.15)}\right),\ \text{kg/kg} \qquad \text{(A.3.5)}$$

A.4 The thermophysical properties of saturated water liquid from 273.15K to 380K

Density

$$\rho_w = (a + bT + cT^2 + dT^6)^{-1}, \quad kg/m^3 \qquad (A.4.1)$$

$a = 1.49343 \times 10^{-3}$
$b = -3.7164 \times 10^{-6}$
$c = 7.09782 \times 10^{-9}$
$d = -1.90321 \times 10^{-20}$

Specific heat

$$c_{pw} = a + bT + cT^2 + dT^6, \quad J/kgK \qquad (A.4.2)$$

$a = 8.15599 \times 10^{3}$
$b = -2.80627 \times 10$
$c = 5.11283 \times 10^{-2}$
$d = -2.17582 \times 10^{-13}$

Dynamic viscosity [82AN1]

$$\mu_w = a10^{b/(T-c)}, \quad kg/ms \qquad (A.4.3)$$

$a = 2.414 \times 10^{-5}$
$b = 247.8$
$c = 140$

Thermal conductivity

$$k_w = a + bT + cT^2 + dT^4, \quad W/mK \tag{A.4.4}$$

$a = -6.14255 \times 10^{-1}$
$b = 6.9962 \times 10^{-3}$
$c = -1.01075 \times 10^{-5}$
$d = 4.74737 \times 10^{-12}$

Latent heat of vaporization

$$i_{fg} = a + bT + cT^2 + dT^3, \quad J/kg \tag{A.4.5}$$

$a = 3.4831814 \times 10^6$
$b = -5.8627703 \times 10^3$
$c = 1.2139568 \times 10$
$d = -1.40290431 \times 10^{-2}$

Critical pressure

$$p_{wc} = 22.09 \times 10^6, \quad N/m^2 \tag{A.4.6}$$

Surface tension [70UK1]

$$\sigma = a + bt + cT^2 + dT^3, \quad N/m \tag{A.4.7}$$

$a = 5.148103 \times 10^{-2}$
$b = 3.998714 \times 10^{-4}$
$c = -1.4721869 \times 10^{-6}$
$d = 1.21405335 \times 10^{-9}$

Table A.3: The thermophysical properties of saturated water liquid

T K	ρ_w kg/m^3	c_{pw} J/kgK	μ_w kg/ms x10^4	k_w W/mK	β_w 1/K x10^5	Pr_w
275	1000.03	4211.21	16.5307	0.572471	0.780333	12.1603
280	999.864	4202.04	14.2146	0.581432	6.184114	10.2730
285	999.422	4194.41	12.3510	0.590001	11.45765	8.78055
290	998.721	4188.27	10.8327	0.598179	16.59011	7.58474
295	997.768	4183.53	9.58179	0.605972	21.57093	6.61511
300	996.572	4180.10	8.54057	0.613383	26.38963	5.82026
305	995.141	4177.92	7.66576	0.620417	31.03593	5.16215
310	993.487	4176.87	6.92443	0.627079	35.49975	4.61225
315	991.618	4176.88	6.29125	0.633372	39.77122	4.14887
320	989.547	4177.83	5.74650	0.639300	43.84070	3.75534
325	987.284	4179.63	5.27468	0.644870	47.69877	3.41871
330	984.842	4182.17	4.86348	0.650084	51.33626	3.12881
335	982.232	4185.32	4.50304	0.654948	54.74422	2.87758
340	979.469	4188.98	4.18540	0.659466	57.91392	2.65859
345	976.564	4193.01	3.90407	0.663644	60.83688	2.46665
350	973.532	4197.28	3.65373	0.667486	63.50480	2.29754
355	970.386	4201.67	3.43001	0.670997	65.90961	2.14781
360	967.141	4206.01	3.22924	0.674182	68.04338	2.01462
365	963.811	4210.17	3.04839	0.677046	69.89838	1.89562
370	960.409	4213.99	2.88488	0.679595	71.46697	1.78884
375	956.952	4217.31	2.73656	0.681833	72.74164	1.69263
380	953.453	4219.96	2.60158	0.683767	73.71494	1.60560

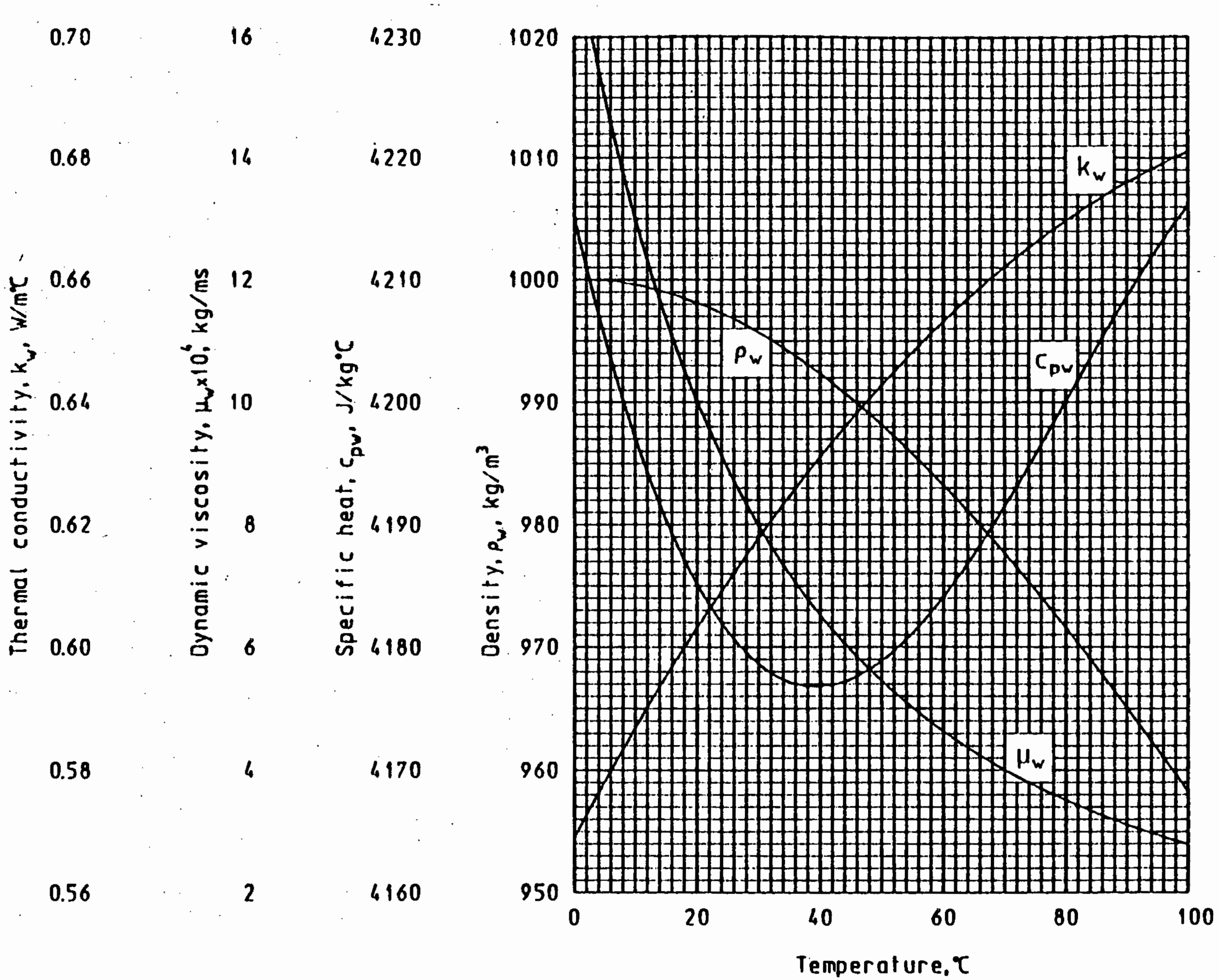

Figure A.3: The thermophysical properties of saturated water liquid

A.5 The thermophysical properties of saturated ammonia vapor.

Vapor pressure [76RA1], (230 K to 395 K)

$$p_{ammv} = a + bT + cT^2 + dT^3 + eT^4, \quad N/m^2 \tag{A.5.1}$$

$a = 1.992448 \times 10^6$
$b = -57.568140 \times 10^3$
$c = 0.5640265 \times 10^3$
$d = -2.337352$
$e = 3.541430 \times 10^{-3}$

Density [76RA1], (260 K to 390 K)

$$\rho_{ammv} = a + bT + cT^2 + dT^3 + eT^4, \quad kg/m^3 \tag{A.5.2}$$

$a = -6.018936 \times 10^2$
$b = 5.361048$
$c = -1.187296 \times 10^{-2}$
$d = -1.161479 \times 10^{-5}$
$e = 4.739058 \times 10^{-8}$

Specific heat [72AS1], (230 K to 325 K)

$$c_{pammv} = a + bT + cT^2 + dT^3, \quad J/kg\ K \tag{A.5.3}$$

$a = -2.7761190256 \times 10^4$
$b = 3.39116449 \times 10^2$
$c = -1.3055687$
$d = 1.728649 \times 10^{-3}$

Dynamic viscosity [72AS1], (240 K to 370 K)

$$\mu_{ammv} = a + bT + cT^2 + dT^3 + eT^4, \quad kg/sm \tag{A.5.4}$$

$a = -2.748011 \times 10^{-5}$
$b = 2.82526 \times 10^{-7}$
$c = -5.201831 \times 10^{-10}$
$d = -6.061761 \times 10^{-13}$
$e = 2.126070 \times 10^{-15}$

Thermal conductivity [72AS1], (245 K to 395 K)

$$k_{ammv} = a + bT + cT^2 + dT^3 + eT^4, \quad W/mK \tag{A.5.6}$$

$a = -0.1390216$
$b = 1.35238 \times 10^{-3}$
$c = -2.532035 \times 10^{-6}$
$d = -4.884341 \times 10^{-9}$
$e = 1.418657 \times 10^{-11}$

Table A.4: The thermophysical properties of saturated ammonia vapor

T K	p_{ammv} N/m^2 x 10^{-3}	ρ_{ammv} kg/m^3	c_{pammv} J/kgK	μ_{ammv} kg/sm x 10^6	k_{ammv} W/mK	Pr_{ammv}
230	60.58		2203.48			
235	79.09		2265.33			
240	102.08		2322.84	9.0376		
245	130.32		2377.30	9.2605	0.019611	1.12261
250	164.65		2430.01	9.4734	0.019920	1.15567
255	205.93		2482.27	9.6774	0.020185	1.19009
260	255.10	1.78881	2535.37	9.8737	0.020414	1.22632
265	313.14	2.56766	2590.61	10.0635	0.020613	1.26477
270	381.09	3.28963	2649.29	10.2480	0.020790	1.30588
275	460.03	3.98405	2712.69	10.4284	0.020954	1.35007
280	551.10	4.68094	2782.13	10.6060	0.021111	1.39774
285	655.51	5.41106	2858.89	10.7822	0.021270	1.44924
290	774.50	6.20583	2944.27	10.9583	0.021439	1.50491
295	909.36	7.09745	3039.57	11.1356	0.021627	1.56503
300	1061.47	8.11876	3146.08	11.3156	0.021843	1.62979
305	1232.21	9.30337	3265.10	11.4997	0.022096	1.69932
310	1423.06	10.68557	3397.93	11.6894	0.022394	1.77368
315	1635.53	12.30036	3545.87	11.8862	0.022748	1.85280
320	1871.19	14.18346	3710.20	12.0917	0.023166	1.93654
325	2131.65	16.37130	3892.24	12.3074	0.023660	2.02466
330	2418.59	18.90102		12.5349	0.024238	
335	2733.73	21.81047		12.7758	0.024911	
340	3078.86	25.13821		13.0318	0.025690	
345	3455.81	28.92352		13.3047	0.026586	
350	3866.46	33.20637		13.5961	0.027608	
355	4312.75	38.02745		13.9078	0.028769	
360	4796.68	43.42818		14.24159	0.030080	
365	5320.28	49.45067		14.5993	0.031551	
370	5885.67	56.13774		14.9827	0.033196	
375	6494.98	63.53294			0.035026	
380	7150.43	71.68050			0.037053	
385	7854.27	80.62539			0.039289	
390	8608.81	90.41328			0.041748	
395	9416.42				0.044441	

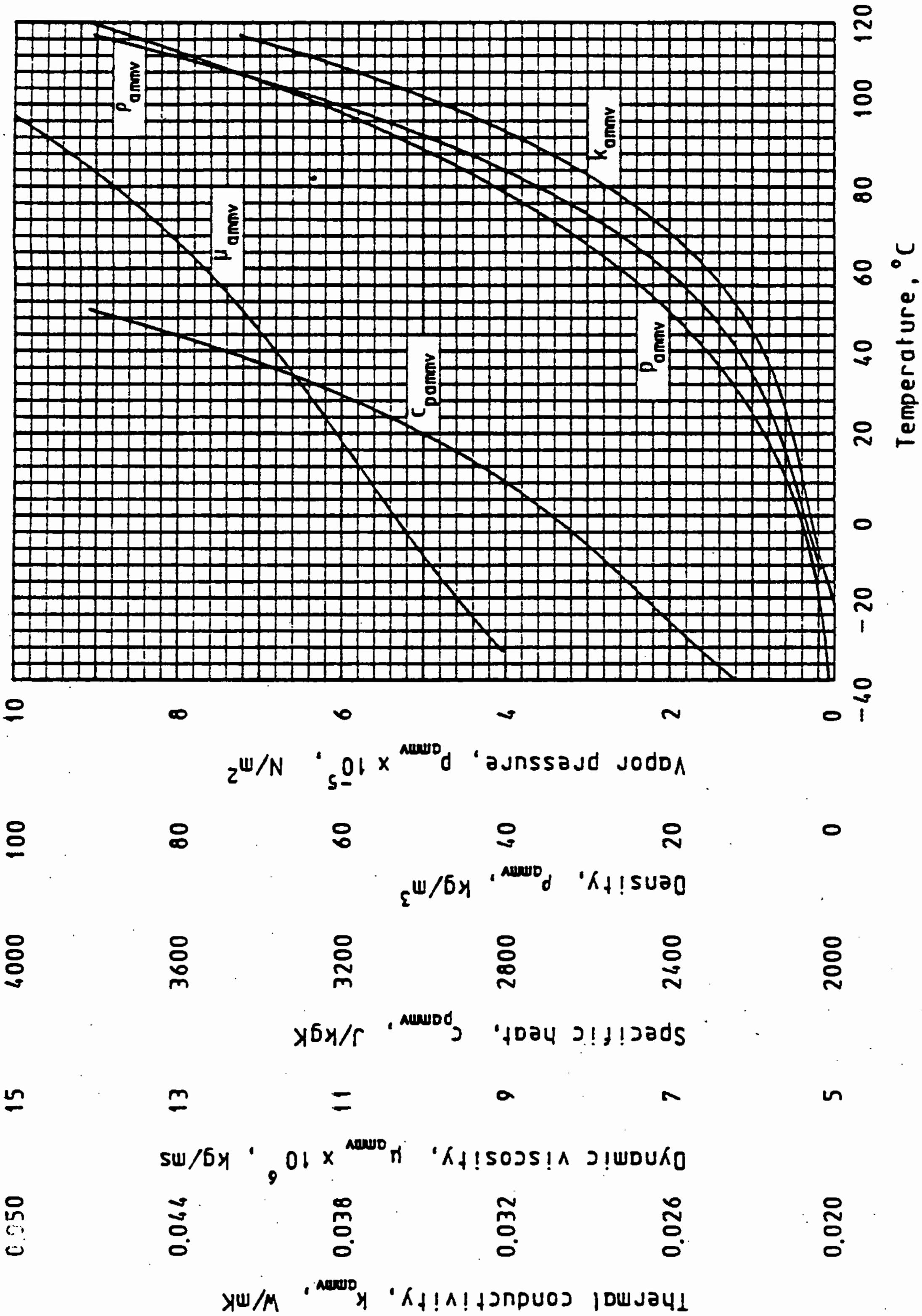

Figure A.4: Thermophysical properties of saturated ammonia vapor

A.6 The thermophysical properties of saturated ammonia liquid from 200 K to 405 K.

Density [77YA1]

$$\rho_{amm} = ab^{\left[-(1-T/T_c)^{0.285714}\right]}, \ \text{kg/m}^3 \qquad \text{(A.6.1)}$$

$a = 2.312 \times 10^2$

$b = 0.2471$

$Tc = 405.5$ K

Specific heat [77YA1], (200 K to 375 K)

$$c_{pamm} = a + bT + cT^2 + dT^3, \quad \text{J/kgK} \qquad \text{(A.6.2)}$$

$a = -2.497276939 \times 10^3$

$b = 7.7813907 \times 10$

$c = -3.006252 \times 10^{-1}$

$d = 4.06714 \times 10^{-4}$

Dynamic viscosity [77YA1]

$$\mu_{amm} = 0.001 \times 10^{(a + b/T + cT + dT^2)}, \quad \text{kg/sm} \qquad \text{(A.6.3)}$$

$a = -8.591$

$b = 876.4$

$c = 0.02681$

$d = -3.612 \times 10^{-5}$

Thermal conductivity [77YA1], (200 K to 375 K)

$$k_{amm} = a + bT + cT^2, \quad W/mK \tag{A.6.4}$$

$a = 1.068229$

$b = -1.576908 \times 10^{-3}$

$c = -1.228884 \times 10^{-6}$

Latent heat of vaporization, [77YA1]

$$i_{fgamm} = a \left[(b-T)/(b-c)\right]^{d}, \quad J/kg \tag{A.6.5}$$

$a = 1.370758 \times 10^{6}$

$b = 405.55$

$c = 239.72$

$d = 0.38$

Table A.5: The thermophysical properties of saturated ammonia liquid

T K	ρ_{amm} kg/m^3	c_{pamm} J/kgK	μ_{amm} kg/sm x10^5	k_{amm} W/mk	Pr_{amm}	i_{fgamm} J/kg x10^{-3}
200	731.094	4294.20	51.0740	0.703692	3.11673	1487.29
205	725.217	4324.69	45.9440	0.693319	2.86583	1473.44
210	719.282	4352.65	41.6429	0.682885	2.65428	1459.37
215	713.288	4378.38	37.9998	0.672389	2.47443	1445.08
220	707.232	4402.21	34.8841	0.661832	2.32034	1430.55
225	701.111	4424.42	32.1948	0.651213	2.18736	1415.78
230	694.923	4445.33	29.8529	0.640532	2.07181	1400.75
235	688.663	4465.24	27.7961	0.629791	1.97076	1385.45
240	682.329	4484.46	25.9749	0.618988	1.88184	1369.87
245	675.918	4503.28	24.3494	0.608123	1.80313	1354.00
250	669.424	4522.03	22.8875	0.597197	1.73306	1337.82
255	662.844	4540.99	21.5630	0.586210	1.67035	1321.31
260	656.173	4560.48	20.3543	0.575161	1.61390	1304.46
265	649.406	4580.79	19.2438	0.564050	1.56284	1287.25
270	642.537	4602.25	18.2170	0.552878	1.51641	1269.65
275	635.560	4625.14	17.2617	0.541645	1.47399	1251.65
280	628.469	4649.78	16.3678	0.530351	1.43503	1233.21
285	621.255	4676.47	15.5271	0.518994	1.39909	1214.31
290	613.912	4705.52	14.7325	0.507577	1.36578	1194.92
295	606.428	4737.23	13.9783	0.496098	1.33478	1175.009
300	598.794	4771.90	13.2596	0.484557	1.30581	1154.52
305	590.999	4809.85	12.5727	0.472955	1.27861	1133.42
310	583.029	4851.37	11.9141	0.461292	1.25300	1111.67
315	574.868	4896.77	11.2814	0.449567	1.22879	1089.19
320	566.500	4946.35	10.6723	0.437781	1.20584	1065.93
325	557.905	5000.43	10.0853	0.425933	1.18401	1041.82
330	549.060	5059.31	9.5189	0.414024	1.16320	1016.75
335	539.936	5123.28	8.9722	0.402054	1.14331	990.644
340	530.501	5192.66	8.4445	0.390022	1.12428	963.355
345	520.717	5267.75	7.9351	0.377928	1.10604	934.743
350	510.534	5348.86	7.4438	0.365773	1.08855	904.625
355	499.893	5436.29	6.9704	0.353557	1.07177	872.776
360	488.718	5530.35	6.5146	0.341279	1.05568	838.908
365	476.910	5631.34	6.0765	0.328940	1.04028	802.648
370	464.337	5739.56	5.6561	0.316539	1.02558	763.498
375	450.814	5855.32	5.2534	0.304077	1.01160	720.764
380	436.074		4.8684			673.437
385	419.697		4.5011			619.951
390	400.967		4.1517			557.630
395	378.461		3.8199			481.199
400	348.445		3.5059			376.981
405	285.772		3.2093			156.612

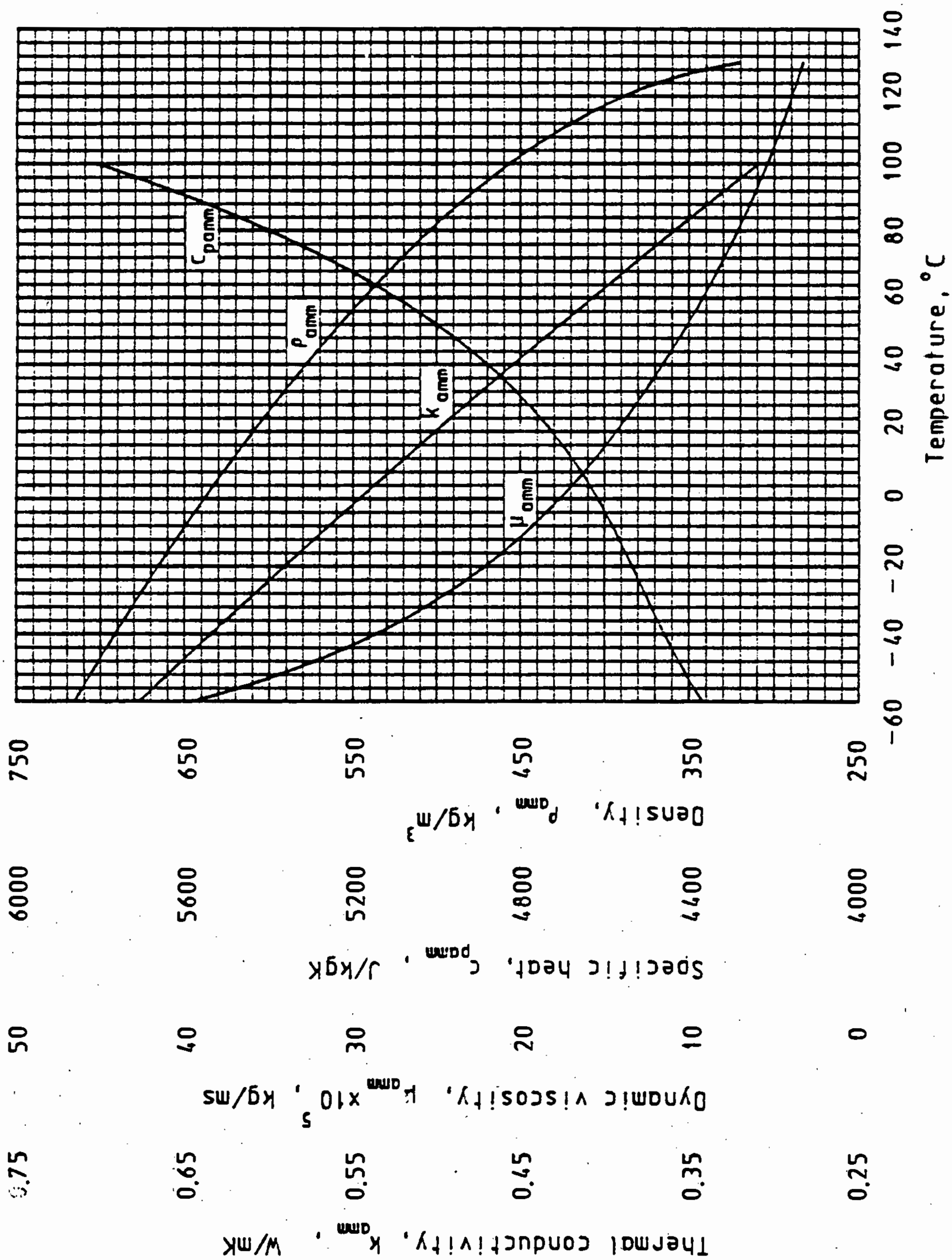

Figure A.5: Thermophysical properties of saturated ammonia liquid

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APPENDIX B

DEFINITION OF LEWIS NUMBER AND THE LEWIS FACTOR

In simultaneous heat and mass transfer the factor $(h_c/h_D c_{pm})$ and the Lewis number are often used as dimensionless parameters. In some of the literature encountered there seems to be some confusion about the definitions of these dimensionless numbers and the factor $(h_c/h_D c_{pm})$ is often incorrectly referred to as the Lewis number. The correct definitions of both these parameters will now be presented to clarify any misconceptions.

Definition of the Lewis number

The rate equations for the transfer of momentum, energy and mass are given by

i) Newton's equation of viscosity,

$$\left[\frac{F}{A}\right] = -\mu\left[\frac{\partial v_x}{\partial y}\right]$$

or

$$\left[\frac{F}{A}\right] = -\nu\left[\frac{\partial\left(\rho v_x\right)}{\partial y}\right] \qquad \text{(B.1)}$$

ii) Fourier's equation of energy conduction,

$$\left[\frac{q}{A}\right] = -k\left[\frac{\partial T}{\partial y}\right]$$

or

$$\left[\frac{q}{A}\right] = -\alpha\left[\frac{\partial\left(\rho c_p T\right)}{\partial y}\right] \qquad \text{(B.2)}$$

iii) and Fick's equation of diffusion,

$$\left[\frac{m}{A} \right] = -D \left[\frac{\partial c}{\partial y} \right] \tag{B.3}$$

The three coefficients ν, α and D in these rate equations all have the dimensions [L^2/T]. Any ratio of these coefficients would result in a dimensionless number.

In the study of systems undergoing simultaneous energy and momentum transfer the ratio of ν to α would be of importance. By definition the Prandtl number is defined as

$$Pr = \frac{\nu}{\alpha} = \frac{c_p \mu}{k} \tag{B.4}$$

In processes where simultaneous momentum and mass transfer occur the Schmidt number is defined as the ratio of ν to D, or

$$Sc = \frac{\nu}{D} \tag{B.5}$$

The ratio of α to D would be important for simultaneous energy and mass transfer processes. This ratio is called the Lewis number and it is expressed as

$$Le = \frac{\alpha}{D} \tag{B.6}$$

These three dimensionless numbers can be seen as a measure of the relative boundary layer thicknesses involved, e.g. the Lewis number can be seen as the relative thickness of the thermal and concentration boundary layers

$$Le = \frac{\delta_t}{\delta_c} \tag{B.7}$$

Similarly we have

$$Pr = \frac{\delta_m}{\delta_t} \tag{B.8}$$

$$Sc = \frac{\delta_m}{\delta_c} \tag{B.9}$$

From the definitions above the Lewis number can be expressed in various forms e.g.

$$Le = \frac{\alpha}{D} = \frac{k}{\rho c_p} = \frac{\delta_t}{\delta_c} = \frac{Sc}{Pr} \tag{B.10}$$

Definition of the Lewis factor

W.K. Lewis [22LE1] tried to prove analytically that

$$\frac{h_c}{h_D\, c_{pm}} = 1 \tag{B.11}$$

for gas/liquid systems.

In a later article Lewis [33LE1] showed that the previous relation does not hold for all mixtures of liquid and gas, but that the relation does, in fact, hold approximately for air/water mixtures.

Peterson [84PE1] concluded that the analytical proof of the Lewis relation given by Lewis [22LE1] was mathematically incorrect.

Although the proof given by Lewis was incorrect the factor $(h_c/h_D c_{pm})$ is today known as the Lewis factor and the relation $h_c/h_D c_{pm} \approx 1$ is known as the Lewis relation.

Many authors, including Arnold [33AR1], Threlkeld [70TH1], Berliner [75BE1], Nahavandi and Oellinger [77NA1], Kettleborough [81KE1], Kern [83KE1] and Majumdar et al. [83MA1] erroneously refer to the Lewis factor as the Lewis number.

The term ($h_c/h_D c_{pm}$) is called the "convective Lewis number" by Close and Banks [74CL1] and Sutherland [83SU1], while Sherwood et al.[75SH1] and Peterson [84PE1] refer to the Lewis factor as the "psychrometric ratio".

Several investigators have studied the Lewis factor and various empirical relations have been proposed.

<u>Chilton and Colburn</u> [34CH1] used experimental data to show that

$$\frac{h_c}{h_D\, c_{pm}} = Le^{0,67} \tag{B.12}$$

According to Cussler [84CU1] the exponent in the Chilton-Colburn relation does not represent the best fit on the experimental data, but it facilitated easier calculations with slide rules.

<u>Bedingfield and Drew</u> [50BE1] obtained data on the heat and mass transfer by studying solid cylinders of volatile solids such as naphthalene in a normal gas flow. The data was correlated by the following relation

$$\frac{h_c}{h_D} \approx 1230,7 \quad (Sc)^{0,56} \tag{B.13}$$

$$\therefore \frac{h_c}{h_D\, c_{pm}} \approx \frac{1230,7\ (Pr)^{0,56}}{c_{pm}}\ (Le)^{0,56}$$

If the non-condensable gas is air this simplifies to

$$\frac{h_c}{h_D\ c_{pm}} \approx (Le)^{0,56} \tag{B.14}$$

in the temperature range normally encountered in evaporative coolers and condensers.

Boelter et al.[65BO1] gave the following relation for the Lewis factor for natural convection systems

$$\frac{h_c}{h_D\ c_{pm}} = (Le)^{C_1} \tag{B.15}$$

where

$$\frac{2}{3} < C_1 < \frac{3}{4}$$

For laminar and turbulent airflow Bosǰnakovic [60BO1] proposed the following correlation for the Lewis factor, i.e.

$$\frac{h_c}{h_D\ c_{pm}} = \frac{(\zeta - 1)}{\ln \zeta} \left(\frac{\nu_a}{\nu_{ma}}\right)^{\left(\frac{5}{12}\right)} (Le)^{0,67} \tag{B.16}$$

where

$$\zeta = \frac{0,622 + w_{asw}}{0,622 + w_a}$$

Assuming that (ν_a / ν_{ma}) ≈ 1 this becomes

$$\frac{h_c}{h_D\ c_{pm}} = (Le)^{0,67} \frac{\left[\dfrac{0,622 + w_{asw}}{0,622 + w_a} - 1\right]}{\ln\left[\dfrac{0,622 + w_{asw}}{0,622 + w_a}\right]} \tag{B.17}$$

According to Berman [61BE1] the Lewis factor can be expressed as

$$\frac{h_c}{h_D\ c_{pm}} = \frac{p_{atm} - p_v}{p_{atm}} \tag{B.18}$$

for air/water systems.

Mizushina et al.[59MI1] assumed the following relation to hold in their study on the operation of spray condensers.

$$\frac{h_c}{h_D\ c_{pm}} = (Le)^{0,5} \tag{B.19}$$

Threlkeld [70TH1] expressed the Lewis factor as

$$\frac{h_c}{h_D\ c_{pm}} \approx (Le)^{C_1} \tag{B.20}$$

where

$0,6 < C_1 < 0,7$

By using an analytical approach Arnold [33AR1] showed that the Lewis factor can be expressed as

$$\frac{h_c}{h_D\ c_{pm}} = \frac{Le\ Pr + \left(\frac{1 - r}{r}\right)}{Pr + \left(\frac{1 - r}{r}\right)} \tag{B.21}$$

where

$$r = \frac{v_{1s1}}{v_\infty}$$

The relation derived by Arnold shows some interesting points. If the

free stream velocity nears zero then the ratio r would approach unity and the Lewis factor would approach the Lewis number. If the free stream velocity increases to infinity the ratio r becomes zero and the Lewis factor would approach a value of unity, regardless of the Lewis number.
The Arnold relation shows that the Lewis factor will have values ranging from the Lewis number to unity depending on the free stream velocity.

Various other investigators expressed the Lewis factor as a constant value, eg.

$$\frac{h_c}{h_D\, c_{pm}} = C_1 \tag{B.22}$$

Foust et al.[80FO1] gave C_1 as $0{,}98 < C_1 < 1{,}13$ for turbulent airflow, while Sherwood [75SH1] reported values of C_1 varying from 0,95 to 1,12.

In cooling tower theory it has been customary to assume a C_1 value of unity since this simplified the theoretical model substantially.

According to the ASHRAE Handbook of Fundamentals [85AS1], the value of C_1 in equation (B.22) should be taken as unity for turbulent air flows since the eddy diffusion in turbulent flow involves the same macroscopic mixing action for heat exchange as for mass exchange, and this completely overwhelms the contribution of molecular diffusion.

APPENDIX C

DEFINITION OF MASS TRANSFER COEFFICIENTS AND MASS TRANSFER DRIVING POTENTIALS

Single phase mass transfer in a binary mixture takes place via a phenomenon known as molecular diffusion. The basic relation describing molecular diffusion is called Fick's law. This states that the mass flux is proportional to the concentration gradient as follows

$$\left(\frac{m}{A}\right)_{rel} = -D\,\frac{\partial c}{\partial y} \tag{C.1}$$

The subscript rel in the massflux term indicates that the massflux given by this relation is expressed in respect to moving coordinates. This is the massflow observed by an observer travelling with the bulk flow.

The absolute mass flux relative to a stationary observer would be given by

$$\left(\frac{m}{A}\right)_{abs} = \left(\frac{m}{A}\right)_{rel} + cv_{bulk} \tag{C.2}$$

Mass transfer between different phases is known as convective mass transfer. Experiments on convective mass transfer have shown that the transfer of mass across an interface can be expressed by a relation of the form:

rate of transfer = transfer coefficient x area x driving potential (C.3)

This form of rate equation corresponds to the form given to governing mass transfer equations in the bulk of the literature. An equation of this form expresses the mass transfer relative to stationary coordinates.

Bird, Stewart and Lightfoot [66BI1] stated that the mass transfer coefficient, as defined by equation (C.3), is independent of the mass transfer rate at only very low mass transfer rates. Thus mass transfer coefficients defined with respect to stationary coordinates would be dependant on the massflow rate at high massflow rates.
This effect arises from the distortion of the velocity and concentration profiles by the high massflow rate across the interface.

Various driving potentials for mass transfer are employed in the literature. The more popular driving potentials used include concentration difference, mass fraction difference, mole fracture difference, vapour pressure difference and thermodynamic activity difference.

In mass transfer processes across a phase interface three resistances to the mass transfer are encountered; the liquid phase, the interface itself and the gas (vapour) phase. Various authors including Treybal [55TR1], Bird et al.[66BI1], Skelland [74SK1] and Foust et al. [80FO1] have studied interphase mass transfer by defining an overall mass transfer coefficient and an overall driving potential.

Treybal [55TR1] used concentration differences as the driving potential in the liquid phase and partial pressure differences as the driving potential in the gas phase to derive a simple governing relation for interphase mass transfer as

$$\left(\frac{m}{A} \right) = K_g \left(p_g - p^* \right) \tag{C.4}$$

where

$$K_g = \left(\frac{1}{k_g} + \frac{a}{k_l} \right)^{-1}$$

The coefficients k_g and k_l are defined by the following single phase mass transfer equations

$$\left(\frac{m}{A}\right) = k_g \left(p_g - p_i\right) \tag{C.5}$$

$$= k_l \left(c_l - c_i\right) \tag{C.6}$$

It is assumed that the vapour pressure at the interface is a linear function of the liquid concentration at the interface as expressed by

$$p_i = ac_i + b \tag{C.7}$$

$$\therefore a = \frac{\partial p_i}{\partial c_i} \tag{C.8}$$

The composition p^* does not physically exist, but it represents a gas (vapour) phase composition which would be in equilibrium with the average liquid composition at the point under consideration.

In cooling tower theory where the mass transfer involves the evaporation of water into air, the driving potentials which are normally used are humidity ratio differences or vapour pressure differences. The governing mass transfer equation can thus be expressed as

$$\left(\frac{m}{A}\right) = h_{Di} \left(w_{asi} - w_a\right) \tag{C.9}$$

or

$$\left(\frac{m}{A}\right) = h_{Dpi} \left(p_{asi} - p_a\right) \tag{C.10}$$

Since the interface temperatures are not always easy to determine in cooling tower applications it has been customary to use the average water temperature instead of the interface temperature to define the mass transfer coefficients. Equations (C.9) and (C.10) could then be written as

$$\left(\frac{m}{A}\right) = h_D \left(w_{asw} - w_a \right) \tag{C.11}$$

and

$$\left(\frac{m}{A}\right) = h_{Dp} \left(p_{asw} - p_a \right) \tag{C.12}$$

Berman [61BE1] showed how h_{Dp} values could be converted to h_D values. Following the method of Berman the relation between h_D and h_{Dp} can now be determined.

From the definition of the absolute humidity ratio it follows that

$$p_a = \left(\frac{w_a}{w_a + 0{,}622} \right) p_{atm}$$

Since the term ($w_a/0{,}622$) is much smaller than unity for air water systems the term [$w_a/(w_a + 0{,}622)$] can be simplified as follows

$$\left(\frac{w_a}{w_a + 0{,}622} \right) \approx \frac{w_a}{0{,}622} \left(1 - \frac{w_a}{0{,}622} \right)$$

$$\therefore p_a \approx \frac{w_a}{0{,}622} \left(1 - \frac{w_a}{0{,}622} \right) p_{atm} \tag{C.13}$$

similarly

$$p_{asw} = \frac{w_{asw}}{0{,}622} \left(1 - \frac{w_{asw}}{0{,}622} \right) p_{atm} \tag{C.14}$$

Setting equations (C.13) and (C.14) into equation (C.12) leads to

$$\left(\frac{m}{A}\right) \approx h_{Dp}\left[\frac{w_{asw}}{0,622}\left(1-\frac{w_{asw}}{0,622}\right)-\frac{w_a}{0,622}\left(1-\frac{w_a}{0,622}\right)\right]p_{atm}$$

$$= h_{Dp}\left[\frac{w_{asw}-w_a}{0,622}-\frac{\left(w_{asw}+w_a\right)\left(w_{asw}-w_a\right)}{(0,622)^2}\right]p_{atm}$$

$$= h_{Dp}\left[\frac{w_{asw}-w_a}{0,622}\left[1-\left(\frac{w_{asw}+w_a}{0,622}\right)\right]\right]p_{atm}$$

Comparing this result equation (C.11) we note that

$$h_D = h_{Dp}\left(\frac{p_{atm}}{0,622}\right)\left[1-\left(\frac{w_{asw}+w_a}{0,622}\right)\right] \tag{C.15}$$

If it is further more assumed that

$$\left[1-\left(\frac{w_{asw}+w_a}{0,622}\right)\right] \approx 1$$

equation (C.15) can be further simplified to

$$h_D = h_{Dp}\left(\frac{p_{atm}}{0,622}\right) \tag{C.16}$$

Berman stressed that care should be taken when converting h_{Dp} values into h_D values, since considerable errors may be introduced because of the simplifications used. The reason for this lies in the fact that the relatively small errors made in the simplifications may be significant when compared to the driving potential ($p_{asw} - p_a$).

Various analytical models for the determination of the mass transfer coefficient exist in the literature, the most prominent models are the "two-film" theory of Whitman [23WH1], the "penetration" model of Higbie [35HI1], the "surface renewal" theory of Danckwerts [51DA1] and the "film penetration" theory of Toor and Marchello [58TO1].

In cooling tower theory empirical relations are normally used to determine the mass transfer coefficient. Chapter 3 gives a summary of available mass transfer coefficient correlations which apply to the operation of evaporative coolers and condensers.

APPENDIX D

SINGLE PHASE PRESSURE DROP ACROSS PLAIN TUBE BUNDLES IN CROSS-FLOW

The pressure drop, Δp, in cross-flow across a tube bundle is given by

$$\Delta p = Kn \frac{\rho v^2}{2} \tag{D.1}$$

where

K = f (Re, geometrical constants) and

$$Re = \frac{\rho v d}{\mu} \tag{D.2}$$

Here, K is the pressure loss coefficient, n characterizes the number of rows in the bundle, Re is the Reynolds number, d is the characteristic length, v is the characteristic velocity, ρ is the density and μ is the dynamic viscosity of the fluid.

Various choices of d, n and v are used in the literature. Equation (D.1) is valid for an ideal tube bundle. An ideal tube bundle is defined as a tube bundle which conforms to the following

i) the velocity in the free cross section is constant,
ii) the velocity is perpendicular to the tube bundle,
iii) the flow is isothermal,
iv) the number of tube rows $\geq$ 10,
v) the number of tubes per row $\geq$ 10,
vi) and the ratio of the tube length to diameter $\geq$ 10.

Deviations from the ideal situation are allowed for by the use of correction factors. The different tube configurations and the geometrical parameters which have an influence on the pressure drop coefficient are shown in Figure D.1.

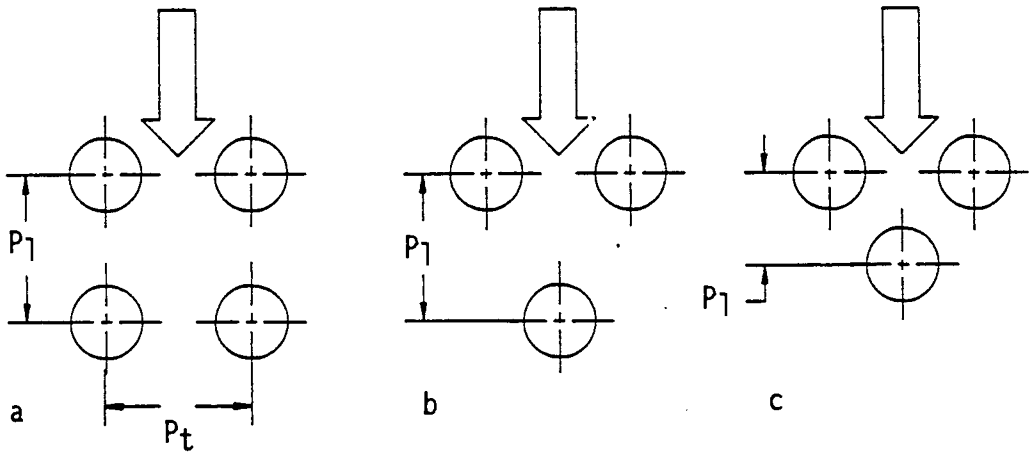

Figure D.1 Tube array configurations: a) in-line, b) staggered with the narrowest cross section perpendicular to the air flow and c) staggered with the narrowest cross section along the diagonals.

The following parameters are used in the calculation of the pressure drop across a tube bundle:

$a = P_t / d_o$, $b = P_l / d_o$, $c = P_d / d_o$

$$d_{eb} = \left\{ \frac{4ab - \pi}{\pi} \right\} d_o \qquad (D.3)$$

For in-line tubes and for staggered tubes with the narrowest cross section perpendicular to the flow the following parameters are used

$$d_{ec} = \left\{ \frac{\pi}{2} \right\} \left\{ \frac{4a - \pi}{\pi} \right\} d_o \qquad (D.4)$$

$$v_m = \frac{1}{\left\{ \frac{4a - \pi}{4a} \right\}} v_\infty \qquad (D.5)$$

$$v_{max} = \frac{a}{(a - 1)} v_\infty \qquad (D.6)$$

For staggered tubes with the narrowest cross section along the diagonal the following parameters are used

$$d_{ec} = \left\{ \frac{\pi}{2} \right\} \left\{ \frac{4c - \pi}{\pi} \right\} d_o \tag{D.7}$$

$$v_m = \frac{(a/(2c))}{\left\{ \frac{4c - \pi}{4c} \right\}} v_\infty \tag{D.8}$$

$$v_{max} = \frac{a}{2 (c - 1)} v_\infty \tag{D.9}$$

According to Bell [63BE1] the flow through the tube bundle will be laminar if Re < 100 and turbulent if Re > 4000. The flow is in the so-called intermediate regime when 100 < Re < 4000.

<u>Chilton and Generaux</u> [33CH1] proposed different equations for pressure loss coefficient in laminar and turbulent flow across tube bundles.

If the flow is **laminar** the proposed relations for this method are

$$n = \frac{L}{d_{eb}} \quad , \quad d = d_{eb} \quad , \quad v = v_{max}$$

and

$$K = \frac{106}{Re}$$

If the flow is **turbulent** then the following relations are to be used

$$n = n_{rows} \quad , \quad d = (a - 1) d_o \quad , \quad v = v_{max}$$

and

$$K = \frac{1{,}32}{Re^{0{,}2}}$$

for a staggered configuration or,

$$K = \frac{3,0}{Re^{0,2}}$$

for an in-line configuration.

Jakob [38JA1] proposed the following equation for determining the pressure drop coefficient when the flow through the tube bundle is **turbulent**.

For a staggered tube layout

$$K = \frac{1}{Re^{\,0,16}} \left\{ 1 + \frac{0,47}{(a - 1)^{1,06}} \right\}$$

and for an in-line tube layout

$$K = \frac{1}{Re^{0,15}} \left\{ 0,176 + \frac{0,32}{(a - 1)^{(0,43 + (1,13/b))}} \right\}$$

where

$$n = n_{rest}, \quad d = d_o \quad , \quad v = v_{max}$$

Gunter and Shaw [45GU1] proposed the following equations to determine the pressure drop coefficient of laminar flow across a tube bundle.

For a staggered layout

$$K = \frac{180}{Re} \left\{ \frac{4\ ab - \pi}{\pi} \right\}^{0,4} \left\{ \frac{b}{a} \right\}^{0,6}$$

and for an in-line layout

$$K = \frac{180}{Re}\left\{\frac{4\ ab - \pi}{\pi}\right\}^{0,4}\left\{\frac{c}{a}\right\}^{0,6}$$

where

$$n = \frac{L}{d_{eb}}, \quad d = d_{eb}, \quad v = v_{max}$$

Gunter and Shaw also proposed equations to determine the pressure drop coefficient when the flow through the tube bank is turbulent.

For a staggered layout

$$K = \frac{1,92}{Re^{0,145}}\left\{\frac{4\ ab - \pi}{\pi}\right\}^{0,4}\left\{\frac{c}{a}\right\}^{0,6}$$

and for an in-line layout

$$K = \frac{1,92}{Re^{\ 0,145}}\left\{\frac{4\ ab - \pi}{\pi}\right\}^{0,4}\left\{\frac{b}{a}\right\}^{0,6}$$

where

$$n = \frac{L}{d_{eb}}, \quad d = d_{eb}, \quad v = v_{max}$$

<u>Bergelin et al</u>.[50BE2] gave the following equation to determine the pressure drop coefficient across a tube bundle if the flow is laminar.

If the layout is staggered with $b \leq \frac{1}{2}(2a + 1)^{0,5}$

$$K = \frac{280}{Re}\left\{\frac{1}{c}\right\}^{1,6}$$

If the layout is in-line or if the layout is staggered with

$$b \geq \tfrac{1}{2} \left(2a + 1 \right)^{0,5}$$

where

$$K = \frac{280}{Re} \left\{ \frac{1}{a} \right\}^{1,6}$$

where

$n = n_{rest}$, $d = d_{eb}$, $v = v_{max}$

Zukauskas [68ZU1] presented graphs to determine the pressure drop coefficient for both laminar and turbulent flow across tube bundles. According to Zukauskas the pressure drop coefficient can be written as

$$K = \frac{K_i}{k_1}$$

where k_1 is a constant which is determined by the geometry of the tube configuration. Zukauskas used the following characteristic values, $n = n_{rows}$, $d = d_o$ and $v = v_{max}$ to determine the Reynolds number and pressure drop coefficient. Figures D.2 and D.3 are reproductions of the graphs given by Zukauskas.

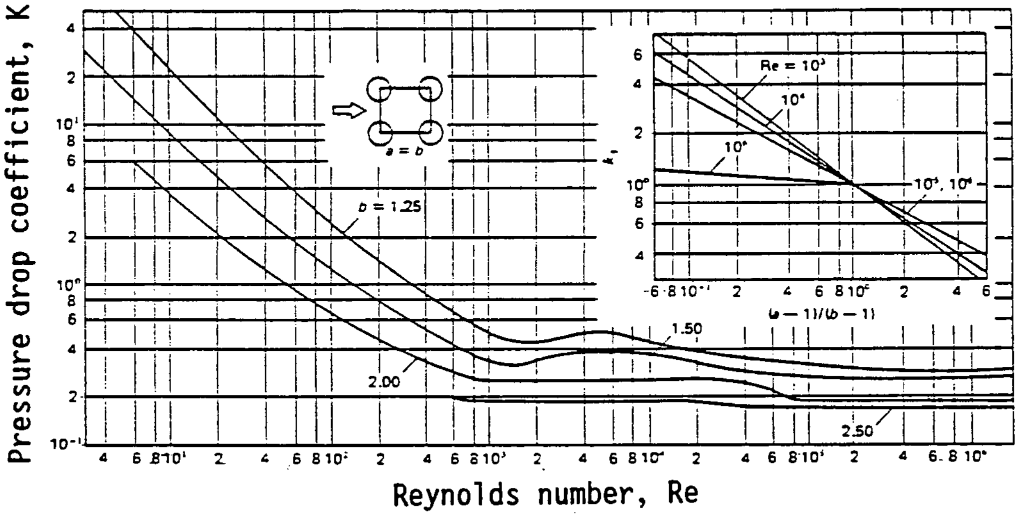

Figure D.2 - Pressure drop coefficient for in-line tube banks.

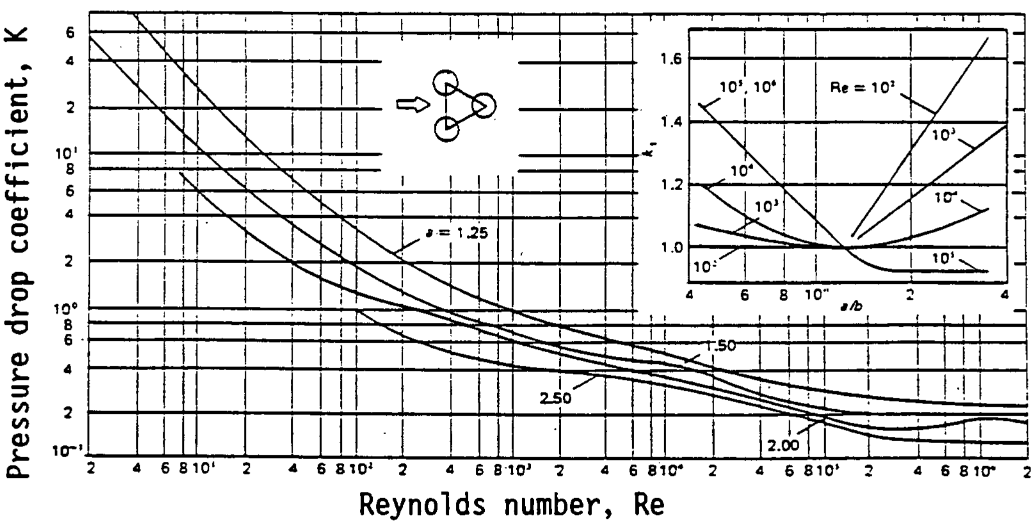

Figure D.3 - Pressure drop coefficient for staggered tube banks.

Zukauskas and Ulinskas [83ZU1] presented previous data [68ZU1] in the form of equations.

Kast [74KA1] proposed an equation for the pressure drop coefficient for staggered tube bundles with the narrowest cross section perpendicular to the direction flow. Charts were given for in-line tube bundles and staggered tube bundles where the narrowest cross section is along the diagonal.

Kast used the following characteristics values for the determination of the Reynolds number and pressure drop:

$$n = \frac{d_o}{d_{ec}} n_{rows}, \qquad d = d_{ec}, \quad v = v_m$$

The equation for K for staggered tube bundles with the narrowest cross section perpendicular to the direction of flow is given as

$$K = \left\{ \frac{128}{Re} + \frac{4}{Re^{0,16}} \right\}$$

Note that this equation holds for all flow regimes from laminar to turbulent. Figure D.4 and Figure D.5 gives the charts for determining the pressure drop coefficient for the in-line tube configuration and the staggered layout (when the narrowest cross section is along the diagonal) respectively.

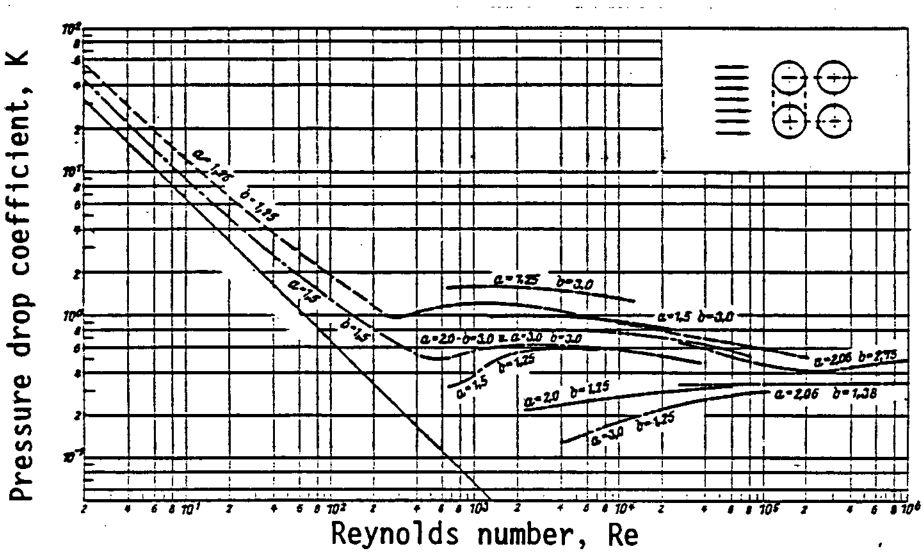

Figure D.4 - Pressure drop coefficient for in-line tube bundles.

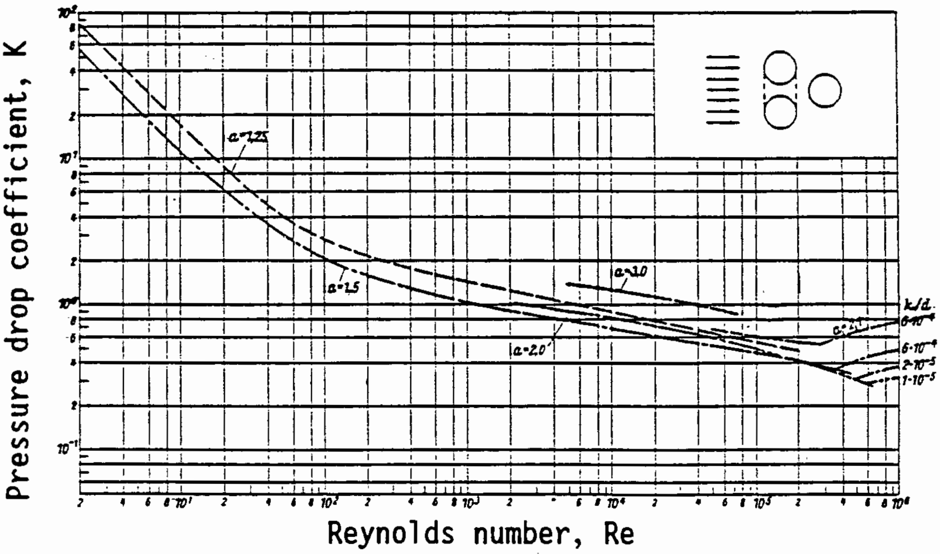

Figure D.5 - Pressure drop coefficient for staggered tube bundles where the narrowest cross section is along the diagonal.

Gaddis and Gnielinski [85GA1] developed comprehensive equations for the pressure drop coefficient through in-line and staggered tube bundles. These equations take into account the effect of number of rows of tubes and the effect of heating/cooling of the fluid.

The following characteristic parameters were used

$n = n_{rest}$, $d = d_o$, $v = v_{max}$

These equations are valid for the following ranges $1 \leq Re \leq 3 \times 10^5$ and $n_{rows} \geq 5$.

For an **in-line** configuration of tubes,

$$K = K_{i,l} \; f_{zn,l} + \left(K_{i,t} \; f_{z,t} + f_{n,t} \right) \left\{ 1 - \exp \left\{ - \frac{Re + 1000}{2000} \right\} \right\}$$

and for a staggered configuration of tubes,

$$K = K_{i,l}\, f_{zn,l} + \left(K_{i,t}\, f_{z,t} + f_{n,t} \right) \left\{ 1 - \exp\left\{ - \frac{Re + 200}{1000} \right\} \right\}$$

where

i) $K_{i,l} = \dfrac{f_{a,l}}{Re}$

For an in-line configuration or a staggered configuration with the narrowest cross section perpendicular to the flow,

$$f_{a,l} = \frac{280\,\pi \left[(b^{0,5} - 0,6)^2 + 0,75 \right]}{(4ab - \pi)a^{1,6}}$$

while for a staggered arrangement with the narrowest cross section along the diagonal

$$f_{a,l} = \frac{280\,\pi \left[(b^{0,5} - 0,6)^2 + 0,75 \right]}{(4\,ab - \pi)\, c^{1,6}}$$

ii) For an in-line configuration

$$K_{i,t} = \frac{f_{a,t,il}}{Re^{0,1}\,(b/a)}$$

and for a staggered configuration

$$K_{i,t} = \frac{f_{a,t,st}}{Re^{0,25}}$$

with

$$f_{a,t,il} = \left\{ 0,22 + 0,12 \frac{\left\{ 1 - \frac{0,94}{b} \right\}^{0,6}}{(a - 0,85)^{1,3}} \right\} \times 10 \left[0,47 \left(b/a - 1,5 \right) \right]$$

$$+ 0,03 (a - 1) (b - 1)$$

and

$$f_{a,t,st} = 2,5 + \frac{1,2}{(a - 0,85)^{1,06}} + 0,4 \left\{ \frac{b}{a} - 1 \right\}^{3} - 0,01 \left\{ \frac{a}{b} - 1 \right\}^{3}$$

iii) If $n_{rows} < 10$ then

$$f_{zn,1} = \left\{ \frac{\mu_w}{\mu} \right\}^{k} \quad \text{where } k = \frac{0,57 \left\{ \frac{n_{rows}}{10} \right\}^{0,25}}{\left\{ \left\{ \frac{4ab}{\pi} - 1 \right\} Re \right\}^{0,25}}$$

and if $n_{rows} \geq 10$ then

$$f_{zn,1} = \left\{ \frac{\mu_w}{\mu} \right\}^{k} \quad \text{where } k = \frac{0,57}{\left\{ \left\{ \frac{4ab}{\pi} - 1 \right\} Re \right\}^{0,25}}$$

iv) $f_{z,t} = \left\{ \frac{\mu_w}{\mu} \right\}^{0,14}$

v) If $5 \leq n_{rows} < 10$ then

$$f_{n,t} = K_o \left\{ \frac{1}{n_{rows}} - \frac{1}{10} \right\}$$

and if $n_{rows} \geq 10$ then

$$f_{n,t} = 0$$

For an in-line configuration and for a staggered configuration with the narrowest cross section perpendicular to the direction of flow

$$K_o = \frac{1}{a^2}$$

and for a staggered configuration with the narrowest cross section along the diagonal

$$K_o = \left\{ \frac{2 (c - 1)}{a (a - 1)} \right\}^2$$

Comparison of the different correlations

The pressure drop across a typical bundle of tubes is evaluated with the different correlations method in order to compare the methods.

Example: n_{rows} = 10 staggered layout

$a = 2$

$b = 3\frac{1}{2}$

$c = 2$

$d_o = 38{,}1$ [mm]

$\mu_w \approx \mu = 1{,}8 \times 10^{-5}$ [kg/ms]

$\rho = 1{,}2$ [kg/m^3]

In order to compare the different equations the pressure drop coefficient is based on the so-called Ry-number, proposed by Kröger [88KR1].

The Ry-number is defined as

$$Ry = \frac{\rho v_\infty}{\mu}$$

*Since the different correlations are based on different characteristic values of n,d and v the product of pressure loss coefficient and n was calculated in order to make the results comparable.

The variation of the product of the pressure loss coefficient and the characteristic number of tube rows vs Ry-number is shown in Figure D.6.

*NOTE : For an in-line configuration and for a staggered configuration where the narrowest cross section is perpendicular to the direction of flow, $n_{rest} = n_{rows}$.

For a staggered configuration in which the narrowest cross section is along the diagonals, $n_{rest} = n_{rows} - 1$.

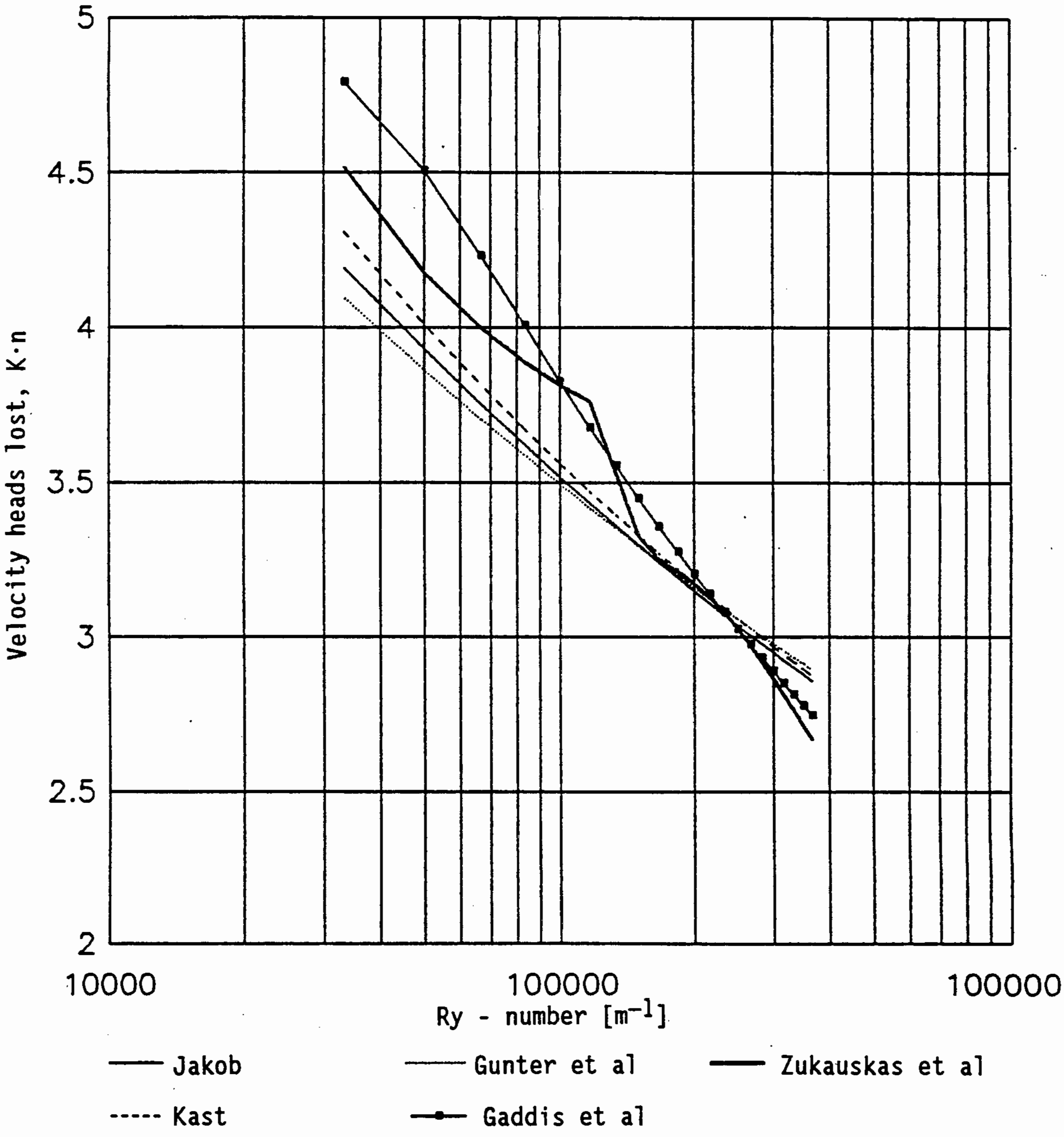

Figure D.6 Single phase pressure drop across a bundle of tubes.

APPENDIX E

DERIVATION OF THE DRAFT EQUATION FOR A NATURAL DRAFT CROSS-FLOW EVAPORATIVE COOLING TOWER

Consider the cross-flow evaporative cooling tower shown in figure E.1, with evaporative cooler units placed around the outer perimeter of the tower base. The density of the heated air inside the tower is lower than the density of the ambient air causing a lower pressure inside the tower than the ambient pressure at the same elevation.
An airflow is induced through the tower as a result of this pressure differential. At the operating point of the tower the air flowrate through the tower would reach a value at which the pressure change due to flow resistances encountered by the airstream and the changes in elevation inside the tower would be in balance with the pressure change, due to elevation change along the outside of the tower.

In the atmosphere outside the tower the following relation describes the pressure change with changing elevation,

$$dp = - \rho g \, dz \qquad (E.1)$$

Assuming air to be a perfect gas the following holds

$$\rho = \frac{p_a}{R_a \, T_a} \qquad (E.2)$$

The dry adiabatic lapse rate in the atmosphere is

$$\frac{dT}{dz} = 0{,}00975 \quad [°C/m]$$

resulting in the following temperature profile in the atmosphere

$$T_a = T_{a1} - 0{,}00975 \, z \qquad (E.3)$$

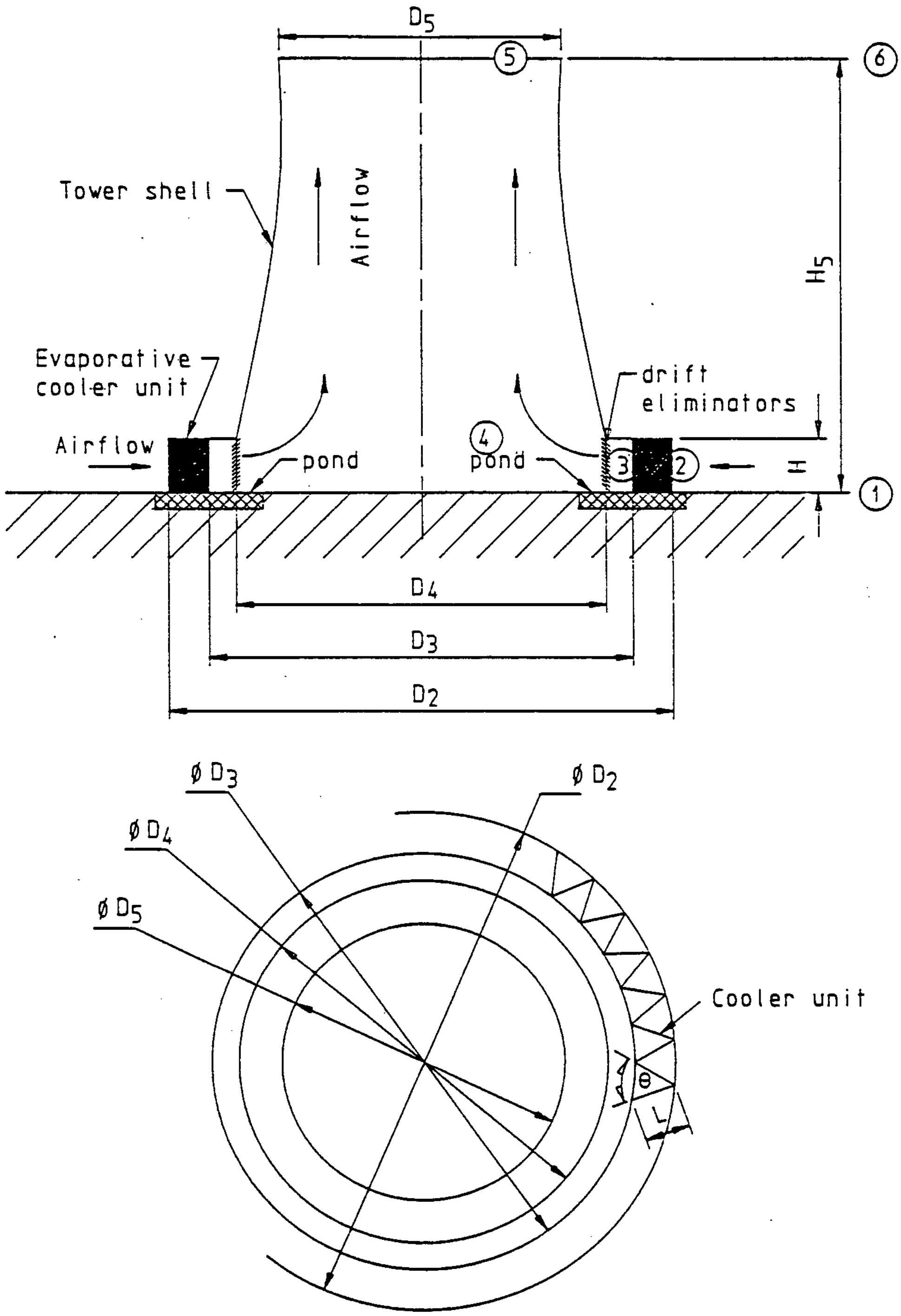

Figure E.1 Layout of natural draft closed circuit cross-flow evaporative cooling tower, showing the reference numbers used in the draft equation.

Substituting equations (E.2) and (E.3) into equation (E.1) and integrating between positions 1 and 6 results in

$$p_{a1} - p_{a6} = p_{a1}\left[1 - \left(1 - \frac{0{,}00975\ H_5}{T_{a1}}\right)^{\left(\frac{g}{0{,}00975\ R_a}\right)}\right] \tag{E.4}$$

with

$R_a \approx 287{,}08$ J/kgK and $g = 9{,}8\ m^2/s$

The air outside the tower accelerates from $v_1 = 0$ m/s at point 1 to v_2 at point 2. Application of the energy equation between points 1 and 2 gives

$$p_{a1} - \left(p_{a2} + \frac{\rho_2\ v_2^2}{2}\right) = p_{a1}\left[1 - \left(1 - \frac{0{,}00975\ H_2}{T_{a1}}\right)^{\left(\frac{g}{0{,}00975 R_a}\right)}\right] \tag{E.5}$$

Between points 2 and 3 the air flows through the evaporative cooler coils and the drop separators. If the coils are positioned in an A-frame configuration there is an additional jetting or oblique flow pressure drop. This can be expressed mathematically as

$$\left(p_{a2} + \frac{\rho_2\ v_2^{\ 2}}{2}\right) - \left(p_{a3} + \frac{\rho_3\ v_3^{\ 2}}{2}\right) = K_{he}\frac{\rho_{23}\ v_{23}^{\ 2}}{2} + K_{de}\frac{\rho_3\ v_3^{\ 2}}{2} + K_{obl}\frac{\rho_{23}\ v_{23}^{\ 2}}{2} \tag{E.6}$$

The flow between positions 3 and 4 changes direction and elevation, expressed by the energy equation as

$$\left(p_{a3} + \frac{\rho_3 v_3^2}{2} \right) - \left(p_{a4} + \frac{\rho_4 v_4^2}{2} \right) = K_{ct} \frac{\rho_4 v_4^2}{2} + \rho_{34} g \left(H_4 - H_3 \right) \tag{E.7}$$

Between positions 4 and 5 the airflow can be described by

$$\left(p_{a4} + \frac{\rho_4 v_4^2}{2} \right) - \left(p_{a5} + \frac{\rho_5 v_5^2}{2} \right) = \rho_{45} g \left(H_5 - H_4 \right) \tag{E.8}$$

The pressure difference between positions 1 and 5 can be determined by substituting equations (E.6), (E.7) and (E.8) into equation (E.5) and simplifying the result as follows,

$$p_{a1} - p_{a5} = K_{he} \frac{\rho_{23} v_{23}^2}{2} + K_{de} \frac{\rho_3 v_3^2}{2} + K_{obl} \frac{\rho_{23} v_{23}^2}{2} + K_{ct} \frac{\rho_4 v_4^2}{2} + \rho_{35} g \left(H_5 - H_3 \right) + \frac{\rho_5 v_5^2}{2} + p_{a1} \left[1 - \left(1 - \frac{0,00975 H_2}{T_{a1}} \right)^{\frac{g}{0,00975 R_a}} \right] \tag{E.9}$$

At the operating point of the tower the pressures inside and outside the tower must be in balance, i.e.

$$p_{a1} - p_{a5} = p_{a1} - p_{a6} \tag{E.10}$$

By substituting equations (E.4) and (E.9) into equation (E.10) the natural draft equation can now be determined as

$$p_{a1}\left[\left(1-\frac{0,00975\ H_2}{T_{a1}}\right)^{\left(\frac{g}{0,00975R_a}\right)}-\left(1-\frac{0,00975\ H_5}{T_{a1}}\right)^{\left(\frac{g}{0,00975R_a}\right)}\right]$$

$$= \rho_{35}\ g\left(H_5 - H_3\right) + K_{he}\frac{\rho_{23}\ v_{23}^{\ 2}}{2} + K_{de}\frac{\rho_3\ v_3^{\ 2}}{2} + K_{obl}\frac{\rho_{23}\ v_{23}^{\ 2}}{2}$$

$$+ K_{ct}\frac{\rho_4\ v_4^{\ 2}}{2} + \frac{\rho_5\ v_5^2}{2} \qquad \text{(E.11)}$$

where

$$\rho_4 = \rho_3$$

$$\rho_{23} = 2\left(\frac{1}{\rho_2} + \frac{1}{\rho_3}\right)^{-1}$$

$$\rho_5 = \frac{p_{a6}}{R_a\left[T_{a3} + \frac{g}{c_{pa}}\left(H_5 - H_2\right)\right]} \approx \frac{p_{a6}}{R_a\left[T_{a3} + 0,00975\left(H_5 - H_2\right)\right]}$$

$$\rho_{35} = \left(\rho_3 + \rho_5\right)/2$$

From continuity it follows that

$$v_{23} = \frac{1}{\rho_{23}}\left(\frac{m_a}{A_{fr}}\right) \qquad \text{(E.12)}$$

$$v_3 = \frac{1}{\rho_3}\left(\frac{m_a}{A_{fr}}\right) \qquad \text{(E.13)}$$

$$v_4 = \frac{1}{\rho_4}\left(\frac{A_{fr}}{A_4}\right)\left(\frac{m_a}{A_{fr}}\right) \qquad \text{(E.14)}$$

$$v_5 = \frac{1}{\rho_5} \left(\frac{A_{fr}}{A_5} \right) \left(\frac{m_a}{A_{fr}} \right) \tag{E.15}$$

Equation (E.11) can be simplified by employing equations (E.12),(E.13), (E.14) and (E.15).

$$\rho_{a1} \left[\left(1 - \frac{0{,}00975\ H_2}{T_{a1}} \right)^{\left(\frac{g}{0{,}00975\ R_a} \right)} - \left(1 - \frac{0{,}00975\ H_5}{T_{a1}} \right)^{\left(\frac{g}{0{,}00975\ R_a} \right)} \right]$$

$$= \left[K_{he} + K_{obl} + K_{de} \left(\frac{\rho_{23}}{\rho_3} \right) + K_{ct} \frac{\rho_{23}}{\rho_4} \left(\frac{A_{fr}}{A_4} \right)^2 + \frac{\rho_{23}}{\rho_5} \left(\frac{A_{fr}}{A_5} \right)^2 \right] \cdot$$

$$\frac{1}{2\ \rho_{23}} \left(\frac{m_a}{A_{fr}} \right)^2 + \rho 35\ g \left(H_5 - H_3 \right) \tag{E.16}$$

Equation (E.16) is the final form of the draft equation for natural convection cooling towers.

The cooling tower loss coefficient for a tower with vertical heat exchangers in the tower inlet was determined by Du Preez and Kröger [88DU1] as

$$K_{ct} = 2{,}98 - 0{,}44 \left(\frac{D_4}{H_4} \right) + 0{,}11 \left(\frac{D_4}{H_4} \right)^2 \tag{E.17}$$

Drift eliminator pressure loss coefficients range between 2,2 and 7,3 according to Chilton [52CH1] and Chan and Golay [77CH1]. A design value of K_{de} = 5 was used throughout this investigation.

The pressure drop and the associated pressure loss coefficient for airflow across a wet tube bundle can be calculated with the correlations presented in Chapter 3.

The oblique flow pressure loss coefficient for a heat exchanger with an A-frame layout was correlated by Kotzé el al. [86KO1] as

$$K_{obl} = \frac{2\rho_3}{\rho_2 + \rho_3} \left(\frac{1}{\sin \theta_m} - 1 \right)^2 + \frac{2\rho_2}{\rho_2 + \rho_3} K_d \qquad (E.18)$$

where

$$\theta_m = 0,0019 \left(\frac{\theta}{2} \right)^2 + 0,9133 \left(\frac{\theta}{2} \right) - 3,1558 \qquad (E.19)$$

and

$$K_d = \exp \left[5,488405 - 0,2131209 \left(\frac{\theta}{2} \right) + 3,533265 \left(\frac{\theta}{2} \right)^2 - 0,2901016 \left(\frac{\theta}{2} \right)^3 \right] \qquad (E.20)$$

APPENDIX F

SOLUTION OF SIMULTANEOUS DIFFERENTIAL EQUATIONS USING THE 4TH ORDER RUNGE-KUTTA METHOD

For the single equation initial value differential equation problem

$$\frac{dy}{dx} = f(x,y)$$

$$y(x_0) = y_0$$

approximate values of y_n must be calculated at point $x_n = x_0 + nh$ where $n = 1,2,3 \ldots$ and h = step size.

The fourth order Runge-Kutta method allows the calculation of y_{n+1} at the point x_{n+1} from the known function value y_n at x_n. According to this method the new function value can be calculated by

$$y_{n+1} = y_n + \left(a_1 + 2a_2 + 2a_3 + a_4 \right) / 6$$

where

$$a_1 = h\, f\left(x_n , y_n \right)$$

$$a_2 = h\, f\left(x_n + h/2, y_n + a_1/2 \right)$$

$$a_3 = h\, f\left(x_n + h/2, y_n + a_2/2 \right)$$

$$a_4 = h\, f\left(x_n + h, y_n + a_3 \right)$$

According to Collatz [86C01] the step size, h, should be chosen such that the values of k_2 and k_3 coincide to within at least two decimal places.

Van Iwaarden [77VA1] shows how the fourth order Runge-Kutta method can be extended to a system of first order initial value problems. Consider the following system of two differential equations and two initial values

$$\frac{dy}{dx} = f(x,y,z)$$

$$\frac{dz}{dx} = g(x,y,z)$$

$$y(x_0) = y_0$$

$$z(x_0) = z_0$$

The fourth order Runge-Kutta method now becomes

$$y_{n+1} = y_n + \left(a_1 + 2a_2 + 2a_3 + a_4 \right) / 6$$

$$z_{n+1} = z_n + \left(b_1 + 2b_2 + 2b_3 + b_4 \right) / 6$$

where

$$a_1 = h\, f\left(x_n, \ y_n, \ z_n \right)$$

$$b_1 = h\, g\left(x_n, \ y_n, \ z_n \right)$$

$$a_2 = h\, f\left(x_n + h/2, \ y_n + a_1/2, \ z_n + b_1/2 \right)$$

$$b_2 = h\, g\left(x_n + h/2, \ y_n + a_1/2, \ z_n + b_1/2 \right)$$

$$a_3 = h\, f\left(x_n + h/2, \ y_n + a_2/2, \ z_n + b_2/2 \right)$$

$$b_3 = h\, g\left(x_n + h/2, \ y_n + a_2/2, \ z_n + b_2/2 \right)$$

$$a_4 = h\, f\left(x_n + h,\ y_n + a_3,\ z_n + b_3 \right)$$

$$b_4 = h\, g\left(x_n + h,\ y_n + a_3,\ z_n + b_3 \right)$$

This method is self starting (no initial estimates are needed) and the new y and z values are calculated after calculating the required a's and b's.

The fourth order Runge-Kutta method can easily be extended to solve any number of simultaneous ordinary differential equations. The following example shows how the fourth order Runge-Kutta method can be used to solve the simultaneous differential equations governing the heat and mass transfer processes of a single element.

The governing differential equations, according to the Merkel model, are

$$dT_p = -K_1 \left(T_p - T_w \right)$$

$$di_a = K_2 \left(i_{asw} - i_a \right)$$

$$dT_w = -K_3 \left(i_{asw} - i_a \right) + K_4 \left(T_p - T_w \right)$$

where $K_1 = \dfrac{U_o\, dA_o}{m_p\, c_{pp}}$

$$K_2 = \frac{h_D\, dA_o}{m_a}$$

$$K_3 = \frac{h_D\, dA_o}{m_w\, c_{pw}}$$

$$K_4 = \frac{U_o\, dA_o}{m_w\, c_{pw}}$$

Assume the following values for the governing variables:

$dA_o = 0{,}25 m^2$
$h_D = 0{,}25$ kg/m^{2}s
$m_a = 0{,}25$ kg/s
$m_w = 0{,}5$ kg/s
$m_p = 0{,}6$ kg/s
$c_{pw} = 4190$ J/kgK
$c_{pp} = 4190$ J/kgK
$T_{pi} = 50°C$
$T_{wi} = 35°C$
$i_{ai} = 55$ kJ/kg
$U_o = 1500$ W/m^{2}K

The constants in the differential equation model can now be determined as

$K_1 = 0{,}1492$
$K_2 = 0{,}25$
$K_3 = 29{,}833 \times 10^{-6}$
$K_4 = 0{,}1790$

The Runge-Kutta method proceeds as follows

<u>Step 1</u>:

$$\Delta i_1 = i_{as}\left(T_{wi}\right) - i_{ai} = 74567 \text{ J/kg}$$

$$\Delta T_1 = T_{pi} - T_{wi} = 15°C$$

$$a_1 = -K_1 \, \Delta T_1 = -0{,}1492\ (15) = -2{,}238$$

$$b_1 = K_2 \, \Delta i_1 = 0{,}25\ (74567) = 18641{,}75$$

$$c_1 = -K_3 \, \Delta i_1 + K_4 \, \Delta T = -29{,}833 \times 10^{-6}(74567) + 0{,}179\ (15) = 0{,}4604$$

Step 2:

$$\Delta i_1 = i_{as}\left[T_{wi} + \frac{c_1}{2}\right] - \left[i_{ai} + \frac{b_1}{2}\right] = 66860{,}125 \text{ J/kg}$$

$$\Delta T_w = \left[T_{pi} + \frac{a_1}{2}\right] - \left[T_{wi} + \frac{c_1}{2}\right] = 13{,}651 \text{ °C}$$

$$\therefore a_2 = -K_1 \, \Delta T_2 = -0{,}1492 \, (13{,}651) = -2{,}0367$$

$$b_2 = K_2 \, \Delta i_2 = 0{,}25 \, (66860{,}125) = 16715{,}031$$

$$c_2 = -K_3 \, \Delta i_2 + K_4 \, \Delta T_2 = -29{,}833 \times 10^{-6} \, (66860{,}125) + 0{,}179 \, (13{,}657) = 0{,}4489$$

Step 3:

$$\Delta i_3 = i_{as}\left[T_{wi} + \frac{c_2}{2}\right] - \left[i_{ai} + \frac{b_2}{2}\right] = 67785{,}095 \text{ J/kg}$$

$$\Delta T_3 = \left[T_{pi} + \frac{a_2}{2}\right] - \left[T_{wi} + \frac{c_2}{2}\right] = 13{,}7573 \text{ °C}$$

$$a_3 = -K_1 \, \Delta T_3 = -0{,}1492 \, (13{,}651) = -2{,}0367$$

$$b_3 = K_2 \, \Delta i_3 = 0{,}25 \, (67785{,}095) = 16946{,}294$$

$$c_3 = -K_3 \, \Delta i_3 + K_4 \, \Delta T_3 = -29{,}833 \times 10^{-6} \, (67785{,}095) + 0{,}179 \, (13{,}7573) = 0{,}4403$$

Step 4:

$$\Delta i_4 = i_{as}\left(T_{wi} + c_3\right) - \left(i_{ai} + b_3\right) = 60645{,}526 \text{ J/kg}$$

$$\Delta T_4 = \left(T_{pi} + a_3\right) - \left(T_{wi} + c_3\right) = 12{,}5071 \text{ °C}$$

$$a_4 = -K_1 \, \Delta T_4 = -0{,}1492 \, (12{,}5071) = -1{,}866$$

$$b_4 = K_2 \, \Delta i_4 = 0{,}25 \, (60645{,}526) = 15161{,}382$$

$$c_4 = -K_3 \, \Delta i_4 + K_4 \, \Delta T_4 = -29{,}833 \times 10^{-6} \, (60645{,}526) + 0{,}179 \, (12{,}5071) = 0{,}4295$$

The other conditions of the element can now be determined as

$$T_{po} = T_{pi} + \left(a_1 + 2 \left(a_2 + a_3 \right) + a_4 \right) / 6 = 47,953°C$$

$$i_{ao} = i_{ai} + \left(b_1 + 2 \left(b_2 + b_3 \right) + b_4 \right) / 6 = 71,854 \text{ kJ/kg}$$

$$T_{wo} = T_{wi} + \left(c_1 + 2 \left(c_2 + c_3 \right) + c_4 \right) / 6 = 35,4447°C$$

APPENDIX G

CORRELATIONS FOR CONVECTIVE AND CONDENSATION HEAT TRANSFER COEFFICIENTS ON THE INSIDE OF TUBES.

Kröger [88KR1] presented a comprehensive summary of the available heat transfer coefficient correlations for the flow of fluids inside ducts, covering the laminar, transitional and turbulent flow regimes.

According to Kays [55KA1] the heat transfer coefficient during laminar flow ($Re_p < 2300$) inside a duct with a constant wall temperature can be expressed by

$$Nu_p = 3,66 + \frac{0,104 \left(Re_p \, Pr_p \left(d_i / L \right) \right)}{1 + 0,016 \left(Re_p \, Pr_p \left(d_i / L \right) \right)^{0,8}} \tag{G.1}$$

Gnielinski [75GN1] proposed the following equation for the heat transfer coefficient on the inside of a tube in the turbulent flow regime

$$Nu_p = \frac{\left(f_D / 8 \right) \left(Re_p - 1000 \right) Pr_p \left[1 + \left(d_i / L \right)^{0,67} \right]}{1 + 12,7 \left(f_D / 8 \right)^{0,5} \left(Pr_p^{\,0,67} - 1 \right)} \tag{G.2}$$

where, the friction factor f_D for smooth tubes is defined by Filonenko [54FI1] as

$$f_D = \left(1,82 \log_{10} Re_p - 1,64 \right)^{-2} \tag{G.3}$$

Equation (G.2) is valid for the following ranges

$2300 < Re_p < 10^6$

$0,5 < Pr_p < 10^4$

$$0 < \left(d_i / L \right) < 1$$

If the fluid properties vary significantly along the flow path the following corrections must be made to the turbulent heat transfer coefficient correlation and the fluid friction factor:

i) The right hand side of equation (G.2) must be multiplied by one of the following correction factors

$$a = \left(Pr / Pr_{wall} \right)_p^{0,11} \qquad \text{(heating)} \qquad (G.4)$$

$$a = \left(Pr / Pr_{wall} \right)_p^{0,25} \qquad \text{(cooling)} \qquad (G.5)$$

ii) The isothermal friction factor must be multiplied by the following correction factor

$$a = \left(\mu_{wall} / \mu \right)_p^{0,25} \qquad (G.6)$$

The following equation proposed by Chato [62CH1] can be used to determine the condensation heat transfer coefficient in essentially horizontal tubes.

$$h_r = 0,555 \left[\frac{g \rho_c \left(\rho_c - \rho_v \right) k_c^3 \, i_{fg}'}{\mu_c \left(T_r - T_{wall} \right) d_i} \right]^{0,25} \qquad (G.7)$$

with

$$i_{fg}' = i_{fg} + 0,68 \, c_{pc} \left(T_r - T_{wall} \right)$$

This equation is only valid for relatively low vapour velocities specified by the range

$$Re_v = \frac{\rho_v v_v d_i}{\mu_v} < 35000 \text{ at the tube inlet.}$$

For higher vapour velocities it is advisable to use the correlation proposed by Shah [79SH1],

$$h_r = h_L \left[0,55 + 2,09 \left(p_{crit} / p_v \right)^{0,38} \right] \tag{G.8}$$

where h_L is given as

$$h_L = 0,023 \, Re_c^{0,8} \, Pr_c^{0,4} \left(k_c / d_i \right) \tag{G.9}$$

with

$$Re_c = \left(\frac{\rho_c \, v_v \, d_i}{\mu_c} \right) \tag{G.10}$$

$$Pr_c = \left(\frac{c_{pc} \, \mu_c}{k_c} \right) \tag{G.11}$$

APPENDIX H

DETERMINATION OF THE AIR/WATER INTERFACE TEMPERATURE

The convective mass transfer coefficient between a water film and an airstream (see Appendix C) is expressed as

$$dm_w = h_{Di} \left(w_{asi} - w_a \right) dA \tag{H.1}$$

but since the interface temperature T_i is difficult to determine, the assumption $T_w \approx T_i$ has often been made in cooling tower theory. The mass transfer is then expressed as

$$dm_w = h_D \left(w_{asw} - w_a \right) dA \tag{H.2}$$

A simple model is now proposed for the determination of the interface temperature. The assumption of $T_i \approx T_w$ does not have to be made when this model is employed. Consider the typical temperature profile in Figure H.1.

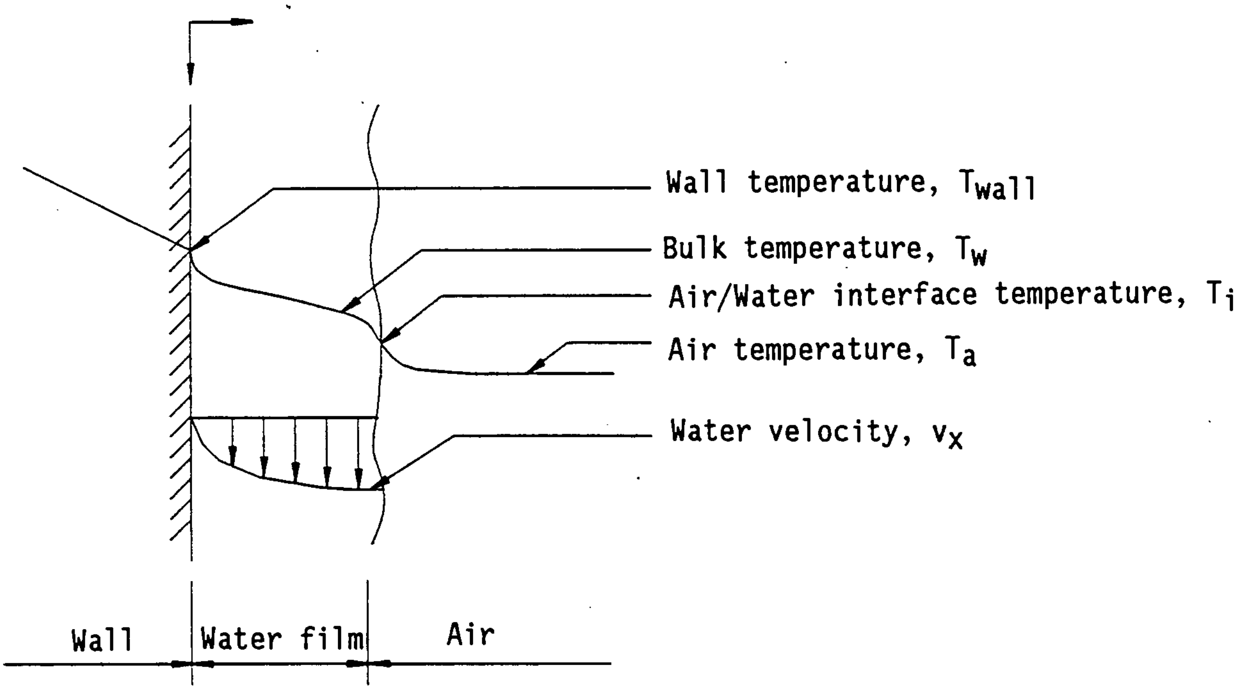

Figure H.1 Schematic representation of the water film flowing down a vertical surface.

If a linear temperature profile is assumed through the water film this profile can be determined as

$$T(y) = \frac{\left(T_i - T_{wall} \right)}{\delta} \, y + T_{wall} \qquad (H.3)$$

From the Nusselt analysis of condensation on an inclined surface the velocity profile in the liquid film is given as

$$v_x = \frac{\left(\rho_w - \rho_a \right) g \sin \theta}{\mu_w} \left[y\delta - \frac{y^2}{2} \right] \qquad (H.4)$$

According to the definition of bulk recirculating water temperature it follows that

$$T_w = \frac{\int_0^\delta \left(c_{pw} \, \rho_w \, v_x \, T \right) dy}{\int_0^\delta \left(c_{pw} \, \rho_w \, v_x \right) dy} \qquad (H.5)$$

By substitution of equations (H.3) and (H.4) into equation (H.5) and assuming c_{pw} to be constant it follows that

$$T_w = \frac{5}{8} \, T_i + \frac{3}{8} \, T_{wall} \qquad (H.6)$$

By defining the film coefficient as

$$q'' = h_{wi} \left(T_{wall} - T_i \right) \qquad (H.7)$$

it follows from equations (H.6) and (H.7) that

$$T_i = T_w - \frac{3}{8} \, \frac{q''}{h_{wi}} \qquad (H.8)$$

If equation (H.8) is used together with the controlling differential equations to evaluate a typical element of an evaporative cooler or condenser, the assumption of $T_w \approx T_i$ does not have to be made.

APPENDIX I

CORRECTION OF HEAT TRANSFER COEFFICIENT AT HIGH MASS TRANSFER RATES

If a water film is in contact with an air stream two heat transfer mechanisms are involved in cooling of the water film, i.e. sensible single phase heat transfer from the water surface to the air and the latent heat transfer associated the mass transfer (evaporation) of a part of the water into the air stream.

This can be mathematically expressed as

$$q'' = h_c F_1 \left(T_i - T_a \right) + h_{Di} \left(w_{asi} - w_a \right) i_v \tag{I.1}$$

The first term in equation (I.1) accounts for the sensible heat transfer and the second term accounts for the latent heat transfer. If only sensible heat transfer took place equation (I.1) would become

$$q'' = h_c \left(T_i - T_a \right) \tag{I.2}$$

Note that the F_1 factor does not appear in equation I.2. The term F_1 accounts for the effect of the mass transfer on the sensible heat transfer when the heat and mass transfer processes take place simultaneously.

Ackermann [37AC1] showed that

$$F_1 = \frac{c_1}{1 - e^{-c_1}} \tag{I.3}$$

where

$$c_1 = \frac{c_{pm} h_{Di} \left(w_{asi} - w_a \right)}{h_c}$$

From the Chilton-Colburn analogy it follows that

$$\frac{h_c}{h_{Di}\ c_{pm}} = (Le)^{2/3}$$

$$\therefore c_1 = \left(w_{asi} - w_a \right) \left(Le \right)^{-2/3} \qquad (I.4)$$

For air/water mixtures at ambient conditions the Lewis number is typically Le = 0,866 ; it follows from equation (I.4) that

$$c_1 \approx 1,1 \left(w_{asi} - w_a \right)$$

In the temperature range 20°C < T_i < 50°C the correction factor typically varies between 1,003 < F_1 < 1,043

The correction F_1 is always larger than unity, since the evaporation of water from the surface gives a net mass flux from the surface in the direction of the heat transfer which increases the heat transfer, because of increased boundary layer activity.

The correction factor is negligible in the normal operating range of an evaporative cooler (20° < T_i < 50°) since it influences the sensible heat transfer, which represents only about 15% of the total heat transfer, by less than 5%.

At higher temperatures this correction factor may become more significant, eg. at $T_i \approx$ 60°C the correction factor is $F_1 \approx$ 1,08 and at $T_i \approx$ 70°C it is $F_1 \approx$ 1,155.

APPENDIX J

EVALUATION OF CONVENTIONAL COOLING TOWER PACKING IN A COMBINATION EVAPORATIVE COOLER

Consider a horizontal slice of a conventional counterflow cooling tower packing with airflow from below and recirculating water flowing through the packing from above. The following Merkel-type equation describes the heat and mass transfer in a section of cooling tower packing of thickness dz,

$$dq = h_D \left(i_{asw} - i_a \right) dA \qquad (J.1)$$

From the energy balance of a section of fill it follows that

$$dq = m_a \, di_a$$

$$= m_w \, c_{pw} \, dT_w \qquad (J.2)$$

By rewriting equations (J.1) and (J.2) the following two controlling differential equations can be found

$$di_a = \frac{h_D}{m_a} \left(i_{asw} - i_a \right) dA \qquad (J.3)$$

and

$$dT_w = \frac{h_D}{m_w \, c_{pw}} \left(i_{asw} - i_a \right) dA \qquad (J.4)$$

For a typical 12mm Munters type extended film packing Cale [77CA1] states that

$$a = 243 \quad m^2/m^3$$

and

$$h_D a = 2,356 \left(\frac{m_w}{A_{fr}} \right) \left(\frac{m_w}{m_a} \right)^{-0,585} \tag{J.5}$$

The surface area of a typical element can be expressed as

$$dA = a \ A_{fr} \ dz \tag{J.6}$$

By employing equation (J.6) equations (J.3) and (J.4) can be expressed as

$$di_a = \frac{h_D \ a}{m_a} \left(i_{asw} - i_a \right) A_{fr} \ dz \tag{J.7}$$

and

$$dT_w = \frac{h_D \ a}{m_w \ c_{pw}} \left(i_{asw} - i_a \right) A_{fr} \ dz \tag{J.8}$$

By using a numerical solution method such as the 4th order Runge-Kutta method these two equations can be numerically integrated through the fill, with the numerical integration starting from the air inlet side. The outlet conditions of the water and air can be determined by a simple iterative search procedure.

APPENDIX K

RESULTS OF COMPUTER SIMULATIONS

A - CROSS-FLOW EVAPORATIVE COOLER SIMULATION

Example No.	Flow pattern	Analytical model	No. of elements
A1	TTB	Merkel	1
A2	TTB	Merkel	5
A3	TTB	Improved Merkel	1
A4	TTB	Poppe	1
A5	FTB	Merkel	1
A6	BTF	Merkel	1
A7	Single pass	Merkel	1
A8	TTB	Simplified model	-

Example A1

Simulation program	:	CROSS
Process water flow layout	:	TOP TO BOTTOM
Analytical model	:	MERKEL

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of cooler unit	=	0.80 m
Width of cooler unit	=	0.80 m
Number of rows of pipes across airstream	=	10
Number of pipes facing the airstream	=	10
Number of elements along a single pipe ..	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Inlet air density	=	1.175 kg/m^3
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.880 kg/s
Air velocity through cooler	=	2.499 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	116.017 kJ/kg
Inlet air humidity ratio	=	0.0120087 kg/kg
Outlet air humidity ratio (saturated) ..	=	0.0323806 kg/kg
Outlet air temperature (saturated)	=	32.844 °C
Outlet air density (saturated)	=	1.132 kg/m^3

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0379 kg/s
Recirculating water temperature in	=	41.214 °C
Recirculating water temperature out	=	41.214 °C

Process water massflow through cooler ..	=	15.000 kg/s
Process water flow velocity in pipes ...	=	1.587 m/s
Process water temperature in	=	50.000 °C
Process water temperature out	=	48.214 °C
Capacity of cooler unit	=	112.383 kW

Example A2

```
Simulation program           : CROSS
Process water flow layout    : TOP TO BOTTOM
Analytical model             : MERKEL

Pipe outer diameter .................... =  38.10 mm
Pipe inner diameter .................... =  34.90 mm
Vertical spacing between pipes ......... =  76.20 mm
Horizontal spacing between pipes ....... =  65.99 mm
Height of cooler unit .................. =   0.80 m
Width of cooler unit ................... =   0.80 m
Number of rows of pipes across airstream =  10
Number of pipes facing the airstream ... =  10
Number of elements along a single pipe . =   5
Fouling coefficient (inside) ........... =     20000.00 W/m2 K
Fouling coefficient (outside) .......... =     20000.00 W/m2 K
Pipe wall conductivity ................. =    43.00 W/mK

Atmospheric pressure .................. =  101.325 kPa
Inlet air temperature (dry bulb) ...... =   25.000 °C
Inlet air temperature (wet bulb) ...... =   19.500 °C
Inlet air density ..................... =    1.175 kg/m3
Dry air massflow through cooler ....... =    1.858 kg/s
Inlet air massflow (inc vapour) ....... =    1.880 kg/s
Air velocity through cooler ........... =    2.499 m/s
Air enthalpy in ....................... =   55.779 kJ/kg
Air enthalpy out (incl. mist) ......... =  116.015 kJ/kg
Inlet air humidity ratio .............. =    0.0120087 kg/kg
Outlet air humidity ratio (saturated) . =    0.0323806 kg/kg
Outlet air temperature (saturated) .... =   32.844 °C
Outlet air density (saturated) ........ =    1.132 kg/m3

Recirc.water massflow / length ........ =  300.0000 kg/m/hr
Inlet recirc.water massflow ........... =    1.3333 kg/s
Recirc. water lost through evaporation  =    0.0379 kg/s
Recirculating water temperature in .... =   41.217 °C
Recirculating water temperature out ... =   41.216 °C

Process water massflow through cooler . =   15.000 kg/s
Process water flow velocity in pipes .. =    1.587 m/s
Process water temperature in .......... =   50.000 °C
Process water temperature out ......... =   48.214 °C
Capacity of cooler unit ............... =  112.380 kW
```

Example A3

```
Simulation program          : CROSS
Process water flow layout   : TOP TO BOTTOM
Analytical model            : IMPROVED MERKEL

Pipe outer diameter .................... =  38.10 mm
Pipe inner diameter .................... =  34.90 mm
Vertical spacing between pipes .......... =  76.20 mm
Horizontal spacing between pipes ........ =  65.99 mm
Height of cooler unit .................. =   0.80 m
Width of cooler unit ................... =   0.80 m
Number of rows of pipes across airstream  =  10
Number of pipes facing the airstream .... =  10
Number of elements along a single pipe .. =   1
Fouling coefficient (inside) ............ =     20000.00 W/m² K
Fouling coefficient (outside) ........... =     20000.00 W/m² K
Pipe wall conductivity .................. =    43.00 W/mK

Atmospheric pressure .................. =  101.325 kPa
Inlet air temperature (dry bulb) ....... =   25.000 °C
Inlet air temperature (wet bulb) ....... =   19.500 °C
Inlet air density ..................... =    1.175 kg/m³
Dry air massflow through cooler ........ =    1.858 kg/s
Inlet air massflow (inc vapour) ........ =    1.880 kg/s
Air velocity through cooler ............ =    2.499 m/s
Air enthalpy in ....................... =   55.779 kJ/kg
Air enthalpy out (incl. mist) .......... =  115.597 kJ/kg
Inlet air humidity ratio ............... =    0.0120087 kg/kg
Outlet air humidity ratio (incl. mist) . =    0.0328293 kg/kg
Outlet air relative humidity ........... =    1.0000000
Outlet air temperature (dry bulb) ...... =   32.738 °C
Outlet air density  .................... =    1.132 kg/m³

Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Inlet recirc.water massflow ............ =    1.3333 kg/s
Outlet recirc. water massflow .......... =    1.2946 kg/s
Recirc. water lost through evaporation . =    0.0387 kg/s
Recirculating water temperature in ..... =   41.242 °C
Recirculating water temperature out .... =   41.241 °C

Process water massflow through cooler .. =   15.000 kg/s
Process water flow velocity in pipes ... =    1.587 m/s
Process water temperature in ........... =   50.000 °C
Process water temperature out .......... =   48.226 °C
Capacity of cooler unit ................ =  111.664 kW
```

Example A4

```
Simulation program          : CROSS
Process water flow layout   : TOP TO BOTTOM
Analytical model            : POPPE

Pipe outer diameter ..................... =  38.10 mm
Pipe inner diameter ..................... =  34.90 mm
Vertical spacing between pipes .......... =  76.20 mm
Horizontal spacing between pipes ........ =  65.99 mm
Height of cooler unit ................... =   0.80 m
Width of cooler unit .................... =   0.80 m
Number of rows of pipes across airstream  =  10
Number of pipes facing the airstream .... =  10
Number of elements along a single pipe .. =   1
Fouling coefficient (inside) ............ =     20000.00 W/m^2 K
Fouling coefficient (outside) ........... =     20000.00 W/m^2 K
Pipe wall conductivity .................. =    43.00 W/mK

Atmospheric pressure ................... =  101.325 kPa
Inlet air temperature (dry bulb) ....... =   25.000 °C
Inlet air temperature (wet bulb) ....... =   19.500 °C
Inlet air density ...................... =    1.175 kg/m^3
Dry air massflow through cooler ........ =    1.858 kg/s
Inlet air massflow (inc vapour) ........ =    1.880 kg/s
Air velocity through cooler ............ =    2.499 m/s
Air enthalpy in ........................ =   55.779 kJ/kg
Air enthalpy out (incl. mist) .......... =  120.665 kJ/kg
Inlet air humidity ratio ............... =    0.0120087 kg/kg
Outlet air humidity ratio (incl. mist) . =    0.0405106 kg/kg
Outlet air relative humidity ........... =    1.0000000
Outlet air temperature (dry bulb) ...... =   33.429 °C
Outlet air density  .................... =    1.129 kg/m^3

Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Inlet recirc.water massflow ............ =    1.3333 kg/s
Outlet recirc. water massflow .......... =    1.2804 kg/s
Recirc. water lost through evaporation . =    0.0530 kg/s
Recirculating water temperature in ..... =   41.291 °C
Recirculating water temperature out .... =   41.290 °C

Process water massflow through cooler .. =   15.000 kg/s
Process water flow velocity in pipes ... =    1.587 m/s
Process water temperature in ........... =   50.000 °C
Process water temperature out .......... =   48.222 °C
Capacity of cooler unit ................ =  111.884 kW
```

Example A5

Simulation program : CROSS
Process water flow layout : FRONT TO BACK
Analytical model : MERKEL

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of cooler unit	=	0.80 m
Width of cooler unit	=	0.80 m
Number of rows of pipes across airstream	=	10
Number of pipes facing the airstream	=	10
Number of elements along a single pipe ..	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK
Atmospheric pressure	=	101.325 kPa
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Inlet air density	=	1.175 kg/m^3
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.880 kg/s
Air velocity through cooler	=	2.499 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	115.784 kJ/kg
Inlet air humidity ratio	=	0.0120087 kg/kg
Outlet air humidity ratio (saturated) ..	=	0.0322668 kg/kg
Outlet air temperature (saturated)	=	32.784 °C
Outlet air density (saturated)	=	1.132 kg/m^3
Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0376 kg/s
Recirculating water temperature in	=	41.524 °C
Recirculating water temperature out	=	41.524 °C
Process water massflow through cooler ..	=	15.000 kg/s
Process water flow velocity in pipes ...	=	1.587 m/s
Process water temperature in	=	50.000 °C
Process water temperature out	=	48.221 °C
Capacity of cooler unit	=	111.952 kW

Example A6

```
Simulation program          : CROSS
Process water flow layout   : BACK TO FRONT
Analytical model            : MERKEL
```

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of cooler unit	=	0.80 m
Width of cooler unit	=	0.80 m
Number of rows of pipes across airstream	=	10
Number of pipes facing the airstream	=	10
Number of elements along a single pipe ..	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK
Atmospheric pressure	=	101.325 kPa
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Inlet air density	=	1.175 kg/m^3
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.880 kg/s
Air velocity through cooler	=	2.499 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	116.025 kJ/kg
Inlet air humidity ratio	=	0.0120087 kg/kg
Outlet air humidity ratio (saturated) ..	=	0.0323806 kg/kg
Outlet air temperature (saturated)	=	32.844 °C
Outlet air density (saturated)	=	1.132 kg/m^3
Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0379 kg/s
Recirculating water temperature in	=	41.481 °C
Recirculating water temperature out	=	41.481 °C
Process water massflow through cooler ..	=	15.000 kg/s
Process water flow velocity in pipes ...	=	1.587 m/s
Process water temperature in	=	50.000 °C
Process water temperature out	=	48.115 °C
Capacity of cooler unit	=	118.646 kW

Example A7

```
Simulation program           : CROSS
Process water flow layout    : STRAIGHT THROUGH
Analytical model             : MERKEL

Pipe outer diameter ..................... =  38.10 mm
Pipe inner diameter ..................... =  34.90 mm
Vertical spacing between pipes .......... =  76.20 mm
Horizontal spacing between pipes ........ =  65.99 mm
Height of cooler unit ................... =   0.80 m
Width of cooler unit .................... =   0.80 m
Number of rows of pipes across airstream  =  10
Number of pipes facing the airstream .... =  10
Number of elements along a single pipe .. =   1
Fouling coefficient (inside) ............ =     20000.00 W/m2 K
Fouling coefficient (outside) ........... =     20000.00 W/m2 K
Pipe wall conductivity .................. =    43.00 W/mK

Atmospheric pressure ................... =  101.325 kPa
Inlet air temperature (dry bulb) ....... =   25.000 °C
Inlet air temperature (wet bulb) ....... =   19.500 °C
Inlet air density ...................... =    1.175 kg/m3
Dry air massflow through cooler ........ =    1.858 kg/s
Inlet air massflow (inc vapour) ........ =    1.880 kg/s
Air velocity through cooler ............ =    2.499 m/s
Air enthalpy in ........................ =   55.779 kJ/kg
Air enthalpy out (incl. mist) .......... =   99.850 kJ/kg
Inlet air humidity ratio ............... =    0.0120087 kg/kg
Outlet air humidity ratio (saturated) .. =    0.0272164 kg/kg
Outlet air temperature (saturated) ..... =   29.925 °C
Outlet air density (saturated) ......... =    1.146 kg/m3

Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Inlet recirc.water massflow ............ =    1.3333 kg/s
Recirc. water lost through evaporation . =    0.0283 kg/s
Recirculating water temperature in ..... =   37.535 °C
Recirculating water temperature out .... =   37.535 °C

Process water massflow through cooler .. =   15.000 kg/s
Process water flow velocity in pipes ... =    0.159 m/s
Process water temperature in ........... =   50.000 °C
Process water temperature out .......... =   48.694 °C
Capacity of cooler unit ................ =   82.230 kW
```

Example A8

Simulation program : SCROSS
Process water flow layout : TOP TO BOTTOM
Cooler type : TUBES ONLY
Process fluid : WATER

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of cooler unit	=	0.80 m
Length of cooler unit	=	0.80 m
Number of pipe rows along the airflow ...	=	10
Number of pipes facing the airstream	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through cooler	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	115.740 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0377 kg/s
Recirculating water temperature in	=	41.60 °C
Recirculating water temperature out	=	41.60 °C

Process fluid massflow through cooler ..	=	15.000 kg/s
Process fluid flow velocity in pipes ...	=	1.587 m/s
Process fluid temperature in	=	50.000 °C
Process fluid temperature out	=	48.222 °C
Capacity of cooler unit	=	111.893 kW

B - COUNTERFLOW EVAPORATIVE COOLER SIMULATION

Example No.	Flow pattern	Analytical model	No. of elements
B1	BTT	Merkel	1
B2	BTT	Merkel	5
B3	BTT	Improved Merkel	1
B4	BTT	Poppe	1
B5	TTB	Merkel	1
B6	BTT+Packing	Merkel	1
B7	TTB+Packing	Merkel	1
B8	TTB	Simplified model	-

Example B1

```
Simulation program          : COUNTER
Process water flow layout   : BOTTOM TO TOP
Analytical model            : MERKEL

Pipe outer diameter .................... =  38.10 mm
Pipe inner diameter .................... =  34.90 mm
Vertical spacing between pipes ......... =  66.00 mm
Horizontal spacing between pipes ....... =  76.20 mm
Width of cooler unit ................... =   0.80 m
Length of cooler unit .................. =   0.80 m
Number of rows of pipes across airstream =  10
Number of pipes facing the airstream ... =  10
Number of elements along a single pipe . =   1

Atmospheric pressure .................. = 101.325 kPa
Inlet air temperature (dry bulb) ...... =  25.000 °C
Inlet air temperature (wet bulb) ...... =  19.500 °C
Inlet air density ..................... =   1.175 kg/m³
Dry air massflow through cooler ....... =   1.858 kg/s
Inlet air massflow (inc vapour) ....... =   1.880 kg/s
Air velocity through cooler ........... =   2.499 m/s
Air enthalpy in ....................... =  55.779 kJ/kg
Air enthalpy out (incl. mist) ......... = 118.231 kJ/kg
Inlet air humidity ratio .............. =   0.0120087 kg/kg
Outlet air humidity ratio (saturated) . =   0.0330753 kg/kg
Outlet air temperature (saturated) .... =  33.203 °C
Outlet air density (saturated) ........ =   1.130 kg/m³

Recirc.water massflow / length ........ = 300.0000 kg/m/hr
Inlet recirc.water massflow ........... =   2.6667 kg/s
Recirc. water lost through evaporation  =   0.0391 kg/s
Recirculating water temperature in .... =  41.39 °C
Recirculating water temperature out ... =  41.38 °C

Process water massflow through cooler . =  15.000 kg/s
Process water flow velocity in pipes .. =   1.587 m/s
Process water temperature in .......... =  50.000 °C
Process water temperature out ......... =  48.237 °C
Capacity of cooler unit ............... = 110.955 kW
```

Example B2

```
Simulation program          : COUNTER
Process water flow layout   : BOTTOM TO TOP
Analytical model            : MERKEL

Pipe outer diameter ..................... =  38.10 mm
Pipe inner diameter ..................... =  34.90 mm
Vertical spacing between pipes .......... =  66.00 mm
Horizontal spacing between pipes ........ =  76.20 mm
Width of cooler unit .................... =   0.80 m
Length of cooler unit ................... =   0.80 m
Number of rows of pipes across airstream  =  10
Number of pipes facing the airstream .... =  10
Number of elements along a single pipe .. =   5

Atmospheric pressure ................... = 101.325 kPa
Inlet air temperature (dry bulb) ....... =  25.000 °C
Inlet air temperature (wet bulb) ....... =  19.500 °C
Inlet air density ...................... =   1.175 kg/m3
Dry air massflow through cooler ........ =   1.858 kg/s
Inlet air massflow (inc vapour) ........ =   1.880 kg/s
Air velocity through cooler ............ =   2.499 m/s
Air enthalpy in ........................ =  55.779 kJ/kg
Air enthalpy out (incl. mist) .......... = 118.222 kJ/kg
Inlet air humidity ratio ............... =   0.0120087 kg/kg
Outlet air humidity ratio (saturated) .. =   0.0330753 kg/kg
Outlet air temperature (saturated) ..... =  33.203 °C
Outlet air density (saturated) ......... =   1.130 kg/m3

Recirc.water massflow / length ......... = 300.0000 kg/m/hr
Inlet recirc.water massflow ............ =   2.6667 kg/s
Recirc. water lost through evaporation . =   0.0391 kg/s
Recirculating water temperature in ..... =  41.38 °C
Recirculating water temperature out .... =  41.39 °C

Process water massflow through cooler .. =  15.000 kg/s
Process water flow velocity in pipes ... =   1.587 m/s
Process water temperature in ........... =  50.000 °C
Process water temperature out .......... =  48.237 °C
Capacity of cooler unit ................ = 110.972 kW
```

Example B3

```
Simulation program              : COUNTER
Process water flow layout       : BOTTOM TO TOP
Analytical model                : IMPROVED MERKEL

Pipe outer diameter ..................... =  38.10 mm
Pipe inner diameter ..................... =  34.90 mm
Vertical spacing between pipes .......... =  66.00 mm
Horizontal spacing between pipes ........ =  76.20 mm
Width of cooler unit .................... =   0.80 m
Length of cooler unit ................... =   0.80 m
Number of rows of pipes across airstream  =  10
Number of pipes facing the airstream .... =  10
Number of elements along a single pipe .. =   1

Atmospheric pressure ................... =  101.325 kPa
Inlet air temperature (dry bulb) ....... =   25.000 °C
Inlet air temperature (wet bulb) ....... =   19.500 °C
Inlet air density ...................... =    1.175 kg/m3
Dry air massflow through cooler ........ =    1.858 kg/s
Inlet air massflow (inc vapour) ........ =    1.880 kg/s
Air velocity through cooler ............ =    2.499 m/s
Air enthalpy in ........................ =   55.779 kJ/kg
Air enthalpy out (incl. mist) .......... =  117.697 kJ/kg
Inlet air humidity ratio ............... =    0.0120087 kg/kg
Outlet air humidity ratio (incl. mist) . =    0.0335399 kg/kg
Outlet air relative humidity ........... =    1.0000000
Outlet air temperature (dry bulb) ...... =   33.092 °C
Outlet air density  .................... =    1.131 kg/m3

Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Inlet recirc.water massflow ............ =    2.6667 kg/s
Outlet recirc. water massflow .......... =    2.6267 kg/s
Recirc. water lost through evaporation . =    0.0400 kg/s
Recirculating water temperature in ..... =   41.43 °C
Recirculating water temperature out .... =   41.43 °C

Process water massflow through cooler .. =   15.000 kg/s
Process water flow velocity in pipes ... =    1.587 m/s
Process water temperature in ........... =   50.000 °C
Process water temperature out .......... =   48.251 °C
Capacity of cooler unit ................ =  110.051 kW
```

Example B4

Simulation program	: COUNTER
Process water flow layout	: BOTTOM TO TOP
Analytical model	: POPPE

Pipe outer diameter	= 38.10 mm
Pipe inner diameter	= 34.90 mm
Vertical spacing between pipes	= 66.00 mm
Horizontal spacing between pipes	= 76.20 mm
Width of cooler unit	= 0.80 m
Length of cooler unit	= 0.80 m
Number of rows of pipes across airstream	= 10
Number of pipes facing the airstream	= 10
Number of elements along a single pipe ..	= 1

Atmospheric pressure	= 101.325 kPa
Inlet air temperature (dry bulb)	= 25.000 °C
Inlet air temperature (wet bulb)	= 19.500 °C
Inlet air density	= 1.175 kg/m³
Dry air massflow through cooler	= 1.858 kg/s
Inlet air massflow (inc vapour)	= 1.880 kg/s
Air velocity through cooler	= 2.499 m/s
Air enthalpy in	= 55.779 kJ/kg
Air enthalpy out (incl. mist)	= 119.121 kJ/kg
Inlet air humidity ratio	= 0.0120087 kg/kg
Outlet air humidity ratio (incl. mist) .	= 0.0337335 kg/kg
Outlet air relative humidity	= 1.0000000
Outlet air temperature (dry bulb)	= 32.897 °C
Outlet air density	= 1.132 kg/m³

Recirc.water massflow / length	= 300.0000 kg/m/hr
Inlet recirc.water massflow	= 2.6667 kg/s
Outlet recirc. water massflow	= 2.6263 kg/s
Recirc. water lost through evaporation .	= 0.0404 kg/s
Recirculating water temperature in	= 41.75 °C
Recirculating water temperature out	= 41.75 °C

Process water massflow through cooler ..	= 15.000 kg/s
Process water flow velocity in pipes ...	= 1.587 m/s
Process water temperature in	= 50.000 °C
Process water temperature out	= 48.319 °C
Capacity of cooler unit	= 105.786 kW

<u>Example B5</u>

Simulation program : COUNTER
Process water flow layout : TOP TO BOTTOM
Analytical model : MERKEL

Pipe outer diameter .... = 38.10 mm
Pipe inner diameter .... = 34.90 mm
Vertical spacing between pipes .... = 66.00 mm
Horizontal spacing between pipes .... = 76.20 mm
Width of cooler unit .... = 0.80 m
Length of cooler unit .... = 0.80 m
Number of rows of pipes across airstream = 10
Number of pipes facing the airstream .... = 10
Number of elements along a single pipe .. = 1

Atmospheric pressure .... = 101.325 kPa
Inlet air temperature (dry bulb) .... = 25.000 °C
Inlet air temperature (wet bulb) .... = 19.500 °C
Inlet air density .... = 1.175 kg/m^3
Dry air massflow through cooler .... = 1.858 kg/s
Inlet air massflow (inc vapour) .... = 1.880 kg/s
Air velocity through cooler .... = 2.499 m/s
Air enthalpy in .... = 55.779 kJ/kg
Air enthalpy out (incl. mist) .... = 118.414 kJ/kg
Inlet air humidity ratio .... = 0.0120087 kg/kg
Outlet air humidity ratio (saturated) .. = 0.0331230 kg/kg
Outlet air temperature (saturated) ..... = 33.228 °C
Outlet air density (saturated) .... = 1.130 kg/m^3

Recirc.water massflow / length .... = 300.0000 kg/m/hr
Inlet recirc.water massflow .... = 2.6667 kg/s
Recirc. water lost through evaporation . = 0.0392 kg/s
Recirculating water temperature in ..... = 41.11 °C
Recirculating water temperature out .... = 111.225 °C

Process water massflow through cooler .. = 15.000 kg/s
Process water flow velocity in pipes ... = 1.587 m/s
Process water temperature in .... = 50.010 °C
Process water temperature out .... = 48.243 °C
Capacity of cooler unit .... = 111.225 kW

Example B6

Simulation program	:	COMBINE
Process water flow layout	:	BOTTOM TO TOP
Cooler type	:	TUBES + 300 mm PACK ABOVE
Process fluid	:	WATER

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	65.99 mm
Horizontal spacing between pipes	=	76.20 mm
Width of cooler unit	=	0.80 m
Length of cooler unit	=	0.80 m
Number of pipes facing the airstream	=	10
Number of pipe rows along the airflow ...	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through cooler	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	133.839 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	2.6667 kg/s
Recirc. water lost through evaporation .	=	0.0485 kg/s
Recirculating water temperature in	=	40.39 °C
Recirculating water temperature out	=	40.39 °C

Process fluid massflow through cooler ..	=	15.000 kg/s
Process fluid flow velocity in pipes ...	=	1.587 m/s
Process fluid temperature in	=	50.000 °C
Process fluid temperature out	=	47.751 °C
Capacity of cooler unit	=	141.559 kW

Example B7

Simulation program	: COMBINE
Process water flow layout	: TOP TO BOTTOM
Cooler type	: TUBES + 300 mm PACK ABOVE
Process fluid	: WATER

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	65.99 mm
Horizontal spacing between pipes	=	76.20 mm
Width of cooler unit	=	0.80 m
Length of cooler unit	=	0.80 m
Number of pipes facing the airstream	=	10
Number of pipe rows along the airflow ...	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK
Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through cooler	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	132.670 kJ/kg
Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	2.6667 kg/s
Recirc. water lost through evaporation .	=	0.0478 kg/s
Recirculating water temperature in	=	39.99 °C
Recirculating water temperature out	=	40.00 °C
Process fluid massflow through cooler ..	=	15.000 kg/s
Process fluid flow velocity in pipes ...	=	1.587 m/s
Process fluid temperature in	=	50.012 °C
Process fluid temperature out	=	47.796 °C
Capacity of cooler unit	=	139.469 kW

Example B8

Simulation program	:	SCOUNT
Process water flow layout	:	TOP TO BOTTOM
Cooler type	:	TUBES ONLY
Process fluid	:	WATER

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	65.99 mm
Horizontal spacing between pipes	=	76.20 mm
Width of cooler unit	=	0.80 m
Length of cooler unit	=	0.80 m
Number of pipes facing the airstream	=	10
Number of pipe rows along the airflow ...	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK
Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through cooler	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through cooler	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	115.740 kJ/kg
Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	2.6667 kg/s
Recirc. water lost through evaporation .	=	0.0377 kg/s
Recirculating water temperature in	=	41.60 °C
Recirculating water temperature out	=	41.60 °C
Process fluid massflow through cooler ..	=	15.000 kg/s
Process fluid flow velocity in pipes ...	=	1.587 m/s
Process fluid temperature in	=	50.000 °C
Process fluid temperature out	=	48.222 °C
Capacity of cooler unit	=	111.893 kW

C - EVAPORATIVE CONDENSER SIMULATION

Example No.	Airflow	Refrigerant
C1	Cross	Steam
C2	Cross	R22
C3	Cross	Ammonia
C4	Counter	Steam
C5	Counter	R22
C6	Counter	Ammonia

Example C1

```
Simulation program            : CSCROSS
Condenser type                : TUBES ONLY
Refrigerant                   : STEAM

Pipe outer diameter ..................... =  38.10 mm
Pipe inner diameter ..................... =  34.90 mm
Vertical spacing between pipes .......... =  76.20 mm
Horizontal spacing between pipes ........ =  65.99 mm
Height of condenser unit ................ =   0.80 m
Length of condenser unit ................ =   0.80 m
Number of pipe rows along the airflow ... =  10
Number of pipes facing the airstream .... =  10
Fouling coefficient (inside) ............ = 20000.00 W/m2 K
Fouling coefficient (outside) ........... = 20000.00 W/m2 K
Pipe wall conductivity .................. =    43.00 W/mK

Atmospheric pressure ................... =  101.325 kPa
Altitude (above sea level) ............. =    0.000 m
Inlet air temperature (dry bulb) ....... =   25.000 °C
Inlet air temperature (wet bulb) ....... =   19.500 °C
Dry air massflow through condenser ..... =    1.858 kg/s
Inlet air massflow (inc vapour) ........ =    1.881 kg/s
Air velocity through condenser ......... =    2.500 m/s
Air enthalpy in ........................ =   55.779 kJ/kg
Air enthalpy out (incl. mist) .......... =  119.749 kJ/kg

Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Inlet recirc.water massflow ............ =    1.3333 kg/s
Recirc. water lost through evaporation . =    0.0400 kg/s
Recirculating water temperature (ave) .. =   42.46 °C

Refrigerant massflow through condenser . =  0.04989 kg/s
Condensing temperature ................. =   50.000 °C
Capacity of condenser unit ............. =    118.8800 kW
```

Example C2

Simulation program	:	CSCROSS
Condenser type	:	TUBES ONLY
Refrigerant	:	R22 (Freon 22)

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of condenser unit	=	0.80 m
Length of condenser unit	=	0.80 m
Number of pipe rows along the airflow ...	=	10
Number of pipes facing the airstream	=	10
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through condenser	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through condenser	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	87.996 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0215 kg/s
Recirculating water temperature (ave) ..	=	34.14 °C

Refrigerant massflow through condenser .	=	0.38896 kg/s
Condensing temperature	=	50.000 °C
Capacity of condenser unit	=	59.8717 kW

Example C3

Simulation program	: CSCROSS
Condenser type	: TUBES ONLY
Refrigerant	: R717 (Ammonia)

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of condenser unit	=	0.80 m
Length of condenser unit	=	0.80 m
Number of pipe rows along the airflow ...	=	10
Number of pipes facing the airstream	=	10
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through condenser	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through condenser	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	117.180 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	1.3333 kg/s
Recirc. water lost through evaporation .	=	0.0385 kg/s
Recirculating water temperature (ave) ..	=	41.91 °C

Refrigerant massflow through condenser .	=	0.10860 kg/s
Condensing temperature	=	50.000 °C
Capacity of condenser unit	=	114.1063 kW

Example C4

(Simplified Merkel model.)

Simulation program	: CSCOUNT
Condenser type	: TUBES ONLY
Refrigerant	: STEAM

Pipe outer diameter	= 38.10 mm
Pipe inner diameter	= 34.90 mm
Vertical spacing between pipes	= 65.99 mm
Horizontal spacing between pipes	= 76.20 mm
Width of condenser unit	= 0.80 m
Length of condenser unit	= 0.80 m
Number of pipes facing the airstream	= 10
Number of pipe rows along the airflow ...	= 10
Order of tube serpentining	= 1
Fouling coefficient (inside)	= 20000.00 W/m^2 K
Fouling coefficient (outside)	= 20000.00 W/m^2 K
Pipe wall conductivity	= 43.00 W/mK

Atmospheric pressure	= 101.325 kPa
Altitude (above sea level)	= 0.000 m
Inlet air temperature (dry bulb)	= 25.000 °C
Inlet air temperature (wet bulb)	= 19.500 °C
Dry air massflow through condenser	= 1.858 kg/s
Inlet air massflow (inc vapour)	= 1.881 kg/s
Air velocity through condenser	= 2.500 m/s
Air enthalpy in	= 55.779 kJ/kg
Air enthalpy out (incl. mist)	= 119.412 kJ/kg

Recirc.water massflow / length	= 300.0000 kg/m/hr
Inlet recirc.water massflow	= 2.6667 kg/s
Recirc. water lost through evaporation .	= 0.0399 kg/s
Recirculating water temperature (ave) ..	= 42.39 °C

Refrigerant massflow through condenser .	= 0.04963 kg/s
Condensing temperature	= 50.000 °C
Capacity of condenser unit	= 118.2548 kW

Example C5

Simulation program	:	CSCOUNT
Condenser type	:	TUBES ONLY
Refrigerant	:	R22 (Freon 22)

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	65.99 mm
Horizontal spacing between pipes	=	76.20 mm
Width of condenser unit	=	0.80 m
Length of condenser unit	=	0.80 m
Number of pipes facing the airstream	=	10
Number of pipe rows along the airflow ...	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through condenser	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through condenser	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	87.929 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	2.6667 kg/s
Recirc. water lost through evaporation .	=	0.0215 kg/s
Recirculating water temperature (ave) ..	=	34.12 °C

Refrigerant massflow through condenser .	=	0.38816 kg/s
Condensing temperature	=	50.000 °C
Capacity of condenser unit	=	59.7479 kW

Example C6

Simulation program	:	CSCOUNT
Condenser type	:	TUBES ONLY
Refrigerant	:	R717 (Ammonia)

Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	65.99 mm
Horizontal spacing between pipes	=	76.20 mm
Width of condenser unit	=	0.80 m
Length of condenser unit	=	0.80 m
Number of pipes facing the airstream	=	10
Number of pipe rows along the airflow ...	=	10
Order of tube serpentining	=	1
Fouling coefficient (inside)	=	20000.00 W/m^2 K
Fouling coefficient (outside)	=	20000.00 W/m^2 K
Pipe wall conductivity	=	43.00 W/mK

Atmospheric pressure	=	101.325 kPa
Altitude (above sea level)	=	0.000 m
Inlet air temperature (dry bulb)	=	25.000 °C
Inlet air temperature (wet bulb)	=	19.500 °C
Dry air massflow through condenser	=	1.858 kg/s
Inlet air massflow (inc vapour)	=	1.881 kg/s
Air velocity through condenser	=	2.500 m/s
Air enthalpy in	=	55.779 kJ/kg
Air enthalpy out (incl. mist)	=	117.181 kJ/kg

Recirc.water massflow / length	=	300.0000 kg/m/hr
Inlet recirc.water massflow	=	2.6667 kg/s
Recirc. water lost through evaporation .	=	0.0385 kg/s
Recirculating water temperature (ave) ..	=	41.91 °C

Refrigerant massflow through condenser .	=	0.10860 kg/s
Condensing temperature	=	50.000 °C
Capacity of condenser unit	=	114.1073 kW

D - CROSS-FLOW NATURAL DRAFT COOLING TOWER SIMULATION

Example D1

Simulation program	:	TOWER
Process water flow layout	:	FRONT TO BACK
Analytical model	:	MERKEL

Total tower height	=	147.00 m
Inlet height at pond	=	8.00 m
Diameter of tower at pond	=	105.00 m
Diameter of tower at outlet	=	60.85 m
Number of A-frames around tower	=	24
Included A-frame angle	=	60.00
Face length of each A-frame side	=	15.00 m
Pipe outer diameter	=	38.10 mm
Pipe inner diameter	=	34.90 mm
Vertical spacing between pipes	=	76.20 mm
Horizontal spacing between pipes	=	65.99 mm
Height of cooler unit	=	8.00 m
Width of cooler unit	=	15.00 m
Number of rows of pipes across airstream	=	40
Number of pipes facing the airstream	=	104
Number of elements along a single pipe ..	=	1

Dry air massflow through A-frame side ..	=	182.81 kg/s
Dry air massflow through tower	=	8775.00 kg/s
Air velocity through A-frame side	=	1.53 m/s
Pressure drop across cooler	=	48.95 Pa
Pressure drop (oblique flow)	=	7.30 Pa
Pressure drop across drift eliminators .	=	6.07 Pa
Tower pressure drop	=	8.68 Pa
Tower outlet pressure loss	=	4.79 Pa
Total pressure loss through tower	=	75.79 Pa
Total available buoyancy	=	76.28 Pa
Atmospheric pressure	=	84.000 kPa
Inlet air temperature (dry bulb)	=	15.450 °C
Inlet air temperature (wet bulb)	=	11.050 °C
Inlet air density	=	1.009 kg/m3
Inlet air massflow (inc vapour)	=	184.302 kg/s
Dry air massflow through cooler	=	182.813 kg/s
Dry air massflow through tower	=	8775.002 kg/s
Air enthalpy in	=	36.165 kJ/kg
Air enthalpy out (incl. mist)	=	102.956 kJ/kg
Inlet air humidity ratio	=	0.0081457 kg/kg
Outlet air humidity ratio (saturated) ..	=	0.0293051 kg/kg
Outlet air temperature (saturated)	=	27.907 °C
Outlet air density (saturated)	=	0.955 kg/m^3

```
Recirc.water massflow / length ......... =  300.0000 kg/m/hr
Total inlet recirc.water massflow ...... = 4800.0000 kg/s
Recirc. water lost through evaporation . =  185.6741 kg/s
Recirculating water temperature in ..... =   29.45 °C
Recirculating water temperature out .... =   29.46 °C

Process water massflow through tower ... = 12500.000 kg/s
Process water massflow through cooler .. =   260.417 kg/s
Process water flow velocity in pipes ... =     2.637 m/s
Process water temperature in ........... =    39.440 °C
Process water temperature out .......... =    28.216 °C
Capacity of cooler unit ................ = 12189.388 kW
Total capacity of tower ................ =   585.091 MW
```

APPENDIX L

FORTRAN CODE FOR CROSS-FLOW EVAPORATIVE COOLER SIMULATION PROGRAM

```
C     ================================================
C     =                                              =
C     =              EVALUATION OF A                 =
C     =          CLOSED CIRCUIT CROSSFLOW            =
C     =             EVAPORATIVE COOLER               =
C     =                                              =
C     =     The evaluation of a evaporative cooler   =
C     =     unit using a finite difference method.   =
C     =    The evaporative cooler units are devided  =
C     =     into small three dimensional blocks.     =
C     =  The blocks are evaluated from a point where =
C     =    all the initial values for the block are  =
C     =     known ; by continuing from this block    =
C     =    the whole cooler unit may be evaluated.   =
C     =                                              =
C     =             ********************             =
C     =                                              =
C     =     Written by  A.A.DREYER (8312818)         =
C     =                                              =
C     =    DEPARTMENT OF MECHANICAL ENGINEERING      =
C     =       UNIVERSITY OF STELLENBOSCH             =
C     =      VAX 785 - FORTRAN Version 4.5           =
C     =                                              =
C     =      LAST REVISION   :  18 August 1988       =
C     =                                              =
C     ================================================
C
C  Tsp       -  Process water temperature    (array)                [ C]
C  Tsw       -  Recirc. water temperature    (array)                [ C]
C  Tsa       -  Dry bulb air temperature     (array)                [ C]
C  sisa      -  Enthalpy of air              (array)              [kJ/kg]
C  swsa      -  Humidty of air               (array)    [kg water/kg air]
C  smsw      -  Recirc. water massflow rate  (array)               [kg/s]
C  Tspi      -  Inlet process water temperature for element         [ C]
C  Tspo      -  Outlet process water temperature for element        [ C]
C  Tswi      -  Inlet recirc. water temperature for element         [ C]
C  Tswo      -  Outlet recirc. water temperature for element        [ C]
C  Tsai      -  Inlet air temperature for element                   [ C]
C  Tsao      -  Outlet air temperature for element                  [ C]
C  sisai     -  Inlet air enthalpy for element                    [kJ/kg]
C  sisao     -  Outlet air enthalpy for element                   [kJ/kg]
C  swsai     -  Inlet air humidity for element          [kg water/kg air]
C  swsao     -  Outlet air humidity for element         [kg water/kg air]
C  smswi     -  Inlet recirc. water massflow for element           [kg/s]
C  smswo     -  Outlet recirc. water massflow for element          [kg/s]
C  Tsa       -  Air temperature                                     [ C]
C  Tsai      -  Air temperature into element                        [ C]
C  Tsao      -  Air temperature from element                        [ C]
C  Tspil     -  Inlet temperature of hot process water              [ C]
C  Tswil     -  Inlet temperature of reciculating water             [ C]
C  TR        -  Right boundary of temperature interval              [ C]
C  TL        -  Left boundary of temperature interval               [ C]
```

```
C  Tsadb      -  Dry bulb temperature of air                          [ C]
C  Tsawb      -  Wet bulb temperature of air                          [ C]
C  spsatm     -  Atmospheric pressure                                 [Pa]
C  L          -  Length of pipe (perpendicular to airstream)           [m]
C  H          -  Heigth of cooler (perpendicular to airstream)         [m]
C  sdsi       -  Inner diameter of pipe                                [m]
C  sdso       -  Outer diameter of pipe                                [m]
C  Kmax       -  Number of pipe rows along direction of airstream
C  Lmax       -  Number of pipes per row
C  Mmax       -  Number of elements along the length of a single pipe
C  clrtype    -  1-Recirc. cooling water, 2-Cooling water once through
C  gradfile   -  Logical variable (1-print temp gradients,0-print nothing)
C  rhosa      -  Density of air                                   [kg/m^3]
C  rhosw      -  Density of water                                 [kg/m^3]
C  rhosv      -  Density of water vapour                          [kg/m^3]
C  rhosav     -  Density of air/vapour mixture                    [kg/m^3]
C  musa       -  Dynamic viscosity of air                          [kg/ms]
C  musw       -  Dynamic viscosity of water                        [kg/ms]
C  musv       -  Dynamic viscosity of water vapour                 [kg/ms]
C  musav      -  Dynamic viscosity of air/vapour mixture           [kg/ms]
C  smsa       -  Massflow of air                                    [kg/s]
C  smsp       -  Massflow of process water                          [kg/s]
C  smsael     -  Massflow of air through single element             [kg/s]
C  smspel     -  Massflow of process water through element          [kg/s]
C  smswel     -  Massflow of recirculating water through element    [kg/s]
C  hspas      -  Horizontal spacing of tubes                           [m]
C  vspas      -  Vertical spacing of tubes                             [m]
C  svsa       -  Velocity of air                                     [m/s]
C  svsp       -  Velocity of process water                           [m/s]
C  shsf1      -  Fouling heat transfer coefficient (inner)        [W/m^2K]
C  shsf2      -  Fouling heat transfer coefficient (outer)        [W/m^2K]
C  skst       -  Thermal conductivity of tube wall                  [W/mK]
C  gamma      -  Recirculating water massflow/unit length        [kg/m/hr]
C  sa         -  Contact area/ unit volume                       [m^2/m^3]
C  dA         -  Air/water contact area per element                  [m^2]
C  Aspi       -  Cross area of pipe (inner diameter)                 [m^2]
C  Aspo       -  Cross area of pipe (outer diameter)                 [m^2]
C  ReyC       -  Constant used in definition of Reynoldsnumber
C  sdsperp    -  Gap between pipes perpendicular to airflow            [m]
C  sdsdiag    -  Gap on diagonal between pipes                         [m]
C  Vstot      -  Total air volumeflow through cooler               [m^3/s]
C  Vseff      -  Effective air volumeflow through a row of pipes   [m^3/s]
C  Vseff2     -  Effective air volumeflow through the cooler       [m^3/s]
C  Tsaodb     -  Dry bulb temperature of outlet air                   [ C]
C  Tsaowb     -  Wet bulb temperature of outlet air                   [ C]
C  model      -  Solution model to be used (1-Merkel,2-Impr Merkel,3-Poppe)
C  spssat     -  Saturation pressure of water                         [Pa]
C  scspv      -  Specific heat of water vapour                     [J/kgK]
C  scspa      -  Specific heat of dry air                          [J/kgK]
C  scspw      -  Specific heat of saturated water                  [J/kgK]
C  sissat     -  Enthalpy of saturated air                         [kJ/kg]
C  sisvap     -  Enthalpy of water vapour                          [kJ/kg]
```

```
C  sksw        -  Thermal conductivity of saturated water              [W/mK]
C  Reysp       -  Reynolds number for process water flow
C  Reysw       -  Reynolds number for recirculating water flow
C  Reysa       -  Reynolds number for air flow
C  Pra         -  Prandtl number
C  Lew         -  Lewis factor
C  shsp        -  Heat transfer coefficient (process water - pipe)  [W/m^2K]
C  shsw        -  Heat transfer coefficient (recirc. water - pipe   [W/m^2K]
C  koga        -  Overall capacity coefficient of mass transfer    [kg/m^3s]
C  kog         -  Mass transfer coefficient                        [kg/m^2s]
C  Uo          -  Overall heat transfer coefficient                 [W/m^2K]
C  K           -  Constant
C  a           -  Runge-Kutta constant                                  [kg]
C  b           -  Runge-Kutta constant                               [kg/kg]
C  c           -  Runge-Kutta constant                               [kJ/kg]
C  d           -  Runge-Kutta constant                                  [ C]
C  e           -  Runge-Kutta constant                                  [ C]
C  ww          -  Temporary variable for Runge-Kutta approximation   [kg/kg]
C  ii          -  Temporary variable for Runge-Kutta approximation   [kJ/kg]
C  TT          -  Temporary variable for Runge-Kutta approximation      [ C]
C  www         -  Temporary variable for Runge-Kutta approximation   [kg/kg]
C  swsasw      -  Saturation air humidity at Tsw                     [kg/kg]
C  swsasa      -  Saturation air humidity at Tsa                     [kg/kg]
C  sisasw      -  Saturation enthalpy of air at Tsw                  [kJ/kg]
C  sisasa      -  Saturation enthalpy of air at Tsa                  [kJ/kg]
C  Power       -  Capacity of cooler unit                               [kW]
C  flowlayout-  Process water flow layout through unit (1,2,3 or 4)

C Reserve storage space for five arrays i.e. Tsp,Tsw,sisa,swsa and smsw
      DIMENSION Tsp(40,400,10)
      DIMENSION Tsw(40,400,10)
      DIMENSION Tsa(40,400,10)
      DIMENSION sisa(40,400,10)
      DIMENSION swsa(40,400,10)
      DIMENSION smsw(40,400,10)

C Declare the changed data types
      INTEGER gradfile,clrtype,flowlayout,gradplot
      REAL L
      CHARACTER*10 char

C Set the initial array values equal to zero
      DO 30 i=1,Kmax+1
        DO 20 j=1,Lmax+1
          DO 10 k=1,Mmax+2
            Tsp(i,j,k)=0.0
            Tsw(i,j,k)=0.0
            Tsa(i,j,k)=0.0
            sisa(i,j,k)=0.0
            swsa(i,j,k)=0.0
            smsw(i,j,k)=0.0
   10     CONTINUE
```

```
 20   CONTINUE
 30 CONTINUE

C Call subroutine to set default values for a typical cooler
      CALL INITIAL(spsatm,Tsadb,Tsawb,L,H,sdso,sdsi,Kmax,
     +     Lmax,Mmax,vspas,hspas,smsp,PI,gamma,skst,Tspil,
     +     clrtype,model,smsa,flowlayout,Tswil,shsfl,shsf2)

C Call subroutine to edit coolertype,flowlayout,model,size etc.
    5 CALL MENU1(clrtype,model,H,L,spsatm,Tsadb,Tsawb,flowlayout,
     +     Tswil,shsfl,shsf2)

C Call subroutine to edit the cooler dimensions and operating parameters
      CALL MENU2(sdso,sdsi,H,L,PI,svsa,vspas,hspas,Lmax,Kmax,Mmax,
     +     spsatm,Tsadb,Tsawb,gamma,skst,Tspil,smsp,svsp,sa,Aspi,
     +     Aspo,clrtype,dA,model,smsa,flowlayout,Tswil,shsfl,shsf2)

C Open result files for program results and cooler temperature gradients
      gradfile=0          ! 0 - print nothing , 1 - print gradients
      gradplot=0          ! 0 - print nothing , 1 - print gradients
      OPEN (UNIT=1, FILE='CROSS.RES', STATUS='NEW')
      IF (gradfile.EQ.1) THEN
        OPEN (UNIT=4, FILE='CROSS.GRA', STATUS='NEW')
      ELSE IF (gradplot.EQ.1) THEN
        OPEN (UNIT=5, FILE='CROSS.PLO', STATUS='NEW')
      END IF

C Determine the air flow parameters for cooler
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsail)
      CALL AirVapMixdensity(Tsadb,swsail,spsatm,rhosail)
      svsa=(smsa*(1.0+swsail))/(rhosail*L*(Lmax+0.5)*vspas)
      ReyC=svsa*(vspas/sdso)*(vspas-sdso)

C Determine the massflow of each fluid for a typical element
      CALL Waterdensity(Tspil,rhosw)
      smsael=smsa/(Mmax*(Lmax+0.5))       ! Air massflow / element
      smspel=svsp*rhosw*Aspi              ! Process water massflow/element
      smswel=2.0*gamma*L/Mmax             ! Recirc. water massflow/element
      smswil=2.0*gamma*L*Kmax             ! Total inlet recirc. water massflow

C Evaluate cooler with given flowlayout
C clrtype = 1  ==> Recirculating cooling water
C clrtype = 2  ==> Cooling water makes only single pass through cooler
      CALL LIB$ERASE_PAGE(1,1)
      WRITE(*,*)' ITERATIVE CALCULATION IN PROGRESS'
      WRITE(*,*)'*********************************'
      IF (clrtype.EQ.1) THEN
        TR=Tspil                          ! Set upper value for Tw(in)
        TL=Tsawb                          ! Set lower value for Tw(in)
 40     Tswil=(TR+TL)/2.0                 ! Halve the Tw(in) interval
        IF (flowlayout.EQ.1) THEN
          CALL FRONTTOBACK (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
```

```
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
        ELSE IF (flowlayout.EQ.2) THEN
          CALL BACKTOFRONT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
        ELSE IF (flowlayout.EQ.3) THEN
          CALL TOPTOBOTTOM (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2,
     +         gradplot)
        ELSE IF (flowlayout.EQ.4) THEN
          CALL STRAIGHT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
        END IF
        IF (ABS(Tswil-Tswol).GT.0.001) THEN
          IF (Tswil.LT.Tswol) THEN
            TL=Tswol                    ! TL=Tswil
            IF (TL.GT.Tspil) TL=Tspil
          ELSE IF (Tswil.GT.Tswol) THEN
            TR=Tswol                   ! TR=Tswil
          END IF
          GO TO 40
        END IF
      ELSE IF (clrtype.EQ.2) THEN
        IF (flowlayout.EQ.1) THEN
          CALL FRONTTOBACK (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
        ELSE IF (flowlayout.EQ.2) THEN
          CALL BACKTOFRONT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
        ELSE IF (flowlayout.EQ.3) THEN
          CALL TOPTOBOTTOM (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +         smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +         swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +         spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +         gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2,
```

```
     +          gradplot)
           ELSE IF (flowlayout.EQ.4) THEN
             CALL STRAIGHT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +          smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +          swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +          spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +          gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)
           END IF
         END IF

C Print final temperature,enthalpy etc. profiles
         IF (flowlayout.EQ.3) THEN    !  TTB flow pattern
           DO j=1,Lmax+1
             IF (j.EQ.1) THEN
               tpi=Tspil
               tpo=Tspil
             ELSE
               iflag=j-2.0*INT(j/2.0)
               IF (iflag.EQ.0) THEN
                 tpi=Tsp(1,j-1,3)
                 tpo=Tsp(Kmax,j-1,3)
               ELSE
                 tpi=Tsp(1,j-1,1)
                 tpo=Tsp(Kmax,j-1,1)
               END IF
             END IF
             twi=Tsw(1,j,2)
             two=Tsw(Kmax,j,2)
             siao=sisa(Kmax+1,j,2)
             swao=swsa(Kmax+1,j,2)
             WRITE(10,*)j,tpi,twi,tpo,two,siao,swao
           ENDDO
           WRITE(10,*)' '
           DO i=1,Kmax
             WRITE(10,*)i,Tsp(i,Lmax,3),Tsw(i,Lmax+1,2)
           ENDDO
         ELSE IF ((flowlayout.EQ.1).OR.(flowlayout.EQ.2)) THEN     ! FTB&BTF
           DO j=1,Lmax+1
             tpi=Tsp(1,j,2)
             tpo=Tsp(Kmax,j,1)
             twi=Tsw(1,j,2)
             two=Tsw(Kmax,j,2)
             siao=sisa(Kmax+1,j,2)
             swao=swsa(Kmax+1,j,2)
             WRITE(11,*)j,tpi,twi,tpo,two,siao,swao
           ENDDO
           WRITE(11,*)' '
           DO i=1,Kmax
             jflag=i-2.0*INT(i/2.0)
             IF (jflag.EQ.1) THEN
               tp=Tsp(i,Lmax,3)
             ELSE
```

```
          tp=Tsp(i,Lmax,1)
        END IF
        WRITE(11,*)i,tp,Tsw(i,Lmax+1,2)
      ENDDO
    END IF

C Print solution model used, ambient conditions and results
    IF (model.EQ.1) THEN
      TR=Tspil
      TL=Tsawb
 50   Tsaol=(TR+TL)/2.0
      CALL Satenthalpy(Tsaol,spsatm,sisasa)
      IF (ABS(sisasa-sisaol).GT.0.1) THEN
        IF (sisasa.GT.sisaol) THEN
          TR=Tsaol
        ELSE
          TL=Tsaol
        END IF
        GO TO 50
      END IF
      CALL Airhumidity(Tsaol,Tsaol,spsatm,swsaol)
      CALL AirVapMixdensity(Tsaol,swsaol,spsatm,rhosaol)
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsal)
      CALL AirVapMixdensity(Tsadb,swsal,spsatm,rhosail)
      smswol=smswil-(swsaol-swsal)*smsa
    ELSE IF (model.EQ.2) THEN
      CALL Satvappressure(Tsaol,spssat)
      spsvap=spsatm*swsaol/(1.005*(0.62198+swsaol))
      phio=spsvap/spssat
      IF (phio.GT.1.0) THEN
        phio=1.0
        CALL Airhumidity(Tsaol,Tsaol,spsatm,swsao2)
        CALL AirVapMixdensity(Tsaol,swsao2,spsatm,rhosaol)
      ELSE
        CALL AirVapMixdensity(Tsaol,swsaol,spsatm,rhosaol)
      END IF
    ELSE IF (model.EQ.3) THEN
      CALL Satvappressure(Tsaol,spssasa)
      spsvap=spsatm*swsaol/(1.005*(0.62198+swsaol))
      phio=spsvap/spssasa
      IF (phio.GT.1.0) THEN
        phio=1.0
        CALL Airhumidity(Tsaol,Tsaol,spsatm,swsao2)
        CALL AirVapMixdensity(Tsaol,swsao2,spsatm,rhosaol)
      ELSE
        CALL AirVapMixdensity(Tsaol,swsaol,spsatm,rhosaol)
      END IF
    END IF
    CALL Airhumidity(Tsadb,Tsawb,spsatm,swsal)
    CALL AirVapMixdensity(Tsadb,swsal,spsatm,rhosail)
    CALL Cpw(Tspil,scsppi)
    CALL Cpw(Tspol,scsppo)
```

```
      Power=smsp*((Tspil)*scsppi-(Tspol)*scsppo)/1000.0
      CALL PRINT_RESULTS(Tspil,Tspol,smsp,sdsi,sdso,vspas,hspas,
     +     Kmax,Lmax,Mmax,gamma,Vstot,rhosail,Vseff2,sisail,sisaol,
     +     Tswil,Tswol,svsp,flowlayout,H,L,spsatm,PI,Tsawb,Tsadb,svsa,
     +     swsail,swsaol,smswil,smswol,model,Tsaol,rhosaol,phio,Power,
     +     shsfl,shsf2,skst,smsa)

C Rerun program or return to DCL
      WRITE(*,100)
  100 FORMAT(' RERUN program or return to DCL (R/D) ?',$)
      READ(*,'(A)')char
      CLOSE (UNIT=1)
      CLOSE (UNIT=4)
      CLOSE (UNIT=5)
      IF ((char.EQ.'R').OR.(char.EQ.'r')) GO TO 5

C End of main program
      END


C *******************************************************************
C *                                                                 *
C *         COUNTER FLOW (FROM BACK TO FRONT OF COOLER)             *
C *                                                                 *
C *******************************************************************
C Subroutine to evaluate a cooler layout where the process fluid flows
C in a direction counter to the direction of the airstream
      SUBROUTINE BACKTOFRONT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +            smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +            swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +            spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +            gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsfl,shsf2)

      DIMENSION Tsp(40,400,10)
      DIMENSION Tsw(40,400,10)
      DIMENSION Tsa(40,400,40)
      DIMENSION sisa(40,400,10)
      DIMENSION swsa(40,400,10)
      DIMENSION smsw(40,400,10)

      REAL L
      INTEGER flag,flag2,gradfile

C Choose an average temperature for the outlet process water
      Tspol=(Tspil+Tswil)/2.0
      DO 50 j=1,Lmax
        Tsp(1,j,2)=Tspol
   50 CONTINUE

C Initialize the arrays with the known temperature and enthalpy values
  999 CALL Enthalpy(Tsadb,Tsawb,spsatm,sisail)
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsail)
```

```
      DO 20 j=1,Lmax
        DO 10 k=2,Mmax+1
          sisa(1,j,k)=sisail
          swsa(1,j,k)=swsail
          Tsa(1,j,k)=Tsadb
   10   CONTINUE
   20 CONTINUE

      DO 40 i=1,Kmax
        DO 30 k=2,Mmax+1
          Tsw(i,1,k)=Tswil
          smsw(i,1,k)=smswel
   30   CONTINUE
   40 CONTINUE

C N.B. flag=1 for backward process fluid flow
C L.W. flag=0 for forward process fluid flow
      flag=0

C Start of the outer loop to evaluate each i-level of the model
      DO 60 i=1,Kmax
        flag2=i-2*INT(i/2.0)
C Flag2=1 in the first row,0 in the second row etc.

C Start of the middle loop to evaluate each j-level of the model
        DO 70 j=1,Lmax

C Start of the inner loop to evaluate each each element of the model
          IF (flag.EQ.0) THEN
C Process water flow is in a forward direction
            DO 80 k=2,Mmax+1

C Determine the input values for each element
              Tspo=Tsp(i,j,k)
              IF((k.EQ.2).AND.(i.NE.1)) Tspo=Tsp(i-1,j,k-1)
              Tswi=Tsw(i,j,k)
              Tsai=Tsa(i,j,k)
              sisai=sisa(i,j,k)
              swsai=swsa(i,j,k)
              smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
              IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
                IF (j.EQ.1) THEN
                  sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                  swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                  Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
                ELSE
                  sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
                  swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
                  Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
                END IF
```

```
            END IF
            IF (flag2.EQ.0) THEN
              IF (j.EQ.Lmax) THEN
                sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
              END IF
            END IF

C Call subroutine to determine outlet conditions of each element
            IF (model.EQ.1) THEN
              CALL MERKEL2 (Tspi,Tswi,sisai,swsai1,L,H,sdsi,sdso,dA,
     +                 Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +                 sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +                 ReyC,shsf1,shsf2,Kmax)
            ELSE IF (model.EQ.2) THEN
              CALL IMPMERKEL2 (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +                 Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +                 sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +                 ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Kmax)
            ELSE
              CALL POPPE2 (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +                 sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +                 smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +                 smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Kmax)
            END IF

C Determine the exit values for each element
            Tsp(i,j,k+1)=Tspi
            Tsw(i,j+1,k)=Tswo
            Tsa(i+1,j,k)=Tsao
            sisa(i+1,j,k)=sisao
            swsa(i+1,j,k)=swsao
            smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
            IF (gradfile.EQ.1) THEN
              WRITE(4,*)i,j,k-1
              WRITE(4,*)Tspo,Tspi
              WRITE(4,*)Tswi,Tswo
              WRITE(4,*)sisai,sisao
              IF (model.NE.1) THEN
                WRITE(4,*)swsai,swsao
                WRITE(4,*)smswi,smswo
                IF (model.EQ.3) THEN
                  WRITE(4,*)Tsai,Tsao
                END IF
              END IF
```

```
          END IF
 80     CONTINUE

        ELSE IF (Flag.EQ.1) THEN
C Start of the inner loop to evaluate each each element of the model
C Process water flow is backwards to the origin
          DO 90 k=Mmax+1,2,-1

C Determine the input values for each element
            Tspo=Tsp(i,j,k)
            IF (k.EQ.(Mmax+1)) Tspo=Tsp(i-1,j,k+1)
            Tswi=Tsw(i,j,k)
            Tsai=Tsa(i,j,k)
            sisai=sisa(i,j,k)
            swsai=swsa(i,j,k)
            smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
            IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
              IF (j.EQ.1) THEN
                sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
              END IF
            END IF
            IF (flag2.EQ.0) THEN
              IF (j.EQ.Lmax) THEN
                sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
              END IF
            END IF

C Call subroutine to determine outlet conditions of each element
            IF (model.EQ.1) THEN
              CALL MERKEL2 (Tspi,Tswi,sisai,swsai1,L,H,sdsi,sdso,dA,
     +             Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +             sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +             ReyC,shsf1,shsf2,Kmax)
            ELSE IF (model.EQ.2) THEN
              CALL IMPMERKEL2 (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +             Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +             sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +             ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Kmax)
```

```
              ELSE
                CALL POPPE2 (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +               sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +               smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +               smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Kmax)
              END IF

C Determine the exit values for each element
              Tsp(i,j,k-1)=Tspi
              Tsw(i,j+1,k)=Tswo
              Tsa(i+1,j,k)=Tsao
              sisa(i+1,j,k)=sisao
              swsa(i+1,j,k)=swsao
              smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
              IF (gradfile.EQ.1) THEN
                WRITE(4,*)i,j,k-1
                WRITE(4,*)Tspo,Tspi
                WRITE(4,*)Tswi,Tswo
                WRITE(4,*)sisai,sisao
                IF (model.NE.1) THEN
                  WRITE(4,*)swsai,swsao
                  WRITE(4,*)smswi,smswo
                  IF (model.EQ.3) THEN
                    WRITE(4,*)Tsai,Tsao
                  END IF
                END IF
              END IF
   90       CONTINUE
          END IF
   70   CONTINUE
        IF (flag.EQ.0) THEN
          flag=1
        ELSE
          flag=0
        END IF
   60 CONTINUE

C Determine the average inlet temperature of process water
      sum1=0.0
      sum2=0.0
      rem=Mmax+2
      IF (flag.NE.1) rem=1
      DO 120 j=1,Lmax
        CALL Cpw(Tsp(Kmax,j,rem),scspp)
        sum1=sum1+Tsp(Kmax,j,rem)*scspp
        sum2=sum2+Tsp(Kmax,j,rem)
  120 CONTINUE
      CALL Cpw((sum2/Lmax),scspp)
      Tspi2=sum1/(Lmax*scspp)
```

```
C Determine if the inlet conditions satisfies the known outlet condition
C If not choose an new inlet condition and repeat from 999
      rem=Mmax+2
      IF (flag.NE.1) rem=1
      sum4=0.0
      DO 121 j=1,Lmax
        sum4=sum4+ABS(Tspi1-Tsp(Kmax,j,rem))
  121 CONTINUE
      WRITE(*,125)Tspi2,sum4/Lmax
  125 FORMAT(' Tp(in) calculated =',F6.2,
     +       '          Average Deviation =',F10.6)
      IF ((sum4/Lmax).GT.0.1) THEN
        DO 122 j=1,Lmax
          dT=(Tspi1-Tsp(Kmax,j,rem))/2.0
          Tsp(1,j,2)=Tsp(1,j,2)+dT
  122   CONTINUE
        GOTO 999
      END IF

C Determine the average exit temperature of the process water
      sum1=0.0
      sum2=0.0
      DO 51 j=1,Lmax
        CALL Cpw(Tsp(1,j,2),scspp)
        sum1=sum1+Tsp(1,j,2)*scspp
        sum2=sum2+Tsp(1,j,2)
   51 CONTINUE
      CALL Cpw((sum2/Lmax),scspp)
      Tspo1=sum1/(Lmax*scspp)

C Determine the average exit temperature of recirculating water
      sum1=0.0
      sum2=0.0
      DO 110 i=1,Kmax
        DO 100 k=2,Mmax+1
          CALL Cpw(Tsw(i,Lmax+1,k),scspw)
          sum1=sum1+Tsw(i,Lmax+1,k)*scspw
          sum2=sum2+smsw(i,Lmax+1,k)
  100   CONTINUE
  110 CONTINUE
      CALL Cpw(Tswi1,scspw)
      Tswo1=sum1/(Mmax*Kmax*scspw)
      smswo1=sum2

C Determine the average exit enthalpy of the air
      sum1=0.0
      sum2=0.0
      sum3=0.0
      DO 140 j=1,Lmax
        DO 130 k=2,Mmax+1
          sum1=sum1+sisa(Kmax+1,j,k)
          sum2=sum2+swsa(Kmax+1,j,k)
```

```
          sum3=sum3+Tsa(Kmax+1,j,k)
  130   CONTINUE
  140 CONTINUE
      DO 150 k=2,Mmax+1
        IF (flag2.EQ.0) THEN
          sum1=sum1+sisa(Kmax,1,k)/2.0
          sum2=sum2+swsa(Kmax,1,k)/2.0
          sum3=sum3+Tsa(Kmax,1,k)/2.0
        ELSE
          sum1=sum1+sisa(Kmax,Lmax,k)/2.0
          sum2=sum2+swsa(Kmax,Lmax,k)/2.0
          sum3=sum3+Tsa(Kmax,Lmax,k)/2.0
        END IF
  150 CONTINUE
      sisaol=sum1/(Mmax*(Lmax+.5))
      swsaol=sum2/(Mmax*(Lmax+.5))
      Tsaol=sum3/(Mmax*(Lmax+.5))

C Print the recirc.water inlet and outlet temperatures on the screen
      WRITE(*,160)Tswil,Tswol
  160 FORMAT(' ',/' Tw(in) = ',F7.3,'      Tw(out) = ',F7.3/)
      RETURN
      END

C *******************************************************************
C *                                                                 *
C *          PARALLEL FLOW (FROM FRONT TO BACK OF COOLER)           *
C *                                                                 *
C *******************************************************************
C Subroutine to evaluate a cooler layout where the process fluid flows
C in a direction parallel to the direction of the airstream
      SUBROUTINE FRONTTOBACK (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,
     +           smspel,smswel,sisail,sisaol,Tspil,Tspol,Tswil,
     +           Tswol,swsail,swsaol,smswil,smswol,L,H,sdsi,sdso,
     +           dA,Tsadb,Tsawb,spsatm,gamma,Vstot,sa,skst,
     +           svsp,Aspi,Aspo,ReyC,gradfile,Kmax,Lmax,Mmax,PI,
     +           model,Tsaol,shsf1,shsf2)

      DIMENSION Tsp(40,400,10)
      DIMENSION Tsw(40,400,10)
      DIMENSION Tsa(40,400,10)
      DIMENSION sisa(40,400,10)
      DIMENSION swsa(40,400,10)
      DIMENSION smsw(40,400,10)

      REAL L
      INTEGER flag,flag2,gradfile

C Initialize the three arrays with the known temperature and enthalpy values
      CALL Enthalpy(Tsadb,Tsawb,spsatm,sisail)
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsail)
      DO 20 j=1,Lmax
```

```
        DO 10 k=2,Mmax+1
          sisa(1,j,k)=sisai1
          swsa(1,j,k)=swsai1
          Tsa(1,j,k)=Tsadb
 10     CONTINUE
 20   CONTINUE

      DO 40 i=1,Kmax
        DO 30 k=2,Mmax+1
          Tsw(i,1,k)=Tswi1
          smsw(i,1,k)=smswe1
 30     CONTINUE
 40   CONTINUE

      DO 50 j=1,Lmax
        Tsp(1,j,2)=Tspi1
 50   CONTINUE

C N.B. flag=1 for backward process fluid flow
C L.W. flag=0 for forward process fluid flow
      flag=0

C Start of the outer loop to evaluate each i-level of the model
      DO 60 i=1,Kmax
        flag2=i-2*INT(i/2.0)
C Flag2=1 in the first row,0 in the second row etc.

C Start of the middle loop to evaluate each j-level of the model
        DO 70 j=1,Lmax

C Start of the inner loop to evaluate each each element of the model
          IF (flag.EQ.0) THEN
C Process water flow is in a forward direction
            DO 80 k=2,Mmax+1

C Determine the input values for each element
              Tspi=Tsp(i,j,k)
              IF((k.EQ.2).AND.(i.NE.1)) Tspi=Tsp(i-1,j,k-1)
              Tswi=Tsw(i,j,k)
              Tsai=Tsa(i,j,k)
              sisai=sisa(i,j,k)
              swsai=swsa(i,j,k)
              smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
              IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
                IF (j.EQ.1) THEN
                  sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                  swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                  Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
                ELSE
                  sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
```

```
              swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
              Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
            END IF
          END IF
          IF (flag2.EQ.0) THEN
            IF (j.EQ.Lmax) THEN
              sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
              swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
              Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
            ELSE
              sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
              swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
              Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
            END IF
          END IF

C Call subroutine to determine outlet conditions of each element
          IF (model.EQ.1) THEN
            CALL MERKEL (Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +           ReyC,shsf1,shsf2,Kmax)
          ELSE IF (model.EQ.2) THEN
            CALL IMPMERKEL (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +           ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Kmax)
          ELSE
            CALL POPPE (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +           sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +           smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +           smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Kmax)
          END IF

C Determine the exit values for each element
          Tsp(i,j,k+1)=Tspo
          Tsw(i,j+1,k)=Tswo
          Tsa(i+1,j,k)=Tsao
          sisa(i+1,j,k)=sisao
          swsa(i+1,j,k)=swsao
          smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
          IF (gradfile.EQ.1) THEN
            WRITE(4,*)i,j,k-1
            WRITE(4,*)Tspi,Tspo
            WRITE(4,*)Tswi,Tswo
            WRITE(4,*)sisai,sisao
            IF (model.NE.1) THEN
              WRITE(4,*)swsai,swsao
              WRITE(4,*)smswi,smswo
              IF (model.EQ.3) THEN
```

```
                 WRITE(4,*)Tsai,Tsao
               END IF
             END IF
           END IF
   80    CONTINUE

        ELSE IF (Flag.EQ.1) THEN
C Start of the inner loop to evaluate each each element of the model
C Process water flow is backwards to the origin
          DO 90 k=Mmax+1,2,-1

C Determine the input values for each element
            Tspi=Tsp(i,j,k)
            IF (k.EQ.(Mmax+1)) Tspi=Tsp(i-1,j,k+1)
            Tswi=Tsw(i,j,k)
            Tsai=Tsa(i,j,k)
            sisai=sisa(i,j,k)
            swsai=swsa(i,j,k)
            smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
            IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
              IF (j.EQ.1) THEN
                sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
              END IF
            END IF
            IF (flag2.EQ.0) THEN
              IF (j.EQ.Lmax) THEN
                sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
              END IF
            END IF

C Call subroutine to determine outlet conditions of each element
            IF (model.EQ.1) THEN
              CALL MERKEL (Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +             Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +             sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +             ReyC,shsfl,shsf2,Kmax)
            ELSE IF (model.EQ.2) THEN
              CALL IMPMERKEL (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
```

```
     +              Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +              sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +              ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Kmax)
              ELSE
                CALL POPPE (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +              sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +              smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +              smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Kmax)
              END IF

C Determine the exit values for each element
              Tsp(i,j,k-1)=Tspo
              Tsw(i,j+1,k)=Tswo
              Tsa(i+1,j,k)=Tsao
              sisa(i+1,j,k)=sisao
              swsa(i+1,j,k)=swsao
              smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
              IF (gradfile.EQ.1) THEN
                WRITE(4,*)i,j,k-1
                WRITE(4,*)Tspi,Tspo
                WRITE(4,*)Tswi,Tswo
                WRITE(4,*)sisai,sisao
                IF (model.NE.1) THEN
                  WRITE(4,*)swsai,swsao
                  WRITE(4,*)smswi,smswo
                  IF (model.EQ.3) THEN
                    WRITE(4,*)Tsai,Tsao
                  END IF
                END IF
              END IF
   90       CONTINUE
          END IF
   70   CONTINUE
        IF (flag.EQ.0) THEN
          flag=1
        ELSE
          flag=0
        END IF
   60 CONTINUE

C Determine the average exit temperature of recirculating water
      sum1=0.0
      sum2=0.0
      DO 110 i=1,Kmax
        DO 100 k=2,Mmax+1
          CALL Cpw(Tsw(i,Lmax+1,k),scspw)
          sum1=sum1+Tsw(i,Lmax+1,k)*scspw
          sum2=sum2+smsw(i,Lmax+1,k)
  100   CONTINUE
  110 CONTINUE
```

```
      CALL Cpw(Tswil,scspw)
      Tswol=suml/(Mmax*Kmax*scspw)
      smswol=sum2

C Determine the average exit temperature of process water
      suml=0.0
      sum2=0.0
      rem=Mmax+2
      IF (flag.NE.1) rem=1
      DO 120 j=1,Lmax
        CALL Cpw(Tsp(Kmax,j,rem),scspp)
        suml=suml+Tsp(Kmax,j,rem)*scspp
        sum2=sum2+Tsp(Kmax,j,rem)
  120 CONTINUE
      CALL Cpw((sum2/Lmax),scspp)
      Tspol=suml/(Lmax*scspp)

C Determine the average exit enthalpy of the air
      suml=0.0
      sum2=0.0
      sum3=0.0
      DO 140 j=1,Lmax
        DO 130 k=2,Mmax+1
          suml=suml+sisa(Kmax+1,j,k)
          sum2=sum2+swsa(Kmax+1,j,k)
          sum3=sum3+Tsa(Kmax+1,j,k)
  130   CONTINUE
  140 CONTINUE
      DO 150 k=2,Mmax+1
        IF (flag2.EQ.0) THEN
          suml=suml+sisa(Kmax,1,k)/2.0
          sum2=sum2+swsa(Kmax,1,k)/2.0
          sum3=sum3+Tsa(Kmax,1,k)/2.0
        ELSE
          suml=suml+sisa(Kmax,Lmax,k)/2.0
          sum2=sum2+swsa(Kmax,Lmax,k)/2.0
          sum3=sum3+Tsa(Kmax,Lmax,k)/2.0
        END IF
  150 CONTINUE
      sisaol=suml/(Mmax*(Lmax+.5))
      swsaol=sum2/(Mmax*(Lmax+.5))
      Tsaol=sum3/(Mmax*(Lmax+.5))

C Print the recirc.water inlet and outlet temperatures on the screen
      WRITE(*,160)Tswil,Tswol
  160 FORMAT(' ','Tw(in) = ',F7.3,'      Tw(out) = ',F7.3)
      RETURN
      END
```

```
C ********************************************************************
C *                                                                  *
C *     IMPROVED MERKEL METHOD TO EVAUALTE A SINGLE ELEMENT          *
C *                                                                  *
C ********************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the three
C Merkel equations and one additional equation which controls the
C state of a single element
      SUBROUTINE IMPMERKEL(Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +           ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,nrow)

      REAL L,musav,musw,kog,koga,K1,K2,K3,K4

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspi,musw)
      CALL Waterdensity(Tspi,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)        ! Reynoldsnumber of process water
      CALL AirVapMixdensity(Tsai,swsai,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsai,swsai,spsatm,musav)
      Reysa=ReyC*rhosav/musav              ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      gammal=gamma*(smswi/smswel)
      Reysw=4.0*gammal/musw                ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspi,sksp)
      CALL Prandtl(Tspi,Prasp)
      shsw=4.186*118.0*((gammal*3600.0/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
        sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
        term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
        term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
        shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
      END IF
      koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
      kog=koga/sa   !  Mass-transfer coefficient
      Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsf1))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3 and K4
      CALL Cpw(Tspi,scspp)
      CALL Cpw(Tswi,scspw)
      K1=kog*dA/(smsael)
      K2=Uo*dA/(smswi*scspw)
      K3=kog*dA*1000.0/(smswi*scspw)
      K4=Uo*dA/(smspel*scspp)
```

```
C Determine the Runge-Kutta coefficients
      CALL Satenthalpy(Tswi,spsatm,sisaswl)
      al=Kl*(sisaswl-sisai)
      bl=K2*(Tspi-Tswi)-K3*(sisaswl-sisai)
      cl=-K4*(Tspi-Tswi)

      CALL Satenthalpy((Tswi+bl/2.0),spsatm,sisasw2)
      a2=Kl*(sisasw2-(sisai+al/2.0))
      b2=K2*((Tspi+cl/2.0)-(Tswi+bl/2.0))-K3*(sisasw2-(sisai+al/2.0))
      c2=-K4*((Tspi+cl/2.0)-(Tswi+bl/2.0))

      CALL Satenthalpy((Tswi+b2/2.0),spsatm,sisasw3)
      a3=Kl*(sisasw3-(sisai+a2/2.0))
      b3=K2*((Tspi+c2/2.0)-(Tswi+b2/2.0))-K3*(sisasw3-(sisai+a2/2.0))
      c3=-K4*((Tspi+c2/2.0)-(Tswi+b2/2.0))

      CALL Satenthalpy((Tswi+b3),spsatm,sisasw4)
      a4=Kl*(sisasw4-(sisai+a3))
      b4=K2*((Tspi+c3)-(Tswi+b3))-K3*(sisasw4-(sisai+a3))
      c4=-K4*((Tspi+c3)-(Tswi+b3))

      CALL Airhumidity(Tswi,Tswi,spsatm,swsaswl)
      CALL Airhumidity((Tswi+bl/2.0),(Tswi+bl/2.0),spsatm,swsasw2)
      CALL Airhumidity((Tswi+b2/2.0),(Tswi+b2/2.0),spsatm,swsasw3)
      CALL Airhumidity((Tswi+b3),(Tswi+b3),spsatm,swsasw4)
      dl=Kl*(swsaswl-swsai)/(1.0-swsaswl)
      d2=Kl*(swsasw2-(swsai+dl/2.0))/(1.0-swsasw2)
      d3=Kl*(swsasw3-(swsai+d2/2.0))/(1.0-swsasw3)
      d4=Kl*(swsasw4-(swsai+d3))/(1.0-swsasw4)

C Determine the exit conditions of the element
      sisao=sisai+(al+2.0*(a2+a3)+a4)/6.0
      Tswo=Tswi+(bl+2.0*(b2+b3)+b4)/6.0
      Tspo=Tspi+(cl+2.0*(c2+c3)+c4)/6.0
      swsao=swsai+(dl+2.0*(d2+d3)+d4)/6.0
      smswo=smswi-smsael*(swsao-swsai)

C Determine the air outlet temperature and saturation enthalpy
      TR=Tspi
      TL=0.0
   10 Tsao=(TR+TL)/2.0
      CALL Cpv(Tsao,scspv)
      CALL Cpa(Tsao,scspa)
      CALL Cpw(Tsao,scspw)
      CALL Airhumidity(Tsao,Tsao,spsatm,swsasa)
      IF (swsasa.GT.swsao) THEN
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
      ELSE
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
     +            +scspw*(swsao-swsasa)*Tsao/1000.0
      END IF
      IF ((ABS(sisao-sisasa)).GT.0.1) THEN
```

```
        IF (sisao.LT.sisasa) THEN
          TR=Tsao
        ELSE
          TL=Tsao
        END IF
        GO TO 10
      END IF
      RETURN
      END


C ******************************************************************
C *                                                                *
C *      IMPROVED MERKEL METHOD(2) TO EVAUALTE A SINGLE ELEMENT    *
C *                                                                *
C ******************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the three
C Merkel equations and one additional equation which controls the
C state of a single element; BACKTOFRONT FLOW CASE
      SUBROUTINE IMPMERKEL2(Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +           ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,nrow)

      REAL L,musav,musw,kog,koga,K1,K2,K3,K4

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspo,musw)
      CALL Waterdensity(Tspo,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)        ! Reynoldsnumber of process water
      CALL AirVapMixdensity(Tsai,swsai,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsai,swsai,spsatm,musav)
      Reysa=ReyC*rhosav/musav             ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      gammal=gamma*(smswi/smswel)
      Reysw=4.0*gammal/musw               ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspo,sksp)
      CALL Prandtl(Tspo,Prasp)
      shsw=4.186*118.0*((gammal*3600.0/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
        sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
        term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
        term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
        shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
      END IF

      koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
```

```
      kog=koga/sa    !  Mass-transfer coefficient
      Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsfl))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3 and K4
      CALL Cpw(Tspo,scspp)
      CALL Cpw(Tswi,scspw)
      K1=kog*dA/(smsael)
      K2=Uo*dA/(smswi*scspw)
      K3=kog*dA*1000.0/(smswi*scspw)
      K4=Uo*dA/(smspel*scspp)

C Determine the Runge-Kutta coefficients
      CALL Satenthalpy(Tswi,spsatm,sisasw1)
      a1=K1*(sisasw1-sisai)
      b1=K2*(Tspo-Tswi)-K3*(sisasw1-sisai)
      c1=K4*(Tspo-Tswi)

      CALL Satenthalpy((Tswi+b1/2.0),spsatm,sisasw2)
      a2=K1*(sisasw2-(sisai+a1/2.0))
      b2=K2*((Tspo+c1/2.0)-(Tswi+b1/2.0))-K3*(sisasw2-(sisai+a1/2.0))
      c2=K4*((Tspo+c1/2.0)-(Tswi+b1/2.0))

      CALL Satenthalpy((Tswi+b2/2.0),spsatm,sisasw3)
      a3=K1*(sisasw3-(sisai+a2/2.0))
      b3=K2*((Tspo+c2/2.0)-(Tswi+b2/2.0))-K3*(sisasw3-(sisai+a2/2.0))
      c3=K4*((Tspo+c2/2.0)-(Tswi+b2/2.0))

      CALL Satenthalpy((Tswi+b3),spsatm,sisasw4)
      a4=K1*(sisasw4-(sisai+a3))
      b4=K2*((Tspo+c3)-(Tswi+b3))-K3*(sisasw4-(sisai+a3))
      c4=K4*((Tspo+c3)-(Tswi+b3))

      CALL Airhumidity(Tswi,Tswi,spsatm,swsasw1)
      CALL Airhumidity((Tswi+b1/2.0),(Tswi+b1/2.0),spsatm,swsasw2)
      CALL Airhumidity((Tswi+b2/2.0),(Tswi+b2/2.0),spsatm,swsasw3)
      CALL Airhumidity((Tswi+b3),(Tswi+b3),spsatm,swsasw4)
      d1=K1*(swsasw1-swsai)/(1.0-swsasw1)
      d2=K1*(swsasw2-(swsai+d1/2.0))/(1.0-swsasw2)
      d3=K1*(swsasw3-(swsai+d2/2.0))/(1.0-swsasw3)
      d4=K1*(swsasw4-(swsai+d3))/(1.0-swsasw4)

C Determine the exit conditions of the element
      sisao=sisai+(a1+2.0*(a2+a3)+a4)/6.0
      Tswo=Tswi+(b1+2.0*(b2+b3)+b4)/6.0
      Tspi=Tspo+(c1+2.0*(c2+c3)+c4)/6.0
      swsao=swsai+(d1+2.0*(d2+d3)+d4)/6.0
      smswo=smswi-smsael*(swsao-swsai)

C Determine the air outlet temperature and saturation enthalpy
      TR=Tspi
      TL=0.0
```

```
 10 Tsao=(TR+TL)/2.0
    CALL Cpv(Tsao,scspv)
    CALL Cpa(Tsao,scspa)
    CALL Cpw(Tsao,scspw)
    CALL Airhumidity(Tsao,Tsao,spsatm,swsasa)
    IF (swsasa.GT.swsao) THEN
      sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
    ELSE
      sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
   +          +scspw*(swsao-swsasa)*Tsao/1000.0
    END IF
    IF ((ABS(sisao-sisasa)).GT.0.1) THEN
      IF (sisao.LT.sisasa) THEN
        TR=Tsao
      ELSE
        TL=Tsao
      END IF
      GO TO 10
    END IF
    RETURN
    END

C  ************************************************************
C  *                                                          *
C  *        INITIALIZE ALL THE NEEDED PARAMETERS              *
C  *                                                          *
C  ************************************************************
C Subroutine to set default values for a typical cooler
    SUBROUTINE INITIAL(spsatm,Tsadb,Tsawb,L,H,sdso,sdsi,Kmax,
   +           Lmax,Mmax,vspas,hspas,smsp,PI,gamma,skst,Tspil,
   +           clrtype,model,smsa,flowlayout,Tswil,shsfl,shsf2)
    REAL L
    INTEGER clrtype,flowlayout

    spsatm=101325.0          !  Atmospheric pressure                [Pa]
    Tsadb=25.0               !  Dry-bulb temperature of air         [ C]
    Tsawb=19.5               !  Dry-bulb temperature of air         [ C]
    sdso=38.1/1000.0         !  Pipe Outer Diameter                  [m]
    sdsi=34.9/1000.0         !  Pipe Inner Diameter                  [m]
    Kmax=10                  !  Number of pipe rows
    Mmax=1                   !  Number of elements along each pipe
    vspas=2.0*sdso           !  Vertical spacing between pipes       [m]
    hspas=SQRT(3.0)*sdso     !  Horizontal spacing between pipes     [m]
    PI=4.0*ATAN(1.0)         !  Pi
    gamma=300.0/3600.0       !  Recirc.water massflow/length    [kg/m/s]
    shsfl=20000.0            !  Fouling heat transfer coeff.   [W/m^2 K]
    shsf2=20000.0            !  Fouling heat transfer coeff.   [W/m^2 K]
    skst=43.0                !  Thermal conductivity of tube     [W/m K]
    Tspil=50.0               !  Process water inlet temperature     [ C]
    smsp=15.0                !  Total process water-massflow      [kg/s]
    smsa=11.75388            !  Total air massflow                [kg/s]
```

```
      H=2.0                      !  Inlet height of cooler             [m]
      L=2.0                      !  Length of each pipe                [m]

      flowlayout=3               !  1 - front to back
                                 !  2 - back to front
                                 !  3 - top to bottom
                                 !  4 - straight through
      model=1                    !  1 - Merkel model
                                 !  2 - improved Merkel model
                                 !  3 - Poppe model
      clrtype=1                  !  1 - Recirc. cooling water
                                 !  2 - Single pass cooling water flow
      Tswil=35.0                 !  Inlet cooling water temperature    [ C]

      RETURN
      END

C ********************************************************************
C *                                                                  *
C *           MENU (1) :  EDIT CURRENT COOLER PARAMETERS             *
C *                                                                  *
C ********************************************************************
C Subroutine to edit the cooler dimensions
      SUBROUTINE MENU1(clrtype,model,H,L,spsatm,Tsadb,Tsawb,
     +           flowlayout,Tswil,shsf1,shsf2)

C Declare new variable types
      REAL L
      INTEGER clrtype,flowlayout

C Display the current cooler parameters on the screen
   10 CALL LIB$ERASE_PAGE(1,1)
      WRITE(*,15)
   15 FORMAT(' ',
     + 'CROSSFLOW EVAPORATIVE COOLER  -  Menu 1'/
     + ' ---------------------------------------')
      IF (clrtype.EQ.1) THEN
        WRITE(*,*)'Cooling water flow          : RECIRCULATING'
      ELSE IF (clrtype.EQ.2) THEN
        WRITE(*,*)'Cooling water flow          : SINGLE PASS'
      END IF
      IF (flowlayout.EQ.1) THEN
        WRITE(*,*)'Process water flow layout : FRONT TO BACK'
      ELSE IF (flowlayout.EQ.2) THEN
        WRITE(*,*)'Process water flow layout : BACK TO FRONT'
      ELSE IF (flowlayout.EQ.3) THEN
        WRITE(*,*)'Process water flow layout : TOP TO BOTTOM'
      ELSE IF (flowlayout.EQ.4) THEN
        WRITE(*,*)'Process water flow layout : STRAIGHT THROUGH'
      END IF
      IF (model.EQ.1) THEN
```

```
        WRITE(*,*)'Analytical model           : MERKEL'
      ELSE IF (model.EQ.2) THEN
        WRITE(*,*)'Analytical model           : Improved MERKEL'
      ELSE
        WRITE(*,*)'Analytical model           : POPPE'
      END IF
      WRITE(*,*)' '

      WRITE(*,20)H,L,spsatm/1000.0,Tsadb,Tsawb,shsf1,shsf2
   20 FORMAT(
     + ' 0 - Change cooling water flow (single pass/recirc.)'/
     + ' 1 - Change process water flow pattern'/
     + ' 2 - Change solution model (MERKEL/Improved MERKEL/POPPE)'/
     + ' 3 - Cooler height .................... = ',F8.2,' m'/
     + ' 4 - Cooler length (across airflow) ... = ',F8.2,' m'/
     + ' 5 - Atmospheric pressure ............. = ',F8.2,' kPa'/
     + ' 6 - Inlet air temperature (dry bulb) . = ',F8.2,' C'/
     + ' 7 - Inlet air temperature (wet bulb) . = ',F8.2,' C'/
     + ' 8 - Fouling coefficient inside tube .. = ',F12.2,' W/m^2 K'/
     + ' 9 - Fouling coefficient outside tube . = ',F12.2,' W/m^2 K')

C Display inlet cooling water temperature
      IF (clrtype.EQ.2) THEN
        WRITE(*,30)Tswil
   30   FORMAT(' 10- Cooling water inlet temperature .. = ',F8.2,' C')
      END IF

C Read keyboard to determine which dataset has to be changed
      WRITE(*,35)
   35 FORMAT(/' Which value has to be changed (15 - CONTINUE) ? ',$)
      READ(*,*)number

C Change cooling/recirc. water option
  999 IF (number.EQ.0) THEN
        WRITE(*,*)' '
        WRITE(*,*)'The following options are available'
        WRITE(*,*)' 1 - Recirculating cooling water flow'
        WRITE(*,*)' 2 - Single pass cooling water flow'
        WRITE(*,*)' '
        WRITE(*,*)'Enter choice (1 or 2) ?'
        READ(*,*)clrtype
        IF ((clrtype.GT.2).OR.(clrtype.LT.1)) THEN
          number=0
          GOTO 999
        END IF
        GO TO 10

C Change process water flow pattern
      ELSE IF (number.EQ.1) THEN
        WRITE(*,*)' '
        WRITE(*,*)'The following cooler layouts are available'
        WRITE(*,*)' 1 - fronttoback'
```

```
        WRITE(*,*)' 2 - backtofront'
        WRITE(*,*)' 3 - toptobottom'
        WRITE(*,*)' 4 - straight through'
        WRITE(*,*)' '
        WRITE(*,*)'Enter choice (1,2,3 or 4) ?'
        READ(*,*)flowlayout
        IF ((flowlayout.GT.4).OR.(flowlayout.LT.1)) THEN
          number=1
          GOTO 999
        END IF
        GO TO 10

C Change model type
      ELSE IF (number.EQ.2) THEN
        WRITE(*,*)' '
        WRITE(*,*)'The following models are available : '
        WRITE(*,*)'  1 - MERKEL model ( 3 Equation )'
        WRITE(*,*)'  2 - Improved MERKEL model ( 4 Equation )'
        WRITE(*,*)'  3 - POPPE model'
        WRITE(*,*)' '
        WRITE(*,*)'Enter choice (1,2 or 3) ?'
        READ(*,*)model
        IF ((model.GT.3).OR.(model.LT.1)) THEN
          number=2
          GOTO 999
        END IF
        GO TO 10

C Change cooler height
      ELSE IF (number.EQ.3) THEN
        WRITE(*,*)' '
        WRITE(*,*)'What is the new cooler height in m ?'
        READ(*,*)H
        IF (H.LE.0.0) THEN
          number=3
          GO TO 999
        END IF
        GO TO 10

C Change cooler width
      ELSE IF (number.EQ.4) THEN
        WRITE(*,*)' '
        WRITE(*,*)'What is the new cooler width in m ?'
        READ(*,*)L
        IF (L.LE.0.0) THEN
          number=4
          GO TO 999
        END IF
        GO TO 10

C Change the value of atmospheric pressure
      ELSE IF (number.EQ.5) THEN
```

```
      WRITE(*,*)' '
      WRITE(*,*)'What is the new atmospheric pressure in  kPa ? '
      READ(*,*)spsatm
C Ensure that input is not an absurd value
      IF (spsatm.LT.60.0) THEN
        WRITE(*,*)'The atmospheric pressure must be above 60 kPa'
        number=5
        GO TO 999
      END IF
      spsatm=spsatm*1000.0
      GO TO 10

C Change the value of the inlet air temperature (dry bulb)
    ELSE IF (number.EQ.6) THEN
      WRITE(*,*)' '
      WRITE(*,*)'Give the air inlet temperature (dry bulb C) ? '
      READ(*,*)Tsadb
      IF (Tsadb.LT.Tsawb) THEN
        WRITE(*,*)'Wet bulb temperature > Dry bulb temperature'
        WRITE(*,*)'T(dry bulb) larger (0) / T(wet bulb) smaller (1) ?'
        READ(*,*)number3
        IF (number3.EQ.0) THEN
          number=6                ! Choose new dry bulb temperature
          GO TO 999
        ELSE
          number=7                ! Choose new wet bulb temperature
          GO TO 999
        END IF
      ELSE IF (Tsadb.GT.100) THEN
        WRITE(*,*)'Air temperature (dry bulb) must be < 100 C'
        number=6
        GO TO 999
      END IF
      GO TO 10

C Change the value of the inlet air temperature (wet bulb)
    ELSE IF (number.EQ.7) THEN
      WRITE(*,*)' '
      WRITE(*,*)'Give the air inlet temperature (wet bulb C) ? '
      READ(*,*)Tsawb
      IF (Tsadb.LT.Tsawb) THEN
        WRITE(*,*)'Wet bulb temperature > Dry bulb temperature'
        WRITE(*,*)'T(dry bulb) larger (0) / T(wet bulb) smaller (1) ?'
        READ(*,*)number3
        IF (number3.EQ.0) THEN
          number=6             ! Choose new dry bulb temperature
          GO TO 999
        ELSE
          number=7             ! Choose new wet bulb temperature
          GO TO 999
        END IF
      END IF
```

```
        GO TO 10

C Change cooling water inlet temperature
      ELSE IF (number.EQ.8) THEN
        WRITE(*,*)' '
        WRITE(*,*)'What is the new outer fouling coefficient ?'
        READ(*,*)shsf1
        IF (shsf1.LE.1000.0) THEN
          WRITE(*,*)'Fouling coefficient > 1000 W/m^2 K'
          number=8
          GO TO 999
        END IF
        GO TO 10

C Change cooling water inlet temperature
      ELSE IF (number.EQ.9) THEN
        WRITE(*,*)' '
        WRITE(*,*)'What is the new outer fouling coefficient ?'
        READ(*,*)shsf2
        IF (shsf2.LE.1000.0) THEN
          WRITE(*,*)'Fouling coefficient > 1000 W/m^2 K'
          number=9
          GO TO 999
        END IF
        GO TO 10

C Change cooling water inlet temperature
      ELSE IF (number.EQ.10) THEN
        WRITE(*,*)' '
        WRITE(*,*)'What is the new cooling water inlet temperature ?'
        READ(*,*)Tswil
        IF ((Tswil.LE.0.0).OR.(Tswil.GT.100.0)) THEN
          number=10
          GO TO 999
        END IF
        GO TO 10
      END IF
      RETURN
      END

C ******************************************************************
C *                                                                *
C *          MENU (2) :  EDIT CURRENT COOLER PARAMETERS            *
C *                                                                *
C ******************************************************************
C Subroutine to edit the cooler dimensions
      SUBROUTINE MENU2(sdso,sdsi,H,L,PI,svsa,vspas,hspas,Lmax,Kmax,
     +           Mmax,spsatm,Tsadb,Tsawb,gamma,skst,Tspil,smsp,
     +           svsp,sa,Aspi,Aspo,clrtype,dA,model,smsa,flowlayout,
     +           Tswil,shsf1,shsf2)
```

```
C Initialize data types
      REAL L
      INTEGER clrtype,flowlayout

C Display the current cooler parameters on the screen
   10 CALL LIB$ERASE_PAGE(1,1)
      WRITE(*,15)
   15 FORMAT(' ',
     + 'CROSSFLOW EVAPORATIVE COOLER  -  Menu 2'/
     + ' ------------------------------------')
      IF (clrtype.EQ.1) THEN
        WRITE(*,*)'Cooling water flow          : RECIRCULATING'
      ELSE IF (clrtype.EQ.2) THEN
        WRITE(*,*)'Cooling water flow          : SINGLE PASS'
      END IF
      IF (flowlayout.EQ.1) THEN
        WRITE(*,*)'Process water flow layout : FRONT TO BACK'
      ELSE IF (flowlayout.EQ.2) THEN
        WRITE(*,*)'Process water flow layout : BACK TO FRONT'
      ELSE IF (flowlayout.EQ.3) THEN
        WRITE(*,*)'Process water flow layout : TOP TO BOTTOM'
      ELSE IF (flowlayout.EQ.4) THEN
        WRITE(*,*)'Process water flow layout : STRAIGHT THROUGH'
      END IF
      IF (model.EQ.1) THEN
        WRITE(*,*)'Analytical model            : MERKEL'
      ELSE IF (model.EQ.2) THEN
        WRITE(*,*)'Analytical model            : Improved MERKEL'
      ELSE
        WRITE(*,*)'Analytical model            : POPPE'
      END IF
      WRITE(*,*)' '

      WRITE(*,20)sdso*1000.0,sdsi*1000.0,vspas*1000.0,hspas*1000.0
   20 FORMAT(
     + ' 0 - Go back to previous menu '/
     + ' 1 - Outer diameter of pipe ............... = ',F7.2,' mm'/
     + ' 1 - Inner diameter of pipe ............... = ',F7.2,' mm'/
     + ' 2 - Vertical spacing between pipes ....... = ',F7.2,' mm'/
     + ' 2 - Horizontal spacing between pipes ..... = ',F7.2,' mm')

C Check whether the array-dimensions were large enough
      Lmax=INT((H-0.5*vspas+0.001)/vspas)
      IF (Lmax.GT.400) THEN
        WRITE(*,*)'Max number of elements in vertical direction'
        WRITE(*,*)'permitted = 400 '
        WRITE(*,*)'Choose a larger vertical spacing (0)'
        WRITE(*,*)'or change the DIMENSION of the array (1)'
        READ(*,*)iantw
        IF (iantw.EQ.0) THEN
          number=2
          GO TO 999          ! Choose bigger vertical spacing
```

```
        ELSE
          WRITE(*,*)'REMEMBER TO CHANGE THIS CONDITION AS WELL'
          STOP                ! Change the DIMENSION statement
        END IF

C Check whether the new vertical spacing is allowable
      ELSE IF (Lmax.LT.1) THEN
        WRITE(*,*)'Vertical spacing too large to fit at least one'
        WRITE(*,*)'pipe into the cooler'
        WRITE(*,*)'Choose new vertical spacing'
        number=2
        GO TO 999
      END IF

      sa=PI*sdso/(vspas*hspas)      ! Coolerarea/unit volume
      dA=L*PI*sdso/(Mmax)            ! Coolerarea/element
      Aspi=PI*(sdsi/2.0)**2.0        ! Pipe inner area
      Aspo=PI*(sdso/2.0)**2.0        ! Pipe outer area

C Determine the water veloctity inside tubes and massflow needed to give
C a water velocity of 1 m/s in tubes
      CALL Waterdensity(Tspil,rhosw)
      IF ((flowlayout.EQ.1).OR.(flowlayout.EQ.2)) THEN
        svsp=smsp/(Aspi*rhosw*Lmax)
        svspl=rhosw*Aspi*Lmax*1.0
      ELSE IF (flowlayout.EQ.3) THEN
        svsp=smsp/(Aspi*rhosw*Kmax)
        svspl=rhosw*Aspi*Kmax*1.0
      ELSE IF (flowlayout.EQ.4) THEN
        svsp=smsp/(Aspi*rhosw*Kmax*Lmax)
        svspl=rhosw*Aspi*Kmax*Lmax*1.0
      END IF

C Print the variable values on the screen in order to edit them if needed
      WRITE(*,30)Kmax,Lmax,Mmax,Tspil,smsp,svsp,gamma*3600.0,
     +            gamma*Kmax*L*2.0,smsa,skst
   30 FORMAT(
     + ' 3 - Number of pipe rows (passes) ......... = ',I3/
     + '     Number of pipes facing the airstream . = ',I3/
     + ' 4 - Number of elements along a single pipe = ',I3/
     + ' 5 - Process water inlet temperature ...... = ',F7.2,' C'/
     + ' 6 - Process water massflow ............... = ',F7.2,' kg/s'/
     + '     Process water flow velocity in pipes . = ',F7.2,' m/s'/
     + ' 7 - Recirc.water massflow / length ....... = ',F7.2,' kg/m.hr'/
     + '     Recirculating water massflow ......... = ',F7.2,' kg/s'/
     + ' 8 - Dry air massflow rate ................ = ',F7.2,' kg/s'/
     + ' 9 - Thermal conductivity of tube wall .... = ',F7.2,' W/m K'/)

C Read keyboard to determine which dataset has to be changed
      WRITE(*,35)
   35 FORMAT(' Which value has to be changed (15 - CONTINUE) ? ',$)
      READ(*,*)number
```

```
C Change cooler unit layout
  999 IF (number.EQ.0) THEN
        CALL MENU1(clrtype,model,H,L,spsatm,Tsadb,Tsawb,flowlayout,
     +      Tswi1,shsf1,shsf2)
        GO TO 10

C Change the pipe dimensions
      ELSE IF (number.EQ.1) THEN
        WRITE(*,*)' '
        WRITE(*,*)'Enter the pipe outer diameter in mm ?'
        READ(*,*)sdso
        WRITE(*,*)'Enter the pipe inner diameter in mm ?'
        READ(*,*)sdsi
        sdso=sdso/1000.0
        sdsi=sdsi/1000.0
C Check if the pipe size is physically allowable with given configuration
        IF (sdsi.GE.sdso) THEN
           WRITE(*,*)'inner diameter >= outer diameter'
           number=1
           GO TO 999
        ELSE IF (sdso.GT.vspas) THEN
           WRITE(*,*)'Element boandaries interfere !!!'
           WRITE(*,*)'Choose smaller pipe diameter (0)   '
           WRITE(*,*)'or choose a larger vertical spacing (1) ? '
           READ(*,*)number2
           IF (number2.EQ.1) THEN
             number=2                        ! Change spacing
           ELSE
             number=1                        ! Change pipe diameter
           END IF
           GO TO 999
        ELSE IF (sdso.GT.hspas) THEN
           WRITE(*,*)'Element boundaries interfere !!!'
           WRITE(*,*)'Choose smaller pipe diameter (0)   '
           WRITE(*,*)'or choose a larger horizontal spacing (1) ? '
           READ(*,*)number2
           IF (number2.EQ.1) THEN
             number=2                        ! Change spacing
           ELSE
             number=1                        ! Change pipe diameter
           END IF
           GO TO 999
        END IF
        vspas=2.0*sdso
        hspas=SQRT(3.0)*sdso
        GO TO 10

C Change the spacing of the pipe array
      ELSE IF (number.EQ.2) THEN
        WRITE(*,*)' '
        WRITE(*,*)'Give the vertical spacing between pipes in mm ?'
```

```
      READ(*,*)vspas
      WRITE(*,*)'Give the horizontal spacing between pipes in mm ?'
      READ(*,*)hspas
      vspas=vspas/1000.0
      hspas=hspas/1000.0
C Check whether this configuration is physically possible
C with the chosen pipes
      IF (sdso.GT.vspas) THEN
        WRITE(*,*)'Element boundaries interfere !!!'
        WRITE(*,*)'Choose smaller pipe diameter (0)   '
        WRITE(*,*)'or choose a larger vertical spacing (1) ? '
        READ(*,*)number2
        IF (number2.EQ.0) THEN
          number=1                      ! Change pipe diameter
        ELSE
          number=2                      ! Change spacing
        END IF
        GO TO 999
      ELSE IF (sdso.GT.hspas) THEN
        WRITE(*,*)'Element boundaries interfere !!!'
        WRITE(*,*)'Choose smaller pipe diameter (0)   '
        WRITE(*,*)'or choose a larger horizontal spacing (1) ? '
        READ(*,*)number2
        IF (number2.EQ.0) THEN
          number=1                      ! Change pipe diameter
        ELSE
          number=2                      ! Change spacing
        END IF
        GO TO 999
      END IF
      GO TO 10

C Change the number of pipe rows
     ELSE IF (number.EQ.3) THEN
      WRITE(*,*)' '
      WRITE(*,*)'Number of pipe rows ? '
      READ(*,*)Kmax
C Ensure that there is a positive number of pipe rows
      IF (Kmax.LT.1) THEN
        WRITE(*,*)'Minimum number of pipe = 1 - Choose again'
        number=3
        GO TO 999
C Check if array-DIMENSION size was sufficient
      ELSE IF (Kmax.GT.40) THEN
        WRITE(*,*)'Max number of pipe rows = 40'
        WRITE(*,*)'Choose less pipe rows (0) OR '
        WRITE(*,*)'change array DIMENSION (1) ?'
        READ(*,*)iantw
C Determine if DIMENSION or number of rows must be changed
        IF (iantw.EQ.0) THEN
          number=3          ! Choose less then 40 pipe rows
          GO TO 999
```

```
        ELSE
          WRITE(*,*)'Change array DIMENSION in source code'
          WRITE(*,*)'REMEMBER to change this condition too'
          STOP
        END IF
      END IF
      GO TO 10

C Change the number of elements across a single pipe
     ELSE IF (number.EQ.4) THEN
      WRITE(*,*)' '
      WRITE(*,*)'Number of elements across a single pipe ? '
      READ(*,*)Mmax
C Ensure a positive number of elements
      IF (Mmax.LT.1) THEN
        WRITE(*,*)'Minimum number of elements = 1 - Choose again'
        number=4
        GO TO 999
C Check if array-DIMENSION size was sufficient
      ELSE IF (Mmax.GT.10) THEN
        WRITE(*,*)'Max number of pipe rows = 10'
        WRITE(*,*)'Choose less elements along pipe (0) OR '
        WRITE(*,*)'change array DIMENSION (1) ?'
        READ(*,*)iantw
C Determine if DIMENSION or number of elements must be changed
        IF (iantw.EQ.0) THEN
          number=4           ! Choose less than 10 elements
          GOTO 999
        ELSE
          WRITE(*,*)'Change array DIMENSION in source code'
          WRITE(*,*)'REMEMBER to change this condition too'
          STOP
        END IF
      END IF
      GO TO 10

C Change the process water inlettemperature
     ELSE IF (number.EQ.5) THEN
      WRITE(*,*)' '
      WRITE(*,*)'Process water inlet temperature ( C) ? '
      READ(*,*)Tspil
      IF (Tspil.GT.100) THEN
        WRITE(*,*)'Process water inlet temperature must be < 100 C'
        number=5
        GO TO 999
      ELSE IF (Tspil.LE.0) THEN
        WRITE(*,*)'Process water inlet temperature must be > 0 C'
        number=5
        GO TO 999
      END IF
      GO TO 10
```

```
C Change the process water massflow rate
      ELSE IF (number.EQ.6) THEN
        WRITE(*,*)' '
        WRITE(*,40)smsp,svspl
   40   FORMAT(
     +      ' Process water massflow for cooler  .. = ',F9.2,' kg/s'/
     +      ' Massflow needed for velocity of 1 m/s = ',F9.2,' kg/s'/)
        WRITE(*,*)'Give new total massflow (kg/s) ? '
        READ(*,*)smsp
        IF (smsp.LE.0) THEN
          WRITE(*,*)'Total process water massflow must be > 0 kg/s'
          number=6
          GO TO 999
        END IF
        GO TO 10

C Change the recirculating water massflow rate
      ELSE IF (number.EQ.7) THEN
        WRITE(*,*)' '
        WRITE(*,45)gamma*3600
   45   FORMAT(' Previous recirc. water massflow (kg/m/hr) = ',F8.2,/
     +         ' New recirc. water massflow (kg/m/hr) ? ')
        READ(*,*)gamma
        gamma=gamma/3600.0
        IF (gamma.LT.1.5*700.*sdso/3600.) THEN
          WRITE(*,46)1.5*700.*sdso
          number=7
          GO TO 999
        END IF
   46   FORMAT(' Recirc. water massflow must be > ',F8.2,' kg/m/hr')
        GO TO 10

C Change the air massflow rate
      ELSE IF (number.EQ.8) THEN
        WRITE(*,*)' '
        WRITE(*,*)'Dry air massflow (kg/s) ? '
        READ(*,*)smsa
        IF (smsa.LT.0) THEN
          WRITE(*,*)'Total dry air massflow must be >= 0 kg/s'
          number=8
          GO TO 999
        END IF
        GO TO 10

C Change the tube wall thermal conductivity
      ELSE IF (number.EQ.9) THEN
        WRITE(*,*)' '
        WRITE(*,*)'Thermal conductivity for different tube materials'
        WRITE(*,*)'Kt (Aluminium)      -  204  W/mK'
        WRITE(*,*)'Kt (Steel 0.5% C)   -  54   W/mK'
        WRITE(*,*)'Kt (Steel 1.0% C)   -  43   W/mK'
        WRITE(*,*)'Kt (Steel 1.5% C)   -  36   W/mK'
```

```
      WRITE(*,*)'Kt (Copper)           -  376 W/mK'
      WRITE(*,*)'Tube wall thermal conductivity (W/mK) ? '
      READ(*,*)skst
      IF (skst.LT.10.0) THEN
        WRITE(*,*)'Conductivity must be > 10'
        number=9
        GO TO 999
      END IF
      GO TO 10
    END IF
    RETURN
    END

C ******************************************************************
C *                                                                *
C *        MERKEL METHOD TO EVAUALTE A SINGLE ELEMENT              *
C *                                                                *
C ******************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the three
C Merkel equations which controls the state of a single element
      SUBROUTINE MERKEL(Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +           ReyC,shsf1,shsf2,nrow)

      REAL L,musav,musw,kog,koga,K1,K2,K3,K4

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspi,musw)
      CALL Waterdensity(Tspi,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)        ! Reynoldsnumber of process water
      Tsa=Tsadb
      CALL AirVapMixdensity(Tsa,swsail,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsa,swsail,spsatm,musav)
      Reysa=ReyC*rhosav/musav             ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      Reysw=4.0*gamma/musw                ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspi,sksp)
      CALL Prandtl(Tspi,Prasp)
      shsw=4.186*118.0*((gamma*3600/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
        sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
        term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
        term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
        shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
      END IF
      koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
```

```
      kog=koga/sa   !  Mass-transfer coefficient
      Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsf1))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3 and K4
      CALL Cpw(Tspi,scspp)
      CALL Cpw(Tswi,scspw)
      K1=kog*dA/(smsael)
      K2=Uo*dA/(smswel*scspw)
      K3=kog*dA*1000.0/(smswel*scspw)
      K4=Uo*dA/(smspel*scspp)

C Determine the Runge-Kutta coefficients
      CALL Satenthalpy(Tswi,spsatm,sisasw1)
      a1=K1*(sisasw1-sisai)
      b1=K2*(Tspi-Tswi)-K3*(sisasw1-sisai)
      c1=-K4*(Tspi-Tswi)

      CALL Satenthalpy((Tswi+b1/2.0),spsatm,sisasw2)
      a2=K1*(sisasw2-(sisai+a1/2.0))
      b2=K2*((Tspi+c1/2.0)-(Tswi+b1/2.0))-K3*(sisasw2-(sisai+a1/2.0))
      c2=-K4*((Tspi+c1/2.0)-(Tswi+b1/2.0))

      CALL Satenthalpy((Tswi+b2/2.0),spsatm,sisasw3)
      a3=K1*(sisasw3-(sisai+a2/2.0))
      b3=K2*((Tspi+c2/2.0)-(Tswi+b2/2.0))-K3*(sisasw3-(sisai+a2/2.0))
      c3=-K4*((Tspi+c2/2.0)-(Tswi+b2/2.0))

      CALL Satenthalpy((Tswi+b3),spsatm,sisasw4)
      a4=K1*(sisasw4-(sisai+a3))
      b4=K2*((Tspi+c3)-(Tswi+b3))-K3*(sisasw4-(sisai+a3))
      c4=-K4*((Tspi+c3)-(Tswi+b3))

C Determine the exit conditions of the element
      sisao=sisai+(a1+2.0*(a2+a3)+a4)/6.0
      Tswo=Tswi+(b1+2.0*(b2+b3)+b4)/6.0
      Tspo=Tspi+(c1+2.0*(c2+c3)+c4)/6.0
      RETURN
      END


C ********************************************************************
C *                                                                  *
C *        MERKEL(2) METHOD TO EVAUALTE A SINGLE ELEMENT             *
C *                                                                  *
C ********************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the three
C Merkel equations which controls the state of a single element
C BACKTOFRONT FLOW CASE
      SUBROUTINE MERKEL2(Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
```

```
     +            ReyC,shsf1,shsf2,nrow)

      REAL L,musav,musw,kog,koga,K1,K2,K3,K4

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspo,musw)
      CALL Waterdensity(Tspo,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)          ! Reynoldsnumber of process water
      Tsa=Tsadb
      CALL AirVapMixdensity(Tsa,swsail,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsa,swsail,spsatm,musav)
      Reysa=ReyC*rhosav/musav               ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      Reysw=4.0*gamma/musw                  ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspo,sksp)
      CALL Prandtl(Tspo,Prasp)
      shsw=4.186*118*((gamma*3600/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
        sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
        term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
        term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
        shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
      END IF
      koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
      kog=koga/sa    !  Mass-transfer coefficient
      Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsf1))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3 and K4
      CALL Cpw(Tspo,scspp)
      CALL Cpw(Tswi,scspw)
      K1=kog*dA/(smsael)
      K2=Uo*dA/(smswel*scspw)
      K3=kog*dA*1000.0/(smswel*scspw)
      K4=Uo*dA/(smspel*scspp)

C Determine the Runge-Kutta coefficients
      CALL Satenthalpy(Tswi,spsatm,sisaswl)
      a1=K1*(sisaswl-sisai)
      b1=K2*(Tspo-Tswi)-K3*(sisaswl-sisai)
      c1=K4*(Tspo-Tswi)

      CALL Satenthalpy((Tswi+b1/2.0),spsatm,sisasw2)
      a2=K1*(sisasw2-(sisai+a1/2.0))
      b2=K2*((Tspo+c1/2.0)-(Tswi+b1/2.0))-K3*(sisasw2-(sisai+a1/2.0))
      c2=K4*((Tspo+c1/2.0)-(Tswi+b1/2.0))
```

```
      CALL Satenthalpy((Tswi+b2/2.0),spsatm,sisasw3)
      a3=K1*(sisasw3-(sisai+a2/2.0))
      b3=K2*((Tspo+c2/2.0)-(Tswi+b2/2.0))-K3*(sisasw3-(sisai+a2/2.0))
      c3=K4*((Tspo+c2/2.0)-(Tswi+b2/2.0))

      CALL Satenthalpy((Tswi+b3),spsatm,sisasw4)
      a4=K1*(sisasw4-(sisai+a3))
      b4=K2*((Tspo+c3)-(Tswi+b3))-K3*(sisasw4-(sisai+a3))
      c4=K4*((Tspo+c3)-(Tswi+b3))

C Determine the exit conditions of the element
      sisao=sisai+(a1+2.0*(a2+a3)+a4)/6.0
      Tswo=Tswi+(b1+2.0*(b2+b3)+b4)/6.0
      Tspi=Tspo+(c1+2.0*(c2+c3)+c4)/6.0
      RETURN
      END


C *******************************************************************
C *                                                                 *
C *         POPPE METHOD TO EVAUALTE A SINGLE ELEMENT               *
C *                                                                 *
C *******************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the five
C Poppe equations which controls the state of a single element
      SUBROUTINE POPPE (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +           sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +           smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +           smswo,Aspi,Aspo,ReyC,shsf1,shsf2,nrow)

      REAL L,Lew,musav,musw,kog,koga,K1,K2,K3,K4,K5,K6,
     +     ii1,ii2,ii3,ii4,iv

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspi,musw)
      CALL Waterdensity(Tspi,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)          ! Reynoldsnumber of process water
      CALL AirVapMixdensity(Tsai,swsai,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsai,swsai,spsatm,musav)
      Reysa=ReyC*rhosav/musav               ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      gamma1=(smswi/smswel)*gamma
      Reysw=4.0*gamma1/musw   ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspi,sksp)
      CALL Prandtl(Tspi,Prasp)
      shsw=4.186*118.0*((gamma1*3600.0/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
```

```
          sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
          term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
          term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
          shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
        END IF
        koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
        kog=koga/sa    !  Mass-transfer coefficient
        Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsf1))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3,K4,K5 and K6
        CALL Cpw(Tswi,scspw)
        CALL Cpw(Tspi,scspp)
        K1=kog*dA
        K2=kog*dA/smsael
        K3=kog*dA/smswi
        K4=Uo*dA/(smswi*scspw)
        K5=kog*dA/(smswi*scspw)
        K6=Uo*dA/(smspel*scspp)
        CALL Cpv(Tswi,scspv)
        iv=2501.6+scspv*Tswi/1000.0

C Determine the humidity of saturated air
        CALL Airhumidity(Tsai,Tsai,spsatm,swsasa)

C Determine the Lewis factor
        CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
        term=(0.622+swsasw)/(0.622+swsai)
        Lew=(0.90854253)*((term-1.0)/(LOG(term)))

C Determine the Runge-Kutta coefficients
        IF (swsasa.GE.swsai) THEN                ! Air not saturated
          CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
          CALL Satenthalpy(Tswi,spsatm,sisasw)
          ww1=swsasw-swsai
          ii1=(sisasw-sisai)
          TT1=Tspi-Tswi
          a1=-K1*ww1
          b1=K2*ww1
          c1=K2*(Lew*ii1-(Lew-1.0)*ww1*iv)
          d1=K3*Tswi*ww1+K4*TT1-K5*(c1/K2)*1000.0
          e1=-K6*TT1

          CALL Airhumidity((Tswi+d1/2.0),(Tswi+d1/2.0),spsatm,swsasw)
          CALL Satenthalpy((Tswi+d1/2.0),spsatm,sisasw)
          ww2=swsasw-(swsai+b1/2.0)
          ii2=(sisasw-(sisai+c1/2.0))
          TT2=(Tspi+e1/2.0)-(Tswi+d1/2.0)
          a2=-K1*ww2
          b2=K2*ww2
          c2=K2*(Lew*ii2-(Lew-1.0)*ww2*iv)
          d2=K3*(Tswi+d1/2.0)*ww2+K4*TT2-K5*(c2/K2)*1000.0
```

```
      e2=-K6*TT2

      CALL Airhumidity((Tswi+d2/2.0),(Tswi+d2/2.0),spsatm,swsasw)
      CALL Satenthalpy((Tswi+d2/2.0),spsatm,sisasw)
      ww3=swsasw-(swsai+b2/2.0)
      ii3=(sisasw-(sisai+c2/2.0))
      TT3=(Tspi+e2/2.0)-(Tswi+d2/2.0)
      a3=-K1*ww3
      b3=K2*ww3
      c3=K2*(Lew*ii3-(Lew-1.0)*ww3*iv)
      d3=K3*(Tswi+d2/2.0)*ww3+K4*TT3-K5*(c3/K2)*1000.0
      e3=-K6*TT3

      CALL Airhumidity((Tswi+d3),(Tswi+d3),spsatm,swsasw)
      CALL Satenthalpy((Tswi+d3),spsatm,sisasw)
      ww4=swsasw-(swsai+b3)
      ii4=(sisasw-(sisai+c3))
      TT4=(Tspi+e3)-(Tswi+d3)
      a4=-K1*ww4
      b4=K2*ww4
      c4=K2*(Lew*ii4-(Lew-1.0)*ww4*iv)
      d4=K3*(Tswi+d3)*ww4+K4*TT4-K5*(c4/K2)*1000.0
      e4=-K6*TT4
    ELSE IF (swsasa.LT.swsai) THEN                   ! Air saturated
      CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
      CALL Satenthalpy(Tswi,spsatm,sisasw)
      ww1=swsasw-swsasa
      ii1=(sisasw-sisai)
      TT1=Tspi-Tswi
      www1=swsai-swsasa
      a1=-K1*ww1
      b1=K2*ww1
      c1=K2*((Lew*ii1-(Lew-1.0)*ww1*iv)
     +             +www1*Lew*scspw*Tswi/1000.0)
      d1=K3*Tswi*ww1+K4*TT1-K5*(c1/K2)*1000.0
      e1=-K6*TT1

      CALL Airhumidity((Tswi+d1/2.0),(Tswi+d1/2),spsatm,swsasw)
      CALL Satenthalpy((Tswi+d1/2.0),spsatm,sisasw)
      ww2=swsasw-swsasa
      ii2=(sisasw-(sisai+c1/2.0))
      TT2=(Tspi+e1/2.0)-(Tswi+d1/2.0)
      www2=(swsai+b1/2.0)-swsasa
      a2=-K1*ww2
      b2=K2*ww2
      c2=K2*((Lew*ii2-(Lew-1.0)*ww2*iv)
     +               +www2*Lew*scspw*(Tswi+d1/2.0)/1000.0)
      d2=K3*(Tswi+d1/2.0)*ww2+K4*TT2-K5*(c2/K2)*1000.0
      e2=-K6*TT2

      CALL Airhumidity((Tswi+d2/2.0),(Tswi+d2/2),spsatm,swsasw)
      CALL Satenthalpy((Tswi+d2/2.0),spsatm,sisasw)
```

```
        ww3=swsasw-swsasa
        ii3=(sisasw-(sisai+c2/2.0))
        TT3=(Tspi+e2/2.0)-(Tswi+d2/2.0)
        www3=(swsai+b2/2.0)-swsasa
        a3=-K1*ww3
        b3=K2*ww3
        c3=K2*((Lew*ii3-(Lew-1.0)*ww3*iv)
     +                  +www3*Lew*scspw*(Tswi+d2/2.0)/1000.0)
        d3=K3*(Tswi+d2/2.0)*ww3+K4*TT3-K5*(c3/K2)*1000.0
        e3=-K6*TT3

        CALL Airhumidity((Tswi+d3),(Tswi+d3),spsatm,swsasw)
        CALL Satenthalpy((Tswi+d3),spsatm,sisasw)
        ww4=swsasw-swsasa
        ii4=(sisasw-(sisai+c3))
        TT4=(Tspi+e3)-(Tswi+d3)
        www4=(swsai+b3)-swsasa
        a4=-K1*ww4
        b4=K2*ww4
        c4=K2*((Lew*ii4-(Lew-1.0)*ww4*iv)
     +                  +www4*Lew*scspw*(Tswi+d3)/1000.0)
        d4=K3*(Tswi+d3)*ww4+K4*TT4-K5*(c4/K2)*1000.0
        e4=-K6*TT4

      END IF
      smswo=smswi+(a1+2.0*(a2+a3)+a4)/6.0
      swsao=swsai+(b1+2.0*(b2+b3)+b4)/6.0
      sisao=sisai+(c1+2.0*(c2+c3)+c4)/6.0
      Tswo=Tswi+(d1+2.0*(d2+d3)+d4)/6.0
      Tspo=Tspi+(e1+2.0*(e2+e3)+e4)/6.0

C Determine the air outlet temperature and saturation enthalpy
      TR=Tspi
      TL=0.0
   10 Tsao=(TR+TL)/2.0
      CALL Cpv(Tsao,scspv)
      CALL Cpa(Tsao,scspa)
      CALL Cpw(Tsao,scspw)
      CALL Airhumidity(Tsao,Tsao,spsatm,swsasa)
      IF (swsasa.GT.swsao) THEN
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
      ELSE
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
     +          +scspw*(swsao-swsasa)*Tsao/1000.0
      END IF
      IF ((ABS(sisao-sisasa)).GT.0.1) THEN
        IF (sisao.LT.sisasa) THEN
          TR=Tsao
        ELSE
          TL=Tsao
        END IF
        GO TO 10
```

```
      END IF
      RETURN
      END

C ******************************************************************
C *                                                                *
C *      POPPE METHOD(2) TO EVAUALTE A SINGLE ELEMENT              *
C *                                                                *
C ******************************************************************
C Subroutine to apply the Runge-Kutta method of solution to the five
C Poppe equations which controls the state of a single element
C BACKTOFRONT FLOW CASE
      SUBROUTINE POPPE2 (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +           sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +           smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +           smswo,Aspi,Aspo,ReyC,shsf1,shsf2,nrow)

      REAL L,Lew,musav,musw,kog,koga,K1,K2,K3,K4,K5,K6,
     +     ii1,ii2,ii3,ii4,iv

C Determine the neccessary Reynoldsnumbers
      CALL Waterviscosity(Tspo,musw)
      CALL Waterdensity(Tspo,rhosw)
      Reysp=rhosw*sdsi*svsp/(musw)          ! Reynoldsnumber of process water
      CALL AirVapMixdensity(Tsai,swsai,spsatm,rhosav)
      CALL AirVapMixviscosity(Tsai,swsai,spsatm,musav)
      Reysa=ReyC*rhosav/musav               ! Reynoldsnumber of airflow
      CALL Waterviscosity(Tswi,musw)
      gammal=(smswi/smswel)*gamma
      Reysw=4.0*gammal/musw   ! Reynoldsnumber of recirc.water

C Determine the neccessary transfer-coefficients
      CALL Waterconductivity(Tspo,sksp)
      CALL Prandtl(Tspo,Prasp)
      shsw=4.186*118.0*((gammal*3600.0/sdso)**(1.0/3.0))/3.6
      IF (Reysp.LT.2300.0) THEN
        term=Reysp*Prasp*sdsi/(L*nrow)
        shsp=(3.66+0.104*(term)/(1.0+0.016*(term)**(0.8)))*sksp/sdsi
      ELSE
        sfsd=(1.82*LOG10(Reysp)-1.64)**(-2.0)
        term1=Prasp*(1.0+(sdsi/(L*nrow))**(0.67))
        term2=1.0+12.7*((sfsd/8.0)**(0.5))*(Prasp**(0.67)-1.0)
        shsp=((sfsd/8.0)*(Reysp-1000.0)*term1/term2)*sksp/sdsi
      END IF
      koga=1.81E-4*((Reysa)**.9)*((Reysw)**.15)*((sdso)**(-2.6))/3600.
      kog=koga/sa   !  Mass-transfer coefficient
      Uo=1.0/((sdso/sdsi)*((1.0/shsp)+(1.0/shsf1))+(1.0/shsw)
     +       +(1.0/shsf2)+sdso*LOG(sdso/sdsi)/(2.0*skst))

C Determine the controlling constants K1,K2,K3,K4,K5 and K6
      CALL Cpw(Tswi,scspw)
```

```
      CALL Cpw(Tspo,scspp)
      K1=kog*dA
      K2=kog*dA/smsael
      K3=kog*dA/smswi
      K4=Uo*dA/(smswi*scspw)
      K5=kog*dA/(smswi*scspw)
      K6=Uo*dA/(smspel*scspp)
      CALL Cpv(Tswi,scspv)
      iv=2501.6+scspv*Tswi/1000.0

C Determine the humidity of saturated air
      CALL Airhumidity(Tsai,Tsai,spsatm,swsasa)

C Determine the Lewis factor
      CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
      term=(0.622+swsasw)/(0.622+swsai)
      Lew=(0.90854253)*((term-1.0)/(LOG(term)))

C Determine the Runge-Kutta coefficients
      IF (swsasa.GE.swsai) THEN                ! Air not saturated
        CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
        CALL Satenthalpy(Tswi,spsatm,sisasw)
        ww1=swsasw-swsai
        ii1=(sisasw-sisai)
        TT1=Tspo-Tswi
        a1=-K1*ww1
        b1=K2*ww1
        c1=K2*(Lew*ii1-(Lew-1.0)*ww1*iv)
        d1=K3*Tswi*ww1+K4*TT1-K5*(c1/K2)*1000.0
        e1=K6*TT1

        CALL Airhumidity((Tswi+d1/2.0),(Tswi+d1/2.0),spsatm,swsasw)
        CALL Satenthalpy((Tswi+d1/2.0),spsatm,sisasw)
        ww2=swsasw-(swsai+b1/2.0)
        ii2=(sisasw-(sisai+c1/2.0))
        TT2=(Tspo+e1/2.0)-(Tswi+d1/2.0)
        a2=-K1*ww2
        b2=K2*ww2
        c2=K2*(Lew*ii2-(Lew-1.0)*ww2*iv)
        d2=K3*(Tswi+d1/2.0)*ww2+K4*TT2-K5*(c2/K2)*1000.0
        e2=K6*TT2

        CALL Airhumidity((Tswi+d2/2.0),(Tswi+d2/2.0),spsatm,swsasw)
        CALL Satenthalpy((Tswi+d2/2.0),spsatm,sisasw)
        ww3=swsasw-(swsai+b2/2.0)
        ii3=(sisasw-(sisai+c2/2.0))
        TT3=(Tspo+e2/2.0)-(Tswi+d2/2.0)
        a3=-K1*ww3
        b3=K2*ww3
        c3=K2*(Lew*ii3-(Lew-1.0)*ww3*iv)
        d3=K3*(Tswi+d2/2.0)*ww3+K4*TT3-K5*(c3/K2)*1000.0
        e3=K6*TT3
```

```
    CALL Airhumidity((Tswi+d3),(Tswi+d3),spsatm,swsasw)
    CALL Satenthalpy((Tswi+d3),spsatm,sisasw)
    ww4=swsasw-(swsai+b3)
    ii4=(sisasw-(sisai+c3))
    TT4=(Tspo+e3)-(Tswi+d3)
    a4=-K1*ww4
    b4=K2*ww4
    c4=K2*(Lew*ii4-(Lew-1.0)*ww4*iv)
    d4=K3*(Tswi+d3)*ww4+K4*TT4-K5*(c4/K2)*1000.0
    e4=K6*TT4
   ELSE IF (swsasa.LT.swsai) THEN                  ! Air saturated
    CALL Airhumidity(Tswi,Tswi,spsatm,swsasw)
    CALL Satenthalpy(Tswi,spsatm,sisasw)
    ww1=swsasw-swsasa
    ii1=(sisasw-sisai)
    TT1=Tspo-Tswi
    www1=swsai-swsasa
    a1=-K1*ww1
    b1=K2*ww1
    c1=K2*((Lew*ii1-(Lew-1.0)*ww1*iv)
 +               +www1*Lew*scspw*Tswi/1000.0)
    d1=K3*Tswi*ww1+K4*TT1-K5*(c1/K2)*1000.0
    e1=K6*TT1

    CALL Airhumidity((Tswi+d1/2.0),(Tswi+d1/2),spsatm,swsasw)
    CALL Satenthalpy((Tswi+d1/2.0),spsatm,sisasw)
    ww2=swsasw-swsasa
    ii2=(sisasw-(sisai+c1/2.0))
    TT2=(Tspo+e1/2.0)-(Tswi+d1/2.0)
    www2=(swsai+b1/2.0)-swsasa
    a2=-K1*ww2
    b2=K2*ww2
    c2=K2*((Lew*ii2-(Lew-1.0)*ww2*iv)
 +                  +www2*Lew*scspw*(Tswi+d1/2.0)/1000.0)
    d2=K3*(Tswi+d1/2.0)*ww2+K4*TT2-K5*(c2/K2)*1000.0
    e2=K6*TT2

    CALL Airhumidity((Tswi+d2/2.0),(Tswi+d2/2),spsatm,swsasw)
    CALL Satenthalpy((Tswi+d2/2.0),spsatm,sisasw)
    ww3=swsasw-swsasa
    ii3=(sisasw-(sisai+c2/2.0))
    TT3=(Tspo+e2/2.0)-(Tswi+d2/2.0)
    www3=(swsai+b2/2.0)-swsasa
    a3=-K1*ww3
    b3=K2*ww3
    c3=K2*((Lew*ii3-(Lew-1.0)*ww3*iv)
 +                  +www3*Lew*scspw*(Tswi+d2/2.0)/1000.0)
    d3=K3*(Tswi+d2/2.0)*ww3+K4*TT3-K5*(c3/K2)*1000.0
    e3=K6*TT3

    CALL Airhumidity((Tswi+d3),(Tswi+d3),spsatm,swsasw)
```

```
        CALL Satenthalpy((Tswi+d3),spsatm,sisasw)
        ww4=swsasw-swsasa
        ii4=(sisasw-(sisai+c3))
        TT4=(Tspo+e3)-(Tswi+d3)
        www4=(swsai+b3)-swsasa
        a4=-K1*ww4
        b4=K2*ww4
        c4=K2*((Lew*ii4-(Lew-1.0)*ww4*iv)
     +               +www4*Lew*scspw*(Tswi+d3)/1000.0)
        d4=K3*(Tswi+d3)*ww4+K4*TT4-K5*(c4/K2)*1000.0
        e4=K6*TT4
      END IF
      smswo=smswi+(a1+2.0*(a2+a3)+a4)/6.0
      swsao=swsai+(b1+2.0*(b2+b3)+b4)/6.0
      sisao=sisai+(c1+2.0*(c2+c3)+c4)/6.0
      Tswo=Tswi+(d1+2.0*(d2+d3)+d4)/6.0
      Tspi=Tspo+(e1+2.0*(e2+e3)+e4)/6.0

C Determine the air outlet temperature and saturation enthalpy
      TR=Tspi
      TL=0.0
   10 Tsao=(TR+TL)/2.0
      CALL Cpv(Tsao,scspv)
      CALL Cpa(Tsao,scspa)
      CALL Cpw(Tsao,scspw)
      CALL Airhumidity(Tsao,Tsao,spsatm,swsasa)
      IF (swsasa.GT.swsao) THEN
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
      ELSE
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
     +         +scspw*(swsao-swsasa)*Tsao/1000.0
      END IF
      IF ((ABS(sisao-sisasa)).GT.0.1) THEN
        IF (sisao.LT.sisasa) THEN
          TR=Tsao
        ELSE
          TL=Tsao
        END IF
        GO TO 10
      END IF
      RETURN
      END

C *****************************************************************
C *                                                               *
C *         SUBROUTINE TO PRINT RESULTS OF CROSS.FOR              *
C *                                                               *
C *****************************************************************
C Subroutine to print the results of the cooler calculations
      SUBROUTINE PRINT_RESULTS(Tspi1,Tspo1,smsp,sdsi,sdso,vspas,hspas,
     +           Kmax,Lmax,Mmax,gamma,Vstot,rhosai1,Vseff2,sisai1,
```

```
        CALL Satenthalpy((Tswi+d3),spsatm,sisasw)
        ww4=swsasw-swsasa
        ii4=(sisasw-(sisai+c3))
        TT4=(Tspo+e3)-(Tswi+d3)
        www4=(swsai+b3)-swsasa
        a4=-K1*ww4
        b4=K2*ww4
        c4=K2*((Lew*ii4-(Lew-1.0)*ww4*iv)
     +                +www4*Lew*scspw*(Tswi+d3)/1000.0)
        d4=K3*(Tswi+d3)*ww4+K4*TT4-K5*(c4/K2)*1000.0
        e4=K6*TT4
      END IF
      smswo=smswi+(a1+2.0*(a2+a3)+a4)/6.0
      swsao=swsai+(b1+2.0*(b2+b3)+b4)/6.0
      sisao=sisai+(c1+2.0*(c2+c3)+c4)/6.0
      Tswo=Tswi+(d1+2.0*(d2+d3)+d4)/6.0
      Tspi=Tspo+(e1+2.0*(e2+e3)+e4)/6.0

C Determine the air outlet temperature and saturation enthalpy
      TR=Tspi
      TL=0.0
   10 Tsao=(TR+TL)/2.0
      CALL Cpv(Tsao,scspv)
      CALL Cpa(Tsao,scspa)
      CALL Cpw(Tsao,scspw)
      CALL Airhumidity(Tsao,Tsao,spsatm,swsasa)
      IF (swsasa.GT.swsao) THEN
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
      ELSE
        sisasa=scspa*Tsao/1000.0+swsasa*(2501.6+scspv*Tsao/1000.0)
     +          +scspw*(swsao-swsasa)*Tsao/1000.0
      END IF
      IF ((ABS(sisao-sisasa)).GT.0.1) THEN
        IF (sisao.LT.sisasa) THEN
          TR=Tsao
        ELSE
          TL=Tsao
        END IF
        GO TO 10
      END IF
      RETURN
      END


C ******************************************************************
C *                                                                *
C *          SUBROUTINE TO PRINT RESULTS OF CROSS.FOR              *
C *                                                                *
C ******************************************************************
C Subroutine to print the results of the cooler calculations
      SUBROUTINE PRINT_RESULTS(Tspil,Tspol,smsp,sdsi,sdso,vspas,hspas,
     +           Kmax,Lmax,Mmax,gamma,Vstot,rhosail,Vseff2,sisail,
```

```
     +          sisaol,Tswil,Tswol,svsp,flowlayout,H,L,spsatm,PI,Tsawb,
     +          Tsadb,svsa,swsail,swsaol,smswil,smswol,model,Tsaol,
     +          rhosaol,phio,Power,shsf1,shsf2,skst,smsa)

C Initialize variable types
      REAL L
      INTEGER flowlayout

C Print the results on the screen or in file CROSS.RES
C Print cooler layout and dimensions
      CALL LIB$ERASE_PAGE(1,1)
      WRITE(*,10)
      WRITE(1,10)
   10 FORMAT(
     +   '  CROSSFLOW EVAPORATIVE COOLER SIMULATION'/
     +   '******************************************')
C Print process water flow layout
      IF (flowlayout.EQ.1) THEN
        WRITE(*,*)'Process water flow layout  : FRONT TO BACK'
        WRITE(1,*)'Process water flow layout  : FRONT TO BACK'
      ELSE IF (flowlayout.EQ.2) THEN
        WRITE(*,*)'Process water flow layout  : BACK TO FRONT'
        WRITE(1,*)'Process water flow layout  : BACK TO FRONT'
      ELSE IF (flowlayout.EQ.3) THEN
        WRITE(*,*)'Process water flow layout  : TOP TO BOTTOM'
        WRITE(1,*)'Process water flow layout  : TOP TO BOTTOM'
      ELSE IF (flowlayout.EQ.4) THEN
        WRITE(*,*)'Process water flow layout  : STRAIGHT THROUGH'
        WRITE(1,*)'Process water flow layout  : STRAIGHT THROUGH'
      END IF
      IF (model.EQ.1) THEN
        WRITE(1,*)'Analytical model           : MERKEL'
        WRITE(*,*)'Analytical model           : MERKEL'
      ELSE IF (model.EQ.2) THEN
        WRITE(1,*)'Analytical model           : IMPROVED MERKEL'
        WRITE(*,*)'Analytical model           : IMPROVED MERKEL'
      ELSE
        WRITE(1,*)'Analytical model           : POPPE'
        WRITE(*,*)'Analytical model           : POPPE'
      END IF
      WRITE(1,11)sdso*1000.0,sdsi*1000.0,vspas*1000.0,hspas*1000.0,
     +          H,L,Kmax,Lmax,Mmax,shsf1,shsf2,skst
   11 FORMAT(/
     +   ' Pipe outer diameter ..................... = ',F6.2,' mm'/
     +   ' Pipe inner diameter ..................... = ',F6.2,' mm'/
     +   ' Vertical spacing between pipes .......... = ',F6.2,' mm'/
     +   ' Horizontal spacing between pipes ........ = ',F6.2,' mm'/
     +   ' Height of cooler unit ................... = ',F6.2,' m'/
     +   ' Width of cooler unit .................... = ',F6.2,' m'/
     +   ' Number of rows of pipes across airstream  = ',I3/
     +   ' Number of pipes facing the airstream .... = ',I3/
     +   ' Number of elements along a single pipe .. = ',I3/
```

```
     +' Fouling coefficient (inside) ............ = ',F12.2,' W/m^2 K'/
     +' Fouling coefficient (outside) ........... = ',F12.2,' W/m^2 K'/
     +' Pipe wall conductivity .................. = ',F8.2,' W/m K')

C Print ambient conditions and results in result file
      IF (model.EQ.1) THEN
        WRITE(1,20)spsatm/1000.0,Tsadb,Tsawb,rhosail,smsa,
     +             smsa*(1.0+swsail),svsa,sisail,sisaol,swsail
        WRITE(1,30)swsaol,Tsaol,rhosaol,gamma*3600.0,smswil,
     +             smswil-smswol,Tswil,Tswol
      ELSE IF (model.EQ.2) THEN
        WRITE(1,20)spsatm/1000.0,Tsadb,Tsawb,rhosail,smsa,
     +             smsa*(1.0+swsail),svsa,sisail,sisaol,swsail
        WRITE(1,40)swsaol,phio,Tsaol,rhosaol,gamma*3600.0,smswil,
     +             smswol,smswil-smswol,Tswil,Tswol
      ELSE
        WRITE(1,20)spsatm/1000.0,Tsadb,Tsawb,rhosail,smsa,
     +             smsa*(1.0+swsail),svsa,sisail,sisaol,swsail
        WRITE(1,40)swsaol,phio,Tsaol,rhosaol,gamma*3600.0,smswil,
     +             smswol,smswil-smswol,Tswil,Tswol
      END IF
      WRITE(1,50)smsp,svsp,Tspil,Tspol,Power

C Print the results on the screen
      WRITE(*,12)H,L,Lmax,Kmax,spsatm/1000.0,svsa,
     +           smsa,sisail,sisaol,smswil,smswil-smswol,
     +           Tswil,Tswol,smsp,Tspil,Tspol,Power
   12 FORMAT(
     + ' Height of cooler unit .................. = ',F8.3,' m'/
     + ' Length of cooler tubes ................. = ',F8.3,' m'/
     + ' Number of pipe rows along cooler height . = ',I3/
     + ' Number of pipes along the airflow ....... = ',I3/
     + ' Atmospheric pressure ................... = ',F8.3,' kPa'/
     + ' Air velocity through cooler ............ = ',F8.3,' m/s'/
     + ' Dry air massflow through cooler ......... = ',F8.3,' kg/s'/
     + ' Air enthalpy in ........................ = ',F8.3,' kJ/kg'/
     + ' Air enthalpy out (incl. mist) ........... = ',F8.3,' kJ/kg'/
     + ' Inlet recirculating water massflow ...... = ',F8.3,' kg/s'/
     + ' Recirculating water evaporated  ......... = ',F8.3,' kg/s'/
     + ' Recirculating water temperature (in) .... = ',F8.3,' C'/
     + ' Recirculating water temperature (out) ... = ',F8.3,' C'/
     + ' Process fluid massflow through cooler ... = ',F8.3,' kg/s'/
     + ' Process fluid temperature in ............ = ',F8.3,' C'/
     + ' Process fluid temperature out ........... = ',F8.3,' C'/
     + ' Total capacity of cooler unit ........... = ',F8.3,' kW'/)

   20 FORMAT(' ',/
     + ' Atmospheric pressure .................. = ',F8.3,' kPa'/
     + ' Inlet air temperature (dry bulb) ....... = ',F8.3,' C'/
     + ' Inlet air temperature (wet bulb) ....... = ',F8.3,' C'/
     + ' Inlet air density ..................... = ',F8.3,' kg/m^3'/
     + ' Dry air massflow through cooler ........ = ',F8.3,' kg/s'/
```

```
     + ' Inlet air massflow (inc vapour) ........ = ',F8.3,' kg/
     + ' Air velocity through cooler ............ = ',F8.3,' m/s
     + ' Air enthalpy in ........................ = ',F8.3,' kJ/
     + ' Air enthalpy out (incl. mist) .......... = ',F8.3,' kJ/
     + ' Inlet air humidity ratio ............... = ',F12.7,' kg/kg')

   30 FORMAT(
     + ' Outlet air humidity ratio (saturated) .. = ',F12.7,' kg/kg'/
     + ' Outlet air temperature (saturated) ..... = ',F8.3,' C'/
     + ' Outlet air density (saturated) ......... = ',F8.3,' kg/m^3'//
     + ' Recirc.water massflow / length ......... = ',F9.4,' kg/m hr'/
     + ' Inlet recirc.water massflow ............ = ',F9.4,' kg/s'/
     + ' Recirc. water lost through evaporation . = ',F9.4,' kg/s'/
     + ' Recirculating water temperature in ..... = ',F8.3,' C'/
     + ' Recirculating water temperature out .... = ',F8.3,' C')

   40 FORMAT(
     + ' Outlet air humidity ratio (incl. mist) . = ',F12.7,' kg/kg'/
     + ' Outlet air relative humidity ........... = ',F12.7/
     + ' Outlet air temperature (dry bulb) ...... = ',F8.3,' C'/
     + ' Outlet air density  .................... = ',F8.3,' kg/m^3'//
     + ' Recirc.water massflow / length ......... = ',F9.4,' kg/m hr'/
     + ' Inlet recirc.water massflow ............ = ',F9.4,' kg/s'/
     + ' Outlet recirc. water massflow .......... = ',F9.4,' kg/s'/
     + ' Recirc. water lost through evaporation . = ',F9.4,' kg/s'/
     + ' Recirculating water temperature in ..... = ',F8.3,' C'/
     + ' Recirculating water temperature out .... = ',F8.3,' C')

   50 FORMAT(' ',/
     + ' Process water massflow through cooler .. = ',F8.3,' kg/s'/
     + ' Process water flow velocity in pipes ... = ',F8.3,' m/s'/
     + ' Process water temperature in ........... = ',F8.3,' C'/
     + ' Process water temperature out .......... = ',F8.3,' C'/
     + ' Capacity of cooler unit ................ = ',F8.3,' kW'/)

      RETURN
      END

C ********************************************************************
C *                                                                  *
C *                 SINGLE STRAIGHT THROUGH PASS                     *
C *                                                                  *
C ********************************************************************
C Subroutine to evaluate a cooler layout where the process fluid flows
C straight through the cooler in one pass
      SUBROUTINE STRAIGHT (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,smspel,
     +           smswel,sisail,sisaol,Tspil,Tspol,Tswil,Tswol,swsail,
     +           swsaol,smswil,smswol,L,H,sdsi,sdso,dA,Tsadb,Tsawb,
     +           spsatm,gamma,Vstot,sa,skst,svsp,Aspi,Aspo,ReyC,
     +           gradfile,Kmax,Lmax,Mmax,PI,model,Tsaol,shsf1,shsf2)
```

```
      DIMENSION Tsp(40,400,10)
      DIMENSION Tsw(40,400,10)
      DIMENSION Tsa(40,400,10)
      DIMENSION sisa(40,400,10)
      DIMENSION swsa(40,400,10)
      DIMENSION smsw(40,400,10)

      REAL L
      INTEGER flag,flag2,gradfile

C Initialize the three arrays with the known temperature and enthalpy values
      CALL Enthalpy(Tsadb,Tsawb,spsatm,sisail)
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsail)
      DO 10 j=1,Lmax
        DO 20 k=1,Mmax
          sisa(1,j,k+1)=sisail
          swsa(1,j,k+1)=swsail
          Tsa(1,j,k+1)=Tsadb
   20   CONTINUE
   10 CONTINUE

      DO 30 i=1,Kmax
        DO 40 k=1,Mmax
          Tsw(i,1,k+1)=Tswil
          smsw(i,1,k+1)=smswel
   40   CONTINUE
   30 CONTINUE

      DO 50 i=1,Kmax
        DO 60 j=1,Lmax
          Tsp(i,j,1+1)=Tspil
   60   CONTINUE
   50 CONTINUE

C Start of the outer loop to evaluate each i-level of the model
      DO 70 i=1,Kmax
        flag2=i-2*INT(i/2.0)
C Flag2=1 in the first row,0 in the second row etc.

C Start of the middle loop to evaluate each j-level of the model
        DO 80 j=1,Lmax

C Start of the inner loop to evaluate each each element of the model
          DO 90 k=2,Mmax+1

C Determine the input values for a given element
                Tspi=Tsp(i,j,k)
                Tswi=Tsw(i,j,k)
                Tsai=Tsa(i,j,k)
                sisai=sisa(i,j,k)
                swsai=swsa(i,j,k)
                smswi=smsw(i,j,k)
```

```
C Determine the enthalpy of air entering each element in the packed formation
              IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
                 IF (j.EQ.1) THEN
                   sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                   swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                   Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
                 ELSE
                   sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
                   swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
                   Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
                 END IF
              END IF
              IF (flag2.EQ.0) THEN
                 IF (j.EQ.Lmax) THEN
                   sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
                   swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
                   Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
                 ELSE
                   sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
                   swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
                   Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
                 END IF
              END IF

C Call subroutine to determine outlet conditions of each element
              IF (model.EQ.1) THEN
                CALL MERKEL (Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +                Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +                sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +                ReyC,shsfl,shsf2,1)
              ELSE IF (model.EQ.2) THEN
                CALL IMPMERKEL (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +                Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +                sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +                ReyC,smswi,smswo,Tsai,Tsao,shsfl,shsf2,1)
              ELSE
                CALL POPPE (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +                sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +                smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +                smswo,Aspi,Aspo,ReyC,shsfl,shsf2,1)
              END IF

C Determine the exit values for a given element
              Tsp(i,j,k+1)=Tspo
              Tsw(i,j+1,k)=Tswo
              Tsa(i+1,j,k)=Tsao
              sisa(i+1,j,k)=sisao
              swsa(i+1,j,k)=swsao
              smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
```

```
            IF (gradfile.EQ.1) THEN
              WRITE(4,*)i,j,k
              WRITE(4,*)Tspi,Tspo
              WRITE(4,*)Tswi,Tswo
              WRITE(4,*)sisai,sisao
              IF (model.NE.1) THEN
                WRITE(4,*)swsai,swsao
                WRITE(4,*)smswi,smswo
                IF (model.EQ.3) THEN
                  WRITE(4,*)Tswi,Tswo
                END IF
              END IF
            END IF
   90     CONTINUE
   80   CONTINUE
   70 CONTINUE

C Determine the average exit temperature of recirculating water
      sum1=0.0
      sum2=0.0
      DO 100 i=1,Kmax
        DO 110 k=1,Mmax
          CALL Cpw(Tsw(i,Lmax+1,k+1),scspw)
          sum1=sum1+Tsw(i,Lmax+1,k+1)*scspw
          sum2=sum2+smsw(i,Lmax+1,k+1)
  110   CONTINUE
  100 CONTINUE
      CALL Cpw(Tswi1,scspw)
      Tswo1=sum1/(Mmax*Kmax*scspw)
      smswo1=sum2

C Determine the average exit temperature of process water
      sum1=0.0
      sum2=0.0
      DO 120 i=1,Kmax
        DO 130 j=1,Lmax
          CALL Cpw(Tsp(i,j,Mmax+1+1),scspp)
          sum1=sum1+Tsp(i,j,Mmax+1+1)*scspp
          sum2=sum2+Tsp(i,j,Mmax+1+1)
  130   CONTINUE
  120 CONTINUE
      CALL Cpw((sum2/(Kmax*Lmax)),scspp)
      Tspo1=sum1/(Kmax*Lmax*scspp)

C Determine the average exit enthalpy of the air
      sum1=0.0
      sum2=0.0
      sum3=0.0
      DO 140 j=1,Lmax
        DO 150 k=1,Mmax
          sum1=sum1+sisa(Kmax+1,j,k+1)
          sum2=sum2+swsa(Kmax+1,j,k+1)
```

```
         sum3=sum3+Tsa(Kmax+1,j,k+1)
  150   CONTINUE
  140 CONTINUE
      DO 160 k=1,Mmax
        IF (flag2.EQ.0) THEN
          sum1=sum1+sisa(Kmax,1,k+1)/2.0
          sum2=sum2+swsa(Kmax,1,k+1)/2.0
          sum3=sum3+Tsa(Kmax,1,k+1)/2.0
        ELSE
          sum1=sum1+sisa(Kmax,Lmax,k+1)/2.0
          sum2=sum2+swsa(Kmax,Lmax,k+1)/2.0
          sum3=sum3+Tsa(Kmax,Lmax,k+1)/2.0
        END IF
  160 CONTINUE
      sisaol=sum1/(Mmax*(Lmax+.5))
      swsaol=sum2/(Mmax*(Lmax+.5))
      Tsaol=sum3/(Mmax*(Lmax+.5))

C Print the recirc.water inlet and outlet temperatures on the screen
      WRITE(*,170)Tswil,Tswol
  170 FORMAT(' ','Tw(in) = ',F7.3,'      Tw(out) = ',F7.3)
      RETURN
      END


C ******************************************************************
C *                                                                *
C *              TOP TO BOTTOM PROCESS WATER FLOW                  *
C *                                                                *
C ******************************************************************
C Subroutine to evaluate a cooler layout where the process fluid flows
C downwards in a direction perpendicular to the direction of the airstream
      SUBROUTINE TOPTOBOTTOM (Tsp,Tsw,Tsa,sisa,swsa,smsw,smsael,
     +           smspel,smswel,sisail,sisaol,Tspil,Tspol,Tswil,
     +           Tswol,swsail,swsaol,smswil,smswol,L,H,sdsi,sdso,
     +           dA,Tsadb,Tsawb,spsatm,gamma,Vstot,sa,skst,
     +           svsp,Aspi,Aspo,ReyC,gradfile,Kmax,Lmax,Mmax,PI,
     +           model,Tsaol,shsf1,shsf2,gradplot)

      DIMENSION Tsp(40,400,10)
      DIMENSION Tsw(40,400,10)
      DIMENSION Tsa(40,400,10)
      DIMENSION sisa(40,400,10)
      DIMENSION swsa(40,400,10)
      DIMENSION smsw(40,400,10)

      REAL L
      INTEGER flag,flag2,gradfile,gradplot

C Initialize the arrays with the known values
      CALL Enthalpy(Tsadb,Tsawb,spsatm,sisail)
      CALL Airhumidity(Tsadb,Tsawb,spsatm,swsail)
```

```
      DO 20 j=1,Lmax
        DO 10 k=2,Mmax+1
          sisa(1,j,k)=sisail
          swsa(1,j,k)=swsail
          Tsa(1,j,k)=Tsadb
   10   CONTINUE
   20 CONTINUE

      DO 40 i=1,Kmax
        DO 30 k=2,Mmax+1
          Tsw(i,1,k)=Tswil
          smsw(i,1,k)=smswel
   30   CONTINUE
   40 CONTINUE

      DO 50 i=1,Kmax
        Tsp(i,1,2)=Tspil
   50 CONTINUE

C Start of the outer loop to evaluate each i-level of the model
      DO 60 i=1,Kmax
        flag=0
        flag2=i-2*INT(i/2.0)
C Flag2=1 in the first row,0 in the second row etc.

C Start of the middle loop to evaluate each j-level of the model
        DO 70 j=1,Lmax

C N.B. flag=1 for backward process fluid flow
C L.W. flag=0 for forward process fluid flow
          IF (flag.EQ.0) THEN

C Start of the inner loop to evaluate each each element of the model
C Process water flow is in a forward direction
            DO 80 k=2,Mmax+1

C Determine the inlet values for a given element
              Tspi=Tsp(i,j,k)
              IF((k.EQ.2).AND.(j.NE.1)) Tspi=Tsp(i,j-1,k-1)
              Tswi=Tsw(i,j,k)
              Tsai=Tsa(i,j,k)
              sisai=sisa(i,j,k)
              swsai=swsa(i,j,k)
              smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
              IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
                 IF (j.EQ.1) THEN
                   sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
                   swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
                   Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
                 ELSE
```

```
                sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
              END IF
           END IF
           IF (flag2.EQ.0) THEN
              IF (j.EQ.Lmax) THEN
                sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
              ELSE
                sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
                swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
                Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
              END IF
           END IF

C Call subroutine to determine outlet conditions of each element
           IF (model.EQ.1) THEN
             CALL MERKEL (Tspi,Tswi,sisai,swsail,L,H,sdsi,sdso,dA,
     +            Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +            sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +            ReyC,shsf1,shsf2,Lmax)
           ELSE IF (model.EQ.2) THEN
             CALL IMPMERKEL (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +            Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +            sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +            ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Lmax)
           ELSE
             CALL POPPE (Tspi,Tswi,Tsa,sisai,swsai,smswi,L,H,sdsi,
     +            sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +            smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +            smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Lmax)
           END IF

C Determine the exit values for a given element
           Tsp(i,j,k+1)=Tspo
           Tsw(i,j+1,k)=Tswo
           Tsa(i+1,j,k)=Tsao
           sisa(i+1,j,k)=sisao
           swsa(i+1,j,k)=swsao
           smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
           IF (gradfile.EQ.1) THEN
             WRITE(4,*)i,j,k-1
             WRITE(4,*)Tspi,Tspo
             WRITE(4,*)Tswi,Tswo
             WRITE(4,*)sisai,sisao
             IF (model.NE.1) THEN
               WRITE(4,*)swsai,swsao
               WRITE(4,*)smswi,smswo
```

```
              IF (model.EQ.3) THEN
                WRITE(4,*)Tswi,Tswo
              END IF
            END IF
          ELSE IF (gradplot.EQ.1) THEN
            IF (((i.EQ.1).OR.(i.EQ.10)).AND.(k.EQ.2)) THEN
              WRITE(5,3)i,j,k-1,Tspi,Tspo,Tswi,Tswo,sisai,sisao
 3            FORMAT(3I4,6F9.3)
            END IF
          END IF
80      CONTINUE
        flag=1

      ELSE IF (flag.EQ.1) THEN
C Start of the inner loop to evaluate each each element of the model
C Process water flow is backwards to the origin
        DO 90 k=Mmax+1,2,-1

C Determine the inlet values for a given element
          Tspi=Tsp(i,j,k)
          IF (k.EQ.(Mmax+1)) Tspi=Tsp(i,j-1,k+1)
          Tswi=Tsw(i,j,k)
          Tsai=Tsa(i,j,k)
          sisai=sisa(i,j,k)
          swsai=swsa(i,j,k)
          smswi=smsw(i,j,k)

C Determine the enthalpy of air entering each element in the packed formation
          IF ((flag2.EQ.1).AND.(i.NE.1)) THEN
             IF (j.EQ.1) THEN
               sisai=(sisa(i-1,j,k)+sisa(i,j,k))/2.0
               swsai=(swsa(i-1,j,k)+swsa(i,j,k))/2.0
               Tsai=(Tsa(i-1,j,k)+Tsa(i,j,k))/2.0
             ELSE
               sisai=(sisa(i,j,k)+sisa(i,j-1,k))/2.0
               swsai=(swsa(i,j,k)+swsa(i,j-1,k))/2.0
               Tsai=(Tsa(i,j,k)+Tsa(i,j-1,k))/2.0
             END IF
          END IF
          IF (flag2.EQ.0) THEN
             IF (j.EQ.Lmax) THEN
               sisai=(sisa(i,j,k)+sisa(i-1,j,k))/2.0
               swsai=(swsa(i,j,k)+swsa(i-1,j,k))/2.0
               Tsai=(Tsa(i,j,k)+Tsa(i-1,j,k))/2.0
             ELSE
               sisai=(sisa(i,j,k)+sisa(i,j+1,k))/2.0
               swsai=(swsa(i,j,k)+swsa(i,j+1,k))/2.0
               Tsai=(Tsa(i,j,k)+Tsa(i,j+1,k))/2.0
             END IF
          END IF

C Call subroutine to determine outlet conditions of each element
```

```
          IF (model.EQ.1) THEN
            CALL MERKEL (Tspi,Tswi,sisai,swsai1,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,Aspi,Aspo,
     +           ReyC,shsf1,shsf2,Lmax)
          ELSE IF (model.EQ.2) THEN
            CALL IMPMERKEL (Tspi,Tswi,sisai,swsai,L,H,sdsi,sdso,dA,
     +           Tsadb,spsatm,gamma,Vstot,smsael,smspel,smswel,
     +           sa,skst,svsp,Tspo,Tswo,sisao,swsao,Aspi,Aspo,
     +           ReyC,smswi,smswo,Tsai,Tsao,shsf1,shsf2,Lmax)
          ELSE
            CALL POPPE (Tspi,Tswi,Tsai,sisai,swsai,smswi,L,H,sdsi,
     +           sdso,dA,Tsadb,spsatm,gamma,Vstot,smsael,smspel,
     +           smswel,sa,skst,svsp,Tspo,Tswo,Tsao,sisao,swsao,
     +           smswo,Aspi,Aspo,ReyC,shsf1,shsf2,Lmax)
          END IF

C Determine the exit values for a given element
          Tsp(i,j,k-1)=Tspo
          Tsw(i,j+1,k)=Tswo
          Tsa(i+1,j,k)=Tsao
          sisa(i+1,j,k)=sisao
          swsa(i+1,j,k)=swsao
          smsw(i,j+1,k)=smswo

C Write the temperature and enthalpy gradients to file CROSS.GRA
          IF (gradfile.EQ.1) THEN
            WRITE(4,*)i,j,k-1
            WRITE(4,*)Tspi,Tspo
            WRITE(4,*)Tswi,Tswo
            WRITE(4,*)sisai,sisao
            IF (model.NE.1) THEN
              WRITE(4,*)swsai,swsao
              WRITE(4,*)smswi,smswo
              IF (model.EQ.3) THEN
                WRITE(4,*)Tswi,Tswo
              END IF
            END IF
          ELSE IF (gradplot.EQ.1) THEN
            IF (((i.EQ.1).OR.(i.EQ.10)).AND.(k.EQ.2)) THEN
              WRITE(5,4)i,j,k-1,Tspi,Tspo,Tswi,Tswo,sisai,sisao
 4            FORMAT(3I4,6F9.3)
            END IF
          END IF
 90     CONTINUE
        flag=0
      END IF
 70   CONTINUE
 60 CONTINUE

C Determine the average exit temperature of recirculating water
      sum1=0.0
```

```
      sum2=0.0
      DO 110 i=1,Kmax
        DO 100 k=2,Mmax+1
          CALL Cpw(Tsw(i,Lmax+1,k),scspw)
          sum1=sum1+Tsw(i,Lmax+1,k)*scspw
          sum2=sum2+smsw(i,Lmax+1,k)
  100   CONTINUE
  110 CONTINUE
      CALL Cpw(Tswi1,scspw)
      Tswo1=sum1/(Mmax*Kmax*scspw)
      smswo1=sum2

C Determine the average exit temperature of process water
      sum1=0.0
      sum2=0.0
      rem=Mmax+2
      IF (flag.EQ.0) rem=1
      DO 120 i=1,Kmax
        CALL Cpw(Tsp(i,Lmax,rem),scspp)
        sum1=sum1+Tsp(i,Lmax,rem)*scspp
        sum2=sum2+Tsp(i,Lmax,rem)
  120 CONTINUE
      CALL Cpw((sum2/Kmax),scspp)
      Tspo1=sum1/(Kmax*scspp)

C Determine the average exit enthalpy of the air
      sum1=0.0
      sum2=0.0
      sum3=0.0
      DO 130 j=1,Lmax
        DO 130 k=2,Mmax+1
          sum1=sum1+sisa(Kmax+1,j,k)
          sum2=sum2+swsa(Kmax+1,j,k)
          sum3=sum3+Tsa(Kmax+1,j,k)
  130   CONTINUE
  140 CONTINUE
      DO 150 k=2,Mmax+1
        IF (flag2.EQ.0) THEN
          sum1=sum1+sisa(Kmax,1,k)/2.0
          sum2=sum2+swsa(Kmax,1,k)/2.0
          sum3=sum3+Tsa(Kmax,1,k)/2.0
        ELSE
          sum1=sum1+sisa(Kmax,Lmax,k)/2.0
          sum2=sum2+swsa(Kmax,Lmax,k)/2.0
          sum3=sum3+Tsa(Kmax,Lmax,k)/2.0
        END IF
  150 CONTINUE
      sisao1=sum1/(Mmax*(Lmax+.5))
      swsao1=sum2/(Mmax*(Lmax+.5))
      Tsao1=sum3/(Mmax*(Lmax+.5))

C Print the recirc.water inlet and outlet temperatures on the screen
```

```
      WRITE(*,160)Tswi1,Tswo1
  160 FORMAT(' ','Tw(in) = ',F7.3,'      Tw(out) = ',F7.3)
      RETURN
      END

C    ********************************************************************
C    *                                                                  *
C    *      THERMOPHYSICAL PROPERTIES OF AIR, WATER, WATER-VAPOUR       *
C    *                     AND AIR WATER MIXTURES                       *
C    *                                                                  *
C    ********************************************************************

C Subroutine to calculate the saturation vapour-pressure of water
      SUBROUTINE Satvappressure(t1,spssat)
      T=t1+273.16
      a=1.079586E1
      b=5.02808
      c=1.50474E-4
      d=-8.29692
      e=4.2873E-4
      f=4.76955
      g=2.786118312
      x=273.16/T
      z=a*(1-x)+b*LOG10(x)+c*(1-10**(d*((1/x)-1)))+e*(10**(f*(1-x))-1)+g
      spssat=10**z
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the specific heat of water-vapour
      SUBROUTINE Cpv(t1,scspv)
      T=t1+273.16
      a=1.3605E3
      b=2.31334
      c=-2.46784E-10
      d=5.91332E-13
      scspv=a+b*T+c*T**5+d*T**6
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the specific heat of air
      SUBROUTINE Cpa(t1,scspa)
      T=t1+273.16
      a=1.045356E3
      b=-3.161783E-1
      c=7.083814E-4
      d=-2.705209E-7
      scspa=a+b*T+c*T**2+d*T**3
      RETURN
      END

C *******************************************************************
```

```
C Subroutine to calculate the specific heat of water
      SUBROUTINE Cpw(tl,scspw)
      T=tl+273.16
      a=8.15599E3
      b=-2.80627E1
      c=5.11283E-2
      d=-2.17582E-13
      scspw=a+b*T+c*T**2+d*T**6
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the saturation enthalpy of air
      SUBROUTINE Satenthalpy(tt2,spsatm,sissat)
      CALL Satvappressure(tt2,spssw2)
      CALL Cpv(tt2,scspv2)
      CALL Cpa(tt2,scspa2)
      swsa=((0.62198)*1.005*spssw2)/(spsatm-(1.005*spssw2))
      sisvap=swsa*(2501.6+(scspv2*tt2/1000))
      sisdryair=scspa2*tt2/(1000)
      sissat=sisdryair+sisvap
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the enthalpy of air using the wb and db temps.
      SUBROUTINE Enthalpy(ttl,tt2,spsatm,sisa)
      CALL Cpv(ttl,scspv2)
      CALL Cpa(ttl,scspa2)
      CALL Airhumidity(ttl,tt2,spsatm,swsa2)
      sisvap=swsa2*(2501.6+((scspv2*ttl)/(1000)))
      sisdryair=(scspa2*ttl)/(1000)
      sisa=sisdryair+sisvap
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the humidity of air
      SUBROUTINE Airhumidity(tttl,ttt2,spsatm,swsal)
      CALL Satvappressure(ttt2,spssw2)
      swsas=((0.62198*1.005*spssw2))/(spsatm-(1.005*spssw2))
      q0=(2501.6-(2.3263*ttt2))*swsas
      ql=1.00416*(tttl-ttt2)
      q2=(2501.6+(1.8577*tttl)-(4.184*ttt2))
      swsal=(q0-ql)/q2
      RETURN
      END

C *******************************************************************
```

```
C Subroutine to calculate the dynamic viscosity of air
      SUBROUTINE Airviscosity(t1,musa)
      REAL musa
      T=t1+273.16
      a=2.287973E-6
      b=6.259793E-8
      c=-3.131956E-11
      d=8.15038E-15
      musa=a+b*T+c*T**2+d*T**3
      RETURN
      END

C ******************************************************************

C Subroutine to calculate the dynamic viscosity of water
      SUBROUTINE Waterviscosity(t1,musw)
      REAL musw
      T=t1+273.16
      a=2.414E-5
      b=247.8
      c=140
      musw=a*10**(b/(T-c))
      RETURN
      END

C ******************************************************************

C Subroutine to calculate the dynamic viscosity of water vapour
      SUBROUTINE Vapourviscosity(t1,musv)
      REAL musv
      T=t1+273.16
      a=2.562435E-6
      b=1.816683E-8
      c=2.579066E-11
      d=-1.067299E-14
      musv=a+b*T+c*T**2+d*T**3
      RETURN
      END

C ******************************************************************

C Subroutine to calculate the dynamic viscosity of air/water vaopur mix
      SUBROUTINE AirVapMixviscosity(t2,swsal,spsatm,musav)
      REAL musav,musa,musv
      T=t2+273.16
      xa=1.0*5.3824/(1.0+1.608*swsal)
      xv=swsal*4.2445/(swsal+0.622)
      CALL Airviscosity(t2,musa)
      CALL Vapourviscosity(t2,musv)
      musav=(xa*musa+xv*musv)/(xa+xv)
      RETURN
      END

C ******************************************************************
```

```
C Subroutine to calculate water-density
      SUBROUTINE Waterdensity(t1,rhosw)
      T=t1+273.16
      a=1.49343E-3
      b=-3.7164E-6
      c=7.09782E-9
      d=-1.90321E-20
      rhosw=(a+b*T+c*T**2+d*T**6)**(-1)
      RETURN
      END

C *******************************************************************

C Subroutine to calculate air-density
      SUBROUTINE Airdensity(t1,spsatm,rhosa)
      T=t1+273.16
      rhosa=spsatm/(287.08*T)
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the density of an air/water vapour mix
      SUBROUTINE AirVapMixdensity(t2,swsal,spsatm,rhosav)
      T=t2+273.16
      CALL Airdensity(t2,spsatm,rhosa)
      rhosav=(1.0+swsal)*(1.0-swsal/(swsal+0.62198))*rhosa
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the conductivity of water
      SUBROUTINE Waterconductivity(t1,sksw)
      T=t1+273.16
      a=-6.14255E-1
      b=6.9962E-3
      c=-1.01075E-5
      d=4.74737E-12
      sksw=a+b*T+c*T**2+d*T**4
      RETURN
      END

C *******************************************************************

C Subroutine to calculate the Prandtl-number of water
      SUBROUTINE Prandtl(t3,Pra)
      REAL musw2
      CALL Waterconductivity(t3,sksw2)
      CALL Waterviscosity(t3,musw2)
      CALL Cpw(t3,scspp2)
```

```
      Pra=scspp2*musw2/sksw2
      RETURN
      END

C ******************************************************************* 
```