

**PREDICTION AND MEASUREMENT OF THE PERFORMANCE OF SPRAY COOLED
HEAT EXCHANGERS**

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.

(Signature of candidate)

day of

1991

SYNOPSIS

In the present study various mathematical models have been developed for the analysis of spray cooled finned-tube heat exchangers. These models range from simplified models based on the work by Kreid et al. [78KR1], Nakayama et al. [88NA1] and Erens et al. [90ER1] to a more comprehensive model based on the work by Poppe [84PO1].

Computer programs were written to evaluate the performance prediction of these models on spray cooled heat exchangers in the vertical air flow mode.

Experimental tests were conducted on a four-row finned-tube heat exchanger in a vertical air/water mist flow to verify the mathematical models and it was found that the performance of this heat exchanger could be predicted within 20 per cent using the accurate model. Significant performance enhancement (up to 3.5 times the dry performance) was found by spraying relatively small amounts of water onto the heat exchanger. The spray water massflow rate was found to have a significant effect on the two phase pressure drop across the heat exchanger.

The present study indicated certain important factors which have to be taken into consideration when designing spray cooled heat exchanger units. These include the geometry of the finned-tubes, the temperature difference between the process water and the optimum air and the air/spray water ratio.

SINOPSIS

In die tesis is 'n aantal wiskundige modelle ontwikkel wat gebruik kan word vir die analise van sproei verkoelde vinbuis warmte uitruilers. Die wiskundige modelle wissel van eenvoudige modelle wat gebaseer is op die werk van Kreid et al. [78KR1], Nakayama et al. [88NA1] en Erens et al. [90ER1] tot 'n omvattende model wat gebaseer is op die werk van Poppe [84PO1].

Rekenaar programme is geskryf om die dié modelle se voorspellingsvermoë van die kapasiteit van sproei verkoelde warmte uitruilers te evalueer.

'n Eksperimentele ondersoek is gedoen op 'n warmte uitruiler met vier vinbuis rye in 'n toetsseksie met vertikale lugvloei. Die toets resultate is gebruik vir die evaluasie van die wiskundige modelle en het getoon dat die akkurate model se voorspellings binne 20 per sent van die gemete kapasiteit van die uitruiler val. Die eksperimentele werk het verder aangetoon dat deur klein hoeveelhede water op die uitruiler te sproei, 'n toename in kapasiteit tot 3.5 keer die droë werkverrigting moontlik is. Die twee-fase drukval oor die uitruiler is sterk beïnvloed deur die sproei water massavloei.

Die analitiese navorsing wat gedoen is met behulp van die rekenaar programme het aangedui dat daar 'n paar belangrike faktore is wat in ag geneem moet word met die ontwerp van sproei verkoelde warmte uitruilers. Die faktore sluit die vinbuis geometrie, die temperatuur verskil tussen die proses water en lug en optimum lug/sproeiwater massavloei verhouding in.

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NOMENCLATURE

A	Area, [m ²]
A _c	Free flow area, [m ²]
A _{fr}	Frontal area, [m ²]
A _{ev}	Effective wet area ratio, [-]
Bi	Biot number, [-]
c _p	Specific heat at constant pressure, [J/kgK]
d	Diameter (characteristic length), [m]
F _{enh}	Enhancement ratio, [-]
F _T	Correction factor defined in Appendix B, [-]
f	Friction factor, [-]
G	Mass velocity, [kg/s]
g	Gravitational acceleration, [m/s ²]
g _c	Force conversion factor, [g _c = 1 kg·m/N·s ²]
h	Heat transfer coefficient, [W/m ² K]
h _D	Mass transfer coefficient, [kg/m ² s]
h _f	Film heat transfer coefficient, [W/m ² K]
h _a [*]	Heat transfer coefficient defined by eq. (4.2.9), [W/m ² K]
Δi	Enthalpy difference [J/kg]
i	Enthalpy, [J/kg]
i _{fg}	Enthalpy of vaporization, [J/kg]
k	Thermal conductivity, [W/mK]
L	Length, [m]
Le	Lewis number, [-]
Le _f	Lewis factor, [-]
m	Mass flow rate, [kg/s]
M	Mass flux, [kg/m ² /s]
n	Number, [-]
Nu	Nusselt number, [-]
Δp	Pressure drop, [N/m ²]
p	Pressure, [N/m ²]
Pr	Prandtl number, [-]
q	Heat transfer rate, [W]
Re	Reynolds number based on Tube diameter, [-]

Re_H	Reynolds number based on hydraulic diameter, [-]
St	Stanton number, [-]
s	Spacing, [m]
ΔT	Temperature difference, [$^{\circ}C$]
T	Temperature, [$^{\circ}C$] or [K]
t	Thickness, [m] or time, [sec]
U	Overall heat transfer coefficient, [W/m^2K]
U^*	Overall heat transfer coefficient defined by equation (4.2.7), [W/m^2K]
v	Velocity, [m/s]
v_{∞}	Free stream velocity, [m/s]
v^*	Flow reversal parameter defined by eq. (3.2.6)
w	Humidity ratio, [kg water/kg dry air]
x	Height dimension, [m]
δ	Film thickness, [m]
Γ	Recirculating water massflow rate per side per unit length of tube, [kg/ms]
μ	Dynamic viscosity, [kg/ms]
ϵ_f	Fin efficiency, [-]
ϵ_o	Effectiveness of finned surface, [-]
θ	Angle, [rad]
ξ	Transformation parameter defined by eq. (4.2.8), [-]
λ	Factor defined by eq. (4.2.12) and (4.2.13), [-]
Ψ	Factor defined by eq. (4.2.15)
Φ	Two-phase multiplier, [-]
ϕ	Relative humidity of air, [-]
ϕ_D	Dryness parameter defined by equation (2.2.1), [-]
η_i	Factor defined by equations (7.1.5) to (7.1.7), [-]

Subscripts

a	Air
app	Apparent
as	Saturated air
asw	Air saturated at bulk (spray) water temperature
c	Convective or convection
d	Dripping water (due to gravity)
db	Dry bulb
dry	Dry part of surface (not covered by water film)
ev	Evaporated
f	Fouling or fin
i	Inlet or inside
lm	Log mean
m	Moist
max	Maximum
o	Outlet or outside (overall)
p	Process fluid (water) or primary
pa	Process fluid to air
pw	Process fluid to (spray) water film
r	Root
s	Secondary
SS	Steady state
SP	Single phase
t	Tube
tr	Tube rows
TP	Two-phase
w	Water film
sw	Spray water
wa	Water film to air
wb	Wet bulb
wet	Wetted by water film

1. INTRODUCTION

For the last century dry surface heat exchangers have been used extensively in industry, but with the recent emphasis on effective and compact heat exchangers, wet surface heat exchangers are receiving more attention.

In an evaporative cooling process the lowest theoretical recool temperature is the ambient wet bulb temperature whereas in dry surface heat exchangers it is the ambient dry bulb temperature. This factor and the combined mass and heat transfer process of evaporative cooling gives the wet surface heat exchanger the advantage of a larger available driving force and lower heat transfer resistance. The main disadvantages of the evaporative cooler is the loss of cooling water to the atmosphere and problems which arise through the deposition of salts on the heat exchanger surface and the consequent possibility of corrosion.

The loss of large quantities of water during the cooling process can be a decisive factor for industries in arid areas where the larger and more expensive dry coolers may well pay off in the long run. These are also the areas most likely to experience extreme conditions of ambient temperature variation which might lead to under-performance of dry coolers. This problem can be solved by spraying small amounts of water onto the heat exchanger surface to create a larger driving force and improved performance of the heat exchanger during peak loads and high temperature periods.

The spray cooled heat exchanger thus has the advantage of increased cooling capacity during extreme ambient conditions and it also minimizes water loss by using the large heat transfer surface of the dry cooler when the potential for dry cooling is large enough. It is thus more compact and can be cheaper than the conventional dry surface heat exchanger.

The combination of wet and dry surface cooling leads to a whole new field of heat exchanger design. To determine the best heat exchanger design to satisfy both the economic and geometric needs of the consumer, we need to know how the spray cooled heat exchanger performs in terms of capacity and running costs. This calls for accurate modelling of the behaviour of the various types of coolers and the systems in which they are used.

In this report the results of experimental work on a vertical air-flow spray cooled finned-tube heat exchanger are presented together with analytical models for the evaluation of vertical

air-flow spray coolers. These models have been incorporated into computer programs and their performance evaluated with the aid of the experimental results obtained during the testing of a commercial heat exchanger.

The physical behaviour of the vertical air-flow spray cooled heat exchanger is examined experimentally using the experimental finned-tube model and also analytically with the help of the computer programs. By having a better understanding of the physical behaviour of the spray cooled heat exchanger, more effective planning and design of spray cooled installations is possible.

2. LITERATURE SURVEY

Evaporative cooling phenomena have been the subject of study for many researchers, but the idea of combining evaporative cooling and dry air cooling has attracted attention more recently. Experimental work to study the effect which the injection of spray water or atomized water into the air stream has on the heat transfer between the surface and the air flowing over smooth tubes, tube banks or flat plates, substantiates the promising results predicted by analytical studies.

Analytical simulation of practical applications for spray cooling in conjunction with finned-tube banks have not received much attention because there are such a vast number of different configurations which complicates the derivation of an effective and accurate model which can be used for a majority of industrial configurations. In addition the probability of surface corrosion, a result of spray cooling, is not accounted for in the design of industrial finned-tube heat exchangers.

2.1 Theoretical work

Harris [64HA1] is one of the first authors who described the advantages of the wetted-fin evaporative cooler above other arrangements such as the humidification of the inlet air with the use of spray water or evaporative cooling followed by demisting of the air. He proposes that the unit operates by evaporation in hot weather and as a dry cooler in cooler weather.

The main advantage given is the compactness of this cooler as it combines evaporative- and air-cooling into one section. This combination has a lower discharge air temperature and thus also a smaller effective log-mean-temperature difference. As a result the same amount of cooling can be achieved with less heat exchanger surface. Experimental work by Harris indicates that the heat transfer rate of dry coolers can be increased up to four times by the implementation of spray cooling.

M.E. Goldstein, W-J. Yang and J.A. Clark [67GO1] present a theoretical model for the prediction of the heat transfer and friction characteristics in a two phase flow over a single cylinder.

This analytical study is formulated within the framework of laminar boundary layer theory with the assumption of a stable liquid film on the cylinder between the forward stagnation

point and the point of separation. The effects of gravity, dissipation heating, surface tension, waves on the liquid surface, liquid splashing and evaporation are neglected.

No correlations are presented for the calculation of heat transfer coefficients. In order to determine the local heat transfer coefficients for a specific setup the governing equations have to be solved numerically.

Based on previous experimental work in this field done by them, J.W. Hodgson and J.E. Sunderland [68HO2] present an analytical model for the local heat transfer coefficient on the periphery of an isothermal smooth cylinder exposed to mist flow.

A wall temperature lower than saturation temperature of the liquid phase is assumed as well as the existence of a laminar liquid film on the leading 160° of the cylinder. The flow is considered incompressible and assumptions of negligible evaporation and straight droplet trajectories are made. The effects of gravity, dissipation heating, surface tension of the liquid layer, waves on the liquid layer and liquid splashing are neglected by this analytical model.

With the above assumptions made, an analytical correlation for the local two-phase Nusselt number is obtained which does not require any numerical integration or differentiation, although the evaluation of some series terms are required. Good agreement is found with previous experimental work by the same authors. However, at low water/air ratios the analytical model underestimates the heat transfer. The rate of film evaporation is decreased by the influx of cold liquid droplets and this effect increases at higher liquid-to-air mass flow ratios. This underestimation of the heat transfer at low water/air ratios might be the result of the model's ignorance of the effect of film evaporation on the heat exchanger surface.

The influence of bouncing and splashing of the spray water droplets on evaporation and convection at the free surface of the liquid layer on a cylinder in an air-water mist flow were examined by I.C. Finlay [71FI1].

For effective modelling the following assumptions are made: (i) The liquid film on the cylinder is laminar. (ii) The flow is considered incompressible and droplet trajectories are straight. (iii) The heat transfer from the tube is influenced by the heat transport by the liquid film and convection and evaporation from the free surface. (iv) The effects of gravity,

dissipation heating, surface tension of the liquid layer and waves on the liquid layer are negligible. (v) The effect of liquid splashing is taken into account by a simple ratio, β , which determines the amount of liquid which stays in the liquid film and the amount which splashes from the film surface. (vi) Steady state conditions prevail and flow in the radial and axial directions are negligible.

The results of his analytical investigation show that the film thickness is almost constant up to 60° from the forward stagnation point and thereafter increases rapidly as the shearing stress becomes smaller. With $\beta = 1$ good agreement between the analytical and experimental heat transfer coefficient values, [67FI2], is found at air velocities ranging from 23 to 34 m/s. At higher air velocities and high liquid loadings the theoretical model overestimates the heat transfer coefficient. This overestimation of the local Nusselt number is the result of the assumption that liquid displaced from the liquid film by the splashing action of the impinging droplets is at the mixed mean temperature of the film. For the relatively high spray density used in the experiments of Finlay, the influence of evaporation and convection from the free liquid surface are found to be of secondary importance.

Although the simplified mixed model proposed by Finlay ignores the effects of convection and evaporation at the free liquid surface, it gives a fair approximation of the heat transfer coefficients compared to experimental values, except at low liquid loading where the model tends to give zero heat transfer coefficients instead of dry gas-only values. The accurate laminar flow model requires numerical integration to obtain the local heat transfer coefficients, but the simplified "mixed flow" model does not require any numerical integration and yields results which are in very close agreement with that of the more accurate model.

A method to design a spray cooled heat exchanger is proposed by T. Oshima, S. Iuchi, A. Yoshida and K. Takamatsu [72OS1] but is applicable only to a restricted range of commercial heat exchangers.

This method assumes that (i) the air flowing through the heat exchanger is saturated at the inlet and throughout the heat exchanger, (ii) that the finned-tube surface is wetted uniformly and (iii) that the heat and mass transfer coefficients and tube and fin temperatures are uniform around the tube. A simple three-point design method is described where the heat and mass transfer coefficients, temperature and vapour pressure are calculated at the inlet,

the middle and outlet of the heat exchanger. From the energy balance the air/water interface temperature can be determined which leads to the calculation of the required heat transfer surface.

The assumptions made in the simplification of the analytical model were found to hold in the following regions :

- i) air flow rate : $2.78 - 3.34 \text{ kg/m}^2\text{s}$
- ii) water spray rate : $> 350 \text{ kg/m}^2\text{hr}$
- iii) $T_{\text{spray}} < T_{\text{air}} + 15 \text{ }^\circ\text{C}$

H.C. Simpson, G.C. Beggs and G.N. Sen [74SI1] report on experimental work done on a single row of finned-tubes in a horizontal test section. Constant tube temperature was assured by using condensing steam in the tubes. The mean spray water droplet diameter for this experiment was $71.5 \mu\text{m}$ and the air/water ratio varied from 0 to 5 %.

Observations made during these tests showed that droplet deposition during spray cooling is non-uniform and depends on both the mist quality and approach velocity. The rear of the tube and the region of the fin shielded by it, is dry at low mist qualities, but flooding occurred here at high mist qualities and high air velocities.

The heat transfer coefficients were found to decrease with increase in fin height and decrease in spacing, although, particularly for wider fin spacing and higher mist quality, this trend is reversed for the longest fins. At higher mist qualities and approach velocities, the heat transfer contribution of the root (primary) surface became more important. The heat transfer coefficient enhancement ratio was found to be between 5 and 16 times the corresponding dry heat exchanger value, but the enhancement ratio increased at a diminishing rate as the mist quality increased. The highest enhancement ratios are observed for geometries with small fin heights and wide fin spacing. This is primarily the result of these configurations having low heat transfer coefficients for normal dry operation.

The analytical work consisted of two models : Model 1 in which the wall heat transfer coefficient was assumed to have the same value for the root and the fin surfaces and Model 2 in which different coefficients were used for these surfaces. Graphical comparison between the experimental and theoretical heat transfer and temperature distribution data indicated that at low mist qualities and approach velocities, the secondary surface played a

more significant part in the heat transfer process, but with increased mist quality and approach velocity the contribution of the primary surface became more important. Analysis of the data using Model 1, gave much higher fin surface coefficients than Model 2 which emphasise the importance of the primary surface in wet surface heat transfer.

D.R. Tree, V.W. Goldschmidt, R.W. Garrett and E. Kach [78TR1] present a simple semi-empirical theoretical model which is based on the experimental work done by them.

A 4-row finned-tube heat exchanger with a staggered tube arrangement and fin spacing of 2.1 mm were used for the experimental work. The test section consisted of an open test tunnel with horizontal air-mist flow ranging from 6.5 to 19.5 kg/m²hr. The droplet diameters were carefully controlled and three droplet sizes, 64 μm, 440 μm and 3300 μm, were used.

Although the temperature of the coil is not given, the results of the experiments show that the evaporation of the spray water droplets played a major role in the heat transfer process which indicated that the temperature of the coil exceeded 50 °C. From the results it is also clear that the mass flow rate of the spray water has the greatest effect on the heat transfer enhancement which increases as more water is sprayed on the coil. Spray water droplet diameter had a small effect in the ranges studied. Typical performance enhancement ratios of 1.2 - 1.5 were found for the spray cooled heat exchanger.

The simple semi-empirical theoretical model proposed for the calculation of the performance of the spray cooled heat exchanger gives the enhancement ratio as a simple function of droplet diameter, spray water mass flux and gas phase Reynolds number. The empirical correlation for the heat transfer coefficients needed to calculate the enhancement ratio is derived from their experimental work and would only be applicable to the particular arrangement used in their experiments.

The analogy between the wet and dry heat transfer mechanisms is used by D.K. Kreid, B.M. Johnson and D.W. Faletti [78KR1] to derive a model for the calculation of the performance of deluged extended surface heat exchangers.

The analytical procedure is derived from the expressions for mass and energy conservation for a differential element of the heat exchanger surface. The result contains a driving

potential for the wet surface based on the enthalpy differences, whereas in the analogous equation for heat transfer from dry surfaces the driving potential is the temperature difference.

$$\frac{dq_o}{dA_o} = h_a(T_f - T_w) \quad (2.1.1)$$

$$\frac{dq_o}{dA_o} = \frac{h_a^*}{c_{pa}}(i_{asw} - i_a) \quad (2.1.2)$$

To arrive at this result the assumptions of a stable and thin liquid film (to ensure a small temperature variation over the film) and a Lewis factor of 1 (which is approximately valid for air/water systems) are made. To simplify the model the heat transferred by the spray water is neglected which in cases of very high spray water mass-flow will result in less accurate predictions. Although this model predicts the performance of the deluged heat exchanger accurately, the assumptions of a thin and continuous film of spray water render it inaccurate for moderate to low spray densities as well as very high spray water mass flow.

In order to use the enthalpy driving potential for the evaluation of the heat transfer, a transformation parameter is defined in equation (2.1.3) with which each thermal conductance in the path of the heat to the fin surface may be transformed as shown in equation (2.1.4).

$$e = \frac{1}{c_{pa}} \frac{di}{dT} \quad (2.1.3)$$

$$h^* = \frac{h}{e} \quad (2.1.4)$$

By this analogy, the data for dry operation may then be used to predict the performance of the wet surface heat exchangers.

Using this analogy, the authors show that wet operation can result in an increase in the heat transfer by a factor of about three and possibly as high as eight. However some concerns are raised by the authors that the analogy may overpredict as it is based primarily on theoretical and intuitive considerations and has yet to be confirmed by experimental work.

S.D. Wilson and A.F. Jones [78WI1] undertook a theoretical investigation of the heat transfer to a single smooth horizontal tube set transversely in a cross flow of water mist in air.

The analysis is a refined version of the model proposed by Hodgson et al. [67HO2]. The temperature of the cylinder wall is assumed to be less than saturation temperature of the liquid phase and only the front half of the cylinder, which is by assumption always completely covered with a liquid film, is considered when calculating the heat transfer from the air stream to the cylinder. The model includes corrections to handle the effect of droplet deflection and drag on the film due to the gas flow. The effects of gravity, dissipation heating, surface tension of the liquid layer, waves on the surface of the liquid layer, liquid splashing and evaporation are neglected as before.

The equations for the flow of the liquid film on the cylinder and for the heat transport are solved by approximate integral methods, yielding simple closed form equations for the determination of the local Nusselt numbers and the Nusselt number at the forward stagnation point.

From the analysis it is clear that in order to maximize the heat transfer, the droplet diameter, the spray water mass flow ratio and the gas Reynolds number must be as high as possible. However, it is noted that a too thick liquid layer on the cylinder might have an insulating effect and thus reduce the heat transfer. This model compares fairly well to experimental results obtained by Hodgson et al. [67HO1].

A similar heat exchanger configuration was analyzed by C.C. Lu and J.W. Heyt [80LU1] who also derived a theoretical model for the single smooth isothermal horizontal tube. The governing equations are solved for drop trajectories upstream of the cylinder, the laminar liquid boundary layer adjacent to the cylinder surface, and the outer laminar vapour boundary layer. In this model the effects of evaporation and droplet trajectory are also considered and the inlet air is assumed to be saturated as is normally the case in practical wet surface heat exchangers.

Using this model Lu et al. concluded that the velocity, temperature, and mass concentration



profiles throughout the double boundary layers depend on flow parameters involving the spray water drop size and air velocity. The critical conditions for which the liquid film dries out, are identified for a range of drop sizes and air velocities, thus defining a transition zone from the wet surface heat transfer to dry surface heat transfer. It was concluded that by increasing the inertia force of the droplets (higher air velocities and/or larger droplets) a significant increase in heat transfer can be achieved. Whereas theory based on straight drop trajectories tends to overestimate the heat transfer at higher Reynolds numbers and water mass flow rate, this theory which includes the effect of trajectory curvature, is in agreement with the trend of experimental data. The theoretical results for heat transfer are shown to correlate well with existing experimental data as well (Hodgson et al. [68HO1]).

M. Pawlowski and B. Siwon [88PA1] went one step further by deriving a theoretical model for the determination of the heat transfer coefficient for a single row of smooth cylinders in an air/water mist flow, and also conducted some experiments to validate it.

The analysis assumes the existence of a liquid film from the forward stagnation point to the separation point and handles the following three regimes separately, (i) $0^\circ < \theta < 90^\circ$, which is the front part of the cylinder and collects the water droplets directly from the air and is therefore completely wet, (ii) $90^\circ < \theta < \text{separation point}$ and (iii) $\text{separation point} < \theta < 180^\circ$, which is the biggest part of the rear half of the tube and is considered to be completely dry. The continuity, momentum and energy equations were rewritten and simplified using various assumptions, including a Rosin-Rammler distribution of droplets, negligible gravitational effect and laminar flow in the boundary layers and water film. The effect of droplet trajectory and liquid bouncing was incorporated into the model through the use of a dimensionless average volume ratio of liquid entering the film to the amount of liquid directed at the cylinder, k_1 , introduced by Pawlowski in a previous paper (1984). The equations have to be solved numerically at every point around the circumference of a cylinder surface to obtain the heat transfer coefficient.

The experiments performed by them were set up to simulate the analytical model as closely as possible. The single row of electrically heated cylinders (35 mm and 45 mm in diameter) were placed in a small horizontal wind tunnel and sprayed with water from a pneumatic atomizer. The mass flux of the spray water varied from 75.6 to 4226.4 kg/m²hr and the free stream air velocity ranged from 5 to 10 m/s. The water temperature was controlled to be the same as the wetbulb temperature of the air to minimize evaporation from the surface

of the droplets.

Good agreement was found between the analytical model and the experimental data. From the experimental results and the analysis it can be deduced that the heat transfer coefficients for a row of cylinders are higher than for a single cylinder placed in a air/water mist flow. The fraction of the surface of the cylinder covered by a liquid layer, increases with decreasing distance between the cylinders and for cylinder pitches greater than 5 times the cylinder diameter the heat transfer coefficient assumes the value for a single cylinder. An important conclusion which follows from their work is that the heat transfer enhancement resulting from spray cooling, reaches a maximum value as the air velocity increases.

2.2 Experimental work

I.C. Finlay [67FI1] and [67FI2] reports on experimental work using a single horizontal tube in a horizontal air-mist flow with a spray water droplet size of 10-160 μm . Experiments were conducted at air velocities ranging from 20 to 75 m/s while the air/water mass flow ratio were varied from 0 to 9 %.

During the experiments a liquid film was observed between the forward stagnation point and the point of separation where severe bouncing and splashing occurred at high Reynolds numbers. Local and overall heat transfer coefficients were determined, and the region of highest heat transfer coefficients were found to be the front half of the tube where the liquid film is observed. Heat transfer data correlates strongly in terms of the quality of the mixture and the Reynolds number. Average two-phase heat transfer coefficients of up to 20 times the single-phase values were recorded at the high liquid flow rates.

J.W. Hodgson, R.T Saterbak and J.E. Sunderland [68HO1] performed an experimental study on a horizontal smooth tube exposed to a downward cross flow of spray water and air. For air velocities ranging from 6 to 24 m/s the air/water mass flow ratio varied from 0 to 13 % with spray water droplet sizes from 20 to 700 μm .

The local heat transfer coefficients determined were graphically compared to the results of previous studies with which good correlation is reported. Average two-phase heat transfer coefficients of up to 30 times the single-phase (air only) values were recorded. They observed the same strong correlation of the Nusselt number to the air-water ratio and their

results indicate that the Prandtl number of the spray medium has little influence on the Nusselt number. The heat transfer was found to be directly proportional to the temperature difference between the spray water and the cylinder wall. The following correlation presented by Acrivos et al. [64AC1], agrees well with their data.

$$h_{tw}(\theta) = G_w c_{pw} \cos(\theta) \quad (2.2.1)$$

R.L. Mednick and C.P. Colver [69ME1] did experimental work similar to the work of the authors mentioned above, but used a vertical tube, 25 mm in diameter, in a spray water and air cross flow setup. An air velocity of 40 m/s with high spray densities ($> 10000 \text{ kg/m}^2\text{h}$) were used in the experimental work.

A simple correlation is presented for the average Nusselt number on a single tube in the experimental spray density range which holds for dry and wet cylinders surfaces.

In addition to their experimental work an elemental modelling method is also presented by I.C. Finlay and T. McMillan [70FI1] for the thermal design of a spray cooled tube bank.

The experimental model included a tube bank which were seven rows deep spaced in a $1.5x d_0$ triangular array with air and atomized water flowing horizontally over the bank. Air velocities ranged from 8 to 14 m/s and air-water mass flow ratios of 0 to 10 %.

Finley et al. reports large increases in surface heat transfer coefficients with the addition of small amounts of water into the air stream with the combined advantage of small increases in pressure drop. Although the heat transfer coefficients measured in this experiment are lower than the corresponding values obtained with a single cylinder, the variation of the heat transfer coefficient with water injection is similar.

This result conflicts with the results from Pawlowski et al. [88PA1] as discussed in the previous section who found that the heat transfer coefficient for a single row of tubes was higher than the coefficient for a single tube. The reason for this irregularity in results is not clear.

The modelling approach is elemental, similar to that used in the modelling of evaporative coolers or condensers. This model uses a set of differential equations to describe the

transfer processes which take place in a single element. The whole tube is assumed to be covered by the water film. The coefficients used include (i) h_a (taken as the dry tube heat transfer coefficient) which describes the heat transfer from the water film to the air, (ii) the mass transfer coefficient h_D (obtained with the heat/mass transfer analogy and h_a) which describes the mass transfer from the water film to the air stream and (iii) h_f (from the proposed correlation for two-phase heat transfer) which describes the heat transfer between the tube wall and the water film.

M.G. Scherberg, H.E Wright and W.C. Elrod [72SC1] examined the heat transfer characteristics of different heat exchanger geometries in an air-mist flow and the interaction of the liquid droplet with the liquid film on the heated surface.

The results obtained for cylindrical tubes in an air-mist flow is similar to the results of previous authors. Experiments with an elliptical tube and a composite tube, which provide larger areas to be wetted by the liquid film than the cylinder, show a marked increase in the heat transfer coefficient compared to the cylinder.

Liquid droplet interaction with the liquid layer is divided into three regimes according to the momentum of the droplet normal to the liquid film. In the first regime of low droplet momentum, bouncing of the droplets are observed. In the second regime of slightly higher droplet momentum, the droplet will cause a splash and bounce from the liquid film surface. In the third regime splashing also occurs, but the droplet is absorbed into the liquid layer as a result of the high droplet momentum. From this model it follows that the droplets in the third regime contribute the most to the heat transfer enhancement.

In his thesis G.N. Sen [73SE1] reports on experimental work to determine the heat transfer between a series of vertical single row finned-tube banks and an air-water mist flowing across the tubes.

The fin heights of the annular finned-tubes ranged from 10 mm to 22 mm with fin spacing varying between 3.2 mm and 14.3 mm. Air-mist approach velocity varied from 5 m/s to 10 m/s with a mixture quality in the range 0 to 5 per cent by weight. Super heated steam at 34.5 kPa and approximately 90 °C was used to provide a constant heat flux, thus creating a relatively high heat transfer surface temperature.

During testing some conspicuous dryout regions at low mixture qualities were observed on both the primary and secondary surfaces of the all the tubes. These dryout areas formed on the secondary surface directly in front of the tube wall, behind it in the wake of the tube and on the primary surface on the downstream side of the tube wall. The size of the dryout regions seemed to be a function of the air flow rate and spray water mass flux only, but other authors found a strong dependence on surface temperature, which this study does not indicate as it was conducted at constant wall temperature. At mixture qualities higher than $500 \text{ kg/m}^2\text{h}$ (3% and higher), the dryout regions disappeared and agglomerates of liquid formed at discrete locations on the surface.

Visual observations during tests with wider fin spacing revealed that the liquid film does not adhere to the fin surface as strongly as with small fin spacing. As a result, less flooding occurs which in turn increases the heat transfer from the fin surface to the air stream. The behaviour of the liquid film on extended surfaces, which gives a better water distribution on the tube surface than the plain tube bank, suggests that unlike the plain tube bank, considerable heat transfer occurs at the rear section of the finned-tube in air-water mist flow conditions.

From the test data it is clearly evident that the average heat transfer coefficient is strongly dependent on both mixture quality and air velocity. All the data show a levelling off characteristic at a mixture quality of approximately 4 per cent, and it can therefore be concluded that the most effective mixture quality for two phase heat transfer lies in the range of 1 to 4 per cent. This levelling off characteristic of the enhancement ratio or heat transfer coefficient is in accordance with the work of other authors, but the mixture quality at which this happens is strongly dependent on the temperature of the heat transfer surface and will be much lower for moderate temperatures. One surprising result is that the heat transfer coefficient increased with fin height, beyond fin heights of 15 mm, as the approach velocity and mixture quality were increased, and this tendency is even more pronounced with wide spaced fins. In spite of this, the performance of these long fins seem to decrease at higher mist qualities which suggests low fin effectiveness.

The maximum performance enhancement obtained varied from 4 times the dry performance for finned-tubes to 20 times the dry performance for smooth tubes at the highest mixture quality of 5 per cent. The rate of performance enhancement for the finned-tubes decreases with increasing mist quality and level off at approximately 2.5 per cent. The lower fin

surface temperature caused by flooding, which dominates at higher mist qualities, is the most likely reason for the decrease in the rate of performance enhancement.

W-J Yang and D.W. Clark [75YA1] studied the heat transfer enhancement as a result of spray cooling on automotive heat exchangers. Three types of heat exchanger fins, plain, perforated and louvred, were used. The heat exchangers were sprayed either with water or ethylene glycol at a rate of 0 to 21 kg/m²hr, in a horizontal air-flow test section. The temperature difference between the wall and spray were less than 30 K and the maximum wall temperature was 88 °C.

Yang et al. found that the pressure drop is not affected by the sprays of such a low density, but the heat transfer performance increase substantially. Improvements of 40-45% in the heat transfer coefficients are observed at the lower air flow rates. The heat transfer mechanism on the wet heat exchanger surface is a combination of (i) evaporation at the interface, (ii) forced convection at the interface and (iii) the interaction of the liquid droplets with the liquid film. The sprays of water and ethylene glycol yield essentially the same results in spite of the fact that ethylene glycol evaporates at a temperature of 197 °C, two times higher than water. It was thus concluded that the improvement in heat transfer on the air-side was mainly due to the formation of a liquid film on the heat transfer surface while the contribution of evaporation is negligible.

The results of their experimental work are presented in graphical form and show a tendency of decreasing enhancement of the heat transfer with increasing Reynolds number.

P.G. Kosky's [76KO1] experiments on a singular horizontal tube showed that the recorded heat-transfer data for an evaporating mist can be grouped according to the moisture content in three regimes : fully wet, partially dry and fully dry.

The tests were done in an open test tunnel with horizontal air-mist flow (ratio of 0.5% to 20%). The air was heated and then saturated with steam before the water spray were introduced to ensured that the tube wall temperature was lower than the saturation temperature of the air.

The average heat transfer coefficient is characterized by a "dryness parameter".

When this dryness parameter is very small (<0.5), the tube behaves as if it is fully wet. On

$$\phi = \frac{q_{to} A_o}{G f d i_{fg}} \quad (2.2.2)$$

the other hand, if this parameter is very large (> 1.0) the tube behaves as if totally dry. In the fully wet regime, a strong variation of the heat transfer coefficient with Reynolds number is observed and the moisture content has almost no effect. It was found that the mass-transfer resistance is a negligible fraction of the measured thermal resistance in this case. Simple-to-use correlations for the average Nusselt numbers were derived for each of the three flow regimes.

Following an analytical study of deluged heat exchangers, D.K. Kreid, H.L. Parry, L.J. McGowen and B.M. Johnson [79KR1] evaluated the performance of a finned-tube heat exchanger experimentally. The deluge concept suggests that enough water ($\pm 1250 \text{ kg/m}^2\text{hr}$ in this study) is sprayed on the frontal surface area to completely wet the entire surface of the heat exchanger.

A HöTERV test core with staggered tubes and plate fins with a 28 mm spacing were used for the experiments. The air velocity ranged from 1 to 2 m/s and the recirculating deluge flow were kept constant to ensure an equilibrium for the experiment. The results were presented as a comparison between the performance in the dry and wet mode using the enhancement ratio as dependant variable. The heat transfer performance of the deluged surface was compared to the dry surface by adjusting the dry airflow to provide the same pressure drop observed in the deluged test, which typically resulted in a dry airflow of about double that of the deluge airflow.

The performance enhancement ratio found experimentally was seen to be a strong function of the initial temperature difference between the air and primary fluid, but the effect of air velocity was less pronounced than expected because the comparisons were made at constant pressure drop. Low relative air humidity increased the performance enhancement ratio at low initial temperature differences, but the effect of the air humidity on the performance enhancement ratio decreased with increasing initial temperature difference.

Although uncertainties did exist in obtaining accurate data from the experiments and in the nature of the approximations of the analytical model, the agreement between the theoretical model and the experimental model were reasonably good. This indicates that the analytical

method given by the same authors in their previous paper [KR781] can be used with confidence to predict the maximum performance of wet finned-tube heat exchangers.

In the experiments performed by H. Kuwahara, W. Nakayama and Y. Mori [81KU1], the emphasis were placed on *low* air velocities and spray water mass flow rates. These two factors make the evaporative mode of heat transfer an important factor in the evaluation of wetted heat exchangers.

A single horizontal tube (31 mm in diameter) in a open test tunnel were used to test three different surfaces: Smooth tube surface, a tube surface with spiral-radial fins and a tube surface with pores. The fins were sharp-edged and small (1 mm high) with a fin spacing of less than 1 mm to ensure that the water on the surface is well distributed.

The tube wall temperature was controlled electrically within the range of 20-80 °C and the air velocity were varied from 1.7-11 m/s. Low spray water densities, 30-453 kg/m²hr, were used to comply with the low pumping power requirement.

From the experimental results it was found that the circumferential heat conduction is not negligible. Only the leading half of the smooth tube was covered with a liquid film and the rear surface remained dry even though occasional drops did hit this surface. However, on the finned and the pored tube surfaces, the liquid film spread also over the rear half of the cylinder, which can be seen from the increased heat transfer on these surfaces. The heat transfer from these surfaces was however decreased by the presence of a "dead region" formed from the drops hanging between the fins at the bottom of these tubes. It was noted that the finned-tubes started to dry at the upper separation point when wall temperatures were increased and/or spray water massflow rates decreased.

An analytical prediction method based on the enthalpy difference between the saturated air at the wall temperature and the free stream air are presented. A Lewis factor of one was assumed to simplify the estimation of the mass transfer coefficient. A further assumption made was that no dry-out occurred, which means that the surface was considered to be completely covered with a thin water film. Because of these assumptions, the correlations required for this prediction method are relatively simple and are presented for the smooth and finned-tubes in the following form:

$$q = f(d_i, m_w, v_a) \quad (2.2.3)$$

The experiments H.C. Simpson, G.C. Beggs and M.I.N. Raouf [84SI1] conducted on a vertical finned-tube bank, provide valuable insight to aid the understanding of the problem of two phase heat transfer in tube banks.

A bank of machined copper tubes was tested in a horizontal test section with air-water mist flow velocities ranging from 2 to 10 m/s and spray water mass flow rates of up to 820 kg/m²hr. The tube bank consisted of three rows of tubes, either 20 mm, 26 mm or 32 mm in diameter, in staggered formation with pitches of 72 mm, 82 mm or 90 mm respectively. To obtain a variety of geometries for their experimental work, the fins were machined down after completion of each test, to provide a new geometry. The fin geometries tested varied from fin heights of 15mm (fin thickness of 1.6 mm) spaced at 4 mm intervals down to 10 mm and 5 mm high fins spaced at 9.6 mm and 20.8 mm respectively.

The results of their experimental work show that the addition of spray water has a negligible effect on the pressure drop across the tube bank. This fact is supported by most studies in this field although it must be noted that the experimental data provided by them shows a considerable increase in pressure drop for the configuration with the longest fins and smallest fin pitch. This may be the result of spray water drops that form pellets of water between neighbouring fins, thus partly blocking the fin gaps and restricting the free flow of air.

The heat transfer mechanism appeared to be related to the nature of the water film on the surface, to the splashing or bouncing-off action of the water particles and to any drying out of the water film on the surface. When the surface temperature exceeds the air temperature by 30 °C or more, dry patches develop on the surface of the fins and tube resulting in a sharp increase of the average wall temperature. For this reason the heat transfer data was correlated by two sets of equations depending on the initial temperature difference.

To determine the performance of extended surface heat exchangers in a wet-dry operating mode, O. Fischer and A. Sommer [88FI1] tested an experimental plate-fin heat exchanger in horizontal air stream with the fins in both the vertical and horizontal positions.

Decarbonated water was sprayed in the air before the heat exchanger at very moderate rates ($< 120 \text{ kg/m}^2\text{h}$). The air pressure drop, spraying water holdup in the cooling element and exit conditions were measured at steady state and preset inlet conditions. The inlet conditions that were varied between tests were the air velocity, water spraying rate, initial temperature difference and different positional orientations of the test core.

It was found that the spray water holdup reaches a maximum at a specific air velocity. The spray water holdup reduces abruptly above the critical air velocity (which was found to be approximately 2.5 m/s for this heat exchanger). The critical velocity for both the heat exchangers (one with 25 mm tube pitch and the other with 45 mm tube pitch) tested, were found to be the same, thus suggesting that the fin gap and tube diameter are the geometrical parameters controlling the critical velocity.

The wetting behaviour of the fins were modelled by assuming a mean contact angle of 50° and it was found that water pellets will form between the fins if the fin spacing is less than 3.1 mm . This situation increases the pressure drop over the wet heat exchanger and in particular for the low air velocity situation with maximum water holdup. The vertical fin geometry showed little increase of the pressure drop over the dry operation.

The performance of the wet heat exchanger showed a marked increase in comparison with the dry operation of the heat exchanger and performance enhancements of 1.4 at high air velocities and up to 2.2 at low air velocities were recorded. The increase in heat transfer is accompanied by a proportionally lower increase in air pressure drop but it was found that the poor surface wettability and the lack of a long-term and large-scale solution for this problem reduces the enhancement potential.

The experimental study involving smooth and finned-tube banks in air-mist flow by W. Nakayama, H. Kuwahara and S. Hirasawa [88NA1], led to concept of using an effective wet area and considering the rest of the tube bank to be dry.

The heat exchanger core used for the experimental work consisted of horizontal tubes arranged in a bank deep in the vertical direction (8-11 tubes) and short in the direction of the air stream (4 tube rows in staggered formation). To study the effects of spray cooling on extended surfaces as well, three different tube surfaces were used in the experiments: Smooth, microfinned and finned. The finned-tubes, which were machined from aluminium,

had fins 5 mm in height with a pitch of 2.4 mm while the microfins were only 1 mm high and spaced with a pitch of 0.7 mm.

The air velocity and spray water mass flux over the heat exchanger were varied from 1 to 3 m/s and from 50 to 390 kg/m²hr respectively. Low air velocities were used to accommodate the demand on reduced air velocities in industrial applications. The result of the low air velocity was that the water drainage in the tube bank was largely vertical which in turn effected the heat transfer on the tubes at the lower levels.

From inspection it was found that the tube surface is primarily wetted by water droplets from the air stream colliding with the tube surface and drainage from tubes in the upper rows. Four regions in the tube bank, in which the heat transfer characteristics depend on the mass flux of spray water and the temperature difference between the air and the tube surface temperature, were identified. The first region contains the tubes of the first row and the heat transfer characteristics here are similar to that of a single tube. The second region contains the tubes in the upper part of the second and third tube rows, which receives less spray water and thus show a lower heat transfer capacity, while the tubes in the lower part of these two rows forms the third region which has a higher heat transfer capacity due to the drainage water supplied from regions one and two. The fourth region corresponds to the fourth tube row in the bank where the water supply rate is small. The tubes in this region tend to dry out at low spray water mass fluxes and the heat transfer capacity thus compares to that for dry tubes.

Due to the smaller free flow area and the large surface area of the finned-tubes, these surfaces were more effective in arresting water drops from the air. This resulted in a relatively lean water supply on the tube rows behind the first row in the finned-tube bank. Although the heat transfer coefficients of the finned-tubes were found to be higher than those of the smooth tubes, the performance enhancement in the air-mist flow were less for the finned-tubes than for the smooth tubes. This indicates that the effect extended surfaces have on the heat transfer potential of an heat exchanger are less pronounced in two phase flow than in single phase flow.

The theoretical model used for the data analysis is an adaptation of a model proposed in a previous paper by the same authors [81KU1]. The model allows for water flowing downwards from one tube to the next but neglects to take into account the evaporation from

the surface of the spray water droplets in the air stream as it flows over the tube bank. The assumption that the efficiency of the fins in mist flow is consistent with that of fins in dry air flow, were proven by A.H. Elmahdy and R.C. Biggs [83EL1] to be incorrect and may lead to overestimation of the capacity of the heat exchanger.

The correlations presented can be used with the theory discussed to evaluate spray cooled heat exchangers to a high degree of accuracy given that it resembles the configurations used by the authors closely.

The work of Nakayama et al [88NA1] served as a basis for the experimental study by P.J. Erens, A.A. Dreyer and D.E. Kriel [90ER1]. The heat exchanger used for spray cooling experiments consisted of 220 smooth tubes (38 mm OD) packed in a $2 \times d_o$ triangular configuration in ten vertical rows of 22 tubes each. Ambient air was drawn through an open end tunnel at velocities that varied from 1 to 4 m/s while spray water was supplied to the air at mass fluxes that ranged from 0 to 570 kg/m²/hr.

The two phase heat transfer data is presented in terms of an enhancement factor and an evaporation efficiency which is defined as follows

$$F_{\text{enh}} = \frac{\text{Experimental two phase capacity}}{\text{Calculated single phase capacity}} \quad (2.2.4)$$

$$e_{\text{evap}} = \frac{\text{Actual evaporation rate}}{\text{Theoretical maximum evaporation rate}} \quad (2.2.5)$$

From the experimental data it was found that the maximum evaporation efficiencies occurred at very high and very low air velocity rates. It was thus concluded that this is the result of a long residence time of the air in the wetted part of the exchanger in the case of the low air velocity, and a large effective wet area that result from droplet migration at high air velocities.

The model used for evaluation of the data is set up along the same lines as the model proposed by Nakayama [88NA1]. Certain modifications were however necessary to compensate for the inadequacies of Nakayama's model concerning heat exchangers with more than 4 tube rows and higher air velocities in the heat exchanger. To compensate for the effect of horizontal flowing air stream has on the falling water drops, the Yung [80YU1] model for falling droplets in an transverse air stream was incorporated into a new model

developed by the authors to allow the water to migrate through the exchanger as it drained from the upper tubes to the lower tubes. To compensate for water evaporation from the surface of the droplets into the air stream, this model assumes that the air flowing over the heat exchanger surface is saturated at all times if there are any water droplets present in the air. With this new model Erens et al. [90ER1] were able to predict the enhancement factor for the spray cooled heat exchanger with smooth tubes within 20%.

2.3 Discussion

For an overview of the experimental work done in this particular field, Table 2.1 has been included. This table shows the range and field of experimental work from which it was possible to draw some important conclusions regarding the physical behaviour of the spray cooled heat exchanger to supplement and confirm the results obtained in this study. These conclusions are discussed in Chapter 8.

The fact that spray cooling provides a practical method to enhance the performance of industrial dry heat exchangers is the most important conclusion which can be drawn from the work of all the authors cited in Table 2.1. Large increases in heat transfer combined with low pressure drops were reported in all the experimental work involving small spray water mass flow rates. The increase in heat transfer was found to be more pronounced on smooth heat transfer surfaces than on extended surfaces such as finned-tubes, but the distribution of the spray water film was better on the extended surfaces.

The analytical approaches of the authors mentioned in this chapter can be divided into two categories:

- (i) The analytical models of this category use the dry air heat transfer potential, which is the temperature difference between the air and heat transfer surface, to calculate the performance of the heat exchanger. The heat transfer coefficient is modified to take into account the additional heat transfer that is the result of the mass transfer and sensible heat transfer from the liquid film on the heat transfer surface.
- (ii) The analytical models of the second category use the enthalpy difference between the air and the liquid film on the heat transfer surface as heat transfer potential. This theory was introduced by Merkel [26ME1] and is also used for deluged evaporative cooling. The heat transfer process is described with the use of two

equations, one describing the mass transfer between the air and liquid film, and the other describing the convective heat transfer between the air and the surface of the liquid film. These two equations are coupled by the Lewis factor and together describe the heat transfer process without modification of the heat transfer coefficient of the heat exchanger.

Table 1 - Comparison of experimental work

Reference		Configuration	v_w m/s	m_w/m_a	F_{enh}
Elperin	61EL1	Tube bundle	-	0.2	17
Acrivos	64AC1	Vertical cylinder	60-100	0.15	9
Finlay	67FI1	Horiz. cylinder	23-76	0.1	17
Finlay	68FI1	Horiz. cylinder	23-76	0.1	17
Hodgson	68HO2	Horiz. cylinder	6-25	0.13	30
Mednick	69ME1	Vert. cylinder	18-43	0.25	36
Finlay	70FI1	Smooth tube bank	8-13	0.1	5.5
Oshima	72OS1	Finned-tube bank	0.5-3	0.03	4
Scherberg	72SC1	Cylinder, ellipse	20-44	0.06	15
Sen	73SE1	Finned tube bank	5-10	0.05	4
Simpson	74SI2	Finned-tube row	2-10	0.05	20
Yang	75YA1	Compact H.X.	1.5-13	0.01	20
Kosky	76KO1	Horiz. cylinder	-	0.2	44
Tree	78TR1	Finned-tube bank	0.8-2.3	0.006	1.4
Kreid	79KR1	Finned-tube bank	1-2	0.15	3.5
Kuwahara	81KU1	Cylinder-sm/fin	1.7-11	0.06	-
Simpson	84SI1	Finned-tube bank	2-10	0.06	1.3
Fischer	88FI1	Plate fin H.X.	1-5	0.006	1.8
Nakayama	88NA1	Tube banks-sm/fin	1-3	0.09	5
Pawlowski	88PA1	Smooth tube row	5-12	0.12	
Erens	90ER1	Smooth tube bank	1-4	0.09	5

The main advantage of the second method is that it allows a more general analysis of spray cooled heat exchangers. This is the result of the model's use of the heat and mass transfer coefficients of the exchanger without any modifications, whereas the first category of models which uses a modified heat transfer coefficient applies only to the specific geometry and

range of conditions for which the modified heat transfer correlation has been determined experimentally. It has the added advantage of contributing to understanding of the physical behaviour of wet surface heat transfer processes.

The knowledge on the spray cooled heat exchangers is so far incomplete. The present study looks at only a fraction of the applications of spray cooling, the spray cooling of a finned-tube heat exchanger in the vertical air flow mode. The models presented here fall into the second category of heat transfer driven by enthalpy potential. By combining some of the analytical models presented by other authors and using the information available from experimental work about the physical behaviour of the heat transfer surface in air-mist flow, it will be shown that the models developed for the prediction of performance of spray cooled heat exchangers in the present study is quite adequate for engineering purposes.

3. MATHEMATICAL MODELLING OF SPRAY COOLED HEAT EXCHANGERS

The first part of this chapter describes the basic theory for the control volume of an evaporative cooler to give a better understanding of the physics involved in wet surface heat transfer. Following this, it will be shown how the control volume theory can be modified to be applicable to the specific case of the spray cooled heat exchanger with extended surface in the vertical air flow mode. Lastly a simplified method of performance prediction based on work done by Nakayama et al. [89NA1] and Erens et al. [90ER1] will be presented.

3.1 Basic theory for evaporative coolers

To develop a sensible control volume model which will produce reasonably accurate predictions of the heat and mass transfer taking place in it, the following simplifying assumptions are necessary :

- i) The system is in a steady state.
- ii) Radiative heat transfer is negligible.
- iii) The analytical model needs only to consider one dimensional processes as the variations in mass and energy in the second dimensional direction are much smaller, i.e. the air temperature is uniform along a certain height in the heat exchanger and the process fluid temperature is uniform over any cross section of a tube.
- iv) The spray water and air will reach an equilibrium state before it reaches the first tube of the exchanger and then the spray water droplets will always assume the wet bulb temperature of the air as it moves through the heat exchanger.
- v) The air will always be saturated if there are spray water droplets present in the air.
- vi) The heat and mass transfer coefficients are not influenced by the rates of these transfer processes.
- vii) The surface area of the tube and water film, for heat and mass transfer purposes, is the same as that of the dry finned-tube.
- viii) The spray water droplets in the air will follow a straight path and the mass flux will have a uniform density along any height in the heat exchanger.
- ix) The Lewis factor is equal to unity.

$$Le_f = 1 \quad \therefore \quad h_D = \frac{h_c}{c_{pa}} \quad (3.1.1)$$

- x) The temperature of the interface surface between the water film on the tube surface and the air is equal to the average bulk temperature of the film.

The analytical method presented here uses the same approach as Poppe [84PO1] and Bourillot [83BO1] to describe the heat and mass transfer processes in the conventional cooling tower.

To quantify the energy transfer that takes place, we use the first law of thermodynamics for a control volume to give us the equations required for the model. The first step in this analysis is to define the control volume and its boundaries. A typical element of the cooler illustrating the control volume, is shown in Figure 3.1.

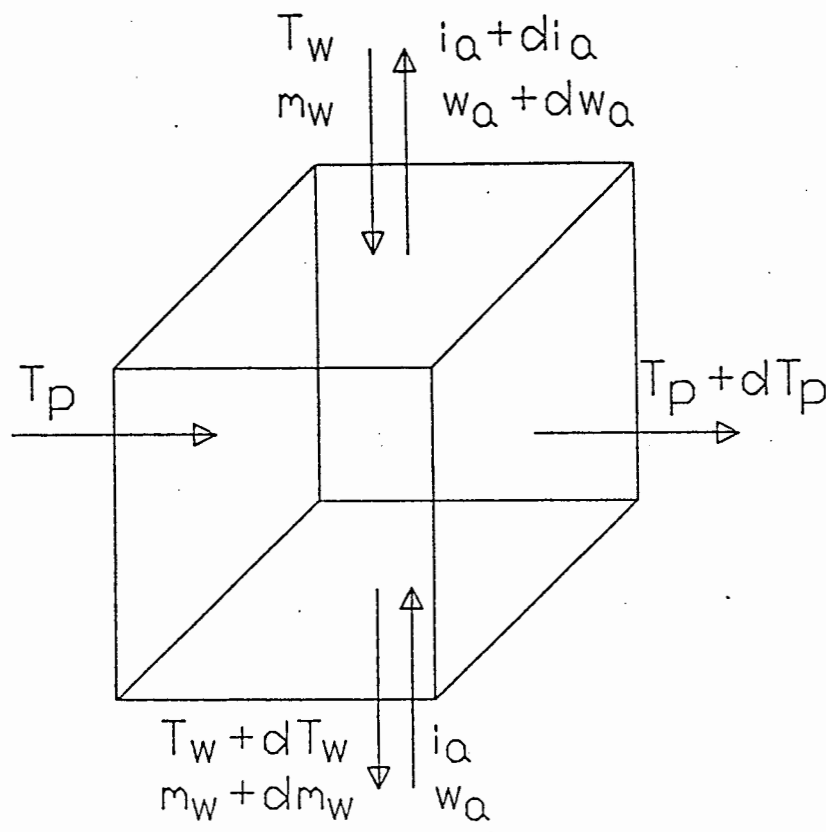


Figure 3.1 - Energy balance for a typical evaporative cooler element.

From the mass balance of the element it follows

$$m_a dw_a + dm_w = 0$$

$$\therefore dw_a = -\frac{dm_w}{m_a} \quad (3.1.2)$$

The energy balance of the system gives

$$m_a i_a + m_w c_{pw} T_w + m_p c_{pp} T_p = m_a (i_a + di_a) + (m_w + dm_w) c_{pw} (T_w + dT_w) + m_p c_{pp} (T_p + dT_p)$$

which by ignoring the second order terms, can be simplified to

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

$$\therefore dT_w = \frac{1}{m_w c_{pw}} (-m_a di_a - dm_w c_{pw} T_w - m_p c_{pp} dT_p) \quad (3.1.3)$$

In order to determine the changes in the temperature of the process fluid, the enthalpy of the air and the mass flow of the air, it is necessary to consider these energy changes more closely. The heat transfer from the process fluid to the film of spray water on the outer surface of the tube is given by

$$dq = -U_o (T_p - T_w) dA_o \quad (3.1.4)$$

If the elliptical tube and fin construction of the heat exchanger under discussion is considered to resemble straight fin construction on a flat plate the overall heat transfer coefficient is given by:

$$\frac{1}{U_o} = \left\{ \frac{1}{h_p} \frac{A_o}{A_i} + \frac{A_o}{n_f n_{fr} L_t} \left(\frac{t_t}{k_f P_t} + \frac{t_r}{k_f P_r} \right) + \frac{1}{e_o h_w} \right\} \quad (3.1.5)$$

Using the energy balance equation for the process water

$$dq = m_p c_{pp} dT_p \quad (3.1.6)$$

together with equation (3.1.4), the temperature change of the process water can be described as

$$dT_p = - \frac{U_o}{m_p c_{pp}} (T_p - T_w) dA_o \quad (3.1.7)$$

In order to describe the changes in the enthalpy of the air and the mass flow of the spray water, the equations which describe the simultaneous heat and mass transfer processes are required. The form of these equations depend on whether there are spray water drops present in the air or not.

Case 1 - Water drops present in the air

Using Dalton's evaporation law, the mass flow rate of the water evaporating from the film on the tube into the air can be expressed as

$$dm_w = - h_d (w_{asw} - w_{as}) dA_o$$

$$dm_w = - \frac{h_c}{c_{pm}} (w_{asw} - w_{as}) dA_o \quad (3.1.8)$$

The simultaneous heat and mass transfer at the air/water interface is given by

$$dq = - i_v dm_w + h_c (T_w - T_a) dA_o \quad (3.1.9)$$

The term representing the change in the mass flow of the water film, dm_w can now be replaced in this equation by equation (3.1.8) and noting that $dq = m_a di_a$ it follows that

$$m_a di_a = \frac{h_c}{c_{pm}} (w_{asw} - w_{as}) i_v dA_o + h_c (T_w - T_a) dA_o \quad (3.1.10)$$

For the simplification of this equation the following auxiliary equations are required:

$$c_{pm} = c_{pa} + w_{as} c_{pv} + (w_a - w_{as}) c_{pw} \quad (3.1.11)$$

$$i_v = i_{vo} + c_{pv} T_w \quad (3.1.12)$$

$$i_a = c_{pa}T_a + w_{as}(i_{vo} + c_{pv}T_a) + (w_a - w_{as})c_{pw}T_a \quad (3.1.13)$$

$$i_{asw} = c_{pa}T_w + w_{as}(i_{vo} + c_{pv}T_w) + (w_{asw} - w_{as})(i_{vo} + c_{pv}T_w) \quad (3.1.14)$$

Because of the assumption made earlier that the spray water drops in the air assume the wet bulb temperature, these equations are formulated to account for the spray water drops in the super-saturated air and thus allow the energy balance calculations of the spray water drops and air to be achieved in one step.

By subtracting equation (3.1.13) from equation (3.1.14), it follows that

$$\begin{aligned} (i_{asw} - i_a) &= [c_{pa} + w_{as}c_{pv} + (w_a - w_{as})c_{pw}](T_w - T_a) + \\ &\quad (w_{asw} - w_{as})i_v - (w_a - w_{as})c_{pw}T_w \\ \therefore (T_w - T_a) &= \frac{(i_{asw} - i_a) - (w_{asw} - w_{as})i_v + (w_a - w_{as})c_{pw}T_w}{c_{pm}} \end{aligned} \quad (3.1.15)$$

Combining equations (3.1.10) and (3.1.15) gives the following result:

$$\begin{aligned} m_a di_a &= \frac{h_c dA_o}{c_{pm}} [(w_{asw} - w_{as})i_v + (i_{asw} - i_a) - \\ &\quad (w_{asw} - w_{as})i_v + (w_a - w_{as})c_{pw}T_w] \\ di_a &= \frac{h_c dA}{c_{pm} m_a} [(i_{asw} - i_a) + (w_a - w_{as})c_{pw}T_w] \end{aligned} \quad (3.1.16)$$

Case 2 - No water drops in the air

In this case, the air is not super-saturated but the film of water on the tube surface still exists. Using Dalton's evaporation law as in Case 1, the mass flow rate of the water film on the tube into the air can be expressed as

$$dm_w = -h_d(w_{asw} - w_a)dA_o$$

$$dm_w = - \frac{h_c}{c_{pm}} (w_{asw} - w_a) dA_o \quad (3.1.17)$$

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

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

The term representing the change in the mass flow of the water film, dm_w can now be replaced in this equation by equation (3.1.17) and noting that $dq = m_a di_a$ it follows that

$$m_a di_a = \frac{h_c}{c_{pm}} (w_{asw} - w_a) i_v dA_o + h_c (T_w - T_a) dA_o \quad (3.1.18)$$

For the simplification of this equation we need the following supplementary equations:

$$c_{pm} = c_{pa} + w_{asw} c_{pv} \quad (3.1.19)$$

$$i_v = i_{vo} + c_{pv} T_w \quad (3.1.20)$$

$$i_a = c_{pa} T_a + w_a (i_{vo} + c_{pv} T_a) \quad (3.1.21)$$

$$i_{asw} = c_{pa} T_w + w_a (i_{vo} + c_{pv} T_w) + (w_{asw} - w_a) (i_{vo} + c_{pv} T_w) \quad (3.1.22)$$

These equations are the same as equations (3.1.11) to (3.1.14) but do not make any provision for water drops in the air.

By subtracting equation (3.1.21) from equation (3.1.22), it follows that

$$(i_{asw} - i_a) = (c_{pa} + w_a c_{pv}) (T_w - T_a) + (w_{asw} - w_a) (i_{vo} + c_{pv} T_w)$$

$$(i_{asw} - i_a) = c_{pm} (T_w - T_a) + (w_{asw} - w_a) i_v$$

$$\therefore (T_w - T_a) = \frac{(i_{asw} - i_a) - (w_{asw} - w_a) i_v}{c_{pm}} \quad (3.1.23)$$

Combining equations (3.1.18) and (3.1.23) gives the following result:

$$m_a di_a = \frac{h_c dA}{c_{pm}} [(w_{asw} - w_a) i_v + (i_{asw} - i_a) - (w_{asw} - w_a) i_v]$$

$$di_a = \frac{h_c dA}{c_{pm} \dot{m}_a} (i_{asw} - i_a) \quad (3.1.24)$$

The full set of five differential equations describing the changes in temperature, enthalpy and mass flow of the water and air flowing through the system is now given for **Case 1** as equations (3.1.2), (3.1.3), (3.1.6), (3.1.8) and (3.1.16) and for **Case 2** as equations (3.1.2), (3.1.3), (3.1.6), (3.1.17) and (3.1.24). Solving either of these two sets of equations simultaneously it is now possible to determine the performance of a wet surface heat exchanger in super-saturated or non-saturated air flow.

3.2 Modelling procedures for spray cooled heat exchangers with extended surfaces

The analytical modelling of spray cooled heat exchangers depends mainly on the physical behaviour of the heat exchanger during operation. One of the primary participants is the spray water film on the surface of the spray cooled heat exchanger. The spray water film may be flowing upwards or downwards on the heat exchanger surface, some tubes will be only partially wet, others completely wet and some others completely dry. As a result of the inconsistent behaviour of the spray water film on the heat exchanger surface, the modelling of the spray cooled heat exchanger is very complicated.

Many approaches to the problem of spray cooled heat exchanger modelling are possible. Some of them are referred to in Chapter 2, but they all have a very important shortcoming in that the models developed so far are only applicable for (i) a very limited range of spray water flow rates which ensures that the complete heat transfer surface is wet or (ii) a specific geometry of the heat exchanger surface through empirical equations developed from experimental results. This makes these methods very difficult to apply in general industrial applications. In this section two models will be presented which will predict the performance of any normal dry air heat exchangers with extended surface operated with spray cooling in the vertical air flow mode, with reasonable accuracy. These models include certain simplifying assumptions described earlier in section 3.1, but no empirical equations except those giving the value for the dry heat transfer coefficient are used. The models are applicable for the range of spray water mass fluxes from 0 to 500 kg/m²h.

3.2.1. An accurate model for vertical air-flow heat exchangers with extended surfaces

The modelling approach is based on the equations which were derived in the previous section, but a few modifications are necessary to calculate of the performance of the finned-tube heat exchanger in a vertical air-mist flow setup possible.

There are basically four modes of operation for the spray cooled heat exchanger:

- i) The surface of the heat exchanger is wet (the thin water film exists) and the air is super-saturated (there are spray water droplets in the air).
- ii) The surface of the heat exchanger is wet and the air is non-saturated (all the spray water droplets in the air have evaporated or become part of the water film on the heat exchanger surface).
- iii) The surface of the heat exchanger is dry (the thin water film has dried out) and the air is super-saturated.
- iv) The surface of the heat exchanger is dry and the air is non-saturated.

The third and fourth modes can be combined into one by allowing dry heat transfer to take place in super-saturated air as in non-saturated air with the assumption that the air will always be saturated if spray water drops is present in the air. This leaves only three modes of operation to be dealt with in a combined model which will be able to handle the combination of all three modes in one heat exchanger thus making it possible to predict the performance of the spray cooled heat exchanger. The analytical model using this technique is presented below as model ACCURATE.

Mode 1 - wet surface and super-saturated air

The outlet conditions of the air, spray water and process water can be calculated for an element of the heat exchanger in Mode 1 by simultaneously solving the differential equations of **Case 1**. A problem which arises when solving these equations is the disappearance of the water film on the tube surface which implies that the value of m_w in equation (3.1.3) becomes zero or almost zero. As a result the set of equations being solved will become unstable. To solve this problem, equation (3.1.3) can be rewritten for the element as

$$q_{(i)} = m_a(i_{a(i+1)} - i_{a(i)}) + c_{pw} T_w (m_{w(i+1)} - m_{w(i)}) + m_w c_{pw} (T_{w(i+1)} - T_{w(i)}) \quad (3.2.1)$$

By rewriting equation (3.1.3) and using the value for $q_{(i)}$ calculated from equation (3.2.1) the average film temperature can be calculated as:

$$T_{wav} = T_{pav} - \frac{q_{(i)}}{U_o A_{o(i)}} \quad (3.2.2)$$

To obtain an accurate value for the film temperature an iteration process is necessary which involves assuming a film temperature and then solving equations (3.1.2), (3.1.6), (3.1.8) and (3.1.16) simultaneously to enable the calculation of a new film temperature using equations (3.2.1) and (3.2.2).

Mode 2 - wet surface and non-saturated air

The outlet conditions of the air, spray water and process water can be calculated for an element of the heat exchanger in Mode 2 by simultaneously solving the differential equations of **Case 2**. The problem that arises when solving these equations is the same as for Mode 1 and is dealt with in the same way.

Mode 3 - dry surface

When the surface of the heat exchanger is dry, there remain only two differential equations to be solved

$$dT_p = - \frac{U_o}{m_p c_{pp}} (T_p - T_a) dA_o \quad (3.2.3)$$

$$dT_a = \frac{U_o}{m_a c_{pa}} (T_p - T_a) dA_o \quad (3.2.4)$$

These two equations completely describe all the changes that takes place in the dry heat exchanger.

If there are spray water droplets in the air, the outlet conditions of the air and spray water droplets can be determined by solving the following energy balance equation assuming that the air containing water droplets is saturated.

$$i_{\text{tot}} = i_{\text{as}} + (w_{\text{a}} - w_{\text{as}})c_{\text{pw}}T_{\text{as}} \quad (3.2.5)$$

where

$$i_{\text{as}} = c_{\text{pa}}T_{\text{as}} + w_{\text{as}}(i_{\text{vo}} + c_{\text{pv}}T_{\text{as}})$$

Determination of the flow direction of the water film on the heat exchanger surface

The problem of flow direction is an important one since the water film is the primary contributor to heat transfer enhancement in the spray cooled heat exchanger. To determine whether the water film flows upwards or downwards through the heat exchanger it is first necessary to determine the flow reversal point for the two phase flow situation that exists on the surface of the heat exchanger.

There are a large number of empirical equations available to determine the flooding point and flow reversal point for two phase flow, but the following equation by Wallis [63WA1] is used because of its simplicity and accuracy.

$$v_{\text{a}}^* = v_{\text{a}}\sqrt{\rho_{\text{a}}}\sqrt{gd_{\text{h}}(\rho_{\text{w}} - \rho_{\text{a}})} \quad (3.2.6)$$

Flow reversal will occur when $v_{\text{a}}^* = 1$. The critical velocity, V_{crit} at which the spray water film will change flow direction, can be determined by calculating the maximum air velocity for which equation (3.2.6) will be equal to one. The maximum air velocity is calculated at the smallest area of the heat exchanger and thus the real flow reversal velocity is not a specific value but includes a range of velocities. A transition zone in which the water film is not definitely flowing up or down exists around the critical velocity. For this model it was assumed that the film flows upwards on the heat exchanger surface if

$$v_{\text{max}} > 1.2 \times v_{\text{crit}} \quad (3.2.7)$$

Calculation of the wetted area for the spray cooled heat exchanger with extended surface - Model ACCURATE

Contrary to the ideal model the surface of the spray cooled heat exchanger is never uniformly wet. This is the result of many factors, the main reasons for this phenomenon being the poor wettability due to surface tension effects and the fact that liquid entrapment by the upstream surface of the tube is much larger than the downstream surface. This surface wetting behaviour has been found by other researchers such as Nakayama et al [88NA1] and Sen [73SE1] whose work is discussed in Chapter 2.

To enable the model to take into account the fact that the wetted area is some ratio of the total area, an effective wet area ratio is defined. This ratio approximates the fraction of the area that can be considered wet when the spray density is high enough to prevent dryout of the spray water film as a result of evaporation. It is also dependant on the critical velocity discussed in the previous section because the wetted surface will increase as the air drags the water film to the downstream side of the tube surface at higher air velocities.

It is not possible to determine the actual size of the wetted area in comparison to the outer tube area for all possible geometries and spray water densities, but some sensible assumptions can be made. For low spray water densities, 0 - 250 kg/m²h, the size of the effective area is not important at all, as the model automatically lets part of the tube surface dry out as the water film on the tube surface evaporates. For higher spray densities there is enough water to sustain the film on the tube surface, and the effective area ratio becomes an important factor in the calculation of the heat transfer rate. The logical factor to use for sub-critical air velocities emerges from the argument that as the upstream side of the tube intercepts most of the droplets and the water film flows downwards, only 50% of the tube surface will be wet. For air velocities higher than the critical velocity, the film flows upwards and therefore flows over the downstream side of the tube as well. The ratio for the effective area depends mainly on the size of the heat exchanger surface which means that if the area is large, ie. closely spaced long fins, the ratio will tend to be low, approximately 70%. If the fins are short and widely spaced the ratio will go up to 90%. These figures were also reported by Nakayama [88NA1] (short fins) and Sen [73SE1] (long fins). For the heat exchanger under discussion the following applied:

$$A_{ev} = 0.5 \quad \text{if} \quad V_{max} < 0.8 \times V_{crit} \quad (3.2.8)$$

$$A_{ev} = 0.7 \quad \text{if} \quad V_{max} > 0.8 \times V_{crit} \quad (3.2.9)$$

The effective area ratio is multiplied by the value of the outside area to produce a value for the wetted area. Equations (3.1.8) and (3.1.16) are thus modified as follows

$$dm_w = - \frac{h_c}{c_{pm}} (w_{asw} - w_{as}) dA_o A_{ev} \quad (3.2.10)$$

$$di_a = \frac{h_c dA_o}{c_{pm} m_a} [(i_{asw} - i_a) + (w_a - w_{as}) c_{pw} T_w + (w_{asw} - w_{as}) i_v (A_{ev} - 1)] \quad (3.2.11)$$

for Case 1, and for Case 2 equations (3.1.17) and (3.1.24) becomes

$$dm_w = - \frac{h_c}{c_{pm}} (w_{asw} - w_a) dA_o A_{ev} \quad (3.2.12)$$

$$di_a = \frac{h_c dA_o}{c_{pm} m_a} [(i_{asw} - i_a) + (w_{asw} - w_a) i_v (A_{ev} - 1)] \quad (3.2.13)$$

Determination of the thickness of the water film on the surface of the spray cooled heat exchanger

The thickness of the water film on the surface of the tubes plays an important role in the spray water droplet arresting rate and heat transfer resistance. To determine the thickness of this film empirically is impossible due to the inconsistency of the heat transfer surface and flow direction. The frontal area of the tube which is responsible for droplet entrapment, will be greatly increased if a very thick film exists in the surface of the tube. The film heat transfer coefficient is also dependent on the film thickness since the heat must be conducted through it as shown by the following equation

$$h_f = \frac{k_w}{\delta} \quad (3.2.14)$$

By ignoring the dependence of the film heat transfer coefficient on the film thickness and assuming a constant value for the film heat transfer coefficient, and including the concept of "thermally dead areas" for areas where the film is very thick, the exact film thickness need not to be known. Liquid retention occurs on the lower part of the first tube as it collects most of the spray water drops from the air and is flooded by a thick liquid film. This is in line with the condensate retention model presented by T.M. Rudy [85RU1] and the experimental findings of A.M. Jacobi [90JA1]. In the transition zone described in the previous section the water film on the surface of the heat exchanger will be more or less stationary. The water drops that is seperated from the film on the upper tube rows fall back onto these tube surfaces as soon as the drops enter the free stream above the heat exchanger. This behaviour of the film will cause an accumulation of water on the upper tubes of the heat exchanger. In view of this, the dead area is defined as follows: (i) The first tube has a dead area of 25% independent of air velocity. (ii) The top two tube rows have a dead area of 25% for maximum air velocities in the transition zone and slightly higher.

$$1.1 \times v_{\text{crit}} < v_{\text{max}} < 1.4 \times v_{\text{crit}} \quad (3.2.15)$$

For the calculation of the spray water arresting rate, the film thickness has to be approximated. The film thickness for velocities well below the critical velocity is dependent on the fin spacing. If two neighbouring wetted surfaces are close enough, droplets on these surfaces will tend to minimize their surface and transform into pellets which contact both surfaces. It was found by O. Fischer [88FI1] that the capillary pressure can compensate for the hydrostatic pressure head equivalent to a fin gap of 3.1 mm. The film thickness for low air velocities is thus defined as follows

$$\text{For } v_{\text{max}} < 0.8 \times v_{\text{crit}}$$

$$t_f = 0.5 \times s_{\text{fin}} \quad \text{if } s_{\text{fin}} < 3.1 \text{ mm} \quad (3.2.16)$$

$$t_f = 1.5 \text{ mm} \quad \text{if } s_{\text{fin}} > 3.1 \text{ mm}$$

For higher velocities the film thickness is assumed to be negligible for the calculation of the frontal area.

With the use of the modifications discussed in the sections above, it is possible to create an accurate model for the prediction of the capacity of spray cooled heat exchangers. This

model will be discussed in more detail in Chapter 5 when the computer simulation of such a model is presented.

3.2.2. Simplified model for vertical air flow heat exchangers with extended surface

The model described in the previous section, ACCURATE, requires a large amount of computer time to produce accurate results. The simplified model presented below is not as accurate and is more dependent on the choice of the effective wetted area but produces reasonably accurate results using considerably less computer time.

This model, SIMPLE, is based on the same principles of mass and energy conservation required by the first law of Thermodynamics as the model ACCURATE but it is solved in a macroscopic sense as each pipe row of the cooler is evaluated in one step whereas the accurate model considers the heat exchanger as a homogeneous surface which is divided into any number of elements which are evaluated in turn.

The analysis centres around the determination of the value of an imaginary average temperature of the dry part of the tube wall and film temperature which is called the wall temperature. This value is obtained iteratively using the energy balance equations. The evaporation from the water film on the surface of the i 'th tube row is calculated as follows

$$m_{ev[i]} = \frac{h_c}{c_{pm}} (w_{asw[i]} - w_{a[i]}) A_o A_{ev[i]} \quad (3.2.17)$$

The latent heat transfer from the tube to the air can thus be described as

$$q_{l[i]} = m_{ev[i]} i_v \quad (3.2.18)$$

The convection heat transfer from the tube to the air is

$$q_{c[i]} = h_c (T_{wall[i]} - T_{a[i]}) A_o \quad (3.2.19)$$

and the sensible heat transferred from the water drops trapped by the tube to the tube is

$$q_{s[i]} = m_{sw[i]} A_{fr} c_{pw} (T_{wall[i]} - T_{awb[i]}) \quad (3.2.20)$$

This allows the total heat transferred from the tube to the surroundings to be expressed as the sum of equations (3.2.18), (3.2.19) and (3.2.20)

$$q_{t[i]} = q_{l[i]} + q_{c[i]} + q_{s[i]} \quad (3.2.21)$$

The new wall temperature can now be obtained by rewriting the following equation

$$q_{t[i]} = A_o U_o (T_{p[i]} - T_{wall[i]})$$

$$\therefore T_{wall[i]} = T_{p[i]} - \frac{q_{t[i]}}{U_o A_o} \quad (3.2.22)$$

When the correct value for the wall temperature has been found the inlet conditions for the next pipe row can be determined as

$$T_{p[i+1]} = T_{p[i]} - \frac{q_{t[i]}}{m_p c_{pp}} \quad (3.2.23)$$

$$i_{a[i+1]} = i_{a[i]} + \frac{q_{c[i]} + q_{l[i]}}{m_a} \quad (3.2.24)$$

$$w_{a[i+1]} = w_{a[i]} + \frac{m_{ev[i]}}{m_a} \quad (3.2.25)$$

$$m_{sw[i+1]} = m_{sw[i]} - \frac{A_c}{A_{fr}} m_{sw[i]} \quad (3.2.26)$$

To satisfy the assumption made earlier that the air will always be saturated if spray water drops are present in the air, the values for air enthalpy, air humidity and spray water mass flux are used in an energy balance equation to obtain the saturated air temperature, the saturated air humidity and the mass flux of spray water remaining in the air.

The direction of the water film flow on the surface of the heat exchanger is determined in exactly the same way as is done for the accurate model discussed earlier above. The amount of water that flows over a certain tube is calculated as follows

$$m_{wd[i]} = m_{sw[i]}A_c - m_{ev[i]} + m_{wd[i(\pm 1)]}$$

$$m_{wt[i+1]} = m_{sw[i+1]}A_c + m_{wd[i+1(\pm 1)]} \quad (3.2.27)$$

With the inlet condition for the next pipe row known, the tube wall temperature for the next tube can be calculated using equations (3.2.17) to (3.2.22). The capacity of the heat exchanger can thus be calculated as the sum of the total heat transfer values for each pipe row, multiplied by the number of tubes per pipe row.

$$Q = n_t \times \sum q_{t[i]} \quad (3.2.28)$$

Determination of the wetted area for the spray cooled heat exchanger - model SIMPLE

The main factors in the calculation of the wetted area of the heat exchanger surface are: (i) The wettability of the tube surface, (ii) the velocity of the air that flows over the tube and (iii) the amount of water that flows over the tube. The first two factors play a major role in the determination of the wetted area of the model ACCURATE discussed in the previous section, but when the wetted area of the whole tube is considered the third factor, water flow rate, plays a primary role. The wettability of the tube surface still plays a role, but this can be accounted for by recognising that the surface will never be completely wet. This leaves two variables to control the wetted area:

$$\text{If } V_{\max} > 1.1 \times V_{\text{crit}}$$

$$\text{for } \frac{m_{wt[i]}}{A_o} > 0.002 \quad \Rightarrow \quad A_{ev[i]} = 0.8$$

$$\text{for } 0.002 > \frac{m_{wt[i]}}{A_o} > 0.0005 \quad \Rightarrow \quad A_{ev[i]} = 0.2 \quad (3.2.29)$$

$$\text{for } \frac{m_{wt[i]}}{A_o} < 0.0005 \quad \Rightarrow \quad A_{ev[i]} = 0.1$$

For the low air velocity situations the wetted area is defined as

If $V_{\max} < 1.1 \times V_{\text{crit}}$

$$\text{for } \frac{m_{\text{wt}[i]}}{A_o} > 0.002 \quad \Rightarrow \quad A_{\text{cv}[i]} = 0.4$$

$$\text{for } 0.002 > \frac{m_{\text{wt}[i]}}{A_o} > 0.0005 \quad \Rightarrow \quad A_{\text{cv}[i]} = 0.2 \quad (3.2.30)$$

$$\text{for } \frac{m_{\text{wt}[i]}}{A_o} < 0.0005 \quad \Rightarrow \quad A_{\text{cv}[i]} = 0.1$$

4. HEAT/MASS TRANSFER COEFFICIENT AND PRESSURE DROP CORRELATIONS

The heat transfer coefficient, and when the surface of the heat exchanger is wet, the mass transfer and film heat transfer coefficients are required to evaluate the heat exchanger performance. These coefficients are dependent on the air velocity in the heat exchanger which, implies that a pressure drop correlation is required to determine the fan power consumption.

Because the spray-cooled heat exchanger's physical behaviour is so complex, the determination of accurate correlations for heat and mass transfer are very difficult. It would therefore be advantageous if it were possible to analyze the spray cooled heat exchanger with the use of the correlations for the same heat exchanger in dry air. In the following sections the correlations controlling the operation of the heat exchanger under discussion will be determined and the coefficients for the dry and wet operation will be discussed.

4.1 Dry operation of a finned-tube heat exchanger

For the determination of the correlations controlling the heat exchanger operation in dry air, experimental data over a range of air velocities for the heat exchanger is required. The results of experimental tests which were carried out for the heat exchanger of the present study in dry air are given in Tables 4.1, 4.2, 4.3 and 4.4.

Table 4.1 - Test 0407, Dry operation ($P_{atm} = 100.92$ kPa)

dP_t	T_{pi}	T_{po}	T_{ai}	T_{wbai}	T_{ao}	T_{wbao}	m_a	m_p
54	50.15	47.04	12.62	11.72	42.07	21.44	3.41	8.34
90	49.84	46.21	12.76	11.72	41.37	21.56	4.29	8.37
125	49.35	45.17	12.99	11.86	39.48	21.06	5.36	8.35
150	50.69	45.82	13.04	11.75	39.23	20.87	6.03	8.00
177	50.23	45.14	13.25	11.87	38.10	20.57	6.74	8.03
213	49.78	44.29	13.39	11.90	36.78	20.14	7.59	7.99
270	49.35	43.49	13.71	11.85	35.20	19.50	8.91	8.02
327	48.66	42.41	14.09	12.34	33.60	19.27	10.13	7.82
379	47.90	41.58	14.43	12.52	32.57	18.96	11.10	7.81
443	47.55	41.08	15.06	12.80	31.95	18.77	2.27	7.82
62	46.67	44.79	12.97	11.96	41.29	21.58	3.37	12.63
91	46.55	44.30	13.10	12.00	39.86	21.23	4.36	12.61
123	46.38	43.80	13.20	11.96	38.20	20.69	5.34	12.63
145	46.20	43.44	13.27	12.02	37.39	20.47	5.96	12.64
171	45.89	42.97	13.42	12.13	36.35	20.18	6.64	12.62
210	45.63	42.49	13.65	12.26	35.09	19.84	7.60	12.65
270	45.31	41.91	13.94	12.43	33.81	19.46	8.94	12.63
327	45.07	41.48	14.39	12.64	32.89	19.18	10.14	12.65
384	44.67	40.97	14.72	12.78	32.04	18.90	11.23	12.64
443	44.35	40.52	15.00	12.83	31.31	18.61	12.31	12.60

Table 4.2 - Test 0507, Dry operation ($P_{atm} = 100.85$ kPa)

dP_t	T_{pi}	T_{po}	T_{ai}	T_{wbai}	T_{ao}	T_{wbao}	m_a	m_p
78	51.77	49.00	15.53	12.44	44.91	22.07	3.87	10.12
108	51.55	48.37	15.70	12.58	43.25	21.83	4.81	10.15
136	51.32	47.84	15.91	12.72	41.63	21.34	5.62	10.16
159	51.11	47.43	15.91	12.57	40.65	20.91	6.24	10.17
195	50.92	46.92	16.01	12.58	39.54	20.66	7.15	10.15
250	50.67	46.32	16.39	12.74	37.97	20.04	8.37	10.18
302	50.38	45.76	16.70	12.94	36.80	19.82	9.50	10.13
358	49.88	45.03	16.83	13.02	35.71	19.55	10.64	10.18
413	49.39	44.48	17.39	13.27	34.86	19.32	11.64	10.15
75	48.45	42.48	15.79	12.61	40.51	20.93	3.91	3.90
103	48.32	41.58	15.87	12.64	38.76	20.31	4.77	3.91
127	48.06	40.76	15.86	12.59	37.22	19.94	5.52	3.91
153	47.94	40.19	15.93	12.60	36.04	19.57	6.22	3.92
187	47.75	39.43	15.90	12.58	34.68	19.18	7.12	3.90
242	47.64	38.72	16.15	12.74	33.22	18.76	8.42	3.93
299	47.43	38.07	16.51	12.99	32.09	18.50	9.63	3.92
352	47.28	37.56	16.79	13.11	31.22	18.20	10.72	3.90
405	47.12	37.16	17.09	13.23	30.58	17.99	11.70	3.90

Table 4.3 - Test 0607, Dry operation ($P_{atm} = 100.38$ kPa)

dP_t	T_{pi}	T_{po}	T_{ai}	T_{wbai}	T_{ao}	T_{wbao}	m_a	m_p
77	57.76	53.77	16.95	12.12	50.28	23.18	3.78	7.52
110	57.46	52.77	17.26	12.20	47.88	22.44	4.78	7.52
144	57.08	51.87	17.54	12.26	45.81	21.79	5.73	7.51
193	56.80	51.01	17.85	12.48	43.72	21.27	6.98	7.52
237	56.58	50.36	18.15	12.65	42.27	20.86	7.99	7.50
286	56.39	49.83	18.57	12.88	41.04	20.54	9.02	7.52
348	56.06	49.14	18.98	12.96	39.66	20.05	10.27	7.51
446	55.80	48.51	19.67	13.33	38.45	19.75	12.03	7.59
447	55.29	50.11	19.78	13.33	39.15	19.91	11.99	11.00
362	54.83	50.05	19.30	13.20	39.88	20.19	10.51	11.01
280	54.43	50.10	18.76	12.98	40.98	20.52	8.90	11.03
236	54.17	50.08	18.31	12.83	41.63	20.75	7.99	11.03
192	53.94	50.17	18.05	12.63	42.49	20.92	6.97	11.04
150	53.65	50.28	17.80	12.44	43.70	21.19	5.96	11.02
107	53.44	50.54	17.59	12.42	45.45	21.75	4.75	11.02
75	53.24	50.81	17.39	12.32	47.06	22.11	3.76	11.00

Table 4.4 - Isothermal pressure drop data

m_a	Δp_t
3.198	46.5
3.646	57
4.091	69
4.577	82
5.202	100
5.789	118.2
6.351	137
6.894	156
7.472	177
7.973	196
8.728	227
9.521	259.7
10.113	285.2
10.574	307.5

4.1.1 Heat transfer coefficient

The performance of the dry heat exchanger can be calculated by rewriting equation (3.2.3) as follows

$$q = U_o(T_p - T_a)dA_o \quad (4.1.1)$$

This equation can in turn be written as

$$q = F_T U_o A_o \Delta T_{lm} \quad (4.1.2)$$

where

$$\Delta T_{lm} = \frac{(T_{po} - T_{ai}) - (T_{pi} - T_{ao})}{\ln[(T_{po} - T_{ai}) / (T_{pi} - T_{ao})]} \quad (4.1.3)$$

for the counter flow heat exchanger. The correction factor F_T can be calculated by the method proposed by Roetzel [79RO1], [80RO1] and [84RO1] which is given in Appendix B. With all the other variables in equation (4.1.2) known, U_o can be calculated using the data in Tables 4.1 to 4.3. To find the air side heat transfer coefficient, equation (3.1.5) can be rewritten as follows

$$h_a = \frac{1}{\epsilon_o} \left\{ \frac{1}{U_o} - \frac{A_o}{h_p A_i} - \frac{A_o}{n_t n_u L_t} \left(\frac{t_t}{k_t P_t} + \frac{t_r}{k_r P_r} \right) \right\}^{-1} \quad (4.1.4)$$

The heat transfer coefficient controlling the heat transfer process on the inside of the tube, h_p , can be calculated with the long established Petukhov [70PE1] equation for fully developed turbulent flow in smooth pipes

$$Nu_d = \frac{(f/8) Re_d Pr}{1.07 + 12.7(f/8)^{1/2} (Pr^{2/3} - 1)} \quad (4.1.5)$$

$$f = (1.82 \log_{10}(Re_d) - 1.64)^{-2}$$

The finned surface effectivity, ϵ_o , is calculated by assuming that the elliptical finned-tube configuration is approximated by a straight fin configuration on a flat plate. This assumption and the calculation of the fin effectivity are explained in Appendix C. The correlation obtained from the data in Tables 4.1 to 4.3 were the following

$$StPr^{2/3} = 4.038075 Re_H^{-0.5205963} \quad (4.1.6)$$

The correlation's close approximation of the experimental data can be seen on Figure 4.1.

4.1.2 Pressure drop

The pressure drop correlation is calculated from the data in Table 4.4 which presents the measured pressure drop over the heat exchanger in isothermal conditions. The correlation for the friction factor is found using the well known equation

$$\Delta p = \frac{G^2}{2 \rho g_c} f_a \frac{A_{fr}}{A_c} \quad (4.1.7)$$

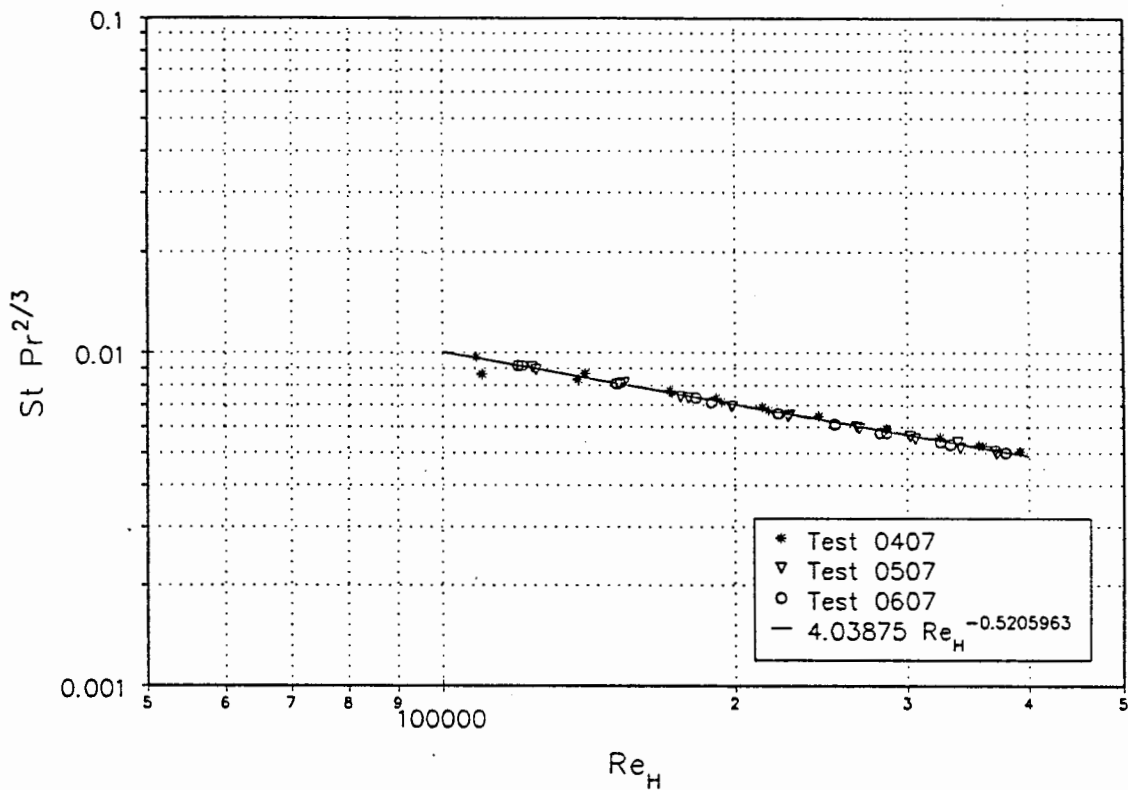


Figure 4.1 - Heat transfer coefficient for dry operation of heat exchanger.

The correlation that gives the friction factor for the present heat exchanger was found to be

$$f_h = 498.944 Re_H^{-0.419123} \quad (4.1.8)$$

The data and correlation are plotted on Figure 4.2. Using these two correlations, equation (4.1.6) and equation (4.1.8), the performance of the air cooled heat exchanger can be accurately predicted.

4.2 Spray cooled finned-tube heat exchanger

The earlier research in this field centred around the development of correlations for the enhanced heat transfer coefficient which assumes that the performance enhancement exhibited by spray cooling is the result of an increase in convective heat transfer. This meant that the correlations for each heat exchanger had to be determined experimentally. Later research, and the work done in the present study, looked at

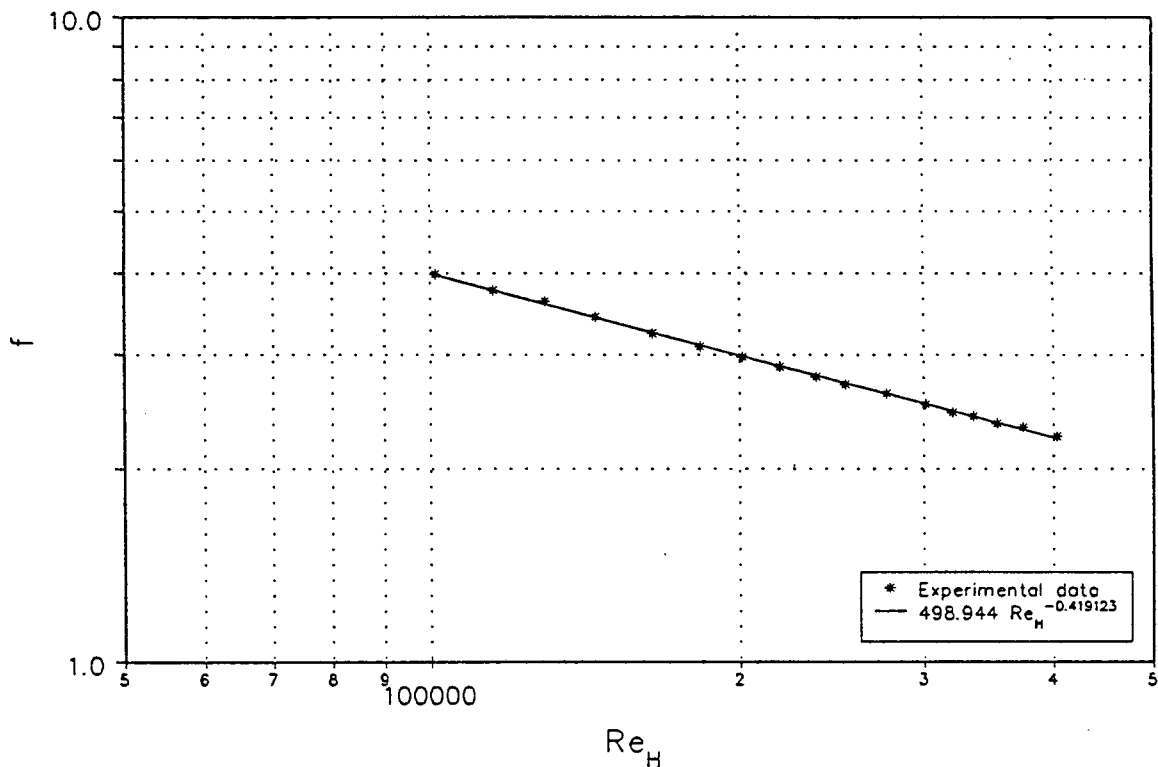


Figure 4.2 - Friction factor for the dry air operation of heat exchanger.

combined heat and mass transfer which needs no new correlations, but uses the correlations for dry operation to predict the performance of the spray cooled heat exchanger.

4.2.1 Heat transfer coefficient

Correlations for the enhanced heat transfer coefficient for spray cooled fin tube heat exchangers are not numerous as very little experimental work has been done in this field. Even less information is available regarding the film heat transfer coefficient on spray cooled extended heat transfer surfaces, but a few equations which were derived for smooth pipes are discussed.

For the film heat transfer coefficient, h_w , McAdams [54MC1] proposed the following equation for a water film on flat plates

$$Nu_w = 0.023 Re_w^{0.8} Pr_w^{0.4} \quad (4.2.1)$$

$$\text{Re}_w = \frac{v_w L \rho}{\mu} \quad (4.2.2)$$

This equation indicates that a film heat transfer coefficient of approximately 1000 W/m²K is applicable for $\text{Re}_w = 5000$.

A different approach by Mizushina [66MI1] for the calculation of film heat transfer coefficient in a counterflow horizontal tube evaporative cooler is

$$h_w = 2202.9 \left(\frac{\Gamma}{d_o} \right)^{1/3} \quad (4.2.3)$$

with

$$0.195 < \frac{\Gamma}{d_o} < 5.556$$

These equations is not applicable for the small amounts of water that spray cooling uses but it gives a minimum value for the film heat transfer coefficient of 1200 W/m²K that was found in experimental work on wet surfaces done by Mizushina et al.

Oshima's [72OS1] experimental work on compact heat exchangers assumed that the heat transfer coefficient, h_a , is replaced by an apparent heat transfer coefficient, $h_{a,app}$ which takes the convective and latent heat transfer into account. A constant value of 5500 W/m²K for the film heat transfer coefficient was used as Oshima correctly assumed that it is not the controlling resistance and contributes less than 10% to the total resistance.

Simpson, Beggs and Sen [74SI1] found the following correlation for the heat transfer coefficient of a single finned-tube row

$$\text{Nu}_a = 0.0059 \text{Re}^{0.94} \text{Pr}^{1/3} \left(\frac{s_f}{t_f} \right)^{0.12} \quad (4.2.4)$$

This equation is valid for air velocities ranging from 4.5 m/s to 9.1 m/s and surface configurations with fin height varying from 9.5 mm to 22.2 mm and fin spacing varying from 3.2 mm to 14.3 mm.

More experimental work on compact heat exchangers were done by Yang and Clark [75YA1] who did not publish any correlations. Their work was based on the same principle as Oshima's namely that the increase in heat transfer in the spray cooled heat exchanger is attributed solely to an increase in the convective contribution.

Experiments by Tree, Goldschmidt, Garrett and Kach [78TR1] produced the following empirical equation for a finned-tube bank four staggered tube rows deep.

$$\frac{Nu_{wet}}{Nu_{dry}} = \frac{63.3 m_w [1 - 0.0583 m_w (d_d \times 10^6)^{0.04}]}{Re_H} \quad (4.2.5)$$

This equation indicates a heat transfer coefficient enhancement of 40% for their experimental model. It is however not relevant for any other heat exchanger than the specific one used in their experiments, as the spray water mass flow rates in equation (4.2.5) represents the total amount of spray water that flows onto the heat exchanger coil.

A different approach used by Kreid, Parry, McGowen and Johnson, centres around the analogy between heat and mass transfer. The heat transfer that takes place in the spray cooled heat exchanger is described as:

$$q_t = U^* A_o \frac{\Delta i_{lm}}{c_{pa}} \quad (4.2.6)$$

In this equation no mass transfer coefficient is needed, but the overall resistance U_o is transformed to a resistance for enthalpy transfer:

$$\frac{1}{U^*} = \xi \left\{ \frac{A_o}{h_p A_i} + \frac{t_i A_o}{k_t A_i} + \frac{1}{e^* h_a^*} \right\} \quad (4.2.7)$$

where

$$\xi = \frac{i_{aspi} - i_{ai}}{c_{pa}(T_{pi} - T_{ai})} \quad (4.2.8)$$

$$h_a^* = \frac{\xi h_a}{1 + Bi_w} \quad (4.2.9)$$

$$Bi_w = \frac{\xi h_a}{h_D} \quad (4.2.10)$$

The problem with this method for the prediction of performance of spray cooled heat exchangers, is its assumption that the heat transfer surface is completely wet, which results in overestimation of the performances of spray cooled heat exchangers with spray water rates that are not high enough to sustain a stable spray water film on the surface of the heat exchanger.

The enthalpy difference as heat transfer potential is also used by Kuwahara and Nakayama [81KU1] to predict the heat transfer from a spray cooled finned-tube. This empirical equation deduced by them also takes the mass flux of the spray water and air velocity into account.

$$\frac{q_t}{\Delta i} = 6.7 \times 10^{-6} \Delta i^{-0.2} \left(\frac{m_{sw}}{3600 \times A_{fr}} \right)^{0.25} v_a^{0.62} \quad (4.2.11)$$

This equation is valid for a tube geometry with 1 mm high fins spaced at 0.7 mm intervals.

Simpson, Beggs and Raouf [84SI1] determined a correlation for the enhanced heat transfer coefficient for a bank of vertical finned-tubes in air-mist flow.

For $45^\circ \geq (T_{wall} - T_a)$

$$Nu_{TP} = Nu_{SP} + 2.144 Re_a^{0.488} Re_w^{0.317} \lambda_{TP} \quad (4.2.12)$$

$$\lambda_{TP} = \left(1 + \frac{L_f}{d_r}\right)^{-2.18} \left(1 + \frac{d_r}{s_f}\right)^{-0.637} \left(\frac{P_t}{d_r}\right)^{0.842}$$

For $45^\circ < (T_{wall} - T_a)$

$$Nu_{TP} = Nu_{SP} + 3.232 Re_a^{0.337} Pr_w^{0.711} \lambda_{TP} \quad (4.2.13)$$

$$\lambda_{TP} = \left(1 + \frac{L_f}{d_r}\right)^{-3.508} \left(1 + \frac{d_r}{s_f}\right)^{-0.887} \left(\frac{P_t}{d_r}\right)^{1.439}$$

These equations are valid for 1.6 mm thick fins with geometries as follows: Fin spacings from 4 to 20.8 mm and fin lengths from 5 to 15 mm.

Nakayama, Kuwahara and Hirasawa [88NA1] assume that the Lewis factor is equal to one, and use this to calculate the mass transfer coefficient as:

$$h_D = \frac{h_a}{c_{pa}} \quad (4.2.14)$$

The convective heat transfer coefficient, h_a , on the dry area as well as on the surface of the water film, is given by the correlation of the single phase heat transfer coefficient. To compensate for low spray water mass fluxes, an effective wet area model is proposed to prevent overestimation.

4.2.2 Pressure drop

The experimental work done by the majority of researchers provides few correlations for the two phase pressure drop over spray cooled heat exchangers. The general consensus seems to be that the increase in pressure drop due to spray cooling of a finned-tube bank, is negligible. This would seem logical for heat exchanger geometries with wide fin spacing or large tube pitches, but for the closely spaced fins and tubes pitched closely together as is the case in the present study, the water holdup will be substantial and this will result in increased pressure drop over the heat exchanger.

Simpson, Beggs and Raouff found the following empirical correlation for the friction factor in bank of vertical finned-tubes:

$$f = 76.062 \text{Re}_a^{-0.654} \psi_{\text{TP}} \quad (4.2.15)$$

$$\psi_{\text{TP}} = \left(1 + \frac{L_f}{d_r}\right)^{0.959} \left(1 + \frac{d_r}{s_f}\right)^{0.309} \left(\frac{p_t}{d_r}\right)^{-0.282}$$

With the use of this equation and the correlation for the single phase friction factor, the two phase multiplier, ϕ , is determined

$$\phi = 187.17 \text{Re}_a^{-0.511} \left(1 + \frac{L_f}{d_r}\right)^{0.122} \left(1 + \frac{d_r}{s_f}\right)^{0.178} \left(\frac{p_t}{d_r}\right)^{-0.282} \quad (4.2.16)$$

Equation (4.2.16) confirm the possibility of a two phase pressure drop that will be up to 60% higher than the single phase pressure drop for the configuration of fins, 1.6 mm thick, 10 mm high and spaced at 4 mm intervals.

Fischer and Sommer [88FI1] measured the water holdup weight in their experimental study and found that there exists a maximum holdup point at an air mass-flow rate of approximately 2 kg/m²s. At this critical air velocity, the pressure drop over the spray cooled coil was 100% higher than the pressure drop observed over the dry heat exchanger at the same air velocity.

5. COMPUTER SIMULATION

Three computer programs, based on the models presented in Chapter 3, were written in Turbo Pascal to simulate the operation of a spray cooled heat exchanger. The source codes for these programs are presented in Appendix H.

The first program is based on the analytical model presented by Kreid et al [78KR1]. The objective of this program is to evaluate the enthalpy potential model and determine a maximum performance enhancement figure based on the assumption that the surface of the heat exchanger is completely wet.

The second and third programs are iterative simulations of the spray cooled heat exchanger based on the simplified and accurate models discussed in Chapter 3. The predictive capabilities of these two programs are compared to the experimental data in Chapter 7.

The water and air properties which are required for the prediction of heat and mass transfer in the spray cooled heat exchanger, are calculated with the use of the equations presented in Appendix A.

5.1 Kreid model

The model requires an iterative program which successively solves equation (4.2.6) for the determination of the outlet conditions. This equation is repeated here for convenience

$$q = F_t U_o A_o \frac{\Delta i_{lm}}{c_{pm}} \quad (5.1.1)$$

The value of U_o is determined with the aid of equation (4.2.7). The execution of the program starts by assuming an approximate value for the outlet air enthalpy. By using this value to determine the average air and spray water temperatures, the properties of the water and air can be calculated. To start with, the outlet air is assumed saturated at the inlet temperature of the process water and the outlet enthalpy of the air can now be calculated. The heat transfer from the process water to the air is determined by multiplying the enthalpy change of the air by the mass-flow of the air. This value is used to calculate the process water outlet temperature. With the inlet and outlet conditions of the air and process

water known, the Log-Mean-Enthalpy-Difference can be determined as:

$$\Delta i_{lm} = \frac{(i_{aspi} - i_{ao}) - (i_{aspo} - i_{ai})}{\ln[(i_{aspi} - i_{ao}) / (i_{aspo} - i_{ai})]} \quad (5.1.2)$$

The values of U_o and F_t are now calculated (see Appendixes B and C) and the new value of the heat transfer from the process water to the air is determined with the use of equation (5.1.1). This value will show whether the approximated value assumed for the air outlet enthalpy was too low or too high. The performance of the heat exchanger can be determined by adjusting this value until the heat transfer from the enthalpy increase of the air equals the heat transfer calculated with equation (5.1.1). The flow chart of this program is shown in Figure 5.1.

With the use of this program, it was possible to determine a maximum theoretical performance enhancement for any set of inlet conditions. This was done for the experimental heat exchanger examined in present study (see Appendix E) and the following inlet conditions:

Process fluid inlet temperature	=	28 - 64 °C
Air inlet condition:		
Dry bulb Temperature	=	25 °C
Wet bulb Temperature	=	19.5 °C
Mass-flow rate	=	2 - 10 kg/s

The result of this calculation, presented in Figure 5.2, shows that the performance enhancement decreases with increasing air velocity and increasing process water temperature. If the air velocity and air inlet conditions are held constant and the process water inlet temperature is changed, which effectively changes the enthalpy potential, it can be seen from Figure 5.2 that the performance enhancement initially decreases sharply but becomes almost constant at higher temperatures. The high performance enhancement that occurs at low air velocity and low process water temperature, is the result of the wet surface utilising the greater potential for heat transfer that the wet bulb temperature of the air offers. As the process water temperature increases, the difference between the wet bulb and dry bulb potential diminishes. Typical values for the performance enhancement ranges from 1.4 to 12 times the dry values. These values are in line with the analytical and experimental

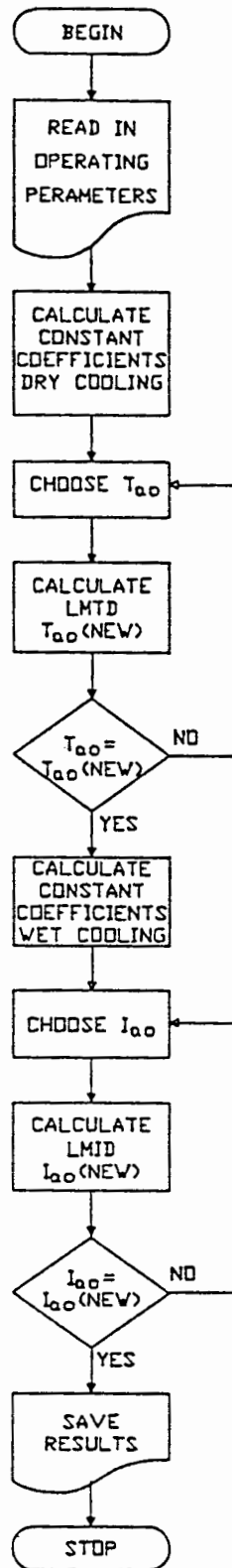


Figure 5.1 - Program logic used for the Kreid computer model.

heat transfer enhancement values obtained by other researchers in the field of spray cooled finned-tube heat exchangers.

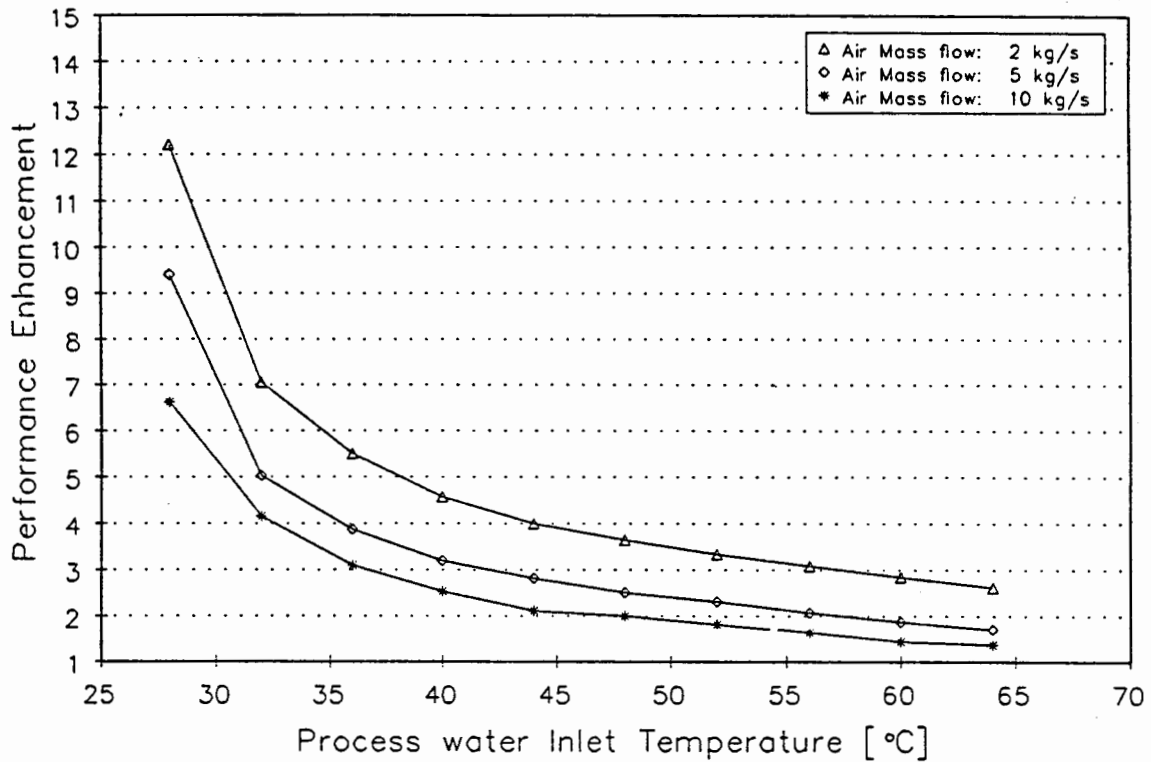


Figure 5.2 - Comparison of wet and dry performance of a heat exchanger.

5.2 Model SIMPLE for the vertical air-flow spray cooled heat exchanger

A one-dimensional model, which only considers the changes in the vertical direction, is sufficient for the simulation of the vertical air-flow spray cooled heat exchanger. This model, SIMPLE, is discussed in section 3.2.2 and the computer program uses the equations given there to simulate the vertical air flow spray cooled heat exchanger.

The exchanger which is evaluated is basically a counterflow setup where the air and mist flows in at the bottom of the heat exchanger and the process water enters at the top tube row of the heat exchanger. This means that at either end of the heat exchanger only some of the conditions are known. As two of the participants, air and spray water, enter at the bottom this seems like the best place to start the calculation process. The program steps

from one tube row to the next, using the calculated outlet conditions of the one tube row as the inlet conditions for the next tube row.

Using equation (3.2.22), the inner iterative loop of the program called the "Wall Temperature Calculation Loop" or WTCL, determines the "average" wall temperature for a specific tube. This fictitious temperature includes the effect of the wet and dry regions and assumes that the dry wall temperature and water film temperature is identical. In this loop the iterative procedure used is the well known interval halving method as there are no instabilities which inhibit the convergence of this loop.

The outer iterative loop determines the outlet temperature of the process water. To start the program execution, an approximate value for the outlet temperature of the process fluid is assumed and using this value to give a complete set of initial values, the program steps through the exchanger until it reaches the top row. The inlet process water temperature that results from the assumed outlet temperature is now compared with the known temperature of the process water at the top of the heat exchanger and the outlet temperature is varied accordingly until the calculated inlet process water temperature is equal to the required value.

The iterative procedure which is used for the outer loop is based on the direct dependence of the process water outlet temperature on the process water inlet temperature. As the inlet temperature of the process water increases, its outlet temperature will increase, but the heat transfer that takes place also increases as a result of the increase in heat transfer potential. The difference between the known value of the process water inlet temperature and the value obtained by stepping through the heat exchanger, is used to determine the correction to the outlet temperature of the process water as follows:

$$\text{If } (T_{pwi,real} - T_{pw,cal}) > 10$$

$$T_{pwi,new} = T_{pwi,old} + \frac{(T_{pwi,real} - T_{pwi,cal})}{3} \quad (5.2.1)$$

$$\text{If } 10 > (T_{pwi,real} - T_{pw,cal}) > 0.1$$

$$T_{pwi,new} = T_{pwi,old} + \frac{(T_{pwi,real} - T_{pwi,cal})}{\sqrt{3}} \quad (5.2.2)$$

If $(T_{pwi,real} - T_{pwi,cal}) < 0.1$

$$T_{pwi,new} = T_{pwi,old} + \frac{(T_{pwi,real} - T_{pwi,cal})}{3} \quad (5.2.3)$$

This method of iteration is necessary to ensure convergence as the model has built-in instabilities i.e. the step like form of the effective area function which prevents convergence in methods like the interval-halving method.

The spray water is considered to be in equilibrium with the air as it moves through the heat exchanger, which means that the air will stay saturated as long as drops of spray water remain present in the air. These droplets will assume the wet bulb temperature of the air at all times. After each tube row the conditions of the air and spray water are modified to ensure that the assumption of equilibrium conditions is maintained. As a result of this procedure, evaporation of the spray water drops on the heat exchanger surface and in the air is considered as heat is transferred from the heat exchanger surface to the air.

The fraction of the spray water that becomes part of the water film on the tube is determined as the product of projected frontal area of the tube and the density of the spray water in the air. The projected frontal area, A_{fr} , includes the thickness of the water film on the tube and fins to ensure that when the water film flows downwards and water pellets form between the fins, the spray water arresting rate increases accordingly. The spray water density of the droplets left in the air, is always considered to be homogeneous at any horizontal plane in the heat exchanger to comply with the assumptions made in section 3.1.

If the air velocity is lower than the critical velocity, v_{crit} which was defined in section 3.2.1, the spray water will be falling downwards through the heat exchanger and the mass flow of the spray water falling from the bottom tube will be unknown. This means that an extra iteration loop is required to ensure stable water film flow rates. However, it was found that it is not necessary for a separate iteration to determine the mass flow of the spray water as it will converge automatically after the amount of iterative loops exceed the number of pipe

rows. To ensure an accurate answer, each iteration is repeated without changing the initial values, as many times as there are tube rows in the heat exchanger. This slows the program down considerably, but prevents conditions of instable film flow and film temperature which can prevent convergence. A flow chart of this program is presented in Figure 5.3.

5.3 Model ACCURATE for the vertical air flow spray cooled heat exchanger

The four differential equations of the accurate model discussed in section 3.2.1 are solved simultaneously using the well known 4th order Runge-Kutta method. This method and an example calculation of the accurate model are presented in Appendix D.

This model considers the heat exchanger as a homogeneous surface which does not consist of a distinct number of finned-tube rows, but any chosen number of identical elements. If each tube row is divided into five elements as the one shown in Figure 3.1, it is important to understand that only a fifth of the process water that flows through a tube will flow through an element that stretches over the length of the heat exchanger and has an area one fifth of the tube area, but all the air and all the spray water in the water film on the tube surface flow over each element. This means that the process water inlet temperature of each element of a tube is identical to all the other elements of the tube and the process water outlet temperature of each tube is the average of the process water outlet temperatures of its elements. On the other hand, the temperatures of the water film on the tube and the air that leaves the heat exchanger is equal to the outlet temperatures of the last element of the last tube.

As the flowchart in Figure 5.4 shows, the simulation of the spray cooled heat exchanger with the accurate model requires constant control of the calculation process to ensure that the correct mode (1,2 or 3 as discussed in section 3.2.1) of heat transfer is being used for any element on the heat exchanger. The choice of the heat transfer mode is managed by the innermost loop of the program.

This loop called the "Simultaneous Differential Equation Solver", or SDES, is concerned with solving one of the three sets of equations for the three modes of heat transfer depending on the condition of the element. The mode of heat transfer is determined from the basic description of each mode: i) Wet heat exchanger surface and super-saturated air, ii) Wet

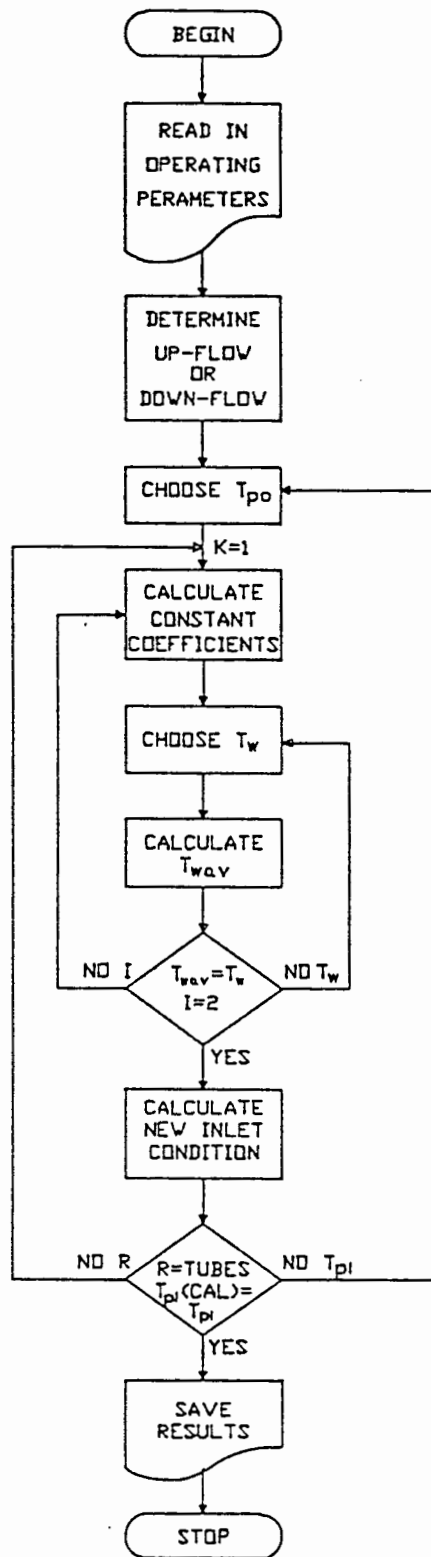


Figure 53 - Program logic used for the simulation of the vertical air-flow spray cooled heat exchanger with the simplified model.

heat exchanger surface and non-saturated air and iii) Dry heat exchanger surface. SDES determines the condition of the surface and the air and then chooses the appropriate set of differential equations to be solved for the element. The Runge-Kutta method solves these equations in four steps over the element, and the condition of the air and heat exchanger surface is verified after each step. If the condition of the air or heat exchanger surface should change after any of the first three steps, SDES terminates the solving process and changes the mode of heat transfer accordingly. It must be noted that if the mode change is from mode 1 or 2 to mode 3 (i.e. when the heat exchanger surface dries out), the excess water on the surface that originally triggered one of the wet surface modes, is allowed to evaporate into the air by adding this water to the spray water drops in the air. If the mode change is from mode 1 to mode 2 the excess water in the air that originally triggered the super-saturated air mode is allowed to evaporate from the surface of the heat exchanger into the air.

The calculation of the film temperature is done iteratively using equation (3.2.2). As in the program for model SIMPLE, this loop called the "Wall Temperature Calculation Loop", or WTCL, uses the interval halving method to ensure convergence of the film temperature. The temperature of the water film that flows over the element is considered to be constant and equal to the average temperature that is calculated in the WTCL routine. The film temperature change that occurs over an element is approximated by the difference between the average film temperature of the element and the average film temperature of the element from which the water film flows onto the element under consideration.

If the velocity of the air is below the critical velocity, the flow direction of the water film is downwards over the tube surface. This fact is ignored by the computer program in the elemental evaluation of the heat exchanger but still holds in the macroscopic evaluation. A simplified flow mode for the downward flowing film was developed in which the computer program considers the flow direction of the water and air to be in the same direction over each element and all the elements of a tube, but the excess water from each tube flows downwards onto the tube below (see "Simplified downward flowing film" in Figure 5.5). The amount of water that flows over the tube (this is the same amount of water that flows onto the first element of the tube) is in this case the sum of the fraction of the spray water that strikes the tube surface and the water falling from the last element of the tube above. When the water flows upwards, the water that flows onto the first element of the tube is the sum of the fraction of spray water that hits the tube and the water flowing from the last element

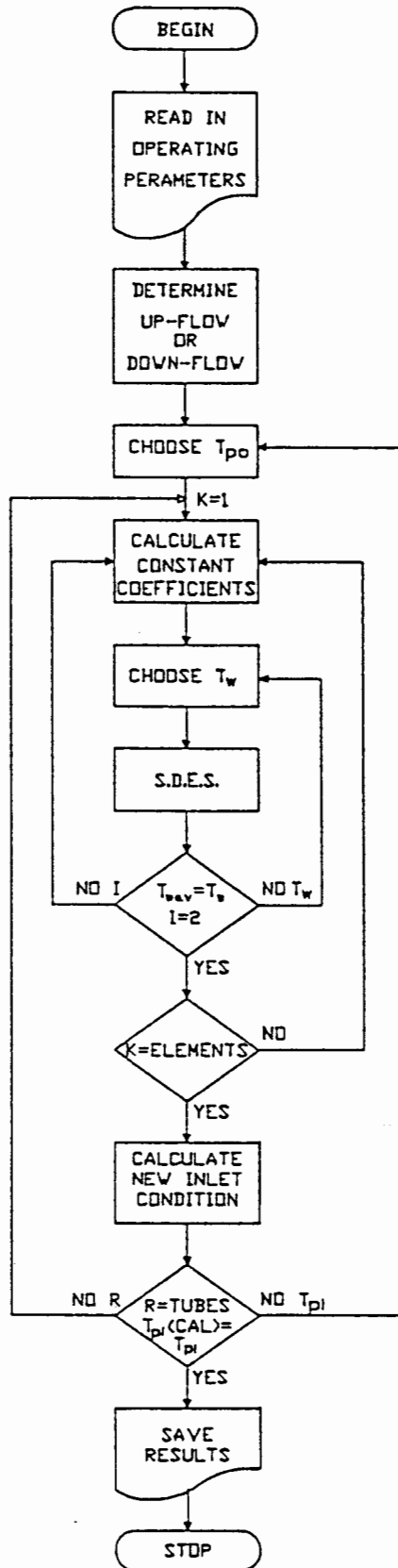


Figure 5.4 - Program logic used for the simulation of the vertical air-flow spray cooled heat exchanger with the accurate model

of the tube below. The film flow modes, normal downward flowing film, simplified downward flowing film and upward flowing film, are graphically compared in Figure 5.5.

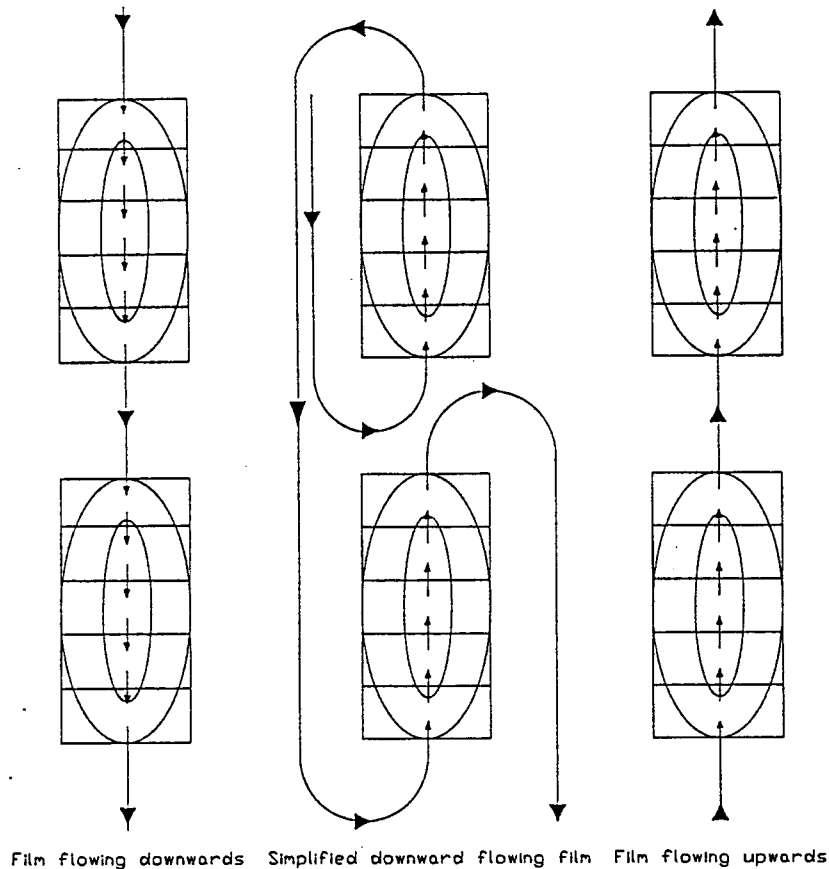


Figure 5.5 - Graphical description of the falling and rising film handling by the accurate model program for vertical air-flow spray cooled heat exchangers.

The fact that the flow direction of the water film in the case of a downward flowing film is upwards over the elements, is important to ensure that the calculation process is kept simple and the execution time is minimized, by eliminating an extra iterative loop for the determination of the inlet mass-flow rate and temperature of the water film that flows over the tube. An additional advantage of this simplification is that the same routines can be used in the program to simulate both the upward and downward film flow directions. It was found that by using the correct flow direction (see Figure 5.5) for the downflow mode, a slightly lower performance for the heat exchanger is obtained. The difference in the calculated value of the process water outlet temperature using the physically correct flow direction is less than 0.2%, but the computing time increases more than 10 times depending on the number of elements used for each tube row. This savings in program execution time justifies the use of the simplified film flow in this program.

As in the program for the model SIMPLE, the outer loop determines the outlet temperature of the process water. The procedure for the iteration process is the same as described in section 5.2 and the correction value is determined using equations (5.2.1) to (5.2.3).

Although the simulation programs were written specifically for heat exchanger rating, both the programs for the accurate and simplified models can easily be adapted to include a heat exchanger selection option as well. As these programs were written specifically to compare the model's performance with results from spray cooling experiments, they were adapted to read the experimental input data automatically and continuously from the files created by the data collection program which is discussed in the next Chapter.

6. EXPERIMENTAL STUDY OF VERTICAL AIR-FLOW FINNED-TUBE HEAT EXCHANGER WITH SPRAY COOLING

The literature discussed in Chapter 2 contains very little experimental data for horizontal air-flow and even less for vertical air-flow spray cooled heat exchangers. The experimental work that was carried out in the present study is important both for the verification of the analytical models presented in Chapter 3, and the better understanding of the physical behaviour of spray cooled heat exchangers.

Various factors influence the performance of the spray cooled heat exchangers including the spray water mass flux, the process water temperature, the air mass flow rate, ambient conditions and heat exchanger geometry. To permit the simultaneous measurement of these variables during operation of the heat exchanger, a test tower was erected at the Department of Mechanical Engineering at the University of Stellenbosch.

6.1 Description of the experimental apparatus

The schematic layout of the vertical test section of the experimental apparatus and a photograph of this section is shown in Figures 6.1 and 6.2. The vertical test section has a plan area of $1.5 \times 1.5 \text{ m}^2$ which was reduced for the present experiment to a plan area of $1.4 \times 1.1 \text{ m}^2$. The total height of the tower is 5.5 m and the length of the test section is approximately 3.5 m. In order to minimize radiation influences the outside walls of the tower is covered with reflective insulation sheets (see Figure 6.2). Visual observation of the upper and lower surfaces of the heat exchanger was made possible by installing Perspex windows in the test section walls.

The heat exchanger has a frontal area of 1.54 m^2 and its length in the direction of the air-flow is 0.218 m. It consists of elliptical steel tubes arranged in a closely packed four pass configuration. The elliptical type L-fins are 9.5 mm long and have been tension wound onto the tubes at a pitch of 2.5 mm. To establish good thermal contact and increase corrosion resistance the finned-tubes were galvanized. A detailed description of the heat exchanger measurements are given in Appendix E.

Ambient air is drawn through the horizontal test section by a centrifugal fan and forced over the heat exchanger. The air mass-flow rate is controlled by means of a stepless electronic

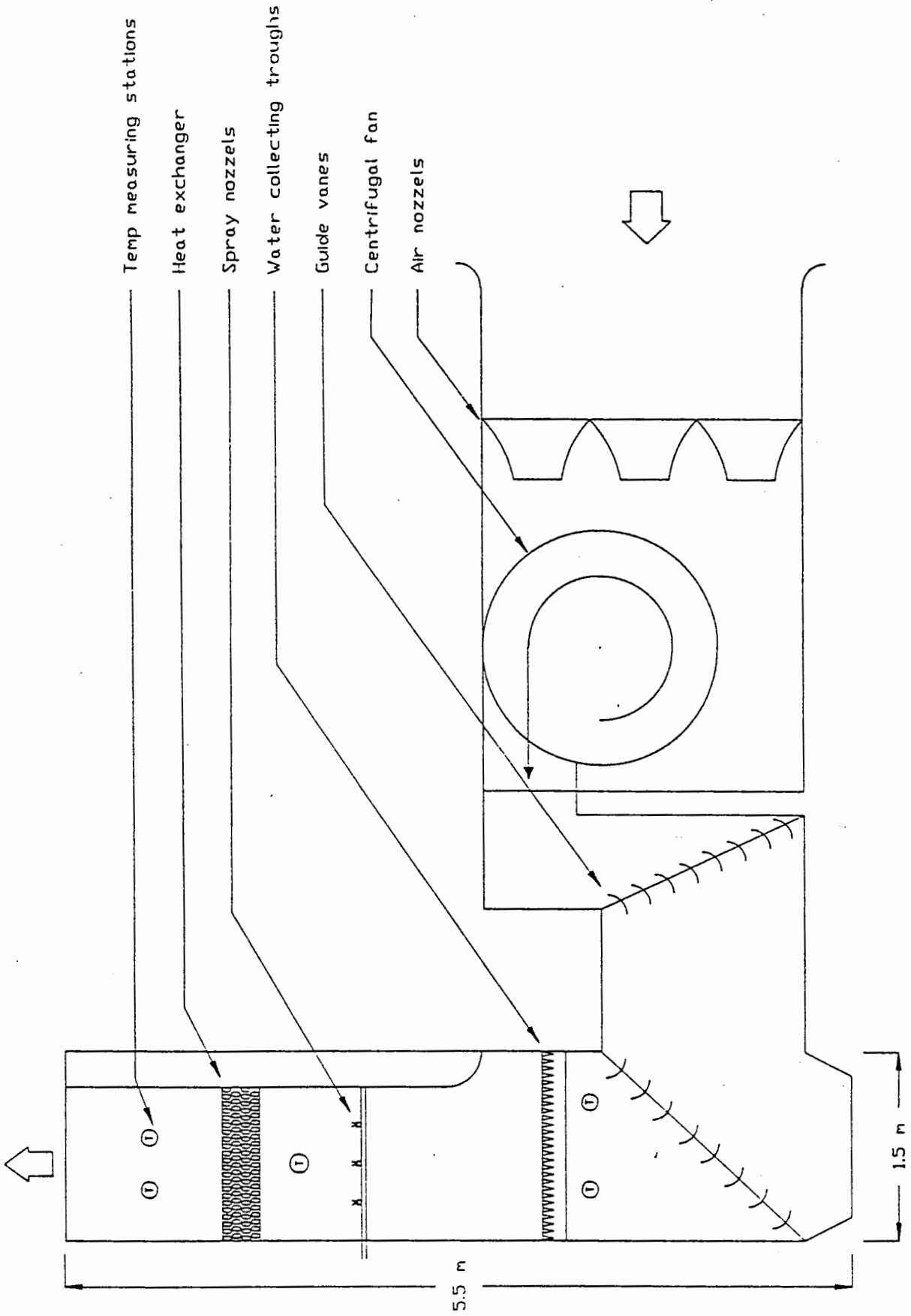


Figure 6.1 - Schematic layout of the vertical test section.

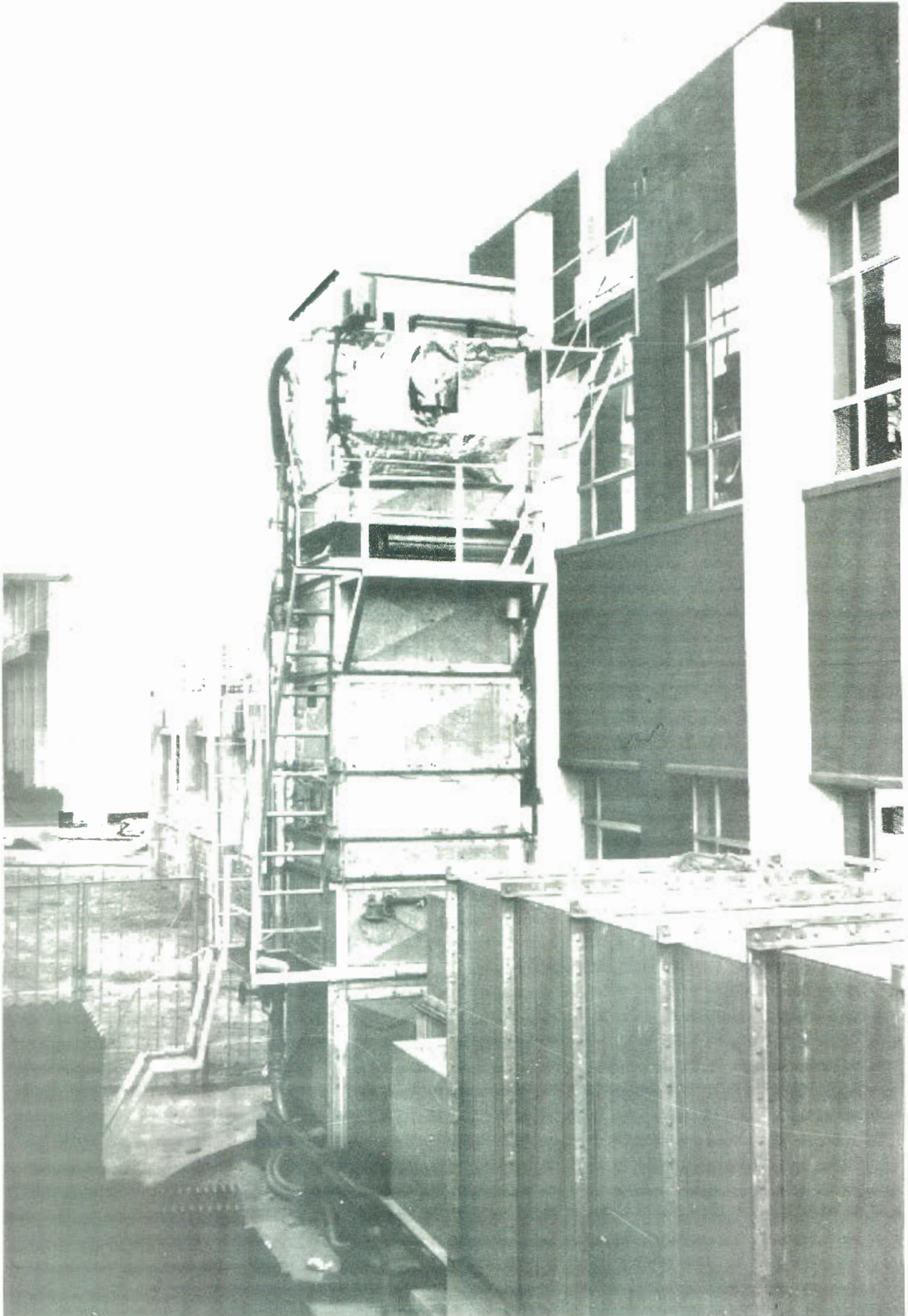


Figure 6.2 - Experimental apparatus.

speed control device which governs the speed of the centrifugal fan motor giving an infinitely variable control. The inlet and outlet air conditions are measured just below the water collecting troughs and approximately 1.5 m above the heat exchanger respectively.

During the wet experiments a fine mist was carried across the heat exchanger by the air from twelve nozzles placed about 0.6 m below it. The spray water system consists of a water supply tap, a high pressure centrifugal pump, a rotameter and a spray distribution manifold and sprayers. The water that drips down from the heat exchanger surface falls into the water collecting troughs and flows from there to a small reservoir under the tower. To prevent excessive fouling of the heat exchanger surface, this water was not recirculated through the spray water system. The troughs are spaced in such a way that most of the descending water is prevented from falling past this section by the counter-flowing air stream (see Figure 6.1). The mass flow and temperature of the spray water flowing from the test section in this way was not measured as it contributes very little to the heat transfer. The temperature of the spray water was measured only at the inlet point of the distribution manifold. To determine the condition of the air and spray water mist just before it reaches the heat exchanger, a single temperature measuring station was placed between the sprayers and the heat exchanger.

The process water is heated by means of a diesoline burning water heater and stored in a 40 m³ underground reservoir. This reservoir holds a large amount of hot water making steady state heat exchanger measurements possible over a reasonable period of time. Hot water is pumped from the surface of the reservoir to the heat exchanger and on return it is mixed with hot water from the boiler as it flows back to the bottom of the reservoir. This procedure ensures that the water in the underground tank stays stratified. The process water enters the heat exchanger at the upper manifold from where it flows through the top tube row before entering the next manifold which leads to the second tube row. After passing through the bottom tube row the water collects in the lower manifold and flows back to the underground reservoir. The capacity of the boiler is much lower than that of the heat exchanger resulting in a gradual temperature decline in the reservoir during testing, and long periods of reheating are necessary. The temperature of the process water was measured at the inlet and outlet manifolds and after each pass of the heat exchanger.

6.2 Instrumentation and data accumulation

The following variables from each test are required for a complete analysis of the experimental heat exchanger's performance:

- a) Process water - mass-flow rate, inlet and outlet temperature, temperature after each pass of the heat exchanger.
- b) Spray water - mass-flow rate, inlet temperature
- c) Air - mass-flow rate, inlet and outlet temperature (dry and wet bulb), pressure drop across heat exchanger.
- d) Other - Atmospheric pressure.

6.2.1 Instrumentation

i) Mass-flow measurements

The process water mass-flow was measured using an orifice plate placed in the supply line between the underground reservoir and the test section. The plate was made and installed according to the BS-1042 standard. The pressure difference across the plate was recorded with two Foxboro differential pressure transducers; one covering the low pressure range and the other the high pressure range. This allowed a wide mass-flow range to be measured without having to change the orifice plate. The pressure transducer delivers a 20 to 40 mA signal which is passed through a high precision 250 ohm resistor in order to obtain a voltage signal between 1 and 5 Volt. The pressure transducer was calibrated using a weighing drum and stopwatch.

The mass-flow of the spray water was measured using a G.E.C. series 2000 rotameter and two size 24 floats. For the high mass-flow rates a stainless steel float was used and for low mass-flow rates a ceramic float. With the use of these two floats mass flow rates as low as 0.012 kg/s (30 kg/m²h) and as high as 0.2 kg/s (450 kg/m²h) could be measured accurately. The rotameters were calibrated with the different floats using a weighing drum and stopwatch. A simple third order polynomial curve was fitted to the data.

The mass-flow rate of air across the heat exchanger was determined by measuring the pressure drop across five standard elliptical nozzles which are positioned upstream of the fan. These nozzles were constructed according to the ASHRAE 51-57 standard [75AS1].

The pressure drops across these nozzles were recorded with the use of a low pressure differential pressure transducer. A Betz manometer connected to the same pressure lines, was used to calibrate this pressure transducer. At low air mass-flow rates one or more of the nozzles were closed up to ensure that the differential pressure transducer never operated too close to its lower limit.

ii) Temperature measurements

All temperature measurements were made with calibrated copper-constantan thermocouples. The voltage signal of the thermocouples were measured at two temperatures, the melting point of ice to determine the zero deviation, and the boiling point of water at atmospheric pressure. This procedure ensured that the copper-constantan calibration curve used by the data logging device, the Digilink, was corrected for each thermocouple at the limits of temperature measurement for the present study.

The temperature of the process water was measured at five points in the heat exchanger with two thermocouples at each point. Two pairs of thermocouples were placed in the inlet and outlet manifold of the heat exchanger. Each pass of the heat exchanger ends in a manifold where the water from all the tubes was collected and returned to the next tube pass. Two thermocouples were placed in each of these manifolds to measure the temperature change of the process water over each pass.

The spray water inlet temperature was measured with two thermocouples at the inlet of the distribution manifold. The temperature of the spray water showed little variation and was usually between 10 °C and 20 °C.

The condition of the air was determined by measuring its wetbulb and drybulb temperatures at the inlet and outlet stations (see Figure 6.1) in the test section. A typical measuring unit consists of a plastic sampling tube containing wet and dry bulb thermocouples. The wick of the wet bulb thermocouple is kept wet by means of a small reservoir containing water. Such a unit is shown in Figure 6.3. The small water reservoir is kept full at all times from a larger outside reservoir which sustains several such units. Air from the immediate surroundings of the unit is drawn over the thermocouples at approximately 3 m/s by means of a small fan and is returned to the test section directly downstream of the measuring units. In order to obtain reliable average air temperatures, four evenly spaced units were installed at each measuring station. The outlet air measuring units are exposed to radiation from the

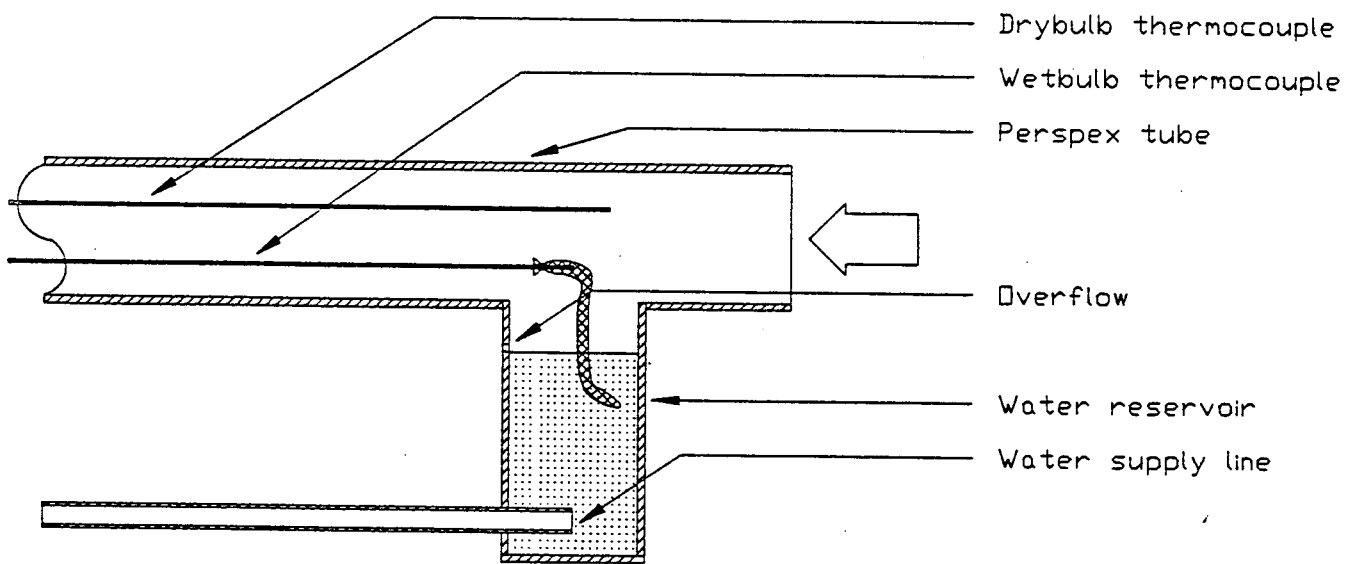


Figure 6.3 - Air temperature measuring unit.

surroundings and were covered by radiation shields after it was found that even indirect radiation increased the measured values of these units by up to 2 °C.

iii) Pressure drop measurements

The measurement of the two phase pressure drop of the air-mist mixture flowing over the heat exchanger is complicated by the positive pressure in the test section. Water on the walls of the tower is forced into the pressure lines by the positive air pressure leading to erroneous pressure measurements. Special pressure tappings were designed to trap the water before it could get into the pressure lines as shown in Figure 6.4. The water which flows through the air vent in the wall, is trapped in the reservoir which has a constant level determined by the height of the overflow tube. Before testing commences, these reservoirs have to be filled with water to ensure correct readings. These pressure tappings are situated 0.5 m below and above the heat exchanger. The pressure drop is recorded using a Betz manometer.

iv) Atmospheric pressure

The atmospheric pressure is recorded before every experiment with a mercury column barometer.

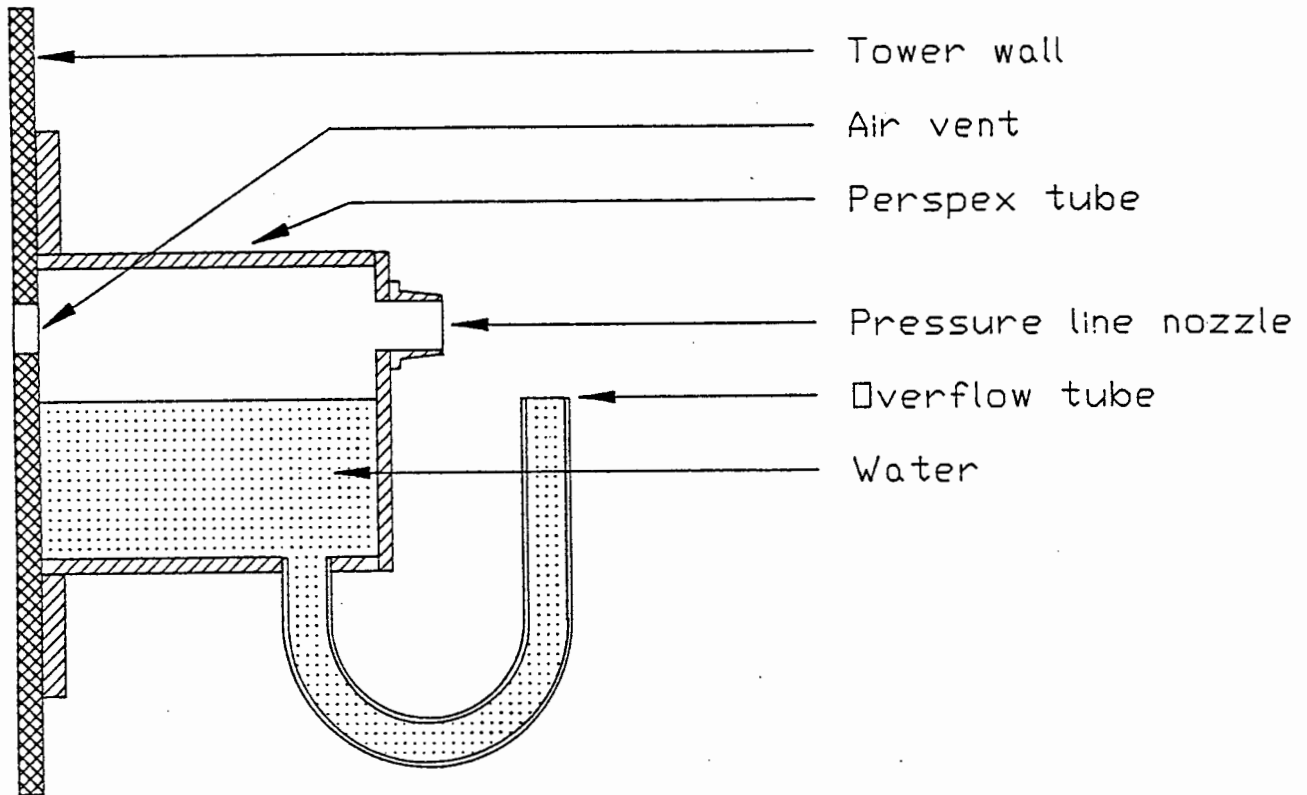


Figure 6.4 - Pressure tapping

6.2.2 Data acquisition

The system used for data acquisition has been described previously by Dreyer [88DR1] and Schultz [90SC1] but for clarity it is briefly discussed below.

The data logging system shown in Figure 6.5 uses a data logger to record all the voltage signals of the thermocouples and pressure transducers. These signals are converted to temperatures and pressures respectively and transferred to a personal computer via a RS232 connection. A data logging program written in Turbo Pascal continuously displayed the measurements and energy balance in graphical form on the screen. This allowed interpretation of the data during tests minimizing the risk of recording bad data. Only the atmospheric pressure and pressure drop across the heat exchanger was recorded manually as these values were relatively constant. In addition visual observations relating to each test, which were later used for interpretation of the numerical data, were noted.

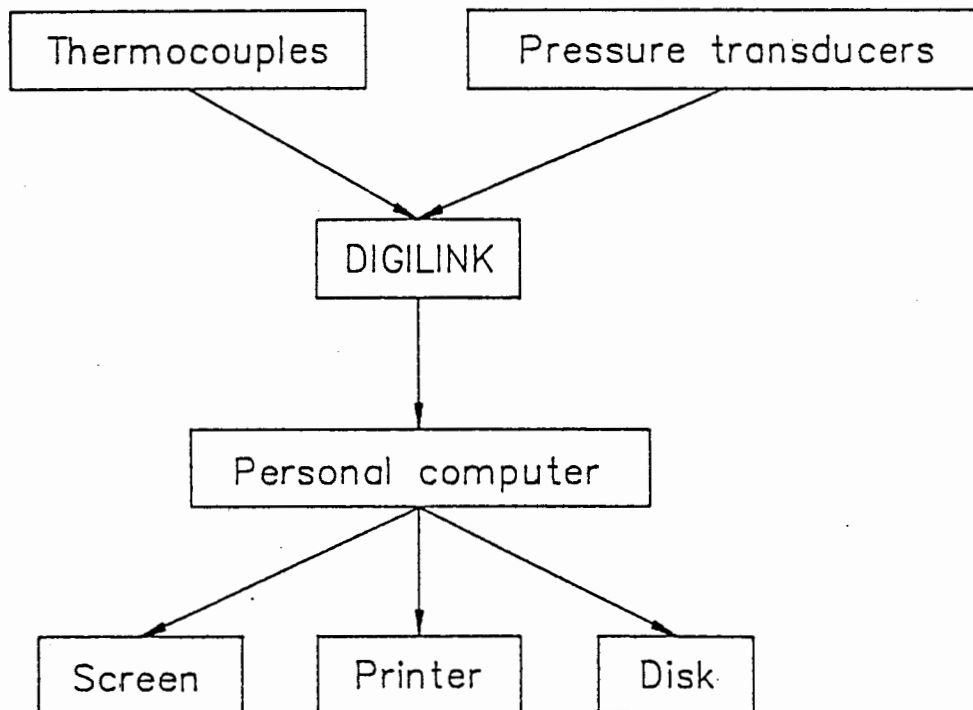


Figure 6.5 - Graphical description of the data logging system.

6.3 Experimental procedure

The performance enhancement of the spray cooled heat exchanger is primarily dependent on three parameters viz the mass-flow rate of the air, the mass flux of the spray water and the heat transfer potential that exists between the air and process water. This can also be deduced from the work of Kreid et al [79KR1] and other researchers as discussed in Chapter 2.

The only parameters which were freely variable were the mass-flow rates of the air, process water and spray water and the temperature of the process water. The temperature of the process water for each experiment was chosen beforehand and the water in the underground reservoir was then heated accordingly.

The experiments were divided in three categories based on the process water temperature. The low heat transfer potential category ranged from process water temperatures of 38 °C to 45 °C, the medium potential category from 45 °C to 55 °C and the high potential category from 55 °C to 65 °C. The second important variable in the performance of the spray cooled heat exchanger is the spray water mass flux. Four sets of spray nozzles were

used to vary the spray water density and droplet size to cover a spray water mass flux range of $35 \text{ kg/m}^2\text{h}$ to $450 \text{ kg/m}^2\text{h}$. Experiments were conducted to ensure that the complete range of spray water mass fluxes were represented in each heat transfer potential category.

The last important parameter, the air mass-flow rate, was easily controlled from inside the laboratory. In addition the energy balance over the test section stabilized rapidly after changing the air mass-flow rate indicating that the latter parameter is a convenient one to change between tests. Each experiment consisted of approximately ten tests which varied only in air velocity over the heat exchanger. The air mass-flow rate was varied between 3 and 12 kg/s which represents a free stream air velocity in the test section of 1.6 to 6.5 m/s. To ensure that representative data were obtained, five data sets for each test were saved on disk and the average of these values were used in the calculation of the performance of the spray cooled heat exchanger.

The calculation of the heat transferred from the process water was critical as this is the only simple and accurate way to determine the performance of the heat exchanger. The mass-flow rate of the process water was controlled manually with the use of a gate valve to ensure that the change of the process water temperature over the heat exchanger was large enough to be accurately measured. The capacity of the heat exchanger ranged from 100 kW for dry tests to 400 kW for high spray densities which represents a process water mass-flow rate of 4 to 12 kg/s if the temperature drop is to remain above $6 \text{ }^\circ\text{C}$.

7. DISCUSSION OF EXPERIMENTAL RESULTS AND COMPARISON WITH THE ANALYTICAL MODELS

The objective of this chapter is to examine the influences that some of the important parameters have on the performance enhancement due to the spray cooling of dry air heat exchangers. It will be shown that these parameters, which include the ambient conditions, spray density, heat transfer potential and air velocity, can give an indication of the viability of spray cooling in a given situation. The effect of spray cooling on the air pressure drop over the heat exchanger is also discussed and the experimental results are presented graphically.

To verify the reliability of the analytical models proposed in Chapter 3, the performance predictions of the computer programs that were discussed in Chapter 5 are compared with the experimental results. Model ACCURATE's simulation of the spray cooled heat exchanger's physical behaviour is graphically presented and discussed to show the similarities and deviations of the analytical models to the experimental data.

With the aid of the model ACCURATE, the spray cooled heat exchanger's behaviour is investigated analytically, to determine the optimum performance points in terms of air velocity and spray density for spray cooled heat exchangers with different process fluid temperatures operating in different ambient conditions.

7.1 Experimental results

The experimental results will be discussed under two headings:

- i) Observations of the physical behaviour of the heat exchanger.
- ii) Results calculated from the performance data.

7.1.1 Observation of the physical behaviour of the spray cooled heat exchanger

Visual observation of the top and bottom surfaces of the heat exchanger were possible through the perspex windows. Complementing to the limited visual observation of the behaviour of the water film on the finned surfaces of the tubes inside the heat exchanger, the temperature drop of the process water over each pass was recorded as described in section 6.2. From these temperature changes some conclusions about the movement of the water film on the surface of the heat exchanger could be made.

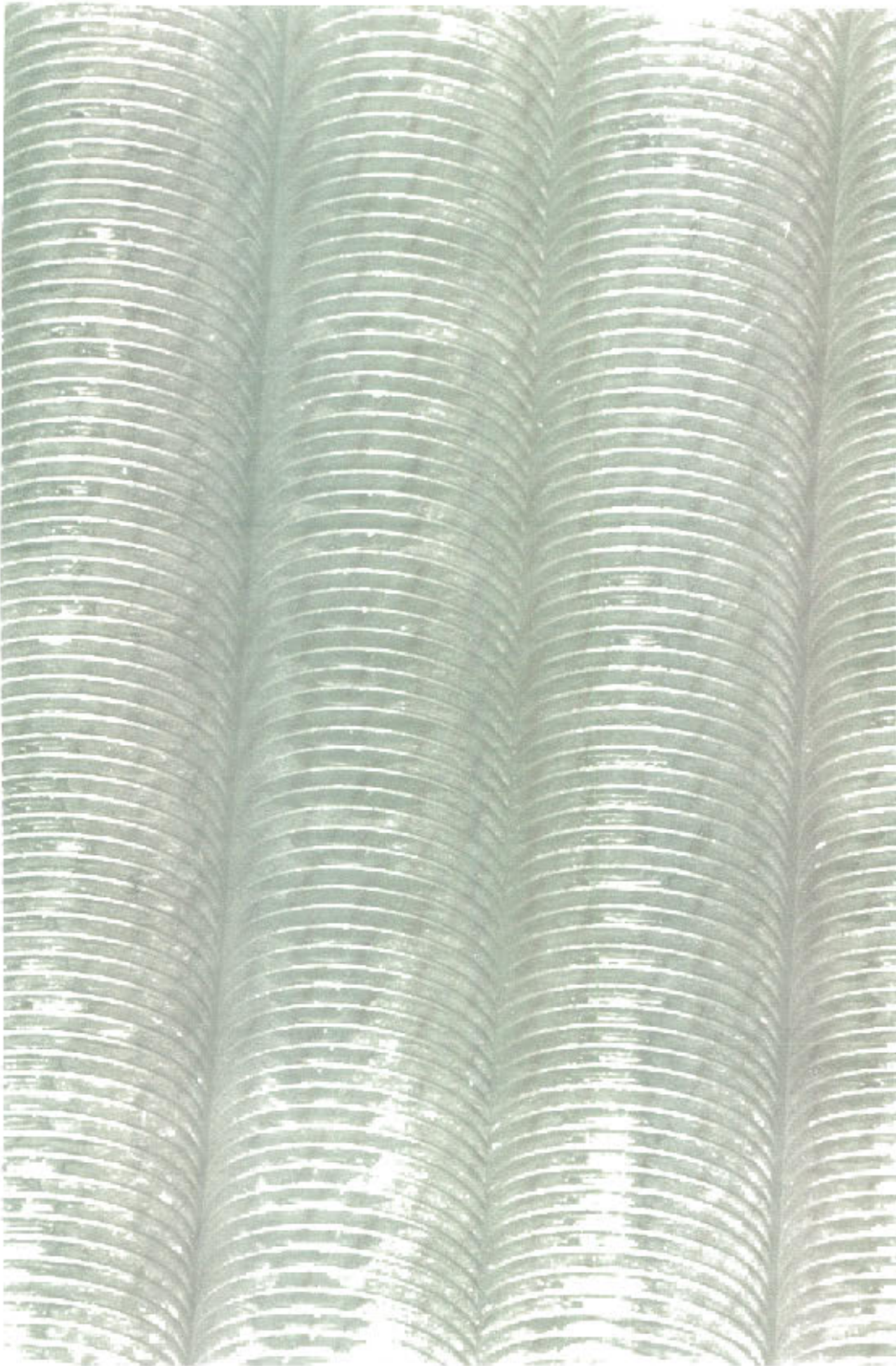


Figure 7.1 - Water pellets forming between the fins of the bottom tube row.

As the work of Fischer et al. [88FI1] predicted, water droplets were observed to be hanging between the fins of the bottom tube at low air velocities. It is clear from the photograph in Figure 7.1 that the lower part of the fins are completely obstructed by the water. This results in the formation of dead areas on the tubes where the resistance to heat transfer is very high. The size of these areas were found by Rudy et al [85RU1] to vary between 15% and 30% of the circumference of the tube and from Figure 6.6 it can be seen that this is also true for the present study. The analytical modelling of this behaviour is described in Chapter 3.

At air velocities above the critical velocity, which is defined in Chapter 3, and high spray water densities, water droplets were separated from the film on the fins of the top row of tubes and were suspended above the heat exchanger or carried out of the tower by the air stream leaving the test section. In the transition zone, it was observed that the water droplets that were separated from the film, dropped back onto the heat exchanger as soon as they entered the free stream leaving the test section.

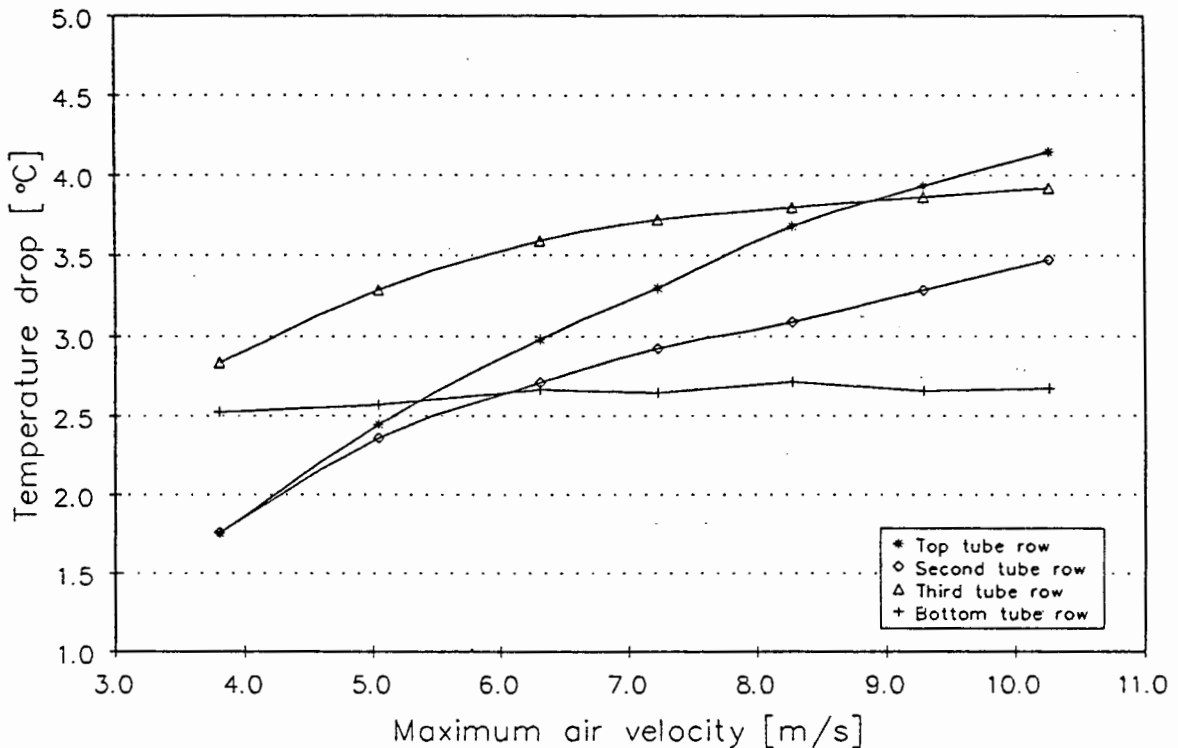


Figure 7.2 - Temperature drop of the process water across each pass of the dry heat exchanger.

Figure 7.2 shows the behaviour of the dry heat exchanger with increasing air velocity. The comparison of this graph with the wet behaviour of the heat exchanger depicted in Figures 7.3 to 7.5, clearly shows the influence which the direction of the flow of the water film has on the heat transfer.

When the spray water density is low, Figure 7.3, the bottom tube row captures most of the spray water droplets from the air and most of the film evaporates on this row. From Figure 7.3 it can be seen that the bottom tube row represents the highest temperature drop in the wet heat exchanger while the third tube row has the largest capacity for heat transfer in the dry heat exchanger. Another feature of the wet heat exchanger is the sudden increase in heat transfer which occurs in the transition zone. This is the result of the water film changing flow direction and therefore creating a larger effective wet area.

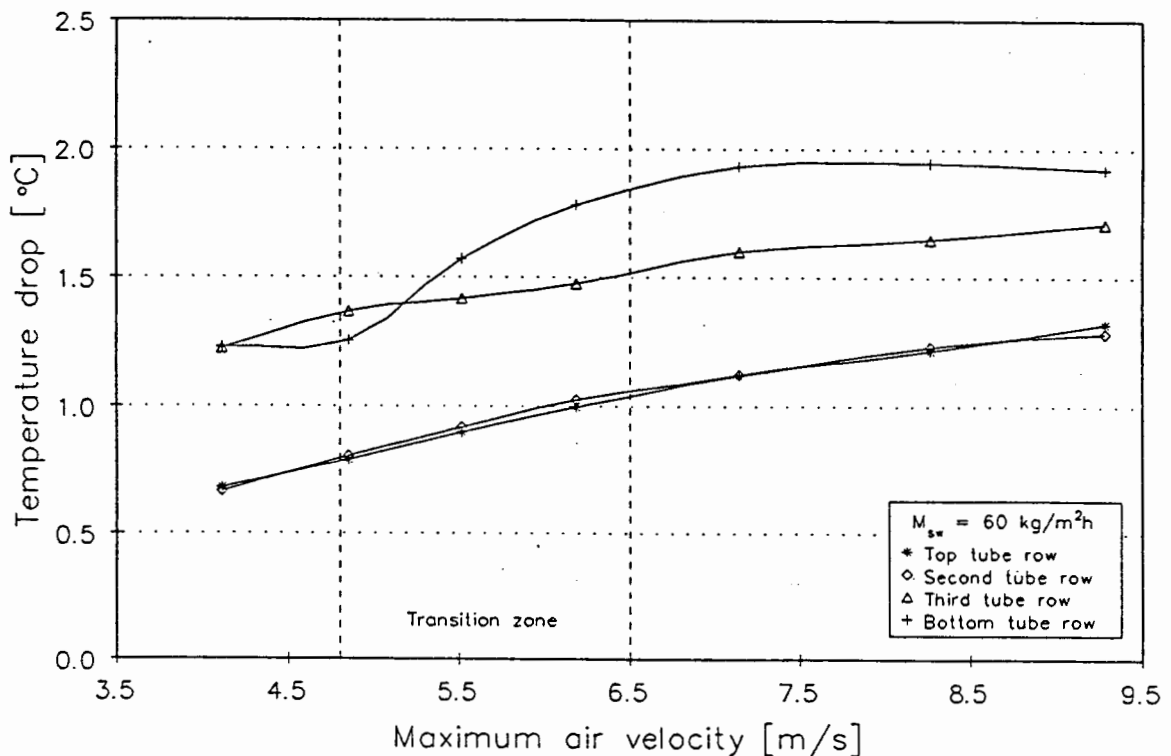


Figure 7.3 - Temperature drop of the process water across each pass of the wet heat exchanger. - Low spray density.

When the spray water density is increased, Figure 7.4 shows that the third tube row now also receives more water from both the air-mist flowing over it and the film flowing from the bottom tube row. At high air mass-flow rates, > 6 kg/s, the graph clearly shows that the heat transfer over the second tube row increases more rapidly than the heat transfer over the top tube row. This is an indication that the water film has reached this tube row before

all the water was evaporated. Once the air velocity has increased beyond the transition zone, the graph shows that a steady state situation is reached where the increases in heat transfer is approximately equal for all the tube rows. The insulation effect that the thick water film has on the bottom tube row at very low air velocities can also be seen from Figure 7.4.

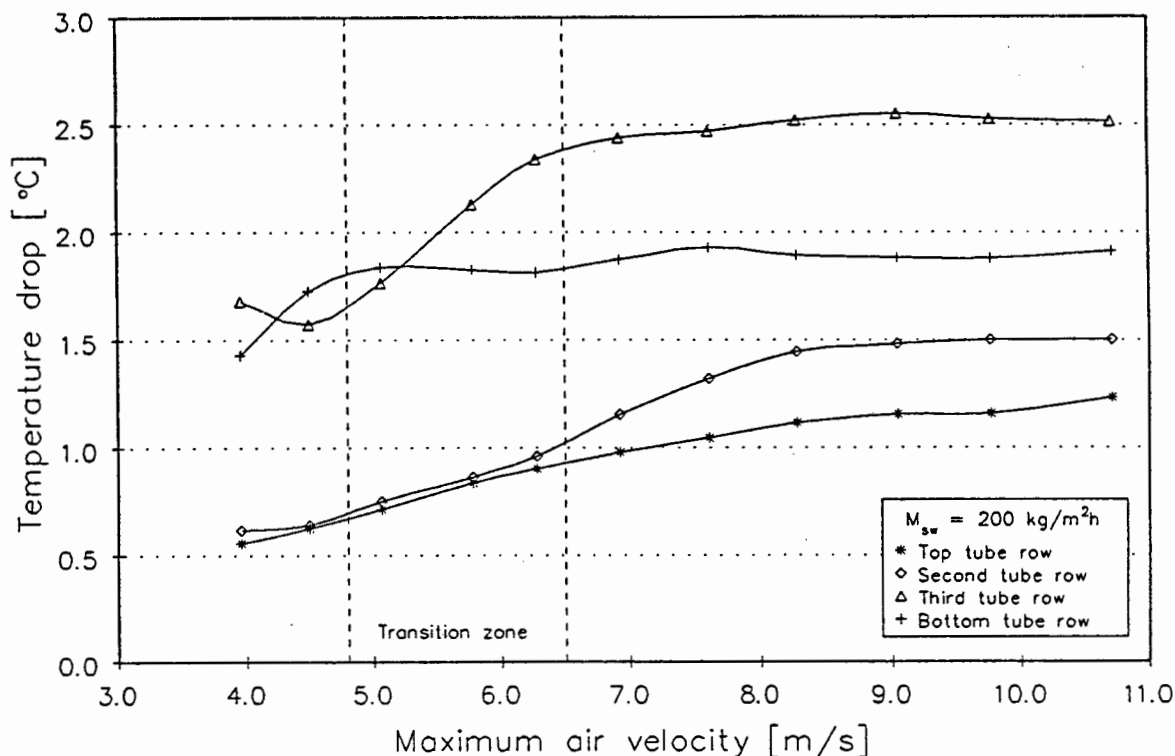


Figure 7.4 - Temperature drop of the process water across each pass of the wet heat exchanger. - Medium spray density.

Figure 7.5 clearly illustrates the behaviour of the water film on the surface of the heat exchanger as all the tube rows are wetted by the spray water at high air velocities. In this figure the area below the transition zone clearly shows that the upper tube rows are prevented by the lower tube rows from getting wet. As in the case of medium spray density the insulating effect which the thick water film has on the bottom tube row is seen from the low heat transfer rate of the bottom tube row.

In the transition zone the water film begins to flow upwards resulting at first in a large increase in the heat transfer over the third tube row and then as the film starts reaching the second and top tube rows, a dramatic increase in heat transfer occurs in the immediate vicinity of the critical velocity (5.4 kg/s). The simultaneous drop in heat transfer over the bottom and third tube rows, is the result of cooler water that now reaches the lower tube

rows and subsequently decreases the heat transfer potential that exists there.

The steady state that exists at air velocities higher than the transition zone upper border can again be seen in Figure 7.5. Because of this phenomenon the modelling of the spray cooled heat exchanger in the upflow zone is much simpler than the downflow or transition zone (see Chapter 3) which results in better predictions by the model of the spray cooled heat exchanger's performance. The experimental results and comparison thereof with the analytical models will be discussed in section 7.2.

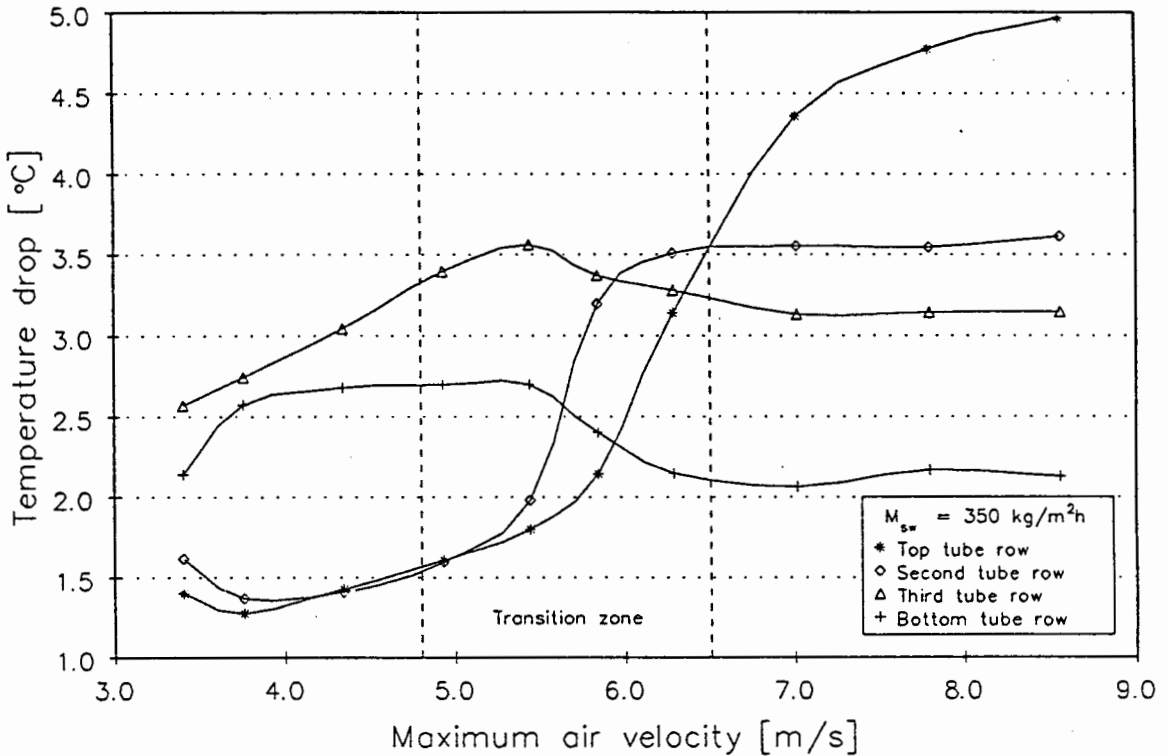


Figure 7.5 - Temperature drop of the process water across each pass of the wet heat exchanger. - High spray density.

7.1.2 Results calculated from the performance data

The experimental data are presented as a function of the dry heat exchanger performance. This allows comparison between tests for the determination of the best point of operation for the spray cooled heat exchanger.

$$F_{\text{enh}} = \frac{q_{\text{tot,wet}}}{q_{\text{tot,dry}}} \quad (7.1.1)$$

The variable ambient conditions and changing process water temperature during testing, made it impossible to obtain test results with constant inlet conditions. To examine the effect of a single parameter on the performance of the heat exchanger, it is necessary to remove the effects of other parameters by keeping their values constant. This was done artificially by first determining the parameters which describe the performance enhancement of the spray cooled heat exchanger, and then creating dimensionless groups with which a correlation for the performance enhancement could be determined.

By repeatedly fitting curves through all the experimental data using combinations of all the variables the basic parameters that control the enhancement were found to be: i) the mass flux of the spray water, ii) the velocity of the air in the heat exchanger and iii) the enthalpy potential that exists between the air and the wet surface of the heat exchanger. Using these parameters as basis, seven dimensionless groups that represent these parameters were determined. These groups are defined as: i) the ratio of the maximum air velocity and critical air velocity, ii) the spray water and air ratio, iii) ratio of the inlet air vapour content and the total spray water/vapour content of the air-mist flow, iv) ratio of the minimum and maximum potential of the fluids (air and process water), which were defined as,

$$q_{\min} = m_p c_{pp} (T_{pi} - T_{wbai}) \quad (7.1.2)$$

$$q_{\max} = m_a (i_{aspi} - i_{ai}) \quad (7.1.3)$$

v) ratio of the measured heat transfer and the minimum potential, vi) ratio of the heat transferred by evaporation and the measured heat transfer and vii) the inlet air relative humidity. With the use of statistical software package the following enhancement function was deduced from the experimental data.

$$F_{\text{enh}} = 6.9676 \left[\frac{v_{\max}}{v_{\text{crit}}} \right]^{0.1002} \times \left[\frac{M_{\text{sw}}}{M_a} \right]^{0.1892} \times \left[\frac{w_{\text{ai,dry}}}{w_{\text{ai,wet}}} \right]^{0.1093} \times \left[\frac{q_{\min}}{q_{\max}} \right]^{0.1757} \times \left[\frac{q_{\text{exp}}}{q_{\min}} \right]^{0.219} \times \left[\frac{m_{\text{sw}} h_{\text{fg}}}{q_{\text{exp}}} \right]^{0.2626} \times \phi^{-0.5828} \quad (7.1.4)$$

The experimental data and performance enhancement for each test are presented in Appendix G.

The air velocity, humidity ratio of the ambient air and spray water density are important parameters that should be taken into consideration when spray cooling is considered. In order to examine separately the influences of these parameters on the spray cooled heat

exchanger, three new dimensionless groups η_1 , η_2 and η_3 is formed in equation using the empirical equation for F_{enh} as defined in equation (7.1.4).

$$\eta_1 = \frac{F_{enh}}{\left[\frac{M_{sw}}{M_a} \right]^{0.1892} \times \left[\frac{w_{ai,dry}}{w_{ai,wet}} \right]^{0.1093} \times \left[\frac{q_{min}}{q_{max}} \right]^{0.1757} \times \left[\frac{q_{exp}}{q_{min}} \right]^{0.219} \times \left[\frac{m_{sw} h_{fg}}{q_{exp}} \right]^{0.2626} \times \phi^{-0.5828}} \quad (7.1.5)$$

$$\eta_2 = \frac{F_{enh}}{\left[\frac{v_{max}}{v_{crit}} \right]^{0.1002} \times \left[\frac{w_{ai,dry}}{w_{ai,wet}} \right]^{0.1093} \times \left[\frac{q_{min}}{q_{max}} \right]^{0.1757} \times \left[\frac{q_{exp}}{q_{min}} \right]^{0.219} \times \left[\frac{m_{sw} h_{fg}}{q_{exp}} \right]^{0.2626} \times \phi^{-0.5828}} \quad (7.1.6)$$

$$\eta_3 = \frac{F_{enh}}{\left[\frac{v_{max}}{v_{crit}} \right]^{0.1002} \times \left[\frac{M_{sw}}{M_a} \right]^{0.1892} \times \left[\frac{w_{ai,dry}}{w_{ai,wet}} \right]^{0.1093} \times \left[\frac{q_{min}}{q_{max}} \right]^{0.1757} \times \left[\frac{q_{exp}}{q_{mi}} \right]^{0.219} \times \left[\frac{m_{sw} h_{fg}}{q_{exp}} \right]^{0.2626}} \quad (7.1.7)$$

The resulting respective enhancement ratios, η_n , show the influence of each of these parameters on the heat transfer enhancement. All the experimental data is plotted against these enhancement ratios and the results is shown in Figure 7.6 to 7.8. From Figure 7.6, which shows the influence of the air velocity on the heat transfer enhancement, it can be seen that the enhancement ratio is not strongly dependent on the air velocity in the heat exchanger. The reason for this is that the performance of the dry heat exchanger increases at approximately the same rate as the wet heat exchanger as the air velocity increases. The slight increase in performance enhancement that occurs with increasing air velocity is the result of a reverse in the direction of flow of the spray water film in the transition zone. This in turn increases the wet area of the heat exchanger which leads to a higher heat transfer rate.

A factor that plays an more important role in the performance enhancement is the relative humidity of the ambient air which can be seen in Figure 7.7.

The performance enhancement that can be obtained by spray cooling in arid ambient conditions is shown to be much higher than in situations where the ambient air is almost saturated. This is the result of the combination of the high potential for evaporation that exists in dry air and the increased heat transfer potential which the low wet bulb temperature of the dry air provides for the wet surface of the heat exchanger. A further factor that increases the performance of the spray cooled heat exchanger in ambient conditions with a low humidity ratio, is the precooling of the warm ambient air by the cold spray water droplets.

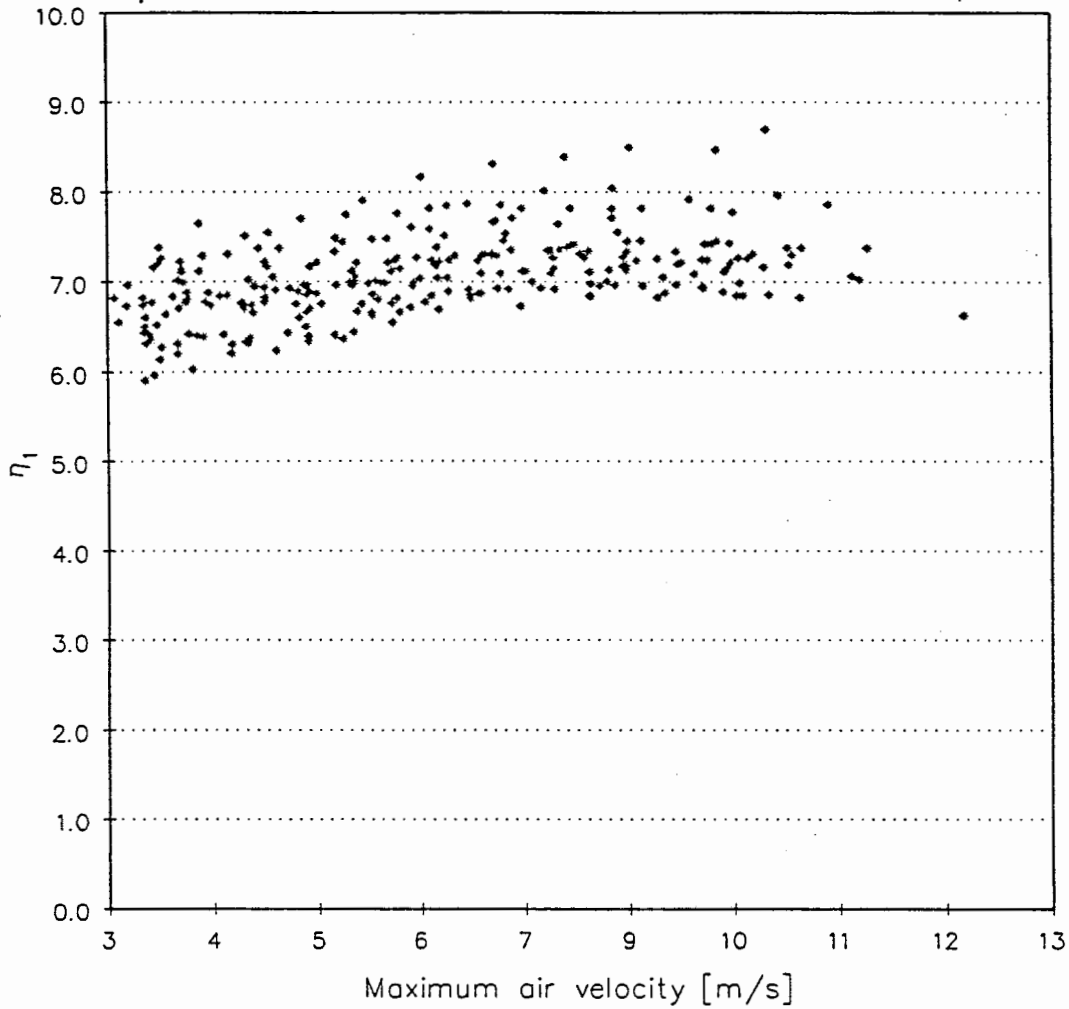


Figure 7.6 - The influence of air velocity on performance enhancement in the spray cooled heat exchanger.

Finally the influence of spray density on performance enhancement is examined. Figure 7.8 gives an indication of the trend that is to be expected of the heat transfer enhancement with varying spray density.

This figure shows the sharp increase in heat transfer performance that occurs at low spray densities. This sharp increase in performance is the result of the evaporation of the spray water film on the surface of the heat exchanger. The rate of increase in performance enhancement declines as the spray density increases because a smaller percentage of the spray water evaporates into the air as the outlet condition of the air reaches saturation point. In the region of very high spray density, the heat transferred by the water film flowing from the heat exchanger plays a significant role in the increase in performance enhancement. This accentuates the fact that the most economical operation point of the

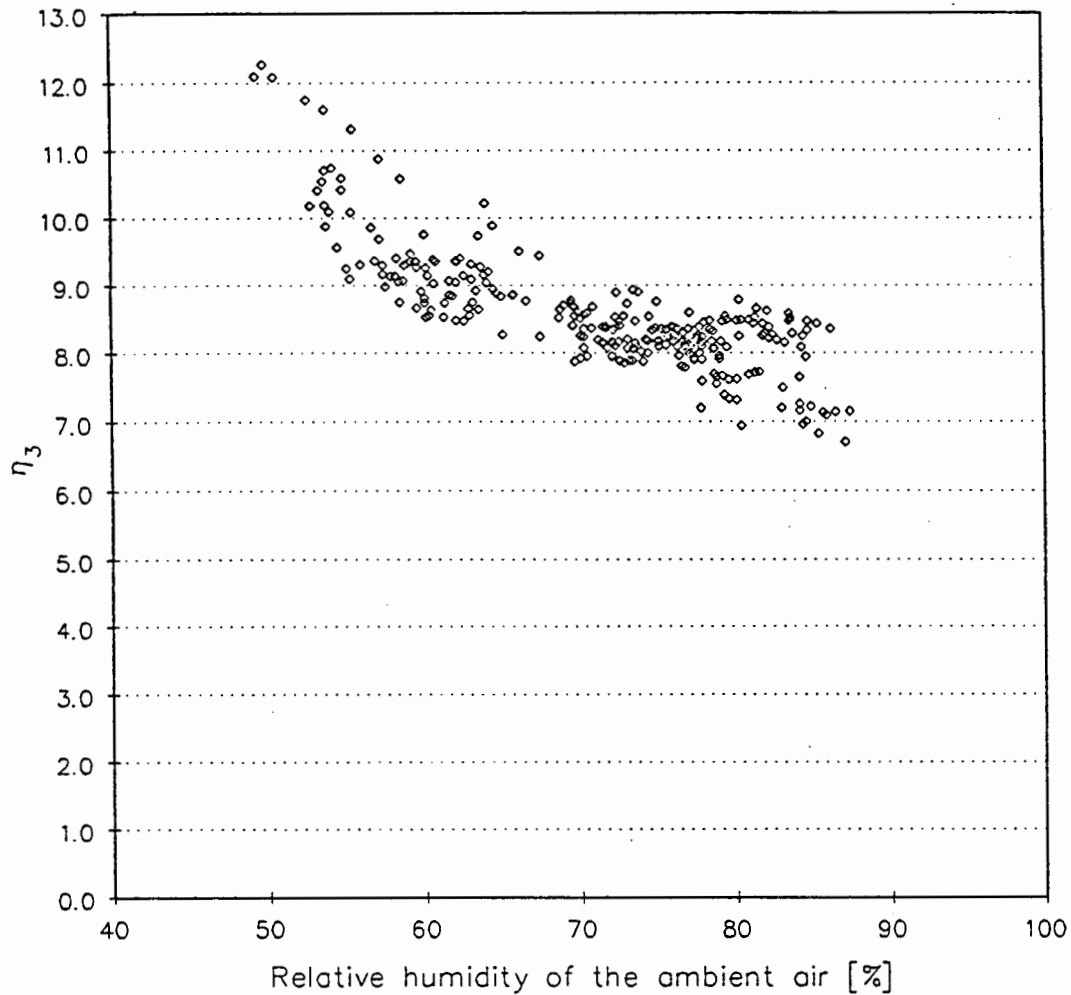


Figure 7.7 - The influence of the humidity ratio on the performance enhancement of the spray cooled heat exchanger.

spray cooled heat exchanger is at low spray densities where less water is wasted and performance enhancement increases rapidly.

The measured pressure drop over the heat exchanger shows a moderate increase with spray cooling as long as the spray density is below $250 \text{ kg/m}^2\text{h}$ and the maximum air velocity in the heat exchanger is above the critical velocity. The graphical evaluation of the experimental data can be seen in Figure 7.9.

From this figure it is clear that a large increase in pressure drop is experienced over the spray cooled heat exchanger at spray densities higher than $250 \text{ kg/m}^2\text{h}$ and low air velocity. This is the result of water congestion that occurs when the weight of the water that is trapped in the heat exchanger increases to such an extent that it cannot be carried from the

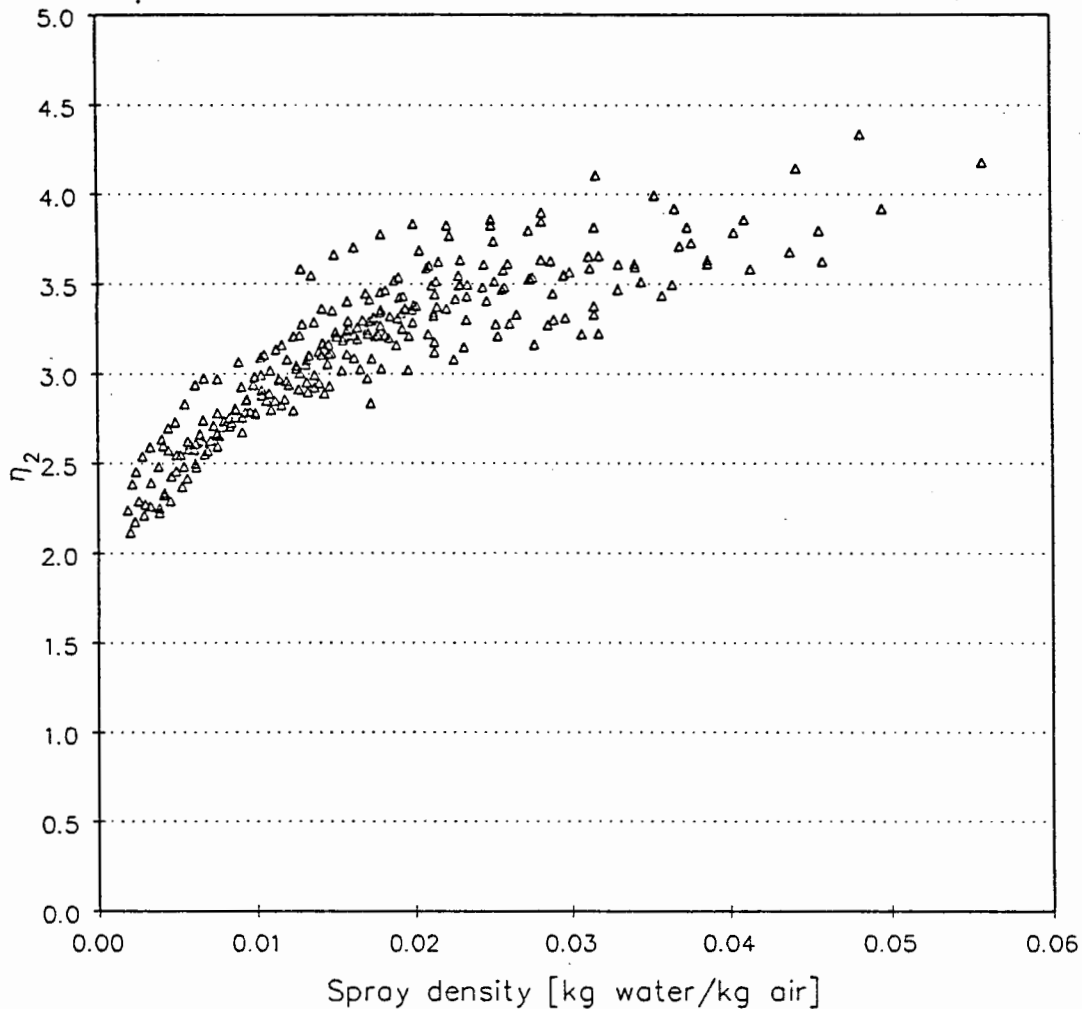


Figure 7.8 - The influence of spray density on the performance enhancement of the spray cooled heat exchanger.

heat exchanger by the air. The water flows into the fin gaps and blocks the air flow which now has to pass through the spaces between the finned-tubes. As a result of the smaller free flow area, the air velocity in the heat exchanger increases which in turn increases the pressure drop over the heat exchanger.

The actual performance enhancement which was measured depended mainly on the spray water mass flux during operation. With very low spray water mass fluxes, 35-100 kg/m²h, it was possible to obtain performance enhancement ratios of 1.2 to 1.5 times the dry performance depending on the humidity ratio and enthalpy potential while very high spray water mass fluxes, 300-480 kg/m²h, produced performance enhancement ratios up to 3.5 times the dry heat exchanger performance at very low humidity ratios and high enthalpy

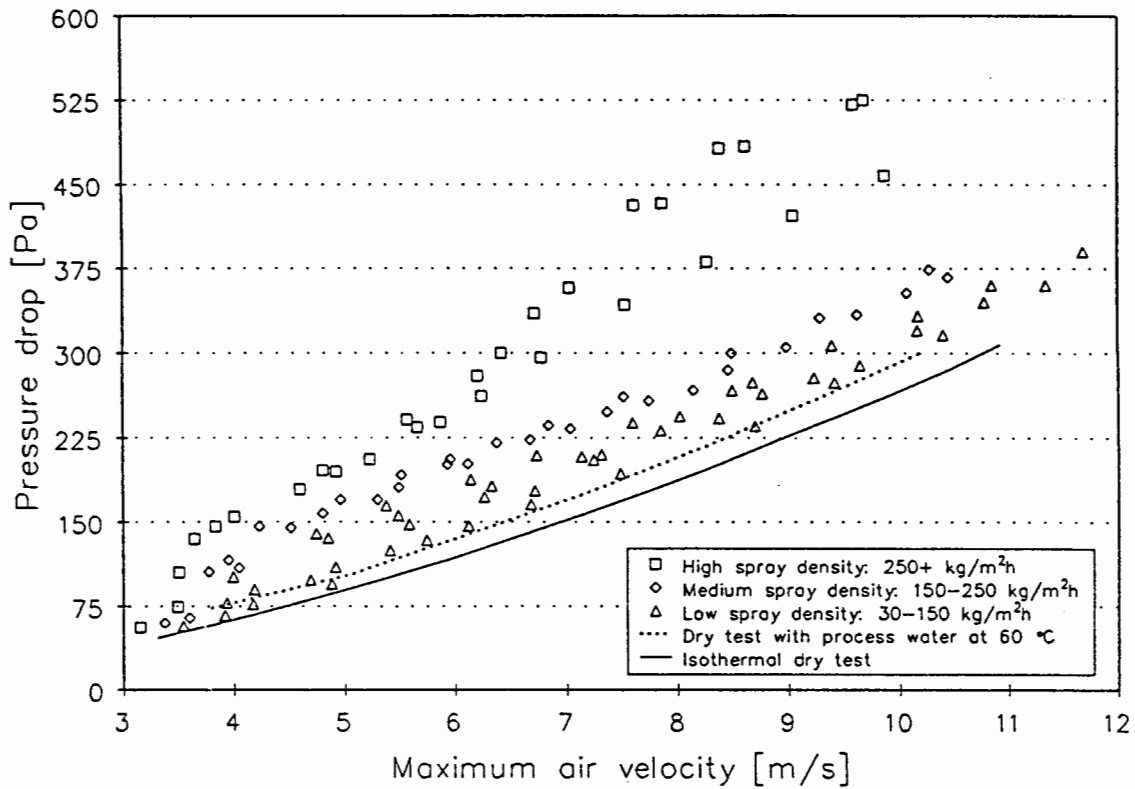


Figure 7.9 - Pressure drop of the air-mist stream flowing over the spray cooled heat exchanger

potential.

7.2 Comparison of experimental results with the predictions of the analytical models

7.2.1 Kreid model

This model, described in Chapter 5, was examined in order to give an indication of the maximum performance enhancement which is possible for the spray cooled heat exchanger. The model assumes that the entire surface of the heat exchanger is covered by a very thin spray water film and the heat transferred to this film is considered negligible. As a result the spray water density has no influence in the calculation of the total heat transfer.

Keeping all but one parameter constant, a number of inlet mass flows and temperatures for the air and process water were chosen to evaluate the influence of a particular parameter. The results are shown in Figures 7.10 and 7.11.

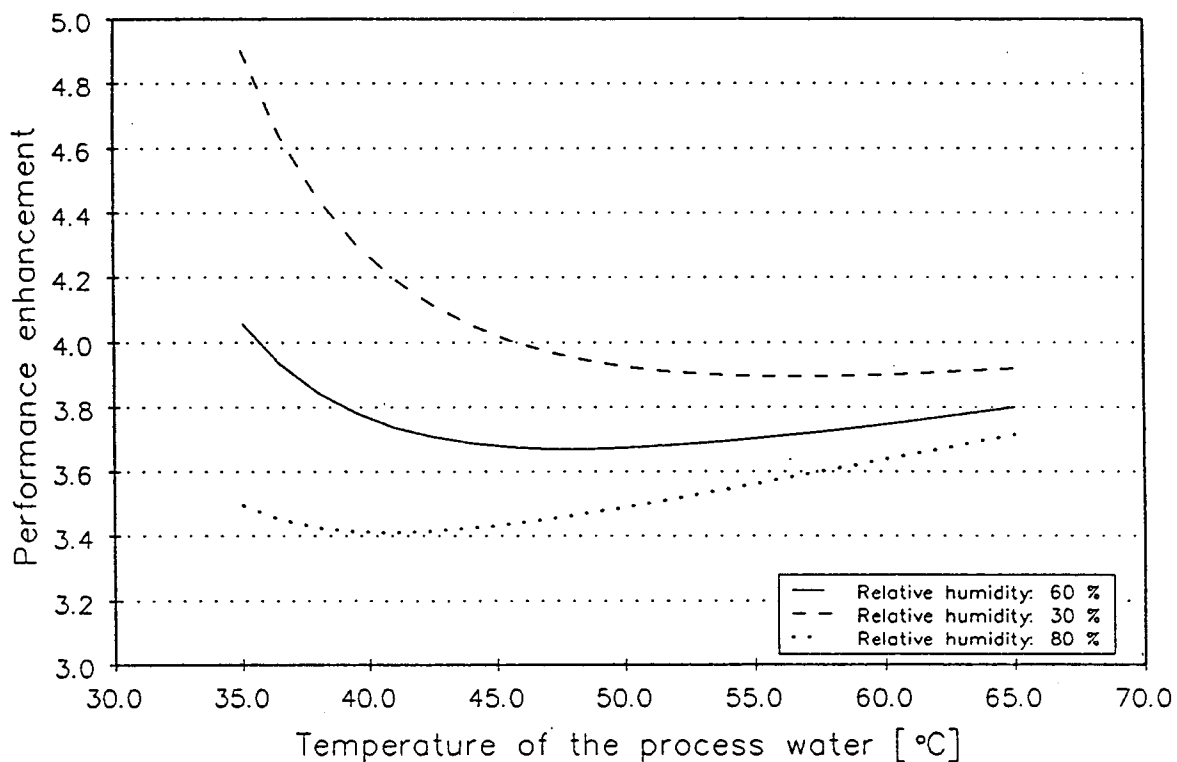


Figure 7.10 - Maximum performance enhancement of the spray cooled heat exchanger with variable heat transfer potential.

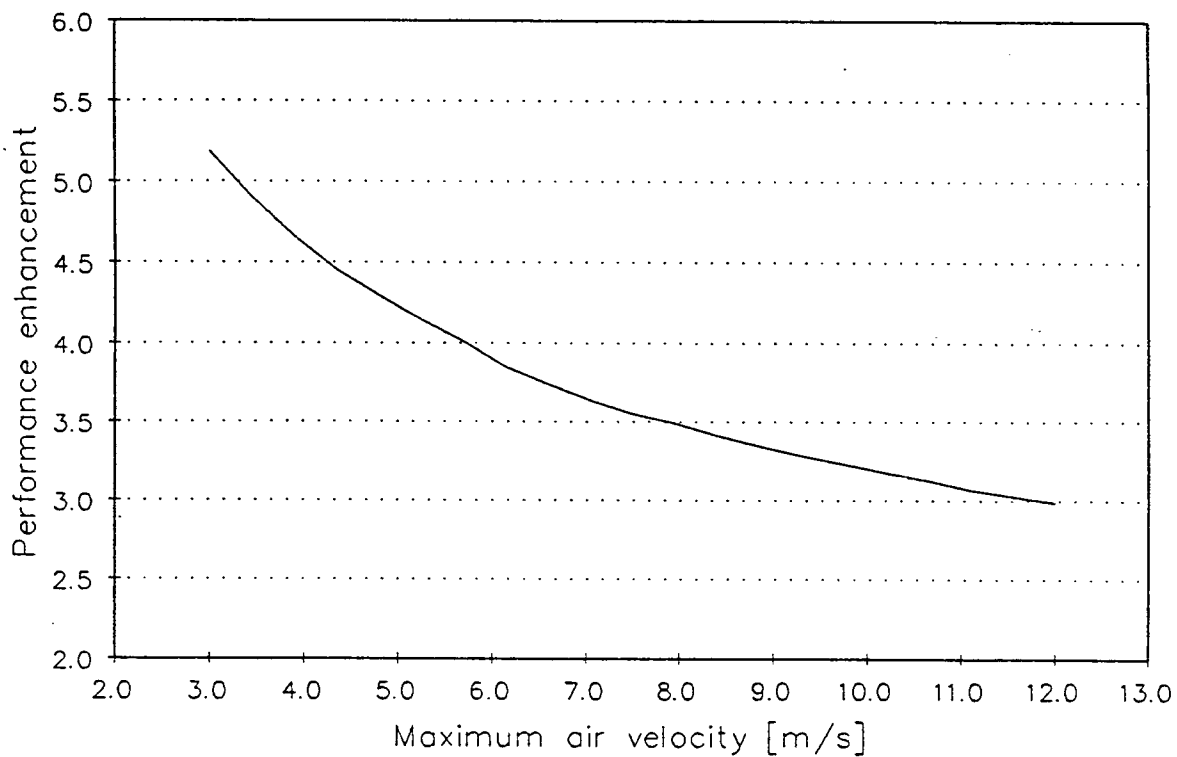


Figure 7.11 - Maximum performance enhancement of the spray cooled heat exchanger with variable air velocity.

Figure 7.10 shows that a performance enhancement of up to five times the performance of the dry air heat exchanger is possible at low heat transfer potential or low process water temperature. The enhancement ratio is strongly dependent on the humidity of the air at low heat transfer potential conditions, but this dependence disappears as the potential increases. These results are similar to the experimental results which were discussed in the previous section.

Figure 7.11 shows the variation of the performance enhancement ratio with air velocity. The decrease in performance enhancement with increasing air velocity is the result of the sharp increase in dry heat exchanger performance compared to the relatively slower increase in performance exhibited by the wet surface heat exchanger. Since the wet heat transfer area or film flow direction is not a factor in this model, the graphical trend differs from the trend that the experimental data shows in Figure 7.6. The model over-predicts the performance enhancement at low air velocities and low spray density but gives a good indication of the maximum performance enhancement that can be expected at conditions with high spray water density and high air velocity in the heat exchanger.

7.2.2 Model ACCURATE

The performance prediction of the Accurate model is compared to the experimental data in Figures 7.12 to 7.14. These graphs show that this model predicts the performance of the spray cooled heat exchanger for all the experiments within $\pm 20\%$. The average absolute error of prediction was found to be 5,88%. Of all the experimental data evaluated, 80% of

Table 7.1 Prediction statistics of the Accurate model.

Spray water mass flux [kg/m ² h]	Average error [%]	Predict. within 10%	Max. error [%]	Min. error [%]
0 - 50	3.20	100	6.4	-2.1
50 - 100	3.72	88	17.13	-4.69
100 - 150	4.29	91	17.84	-8.34
150 - 200	4.48	87.5	15.08	-13.22
200 - 250	6.15	80.5	12.65	-11.93
250 - 300	7.39	64	15.79	-13.91
300 - 350	6.45	76	14.76	-9.55
350 +	7.33	67.5	15.25	-15.9
All data	5.88	79.3	17.83	-15.9

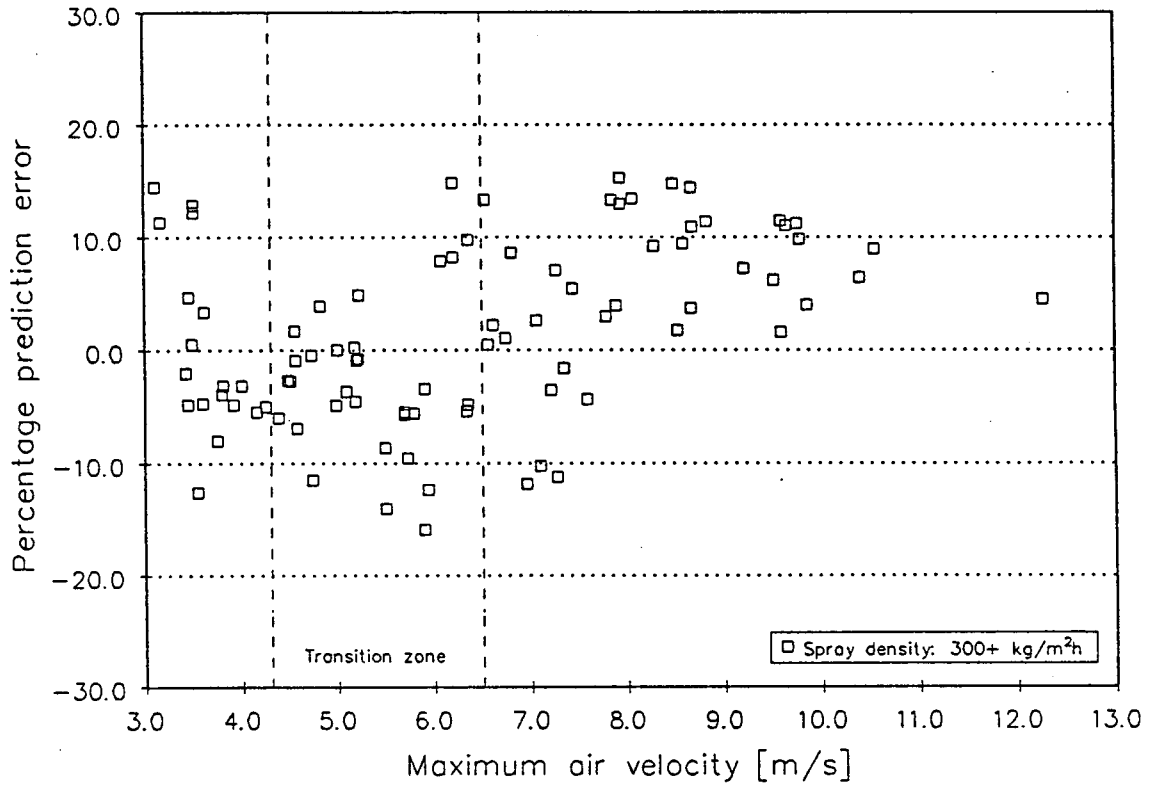


Figure 7.12 - Graphical evaluation of the predicted performance model ACCURATE for high spray density.

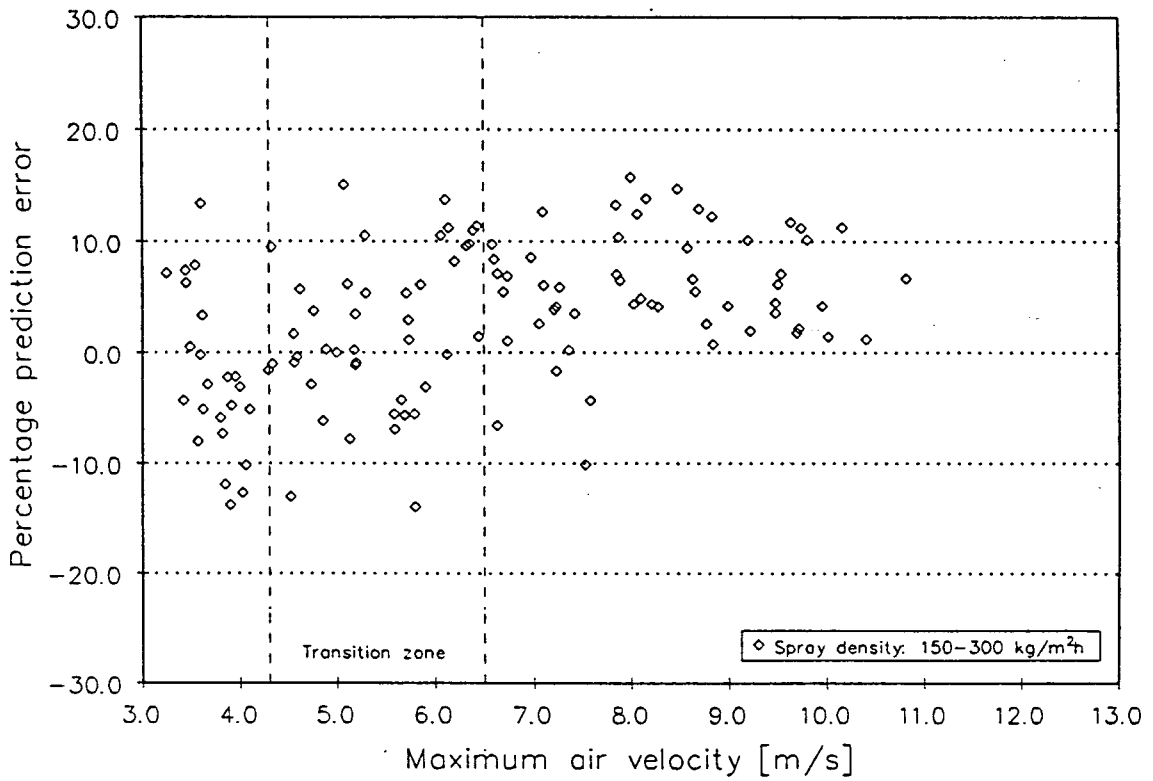


Figure 7.13 - Graphical evaluation of the predicted performance of model ACCURATE for medium spray density.

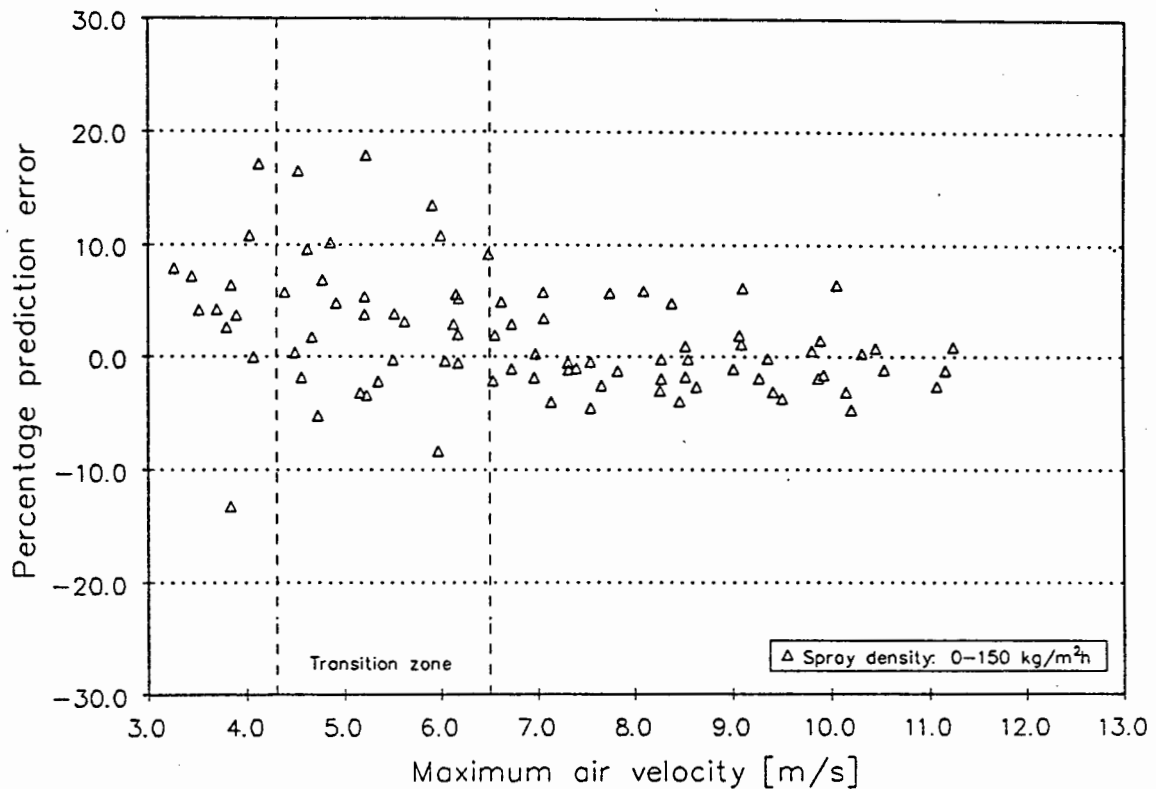


Figure 7.14 - Graphical evaluation of the predicted performance of model ACCURATE for low spray density.

all predictions were within 10%, and 97% of the predictions was within 15% of the measured values. Table 7.1 gives a summary of the performance prediction of this model.

From these figures and the statistical analysis in Table 7.1, it can be seen that the predicted performance of model ACCURATE is better at low spray densities and high air velocities. The higher accuracy in the post-transition zone is the result of the stable behaviour of the spray cooled heat exchanger in this zone. In the transition zone and down flow zone, the uncertainty concerning the behaviour of the water film on the surface of the heat exchanger is the cause of the more erratic predictions by the Accurate model. The fact that the performance prediction of the model decreases with increasing spray density, can be explained by the model's greater dependency on the effective wet area ratio to determine the amount of evaporation from the surface of the tubes in high density spray. Model ACCURATE assumes a value for the effective wet area ratio for the first element of the tube and the evaporation rate from the surface of this element is calculated using this value. The next element receives the remaining water and when the mass of water which flows over the surface of a tube is small, some of the elements may receive no water at all, in which case the model assumes that the value of the effective wet area ratio is zero. The average of these ratios for the elements of a tube give the effective wet area ratio for the tube which means that for low density spray water conditions, the average effective wet area ratio for

a tube is self-correcting whereas this ratio is constant when the spray density is high enough to sustain the water film on the surface of the heat exchanger.

To evaluate this model further, its prediction of the physical behaviour of the spray cooled heat exchanger is examined next. The behaviour of the spray water film on the heat exchanger surface is best described in Figures 7.3 to 7.5. The same inlet conditions were used to create curves of the temperature drop across each pass as predicted by the accurate model.

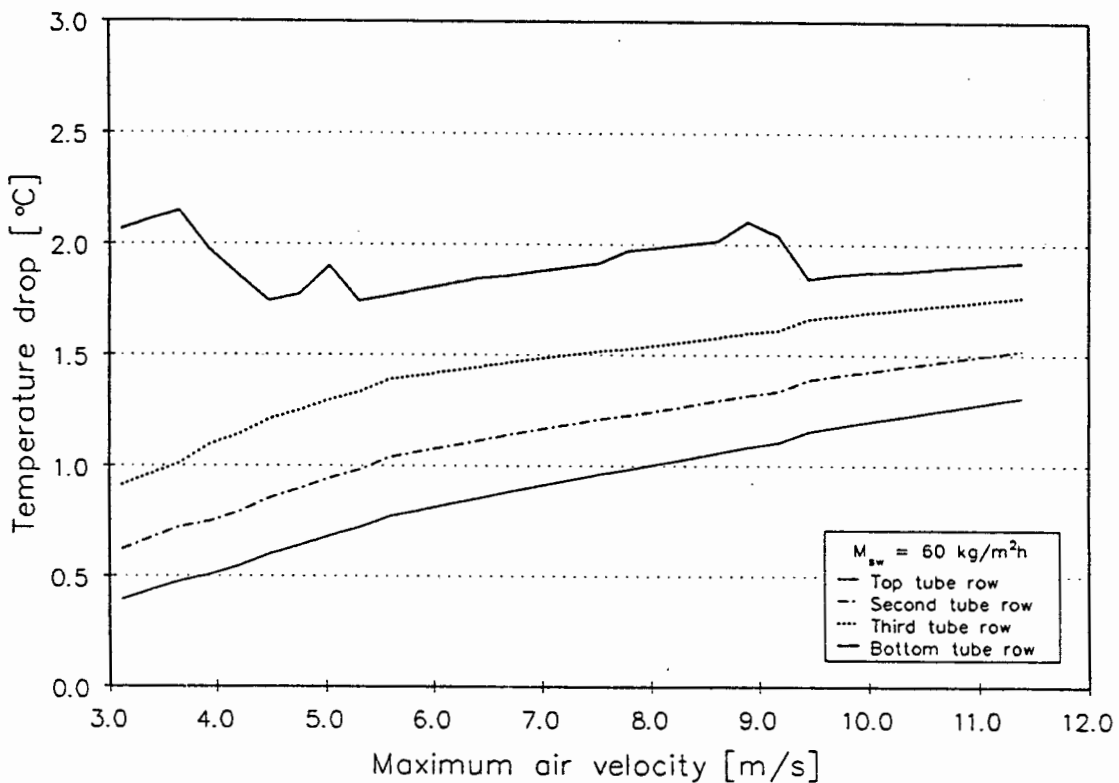


Figure 7.15 - Temperature drop of the process water across each pass of the heat exchanger at low spray density - model ACCURATE.

From Figures 7.15 to 7.17 it can be seen that although the model does not give an exact prediction of the behaviour of the spray cooled heat exchanger, the same physical phenomena that were observed in the experimental work are shown by the analytical model. The steplike form of the graphs are a result of the changes in film flow direction and effective wet area ratio that the model must compensate for with the use of step functions. The first step which occurs at a air velocity of approximately 4.7 m/s, takes place in the lower tube rows as the effective wet area ratio increase as a result of a thinner spray water film on the surface of the tube. The second step at approximately 5.6 m/s, has a influence on all the tube rows as this is the point where the spray water film starts flowing upwards on the surface of the heat exchanger. The heat transfer on the bottom tube row decreases here as the heat transfer on the upper rows increases sharply. The last step occurs at the

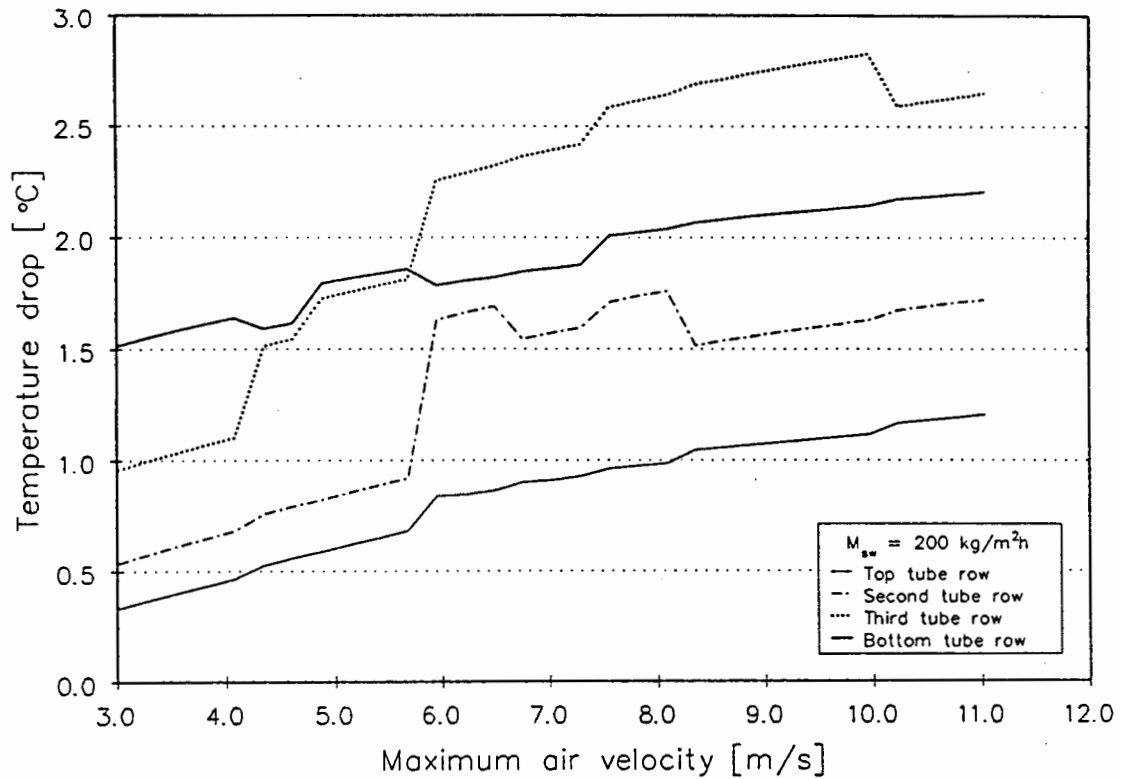


Figure 7.16 - Temperature drop of the process water across each pass of the heat exchanger at medium spray density - model ACCURATE.

end of the transition zone, approximately 7.8 m/s, and from here a thin water film that flows upwards exists on the surface of the heat exchanger. Other smaller steps are also noticeable and is the result of film dryout that occurs on the surface. From Figure 7.15 it is clear that one of the problem areas is the over-prediction of the performance of the bottom tube row at very low air velocities. Figures 7.16 and 7.17 show that the performance prediction of the bottom tube row is too high at almost all air velocities and spray densities which is probably the result of underestimating the spray water film thickness on this tube by the use of a constant film heat transfer coefficient, h_f . The fact that the model simulates the physical tendencies that occur in a experimental heat exchanger reasonably well, is an indication that this model will give accurate predictions for spray cooled heat exchangers of any geometry.

The performance enhancement of the spray cooled heat exchanger with constant inlet conditions can also be evaluated with the use of model ACCURATE. Fictitious data with constant inlet conditions was generated for evaluation with the computer program discussed in Chapter 5 and the performance enhancement results are presented as a functions of the air velocity, spray water mass flux and inlet process water temperature in Figures 7.18 to 7.20.

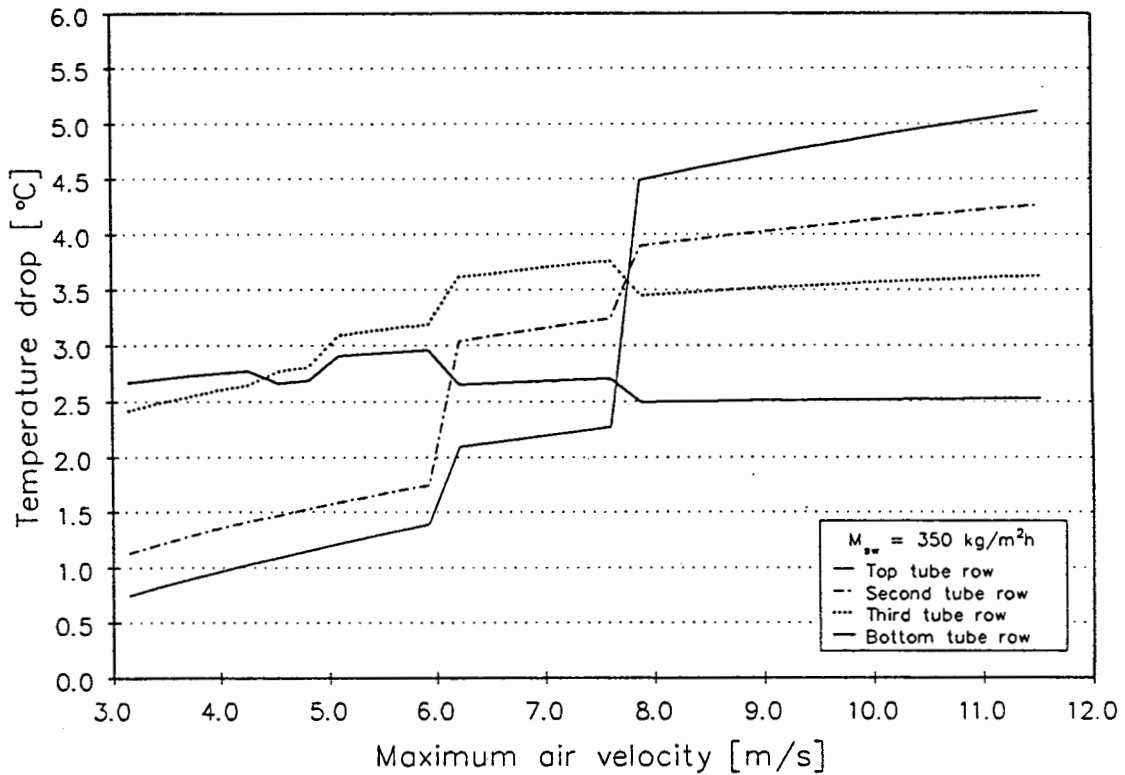


Figure 7.17 - Temperature drop of the process water across each pass of the heat exchanger at high spray density - model ACCURATE.

The steplike changes depicted in Figure 7.18 are also due to the nature of the analytical model which is described above, and the real behaviour of the heat exchanger can of course be presented as a smooth curve. This figure shows that for all spray densities except the lowest, there is a significant increase in heat transfer enhancement in the transition zone. This is the direct result of the change of flow direction of the spray water film which increases the effective wetted area of the heat exchanger. The maximum enhancement that can be achieved is according to Figure 7.18 dependent on the spray density and air velocity.

At very high spray densities the maximum point of performance enhancement is at the high end of the transition zone, in this case 7.5 m/s, while this point moves back to the lower end of the transition zone as the spray density decreases.

The next figure, Figure 7.19, show more clearly the variation of the performance enhancement as the spray density increases. An important feature of the spray cooled heat exchanger is the fact that the performance enhancement reaches a maximum value as the spray density increases and then becomes constant. The importance of this for the design of a spray cooled heat exchangers is obvious in that the choice of the optimum spray density is dependent on the maximum air velocity in the heat exchanger. For example, if an air

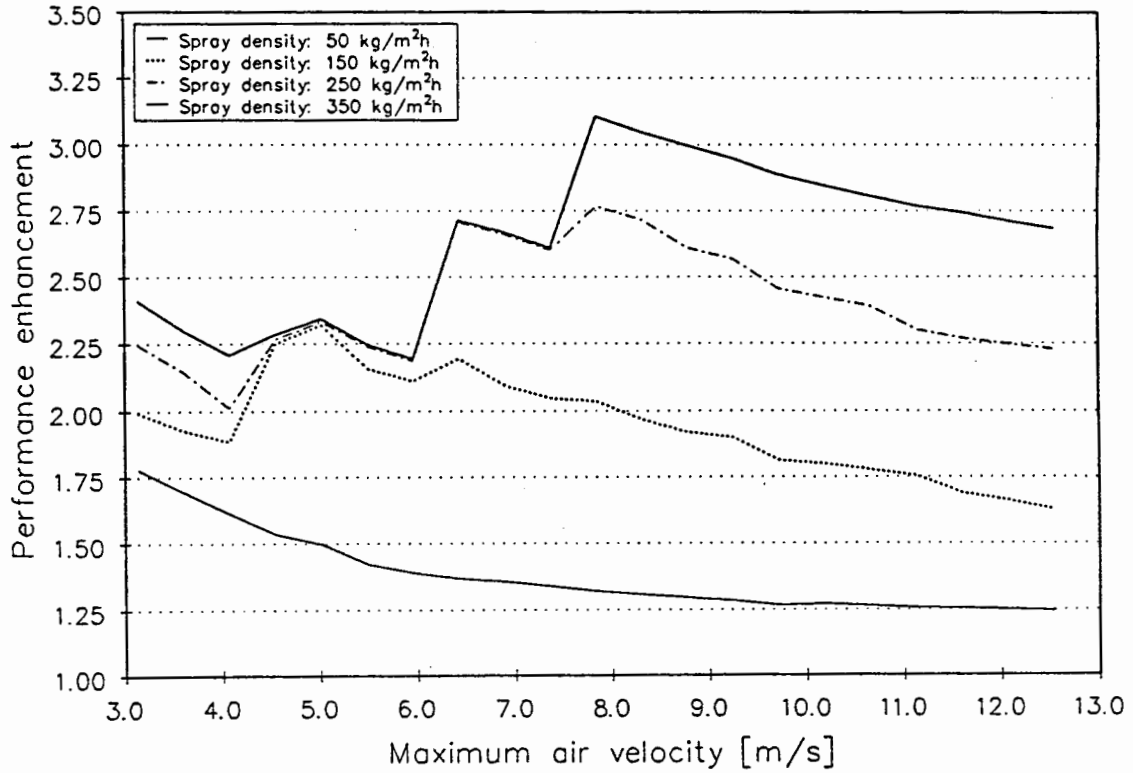


Figure 7.18 - Performance enhancement of the spray cooled heat exchanger with variable air velocity as predicted by the accurate model.

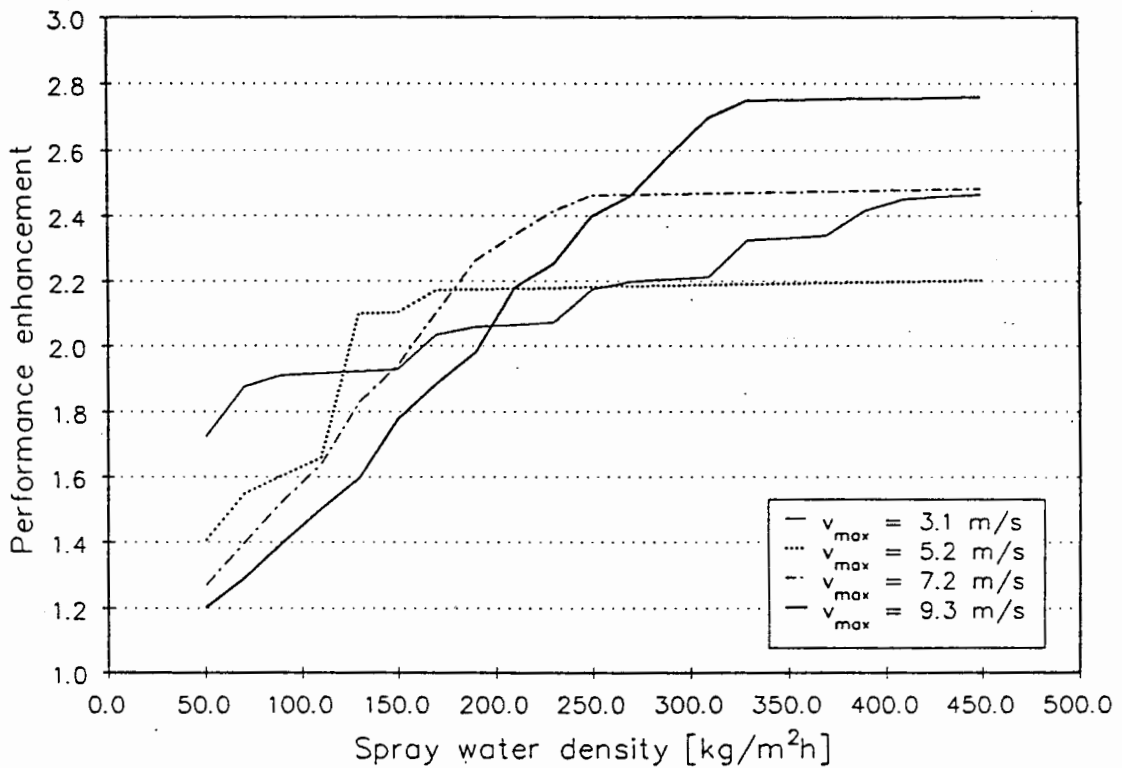


Figure 7.19 - Performance enhancement of the spray cooled heat exchanger with variable spray density as predicted by the accurate model.

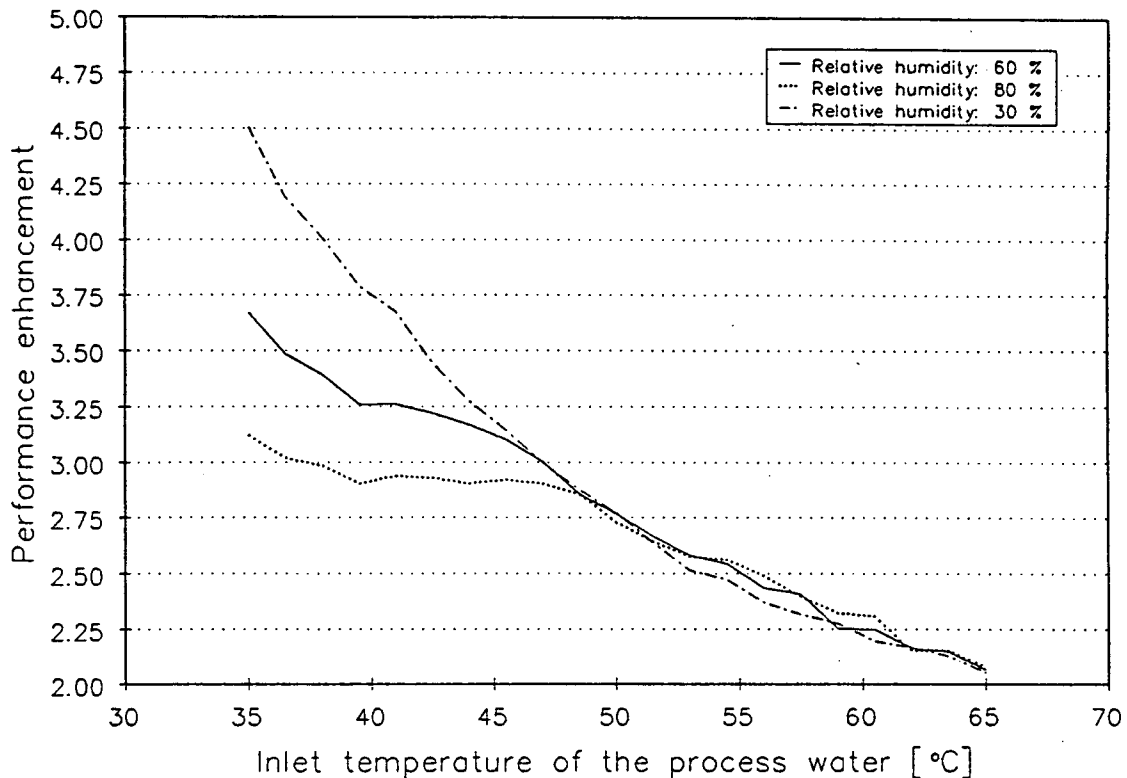


Figure 7.20 - Performance enhancement of the spray cooled heat exchanger with variable process water inlet temperature as predicted by the accurate model.

velocity in the transition zone, e.g. 5.2 m/s in Figure 7.19, is considered, it would make no difference to the performance enhancement in this specific case if more than 150 kg/m²h water is sprayed on the surface of the heat exchanger. On the other hand, for a higher air velocity of 9.3 m/s with the same inlet conditions, a spray density of 325 kg/m²h must be used to obtain the maximum performance enhancement.

From Figure 7.20 it can be seen that the performance enhancement obtained by spray cooling decreases sharply as the process water temperature increases. This result is to be expected as the ratio of the heat transfer potentials of the wet surface and dry surface is much higher at low temperatures than at high temperatures. Another interesting result is the fact that the air humidity plays an important role in the performance enhancement of spray cooling up to a certain temperature, approximately 45 °C in this case, after which it has little influence.

7.2.3 Model SIMPLE

The main advantage of the model SIMPLE is the decrease in computer processing time, relative to the accurate model. The calculation of the performance of spray cooled heat exchangers is up to eight times faster when the model SIMPLE is used. The main drawbacks are the loss of accuracy in both performance prediction and behaviour simulation.

The average absolute error of prediction was found to be 6.91%. Of all the experimental data evaluated, 79% of all predictions were within 10% and 91% within 15% of the measured values. The accuracy of the performance prediction of the simplified model is evaluated in Table 7.2.

Table 7.2 Prediction statistics of the simplified model.

Spray water mass flux [$\text{kg}/\text{m}^2\text{h}$]	Average error [%]	Predict. within 10%	Max. error [%]	Min. error [%]
0 - 50	4.75	95	12.1	-1.77
50 - 100	4.55	92	13.6	-0.91
100 - 150	5.78	91	13.08	-1.5
150 - 200	5.46	90	14.18	-15.57
200 - 250	7.46	74	19.47	-16.16
250 - 300	6.84	80	20.49	-17.16
300 - 350	10.04	59	25.03	-10.74
350 +	7.77	70.2	23.85	-13.94
All data	6.91	79.3	25.49	-17.16

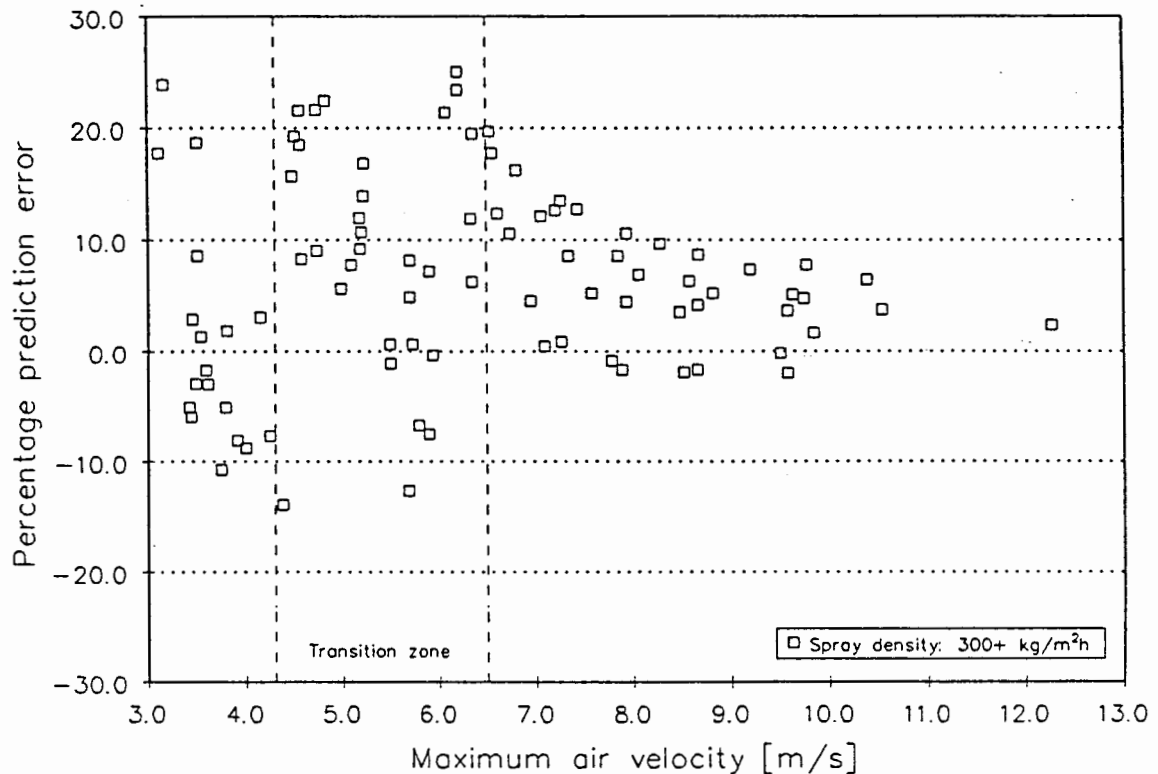


Figure 7.21 - Graphical evaluation of the predicted performance of model SIMPLE for high spray density.

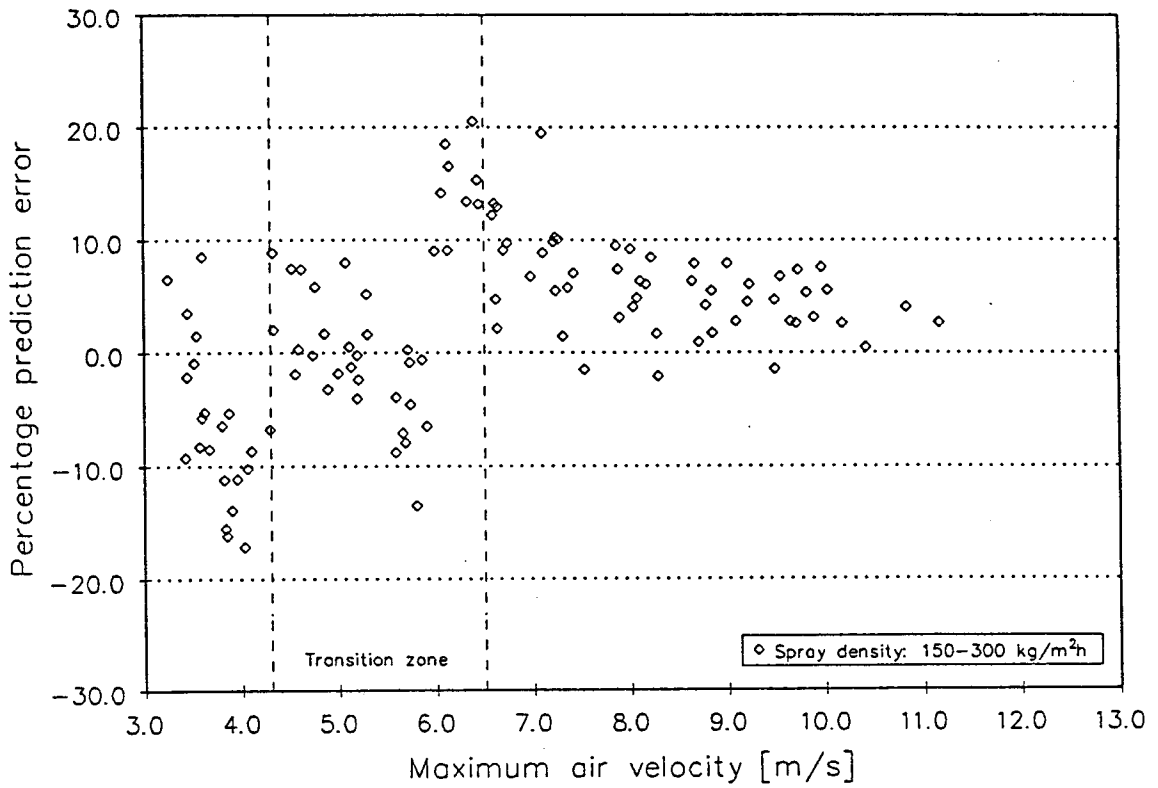


Figure 7.22 - Graphical evaluation of the predicted performance of model SIMPLE for medium spray density.

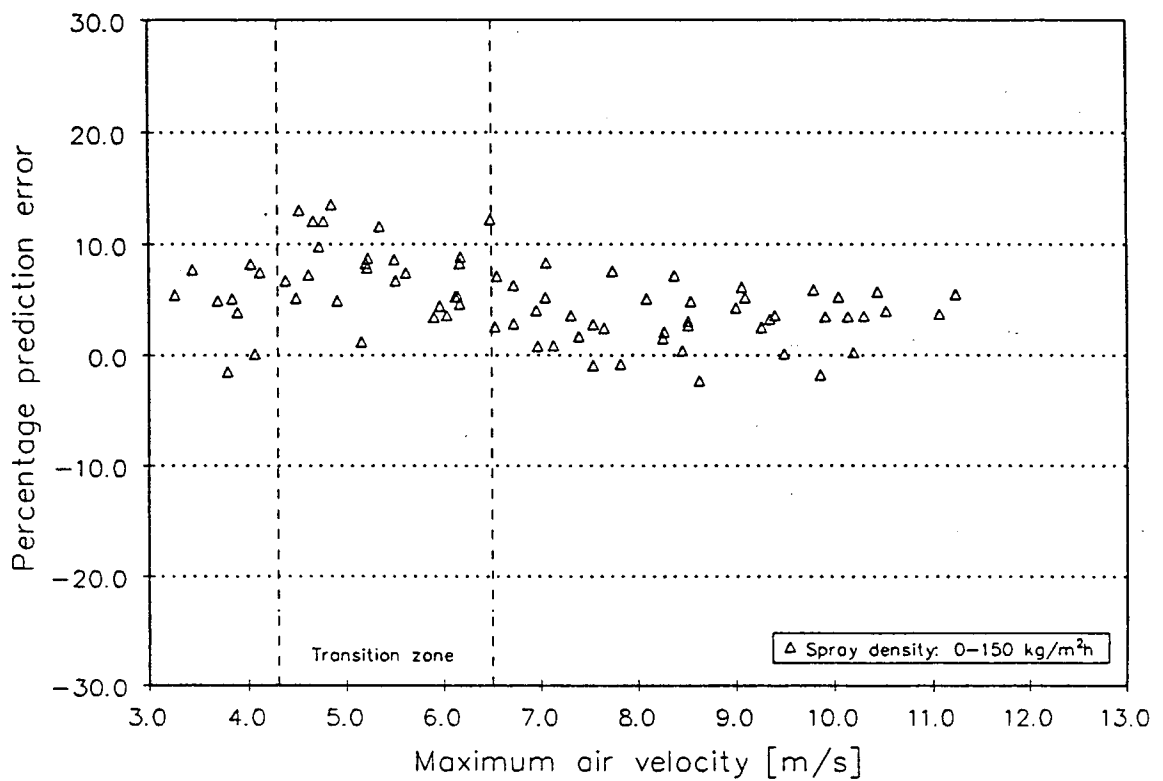


Figure 7.23 - Graphical evaluation of the predicted performance of model SIMPLE for low spray density.

The prediction error of this model is graphically presented in Figures 7.21 to 7.23. From these figures it can be seen that this model performs badly in the transition zone and in the film down flow zone when the high and medium spray density conditions are evaluated. The model's tube-by-tube evaluation of the spray cooled heat exchanger is too crude to give accurate predictions in such an unstable zone. In contrast its performance in the region of up-flowing spray water film is very good. This is mainly the result of the stable behaviour of the spray cooled heat exchanger in this region which can be accurately approximated by the tube-by-tube evaluation method.

The Simplified model's simulation of the physical behaviour of the spray cooled heat exchanger is not as accurate as that of the Accurate model. As Figures 7.24 to 7.26 show, there are serious discrepancies between the predicted temperature drop across each tube row and the measured values in the low, medium and high spray ranges. The graphs show the same steplike changes that was evident in the simulation of model ACCURATE and it is the result of the same step functions controlling the flow of the spray water film and effective wet area ratio.

By comparing Figure 7.24 with the measured temperature drop across the tubes shown in Figure 7.3 it can be seen that heat transfer at low spray densities is over-estimated at low air velocities in the case of the bottom tube row and under-estimated in the case of the third tube row. In the case of medium spray density, Figure 7.24 clearly shows that the heat transfer rate of the third tube row is under-estimated at all air velocities. This results in the prediction of a high heat transfer rate in the bottom tube row.

The prediction of a very high heat transfer rate on the bottom tube row is more pronounced in the high spray water density range. At air velocities above the transition zone, the heat transfer rate of each of the tube passes is almost the same, which is the result of the constant effective area ratio that this model assumes here.

From these results it can be seen that the most important disadvantages of the simplified model are its erratic performance predictions in the transition zone and inferior simulation of the spray cooled heat exchanger's behaviour which is coupled to the assumed value for the wetted area ratio.

This does not mean that the simplified model cannot be used for performance prediction of spray cooled heat exchangers, but care must be taken in the choice of the effective wet

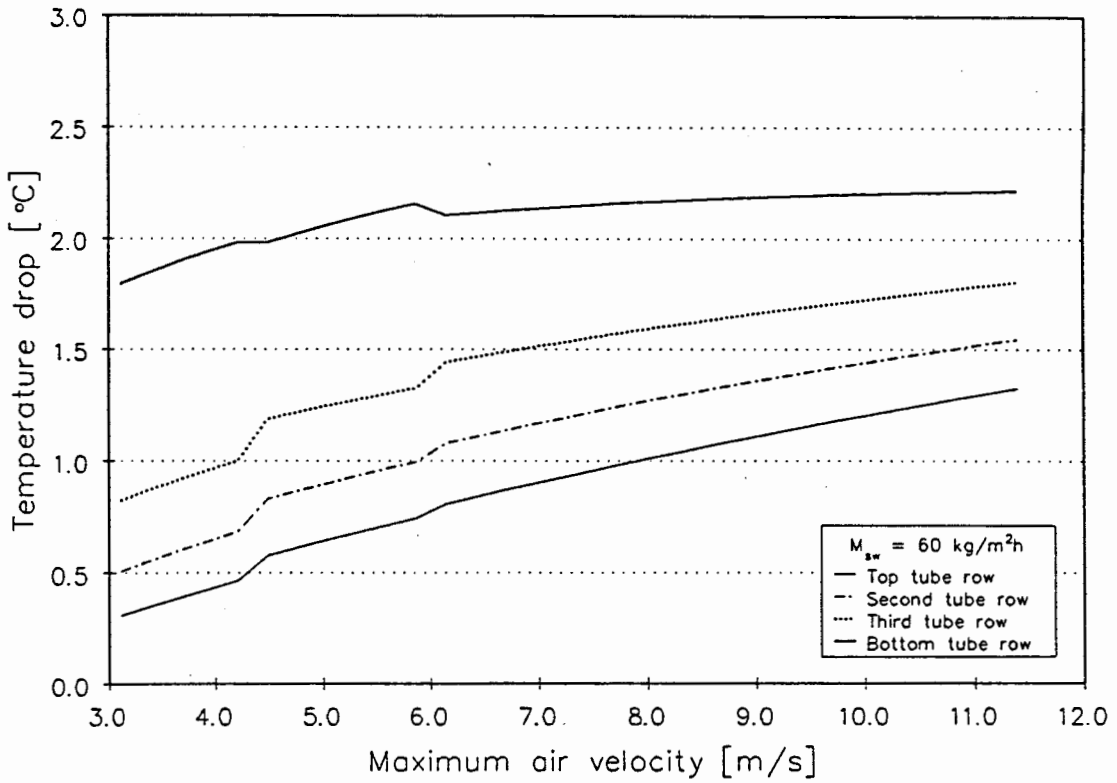


Figure 7.24 - Temperature drop of the process water across each pass of the heat exchanger at low spray water density - model SIMPLE.

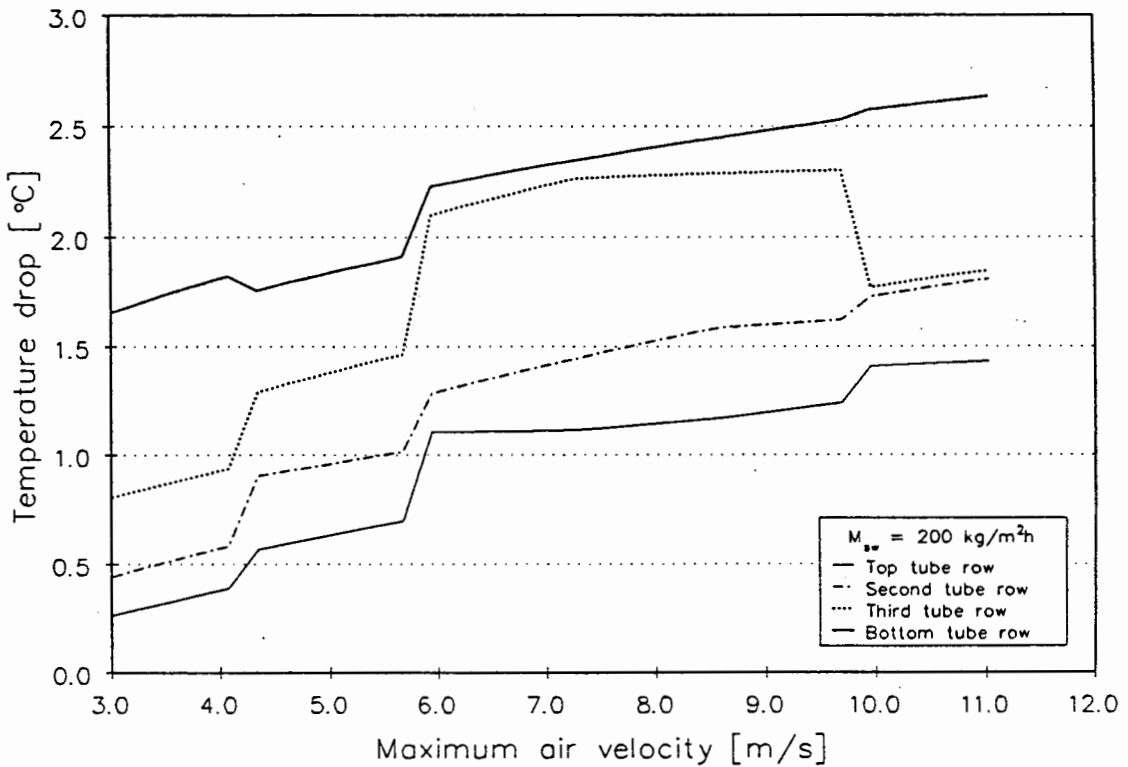


Figure 7.25 - Temperature drop of the process water across each pass of the heat exchanger at medium spray density - model SIMPLE.

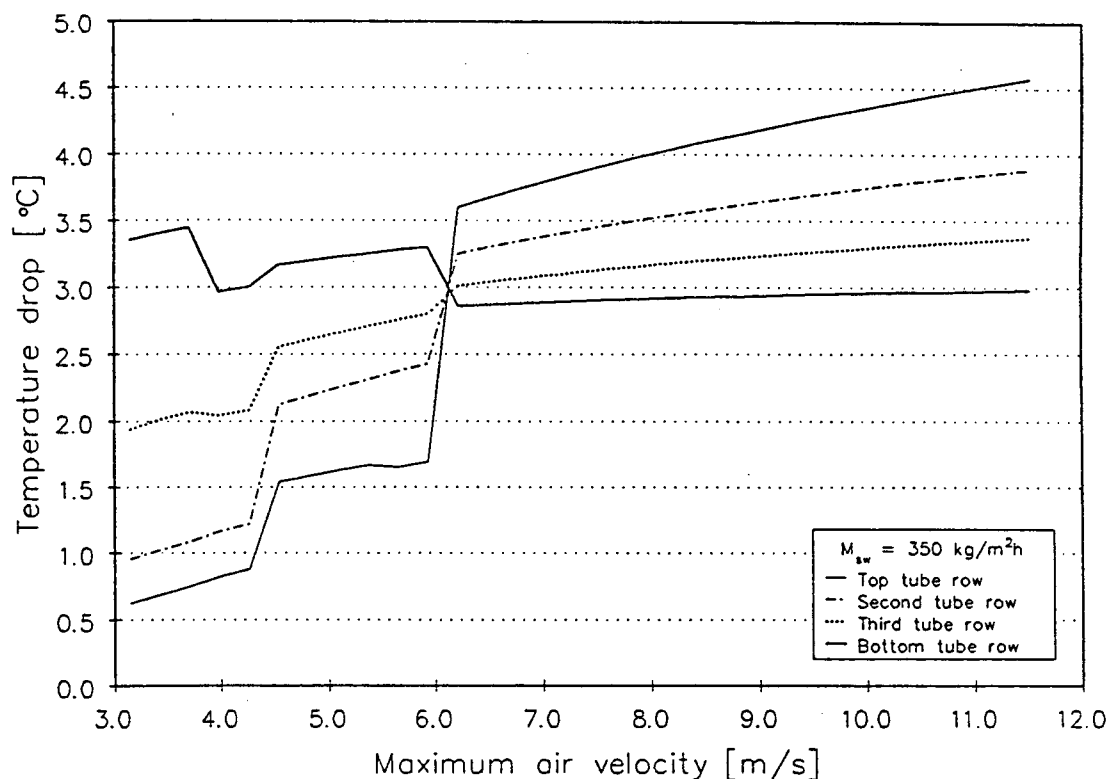


Figure 7.26 - Temperature drop of the process water across each pass of the heat exchanger at high spray density - model SIMPLE.

area ratio. Performance prediction of spray cooled heat exchangers with a maximum air velocity below the critical velocity should be avoided.

To give a direct comparison of the performance prediction of the two models, ACCURATE and SIMPLE, the measured performance against air velocity of a three experiments with different spray densities are graphically shown with the performance predicted by each model.

Figures 7.27 to 7.29 show how the performance of the spray cooled heat exchanger increases with increasing air velocity for different spray densities. The steplike behaviour of the models can also be seen from these figures. The model ACCURATE clearly uses more and smaller steps while the simplified model uses a few large steps with increasing air velocity.

This results in the smaller performance prediction errors of the model ACCURATE that can be seen when Table 7.1 and the graphical evaluation of model ACCURATE in Figures 7.12 to 7.14 is compared to Table 7.2 and the graphical evaluation of model SIMPLE in Figures 7.21 to 7.23.

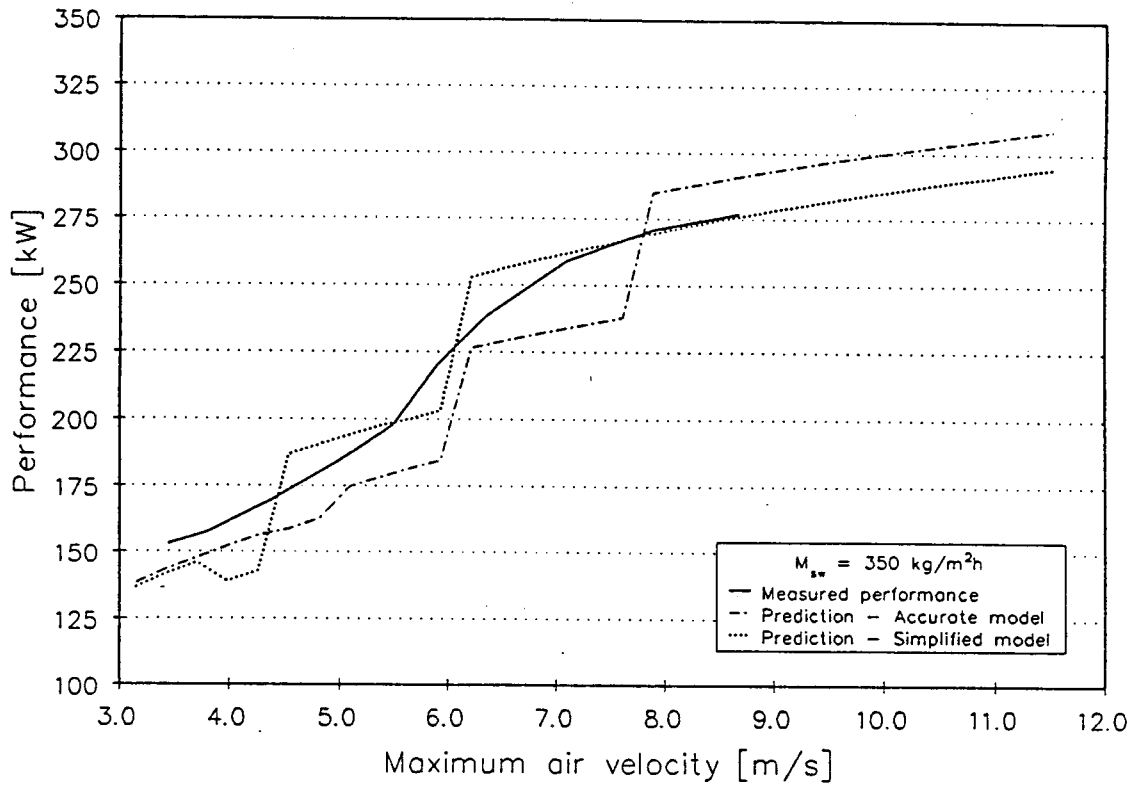


Figure 7.27 - Performance of the spray cooled heat exchanger with increasing air velocity - High spray density.

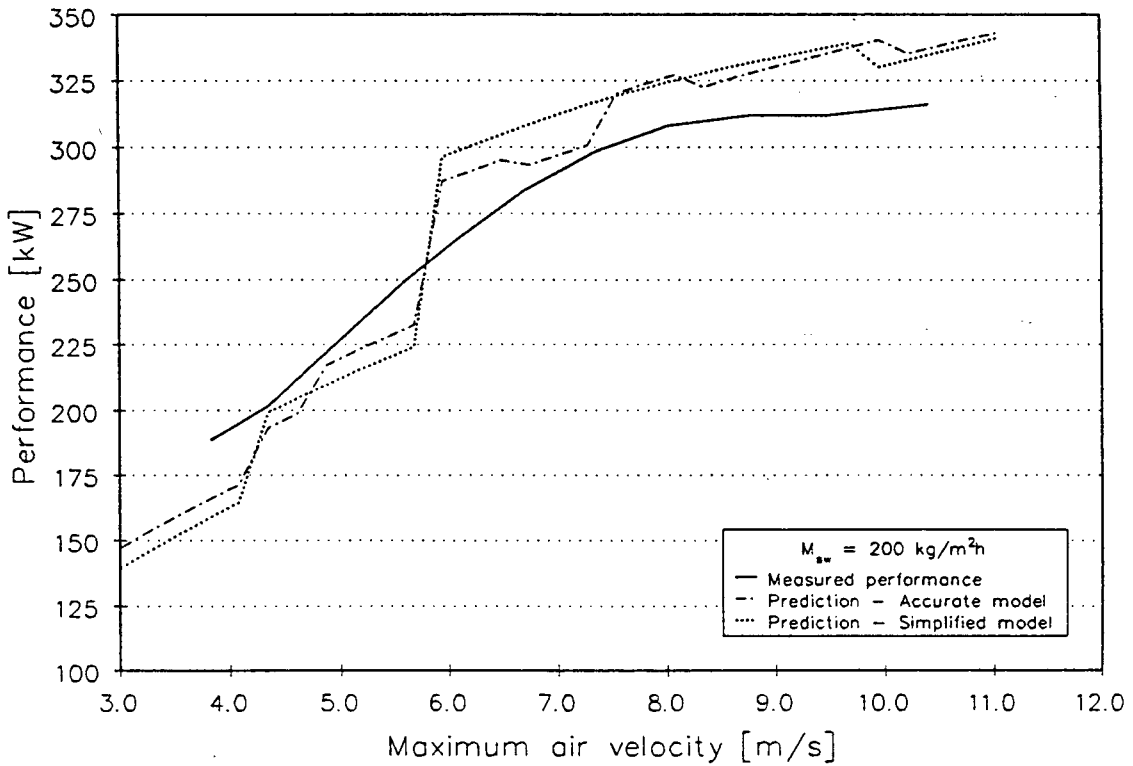


Figure 7.28 - Performance of the spray cooled heat exchanger with increasing air velocity - Medium spray density.

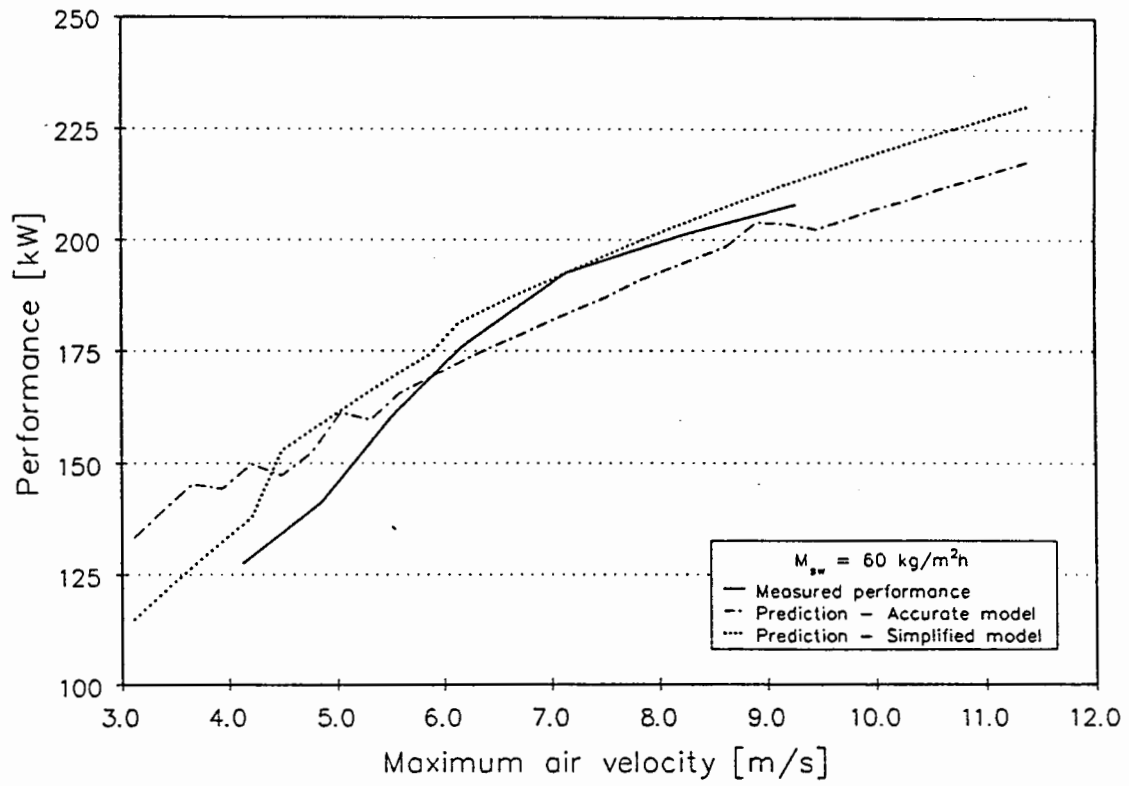


Figure 7.29 - Performance of the spray cooled heat exchanger with increasing air velocity - Low spray density.

8. CONCLUSIONS

From the discussion of the results in Chapter 7, it is clear that the analytical model ACCURATE developed in the present study, can predict the performance of the vertical air-flow spray cooled heat exchanger accurately enough for engineering design purposes. An added advantage of analytical models similar to model ACCURATE which are based on the enthalpy potential as driving force for heat and mass transfer, is that it should predict the performance of heat exchangers with different geometries just as well. This can be seen from model ACCURATE's simulation of the physical behaviour of the spray cooled heat exchanger which is discussed in Chapter 7.

The experimental results indicate that spray cooling of a finned-tube heat exchanger will increase its performance from 1.4 to 3.5 times the dry performance depending on the inlet conditions and spray density. The enhancement rate may be higher for heat exchanger geometries with wider spaced or shorter fins but will be lower for long fins with a small pitch. The increase in pressure drop across the finned-tube heat exchanger remains moderate when spray densities lower than $250 \text{ kg/m}^2\text{h}$ are used. When the spray density is increased beyond this point, a considerable increase in the pressure drop is observed at high air velocities. The increase in pressure drop in this region can be as high as 75% and is the result of spray water congestion in the fin gaps.

The physical behaviour of the vertical air-flow spray cooled heat exchanger can be divided into three regimes depending on the maximum air velocity in the heat exchanger. The first regime includes sub-critical air velocities which means that the spray water film will flow downwards over the heat transfer surface. In this zone only the lower tube rows are wetted while the spray cooling has no effect on the heat transfer rate of the upper tube rows. The second regime is the transition regime which includes air velocities in the region of the critical air velocity, v_{crit} . In this regime the flow direction of the spray water film on the surface of the heat exchanger is uncertain and could be upwards or downwards. As a result of the high air velocities in the heat exchanger in the post-transition regime, the flow direction of the spray water film is upwards. High heat transfer rates which are the result of an increased effective wet area is typical of this regime. This makes it a logical choice for the operation point when designing spray cooled heat exchangers.

The performance enhancement of spray cooled heat exchangers depends on more than just the mass flux of the spray water. The optimum spray density is an important design

parameter which can be determined with the use of an analytical model such as the model ACCURATE presented in the present study. The optimum spray density for a specific heat exchanger depends on the following variables: i) the maximum air velocity ii) the relative humidity of the air and iii) the inlet temperature or heat transfer potential that exists between the air and heat exchanger surface. The pressure drop across the heat exchanger can increase sharply when high density spray cooling is used or heat exchangers with low air velocities across the heat exchanger surface are spray cooled. It is thus important that the pressure drop increase as a result of spray cooling is considered to ensure that the air mass flow rate for which the heat exchanger was designed can be attained. By calculating the optimum spray density, the spray cooled heat exchanger designed for a specific situation will be able to meet the cooling demands in the most economical way possible.

By adapting the computer programs, the analytical models developed in the present study can also be used for horizontal air-flow spray cooled heat exchangers or heat exchangers that are placed at an angle to the air-flow. Further research in these fields is necessary.

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APPENDIX A
PROPERTIES OF FLUIDS

A.1 The thermophysical properties of dry air from 220 K to 380 K at standard atmospheric pressure (101325 N/m²)

Density

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

where $R = 287.08 \text{ J/kgK}$

Specific heat [82AN1]

$$c_{pa} = a + bT + cT^2 + dT^3, \text{ J/kgK} \quad (\text{A.1.2})$$

$$a = 1.045356 \times 10^3$$

$$b = -3.161783 \times 10^{-1}$$

$$c = 7.083814 \times 10^{-4}$$

$$d = -2.705209 \times 10^{-7}$$

Dynamic viscosity [82AN1]

$$\mu_a = a + bT + cT^2 + dT^3, \text{ kg/sm} \quad (\text{A.1.3})$$

$$a = 2.287973 \times 10^{-6}$$

$$b = 6.259793 \times 10^{-8}$$

$$c = -3.131956 \times 10^{-11}$$

$$d = 8.150380 \times 10^{-15}$$

Thermal conductivity

$$k_a = a + bT + cT^2 + dT^3, \text{ W/mK} \quad (\text{A.1.4})$$

$$a = -4.937787 \times 10^{-4}$$

$$b = 1.018087 \times 10^{-4}$$

$$c = -4.627937 \times 10^{-8}$$

$$d = 1.250603 \times 10^{-11}$$

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 W/mK $\times 10^5$	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
280	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

A.4

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	0.0311098	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

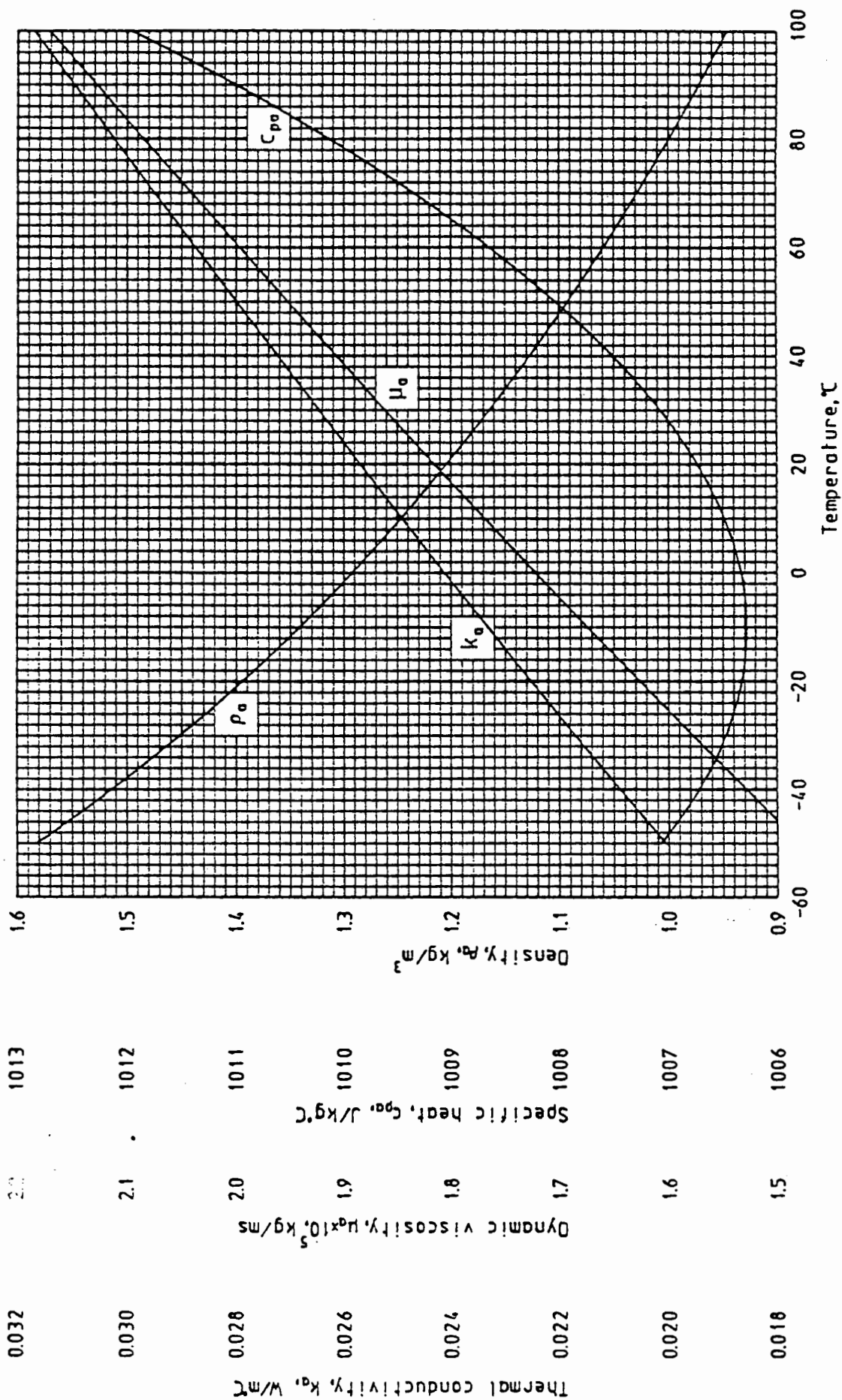


Figure A.1: The thermophysical properties of dry air at standard atmospheric pressure (101325 N/m 2)

A.2 The thermophysical properties of saturated water vapor from 273.15 K to 380 K

Vapor pressure [46GO1]

$$p_v = 10^z, \text{ N/m}^2 \quad (\text{A.2.1})$$

$$z = a(1 - x) + b \log_{10}(x) + c[1 - 10^{d((1/x) - 1)}] + e(10^{f(1 - x)} - 1) + g$$

$$x = 273.16/T$$

$$a = 1.079586 \times 10$$

$$b = 5.028080$$

$$c = 1.504740 \times 10^{-4}$$

$$d = -8.296920$$

$$e = 4.287300 \times 10^{-4}$$

$$f = 4.769550$$

$$g = 2.786118312$$

Specific heat

$$c_{pv} = a + bT + cT^5 + dT^6, \text{ J/kgK} \quad (\text{A.2.2})$$

$$a = 1.3605 \times 10^3$$

$$b = 2.31334$$

$$c = -2.46784 \times 10^{-10}$$

$$d = 5.91332 \times 10^{-13}$$

Dynamic viscosity

$$\mu_v = a + bT + cT^2 + dT^3, \text{ kg/ms} \quad (\text{A.2.3})$$

$$a = 2.562435 \times 10^{-6}$$

$$b = 1.816683 \times 10^{-8}$$

$$c = 2.579066 \times 10^{-11}$$

$$d = -1.067299 \times 10^{-14}$$

Thermal conductivity [82AN1]

$$k_v = a + bT + cT^2 + dT^3, \text{ W/mK} \quad (\text{A.2.4})$$

$$a = 1.3046 \times 10^{-2}$$

$$b = -3.756191 \times 10^{-5}$$

$$c = 2.217964 \times 10^{-7}$$

$$d = -1.111562 \times 10^{-10}$$

Vapor density [70UK1]

$$\rho_v = a + bT + cT^2 + dT^3 + eT^4 + fT^5, \text{ kg/m}^3 \quad (\text{A.2.5})$$

$$a = -4.062329056$$

$$b = 0.10277044$$

$$c = -9.76300388 \times 10^{-4}$$

$$d = 4.475240795 \times 10^{-6}$$

$$e = -1.004596894 \times 10^{-8}$$

$$f = 8.9154895 \times 10^{-12}$$

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 $\times 10^6$	k_v W/mK	α_{v5} $\times 10^5$	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	67.2777	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	1985.92	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

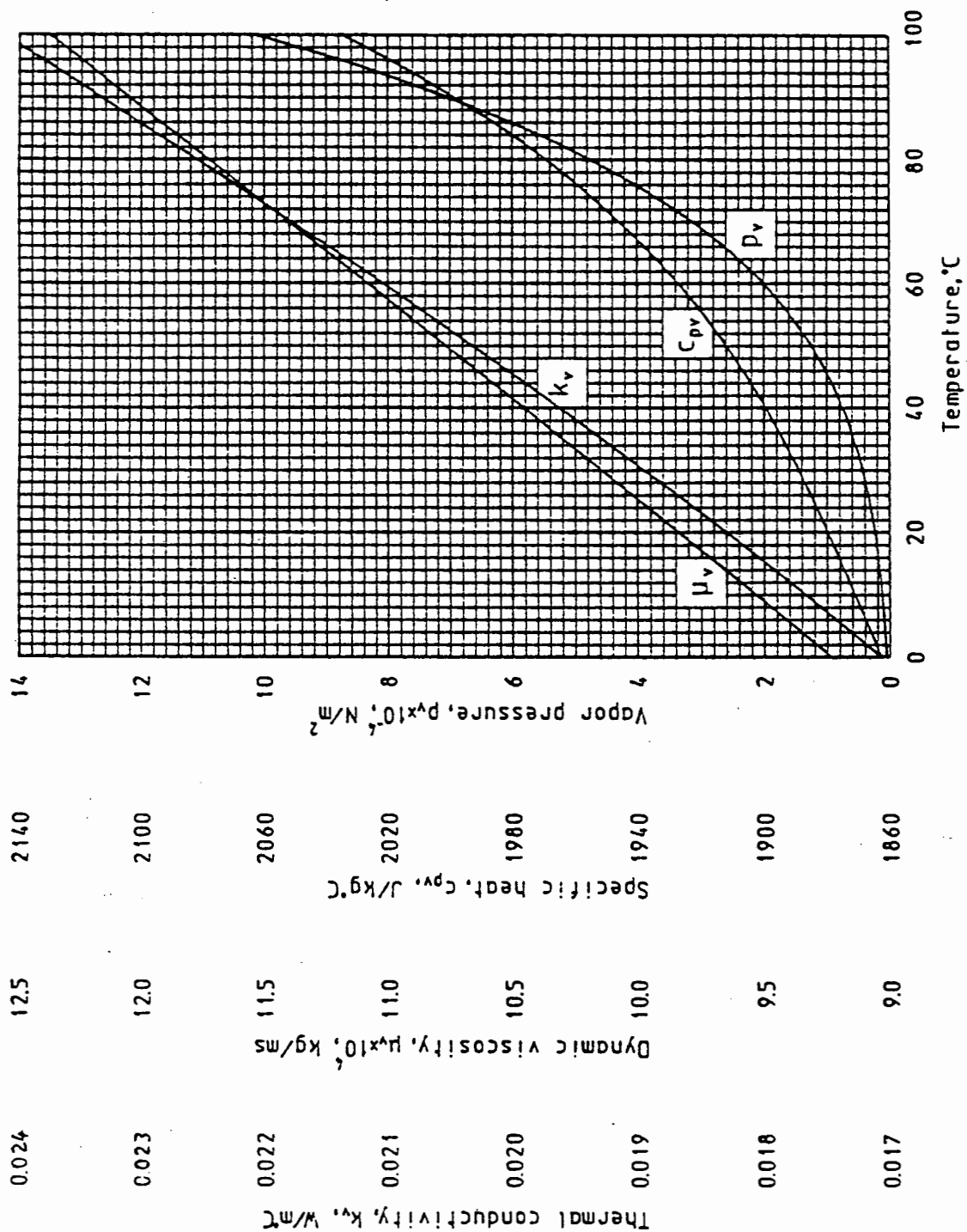


Figure A.2: The thermophysical properties of saturated water vapor

A.3 The thermophysical properties of mixtures of dry air and water vapor.

Density [72AS1]

$$\rho_{av} = (1 + w) [1 - w/(w + 0.62198)] (p_{abs}/RT), \text{kg/m}^3 \quad (\text{A.3.1})$$

where $R = 287.08 \text{ J/kgK}$

Specific heat [78FA1]

$$c_{pav} = (c_{pa} + wc_{pv})/(1 + w), \text{J/kgK} \quad (\text{A.3.2})$$

Dynamic viscosity [54GO1]

$$\mu_{av} = (X_a \mu_a M_a^{0.5} + X_v \mu_v M_v^{0.5}) / (X_a M_a^{0.5} + X_v M_v^{0.5}), \text{kg/ms} \quad (\text{A.3.3})$$

Thermal conductivity [57LE1]

$$k_{av} = (X_a k_a M_a^{0.33} + X_v k_v M_v^{0.33}) / (X_a M_a^{0.33} + X_v M_v^{0.33}), \text{W/mK} \quad (\text{A.3.4})$$

where $M_a = 28.97 \text{ kg/mole}$, $M_v = 18.016 \text{ kg/mole}$, $X_a = 1/(1 + 1.608 w)$ and $X_v = w/(w + 0.622)$

Humidity ratio

$$w = \left(\frac{2501.6 - 2.3263(T_{wb} - 273.15)}{2501.6 + 1.8577(T_{db} - 273.15) - 4.184(T_{wb} - 273.15)} \right) \times \left(\frac{0.62509 p_{vwb}}{p_{abs} - 1.005 p_{vwb}} \right) - \left(\frac{1.00416 (T_{db} - T_{wb})}{2501.6 + 1.8577(T_{db} - 273.15) - 4.184(T_{wb} - 273.15)} \right), \text{kg/kg} \quad (\text{A.4.3})$$

Enthalpy

$$i_{av} = c_{pa} T + w(i_{fgw} + c_{pv} T), \text{ J/kg} \quad (\text{A.3.6})$$

A.4 The thermophysical properties of saturated waterliquid from 273.15K to 380 K

Density

$$\rho_w = (a + bT + cT^2 + dT^6)^{-1}, \text{ kg/m}^3 \quad (\text{A.4.1})$$

$$a = 1.49343 \times 10^{-3}$$

$$b = -3.7164 \times 10^{-6}$$

$$c = 7.09782 \times 10^{-9}$$

$$d = -1.90321 \times 10^{-20}$$

Specific heat

$$c_{pw} = a + bT + cT^2 + dT^6, \text{ J/kgK} \quad (\text{A.4.2})$$

$$a = 8.15599 \times 10^3$$

$$b = -2.80627 \times 10$$

$$c = 5.11283 \times 10^{-2}$$

$$d = -2.17582 \times 10^{-13}$$

Dynamic viscosity [82AN1]

$$\mu_w = a10^{b/(T - c)}, \text{ kg/ms} \quad (\text{A.4.3})$$

$$a = 2.414 \times 10^{-5}$$

$$b = 247.8$$

$$c = 140$$

Thermal conductivity

$$k_w = a + bT + cT^2 + dT^4, \text{ W/mK} \quad (\text{A.4.4})$$

$$a = -6.14255 \times 10^{-1}$$

$$b = 6.9962 \times 10^{-3}$$

$$\begin{aligned}c &= -1.01075 \times 10^{-5} \\d &= 4.74737 \times 10^{-12}\end{aligned}$$

Latent heat of vaporization

$$i_{fgw} = a + bT + cT^2 + dT^3, \text{ J/kg} \quad (\text{A.4.5})$$

$$\begin{aligned}a &= 3.4831814 \times 10^6 \\b &= -5.8627703 \times 10^3 \\c &= 1.2139568 \times 10 \\d &= -1.40290431 \times 10^{-2}\end{aligned}$$

Critical pressure

$$p_{wc} = 22.09 \times 10^6, \text{ N/m}^2 \quad (\text{A.4.6})$$

Surface tension [70UK1]

$$\sigma = a + bt + cT^2 + dT^3, \text{ N/m} \quad (\text{A.4.7})$$

$$\begin{aligned}a &= 5.148103 \times 10^{-2} \\b &= 3.998714 \times 10^{-4} \\c &= -1.4721869 \times 10^{-6} \\d &= 1.21405335 \times 10^{-9}\end{aligned}$$

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 x 10 ⁴	k_w W/mK	β_w 1/K x 10 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	976.141	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

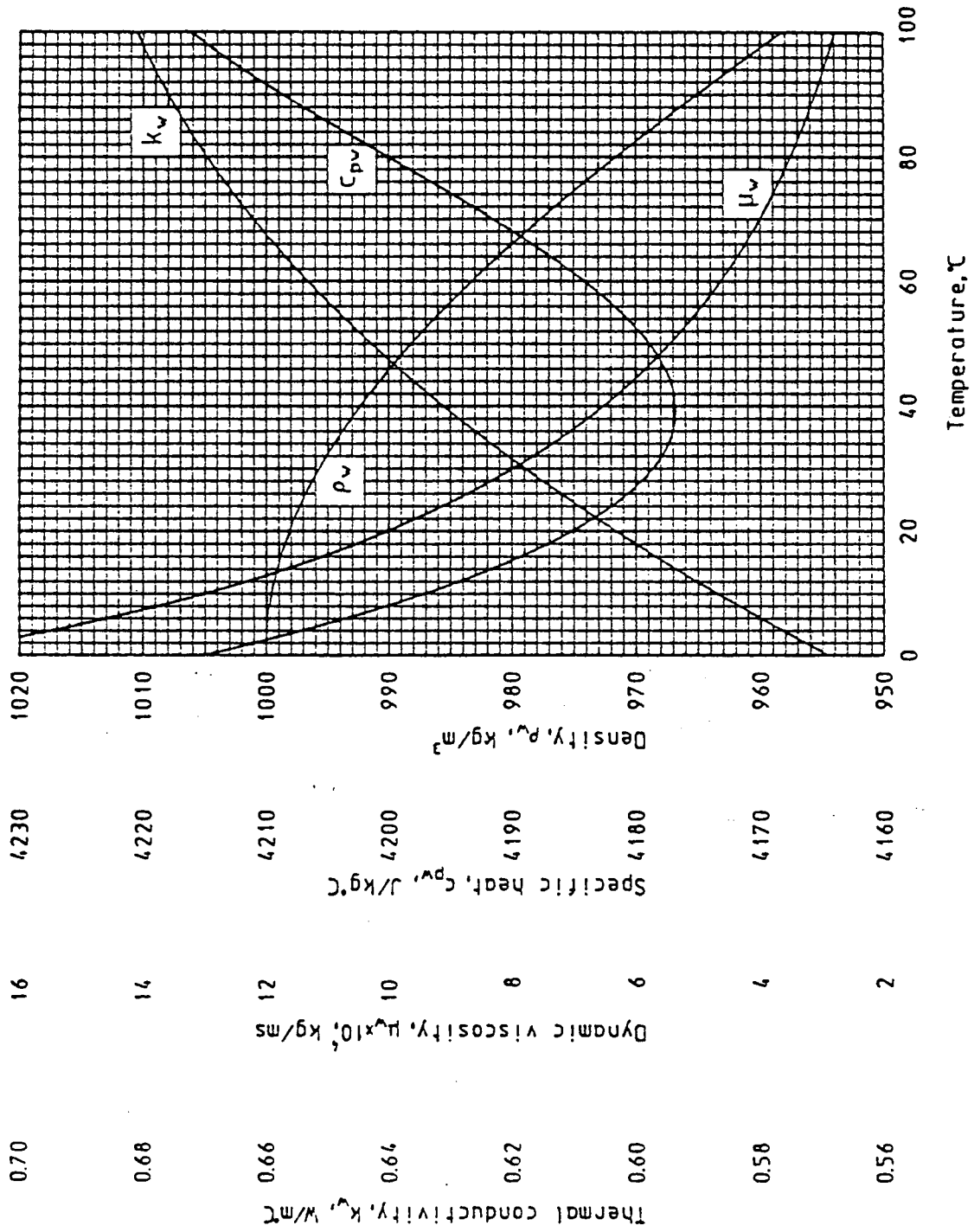


Figure A.3: The thermophysical properties of saturated water liquid

A.5 The thermophysical properties of saturated ammonia vapor.

Vapor pressure [76RA1], (230 K to 395 K)

$$P_{\text{ammv}} = a + bT + cT^2 + dT^3 + eT^4, \text{ N/m}^2 \quad (\text{A.5.1})$$

$$a = 1.992448 \times 10^6$$

$$b = -57.568140 \times 10^3$$

$$c = 0.5640265 \times 10^3$$

$$d = -2.337352$$

$$e = 3.541430 \times 10^{-3}$$

Density [76RA1], (260 K to 390 K)

$$\rho_{\text{ammv}} = a + bT + cT^2 + dT^3 + eT^4, \text{ kg/m}^3 \quad (\text{A.5.2})$$

$$a = -6.018936 \times 10^2$$

$$b = 5.361048$$

$$c = -1.187296 \times 10^{-2}$$

$$d = -1.161479 \times 10^{-5}$$

$$e = 4.739058 \times 10^{-8}$$

Specific heat [72AS1], (230 K to 325 K)

$$c_{p\text{ammv}} = a + bT + cT^2 + dT^3; \text{ J/kgK} \quad (\text{A.5.3})$$

$$a = -2.7761190256 \times 10^4$$

$$b = 3.39116449 \times 10^2$$

$$c = -1.3055687$$

$$d = -1.728649 \times 10^{-3}$$

Dynamic viscosity [72AS1], (240 K to 370 K)

$$\mu_{\text{ammv}} = a + bT + cT^2 + dT^3 + eT^4, \text{ kg/sm} \quad (\text{A.5.4})$$

$$a = -2.748011 \times 10^{-5}$$

$$b = 2.82526 \times 10^{-7}$$

$$c = -5.201831 \times 10^{-10}$$

$$d = -6.061761 \times 10^{-13}$$

$$e = 2.126070 \times 10^{-15}$$

Thermal conductivity [72AS1], (245 K to 395 K)

$$k_{\text{ammv}} = a + bT + cT^2 + dT^3 + eT^4, \text{ W/mK} \quad (\text{A.5.6})$$

$$a = -0.1390216$$

$$b = 1.35238 \times 10^{-3}$$

$$c = -2.532035 \times 10^{-6}$$

$$d = -4.884341 \times 10^{-9}$$

$$e = 1.418657 \times 10^{-11}$$

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	$c_{p\text{ammv}}$ J/kgK	μ_{ammv} kg/sm x 10 ⁶	k_{ammv} W/mK	Pr_{ammv}
230	60.58		2230.48			
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	1061.74	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	

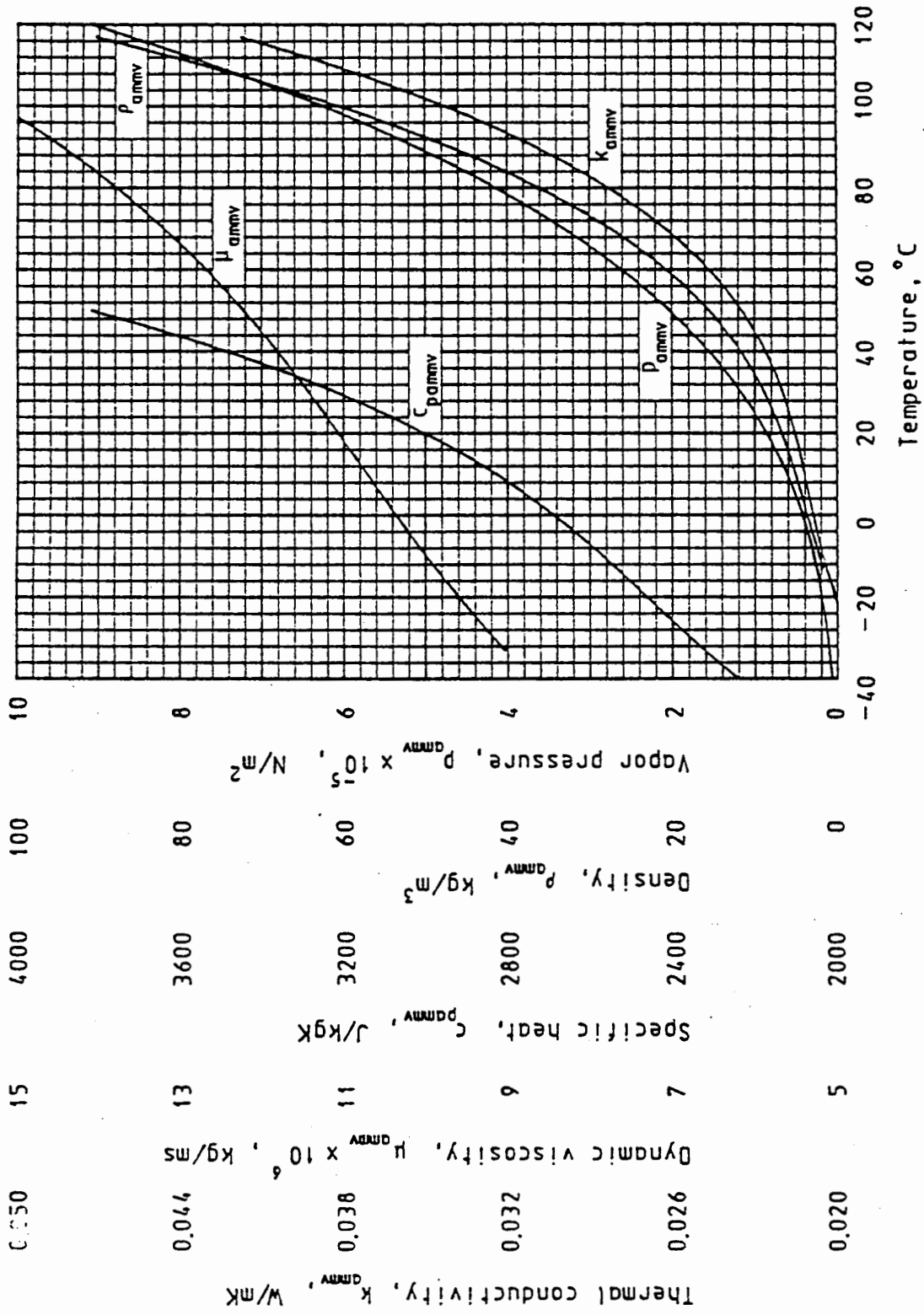


Figure A.4: Thermophysical properties of saturated ammonia vapor

A.6. The thermophysical properties of saturated ammonia liquid from 200 K to 405 K.

Density [77YA1]

$$\rho_{\text{amm}} = ab \left[- (1 - T/T_c)^{0.285714} \right], \text{kg/m}^3 \quad (\text{A.6.1})$$

$$a = 2.312 \times 10^2$$

$$b = 0.2471$$

$$T_c = 405.5 \text{ K}$$

Specific heat [77YA1], (200 K to 375 K)

$$c_{\text{pamm}} = a + bT + cT^2 + dT^3, \text{J/kgK} \quad (\text{A.6.2})$$

$$a = -2.497276939 \times 10^3$$

$$b = 7.7813907 \times 10$$

$$c = -3.006252 \times 10^{-1}$$

$$d = 4.06714 \times 10^{-4}$$

Dynamic viscosity [77YA1]

$$\mu_{\text{amm}} = 0.001 \times 10^{(a + b/T + cT + dT^2)}, \text{kg/sm} \quad (\text{A.6.3})$$

$$a = -8.591$$

$$b = 876.4$$

$$c = 0.02681$$

$$d = -3.612 \times 10^{-5}$$

Thermal conductivity [77YA1], (200 K to 375 K)

$$k_{\text{amm}} = a + bT + cT^2, \text{W/mK} \quad (\text{A.6.4})$$

$$a = 1.068229$$

$$b = -1.576908 \times 10^{-3}$$

$$c = -1.228884 \times 10^{-6}$$

Latent heat of vaporization [77YA1]

$$i_{fgamm} = [(b - T)/(b - c)]^d, \text{J/kg} \quad (\text{A.6.5})$$

$$a = 1.370758 \times 10^6$$

$$b = 405.55$$

$$c = 239.72$$

$$d = 0.38$$

Critical pressure

$$P_{ammc} = 11.28 \times 10^6 \text{ N/m}^2 \quad (\text{A.6.6})$$

Surface tension [77YA1]

$$\sigma = 0.0366[(405.55 - T)/177.4]^{1.1548}, \text{N/m} \quad (\text{A.6.7})$$

Table A.5: The thermophysical properties of saturated ammonia liquid.

T K	ρ_{pamm} kg/m ³	c_{pamm} J/kgK	μ_{amm} kg/sm x 10 ⁵	k_{amm} W/mk	Pr_{amm}	i_{fgamm} J/kg x 10 ⁻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	4550.48	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	51.92.66	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

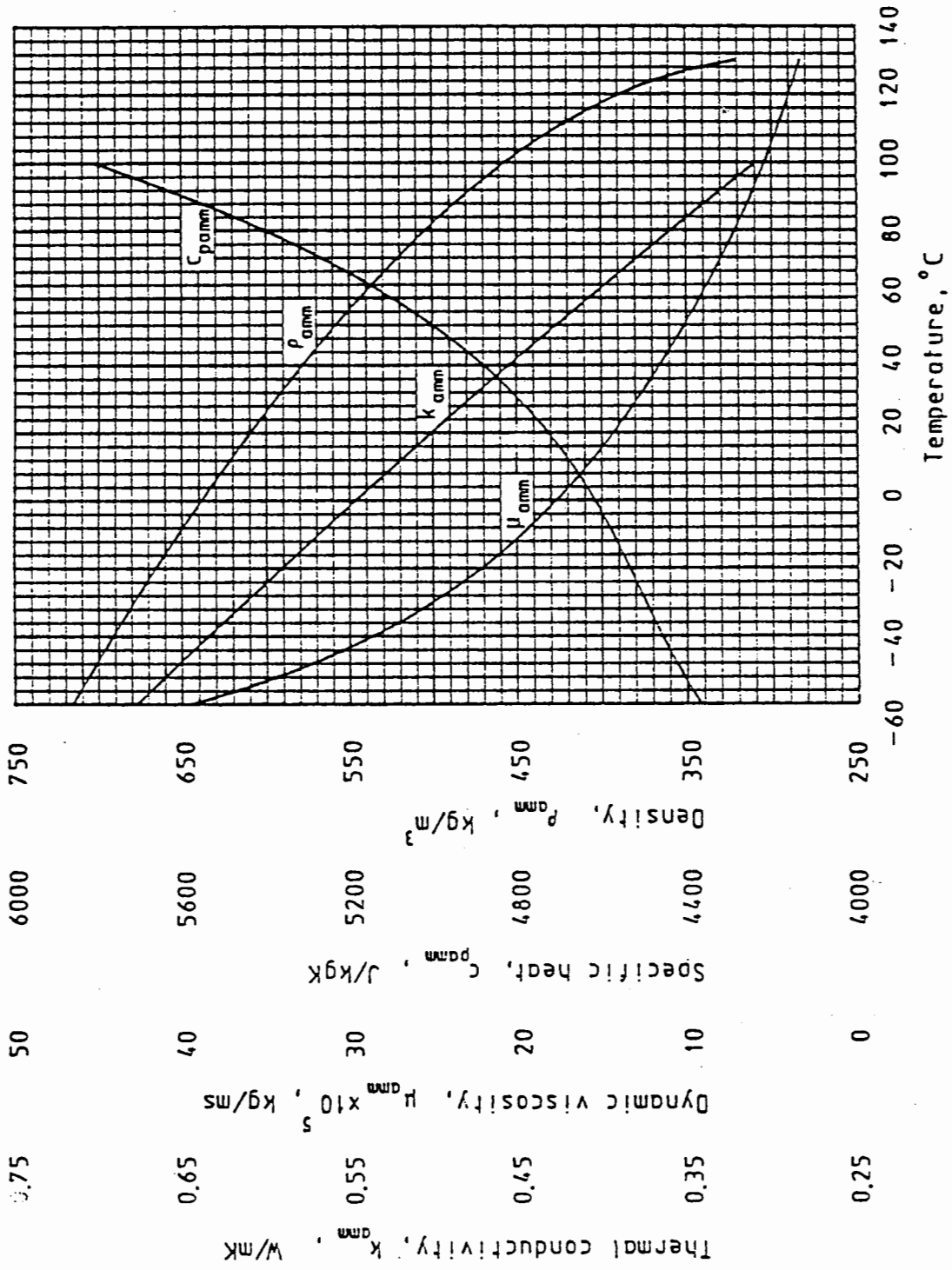


Figure A.5: Thermophysical properties of saturated ammonia liquid

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APPENDIX B**TEMPERATURE CORRECTION FACTOR**

To determine the performance characteristics of heat exchangers it is essential that the mean temperature difference between fluids be accurately determined. Since this difference depends on the geometry and flow pattern through the heat exchanger, relatively simple analytic solutions are not always possible. Roetzel [79RO1], [80RO1], [84RO1] proposed a method that is effective for computational purposes.

For a heat exchanger such as the experimental model in the present study the heat transfer rate is given by

$$q = UA\Delta T_m \quad (\text{B.1})$$

where UA is the conductance of the heat exchanger. The overall heat transfer coefficient is assumed to be constant. From equation (B.1) the mean temperature difference between the hot and cold streams, ΔT_m , can be expressed as

$$\Delta T_m = \frac{Q}{UA} \quad (\text{B.2})$$

Equation (B.2) can be made dimensionless with respect to the largest temperature difference

$$\phi = \frac{Q}{UA(T_{hi} - T_{ci})} \quad (\text{B.3})$$

and the dimensionless temperature changes of the two streams may respectively be defined as

$$\phi_h = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \quad (\text{B.4})$$

$$\phi_c = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \quad (\text{B.5})$$

In the case of counterflow, a dimensionless mean temperature difference can be defined in terms of the logarithmic mean temperature differences i.e.

According to Roetzel [84RO1], a temperature correction factor can in general be expressed as

The values for the empirical constant, $a_{i,k}$ for a crossflow heat exchanger with four tube rows

$$\phi_{cf} = \frac{\Delta T_{lm}}{T_{hi} - T_{ci}} = \frac{\phi_h - \phi_c}{\ln[(1 - \phi_c)/(1 - \phi_h)]} \quad (\text{B.6})$$

$$F_T = 1 - \sum_{i=1}^4 \sum_{k=1}^4 a_{i,k} (1 - \phi_{cf})^k \sin[2i \arctan(\phi_h / \phi_c)] \quad (\text{B.7})$$

and four tube passes, are presented in Table B.1.

Table B.1 - Cross flow with four tube rows and four tube passes

$a_{i,k}$	$i=1$	2	3	4
$k=1$	3.39×10^{-1}	2.77×10^{-2}	1.79×10^{-1}	-1.99×10^{-2}
2	2.38×10^0	-9.99×10^{-2}	-1.21×10^0	4.00×10^{-2}
3	-5.26×10^0	9.04×10^{-2}	2.62×10^0	4.94×10^{-2}
4	3.90×10^0	-8.45×10^{-4}	-1.81×10^0	-9.81×10^{-2}

APPENDIX C

CALCULATION OF FIN EFFECTIVITY

The fin effectivity of elliptical fins is not as simple to calculate as that of radial or straight fins. As the real value of the fin effectivity will lie between these two, the fin effectivity of elliptical fins can be approximated by either radial or straight fins depending on the geometry of the elliptical tube. If the ellipse is more flat than round, the fin effectivity of straight fins can be used and if the ellipse is almost round, the fin effectivity can be approximated by the radial fin effectivity.

The fin height is 9.5 mm and the area of a single elliptical fin is $1.0416 \times 10^{-3} \text{ m}^2$. Using the areas of the tube and fin of the elliptical finned-tube, a fictitious cylindrical finned-tube with the same tube and fin areas and the same fin height can be defined. This resulted in a circular finned-tube with an outer tube diameter of 24.4 mm and fin height of 9.5 mm. This tube can now be used in the calculation of the fin effectivity of the elliptical finned-tube as with the well known equation proposed by Schmidt [46SC1]

$$e_f = \frac{\tanh(b d_r \phi/2)}{b d_r \phi/2} \quad (\text{C.1})$$

where

$$b = \left[\frac{2h_a}{k_f t_f} \right]^{0.5} \quad (\text{C.2})$$

$$\phi = \left(\frac{d_f}{d_r} - 1 \right) \left\{ 1 + 0.35 \ln \left(\frac{d_r}{d_f} \right) \right\} \quad (\text{C.3})$$

If the fin is considered to be straight, the following equation may be used for the calculation of the fin effectivity

$$e_f = \frac{\tanh(mL)}{mL} \quad (\text{C.4})$$

where the expression for mL may be written for one-dimensional flow as

$$mL = \sqrt{\frac{2h_a}{k_f A_m}} L^{2/3} \quad (C.5)$$

$$A_m = L_f t_f \quad (C.6)$$

Depending on the air and process water temperature, the finned surface effectivity, ϵ_o , calculated using equation (C.1) varied from 77% to 87% while the effectivity calculated using equation (C.4) varied from 83% to 90%.

The fin has a total thickness of 0.48 mm and consists of a steel core, with a conductivity of 43 W/m °C, coated on both sides by a 0.07 mm thick layer of zinc with a conductivity of 112 W/m °C. The resistance of the fin root and fin to conduction of heat were calculated for the circular fin and straight fin using the following equations respectively.

$$\sum \frac{R}{A} = \frac{1}{n_t n_g L_t} \left\{ \frac{\ln(d_o/d_i)}{2\pi k_t} + \frac{\ln(d_r/d_o)}{2\pi k_f} \right\} \quad (C.7)$$

$$\sum \frac{R}{A} = \frac{1}{n_t n_g L_t} \left\{ \frac{t_t}{P_t k_t} + \frac{t_f}{P_f k_f} \right\} \quad (C.8)$$

and were found to be 3.4996×10^{-6} °C/W for the circular finned-tubes and 3.0705×10^{-6} °C/W for the straight fins configuration.

Using the data in Tables 4.1 to 4.3 the dry heat transfer coefficient for the experimental model were calculated with the use of the software package Lotus 1-2-3, and the results are shown in Figure C.1. From this figure it can be seen that the difference between these two curves is very small and therefore the choice of straight or radial fins will not influence the accuracy of the calculations to a large extent. For the experimental model of the present study the straight fin approximation was chosen as the shape of the elliptical tubes were very flat.

C3

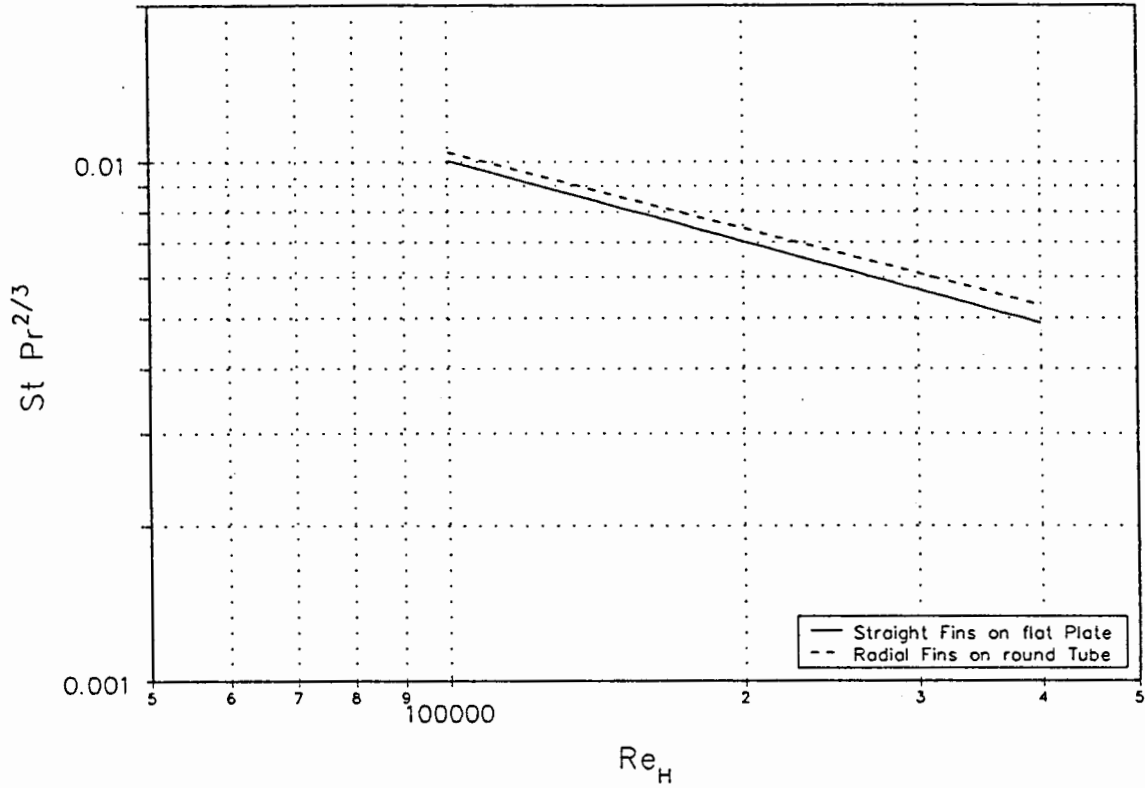


Figure C.1 - Performance characteristics of the dry heat exchanger.

APPENDIX D

SOLUTION OF SIMULTANEOUS DIFFERENTIAL EQUATIONS USING THE 4TH ORDER RUNGE-KUTTA METHOD

The Runge-Kutta method approximates the solution the differential equation along a specified grid. The initial value differential equation problem can be written as

$$\frac{dy}{dx} = f(x,y) \quad (\text{D.1})$$

$$y(x_0) = y_0$$

The function value y_{n+1} can be calculated by

$$y_{n+1} = y_n + \frac{(a_1+2a_2+2a_3+a_4)}{6} \quad \text{for } n = 0,1,2 \dots \quad (\text{D.2})$$

$$a_1 = h f(x_n, y_n)$$

$$a_2 = h f(x_n+h/2, y_n+a_1/2)$$

$$a_3 = h f(x_n+h/2, y_n+a_2/2)$$

$$a_4 = h f(x_n+h, y_n+a_3)$$

This method is self starting (no initial estimates are needed) and the new function values can be calculated directly after calculating the a_n variables. The fourth order Runge-Kutta method can be extended to a system of first order initial value problems as shown by Van Iwaarden [77VA1]. In the present study four differential equations are solved simultaneously. The problem can be written as

$$\frac{dy}{dx} = f(x,y,z,u,v) \quad (\text{D.3})$$

$$y(x_0) = y_0$$

$$\frac{dz}{dx} = g(x,y,z,u,v) \quad (\text{D.4})$$

$$z(x_0) = z_0$$

$$\frac{du}{dx} = k(x,y,z,u,v) \quad (\text{D.5})$$

$$u(x_0) = u_0$$

$$\frac{dv}{dx} = j(x,y,z,u,v) \quad (\text{D.6})$$

$$v(x_0) = v_0$$

The fourth order Runge-Kutta method now becomes

$$y_{n+1} = y_n + \frac{(a_1+2a_2+2a_3+a_4)}{6} \quad (\text{D.7})$$

$$z_{n+1} = z_n + \frac{(b_1+2b_2+2b_3+b_4)}{6} \quad (\text{D.8})$$

$$u_{n+1} = u_n + \frac{(c_1+2c_2+2c_3+c_4)}{6} \quad (\text{D.9})$$

$$v_{n+1} = v_n + \frac{(d_1+2d_2+2d_3+d_4)}{6} \quad (\text{D.10})$$

where

$$a_1 = h f(x_n, y_n, z_n, u_n, v_n)$$

$$b_1 = h g(x_n, y_n, z_n, u_n, v_n)$$

$$c_1 = h k(x_n, y_n, z_n, u_n, v_n)$$

$$d_1 = h j(x_n, y_n, z_n, u_n, v_n)$$

$$a_2 = h f(x_n+h/2, y_n+a_1/2, z_n+b_1/2, u_n+c_1/2, v_n+d_1/2)$$

$$b_2 = h g(x_n+h/2, y_n+a_1/2, z_n+b_1/2, u_n+c_1/2, v_n+d_1/2)$$

$$c_2 = h k(x_n+h/2, y_n+a_1/2, z_n+b_1/2, u_n+c_1/2, v_n+d_1/2)$$

$$d_2 = h j(x_n+h/2, y_n+a_1/2, z_n+b_1/2, u_n+c_1/2, v_n+d_1/2)$$

$$a_3 = h f(x_n+h/2, y_n+a_2/2, z_n+b_2/2, u_n+c_2/2, v_n+d_2/2)$$

$$b_3 = h g(x_n+h/2, y_n+a_2/2, z_n+b_2/2, u_n+c_2/2, v_n+d_2/2)$$

$$c_3 = h k(x_n+h/2, y_n+a_2/2, z_n+b_2/2, u_n+c_2/2, v_n+d_2/2)$$

$$d_3 = h j(x_n+h/2, y_n+a_2/2, z_n+b_2/2, u_n+c_2/2, v_n+d_2/2)$$

$$a_4 = h f(x_n+h/2, y_n+a_3/2, z_n+b_3/2, u_n+c_3/2, v_n+d_3/2)$$

$$b_4 = h g(x_n+h/2, y_n+a_3/2, z_n+b_3/2, u_n+c_3/2, v_n+d_3/2)$$

$$c_4 = h k(x_n+h/2, y_n+a_3/2, z_n+b_3/2, u_n+c_3/2, v_n+d_3/2)$$

$$d_4 = h j(x_n+h/2, y_n+a_3/2, z_n+b_3/2, u_n+c_3/2, v_n+d_3/2)$$

For the case of super-saturated air the controlling equations for the accurate model are

$$dm_w = K_1 (w_{asw} - w_{as}) \tag{D.11}$$

$$dw_a = K_2(w_{asw} - w_{as}) \quad (D.12)$$

$$dT_p = K_3(T_p - T_w) \quad (D.13)$$

$$di_a = K_4(i_{asw} - i_a) + K_5(w_a - w_{as}) + K_6(w_{asw} - w_{as}) \quad (D.14)$$

where

$$K_1 = - \frac{h_c dA_o A_{ev}}{c_{pm}}$$

$$K_2 = \frac{h_c dA_o A_{ev}}{c_{pm} m_a}$$

$$K_3 = - \frac{U_o dA_o}{m_p c_{pp}}$$

$$K_4 = \frac{h_c dA_o}{c_{pm} m_a}$$

$$K_5 = K_4 c_{pw} T_w$$

$$K_6 = K_4 (A_{ev} - 1) i_v$$

The hypothetical case with the following values for the governing variables is considered:

$$\begin{aligned} dA_o &= 0.1776 \text{ m}^2 \\ m_a &= 0.2623 \text{ kg/s (8 kg/s)}^* \\ m_p &= 4.5902 \times 10^{-2} \text{ kg/s (7 kg/s)}^* \\ m_{wi} &= 1.2234 \times 10^{-3} \text{ kg/s (200 kg/m}^2\text{h)}^* \\ c_{pw} &= 4185.64 \text{ J/kgK} \\ c_{pp} &= 4176.76 \text{ J/kgK} \\ T_{po} &= 39.325 \text{ }^\circ\text{C} \\ T_{pi} &= 50.000 \text{ }^\circ\text{C} \\ T_{film} &= 29.382 \text{ }^\circ\text{C} \\ T_w &= 19.439 \text{ }^\circ\text{C} \end{aligned}$$

D5

$$\begin{aligned}
 i_{ai} &= 55805.5 \text{ J/kg} \quad (T_{wba} = 19.5 \text{ }^\circ\text{C}, T_{dba} = 25 \text{ }^\circ\text{C})^* \\
 h_c &= 78.20 \text{ W/m}^2\text{ }^\circ\text{C} \\
 c_{pm} &= 1048.74 \text{ J/kgK} \\
 A_{ev} &= 0.7 \\
 U_o &= 278.09 \\
 i_v &= 2536988 \text{ J/kg} \\
 x_{ai} &= 0.017845 \text{ kg/kg}
 \end{aligned}$$

The above approximated value for the film temperature is corrected by repeating the Runge-Kutta procedure with a progressively better approximation for the film temperature until this value is equal to the film temperature calculated with equation (3.2.2). The constants, K_1 to K_6 can now be calculated.

$$\begin{aligned}
 K_1 &= -9.26777 \times 10^{-3} \\
 K_2 &= 3.5333 \times 10^{-2} \\
 K_3 &= 2.5762 \times 10^{-1} \\
 K_4 &= 5.04762 \times 10^{-2} \\
 K_5 &= 4106.972 \\
 K_6 &= -38417.27
 \end{aligned}$$

The Runge-Kutta procedure to solve the differential equations can now begin. The procedure is repeated two times for each element to obtain the average values of the inlet and outlet temperatures and mass flows for the calculation of the values of these parameters at the next node. The first calculation uses the inlet value as the average value between the two nodes.

Step 1

$$\Delta w_{as} = w_{asw} - w_{as} = 0.0121177 \text{ kg/kg}$$

$$\Delta w_a = w_{aav} - w_{as} = 0.0037589 \text{ kg/kg}$$

$$\Delta i_a = i_{asw} - i_{aav} = 40703.67 \text{ J/kg}$$

$$\Delta T_p = T_{pav} - T_{film} = 9.94302 \text{ }^\circ\text{C}$$

$$a_1 = K_1 \Delta w_{as} = -1.122304 \times 10^{-4}$$

$$b_1 = K_2 \Delta w_{as} = 4.28158 \times 10^{-4}$$

$$c_1 = K_3 \Delta T_p = 2.56152$$

$$d_1 = K_4 \Delta i_a + K_5 \Delta w_a + K_6 \Delta w_{as} = 1604.477$$

Step 2

First the new average values for the enthalpy and humidity of the air and process water are calculated as

$$i_{aav} = (i_{ai} + d_1/2 + i_{ao})/2 = 56206.7$$

$$w_{aav} = (w_{ai} + b_1/2 + w_{ao})/2 = 0.017961$$

$$T_{pav} = (T_{po} + c_1/2 + T_{po})/2 = 39.965$$

With the assumption that the air is always saturated if spray water drops are present the wet-bulb and dry-bulb temperatures of the air are calculated. With this calculation it is also possible to ensure that if the air dries out during the calculation, the procedure starts over by solving the differential equations for dry air. The condition of the air is now known

$$T_{wba} = T_{dba} = 19.4735 \text{ } ^\circ\text{C}$$

$$\Delta w_{as} = w_{asw} - w_{as} = 0.0120009 \text{ kg/kg}$$

$$\Delta w_a = w_{aav} - w_{as} = 0.003749 \text{ kg/kg}$$

$$\Delta i_a = i_{asw} - i_{aav} = 40302.6 \text{ J/kg}$$

$$\Delta T_p = T_{pav} - T_{film} = 10.5834 \text{ } ^\circ\text{C}$$

$$a_2 = K_1 \Delta w_{as} = -1.112213 \times 10^{-4}$$

$$b_2 = K_2 \Delta w_{as} = 4.240311 \times 10^{-4}$$

$$c_2 = K_3 \Delta T_p = 2.726494$$

$$d_2 = K_4 \Delta i_a + K_5 \Delta w_a + K_6 \Delta w_{as} = 1588.68$$

Step 3

Again the new average values for the enthalpy and humidity of the air and process water are calculated as

$$i_{aav} = (i_{ai} + d_1/2 + i_{ao})/2 = 56202.7$$

$$w_{aav} = (w_{ai} + b_1/2 + w_{ao})/2 = 0.017960$$

$$T_{pav} = (T_{po} + c_1/2 + T_{po})/2 = 40.007$$

The condition of the air can be calculated as before as

$$T_{wba} = T_{dba} = 19.4723 \text{ } ^\circ\text{C}$$

$$\Delta w_{as} = w_{asw} - w_{as} = 0.012002 \text{ kg/kg}$$

$$\Delta w_a = w_{aav} - w_{as} = 0.003749 \text{ kg/kg}$$

$$\Delta i_a = i_{asw} - i_{aav} = 40306.5 \text{ J/kg}$$

$$\Delta T_p = T_{pav} - T_{film} = 10.6246 \text{ } ^\circ\text{C}$$

$$a_2 = K_1 \Delta w_{as} = -1.11232 \times 10^{-4}$$

$$b_2 = K_2 \Delta w_{as} = 4.240718 \times 10^{-4}$$

$$c_2 = K_3 \Delta T_p = 2.73712$$

D7

$$d_2 = K_4 \Delta i_a + K_5 \Delta w_a + K_6 \Delta w_{as} = 1588.83$$

Step 4

Again the new average values for the enthalpy and humidity of the air and process water are calculated as

$$i_{aav} = (i_{ai} + d_1/2 + i_{ao})/2 = 56599.95$$

$$w_{aav} = (w_{ai} + b_1/2 + w_{ao})/2 = 0.0180662$$

$$T_{pav} = (T_{po} + c_1/2 + T_{po})/2 = 40.694$$

The condition of the air can be calculated as before as

$$T_{wba} = T_{dba} = 19.6001 \text{ } ^\circ\text{C}$$

$$\Delta w_{as} = w_{asw} - w_{as} = 0.011886 \text{ kg/kg}$$

$$\Delta w_a = w_{aav} - w_{as} = 0.0037393 \text{ kg/kg}$$

$$\Delta i_a = i_{asw} - i_{aav} = 39909.3 \text{ J/kg}$$

$$\Delta T_p = T_{pav} - T_{film} = 11.3116 \text{ } ^\circ\text{C}$$

$$a_2 = K_1 \Delta w_{as} = -1.101569 \times 10^{-4}$$

$$b_2 = K_2 \Delta w_{as} = 4.199732 \times 10^{-4}$$

$$c_2 = K_3 \Delta T_p = 2.914087$$

$$d_2 = K_4 \Delta i_a + K_5 \Delta w_a + K_6 \Delta w_{as} = 1573.20$$

The outlet conditions of the element can now be calculated with the use of equations (D.7) to (D.10) as

$$m_{wo} = 0.00111297 \text{ kg/s}$$

$$w_{ao} = 0.0182782 \text{ kg/kg}$$

$$i_{ao} = 57394.3 \text{ J/kg}$$

$$T_{pi} = 42.0588 \text{ } ^\circ\text{C}$$

These values are used to verify the film temperature which is found to be $31.5943 \text{ } ^\circ\text{C}$ which indicates that the first approximation was too low. After several calculations the value for the film temperature of the first element is found to be $30.3776 \text{ } ^\circ\text{C}$. This value corresponds to the following outlet conditions

$$m_{wo} = 0.00109739 \text{ kg/s}$$

$$w_{ao} = 0.01833467 \text{ kg/kg}$$

$$i_{ao} = 57591.5 \text{ J/kg}$$

$$T_{pi} = 41.785 \text{ } ^\circ\text{C}$$

D8

The calculation is repeated a second time with these outlet conditions of the element to determine the average value for each parameter. The final value of the film temperature is 30.5532 °C and the outlet conditions of the element are

$$m_{w_o} = 0.00109721 \text{ kg/s}$$

$$w_{a_o} = 0.0183354 \text{ kg/kg}$$

$$i_{a_o} = 57593.0 \text{ J/kg}$$

$$T_{p_i} = 42.088 \text{ °C}$$

The calculation procedure of the outlet conditions of the second and successive elements are exactly the same and the inlet temperature of the heat exchanger is found to be 48.9456 °C which is lower than the prescribed 50.000 °C. The calculation is repeated from the first element with a successively better approximation of the outlet temperature of the process water temperature until the correct value is found. The final outlet temperature that corresponds to the inlet temperature of 50.000 °C is 40.0227 °C.

E1

APPENDIX E**DETAILED MEASUREMENTS OF THE EXPERIMENTAL HEAT EXCHANGER
MODEL**

The heat exchanger coil is 1.4 m long and 1.1 m wide and consists of four rows of staggered elliptical finned-tubes. The exact measurements of the heat exchanger tubes are shown in Figure E.1. The tubes are spaced at a pitch of 36.07 mm along the width and a pitch of 54.5 mm along the depth of the heat exchanger which gives it a the total depth of 0.218 m.

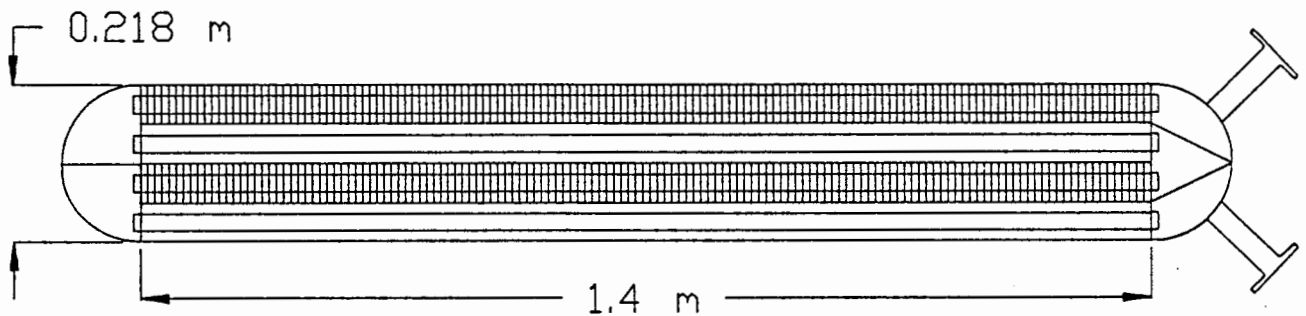


Figure E.1 - Dimensions of the experimental heat exchanger.

From the figure it can be seen that the hot process water enter the heat exchanger at the top tube row, and after four passes through the tubes leaves the heat exchanger at the bottom tube row on the same side that it entered.

The elliptical finned-tubes are made of steel and the type-L steel fins are tension wound on the tubes. The finned-tubes are galvanized to ensure good thermal contact between the fins and tube. The exact dimensions of the tube and fins are given in Figure E.2.

E2

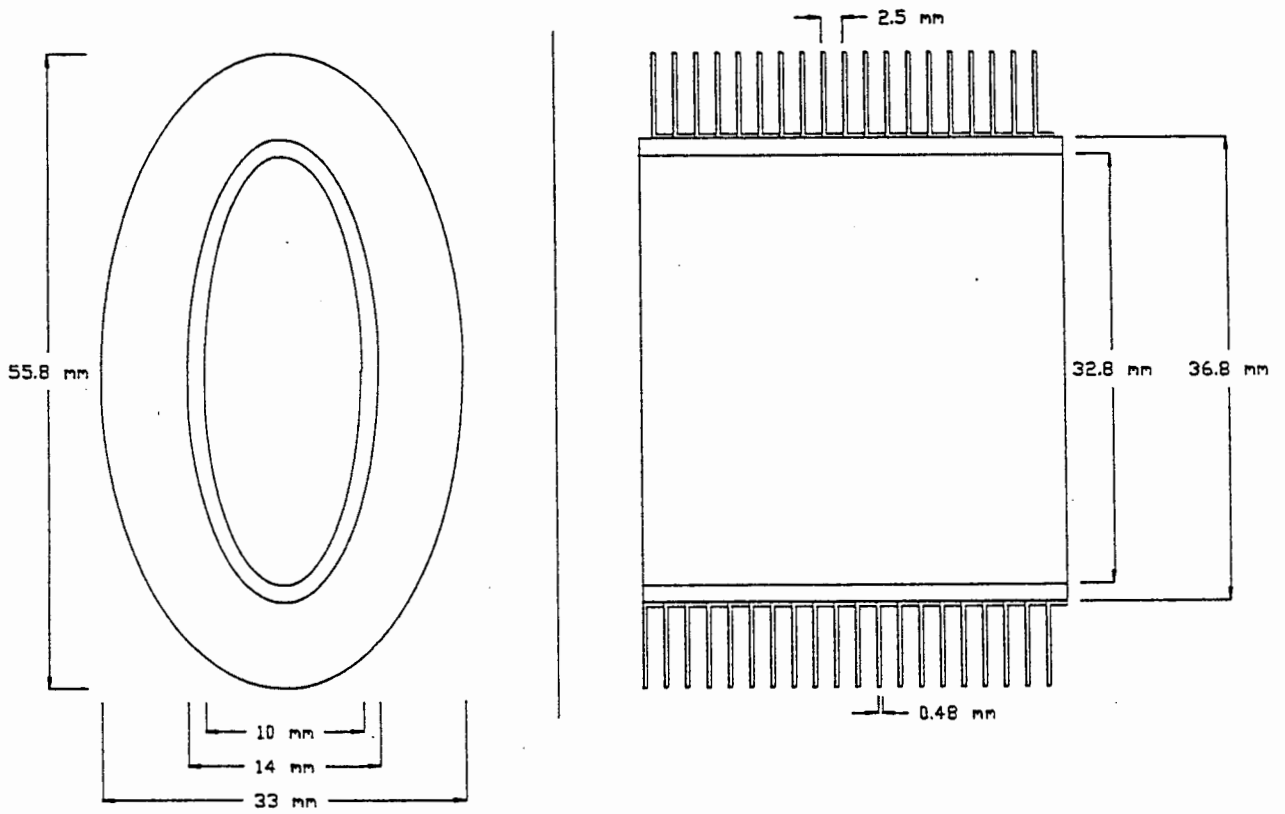


Figure E.2 - Dimensions of the elliptical fin tube.

F1

APPENDIX F**EXPERIMENTAL DATA OF SPRAY COOLED HEAT EXCHANGER**Table F.1: Test 1408 ($p_a = 100.623$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
104.5	287.766	14.787	3.477	19.552	13.381	38.131	27.285	11.207	44.294	40.913
160.0	287.766	15.121	3.885	20.392	13.371	37.603	28.039	11.233	44.055	40.042
190.0	287.766	15.211	4.362	20.733	13.805	36.618	28.186	11.210	43.904	39.485
204.0	287.766	15.315	4.918	22.312	13.814	35.536	27.267	11.195	43.505	38.880
238.0	287.766	15.412	5.548	23.023	14.108	34.233	27.189	11.216	43.303	38.174
270.0	287.766	15.418	6.149	23.653	14.241	32.769	26.999	11.198	43.111	37.502
316.0	287.766	15.442	6.880	24.576	14.569	31.698	26.286	11.192	42.805	36.921
346.0	287.766	15.549	7.602	25.110	14.761	30.966	25.526	11.207	42.369	36.356
373.0	287.766	15.642	8.239	26.055	14.970	30.504	24.971	11.178	42.086	35.996
402.0	287.766	15.585	9.111	26.650	15.147	30.240	24.029	11.182	41.638	35.592
420.0	287.766	15.622	9.599	26.954	15.592	29.901	23.826	11.202	41.110	35.160
66.0	251.532	16.013	3.414	25.683	16.937	41.092	27.782	7.175	46.783	42.557
110.0	251.532	16.203	3.893	25.120	16.610	37.559	27.584	7.194	44.003	39.147
197.0	251.532	16.113	4.733	26.033	16.923	38.080	28.264	7.203	46.189	39.918
222.0	251.532	16.293	5.375	26.245	16.974	36.818	28.022	7.196	46.020	39.075
253.0	251.532	16.314	6.039	26.421	16.774	35.269	27.458	7.182	45.814	38.151
306.0	251.532	16.334	6.827	26.622	16.788	33.666	26.897	7.194	45.595	37.235
336.0	251.532	16.364	7.610	26.905	16.874	32.941	26.097	7.180	45.361	36.737
368.0	251.532	16.323	8.323	27.297	17.102	32.465	25.464	7.189	44.922	36.321
405.0	251.532	16.240	9.237	27.602	17.346	31.931	24.863	7.197	44.548	35.941
446.0	251.532	16.173	10.207	27.333	17.456	30.960	24.262	7.168	43.756	35.095

Table F.2: Test 1607 ($p_a = 100.397$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
464.0	481.792	14.370	7.122	15.365	12.509	27.363	27.041	3.913	55.157	32.867
287.0	481.792	14.289	5.579	14.939	12.292	29.344	28.677	3.989	54.433	35.348
170.0	481.792	14.199	4.492	14.577	12.245	38.300	27.740	3.963	53.489	39.321
100.0	481.792	14.169	3.397	14.436	12.373	41.092	27.894	3.974	52.744	41.477
100.0	481.792	14.166	9.642	15.609	12.727	24.234	23.770	3.993	52.136	30.864
100.0	481.792	14.149	11.975	16.381	12.994	22.836	22.312	3.959	51.803	30.089

F2

Table F.3: Test 1707 ($P_a = 100.064$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
56.0	398.104	14.553	3.053	13.763	11.210	43.592	28.311	8.908	50.558	45.929
135.0	398.104	14.530	3.531	13.769	11.249	41.459	28.888	8.924	50.196	44.359
196.0	398.104	14.476	4.647	13.799	11.203	38.551	28.339	8.904	49.644	42.522
241.0	398.104	14.467	5.392	13.954	11.145	36.163	28.016	8.899	48.968	40.874
300.0	398.104	14.480	6.210	14.254	11.185	32.836	27.840	8.889	48.231	38.835
358.0	398.104	14.450	6.805	14.579	11.425	30.498	27.395	8.912	47.420	37.401
433.0	398.104	14.436	7.614	14.758	11.402	27.662	26.532	8.903	46.287	35.764
484.0	398.104	14.400	8.338	14.813	11.791	26.440	25.501	8.893	45.226	34.756
525.0	398.104	14.346	9.370	14.994	12.016	25.245	24.195	8.891	44.063	33.711
74.5	251.532	14.987	3.382	13.621	11.516	33.361	23.747	3.775	42.700	34.691
155.0	251.532	15.034	3.880	13.814	11.417	32.629	24.190	3.789	42.263	33.041
195.0	251.532	15.051	4.765	14.015	11.434	30.656	22.885	3.794	41.947	31.700
234.0	251.532	15.014	5.480	14.104	11.546	28.927	22.568	3.804	41.505	30.358
280.0	251.532	15.011	6.008	14.175	11.556	26.603	22.615	3.789	41.050	28.718
335.5	251.532	14.967	6.503	14.256	11.627	24.216	22.415	3.777	40.527	27.351
431.5	251.532	14.961	7.371	14.439	12.067	22.242	21.948	3.752	40.150	26.343
482.0	251.532	14.911	8.116	14.365	12.146	21.333	21.106	3.774	39.792	25.846
521.0	251.532	14.891	9.284	14.666	12.155	20.610	20.276	3.777	39.496	25.378

F3

Table F.4: Test 1807 ($p_a = 100.610$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
73.5	0.000	10.284	3.674	11.728	9.798	53.705	23.854	4.114	65.708	56.840
103.0	0.000	10.385	4.873	12.129	9.887	50.653	23.076	4.094	65.619	54.969
145.0	0.000	10.545	6.094	12.680	10.107	47.090	22.001	4.089	65.166	53.227
177.0	0.000	10.762	6.978	13.031	10.245	44.761	21.386	4.113	64.468	51.885
217.5	0.000	10.909	7.992	13.569	10.695	42.622	20.804	4.098	63.858	50.578
260.0	0.000	11.086	8.973	13.942	10.762	40.621	20.226	4.099	63.046	49.309
302.0	0.000	11.287	9.904	14.371	10.906	39.338	19.792	4.081	62.670	48.456
56.5	96.078	15.786	3.225	13.295	10.333	49.985	27.723	4.090	61.921	51.251
85.0	96.078	16.998	3.979	13.370	10.490	48.206	26.141	4.108	61.590	50.237
122.0	96.078	17.075	4.610	13.357	10.384	46.158	25.370	4.096	61.301	48.714
157.0	96.078	17.048	5.440	13.184	10.616	43.547	24.789	4.109	60.940	47.223
180.0	96.078	17.122	6.118	13.014	10.761	41.935	24.221	4.119	60.651	46.143
202.5	96.078	17.038	6.654	13.195	10.791	40.831	23.845	4.123	60.573	45.347
237.0	96.078	16.941	7.454	13.365	10.852	38.907	23.537	4.124	60.421	44.145
272.0	96.078	17.025	8.410	13.707	10.994	37.502	22.765	4.121	60.175	43.448
300.0	96.078	17.139	9.219	14.132	11.080	36.318	22.015	4.125	59.894	42.932
338.0	96.078	17.195	10.145	14.604	11.092	35.390	21.287	4.137	59.497	42.368

F4

Table F.5: Test 1708 ($p_a = 101.180$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
390.0	148.909	15.706	11.128	14.626	11.520	34.387	22.906	5.925	57.978	43.582
360.0	148.909	16.133	10.331	14.850	11.672	34.885	23.309	5.925	57.244	43.390
333.0	148.909	16.223	9.688	14.889	11.636	35.344	23.700	5.913	56.798	43.294
307.0	148.909	16.440	8.940	15.648	12.239	35.541	24.384	5.941	55.087	42.487
274.0	148.909	16.504	8.263	15.661	12.148	36.247	24.829	5.980	54.814	42.721
244.0	148.909	16.584	7.636	15.561	11.984	36.502	24.997	5.963	54.301	42.591
210.0	148.909	16.641	6.962	15.680	11.988	37.245	25.574	5.979	53.977	42.780
178.0	148.909	16.711	6.398	15.442	11.790	38.161	25.662	5.958	53.767	43.213
147.0	148.909	16.761	5.828	15.516	11.857	39.317	25.798	5.973	53.571	43.888
125.0	148.909	16.811	5.152	15.234	11.741	40.514	26.219	5.968	53.305	44.458
98.0	148.909	16.858	4.466	15.304	11.735	42.127	26.697	5.980	53.070	45.130
78.0	148.909	16.905	3.748	15.191	11.793	43.345	27.920	5.989	52.898	45.678
57.0	127.403	17.793	3.372	16.816	12.247	49.095	28.847	7.297	57.369	51.520
90.0	127.403	17.162	3.983	17.272	12.483	47.051	29.039	7.305	56.811	50.095
140.0	127.403	17.132	4.516	17.208	12.467	45.821	28.300	7.300	56.591	49.210
165.0	127.403	17.195	5.117	16.848	12.186	44.120	26.790	7.285	56.126	48.508
188.0	127.403	17.252	5.846	17.047	12.271	42.617	26.367	7.298	56.016	47.704
209.0	127.403	17.285	6.415	17.233	12.381	41.473	26.054	7.310	55.869	47.127
238.0	127.403	17.229	7.235	17.346	12.318	39.790	25.389	7.297	55.623	46.297
267.0	127.403	17.155	8.088	17.375	12.309	38.479	24.610	7.292	55.424	45.845
278.0	127.403	17.195	8.791	18.020	12.581	37.771	23.842	7.284	55.080	45.638
320.0	127.403	17.242	9.678	18.292	12.763	36.793	23.282	7.301	54.801	45.132
345.0	127.403	17.262	10.264	18.527	12.886	36.213	22.940	7.290	54.443	44.751
101.0	106.364	17.806	3.804	17.451	12.544	46.271	28.246	10.224	54.221	49.876
136.0	106.364	17.843	4.619	17.526	12.553	44.811	27.373	10.200	54.079	49.108
156.0	106.364	17.880	5.223	17.605	12.599	43.786	26.535	10.216	53.853	48.737
182.0	106.364	17.907	6.030	17.735	12.674	42.095	25.453	10.253	53.526	48.094
208.0	106.364	17.900	6.795	17.783	12.687	40.719	25.094	10.236	53.340	47.459
231.0	106.364	17.873	7.473	17.732	12.721	39.538	24.624	10.217	53.132	46.952
264.0	106.364	17.880	8.345	17.758	12.613	38.056	23.571	10.223	52.675	46.459
289.0	106.364	17.820	9.182	17.754	12.544	37.047	22.956	10.248	52.467	45.996
316.0	106.364	17.806	9.905	17.806	12.525	36.338	22.471	10.255	52.316	45.698
360.0	106.364	17.803	10.805	18.044	12.569	35.541	22.008	10.229	52.082	45.346

F5

Table F.6: Test 1907 ($p_a = 101.073$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
73.0	312.779	14.383	3.457	15.322	11.277	51.816	30.427	7.308	63.024	55.598
157.5	312.779	14.784	4.199	14.971	11.283	48.782	31.114	7.408	62.202	52.940
184.0	312.779	14.887	4.747	15.929	11.684	47.943	29.575	7.329	61.749	51.878
194.0	312.779	14.880	5.131	16.516	11.973	46.119	28.960	7.398	60.608	50.392
226.0	312.779	14.841	5.802	16.356	11.930	43.683	28.683	7.394	60.192	48.711
262.0	312.779	14.827	6.417	16.531	11.851	41.842	28.256	7.384	59.647	47.265
250.0	312.779	14.830	7.136	16.280	11.544	39.620	27.510	7.415	59.039	45.832
278.0	312.779	14.820	7.801	16.284	11.319	37.937	26.744	7.410	58.335	44.682
310.0	312.779	14.820	8.529	16.264	10.990	36.178	26.238	7.407	57.837	43.799
363.0	312.779	14.790	9.589	16.951	11.556	34.018	25.077	7.411	56.646	42.577
372.5	312.779	14.790	10.187	17.219	11.736	33.286	24.782	7.416	55.951	41.780
73.0	251.532	15.071	3.499	14.771	10.944	42.337	26.631	4.249	54.335	44.386
133.5	251.532	15.071	3.631	14.629	10.782	42.652	27.052	4.245	54.171	42.962
160.0	251.532	15.074	4.253	14.650	10.839	41.525	26.150	4.246	53.957	42.230
200.0	251.532	15.101	5.134	14.679	10.612	38.802	25.157	4.243	53.758	40.245
235.0	251.532	15.101	5.849	14.626	10.520	36.687	24.634	4.236	53.605	38.561
278.0	251.532	15.084	6.570	14.784	10.455	33.999	24.180	4.243	53.372	36.690
316.0	251.532	15.061	7.138	14.734	10.477	32.229	23.633	4.243	53.098	35.717
373.0	251.532	15.041	8.077	14.803	10.565	29.980	23.046	4.225	52.706	34.336
429.0	251.532	15.031	9.100	14.999	10.437	28.538	22.148	4.251	52.343	33.419

F6

Table F.7: Test 2109 ($p_a = 100.568$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
107.0	88.597	19.687	4.359	18.282	14.240	46.601	27.451	5.274	58.380	49.740
147.0	88.597	19.948	5.001	18.766	14.624	45.093	27.082	5.252	58.056	48.624
178.0	88.597	20.238	5.851	19.184	14.662	43.352	26.524	5.318	57.837	47.688
198.0	88.597	20.392	6.501	19.642	14.941	42.102	25.975	5.307	57.652	47.141
226.0	88.597	20.465	7.279	20.211	14.941	40.803	25.463	5.342	57.373	46.263
253.0	88.597	20.569	8.140	20.986	15.376	39.597	24.914	5.318	57.069	45.911
292.0	88.597	20.703	9.124	21.613	15.339	38.778	24.178	5.318	56.739	45.409
319.0	88.597	20.773	9.771	22.333	15.460	38.301	23.786	5.314	56.467	45.200
84.0	44.416	19.593	4.147	25.332	17.052	48.927	26.941	5.665	57.084	51.504
114.0	44.416	19.390	4.930	25.186	16.973	47.490	26.206	5.687	56.983	50.947
152.0	44.416	19.059	5.803	24.950	17.087	45.976	25.602	5.650	56.923	50.195
183.0	44.416	18.842	6.613	24.885	17.082	44.710	25.433	5.681	56.833	49.541
215.0	44.416	18.678	7.418	24.776	17.069	43.355	24.680	5.659	56.741	48.926
245.0	44.416	18.558	8.180	24.849	17.099	42.473	24.535	5.663	56.636	48.415
295.0	44.416	18.491	9.351	25.040	17.179	41.303	24.140	5.658	56.522	47.957

F7

Table F.8: Test 1908 ($p_a = 101.508$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
82.0	193.325	15.104	3.824	12.396	9.260	37.509	26.580	10.538	45.348	41.065
142.0	193.325	14.787	4.339	12.558	9.273	36.548	25.477	10.556	44.681	40.115
185.0	193.325	15.238	4.891	12.629	9.284	35.201	25.137	10.521	44.566	39.500
185.0	193.325	15.301	5.580	12.709	9.254	33.688	24.838	10.540	44.396	38.741
234.0	193.325	15.291	6.062	12.788	9.287	32.712	24.609	10.543	44.222	38.208
264.0	193.325	15.291	6.687	13.075	9.410	31.477	24.266	10.543	44.013	37.573
296.0	193.325	15.311	7.346	13.264	9.506	30.333	23.733	10.558	43.742	36.976
328.0	193.325	15.291	7.997	13.634	9.597	29.390	23.357	10.573	43.553	36.580
360.0	193.325	15.291	8.737	13.871	9.640	28.607	22.606	10.572	43.358	36.294
383.0	193.325	15.291	9.431	14.180	9.828	28.080	21.934	10.580	43.055	35.998
419.0	193.325	15.308	10.339	14.664	9.989	27.383	21.145	10.568	42.796	35.634
78.0	159.896	15.388	3.517	12.731	9.284	38.224	24.921	8.357	45.782	41.294
111.0	159.896	15.682	3.829	13.149	9.231	36.955	26.413	8.414	45.211	39.826
167.0	159.896	15.659	4.550	13.276	9.166	35.799	24.990	8.403	45.092	39.276
189.0	159.896	15.692	5.193	13.480	9.354	34.769	24.141	8.394	44.959	38.806
221.0	159.896	15.672	5.977	13.637	9.313	33.044	23.735	8.406	44.793	37.890
247.0	159.896	15.679	6.597	13.840	9.497	31.972	23.433	8.401	44.686	37.307
276.0	159.896	15.692	7.278	13.969	9.486	30.863	22.966	8.403	44.556	36.742
311.0	159.896	15.722	8.229	14.260	9.623	29.849	22.013	8.388	44.279	36.289
336.0	159.896	15.732	9.030	14.555	9.758	29.129	21.269	8.405	44.001	36.005
368.0	159.896	15.776	9.821	14.789	10.037	28.458	20.709	8.392	43.781	35.704
427.0	159.896	15.772	11.073	15.065	10.193	27.521	19.914	8.403	43.527	35.259

F8

Table F.9: Test 2509 ($p_a = 100.575$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
77.0	61.948	19.450	3.972	20.714	16.109	43.002	25.870	8.021	49.907	46.103
110.0	61.948	19.052	4.682	20.158	15.915	41.437	25.170	8.007	49.836	45.616
148.0	61.948	18.859	5.321	19.938	15.757	40.546	25.250	7.987	49.764	44.959
172.0	61.948	18.682	5.967	19.825	15.759	39.407	24.929	7.987	49.739	44.465
205.0	61.948	18.598	6.895	19.902	15.691	38.132	24.547	7.981	49.624	43.851
242.0	61.948	18.491	7.976	19.836	15.776	36.430	23.903	7.979	49.303	43.264
274.0	61.948	18.401	8.960	19.558	15.604	35.518	23.558	7.976	48.927	42.689
67.0	35.766	18.338	3.731	18.151	14.988	41.489	24.952	5.197	48.566	43.340
95.0	35.766	18.485	4.646	17.432	14.633	38.895	23.600	5.253	48.590	42.724
134.0	35.766	18.488	5.473	17.481	14.650	37.982	23.369	5.238	48.587	41.989
166.0	35.766	18.458	6.364	17.402	14.590	36.629	23.169	5.259	48.653	41.294
193.0	35.766	18.451	7.128	17.457	14.632	35.447	22.719	5.251	48.560	40.914
235.0	35.766	18.465	8.282	17.946	14.817	34.511	22.604	5.232	48.530	40.406

Table F.10: Test 2007 ($p_a = 101.529$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
60.2	16.234	15.402	3.213	15.467	10.608	50.231	30.358	10.417	58.292	53.645
116.0	216.234	15.321	3.765	15.524	10.720	47.775	30.759	10.415	57.441	51.838
158.0	216.234	15.332	4.574	15.567	10.600	46.033	29.567	10.412	57.082	50.768
181.0	216.234	15.368	5.232	15.706	10.581	44.172	28.279	10.426	56.330	49.697
201.0	216.234	15.341	5.650	15.701	10.744	42.845	28.184	10.427	56.135	49.015
223.0	216.234	15.382	6.356	15.966	10.869	40.821	27.296	10.427	55.546	48.000
248.0	216.234	15.341	7.013	16.188	11.035	39.708	25.950	10.453	54.859	47.517
267.0	216.234	15.625	7.755	16.314	11.001	37.929	25.608	10.454	54.335	46.308
305.0	216.234	15.245	8.550	16.481	11.064	36.550	24.805	10.439	53.666	45.416
353.0	216.234	15.271	9.590	16.716	11.242	35.247	23.964	10.451	53.272	44.819
64.5	170.883	15.852	3.427	15.050	10.924	44.603	27.349	8.198	52.350	47.266
109.0	170.883	16.113	3.853	15.023	10.964	43.133	27.592	8.199	52.107	46.270
145.0	170.883	15.916	4.301	14.881	11.062	41.839	27.027	8.162	51.507	45.316
170.0	170.883	16.006	5.047	14.841	10.928	40.400	25.718	8.162	51.300	44.642
202.0	170.883	15.899	5.823	14.934	11.024	38.388	25.715	8.171	51.105	43.504
233.0	170.883	15.736	6.702	14.988	11.112	36.354	24.828	8.178	50.801	42.494
258.0	170.883	15.732	7.372	15.082	11.255	35.103	24.275	8.162	50.493	41.792
285.0	170.883	15.709	8.052	15.117	11.239	34.096	23.486	8.257	50.082	41.279
334.0	170.883	15.702	9.158	15.242	11.312	32.736	22.715	8.231	49.749	40.638
367.0	170.883	15.689	9.951	15.364	11.368	31.714	22.099	8.236	49.240	39.998

F9

Table F.11: Test 1009 ($p_a = 100.024$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
107.0	200.104	17.596	3.320	16.910	9.676	40.635	26.898	7.831	48.429	42.940
143.0	200.104	17.596	3.734	17.059	9.274	39.483	26.881	7.844	48.062	41.785
170.0	200.104	17.566	4.448	17.154	8.895	38.179	24.989	7.836	47.916	41.254
185.0	200.104	17.576	4.955	17.222	8.877	37.335	24.162	7.830	47.788	40.859
207.0	200.104	17.566	5.564	17.249	9.105	35.815	23.953	7.844	47.672	40.112
231.0	200.104	17.623	6.133	17.319	9.075	34.457	23.526	7.832	47.500	39.350
248.0	200.104	17.716	6.764	17.525	9.239	33.393	22.822	7.874	47.313	38.858
292.0	200.104	17.746	7.639	17.668	9.371	32.128	22.047	7.872	47.002	38.074
333.0	200.104	17.713	8.558	17.888	9.357	31.221	21.120	7.895	46.815	37.626
360.0	200.104	17.696	9.386	18.092	9.600	30.722	20.432	7.910	46.635	37.424

Table F.12: Test 2108 ($p_a = 100.668$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
62.0	350.883	14.667	3.423	17.022	10.086	43.193	27.285	9.739	50.792	46.327
142.0	350.883	14.917	3.718	17.263	10.188	42.255	28.962	9.786	50.102	44.839
180.0	350.883	15.037	4.401	17.163	10.173	40.557	28.049	9.777	49.871	43.952
207.0	350.883	15.034	5.064	17.066	10.378	38.728	27.934	9.789	49.646	42.917
248.0	350.883	15.014	5.795	17.185	10.745	36.694	28.038	9.762	49.439	41.773
291.0	350.883	15.037	6.388	17.724	10.891	34.444	28.130	9.799	49.208	40.688
337.0	350.883	14.961	7.008	18.233	11.100	32.759	27.334	9.779	48.728	39.767
384.0	350.883	14.961	7.712	18.521	11.145	30.992	26.931	9.825	48.414	38.977
419.0	350.883	15.001	8.406	19.045	11.303	29.701	26.046	9.805	48.075	38.464
466.0	350.883	15.008	9.282	19.417	11.629	28.860	24.951	9.766	47.728	37.894
504.0	350.883	15.042	10.195	20.057	11.945	27.898	24.302	9.781	47.540	37.558
87.0	300.156	15.351	3.517	18.338	11.190	39.402	27.053	7.632	46.755	41.479
148.0	300.156	15.358	3.886	18.462	11.266	38.867	27.140	7.619	46.581	40.734
177.0	300.156	15.265	4.431	18.584	11.318	37.825	26.528	7.614	46.503	40.164
203.0	300.156	15.265	5.046	18.777	11.388	36.449	26.332	7.626	46.417	39.356
226.0	300.156	15.315	5.524	19.088	11.462	35.279	26.310	7.631	46.260	38.627
256.0	300.156	15.241	6.015	19.254	11.656	33.647	26.373	7.618	45.971	37.635
293.0	300.156	15.201	6.516	19.712	11.922	32.088	26.430	7.603	45.734	36.695
354.0	300.156	15.191	7.320	20.099	12.138	30.189	25.824	7.613	45.478	35.742
409.0	300.156	15.201	8.178	20.520	12.188	28.615	24.982	7.622	45.266	35.119
474.0	300.156	15.181	9.384	21.135	12.261	27.662	23.418	7.603	45.004	34.641

F10

Table F.13: Test 0409 ($p_a = 100.537$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
75.0	300.156	16.898	3.314	24.770	15.794	42.785	29.309	7.047	49.820	44.329
143.0	300.156	17.239	3.717	25.003	15.624	42.248	29.308	7.053	49.549	43.483
176.0	300.156	17.369	4.322	25.181	15.592	41.102	28.788	7.041	49.442	42.813
201.0	300.156	17.336	4.908	25.302	15.637	39.589	28.789	7.045	49.340	41.877
223.0	300.156	17.305	5.488	25.599	15.753	38.194	28.616	7.150	49.026	40.964
260.0	300.156	17.262	6.025	25.675	15.622	36.321	28.513	7.133	48.690	39.788
307.0	300.156	17.205	6.686	25.703	15.690	34.464	28.208	7.144	48.476	38.778
357.0	300.156	17.195	7.426	25.858	15.772	33.095	27.594	7.150	48.349	38.074
395.0	300.156	17.175	8.118	26.047	15.811	32.066	26.936	7.147	48.207	37.623
446.0	300.156	17.148	8.989	26.313	15.909	31.100	26.183	7.152	48.080	37.202
105.0	350.883	16.754	3.287	23.269	15.160	37.021	28.061	4.750	45.316	37.598
151.0	350.883	16.814	3.627	23.146	14.993	37.086	27.631	4.737	45.156	37.205
185.0	350.883	16.805	4.189	23.048	15.007	36.170	27.048	4.753	45.101	36.550
210.0	350.883	16.731	4.759	23.068	14.906	35.018	26.535	4.742	45.049	35.756
237.0	350.883	16.751	5.247	23.204	14.911	33.832	26.319	4.720	44.985	34.957
267.0	350.883	16.785	5.634	23.104	14.907	31.880	26.650	4.737	44.917	33.801
312.0	350.883	16.758	6.066	23.309	14.997	29.755	26.682	4.730	44.847	32.767
390.0	350.883	16.741	6.774	23.322	15.029	27.378	26.231	4.732	44.778	31.669
461.0	350.883	16.718	7.522	23.561	15.132	26.348	25.613	4.756	44.673	31.038
501.0	350.883	16.684	8.265	23.623	15.143	25.503	24.866	4.786	44.578	30.724

F11

Table F.14: Test 0909 ($p_a = 100.409$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
64.0	350.883	16.694	3.030	21.989	15.484	53.384	33.873	9.685	61.477	56.397
123.0	350.883	16.901	3.390	22.000	15.412	51.879	35.557	9.711	60.989	54.573
165.0	350.883	16.915	3.981	21.643	14.944	49.932	32.864	9.680	60.290	53.612
188.0	350.883	16.935	4.537	21.664	14.799	48.452	32.652	9.689	60.050	52.619
218.0	350.883	16.871	5.006	21.570	14.591	46.665	32.611	9.688	59.874	51.703
226.0	350.883	16.835	5.469	21.510	14.443	45.005	32.356	9.690	59.533	50.740
251.0	350.883	16.841	5.954	21.441	14.334	43.457	32.055	9.707	59.311	49.876
285.0	350.883	16.805	6.531	21.399	14.252	41.437	31.583	9.703	58.923	48.789
318.0	350.883	16.744	7.131	21.543	14.364	39.973	31.025	9.696	58.661	48.104
361.0	350.883	16.744	7.943	21.657	14.361	38.365	30.140	9.702	58.160	47.201
404.0	350.883	16.744	8.819	21.924	14.473	37.042	29.086	9.695	57.814	46.509
102.0	275.610	17.072	3.451	19.006	13.319	47.646	31.638	7.155	57.782	50.230
143.0	275.610	16.965	3.779	18.796	13.123	46.563	31.189	7.166	57.368	49.043
185.0	275.610	16.941	4.585	18.809	13.158	44.419	30.032	7.138	57.159	47.898
202.0	275.610	16.925	5.024	18.805	13.207	42.983	29.689	7.154	56.960	47.076
220.0	275.610	16.925	5.477	18.743	13.226	41.561	29.463	7.150	56.764	46.280
242.0	275.610	16.925	5.921	18.705	13.057	40.210	29.036	7.175	56.502	45.389
265.0	275.610	16.975	6.375	18.743	13.046	38.908	28.696	7.144	56.245	44.536
292.0	275.610	16.975	6.879	18.784	13.090	37.778	28.289	7.147	56.076	43.830
333.0	275.610	16.975	7.621	18.885	13.088	36.429	27.436	7.152	55.840	43.196
360.0	275.610	16.965	8.351	19.056	13.189	35.383	26.642	7.170	55.611	42.674
400.0	275.610	16.918	9.227	19.210	13.319	34.393	25.846	7.167	55.289	42.208

F12

Table F.15: Test 1709 ($p_a = 100.914$ kPa)

dP_a	M_w	T_w	m_a	T_{dbai}	T_{wbai}	T_{dbao}	T_{wbao}	m_p	T_{pi}	T_{po}
106.0	237.273	15.268	3.596	12.679	9.562	50.386	31.086	9.140	61.754	54.844
146.0	237.273	16.137	4.032	12.940	9.751	49.413	31.120	9.146	61.423	53.792
170.0	237.273	16.120	4.723	13.226	9.909	47.262	29.990	9.134	60.868	52.729
192.0	237.273	16.160	5.252	13.309	9.677	45.312	29.316	9.127	60.396	51.636
206.0	237.273	16.223	5.673	13.822	9.869	43.976	28.892	9.121	59.991	50.860
220.0	237.273	16.377	6.071	14.457	10.344	42.789	28.544	9.112	59.582	50.192
236.0	237.273	16.507	6.518	14.865	10.416	41.501	28.026	9.120	59.069	49.356
261.0	237.273	16.634	7.153	15.454	10.764	40.196	27.401	9.142	58.708	48.635
300.0	237.273	16.701	8.080	15.715	10.814	38.519	26.299	9.159	58.223	47.836
331.0	237.273	16.591	8.841	16.001	10.995	37.495	25.490	9.195	57.690	47.194
374.0	237.273	16.584	9.787	16.164	11.339	36.262	24.804	9.162	57.292	46.574
105.0	325.403	15.545	3.392	14.058	10.261	45.601	29.394	7.300	55.302	48.154
146.0	325.403	15.515	3.715	13.733	10.071	44.805	29.556	7.277	54.926	47.026
179.0	325.403	15.525	4.451	13.526	9.952	42.500	28.511	7.264	54.729	45.906
206.0	325.403	15.612	5.058	13.395	9.826	40.325	28.312	7.310	54.535	44.725
239.0	325.403	15.645	5.683	13.457	9.840	38.169	28.040	7.302	54.163	43.340
262.0	325.403	15.592	6.040	13.432	9.883	36.751	27.816	7.308	53.940	42.489
296.0	325.403	15.612	6.563	13.557	9.870	34.915	27.357	7.323	53.626	41.391
343.0	325.403	15.642	7.285	13.697	9.884	32.921	26.586	7.319	53.280	40.337
381.0	325.403	15.619	8.002	13.690	9.673	31.113	25.656	7.289	52.839	39.433
422.0	325.403	15.589	8.755	13.723	9.705	29.461	24.709	7.297	52.408	38.614
458.0	325.403	15.552	9.556	13.893	9.751	28.358	23.824	7.301	52.084	38.079

G1

APPENDIX G**PERFORMANCE OF ANALYTICAL MODELS COMPARED TO EXPERIMENTAL DATA**

Table G.1 (Test 1408)

M_w	v_{max}	ϕ_a	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
287.766	3.5935	67.508	44.294	158.27	157.93	-0.2193	149.25	-5.697
287.766	4.0267	64.043	44.055	188.29	164.44	-12.665	155.97	-17.162
287.766	4.5257	64.506	43.904	206.92	180.03	-12.996	222.34	7.450
287.766	5.1307	58.598	43.505	216.27	199.43	-7.787	213.47	-1.292
287.766	5.8022	57.206	43.303	240.25	206.84	-13.907	207.7	-13.530
287.766	6.4442	55.549	43.111	262.31	266.12	1.453	296.82	13.155
287.766	7.2323	53.684	42.805	275.07	270.48	-1.667	303.27	10.253
287.766	8.0058	52.647	42.369	281.45	325.88	15.785	307.29	9.179
287.766	8.7043	50.450	42.086	284.32	321.05	12.921	286.96	0.929
287.766	9.6441	49.271	41.638	282.39	315.51	11.727	290.26	2.787
287.766	10.1717	49.801	41.11	278.39	309.76	11.268	285.59	2.587
251.532	3.6027	58.487	46.783	126.66	143.68	13.433	137.4	8.472
251.532	4.1005	59.231	44.003	145.89	138.36	-5.162	133.27	-8.651
251.532	5.0005	57.233	46.189	188.68	188.68	0.002	185.19	-1.844
251.532	5.6873	56.704	46.02	208.72	196.92	-5.650	192.17	-7.925
251.532	6.3932	55.410	45.814	229.88	255.09	10.96	276.99	20.494
251.532	7.2319	54.809	45.595	251.23	261.68	4.158	265.07	5.508
251.532	8.0688	54.198	45.361	258.66	290.96	12.48	271.15	4.830
251.532	8.8372	53.737	44.922	258.28	290.96	12.282	272.39	5.463
251.532	9.8169	53.608	44.548	258.72	284.98	10.147	272.48	5.316
251.532	10.8316	54.840	43.756	259.27	276.66	6.706	269.72	4.027

Table G.2 (Test 1607)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
481.792	7.271	83.071	55.157	364.32	323.37	-11.247	367.41	0.847
481.792	5.687	84.173	54.433	318.06	251.9	-20.800	277.74	-12.675
481.792	4.573	85.890	53.489	234.57	218.38	-6.904	253.91	8.245
481.792	3.456	87.412	52.744	187.05	195.8	4.677	192.41	2.868
481.792	9.853	82.963	52.136	354.72	368.82	3.974	360.71	1.689
481.792	12.260	80.358	51.803	359.09	375.36	4.532	367.61	2.373

G2

Table G.3 (Test 1707)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
398.104	3.110	84.561	50.558	172.29	197.15	14.424	202.91	17.771
398.104	3.597	84.749	50.196	217.60	207.43	-4.676	213.82	-1.740
398.104	4.734	84.323	49.644	264.91	234.36	-11.531	288.73	8.993
398.104	5.496	83.159	48.968	300.87	258.61	-14.045	297.65	-1.071
398.104	6.337	81.770	48.231	348.88	330.03	-5.402	390.34	11.884
398.104	6.952	81.351	47.420	373.00	328.85	-11.834	389.77	4.498
398.104	7.783	80.300	46.287	391.30	402.83	2.945	387.93	-0.863
398.104	8.525	82.096	45.226	388.88	395.67	1.747	381.41	-1.921
398.104	9.586	82.360	44.063	384.41	390.36	1.549	376.87	-1.960
251.532	3.443	87.089	42.700	126.29	135.6	7.370	123.56	-2.161
251.532	3.953	85.443	42.263	145.95	142.78	-2.173	129.67	-11.150
251.532	4.858	84.430	41.947	162.39	152.36	-6.175	165.08	1.655
251.532	5.589	84.571	41.505	177.10	164.84	-6.923	170.12	-3.941
251.532	6.129	84.238	41.050	195.21	194.9	-0.156	212.88	9.054
251.532	6.635	84.191	40.527	207.90	194.14	-6.618	212.35	2.140
251.532	7.526	85.651	40.150	216.42	194.54	-10.110	213.13	-1.517
251.532	8.285	86.512	39.792	219.87	228.97	4.141	215.27	-2.090
251.532	9.487	84.895	39.496	222.75	232.75	4.489	219.52	-1.453

G3

Table G.4 (Test 1807)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
0	3.696	87.941	65.708	152.67	159.14	4.238	160.14	4.893
0	4.908	86.153	65.619	182.45	191.21	4.800	191.40	4.906
0	6.150	84.320	65.166	204.22	215.71	5.625	215.04	5.299
0	7.052	83.174	64.468	216.47	228.93	5.756	227.69	5.186
0	8.091	82.750	63.858	227.61	241.08	5.917	239.20	5.092
0	9.097	81.133	63.046	235.45	250.06	6.205	247.63	5.174
0	10.055	79.668	62.670	242.53	258.18	6.455	255.25	5.245
96.078	3.262	82.232	61.921	182.48	196.95	7.929	192.44	5.458
96.078	4.026	82.694	61.590	195.04	216.10	10.793	211.10	8.231
96.078	4.664	82.180	61.301	215.59	219.35	1.742	241.65	12.088
96.078	5.500	84.409	60.940	235.63	234.91	-0.305	255.86	8.583
96.078	6.182	86.181	60.651	249.83	262.76	5.175	271.89	8.830
96.078	6.728	85.339	60.573	262.39	270.00	2.902	278.86	6.279
96.078	7.541	84.742	60.421	280.53	279.29	-0.442	288.33	2.780
96.078	8.519	83.658	60.175	288.08	290.73	0.917	296.97	3.085
96.078	9.352	81.844	59.894	292.47	291.95	-0.177	301.95	3.240
96.078	10.309	79.449	59.497	296.11	296.98	0.295	306.63	3.552

G4

Table G.5 (Test 1708)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{simp}	error
148.909	11.244	81.621	57.978	356.48	359.67	0.895	376.06	5.492
148.909	10.447	81.262	57.244	343.02	345.51	0.726	362.63	5.718
148.909	9.798	80.863	56.798	333.64	335.39	0.522	353.33	5.899
148.909	9.065	80.144	55.087	312.76	318.68	1.893	331.92	6.126
148.909	8.380	79.598	54.814	302.19	316.70	4.798	323.8	7.150
148.909	7.740	79.247	54.301	291.77	308.46	5.721	313.85	7.568
148.909	7.060	78.669	53.977	279.76	289.22	3.382	303.03	8.316
148.909	6.483	78.835	53.767	262.75	286.97	9.214	295.02	12.280
148.909	5.907	78.817	53.571	241.67	274.20	13.457	249.88	3.397
148.909	5.217	79.641	53.305	220.63	259.99	17.837	237.93	7.841
148.909	4.523	79.255	53.070	198.39	231.09	16.478	224.34	13.077
148.909	3.795	80.138	52.898	180.69	185.44	2.628	177.97	-1.500
127.403	3.440	74.427	57.369	178.44	191.32	7.221	192.14	7.676
127.403	4.070	73.439	56.811	205.10	205.00	-0.048	205.35	0.121
127.403	4.613	73.658	56.591	225.20	246.79	9.588	241.52	7.245
127.403	5.221	73.977	56.126	231.96	224.07	-3.400	252.27	8.753
127.403	5.968	73.460	56.016	253.52	232.38	-8.337	264.86	4.473
127.403	6.554	73.130	55.869	267.06	272.21	1.927	286.02	7.100
127.403	7.394	72.305	55.623	284.43	281.52	-1.021	289.16	1.662
127.403	8.267	72.133	55.424	291.91	286.20	-1.952	298.02	2.093
127.403	9.005	70.505	55.080	287.42	284.33	-1.075	299.77	4.297
127.403	9.923	70.143	54.801	295.02	290.16	-1.644	305.3	3.484
127.403	10.535	69.676	54.443	295.25	291.86	-1.148	306.97	3.967
106.364	3.891	72.907	54.221	185.70	192.56	3.695	192.93	3.889
106.364	4.726	72.602	54.079	211.91	200.85	-5.219	232.61	9.767
106.364	5.347	72.459	53.853	218.44	213.69	-2.173	243.83	11.624
106.364	6.176	72.220	53.526	232.76	237.35	1.972	252.07	8.293
106.364	6.960	72.066	53.340	251.62	247.02	-1.826	261.79	4.042
106.364	7.653	72.456	53.132	263.85	257.13	-2.544	270.19	2.403
106.364	8.547	71.828	52.675	265.56	264.89	-0.251	278.43	4.844
106.364	9.404	71.524	52.467	277.09	268.42	-3.128	286.99	3.575
106.364	10.146	71.201	52.316	283.59	274.73	-3.124	293.39	3.457
106.364	11.077	70.340	52.082	287.93	280.40	-2.616	298.56	3.690

G5

Table G.6 (Test 1907)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{simp}	error
312.779	3.505	76.794	63.024	227.05	254.77	12.209	246.52	8.575
312.779	4.252	78.579	62.202	286.99	272.69	-4.980	264.92	-7.689
312.779	4.824	75.884	61.749	302.54	314.34	3.897	370.44	22.439
312.779	5.224	74.502	60.608	316.01	331.32	4.845	369.13	16.809
312.779	5.904	75.049	60.192	354.91	342.89	-3.387	380.37	7.175
312.779	6.533	73.836	59.647	382.24	433.13	13.315	457.52	19.695
312.779	7.259	73.518	59.039	409.31	438.27	7.076	464.56	13.500
312.779	7.936	72.414	58.335	422.86	477.69	12.967	467.65	10.592
312.779	8.677	70.937	57.837	434.57	482.15	10.947	472.34	8.689
312.779	9.778	70.501	56.646	435.70	478.62	9.852	469.46	7.749
312.779	10.397	70.147	55.951	439.12	467.31	6.418	467.35	6.428
251.532	3.541	77.828	54.335	176.62	190.51	7.860	179.26	1.490
251.532	3.668	77.702	54.171	198.79	193.09	-2.870	181.91	-8.492
251.532	4.297	77.889	53.957	208.03	204.69	-1.608	193.92	-6.785
251.532	5.187	76.581	53.758	239.53	247.77	3.440	238.96	-0.238
251.532	5.908	76.374	53.605	266.20	257.93	-3.109	248.95	-6.481
251.532	6.641	75.272	53.372	295.70	316.7	7.103	333.91	12.922
251.532	7.213	75.623	53.098	308.08	320.05	3.884	338.32	9.815
251.532	8.164	75.730	52.706	324.17	369.18	13.881	343.88	6.078
251.532	9.205	74.144	52.343	336.04	369.97	10.095	351.23	4.518

G6

Table G.7 (Test 2109)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{simp}	error
88.597	4.487	77.276	58.380	190.50	191.20	0.369	200.31	5.153
88.597	5.157	76.859	58.056	207.08	200.50	-3.176	209.57	1.203
88.597	6.042	75.061	57.837	225.63	224.77	-0.378	233.66	3.560
88.597	6.724	74.273	57.652	233.16	230.67	-1.070	239.82	2.852
88.597	7.543	71.700	57.373	248.05	236.80	-4.534	245.8	-0.90
88.597	8.458	70.296	57.069	248.00	238.25	-3.931	249.01	0.404
88.597	9.501	67.485	56.739	251.81	242.54	-3.680	252.06	0.099
88.597	10.200	65.091	56.467	250.24	238.49	-4.693	250.83	0.237
44.416	4.381	60.155	57.084	132.16	139.78	5.767	140.99	6.676
44.416	5.206	60.374	56.983	143.51	151.26	5.399	155.33	8.237
44.416	6.123	61.675	56.923	158.91	163.48	2.874	167.28	5.265
44.416	6.977	61.894	56.833	173.18	173.54	0.209	174.63	0.837
44.416	7.823	62.249	56.741	184.84	182.48	-1.279	183.32	-0.825
44.416	8.628	62.096	56.636	194.60	189.44	-2.651	190.05	-2.338
44.416	9.870	61.702	56.522	202.58	198.72	-1.905	198.98	-1.776

G7

Table G.8 (Test 1908)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
193.325	3.822	81.157	45.348	188.54	174.74	-7.321	167.45	-11.185
193.325	4.338	80.370	44.681	201.32	199.25	-1.029	205.33	1.995
193.325	4.892	80.057	44.566	222.62	223.21	0.262	215.47	-3.211
193.325	5.582	79.477	44.396	248.95	235.14	-5.544	227.11	-8.771
193.325	6.067	79.246	44.222	264.82	292.67	10.516	302.36	14.175
193.325	6.699	78.412	44.013	283.61	299.17	5.489	309.29	9.057
193.325	7.364	77.956	43.742	298.39	299.14	0.249	315.62	5.771
193.325	8.026	76.563	43.553	307.96	321.53	4.407	320.47	4.061
193.325	8.777	75.615	43.358	311.91	320.04	2.604	325.11	4.231
193.325	9.484	75.051	43.055	311.87	323.12	3.605	326.54	4.703
193.325	10.414	73.528	42.796	316.15	320.01	1.219	317.6	0.457
159.896	3.516	79.523	45.782	156.70	163.26	4.189	155.25	-0.922
159.896	3.834	77.104	45.211	189.25	164.23	-13.219	159.78	-15.568
159.896	4.558	76.133	45.092	204.15	200.37	-1.850	200.26	-1.908
159.896	5.205	76.082	44.959	215.72	223.83	3.763	210.66	-2.345
159.896	5.995	75.102	44.793	242.36	268.58	10.819	264.18	9.003
159.896	6.621	75.042	44.686	258.95	271.68	4.917	271.18	4.725
159.896	7.308	74.357	44.556	274.26	272.84	-0.517	278.22	1.445
159.896	8.271	73.642	44.279	279.92	279.29	-0.225	284.57	1.658
159.896	9.086	72.909	44.001	280.71	283.76	1.086	288.52	2.783
159.896	9.889	73.169	43.781	283.11	287.19	1.440	291.99	3.137
159.896	11.161	72.635	43.527	290.16	286.66	-1.206	297.82	2.641

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Table G.9 (Test 2509)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
61.948	4.123	74.919	49.907	127.48	149.32	17.134	137.01	7.472
61.948	4.851	76.579	49.836	141.18	155.53	10.166	160.38	13.603
61.948	5.508	76.844	49.764	160.33	166.46	3.819	171.09	6.707
61.948	6.175	77.397	49.739	176.03	175.00	-0.580	184.18	4.634
61.948	7.137	76.697	49.624	192.50	184.77	-4.013	194.12	0.841
61.948	8.255	77.431	49.303	201.29	195.18	-3.032	204.26	1.479
61.948	9.264	77.915	48.927	207.86	203.89	-1.910	213.07	2.502
35.766	3.839	81.774	48.566	113.48	120.77	6.420	119.24	5.072
35.766	4.768	83.628	48.590	128.72	137.60	6.900	144.30	12.106
35.766	5.619	83.459	48.587	144.37	148.85	3.108	155.04	7.391
35.766	6.532	83.552	48.653	161.69	158.22	-2.147	165.85	2.570
35.766	7.317	83.493	48.560	167.70	165.74	-1.168	173.69	3.574
35.766	8.516	81.927	48.530	177.54	174.39	-1.772	182.23	2.644

Table G.10 (Test 2007)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
216.234	3.245	72.772	58.292	202.41	216.92	7.168	215.65	6.538
216.234	3.803	73.052	57.441	243.98	229.63	-5.881	228.37	-6.399
216.234	4.620	72.272	57.082	274.83	290.59	5.732	295.09	7.370
216.234	5.288	71.538	56.330	289.08	319.44	10.500	304.12	5.203
216.234	5.711	72.345	56.135	310.29	327.03	5.393	311.04	0.240
216.234	6.430	71.720	55.546	328.88	366.32	11.383	379.19	15.296
216.234	7.100	71.498	54.859	320.76	361.33	12.647	383.22	19.471
216.234	7.855	70.762	54.335	350.66	375.35	7.041	383.93	9.486
216.234	8.665	70.310	53.666	359.89	379.92	5.564	388.46	7.937
216.234	9.727	70.089	53.272	369.16	377.29	2.202	396.32	7.359
170.883	3.449	76.342	52.350	174.14	185.09	6.288	180.25	3.506
170.883	3.877	76.677	52.107	199.98	195.43	-2.272	189.32	-5.329
170.883	4.326	77.885	51.507	211.17	231.17	9.471	229.87	8.857
170.883	5.075	77.394	51.300	227.06	261.31	15.082	245.12	7.951
170.883	5.858	77.423	51.105	259.50	275.34	6.104	257.93	-0.608
170.883	6.743	77.609	50.801	283.81	303.34	6.882	311.41	9.724
170.883	7.419	77.876	50.493	296.71	307.15	3.519	317.55	7.024
170.883	8.105	77.618	50.082	303.67	318.65	4.930	322.96	6.352
170.883	9.223	77.370	49.749	313.29	319.43	1.956	332.25	6.049
170.883	10.025	77.055	49.240	317.94	322.53	1.443	335.56	5.539

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Table G.11 (Test 1009)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
200.104	3.421	62.360	48.429	179.58	171.85	-4.306	162.97	-9.249
200.104	3.848	60.125	48.062	205.68	181.14	-11.933	172.44	-16.159
200.104	4.586	58.252	47.916	218.09	217.19	-0.415	218.70	0.278
200.104	5.110	57.935	47.788	226.63	240.66	6.192	227.86	0.546
200.104	5.739	58.733	47.672	247.73	250.56	1.145	236.48	-4.540
200.104	6.327	58.356	47.500	266.65	292.25	9.603	302.41	13.411
200.104	6.983	58.242	47.313	278.08	301.92	8.575	296.81	6.737
200.104	7.890	58.235	47.002	293.57	312.73	6.526	302.75	3.126
200.104	8.846	57.377	46.815	303.04	305.28	0.738	308.38	1.761
200.104	9.709	57.577	46.635	304.33	309.84	1.811	312.24	2.600

Table G.12 (Test 2108)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
350.883	3.505	63.645	50.792	181.71	205.08	12.858	215.61	18.658
350.883	3.810	63.112	50.102	215.18	208.46	-3.124	219.15	1.843
350.883	4.509	63.448	49.871	241.80	235.27	-2.702	288.41	19.274
350.883	5.186	64.718	49.646	275.19	262.73	-4.526	300.52	9.207
350.883	5.938	65.825	49.439	312.60	273.93	-12.370	311.52	-0.345
350.883	6.558	64.239	49.208	348.73	350.31	0.452	410.69	17.765
350.883	7.207	63.083	48.728	366.03	353.17	-3.513	412.47	12.686
350.883	7.938	62.144	48.414	387.33	446.43	15.258	404.60	4.461
350.883	8.669	60.771	48.075	393.64	450.38	14.414	410.11	4.183
350.883	9.584	60.678	47.728	401.16	447.09	11.447	415.70	3.623
350.883	10.549	59.546	47.540	407.84	444.37	8.957	423.21	3.768
300.156	3.617	63.048	46.755	168.20	173.84	3.353	163.11	-3.029
300.156	3.999	62.878	46.581	186.09	180.29	-3.120	169.77	-8.770
300.156	4.562	62.614	46.503	201.60	199.75	-0.914	238.88	18.494
300.156	5.198	62.149	46.417	224.93	222.81	-0.943	248.98	10.695
300.156	5.697	61.256	46.260	243.28	229.43	-5.692	255.14	4.872
300.156	6.207	61.408	45.971	265.28	287.03	8.197	331.68	25.028
300.156	6.741	60.738	45.734	287.06	290.08	1.052	317.45	10.586
300.156	7.583	60.151	45.478	309.59	296.11	-4.355	325.63	5.180
300.156	8.484	58.797	45.266	323.06	370.74	14.760	334.48	3.536
300.156	9.756	56.889	45.004	329.07	366.06	11.240	344.70	4.749

G10

Table G.13 (Test 0409)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{simp}	error
300.156	3.489	57.412	49.820	161.65	162.47	0.507	156.88	-2.954
300.156	3.916	56.008	49.549	178.71	170.13	-4.800	164.22	-8.110
300.156	4.556	55.305	49.442	195.00	198.23	1.659	237.00	21.538
300.156	5.176	55.065	49.340	219.65	220.20	0.250	245.99	11.989
300.156	5.794	54.506	49.026	240.79	227.40	-5.558	224.62	-6.712
300.156	6.362	53.807	48.690	265.26	291.10	9.741	316.88	19.463
300.156	7.061	53.952	48.476	289.40	296.97	2.618	324.61	12.165
300.156	7.846	53.739	48.349	306.86	347.69	13.304	333.10	8.553
300.156	8.583	53.275	48.207	315.99	345.82	9.441	335.84	6.281
300.156	9.513	52.773	48.080	324.98	345.07	6.179	324.38	-0.186
350.883	3.443	60.321	45.316	153.12	145.81	-4.774	144.04	-5.933
350.883	3.797	60.124	45.156	157.31	151.19	-3.894	149.28	-5.107
350.883	4.384	60.536	45.101	169.75	159.59	-5.980	146.09	-13.935
350.883	4.981	60.069	45.049	184.05	175.15	-4.839	194.34	5.587
350.883	5.494	59.597	44.985	197.72	180.71	-8.604	199.02	0.657
350.883	5.898	59.946	44.917	219.92	184.96	-15.895	203.36	-7.530
350.883	6.355	59.550	44.847	238.64	227.15	-4.811	253.53	6.239
350.883	7.096	59.626	44.778	259.11	232.67	-10.204	260.31	0.464
350.883	7.887	59.166	44.673	270.85	281.52	3.940	266.30	-1.679
350.883	8.667	58.987	44.578	276.94	287.23	3.713	272.27	-1.686

G11

Table G.14 (Test 0909)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{simp}	error
350.883	3.164	66.568	61.477	205.83	229.19	11.349	254.92	23.851
350.883	3.540	66.215	60.989	260.60	227.77	-12.596	264.06	1.330
350.883	4.152	65.672	60.290	270.37	255.67	-5.435	278.65	3.061
350.883	4.733	64.977	60.050	301.08	299.66	-0.473	366.14	21.607
350.883	5.220	64.483	59.874	331.02	328.54	-0.748	377.17	13.940
350.883	5.701	64.100	59.533	356.28	336.99	-5.414	385.29	8.141
350.883	6.206	63.920	59.311	382.92	439.63	14.809	472.56	23.408
350.883	6.806	63.747	58.923	411.10	446.40	8.585	477.76	16.214
350.883	7.436	63.645	58.661	427.93	451.19	5.435	482.55	12.764
350.883	8.285	63.188	58.160	444.41	485.32	9.206	487.24	9.637
350.883	9.207	62.621	57.814	458.11	491.29	7.242	491.72	7.336
275.61	3.568	69.560	57.782	225.91	207.75	-8.038	207.26	-8.252
275.61	3.903	69.588	57.368	249.37	215.12	-13.736	214.58	-13.950
275.61	4.736	69.687	57.159	276.31	268.33	-2.890	275.75	-0.202
275.61	5.190	69.924	56.960	295.57	292.16	-1.153	283.67	-4.025
275.61	5.656	70.286	56.764	313.31	299.86	-4.291	291.15	-7.073
275.61	6.114	69.679	56.502	333.21	379.03	13.751	394.77	18.474
275.61	6.584	69.467	56.245	349.54	383.53	9.724	392.19	12.202
275.61	7.105	69.489	56.076	365.78	388.05	6.089	398.03	8.817
275.61	7.875	69.040	55.840	377.87	417.17	10.401	405.84	7.401
275.61	8.634	68.758	55.611	387.59	413.47	6.675	412.31	6.378
275.61	9.545	68.680	55.289	391.74	419.57	7.104	418.2	6.753

G12

Table G.15 (Test 1709)

M_w	v_{max}	ϕ_{ai}	T_{pi}	q_{exp}	q_{Acc}	error	q_{Simp}	error
237.273	3.619	81.302	61.754	264.22	250.61	-5.151	250.39	-5.232
237.273	4.060	80.949	61.423	291.90	262.18	-10.182	262.00	-10.242
237.273	4.761	80.292	60.868	310.91	322.61	3.762	328.99	5.814
237.273	5.297	78.625	60.396	334.34	352.34	5.384	339.81	1.637
237.273	5.731	77.030	59.991	348.26	358.45	2.926	345.29	-0.852
237.273	6.147	76.311	59.582	357.76	397.83	11.201	416.98	16.553
237.273	6.609	74.685	59.069	370.35	401.44	8.394	419.30	13.215
237.273	7.267	73.593	58.708	384.94	407.62	5.893	423.58	10.037
237.273	8.217	72.613	58.223	397.70	415.21	4.404	431.29	8.447
237.273	9.000	72.158	57.690	403.40	420.45	4.227	435.25	7.895
237.273	9.968	73.064	57.292	410.43	427.74	4.216	441.53	7.578
325.403	3.423	77.871	55.302	218.09	213.64	-2.039	207.00	-5.084
325.403	3.745	78.522	54.926	240.28	221.07	-7.995	214.48	-10.737
325.403	4.484	78.967	54.729	267.84	260.72	-2.660	309.77	15.652
325.403	5.092	78.971	54.535	299.66	288.70	-3.658	322.85	7.738
325.403	5.723	78.727	54.163	330.21	298.67	-9.552	332.38	0.655
325.403	6.082	79.082	53.940	349.67	377.25	7.888	424.43	21.381
325.403	6.611	78.375	53.626	374.35	382.66	2.217	420.63	12.361
325.403	7.343	77.740	53.280	395.74	389.37	-1.609	429.58	8.551
325.403	8.065	76.675	52.839	408.22	462.92	13.400	436.41	6.907
325.403	8.826	76.675	52.408	420.47	468.26	11.365	442.30	5.191
325.403	9.638	76.072	52.084	427.15	474.33	11.046	448.93	5.098

H1

APPENDIX H

FORTRAN CODE FOR COMPUTER PROGRAMS

```

Program      : Mist2.pas (Vertical air flow cooler - model ACCURATE)
Programmer   : D.E. Kriel
Date        : 20/10/90

```

```
Program Mist2;
```

```
($N+,E+)      (use 8087 if present)
uses Crt,UTIL;
```

```
type
```

```

mat = array [0..10,0..80] of double;
col = array [0..80] of double;
bol = array [0..10,0..80] of boolean;

```

```
var
```

```

Tspw,Tdbsa,Tsww,smsww,
xx,isa,Twbsa           : mat;
dT                    : col;
Wet,drops             : bol;
Item                  : StrArr;
K1,K2,K3,K4,K5,K6,K7,K8,
Ac,Afr,Acw,At,Aw,Aev,Asf,Ast,
sksf,spsl,spss,skspiw,
Pisl,hxsl,W,Pist,cdsh,
Fss,Fst,Fsh,Feff,Feffd,
xi,xo,Psa,Mevst,iai,
hsw,hw,hc,hspf,U,Ud,Vcrit,
Cpa,Cpp,Cpw,Cpm,Vsamax,
smssw1,smsswi,smspwsmsa,smsswo,
Tspwoe,Tdbsaoe,Twbsaoe,
isaa,xxa,smswwa,Tfilm,
Tdbsaa,Tspwa,Tswwa,
Tdbsai,Tsswi,Tspwi,Twbsai : double;
i,r,Pirows,Picol,
Selected,FirstPos,step : integer;
Inputfile,Up,Single,Dry : boolean;
Default,DefDir,readfile,
Title,Chosen,Key       : string;
spray,inpf             : text;

```

```
(* * * Subroutine to initialize starting values * * *)
```

```
Procedure Start;
```

```
begin
```

```

Fsh      := 9.5;    { Fin height [mm]}
Fss      := 2.5;    { Fin spacing [mm]}
Fst      := 0.48;   { Fin thickness [mm]}
Pirows   := 4;      { Number of pipe rows }
Picol    := 31;     { Number of pipe coloms }
smspww   := 10.0;   { Process water mass flow [kg/s]}
smsswi   := 450.0;  { Spray water mass flow [kg/m²/hr]}
smsa     := 10.0;   { Air mass flow [kg/s]}
Tspwi    := 50.0;   { Temperature of process water [°C]}
Tsswi    := 15.0;   { Temperature of inlet spray water [°C]}
Tdbsai   := 25.0;   { Drybulb temperature of air [°C]}
Twbsai   := 19.5;   { Wetbulb temperature of air [°C]}
spsl     := 36.3;   { Pipe outer diameter [mm]}
spss     := 13.5;   { Pipe inner diameter [mm]}
skspiw   := 43.0;   { Thermal conductivity of pipe [W/m/°C]}
sksf     := 52.4;   { Thermal conductivity of fin [W/m/°C]}
hxsl     := 0.218;  { Heat exchanger length [m]}
W        := 1.1;    { Heat exchanger width [m]}
Pisl     := 1.4;    { Pipe length [m]}

```

```

Pist      := 1.8;      ( Pipe wall thickness [mm])
hw        := 1500.0;  ( Film heat transfer coefficient [W/m2/°C])
Psa       := 101.325; ( Atmospheric Pressure [kPa])
step      := 5;      ( Steps in calculation for each piperow)
default   := 'SPRAY'; ( Name of output file)
Inputfile := False;
Key       := 'Keyboard';
end;

```

```

(-----)

```

```

(* * * Subroutine to capture input data * * *)

```

```

Procedure Input;

```

```

var

```

```

    smspw0, smspw1 : double;
    V, F, n        : integer;

```

```

begin

```

```

    V := 1;

```

```

    IF InputFile THEN F := 1 ELSE F := 0;

```

```

    repeat

```

```

        window(1,1,80,25);

```

```

        clrscr;

```

```

        IF InputFile THEN

```

```

            Item[1] := 'I - Input from data file..... '+Chosen

```

```

        ELSE

```

```

            Item[1] := 'I - User input data from keyboard';

```

```

        IF Single and InputFile THEN

```

```

            Item[2] := 'F - File processing: Single file '+Chosen

```

```

        ELSE IF InputFile THEN

```

```

            Item[2] := 'F - File processing: Multiple files '+copy(Chosen,1,6)+'*.DAT';

```

```

        Item[2+F] := 'O - Cooler configuration: ';

```

```

        Item[3+F] := '    - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';

```

```

        Item[4+F] := '    - Number of Pipe coloms        = '+ITS(Picol,3)+' [-]';

```

```

        Item[5+F] := '    - Pipe length                = '+RTS(Pisl,8,3)+' [m]';

```

```

        Item[6+F] := '    - Pipe thickness              = '+RTS(Pist,8,3)+' [mm]';

```

```

        Item[7+F] := '    - Film coefficient             = '+RTS(hw,8,3)+' [W/m2/°C]';

```

```

        IF Inputfile THEN

```

```

            Item[8+F] := '    Massflow: '

```

```

        ELSE

```

```

            Item[8+F] := 'M - Massflow: ';

```

```

        Item[9+F] := '    - Massflow of Process water     = '+RTS(smspw,8,3)+' [kg/s]';

```

```

        Item[10+F] := '    - Massflow of Spray water      = '+RTS(smsswi,8,3)+' [kg/m2/hr]';

```

```

        Item[11+F] := '    - Mass flow of Air             = '+RTS(smsa,8,3)+' [kg/s]';

```

```

        IF Inputfile THEN

```

```

            Item[12+F] := '    Temperatures: '

```

```

        ELSE

```

```

            Item[12+F] := 'T - Temperatures: ';

```

```

        Item[13+F] := '    - Temperature of Process water = '+RTS(Tspwi,8,3)+' [°C]';

```

```

        Item[14+F] := '    - Temperature of Spray water   = '+RTS(Tsswi,8,3)+' [°C]';

```

```

        Item[15+F] := '    - Drybulb Temperature of air    = '+RTS(Tdbsai,8,3)+' [°C]';

```

```

        Item[16+F] := '    - Wetbulb Temperature of air    = '+RTS(Twbsai,8,3)+' [°C]';

```

```

        Item[17+F] := 'S - Iteration steps / pipe row   = '+ITS(step,4)+' [-]';

```

```

        Item[18+F] := 'N - Output file name is .....'+default;

```

```

        Item[19+F] := 'C - Continue';

```

```

    Title := 'VERTICAL SPRAY COOLED H.X.';

```

```

    FirstPos:= V;

```

```

    DISPLAYMENU((19+F),FirstPos,Item,Title,Selected);

```

```

    IF not InputFile and (Selected>1) THEN Selected := Selected+1;

```

```

    IF (InputFile)and(Selected>2+F) THEN Selected := Selected+2;

```

```

    CASE Selected OF

```

```

        1 : begin

```

```

            ClrScr;

```

```

            FirstPos := 1;

```

```

            repeat;

```

```

                window(1,1,80,25);

```

```

                ClrScr;

```

```

                Title := 'Input from '+Key;

```

```

                Item[1] := 'N - Choose New Input file';

```

```

Item[2] := 'K - Input data from Keyboard';
Item[3] := 'C - Continue';
DISPLAYMENU(3,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    window(1,1,80,25);
    ClrScr;
    Key := 'Datafile';
    CHOOSEFILE(DefDir,'*.DAT',Chosen);
    if Chosen='' then InputFile:=False else InputFile:=True;
    IF InputFile THEN
      begin
        readfile:=defdir+Chosen;
        assign(inpf,readfile);
        reset(inpf);
        read(inpf,Psa);
        read(inpf,Tspwi);
        read(inpf,Tspwoe);
        read(inpf,Tsswi);
        read(inpf,Tdbsai);
        read(inpf,Twbsai);
        read(inpf,Tdbsaoe);
        read(inpf,Twbsaoe);
        read(inpf,smsa);
        read(inpf,smspw0);
        read(inpf,smspw1);
        read(inpf,smsswi);
        FOR n:=1 TO Pirows DO read(inpf,dT[n]);
        IF smspw0>smspw1 THEN smspw := smspw0 ELSE smspw := smspw1;
        Psa := Psa/1000.0;
        smsswi := smsswi*3600/(Pisl*W);
        close(inpf);
        F := 1;
      end;
      Single := True;
      FirstPos := 3;
    end;
  2 : begin
    InputFile := False;
    Key := 'Keyboard';
    FirstPos := 3;
    Single := True;
    F := 0;
  end;
end;
until Selected=3;
V := 19+F;
end;
2 : begin
  IF Single THEN Single := False ELSE Single := True;
end;
3 : begin
  repeat;
    window(1,1,80,25);
    ClrScr;
    Title := 'COOLER CONFIGURATION';
    FirstPos := 6;
    Item[1] := 'R - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';
    Item[2] := 'O - Number of Pipe coloms         = '+ITS(Picol,3)+' [-]';
    Item[3] := 'L - Pipe length                       = '+RTS(Pisl,8,3)+' [m]';
    Item[4] := 'T - Pipe thickness                   = '+RTS(Pist,8,3)+' [mm]';
    Item[5] := 'F - Film coefficient                       = '+RTS(hw,8,3)+' [W/m²/°C]';
    Item[6] := 'C - Continue!';
    DISPLAYMENU(6,FirstPos,Item,Title,Selected);
    CASE Selected OF
      1 : begin
        repeat
          WINDOWBOX(10,18,60,1,'');
          write('  Give the number of Pipe rows  ');
          Pirows:=readint;
          until (Pirows>=0) and (Pirows<=10);
        end;
      end;
    end;
  end;
end;

```

```

2 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the number of Pipe coloms ');
    Picol:=readint;
    until (Picol>=0) and (Picol<=100);
  end;
3 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the length of the Pipes [m] ');
    Pisl:=readreal;
    until (Pisl<5) and (Pisl>0);
  end;
4 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the thickness of the tube walls [mm] ');
    Pist:=readreal;
    until (Pist<10) and (Pist>0);
  end;
5 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the Film Heat Transfer Coefficient [W/m2/°C]');
    hw:=readreal;
    until (hw<100000) and (hw>100);
  end;
end;
until Selected=6;
V := 19+F;
end;
4 : begin
repeat;
window(1,1,80,25);
ClrScr;
Title := 'MASSFLOW';
FirstPos := 4;
Item[1] := 'P - Massflow of Process water = '+RTS(smspw,8,3)+' [kg/s]';
Item[2] := 'S - Massflow of Spray water = '+RTS(smsswi,8,3)+' [kg/m2/hr] ';
Item[3] := 'A - Mass flow of Air = '+RTS(smsa,8,3)+' [kg/s]';
Item[4] := 'C - Continue!';
DISPLAYMENU(4,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the massflow of Process water [kg/s] ');
      smspw:=readreal;
      until (smspw<100) and (smspw>0);
    end;
  2 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the massflux of Spray water [kg/m2/hr] ');
      smsswi:=readreal;
      until (smsswi>=0) and (smsswi<1000);
    end;
  3 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the massflow of Air [kg/s] ');
      smsa:=readreal;
      until (smsa>=0) and (smsa<100);
    end;
  end;
until Selected=4;
V := 19+F;
end;
5 : begin
repeat;
window(1,1,80,25);

```



```

ClrScr;
Title := 'TEMPERATURES ';
FirstPos := 5;
Item[1] := 'P - Temperature of Process water = '+RTS(Tspwi,8,3)+' ['°C] ' ;
Item[2] := 'S - Temperature of Spray water = '+RTS(Tsswi,8,3)+' ['°C] ' ;
Item[3] := 'D - Drybulb Temperature of air = '+RTS(Tdbsai,8,3)+' ['°C] ' ;
Item[4] := 'W - Wetbulb Temperature of air = '+RTS(Twbsai,8,3)+' ['°C] ' ;
Item[5] := 'C - Continue';
DISPLAYMENU(5,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the temperature of Process water ['°C] ');
      Tspwi:=readreal;
    until (Tspwi<80) and (Tspwi>20);
  end;
  2 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the temperature of Spray water ['°C] ');
      Tsswi:=readreal;
    until (Tsswi<40) and (Tsswi>0);
  end;
  3 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the drybulb temperature of the Air ['°C] ');
      Tdbsai:=readreal;
    until (Tdbsai<=40) and (Tdbsai>=Twbsai);
  end;
  4 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the wetbulb temperature of the Air ['°C] ');
      Twbsai:=readreal;
    until (Twbsai>=-10) and (Twbsai<=Tdbsai);
  end;
  until Selected=5;
  V := 19+F;
end;
6 : begin
  repeat
    WINDOWBOX(15,23,50,1,'');
    write(' Give number of iteration steps / pipe ... ');
    step:=ReadInt;
  until (step<20) and (step>0);
  V := 19+F;
end;
7 : begin
  WINDOWBOX(15,23,50,1,'');
  write(' Give the new file name ... ');
  default:=Readfilename;
  V := 19+F;
end;
end; (case)
until (Selected=8);
end;

```

{* * * Subroutine to read the Input data file * * *}

```

Procedure Readinpf;
var
  smspw0,smspw1 : double;
  n              : integer;
begin
  read(inpf,Psa);

```

```

read(inpf,Tspwi);
read(inpf,Tspwoe);
read(inpf,Tsswi);
read(inpf,Tdbsai);
read(inpf,Twbsai);
read(inpf,Tdbsaoe);
read(inpf,Twbsaoe);
read(inpf,smsa);
read(inpf,smspw0);
read(inpf,smspw1);
read(inpf,smsswi);
FOR n:=1 TO Pirows DO read(inpf,dT[n]);
IF smspw0>smspw1 THEN smspw := smspw0 ELSE smspw := smspw1;
Psa := Psa/1000.0;
smsswi := smsswi*3600.0/(Pisl*W);
close(inpf);
end;

{
}

(* * * Subroutine to calculate the saturation temperature of mist flow * * *)

Procedure SatMist(var Tdbsaii,Twbsaii,Tsswii,mswi:double; var Tni,Tniw,mswo:double);
var
  isai,xii,Isti,Ist,xxi,Tl,
  Tr,Tl1,Tr1,cond,isaii : double;

begin
  isai := enthalpy(Tdbsaii,Twbsaii,Psa*1000)*1000.0;
  xxi := Airhumidity(Tdbsaii,Twbsaii,Psa*1000);
  Ist := smsa*isai+Afr*mswi*Cpwf(Tsswii)*(Tsswii);
  IF Tsswii>Twbsaii THEN
    begin
      IF Tsswii>Tdbsaii THEN
        begin
          Tl1 := Twbsaii;
          Tr1 := Tsswii;
        end
      ELSE
        begin
          Tl1 := Twbsaii;
          Tr1 := Tdbsaii;
        end;
      end
    end
  ELSE
    begin
      Tl1 := Tsswii;
      Tr1 := Tdbsaii;
    end;
  TL := Tl1; Tr := Tr1;
  repeat
    Tni := (Tl+Tr)/2.0;
    xii := Airhumidity(Tni,Tni,Psa*1000.0);
    mswo := mswi*Afr-smsa*(xii-xxi);
    isaii := Enthalpy(Tni,Tni,Psa*1000.0)*1000.0;
    Isti := smsa*isaii+mswo*Cpwf(Tni)*(Tni);
    cond := Isti-Ist;
    IF cond>0.0 THEN Tr:=Tni ELSE Tl:=Tni;
  until Abs(cond)<0.01;
  IF mswo>0.0 THEN
    begin
      Tni := (Tl+Tr)/2.0;
      Tniw := Tni;
      xii := SatAirhumidity(Tni,Psa*1000.0);
      mswo := mswi-smsa*(xii-xxi)/Afr;
    end
  ELSE
    begin
      mswo := 0.0;
      xii := xxi+mswi*Afr/smsa;
      Tni := (Ist/smsa-xii*HfgWater(0.0))/(Cpa+xii*Cpvf(Tdbsaii));
      Tl := Tl1; Tr := Tr1;
    end
  end
end

```

```

repeat
  Tniw := (Tl+Tr)/2.0;
  isaii := smsa*Enthalpy(Tni,Tniw,Psa*1000.0)*1000.0;
  cond := Ist-isaii;
  IF cond>0.0 THEN Tl:=Tniw ELSE Tr:=Tniw;
until Abs(cond)<0.1;
  Tniw := (Tl+Tr)/2.0;
end;
end;
(
(
(* * * Subroutine to calculate the starting conditions * * *)

Procedure Initialcond;

var
  def,sm : double;
  m,n : integer;

begin
  xi := Airhumidity(Tdbsai,Twbsai,Psa*1000.0);
  iai := Enthalpy(Tdbsai,Twbsai,Psa*1000.0)*1000.0;
  smsswi := smsswi/3600.0;
  IF smsswi=0.0 THEN Wet[1,1] := false
  ELSE
  begin
    drops[1,1] := true;
    Wet[1,1] := true;
  end;
  smspw := smspw/(Picol-0.5)/step;
  smsa := smsa/(Picol-0.5);
  Aw := 2.0*Pi*sqrt(0.5*(sqr(spss/2.0-Pist)+sqr(spss/2.0-Pist)))*Pisl/step;
  Afr := W*Pisl/(Picol-0.5);
  Ast := 2.0*Pi*sqrt(0.5*(sqr(spss/2.0)+sqr(spss/2.0)))*Pisl/step;
  Asf := 2.0*Pi*((spss/2.0+Fsh)*(spss/2.0+Fsh)-(spss/2.0)*(spss/2.0))*Int(Pisl/Fss)/step;
  At := Ast+Asf;
  Ac := (spss*Pisl+Int(Pisl/Fss)*Fsh*Fst*2.0)/step;
  def := 4.0*(Fsh*(Fss-Fst)/(2.0*Fsh+(Fss-Fst)));
  Vcrit := sqrt(9.81*def*(Waterdensity(Tsswi)-Airdensity(Tdbsai,Psa*1000.0))
    /sqrt(Airdensity(Tdbsai,Psa*1000.0)));
  Vsamax := smsa/Airdensity(Tdbsai,Psa*1000)/(Afr-Ac*step);
  IF (Vsamax>Vcrit*1.1) THEN Up:=True ELSE Up:=False;
  IF (Vsamax<Vcrit*1.4) THEN hsw := hw*2.0/3.0 ELSE hsw := hw;
  IF (Vsamax<Vcrit*0.8) THEN
  begin
    IF Fss < 3.0e-3 THEN
      Acw := (spss*Pisl+Int(Pisl/Fss)*Fsh*(Fss)*2.0)/step
    ELSE
      Acw := (spss*Pisl+Int(Pisl/Fss)*Fsh*(Fst+3.0e-3)*2.0)/step;
  end
  ELSE Acw := (spss*Pisl+Int(Pisl/Fss)*Fsh*(Fst+4.0e-4)*2.0)/step;

  cdsh := 4.0*(Afr-Ac*step)/Afr*hxsl;

  IF (Vsamax>Vcrit*0.9) and Wet[1,1] THEN Aev := 0.7 ELSE IF Wet[1,1] THEN Aev := 0.5;
  IF Wet[1,1] THEN SatMist(Tdbsai,Twbsai,Tsswi,smsswi,Tdbsa[1,1],Twbsa[1,1],smssw1)
  ELSE
  begin
    Tdbsa[1,1] := Tdbsai;
    Twbsa[1,1] := Twbsai;
    smssw1 := smsswi;
  end;

  smsww[1,1] := smssw1*Acw*step;
  IF Wet[1,1] THEN
  begin
    Tsww[1,1] := Twbsa[1,1];
    IF not Up THEN Tsww[2,1] := Twbsa[1,1];
  end;
end;

```

```

xx[1,1] := Airhumidity(Tdbsa[1,1],Twbsa[1,1],Psa*1000.0)+smssw1*(Afr-Acw*step)/smsa;
isa[1,1] := Enthalpy(Tdbsa[1,1],Twbsa[1,1],Psa*1000.0)*1000.0+smssw1*(Afr-Acw*step)
/smsa*Cpwf(Twbsa[1,1])*Twbsa[1,1];
end;

```

```

(-----)

```

```

(* * * Subroutine to find the fin effectivity * * *)

```

```

Procedure Fineff(var h:double; var aeff:double);
var
  M,eff : double;
begin
  M := sqrt(2.0*h/(sksf*Fst));
  eff := Htan(M*Fsh)/(M*Fsh);
  aeff := 1.0-Asf/At*(1.0-eff);
end;

```

```

(-----)

```

```

(* * * Subroutine to find constant coefficients * * *)

```

```

Procedure Constants;
var
  vsw,Resw,f,Prsa,hs2,iv,
  ksa,vsa,Resa,densai,de : double;
begin
  IF Wet[i,r] and ((i=1) or (Up and (Vsamax<Vcrit*1.4) and (i>2) ) or not Up) THEN
  begin
    IF Up and (Vsamax<Vcrit*1.4) and (i=4) THEN
    begin
      At := At*0.4;
      Asf := Asf*0.4;
      Ast := Ast*0.4;
    end
    ELSE
    begin
      At := At*0.7;
      Asf := Asf*0.7;
      Ast := Ast*0.7;
    end;
  end;
  de := 4.0*(spsl/2.0-Pist)*(spss/2.0-Pist)
/(2.0*sqrt(0.5*((sqr(spsl/2.0-Pist)+sqr(spss/2.0-Pist)))));
  Resw := de*smpw*step/(WaterViscosity(Tspwa)*Pi*(spsl/2.0-Pist)*(spss/2.0-Pist));
  f := 1/sqr(1.82*ln(Resw)/ln(10)-1.64);
  hspf := (Waterconductivity(Tspwa)/spss)*(f/8*Resw*Waterprandtl(Tspwa)/
(1.07+12.7*(p(Waterprandtl(Tspwa),0.67)-1)*sqrt(f/8)));
  vsa := Airviscosity(Tdbsa[i,r]);
  densai := Airdensity(Tdbsa[i,r],Psa*1000.0);
  Resa := cdsh*smsa/((Afr-Ac*step)*vsa);
  Cpa := Cpaf(Tdbsa[i,r]);
  Prsa := airprandtl(Tdbsa[i,r]);
  ksa := airconductivity(Tdbsa[i,r]);
  IF Wet[i,r] THEN
  begin
    Cpw := Cpwf(Tswwa);
    iv := Hfgwater(Tswwa)+Cpw*Tswwa;
  end;
  Cpp := Cpwf(Tspwa);
  IF Wet[i,r] THEN
    Cpm := Cpa+Airhumidity(Tdbsa[i,r],Twbsa[i,r],Psa*1000.0)*Cpvf(Twbsa[i,r])
+(xxa-Airhumidity(Tdbsa[i,r],Twbsa[i,r],Psa*1000.0))*Cpw
  ELSE
    Cpm := Cpa+Airhumidity(Tdbsa[i,r],Twbsa[i,r],Psa*1000.0)*Cpvf(Twbsa[i,r]);
  hc := 4.038075*P(Resa,-0.5205963)*P(Prsa,(-2/3))*Cpa*smsa/(Afr-Ac*step);
  Fineff(hsw,Feff);

```

```
Fineff(hc,Feffd);
```

```
Ud := 1/(1/hspf+Pist/(skspiw*Pisl)+Aw/(At*Feffd*hc));
```

```
U := 1/(1/hspf+Pist/(skspiw*Pisl)+Aw/(At*Feff*hsww));
```

```
IF Wet[i,r] and drops[i,r] THEN
```

```
begin
```

```
  K1 := -hc/Cpm*At*Aev;
```

```
  K2 := -K1/smsa;
```

```
  K3 := U*Aw/(smspw*Cpp);
```

```
  K4 := hc*At/(Cpm*smsa);
```

```
  K5 := Cpw*hc*At/(smsa*Cpm)*Tswwa;
```

```
  K6 := K4*iv*(Aev-1.0);
```

```
end
```

```
ELSE IF Wet[i,r] THEN
```

```
begin
```

```
  K1 := -hc/Cpm*At*Aev;
```

```
  K2 := -K1/smsa;
```

```
  K3 := U*Aw/(smspw*Cpp);
```

```
  K4 := hc*At/(Cpm*smsa);
```

```
  K5 := K4*iv*(Aev-1.0);
```

```
end;
```

```
K7 := Ud*Aw/(smspw*Cpp);
```

```
K8 := Ud*Aw/(smsa*Cpa);
```

```
IF Wet[i,r] and ((i=1) or (Up and (Vsamax<Vcrit*1.4) and (i>2) ) or not Up) THEN
```

```
begin
```

```
  IF Up and (Vsamax<Vcrit*1.4) and (i=4) THEN
```

```
  begin
```

```
    At := At/0.4;
```

```
    Asf := Asf/0.4;
```

```
    Ast := Ast/0.4;
```

```
  end
```

```
  ELSE
```

```
  begin
```

```
    At := At/0.7;
```

```
    Asf := Asf/0.7;
```

```
    Ast := Ast/0.7;
```

```
  end;
```

```
end;
```

```
end;
```

```
{-----}
```

```
(* * * Subroutine to calculate the outlet air temperature of an element * * *)
```

```
Procedure SatAirTemp(var x,ai:double; var Tw,Td:double ;var drop:boolean);
```

```
var
```

```
  Tl,Tr,cond,iit,ias,was : double;
```

```
begin
```

```
  Tw := 18.51963+190.80157666*x+4.501708159/x+7.914675e-3/sqr(x)
```

```
      -5.02569691/(x-0.001)+0.2220918*Psa-2.875155*x*x*Psa;
```

```
  iit := Enthalpy(Tw,Tw,Psa*1000.0)*1000.0;
```

```
  IF iit>ai THEN
```

```
  begin
```

```
    drop := true;
```

```
    Tw := 28.633344+0.0677895e-3*ai+3300.0311e3/ai+72773.807e6/(ai*ai)
```

```
          -5351.2607e3/(ai-6000)+0.1182024*Psa-3.5367202e-9*ai*ai/Psa;
```

```
    Td := Tw;
```

```
  end
```

```
  ELSE
```

```
  begin
```

```
    drop := false;
```

```
    Td := (ai-x*HfgWater(0.0))/(Cpa+x*Cpvf(Tdbsa[i,r]));
```

```
    Tl := Tw-1.0;
```

```
    Tr := Tspw[i,r];
```

```
    repeat
```

```
      Tw := (Tl+Tr)/2.0;
```

```
      iit := Enthalpy(Td,Tw,Psa*1000.0)*1000.0;
```

```
      cond := iit-ai;
```

```
      IF cond>0.0 THEN Tr:=Tw ELSE Tl:=Tw;
```

```
    until Abs(cond)<1.0e-3;
```

```
end;
end;
```

```
var
  Tspw2,Tdbsa2,isa2,
  xx2,smsww2,Tsww2 : double;
```

```
{  
-----  
}
```

```
{* * * Subroutine to solve the diff. eq. using Fourth order Runge Kutta * * *}
```

```
Procedure Element;
```

```
var
  a,b,c,d,dwas,
  dwa,dia,dTpw : array [0..10] of double;
  Twbsan,Tdbsan,x,
  ai,Tl,Tr,cond : double;
  j : integer;
  dropsa : boolean;
```

```
begin
```

```
  IF Wet[i,r] THEN
```

```
    begin
```

```
      Dry := false;
```

```
      IF drops[i,r] THEN
```

```
        begin
```

```
          Tspwa := (Tspw[i,1]+Tspw2)/2.0;
```

```
          isaa := (isa[i,r]+isa2)/2.0;
```

```
          xxa := (xx[i,r]+xx2)/2.0;
```

```
          Satairtemp(xxa, isaa, Twbsan, Tdbsan, dropsa);
```

```
          duas[1] := (Airhumidity(Tfilm, Tfilm, Psa*1000.0) - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
          dwa[1] := (xxa - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
          dia[1] := (Enthalpy(Tfilm, Tfilm, Psa*1000.0)*1000.0 - isaa);
```

```
          dTpw[1] := (Tspwa - Tfilm);
```

```
          a[1] := K1*dwas[1];
```

```
          b[1] := K2*dwas[1];
```

```
          c[1] := K3*dTpw[1];
```

```
          d[1] := K4*dia[1]+K5*dwa[1]+K6*dwas[1];
```

```
        FOR j:=2 TO 3 DO
```

```
          begin
```

```
            Tspwa := ((Tspw[i,1]+c[j-1]/2.0)+Tspw2)/2.0;
```

```
            isaa := ((isa[i,r]+d[j-1]/2.0)+isa2)/2.0;
```

```
            xxa := ((xx[i,r]+b[j-1]/2.0)+xx2)/2.0;
```

```
            Satairtemp(xxa, isaa, Twbsan, Tdbsan, dropsa);
```

```
            duas[j] := (Airhumidity(Tfilm, Tfilm, Psa*1000.0) - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
            dwa[j] := (xxa - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
            dia[j] := (Enthalpy(Tfilm, Tfilm, Psa*1000.0)*1000.0 - isaa);
```

```
            dTpw[j] := (Tspwa - Tfilm);
```

```
            a[j] := K1*dwas[j];
```

```
            b[j] := K2*dwas[j];
```

```
            c[j] := K3*dTpw[j];
```

```
            d[j] := K4*dia[j]+K5*dwa[j]+K6*dwas[j];
```

```
          end;
```

```
        Tspwa := ((Tspw[i,1]+c[3])+Tspw2)/2.0;
```

```
        isaa := ((isa[i,r]+d[3])+isa2)/2.0;
```

```
        xxa := ((xx[i,r]+b[3])+xx2)/2.0;
```

```
        Satairtemp(xxa, isaa, Twbsan, Tdbsan, dropsa);
```

```
        duas[4] := (Airhumidity(Tfilm, Tfilm, Psa*1000.0) - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
        dwa[4] := (xxa - Airhumidity(Twbsan, Twbsan, Psa*1000.0));
```

```
        dia[4] := (Enthalpy(Tfilm, Tfilm, Psa*1000.0)*1000.0 - isaa);
```

```
        dTpw[4] := (Tspwa - Tfilm);
```

```
        a[4] := K1*dwas[4];
```

```
        b[4] := K2*dwas[4];
```

```
        c[4] := K3*dTpw[4];
```

```
        d[4] := K4*dia[4]+K5*dwa[4]+K6*dwas[4];
```

```
    end
```

```
  ELSE
```

```

begin
  Tspw2 := (Tspw[i,1]+Tspw2)/2.0;
  isaa := (isa[i,r]+isa2)/2.0;
  xxa := (xx[i,r]+xx2)/2.0;
  dwas[1] := (Airhumidity(Tfilm,Tfilm,Psa*1000.0)-xxa);
  dia[1] := (Enthalpy(Tfilm,Tfilm,Psa*1000.0)*1000.0-isa2);
  dTpw[1] := (Tspw2-Tfilm);
  a[1] := K1*dwas[1];
  b[1] := K2*dwas[1];
  c[1] := K3*dTpw[1];
  d[1] := K4*dia[1]+K5*dwas[1];

  FOR j:=2 TO 3 DO
  begin
    Tspw2 := ((Tspw[i,1]+c[j-1]/2.0)+Tspw2)/2.0;
    isaa := ((isa[i,r]+d[j-1]/2.0)+isa2)/2.0;
    xxa := ((xx[i,r]+b[j-1]/2.0)+xx2)/2.0;

    dwas[j] := (Airhumidity(Tfilm,Tfilm,Psa*1000.0)-xxa);
    dia[j] := (Enthalpy(Tfilm,Tfilm,Psa*1000.0)*1000.0-isa2);
    dTpw[j] := (Tspw2-Tfilm);
    a[j] := K1*dwas[j];
    b[j] := K2*dwas[j];
    c[j] := K3*dTpw[j];
    d[j] := K4*dia[j]+K5*dwas[j];
  end;

  Tspw2 := ((Tspw[i,1]+c[3])+Tspw2)/2.0;
  isaa := ((isa[i,r]+d[3])+isa2)/2.0;
  xxa := ((xx[i,r]+b[3])+xx2)/2.0;

  dwas[4] := (Airhumidity(Tfilm,Tfilm,Psa*1000.0)-xxa);
  dia[4] := (Enthalpy(Tfilm,Tfilm,Psa*1000.0)*1000.0-isa2);
  dTpw[4] := (Tspw2-Tfilm);
  a[4] := K1*dwas[4];
  b[4] := K2*dwas[4];
  c[4] := K3*dTpw[4];
  d[4] := K4*dia[4]+K5*dwas[4];
end;
smsww[i,r+1] := smsww[i,r]+(a[1]+2.0*a[2]+2.0*a[3]+a[4])/6.0;
IF smsww[i,r+1]<0.0 THEN
begin
  smsww[i,r+1] := 0.0;
  Dry := True;
end
ELSE
begin
  xx[i,r+1] := xx[i,r]+(b[1]+2.0*b[2]+2.0*b[3]+b[4])/6.0;
  Tspw[i,r+1] := Tspw[i,1]+(c[1]+2.0*c[2]+2.0*c[3]+c[4])/6.0;
  isa[i,r+1] := isa[i,r]+(d[1]+2.0*d[2]+2.0*d[3]+d[4])/6.0;

  Satairtemp(xx[i,r+1],isa[i,r+1],Twbsa[i,r+1],Tdbsa[i,r+1],drops[i,r+1]);
  IF not drops[i,r] THEN drops[i,r+1] := false;
  Wet[i,r+1] := true;
end;
end;
IF not Wet[i,r] or Dry THEN
begin
  Tspw2 := (Tspw[i,1]+Tspw2)/2.0;
  Tdbsa := (Tdbsa[i,r]+Tdbsa2)/2.0;

  dTpw[1] := (Tspw2-Tdbsa);
  a[1] := K7*dTpw[1];
  b[1] := K8*dTpw[1];

  FOR j:=2 TO 3 DO
  begin
    Tspw2 := (Tspw[i,1]+a[j-1]/2.0+Tspw2)/2.0;
    Tdbsa := (Tdbsa[i,r]+b[j-1]/2.0+Tdbsa2)/2.0;

    IF drops[i,r] or Dry THEN
  begin

```

```

IF Dry THEN xxa := xx[i,r]+smsww[i,r]/smsa*(j-1.0)/3.0
    ELSE xxa := xx[i,r];
IF Dry THEN isaa := isa[i,r]+Cpa*b[j-1]/4.0+smsww[i,r]/smsa*(j-1.0)
    /3.0*Cpvf(Tdbsa[i,r])*Tdbsa[i,r]
    ELSE isaa := isa[i,r]+Cpa*b[j-1]/4.0;
Satairtemp(xx[i,r],isaa,Twbsan,Tdbsa,dropsa);
end;

dTpw[j] := (Tspwa-Tdbsa);
a[j] := K7*dTpw[j];
b[j] := K8*dTpw[j];
end;

Tspwa := (Tspw[i,1]+a[3]+Tspw2)/2.0;
Tdbsa := (Tdbsa[i,r]+b[3]+Tdbsa2)/2.0;

IF drops[i,r] or Dry THEN
begin
    IF Dry THEN xxa := xx[i,r]+smsww[i,r]/smsa
        ELSE xxa := xx[i,r];
    IF Dry THEN isaa := isa[i,r]+Cpa*b[3]/2.0+smsww[i,r]/smsa*Cpvf(Tdbsa[i,r])*Tdbsa[i,r]
        ELSE isaa := isa[i,r]+Cpa*b[3]/2.0;
    Satairtemp(xx[i,r],isaa,Twbsan,Tdbsa,dropsa);
end;

dTpw[4] := (Tspwa-Tdbsa);
a[4] := K7*dTpw[4];
b[4] := K8*dTpw[4];

Tspw[i,r+1] := Tspw[i,1]+(a[1]+2.0*a[2]+2.0*a[3]+a[4])/6.0;

IF drops[i,r] or Dry THEN
begin
    IF Dry THEN xx[i,r+1] := xx[i,r]+smsww[i,r]/smsa ELSE xx[i,r+1] := xx[i,r];
    IF Dry THEN isa[i,r+1] := isa[i,r]+Cpa*(b[1]+2.0*b[2]+2.0*b[3]+b[4])/6.0
        +smsww[i,r]/smsa*Cpvf(Tdbsa[i,r])*Tdbsa[i,r]
        ELSE isa[i,r+1] := isa[i,r]+Cpa*(b[1]+2.0*b[2]+2.0*b[3]+b[4])/6.0;
    Satairtemp(xx[i,r+1],isa[i,r+1],Twbsa[i,r+1],Tdbsa[i,r+1],drops[i,r+1]);
    IF not drops[i,r] THEN drops[i,r+1] := false;
end
ELSE
begin
    Tdbsa[i,r+1] := Tdbsa[i,r]+(b[1]+2.0*b[2]+2.0*b[3]+b[4])/6.0;
    Tl := Twbsa[i,r]-0.2;
    Tr := Tdbsa[i,r+1]+0.2;
    repeat
        Twbsan := (Tl+Tr)/2.0;
        x := Airhumidity(Tdbsa[i,r+1],Twbsan,Psa*1000.0);
        cond := xx[i,r+1]-x;
        IF cond>0.0 THEN Tl := Twbsan ELSE Tr := Twbsan;
    until Abs(cond)<1.0e-6;
    Twbsa[i,r+1] := Twbsan;
    isa[i,r+1] := Enthalpy(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0)*1000.0;
    xx[i,r+1] := xx[i,r];
    drops[i,r+1] := false;
end;

Wet[i,r+1] := false;
smsww[i,r+1] := 0.0;
end;
end;

```

```
( _____ )
```

```
(* * * Subroutine to calculate the Film temperature on the tube * * *)
```

```
Procedure Filmtemp;
```

```
var
```

```
cond1,cond,T2,Qt,incr : double;
```

```
j : integer;
```



```

begin
  FOR j:=1 TO 2 DO
  begin
    Tsww2 := Tsww[i,r+1];
    Tspw2 := Tspw[i,r+1];
    isa2 := isa[i,r+1];
    xx2 := xx[i,r+1];
    Tdbsa2 := Tdbsa[i,r+1];
    smsww2 := smsww[i,r+1];
    Tswwa := (Tsww[i,r]+Tsww2)/2.0;
    Tspwa := (Tspw[i,r]+Tspw2)/2.0;
    isaa := (isa[i,r]+isa2)/2.0;
    xxa := (xx[i,r]+xx2)/2.0;
    smswwa := (smsww[i,r]+smsww2)/2.0;
    Tdbsa := (Tdbsa[i,r]+Tdbsa2)/2.0;

    Constants;

    IF Wet[i,r] THEN
    begin
      incr := 1.0;
      Tfilm := (Tspw[i,r+1]+Twbsa[i,r])/2.0;
      repeat

        Element;

        Qt := smspw*(Tspw[i,r+1]-Tspw[i,r])*Cpw;
        Qt := smsa*(isa[i,r+1]-isa[i,r])+smsww[i,r+1]*Cpw*(Tfilm-Tsww[i,r])
            +(smsww[i,r+1]-smsww[i,r])*Cpw*Tfilm;

        T2 := (Tspw[i,r]+Tspw[i,r+1])/2.0-Qt/(U*Aw);
        cond := Tfilm-T2;
        IF cond/cond1<0.0 THEN incr := incr/2.0;
        IF cond>0.0 THEN Tfilm := Tfilm-incr ELSE Tfilm := Tfilm+incr;
        cond1 := cond;
        until (Abs(cond)<5.0e-5) or Dry;
        Tsww[i,r+1] := Tfilm
      end;
      IF not Wet[i,r] or Dry THEN Element;
      Dry := false;
    end;
  end;
end;

(-----)

(* * * Subroutine to calculate the inlet conditions of each pipe row * * *)

Procedure Startcond;
var
  tl : double;
  j : integer;

begin
  IF Up and (Wet[i,r+1] or drops[i,r+1]) THEN
  begin
    IF drops[i,r+1] THEN tl := 1.0 ELSE tl := 0.0;
    smsww[i+1,1] := smsww[i,r+1]+tl*smsa*(Acw*step/Afr)
        *(xx[i,r+1]-Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0));
    IF smsww[i+1,1]<0.0 THEN
    begin
      smsww[i+1,1] :=0.0;
      drops[i,r+1] := false;
      Tsww[i+1,1] := Tspw[i,1];
    end
    ELSE
      Tsww[i+1,1] := 1.0/smsww[i+1,1]*(Tsww[i,r+1]*smsww[i,r+1]+tl*Twbsa[i,r+1]*sma*(Acw*step/Afr)
          *(xx[i,r+1]-Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0)));
  end
  ELSE IF Wet[i,r+1] and not Up and (i<=2) THEN
  begin

```

```

IF i=1 THEN
begin
  smsww[i+1,1] := smsa*(xx[i,r+1]-Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0));
  smsww[i,0] := smsww[i,r+1];
  Tsww[i,0] := Tsww[i,r+1];
end
ELSE
begin
  smsww[1,1] := smssw1*Acw*step+smsww[i,r+1];
  Tsww[1,1] := 1.0/smsww[1,1]*(Twbsa[1,1]*smssw1*Acw*step+Tsww[i,r+1]*smsww[i,r+1]);
  Wet[i,r+1] := false;
end;
end
ELSE
begin
  smsww[i+1,1] := 0.0;
  Tsww[i+1,1] := Tspw[i,1];
end;
IF Up and drops[i,r+1] THEN
begin
  xx[i+1,1] := xx[i,r+1]-(Acw*step/Afr)*(xx[i,r+1]
    -Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0));
  isa[i+1,1] := isa[i,r+1]-smsa*(Acw*step/Afr)*(xx[i,r+1]
    -Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0))*Cpwf(Twbsa[i,r+1])*Twbsa[i,r+1];
end
ELSE
begin
  IF not Up and (i=1) THEN
  begin
    xx[i+1,1] := Airhumidity(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0);
    isa[i+1,1] := Enthalpy(Tdbsa[i,r+1],Twbsa[i,r+1],Psa*1000.0)*1000.0;
  end
  ELSE
  begin
    xx[i+1,1] := xx[i,r+1];
    isa[i+1,1] := isa[i,r+1];
  end;
end;
Tdbsa[i+1,1] := Tdbsa[i,r+1];
Twbsa[i+1,1] := Twbsa[i,r+1];
Tspw[i+1,1] := 0.0;
FOR j:=2 TO step+1 DO Tspw[i+1,1] := Tspw[i+1,1]+Tspw[i,j]/step;
IF drops[i,r+1] and Up THEN drops[i+1,1] := true ELSE drops[i+1,1] := false;
IF drops[i+1,1] or Wet[i,r+1] THEN Wet[i+1,1] := true ELSE Wet[i+1,1] := false;
end;

```

```

var
  smsswi,smsswo,Qe,QT,
  smsswto,MsdT,Qac,Qswc : double;
  is : string;

```

```

( * * * Subroutine to print the final results on the screen * * *)

```

```

Procedure Output;

```

```

begin
  Qac := smsa*(Picol-0.5)*(isa[Pirows+1,1]-iai)/1000.0;

  IF not Up and (smsswi>0.0) THEN
  begin
    smsww[Pirows+1,1] := 0.0;
    MsdT := smsww[1,0];
  end
  ELSE MsdT := 0.0;

  xo := Airhumidity(Tdbsa[Pirows+1,1],Twbsa[Pirows+1,1],Psa*1000.0);
  IF smsswi>0.0 THEN
  begin
    Mevst := (Picol-0.5)*smsa*(xo-xi);
    IF drops[Pirows+1,1] and Up THEN smsswo := smsa*(xx[Pirows+1,1]-xo) ELSE smsswo := 0.0;
  end;

```

```

smsswsi := smsswi*W*Pisl;
smsswso := (smssw[Pirows+1,1]+smsswo+MsdT)*(Picol-0.5);
smsswto := smsswso+Mevst;
IF UP THEN
Qswc := (Picol-0.5)*smssw[Pirows+1,1]*Cpwf(Tsswi)*(Tsw[Pirows+1,1]-Tsswi)/1000.0
      +(Picol-0.5)*smsswo*Cpwf((Twbsa[Pirows+1,1]+Tsswi)/2.0)*(Twbsa[Pirows+1,1]-Tsswi)/1000.0
      +Mevst*Cpwf(Tsswi)*Tsswi/1000.0
ELSE
Qswc := (Picol-0.5)*MsdT*Cpwf((Tsw[1,0]+Tsswi)/2.0)*(Tsw[1,0]-Tsswi)/1000.0
      +(Picol-0.5)*smsswo*Cpwf((Twbsa[Pirows+1,1]+Tsswi)/2.0)*(Twbsa[Pirows+1,1]-Tsswi)/1000.0
      +Mevst*Cpwf(Tsswi)*Tsswi/1000.0;
end
ELSE
begin
smsswsi := 0.0;
smsswso := 0.0;
smsswto := 0.0;
Qswc := 0.0;
Mevst := 0.0;
end;

OUTLINE('FINAL RESULTS');
writeln(' For vertical cooler with FINNED tubes and ',3600*smsswi:5:1,' kg/m3/hr spraywater');
writeln;
writeln(' Spraywater massflow is                               : ',smsswsi:5:4,' kg/s');
writeln;
writeln(' Total massflow of spraywater out of H.X.                 : ',smsswso:5:4,' kg/s');
writeln(' Total massflow of spraywater that evaporates             : ',Mevst:5:4,' kg/s');
writeln(' TOTAL                                                       : ',smsswto:5:4,' kg/s');
writeln;
writeln(' Inlet Temperature of the Process water                   : ',Tspwi:7:4,' °C');
IF InputFile THEN
  writeln(' Outlet Temperature of the Process water              : ',Tspw[1,1]:7:4,' (',Tspwoe:7:4,') °C')
ELSE
  writeln(' Outlet Temperature of the Process water                  : ',Tspw[1,1]:7:4,' °C');
writeln(' Mass flow of the Air                                     : ',(smsa*(Picol-0.5)):6:3,' kg/s');
writeln(' Inlet condition of the Air                              : ',Tdb sai:6:3,' °C (Dryb)');
writeln('                                                           : ',Twbsai:6:3,' °C (Wetb)');
IF InputFile THEN begin
  writeln(' Outlet condition of the Air                              : ',Tdbsa[Pirows+1,1]:6:3,
        ' (',Tdb saoe:6:3,') °C (Dryb)');
  writeln('                                                           : ',Twbsa[Pirows+1,1]:6:3,
        ' (',Twbsa oe:6:3,') °C (Wetb)');
end
ELSE begin
  writeln(' Outlet condition of the Air                              : ',Tdbsa[Pirows+1,1]:6:3,' °C (Dryb)');
  writeln('                                                           : ',Twbsa[Pirows+1,1]:6:3,' °C (Wetb)');
end;
writeln;
IF InputFile THEN
  writeln(' Capacity of the cooler is                               : ',QT:7:2,' (',Qe:7:2,') kW')
ELSE
  writeln(' Capacity of the cooler is                               : ',QT:7:2,' kW');
writeln(' Energy change of the Air is                             : ',Qac:7:2,' kW');
writeln(' Energy change of the Spray water is                    : ',Qswc:7:2,' kW');
writeln;
WINDOWBOX(25,23,30,1,'');
write(' Press spacebar to continue');
repeat
  until Keypress;
end;
(-----)
(* * * Subroutine to write the final results to file * * *)

```

Procedure Save;

var

```

Aeff,phi,Perror : double;
j,k              : integer;

```

```

begin
  Aeff := 0.0;
  FOR j := 1 TO Pirows DO FOR k := 1 TO step DO
  begin
    IF Wet[j,k] THEN
    begin
      Aeff := Aeff+1.0;
    end;
  end;
  Aeff := 100.0*Aeff/(step*Pirows);
  ( IF InputFile THEN Perror := 100.0*(QT-Qe)/Qe;
  IF InputFile THEN Perror := 0.0;
  phi := 100.0*xi/Airhumidity(Tdbsai,Tdbsai,Psa*1000.0);

  write(spray,' ',(smsswi*3600.0):7:3);
  write(spray,' ',(smsa*(Picol-0.5)):7:3);
  write(spray,' ',Vsamax:7:4);
  write(spray,' ',phi:7:4);
  write(spray,' ',Tspwi:7:3);
  write(spray,' ',Tspw[1,1]:7:3);
  write(spray,' ',Qe:7:2);
  write(spray,' ',QT:7:2);
  write(spray,' ',Perror:7:4);
  FOR k:=Pirows DOWNT0 1 DO write(spray,' ',(Tspw[k+1,1]-Tspw[k,1]):7:4);
  writeln(spray);
end;

```

```

var
  condm,Tr,Tl,incr,cond1   : double;
  ii,j,code1,error,S      : integer;
  Choice,First            : boolean;
  answer,character        : char;
  istr                    : string;

```

```

(* * * MAIN PROGRAM * * *)

```

```

begin
  GetDir(0,DefDir);
  IF copy(DefDir,length(DefDir),1)<>'\' THEN DefDir:=DefDir+'\'

  Start;

  repeat;                ( * Start of main loop * )
    Input;

    IF (Inputfile)and(not Single) THEN
    begin
      ii:=0;
      repeat
        inc(ii);
        Str(ii,is);
        assign(spray,DefDir+default+is+'.prn');
        ($I-)
        reset(spray);
        ($I+)
        error:=IOresult;
        if error=0 then close(spray);
      until error<>0;
      rewrite(spray);
      writeln(spray,' '+copy(Chosen,1,6));
    end;
    Choice := false;
    spsl := spsl/1000.0;
    spss := spss/1000.0;
    Fsh := Fsh/1000.0;
    Fss := Fss/1000.0;
    Fst := Fst/1000.0;
    Pist := Pist/1000.0;
    window(1,1,80,25);

```

```

ClrScr;
ii := 0;
repeat;
  S := 0;
  cond1 := 0.0;
  First := true;
  IF (not Single) THEN
  begin
    inc(ii);
    Str(ii,istr);
    readfile:=defdir+copy(Chosen,1,6)+istr+'.dat';
    assign(inpfile,readfile);
    {$I-}
    reset(inpfile);
    {$I+}
    error:=IOResult;
    OUTLINE('Multiple File Processing');
    WINDOWBOX(7,4,68,1,'');
    write(' Processing file '+copy(Chosen,1,6)+istr+'.dat: ');
  end
  ELSE IF InputFile THEN
  begin
    error := 0;
    writeln('                      RESULTS');
    writeln;
    write('In Temp ',Tspwi:7:4);
    FOR i:=1 TO Pirows DO write(' dT',i:1,' ',dT[i]:6:3);
    writeln(' Out Temp ',Tspwoe:7:4);
  end
  ELSE
  begin
    error := 0;
    writeln('                      RESULTS');
    writeln;
    writeln('Process Water inlet Temp. ',Tspwi:7:4,' °C   Process water outlet Temp. °C');
  end;
  if error=0 then    ( if file exists --> evaluate )
  begin
    IF (InputFile)and(not Single) THEN Readinpf;

    Initialcond;

    Tspw[1,1] := 0.35*Twbsai+0.65*Tspwi;
    repeat

      FOR i := 1 TO Pirows DO
      begin
        FOR r := 1 TO step DO
        begin
          Tspw[i,r+1] := Tspw[i,r];
          isa[i,r+1] := isa[i,r];
          xx[i,r+1] := xx[i,r];
          Tsww[i,r+1] := Tsww[i,r];
          smsww[i,r+1] := smsww[i,r];
          Tdbsa[i,r+1] := Tdbsa[i,r];

          Filtemp;
        end;

        Startcond;
      end;
      IF Single and Inputfile THEN
      begin
        write('          ',Tspw[Pirows+1,1]:7:4);
        FOR i:=Pirows DOWNT0 1 DO write('          ',(Tspw[i+1,1]-Tspw[i,1]):6:3);
        writeln('          ',Tspw[1,1]:7:4);
      end
      ELSE IF Single THEN writeln('          ',Tspw[Pirows+1,1]:7:4,'          ',Tspw[1,1]:7:4);
      ELSE write('.');

      condm := Tspw[Pirows+1,1]-Tspwi;
      IF (Abs(conda)>10.0) THEN

```

```

Tspw[1,1] := Tspw[1,1]-condm/3.0
ELSE IF (Abs(condm)>0.1) THEN
  Tspw[1,1] := Tspw[1,1]-condm/1.7
ELSE
begin
  IF (Abs((condm+cond1)/condm)<0.5) or (not First) THEN
  begin
    Tspw[1,1] := Tspw[1,1]-condm/10.0;
    IF First THEN First := False;
  end;
  IF Up THEN Tspw[1,1] := Tspw[1,1]-condm/3.0 ELSE Tspw[1,1] := Tspw[1,1]-condm/1.7;
  cond1 := condm;
end;
Inc(S);
until (Abs(condm)<1.0E-3) or (S>16);
IF First THEN Tspw[1,1] := Tspw[1,1]+condm/3.0 ELSE Tspw[1,1] := Tspw[1,1]+condm/10.0;
Qe := smspw*(Picol-0.5)*step*Cpwf((Tspwi+Tspwoe)/2.0)*(Tspwi-Tspwoe)/1000.0;
QT := smspw*(Picol-0.5)*step*Cpwf((Tspw[Pirows+1,1]+Tspw[1,1])/2.0)
      *(Tspw[Pirows+1,1]-Tspw[1,1])/1000.0;

IF (Single)or(not Inputfile) THEN
begin
  Output;
  error := 1;
end
ELSE Save;
end;
until (error<>0);
IF not Single THEN close(spray);
window(1,1,80,25);
ClrScr;
Item[1] := 'R - Return to Main Menu';
Item[2] := 'Q - Quit';
Title := '';
FirstPos := 1;
DISPLAYMENU(2,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    choice := false;
    spsl := spsl*1000.0;
    spss := spss*1000.0;
    Fsh := Fsh*1000.0;
    Fss := Fss*1000.0;
    Fst := Fst*1000.0;
    Pist := Pist*1000.0;
    smsswi := smsswi*3600;
    smspw := smspw*(Picol-0.5)*step;
    smsa := smsa*(Picol-0.5);
  end;
  2 : begin
    choice := true;
  end;
end; { case }

until choice;
ClrScr;
end.

```

```

Program   : Mist1.pas (Vertical air flow cooler - model SIMPLE)
Programmer : D.E. Kriel
Date      : 30/09/90

```

```
Program Mist1;
```

```
{ $N+,E+ }           (use 8087 if present)
uses Crt,UTIL;
```

```
type
```

```
  col = array [0..50] of double;
  bol = array [0..50] of boolean;
```

```
var
```

```

Qst,Tspiwall,Tspw,Tdbsa,Aev,Qsl,
smsswt,Msd,xx,isa,Twbsa,smssw,dT      : col;
Wet                                     : bol;
Item                                    : StrArr;
Ac,hc,Cpa,QT,Afr,Vsamax,sksf,xxa,xi,
QTc,smpw,smsa,spsl,spss,hspf,xl,Aw,
skspiw,Pisl,Psa,Mev,Tsswi,hsw,Tdb sai,
hxsl,Twbsai,Qsc,W,Tspwa,Feff,U,Tdb saa,
Mevst,Fsh,smsswi,Fss,Fst,cdsh,Acw,xo,
Pist,Tspwi,Tspwoe,Tdb saoe,Twbsaoe,A   : double;
r,Pirows,Picol,Selected,FirstPos      : integer;
Finns,InputFile,Up,Single              : boolean;
Default,DefDir,readfile,Title,Chosen,Key : string;
spray,inpfile                           : text;

```

```
(* * * Subroutine to initialize starting values * * *)
```

```
Procedure Start;
```

```
begin
```

```

Finns      := True;    { Finned tubes or smooth pipes}
Fsh        := 9.5;    { Fin height [mm]}
Fss        := 2.5;    { Fin spacing [mm]}
Fst        := 0.48;   { Fin thickness [mm]}
Pirows     := 4;      { Number of pipe rows }
Picol      := 31;     { Number of pipe coloms }
smpw       := 10.0;   { Process water mass flow [kg/s]}
smsswi     := 450.0;  { Spray water mass flow [kg/m²/hr]}
smsa       := 10.0;   { Air mass flow [kg/s]}
Tspwi      := 50.0;   { Temperature of process water [°C]}
Tsswi      := 15.0;   { Temperature of inlet spray water [°C]}
Tdb sai    := 25.0;   { Drybulb temperature of air [°C]}
Twbsai     := 19.5;   { Wetbulb temperature of air [°C]}
spsl       := 36.3;   { Pipe outer diameter [mm]}
spss       := 13.5;   { Pipe inner diameter [mm]}
skspiw     := 43.0;   { Thermal conductivity of pipe [W/m/°C]}
sksf       := 52.4;   { Thermal conductivity of fin [W/m/°C]}
hxsl       := 0.218;  { Heat exchanger length [m]}
W          := 1.1;    { Heat exchanger width [m]}
Pisl       := 1.4;    { Pipe length [m]}
Pist       := 1.8;    { Pipe wall thickness [mm]}
hsw        := 1500.0; { Film heat transfer coefficient [W/m²/°C]}
Psa        := 101.325; { Atmospheric Pressure [kPa]}
default    := 'SPRAY'; { Name of output file}
Inputfile  := False;
Key        := 'Keyboard';

```

```
end;
```

```
(* * * Subroutine to capture input data * * *)
```

```
Procedure Input;
```

```
var
```

```
smspw0,smspw1 : double;
V,F,n         : integer;
```

```
begin
```

```
V := 1;
```

```
IF InputFile THEN F := 1 ELSE F := 0;
```

```
repeat
```

```
  window(1,1,80,25);
```

```
  clrscr;
```

```
  IF InputFile THEN
```

```
    Item[1] := 'I - Input from data file..... '+Chosen
```

```
  ELSE
```

```
    Item[1] := 'I - User input data from keyboard';
```

```
  IF Single and InputFile THEN
```

```
    Item[2] := 'F - File processing: Single file '+Chosen
```

```
  ELSE IF InputFile THEN
```

```
    Item[2] := 'F - File processing: Multiple files '+copy(Chosen,1,6)+'*.DAT';
```

```
  Item[2+F] := 'O - Cooler configuration: ';
```

```
  Item[3+F] := '  - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';
```

```
  Item[4+F] := '  - Number of Pipe coloms         = '+ITS(Picol,3)+' [-]';
```

```
  Item[5+F] := '  - Pipe length                   = '+RTS(Pisl,8,3)+' [m]';
```

```
  Item[6+F] := '  - Pipe thickness                = '+RTS(Pist,8,3)+' [mm]';
```

```
  Item[7+F] := '  - Film coefficient              = '+RTS(hsw,8,3)+' [W/m2/°C]';
```

```
  IF Inputfile THEN
```

```
    Item[8+F] := '    Massflow: '
```

```
  ELSE
```

```
    Item[8+F] := 'M - Massflow: ';
```

```
  Item[9+F] := '  - Massflow of Process water     = '+RTS(smspw,8,3)+' [kg/s]';
```

```
  Item[10+F] := '  - Massflow of Spray water      = '+RTS(smsswi,8,3)+' [kg/m2/hr]';
```

```
  Item[11+F] := '  - Mass flow of Air             = '+RTS(smsa,8,3)+' [kg/s]';
```

```
  IF Inputfile THEN
```

```
    Item[12+F] := '    Temperatures: '
```

```
  ELSE
```

```
    Item[12+F] := 'T - Temperatures: ';
```

```
  Item[13+F] := '  - Temperature of Process water = '+RTS(Tspwi,8,3)+' [°C]';
```

```
  Item[14+F] := '  - Temperature of Spray water   = '+RTS(Tsswi,8,3)+' [°C]';
```

```
  Item[15+F] := '  - Drybulb Temperature of air    = '+RTS(Tdbsai,8,3)+' [°C]';
```

```
  Item[16+F] := '  - Wetbulb Temperature of air   = '+RTS(Twbsai,8,3)+' [°C]';
```

```
  IF Inputfile THEN
```

```
    Item[17+F] := '    Absolute air Pressure      = '+RTS(Psa,8,3)+' [kPa]'
```

```
  ELSE
```

```
    Item[17+F] := 'P - Absolute air Pressure      = '+RTS(Psa,8,3)+' [kPa]';
```

```
  Item[18+F] := 'N - Output file name is .....'+default;
```

```
  Item[19+F] := 'C - Continue';
```

```
Title := 'VERTICAL SPRAY COOLED H.X.';
```

```
FirstPos:= V;
```

```
DISPLAYMENU((19+F),FirstPos,Item,Title,Selected);
```

```
IF not InputFile and (Selected>1) THEN Selected := Selected+1;
```

```
IF (InputFile)and(Selected>2+F) THEN Selected := Selected+3;
```

```
CASE Selected OF
```

```
  1 : begin
```

```
    ClrScr;
```

```
    FirstPos := 1;
```

```
    repeat;
```

```
      window(1,1,80,25);
```

```
      ClrScr;
```

```
      Title := 'Input from '+Key;
```

```
      Item[1] := 'N - Choose New Input file';
```

```
      Item[2] := 'K - Input data from Keyboard';
```

```
      Item[3] := 'C - Continue';
```

```
      DISPLAYMENU(3,FirstPos,Item,Title,Selected);
```

```
      CASE Selected OF
```

```
        1 : begin
```

```
          window(1,1,80,25);
```

```
          ClrScr;
```

```
          Key := 'Datafile';
```



```

CHOOSEFILE(DefDir, '*.DAT', Chosen);
if Chosen='' then InputFile:=False else InputFile:=True;
IF InputFile THEN
begin
  readfile:=defdir+Chosen;
  assign(inpf, readfile);
  reset(inpf);
  read(inpf, Psa);
  read(inpf, Tspwi);
  read(inpf, Tspwoe);
  read(inpf, Tsswi);
  read(inpf, Tdbsai);
  read(inpf, Twbsai);
  read(inpf, Tdbsaoe);
  read(inpf, Twbsaoe);
  read(inpf, smsa);
  read(inpf, smspw0);
  read(inpf, smspw1);
  read(inpf, smsswi);
  FOR n:=Pirows DOWNTO 1 DO read(inpf, dT[n]);
  IF smspw0>smspw1 THEN smspw := smspw0 ELSE smspw := smspw1;
  Psa := Psa/1000.0;
  smsswi := smsswi*3600/(Pisl*W);
  close(inpf);
  F := 1;
end;
Single := True;
FirstPos := 3;
end;
2 : begin
  InputFile := False;
  Key := 'Keyboard';
  FirstPos := 3;
  Single := True;
  F := 0;
end;
end;
until Selected=3;
V := 19+F;
end;
2 : begin
  IF Single THEN Single := False ELSE Single := True;
end;
3 : begin
  repeat;
  window(1,1,80,25);
  ClrScr;
  Title := 'COOLER CONFIGURATION';
  FirstPos := 6;
  Item[1] := 'R - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';
  Item[2] := 'O - Number of Pipe coloms         = '+ITS(Picol,3)+' [-]';
  Item[3] := 'L - Pipe length                       = '+RTS(Pisl,8,3)+' [m]';
  Item[4] := 'T - Pipe thickness                   = '+RTS(Pist,8,3)+' [mm]';
  Item[5] := 'F - Film coefficient                 = '+RTS(hsw,8,3)+' [W/m²/°C]';
  Item[6] := 'C - Continue';
  DISPLAYMENU(6, FirstPos, Item, Title, Selected);
  CASE Selected OF
    1 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the number of Pipe rows ');
        Pirows:=readint;
        Tspw[Pirows+1] := Tspwi;
        until (Pirows>=0) and (Pirows<=100);
      end;
    2 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the number of Pipe coloms ');
        Picol:=readint;
        until (Picol>=0) and (Picol<=100);
      end;
    end;
  end;
end;

```

```

3 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the length of the Pipes [m] ');
    Pisl:=readreal;
    until (Pisl<5) and (Pisl>0);
  end;
4 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the thickness of the tube walls [mm] ');
    Pist:=readreal;
    until (Pist<10) and (Pisl>0);
  end;
5 : begin
  repeat
    WINDOWBOX(10,18,60,1,'');
    write(' Give the Film Heat Transfer Coefficient [W/m2/°C]');
    hsw:=readreal;
    until (hsw<100000) and (hsw>100);
  end;
end;
until Selected=6;
V := 19+F;
end;
4 : begin
  repeat;
  window(1,1,80,25);
  ClrScr;
  Title := 'MASSFLOW';
  FirstPos := 4;
  Item[1] := 'P - Massflow of Process water = '+RTS(smspw,8,3)+' [kg/s]';
  Item[2] := 'S - Massflow of Spray water = '+RTS(smsswi,8,3)+' [kg/m2/hr] ';
  Item[3] := 'A - Mass flow of Air = '+RTS(smsa,8,3)+' [kg/s]';
  Item[4] := 'C - Continue!';
  DISPLAYMENU(4,FirstPos,Item,Title,Selected);
  CASE Selected OF
    1 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflow of Process water [kg/s] ');
        smspw:=readreal;
        until (smspw<100) and (smspw>0);
      end;
    2 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflux of Spray water [kg/m2/hr] ');
        smsswi:=readreal;
        until (smsswi>=0) and (smsswi<1000);
      end;
    3 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflow of Air [kg/s] ');
        smsa:=readreal;
        until (smsa>=0) and (smsa<100);
      end;
    end;
  until Selected=4;
  V := 19+F;
end;
5 : begin
  repeat;
  window(1,1,80,25);
  ClrScr;
  Title := 'TEMPERATURES ';
  FirstPos := 5;
  Item[1] := 'P - Temperature of Process water = '+RTS(Tspwi,8,3)+' [°C] ';
  Item[2] := 'S - Temperature of Spray water = '+RTS(Tsswi,8,3)+' [°C] ';
  Item[3] := 'D - Drybulb Temperature of air = '+RTS(Tdbsai,8,3)+' [°C]';
  Item[4] := 'W - Wetbulb Temperature of air = '+RTS(Twbsai,8,3)+' [°C]';

```

```

Item[5] := 'C - Continue';
DISPLAYMENU(5,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
      repeat
          WINDOWBOX(10,18,60,1,'');
          write(' Give the temperature of Process water [°C] ');
          Tspwi:=readreal;
          until (Tspwi<80) and (Tspwi>20);
      end;
  2 : begin
      repeat
          WINDOWBOX(10,18,60,1,'');
          write(' Give the temperature of Spray water [°C] ');
          Tsswi:=readreal;
          until (Tsswi<40) and (Tsswi>0);
      end;
  3 : begin
      repeat
          WINDOWBOX(10,18,60,1,'');
          write(' Give the drybulb temperature of the Air [°C] ');
          Tdbsai:=readreal;
          until (Tdbsai<40) and (Tdbsai>-10);
      end;
  4 : begin
      repeat
          WINDOWBOX(10,18,60,1,'');
          write(' Give the wetbulb temperature of the Air [°C] ');
          Twbsai:=readreal;
          until (Twbsai>=-10) and (Twbsai<=40);
      end;
  end;
until Selected=5;
V := 19+F;
end;
6 : begin
  repeat
    WINDOWBOX(15,23,50,1,'');
    write(' Give the Air pressure [kPa] ');
    Psa:=readreal;
    until (Psa<103) and (Psa>97);
    V := 19+F;
  end;
7 : begin
  WINDOWBOX(15,23,50,1,'');
  write(' Give the new file name ... ');
  default:=Readfilename;
  V := 19+F;
end;
end; (case)
until (Selected=8);
end;

```

```

{ * * * Subroutine to read the Input data file * * * }

```

```

Procedure Readinpf;

```

```

var

```

```

  smspw0,smspw1 : double;
  n             : integer;

```

```

begin

```

```

  read(inpf,Psa);
  read(inpf,Tspwi);
  read(inpf,Tspwoe);
  read(inpf,Tsswi);
  read(inpf,Tdbsai);
  read(inpf,Twbsai);
  read(inpf,Tdbsaoe);
  read(inpf,Twbsaoe);

```

```

read(inpf,smsa);
read(inpf,smspw0);
read(inpf,smspw1);
read(inpf,smsswi);
FOR n:=Pirows DOWNTO 1 DO read(inpf,dT[n]);
IF smspw0>smspw1 THEN smspw := smspw0 ELSE smspw := smspw1;
Psa := Psa/1000.0;
smsswi := smsswi*3600/(Pisl*W);
close(inpf);
end;

```

```

(-----)

```

```

(* * * Subroutine to calculate the saturation temperature of mist flow * * *)

```

```

Procedure Satair(var Tdbsaii,Twbsaii,Tsswii,mswi:double; var Tni,Tniw,mswo:double);
var

```

```

  isai,xii,Isti,Ist,xxi,Tl,
  Tr,Tl1,Tr1,cond,isaii : double;

```

```

begin

```

```

  isai := enthalpy(Tdbsaii,Twbsaii,Psa*1000)*1000.0;
  xxi := Airhumidity(Tdbsaii,Twbsaii,Psa*1000);
  Ist := smsa*isai+Afr*mswi*Cpwf(Tsswii)*(Tsswii);
  IF Tsswii>Twbsaii THEN

```

```

    begin

```

```

      IF Tsswii>Tdbsaii THEN

```

```

        begin

```

```

          Tl1 := Twbsaii;

```

```

          Tr1 := Tsswii;

```

```

        end

```

```

      ELSE

```

```

        begin

```

```

          Tl1 := Twbsaii;

```

```

          Tr1 := Tdbsaii;

```

```

        end;

```

```

      end

```

```

    ELSE

```

```

      begin

```

```

        Tl1 := Tsswii;

```

```

        Tr1 := Tdbsaii;

```

```

      end;

```

```

    Tl := Tl1; Tr := Tr1;

```

```

  repeat

```

```

    Tni := (Tl+Tr)/2.0;

```

```

    xii := Airhumidity(Tni,Tni,Psa*1000.0);

```

```

    mswo := mswi*Afr-smsa*(xii-xxi);

```

```

    isaii := Enthalpy(Tni,Tni,Psa*1000.0)*1000.0;

```

```

    Isti := smsa*isaii+mswo*Cpwf(Tni)*(Tni);

```

```

    cond := Isti-Ist;

```

```

    IF cond>0.0 THEN Tr:=Tni ELSE Tl:=Tni;

```

```

  until Abs(cond)<0.01;

```

```

  IF mswo>0.0 THEN

```

```

    begin

```

```

      Tni := (Tl+Tr)/2.0;

```

```

      Tniw := Tni;

```

```

      xii := Airhumidity(Tni,Tni,Psa*1000.0);

```

```

      mswo := mswi-smsa*(xii-xxi)/Afr;

```

```

    end

```

```

  ELSE

```

```

    begin

```

```

      mswo := 0.0;

```

```

      xii := xxi+mswi*Afr/smsa;

```

```

      Tni := (Ist/smsa-xii*HfgWater(0.0))/(Cpa+xii*Cpvf(Tdbsaii));

```

```

      Tl := Tl1; Tr := Tr1;

```

```

    repeat

```

```

      Tniw := (Tl+Tr)/2.0;

```

```

      isaii := smsa*Enthalpy(Tni,Tniw,Psa*1000.0)*1000.0;

```

```

      cond := Ist-isaii;

```

```

      IF cond>0.0 THEN Tl:=Tniw ELSE Tr:=Tniw;

```

```

    until Abs(cond)<0.1;

```

```

    Tniw := (Tl+Tr)/2.0;

```

```
end;
end;
```

```
(
  (
    (* * * Subroutine to calculate the starting conditions * * *)

var
  Ast,Asf,def,Vcrit : double;

Procedure Initialcond;

var
  i : integer;

begin
  Tspw[Pirows+1] := Tspwi;
  smsswi := smsswi/3600.0;
  IF smsswi=0.0 THEN Wet[1] := False ELSE Wet[1] := True;
  smspw := smspw/(Picol-0.5);
  smsa := smsa/(Picol-0.5);
  FOR i:=1 TO Pirows DO
  begin
    Aev[i] := 0.0;
    Qsl[i] := 0.0;
    Qst[i] := 0.0;
    Msd[i] := 0.0;
    Tspiwall[i] := 0.0;
  end;
  Aw := 2.0*Pi*sqrt(0.5*(sqr(spsl/2.0-Pist)+sqr(spss/2.0-Pist)))*Pisl;
  Afr := W*Pisl/(Picol-0.5);

  IF Finns THEN
  begin
    Ast := 2.0*Pi*sqrt(0.5*(sqr(spsl/2.0)+sqr(spss/2.0)))*Pisl;
    Asf := 2.0*Pi*((spsl/2.0+Fsh)*(spss/2.0+Fsh)-(spsl/2.0)*(spss/2.0))*Int(Pisl/Fss);
    A := Ast+Asf;
    Ac := spss*Pisl+Int(Pisl/Fss)*Fsh*Fst*2.0;
    Acw := spss*Pisl+Int(Pisl/Fss)*Fsh*(Fst+4.0e-4)*2.0;
  end
  ELSE
  begin
    A := 2.0*Pi*sqrt(0.5*(sqr(spsl/2.0)+sqr(spss/2.0)))*Pisl;
    Ac := spss*Pisl;
  end;

  cdsh := 4.0*(Afr-Ac)/Afr*hxsl;

  IF Wet[1] THEN Satair(Tdbsai,Twbsai,Tsswi,smsswi,Tdbsa[1],Twbsa[1],smssw[1])
  ELSE
  begin
    Tdbsa[1] := Tdbsai;
    Twbsa[1] := Twbsai;
    smssw[1] := smsswi;
  end;

  isa[1] := Enthalpy(Tdbsa[1],Twbsa[1],Psa*1000.0)*1000.0;
  xx[1] := Airhumidity(Tdbsa[1],Twbsa[1],Psa*1000.0);
  def := 4.0*(Fsh*(Fss-Fst)/(2.0*Fsh+(Fss-Fst)));
  Vcrit := sqrt(9.81*def*(Waterdensity(Tsswi)-Airdensity(Tdbsai,Psa*1000.0)))/sqrt(Airdensity(Tdbsai,Psa*1000.0));
  Vsamax := smsa/Airdensity(Tdbsai,Psa*1000)/(Afr-Ac);
  IF Vsamax>Vcrit*1.1 THEN Up:=True ELSE Up:=False;
  IF Finns and (Vsamax<Vcrit*0.8) THEN
  begin
    IF Fss < 3.0e-3 THEN
      Acw := spss*Pisl+Int(Pisl/Fss)*Fsh*(Fss)*2.0
    ELSE
      Acw := spss*Pisl+Int(Pisl/Fss)*Fsh*(Fst+3.0e-3)*2.0;
    end;
  end;
  Msd[Pirows+1] := 0.0;
  Msd[0] := 0.0;
```

```

FOR i:=1 TO Pirows DO
begin
  IF Finns THEN Msd[i] := smssw[i]*Acw ELSE Msd[i] := smssw[i]*Ac;
  smssw[i+1] := smssw[i]-Msd[i]/Afr
end;
IF Up THEN smsswt[1] := smssw[1] ELSE smsswt[1] := smssw[1]+Msd[2]/Ac;
end;

```

```

(-----)

```

```

(* * * Subroutine to calculate the effective wet area of the tubewall * * *)

```

```

Procedure Area;

```

```

begin
  IF smsswt[r]=0.0 THEN
  begin
    Wet[r] := False;
    Aev[r] := 0.0;
  end
  ELSE IF (Msd[r]>0.0)and(Finns)and(smsswt[r]*Ac/A>0.002) THEN
  begin
    Wet[r] := True;
    IF Vsamax>Vcrit*1.1 THEN Aev[r] := 0.8*A ELSE Aev[r] := 0.4*A;
  end
  ELSE IF (Msd[r]>0.0)and(Finns)and(smsswt[r]*Ac/A>0.0005) THEN
  begin
    Wet[r] := True;
    IF Vsamax>Vcrit*1.1 THEN Aev[r] := 0.4*A ELSE Aev[r] := 0.2*A;
  end
  ELSE IF (Msd[r]>0.0)and(not Finns) THEN
  begin
    Wet[r] := True;
    Aev[r] := 0.9*A;
  end
  ELSE IF not Finns THEN
  begin
    Wet[r] := True;
    Aev[r] := 0.5*A;
  end
  ELSE
  begin
    Wet[r] := True;
    Aev[r] := 0.1*A;
  end;
end;
end;

```

```

(-----)

```

```

(* * * Subroutine to find the fin effectivity * * *)

```

```

Procedure Fineff(var h:double; var aeff:double);

```

```

var
  M,eff : double;
begin
  M := sqrt(2.0*h/(sksf*Fst));
  eff := Htan(M*Fsh)/(M*Fsh);
  aeff := 1-Asf/A*(1-eff);
end;

```

```

(-----)

```

```

(* * * Subroutine to find constant coefficients * * *)

```

```

Procedure Constants;

```

```

var
  vsw,Resw,f,Prsa,hsw2,
  ksa,vsa,Resa,densai,de : double;

```

```

begin

```

```

de := 4.0*(spsl/2.0-Pist)*(spss/2.0-Pist)/(2.0*sqrt(0.5*((sqr(spss/2.0-Pist)
+sqr(spss/2.0-Pist))));
Resw := de*smspw/(WaterViscosity(Tspwa)*Pi*(spsl/2.0-Pist)*(spss/2.0-Pist));
f := 1/sqr(1.82*ln(Resw)/ln(10)-1.64);
hspf := (Waterconductivity(Tspwa)/spss)*(f/8*Resw*Waterprandtl(Tspwa)/
(1.07+12.7*(p(Waterprandtl(Tspwa),0.67)-1)*sqrt(f/8)));
vsa := Airviscosity(Tdbsaa);
densai := Airdensity(Tdbsaa,Psa*1000);
IF Finns THEN
    Resa := cdsh*smsa/((Afr-Ac)*vsa)
ELSE Resa := smsa*(de+2.0*Pist)/((Afr-Ac)*vsa);
Cpa := cpaf(Tdbsaa);
Prsa := airprandtl(Tdbsaa);
ksa := airconductivity(Tdbsaa);

IF Finns THEN
    hc := 4.038075*P(Resa,-0.5205963)*P(Prsa,(-2/3))*Cpa*smsa/(Afr-Ac)
ELSE
    hc := ksa/de*P(Resa,0.6)*0.278*P(Prsa,(1/3));

hsw2 := hsw*2.0;
IF Finns and Wet[r] THEN
begin
    IF Aev[r]=0.8*A THEN Fineff(hsw,Feff)
    ELSE Fineff(hsw2,Feff);
end
ELSE IF Finns THEN Fineff(hc,Feff)
ELSE Feff := 1;

IF not Wet[r] THEN
    U := 1/(1/hspf+Pist/(skspiw*Pisl)+Aw/(A*Feff*hc))
ELSE IF (Aev[r]=0.8*A)or(Aev[r]=0.9*A) THEN
    U := 1/(1/hspf+Pist/(skspiw*Pisl)+Aw/(A*Feff*hsw))
ELSE
    U := 1/(1/hspf+Pist/(skspiw*Pisl)+Aw/(A*Feff*hsw*2.0));
end;

```

(* * * Subroutine to find the tubewall temperature * * *)

Procedure Tubewalltemp;

var

Qss,Tspiwall2,Tr,Tl,
cond,hsg,xxm,Tdbm : double;

begin

IF Wet[r] THEN

begin

Tr := Tspwa;

Tl := Tdbsaa;

repeat

xl := SatAirhumidity(Tspiwall[r],Psa*1000.0);

Qsc := hc*(Tspiwall[r]-Tdbsaa)*A;

Tdbm := Tdbsa[r]+Qsc/(smsa*Cpa);

xxm := smsa*(SatAirhumidity(Tdbm,Psa*1000.0)-xx[r]);

IF Up and Finns THEN

Qss := smssw[r]*Acw*cpwf(Tspiwall[r])*(Tspiwall[r]-Twbsa[r])
+ Msd[r-1]*cpwf(Tspiwall[r])*(Tspiwall[r]-Tspiwall[r-1])

ELSE IF Finns THEN

Qss := smssw[r]*Acw*cpwf(Tspiwall[r])*(Tspiwall[r]-Twbsa[r])
+ Msd[r+1]*cpwf(Tspiwall[r])*(Tspiwall[r]-Tspiwall[r+1])

ELSE IF Up THEN

Qss := smssw[r]*Ac*cpwf(Tspiwall[r])*(Tspiwall[r]-Twbsa[r])
+ Msd[r-1]*cpwf(Tspiwall[r])*(Tspiwall[r]-Tspiwall[r-1])

ELSE

Qss := smssw[r]*Ac*cpwf(Tspiwall[r])*(Tspiwall[r]-Twbsa[r])
+ Msd[r+1]*cpwf(Tspiwall[r])*(Tspiwall[r]-Tspiwall[r+1]);

Mev := Aev[r]*hc*(xl-xxa)/Cpa;

IF Mev > xxm THEN Mev := xxm;

```

IF Mev > smsswt[r]*Ac THEN Mev := smsswt[r]*Ac;
hsg := HfgWater(Tspiwall[r])+Cpwf(Tspiwall[r])*Tspiwall[r];
Qsl[r] := Mev*hsg;
Qst[r] := Qsl[r]+Qsc+Qss;

Tspiwall2 := Tspwa-(Qst[r]/(Aw*U));
cond := Tspiwall[r]-Tspiwall2;
IF cond < 0.0 then Tl := Tspiwall[r] ELSE Tr := Tspiwall[r];
Tspiwall[r] := (Tl+Tr)/2.0;
until Abs(cond) < 0.0005;
end
ELSE
begin
Mev := 0.0;
Qsc := Aw*U*(Tspwa-Tdbsa);
Qsl[r] := 0.0;
Qst[r] := Qsc;
Tspiwall[r] := Tspwa-(1/hspf+Pist/(skspiw*Pisl))*Qst[r]/Aw;
end;
end;

(-----)

(* * * Subroutine to calculate the inlet conditions for each pipe * * *)

Procedure Startcond;

var
Tr,Tl,cond,isa2,xxn,Td,Tw,msw,isan : double;

begin
xx[r+1] := xx[r]+Mev/smsa;
Tspw[r+1] := Tspw[r]+Qst[r]/Cpwf(Tspw[r])/smssp;
isan := isa[r]+(Qsl[r]+Qsc)/smsa;
Tdbsa[r+1] := Tdbsa[r]+Qsc/Cpa/smsa;
isa[r+1] := xx[r+1]*(2501600.0+1857.7)+1004.16*Tdbsa[r+1];

Tr := Tdbsa[r+1];
Tl := Twbsa[r];
Twbsa[r+1] := (Tr+Tl)/2.0;
repeat
isa2 := Enthalpy(Tdbsa[r+1],Twbsa[r+1],Psa*1000.0)*1000.0;
cond := isa[r+1]-isa2;
IF cond < 0.0 then Tr := Twbsa[r+1] ELSE Tl := Twbsa[r+1];
Twbsa[r+1] := (Tl+Tr)/2.0;
until Abs(cond) < 1.0E-2;
xxn := Airhumidity(Tdbsa[r+1],Twbsa[r+1],Psa*1000.0);
IF Wet[r] THEN
begin
IF Up and Finns THEN Msd[r] := smssw[r]*Acw-Mev+Msd[r-1]
ELSE IF Finns THEN Msd[r] := smssw[r]*Acw-Mev+Msd[r+1]
ELSE IF Up THEN Msd[r] := smssw[r]*Ac-Mev+Msd[r-1]
ELSE Msd[r] := smssw[r]*Ac-Mev+Msd[r+1];
IF Msd[r] < 0.0000001 THEN Msd[r] := 0.0;
IF Finns THEN smssw[r+1] := smssw[r]-smssw[r]*Acw/Afr
ELSE smssw[r+1] := smssw[r]-smssw[r]*Ac/Afr;

Satair(Tdbsa[r+1],Twbsa[r+1],Twbsa[r],smssw[r+1],Td,Tw,msw);
Tdbsa[r+1] := Td;
Twbsa[r+1] := Tw;
smssw[r+1] := msw;
xx[r+1] := Airhumidity(Tdbsa[r+1],Twbsa[r+1],Psa*1000.0);

IF Up THEN smsswt[r+1] := smssw[r+1]+Msd[r]/Afr
ELSE IF r<4 THEN smsswt[r+1] := smssw[r+1]+Msd[r+2]/Afr
ELSE smsswt[r+1] := smssw[r+1];
end
ELSE
begin
smsswt[r+1] := 0.0;

```



```

Msd[r] := 0.0;
smssw[r+1] := 0.0;
end;
end;

```

```

var
smsswi,smsswo,Qe,
smsswto,MsdT,Qac,Qswc : double;
is : string;

```

```

(-----)

```

```

(* * * Subroutine to print the final results on the screen * * *)

```

```

Procedure Output;

```

```

begin

```

```

Qac := smsa*(Picol-0.5)*(Enthalpy(Tdbsa[Pirows+1],Twbsa[Pirows+1],Psa*1000.0)
-Enthalpy(Tdbsai,Twbsai,Psa*1000.0));

```

```

smsswi := smsswi*W*Pisl;

```

```

smsswo := smssw[Pirows+1]*W*Pisl;

```

```

smsswto := smsswo+MsdT+Mevst;

```

```

IF UP and (smsswi>0.0) THEN

```

```

Qswc := MsdT*cpwf((Tspiwall[Pirows]+Twbsa[1])/2.0)*(Tspiwall[Pirows]-Twbsa[1])/1000.0
+ smssw[Pirows+1]*W*Pisl*cpwf((Twbsa[Pirows+1]+Twbsa[1])/2.0)
* (Twbsa[Pirows+1]-Twbsa[1])/1000.0+Mevst*Cpwf(Tsswi)*Tsswi/1000.0

```

```

ELSE IF smsswi>0.0 THEN

```

```

Qswc := MsdT*cpwf((Tspiwall[1]+Twbsa[1])/2.0)*(Tspiwall[1]-Twbsa[1])/1000.0
+ smssw[Pirows+1]*W*Pisl*cpwf((Twbsa[Pirows+1]+Twbsa[1])/2.0)
* (Twbsa[Pirows+1]-Twbsa[1])/1000.0+Mevst*Cpwf(Tsswi)*Tsswi/1000.0

```

```

ELSE Qswc := 0.0;

```

```

OUTLINE('FINAL RESULTS');

```

```

IF Finns THEN

```

```

writeln(' For vertical cooler with FINNED tubes and ',3600*smsswi:5:1,' kg/m2/hr spraywater')

```

```

ELSE

```

```

writeln(' For vertical cooler with SMOOTH tubes and ',3600*smsswi:5:1,' kg/m2/hr spraywater');

```

```

writeln;

```

```

writeln(' Spraywater massflow is : ',smsswi:5:4,' kg/s');

```

```

writeln;

```

```

writeln(' Spraywater massflow left in outlet air : ',smsswo:5:4,' kg/s');

```

```

writeln(' Total massflow of spraywater dripping off : ',MsdT:5:4,' kg/s');

```

```

writeln(' Total massflow of spraywater that evaporates : ',Mevst:5:4,' kg/s');

```

```

writeln(' TOTAL : ',smsswto:5:4,' kg/s');

```

```

writeln;

```

```

writeln(' Inlet Temperature of the Process water : ',Tspwi:7:4,' °C');

```

```

IF InputFile THEN

```

```

writeln(' Outlet Temperature of the Process water : ',Tspw[1]:7:4,' (',Tspwoe:7:4,') °C')

```

```

ELSE

```

```

writeln(' Outlet Temperature of the Process water : ',Tspw[1]:7:4,' °C');

```

```

writeln(' Mass flow of the Air : ',(smsa*(Picol-0.5)):6:3,' kg/s');

```

```

writeln(' Inlet condition of the Air : ',Tdbsai:6:3,' °C (Dryb)');

```

```

writeln(' : ',Twbsai:6:3,' °C (Wetb)');

```

```

IF InputFile THEN begin

```

```

writeln(' Outlet condition of the Air : ',Tdbsa[r+1]:6:3,

```

```

' (',Tdbsaoe:6:3,') °C (Dryb)');

```

```

writeln(' : ',Twbsa[r+1]:6:3,

```

```

' (',Twbsaoe:6:3,') °C (Wetb)');

```

```

end

```

```

ELSE begin

```

```

writeln(' Outlet condition of the Air : ',Tdbsa[r+1]:6:3,' °C (Dryb)');

```

```

writeln(' : ',Twbsa[r+1]:6:3,' °C (Wetb)');

```

```

end;

```

```

writeln;

```

```

IF InputFile THEN

```

```

writeln(' Capacity of the cooler is (cal. Q of pipes) : ',QT:7:2,' (',Qe:7:2,') kW')

```

```

ELSE

```

```

writeln(' Capacity of the cooler is (cal. Q of pipes) : ',QT:7:2,' kW');

```

```

writeln(' Energy change of the Air is : ',Qac:7:2,' kW');

```

```

writeln(' Energy change of the Spray water is : ',Qswc:7:2,' kW');

```

```
writeln;
WINDOWBOX(25,23,30,1,'');
write(' Press spacebar to continue');
repeat
  until Keypressed;
end;
```

```
(-----)
```

```
(* * * Subroutine to write the final results to file * * *)
```

```
Procedure Save;
```

```
var
```

```
  Aeff,phi,Perror : double;
  i                : integer;
```

```
begin
```

```
  Aeff := 100.0*(Aev[1]+Aev[2]+Aev[3]+Aev[4])/(4.0*A);
```

```
( IF InputFile THEN Perror := 100.0*(QT-Qe)/Qe;)
```

```
  phi := 100.0*Satvappres(Twbsai)/Satvappres(Tdbsai);
```

```
  write(spray, ' ',(smsswi*3600.0):7:3);
```

```
  write(spray, ' ',(smsa*(Picol-0.5)):7:3);
```

```
  write(spray, ' ',Vsamax:7:4);
```

```
  write(spray, ' ',phi:7:4);
```

```
  write(spray, ' ',Tspwi:7:3);
```

```
  write(spray, ' ',Tspw[1]:7:3);
```

```
  write(spray, ' ',Qe:7:2);
```

```
  write(spray, ' ',QT:7:2);
```

```
  write(spray, ' ',Perror:7:4);
```

```
  FOR i:=Pirows DOWNTO 1 DO write(spray, ' ',(Tspw[i+1]-Tspw[i]):7:4);
```

```
  writeln(spray);
```

```
end;
```

```
var
```

```
  condm,Tr,Tl,incr,cond1 : double;
```

```
  ii,i,j,k,code1,error,step : integer;
```

```
  Choice,First            : boolean;
```

```
  answer,character        : char;
```

```
  istr                    : string;
```

```
(-----)
```

```
(* * * MAIN PROGRAM * * *)
```

```
begin
```

```
  GetDir(0,DefDir);
```

```
  IF copy(DefDir,length(DefDir),1)<>'\' THEN DefDir:=DefDir+'\';
```

```
  Start;
```

```
repeat;                ( * Start of main loop * )
```

```
  Input;
```

```
  IF (Inputfile)and(not Single) THEN
```

```
    begin
```

```
      i:=0;
```

```
      repeat
```

```
        inc(i);
```

```
        Str(i,is);
```

```
        assign(spray,DefDir+default+is+'.prn');
```

```
        {$I-}
```

```
        reset(spray);
```

```
        {$I+}
```

```
        error:=IOresult;
```

```
        if error=0 then close(spray);
```

```
      until error<>0;
```

```
      rewrite(spray);
```

```
      writeln(spray, ' ' + copy(Chosen,1,6));
```

```
    end;
```

```

spsl := spsl/1000.0;
spss := spss/1000.0;
Fsh := Fsh/1000.0;
Fss := Fss/1000.0;
Fst := Fst/1000.0;
Pist := Pist/1000.0;
window(1,1,80,25);
ClrScr;
ii := 0;
repeat;
  IF (not Single) THEN
    begin
      inc(ii);
      Str(ii,istr);
      readfile:=defdir+copy(Chosen,1,6)+istr+'.dat';
      assign(inpfile,readfile);
      {$I-}
      reset(inpfile);
      {$I+}
      error:=IOResult;
      OUTLINE('Multiple File Processing');
      WINDOWBOX(7,4,68,1,'');
      write(' Processing file '+copy(Chosen,1,6)+istr+'.dat: ');
    end
  ELSE IF InputFile THEN
    begin
      error := 0;
      writeln('                      RESULTS');
      writeln;
      write('In Temp ',Tspwi:7:4);
      FOR i:=Pirows DOWNTO 1 DO write(' dT',i:1,' ',dT[i]:6:3);
      writeln(' Out Temp ',Tspwoe:7:4);
    end
  ELSE
    begin
      error := 0;
      writeln('                      RESULTS');
      writeln;
      writeln('Process Water inlet Temp. ',Tspwi:7:4,' °C   Process water outlet Temp. ');
    end;
  if error=0 then    ( if file exists --> evaluate )
    begin
      IF (InputFile)and(not Single) THEN Readinpfile;

      cond1 := 0.0;
      First := True;
      Choice := False;
      QT := 0.0;
      MsdT := 0.0;
      step := 0;

      Initialcond;

      IF Finns THEN
        Tspw[1] := 0.35*Tdbsa[1]+0.65*Tspwi
      ELSE
        Tspw[1] := 0.1*Tdbsa[1]+0.9*Tspwi;
      IF Finns and Wet[1] THEN
        FOR i:=1 TO Pirows DO Tspiwall[i] := (0.3*Tspw[1]+0.7*Tdbsa[1])
      ELSE IF Wet[1] THEN
        FOR i:=1 TO Pirows DO Tspiwall[i] := (0.8*Tspw[1]+0.2*Tdbsa[1]);

      repeat
        FOR k := 1 TO Pirows DO
          begin
            FOR r := 1 TO Pirows DO
              begin
                Tspwa := Tspw[r];
                Tdbsa := Tdbsa[r];
                xxa := xx[r];

                Area;

```

```

IF Finns and Wet[1] THEN
  Tspiwall[r] := (0.3*Tspw[r]+0.7*Tdbsa[r])
ELSE IF Wet[1] THEN
  Tspiwall[r] := (0.8*Tspw[r]+0.2*Tdbsa[r]);
FOR j := 1 TO 3 DO
begin

  Constants;

  Tubewalltemp;

  IF (j=1)or(j=2) THEN
  begin
    xx[r+1] := xx[r]+Mev/smsa;
    Tdbsa[r+1] := Tdbsa[r]+Qsc/Cpa/smsa;
    Tspw[r+1] := Tspw[r]+Qst[r]/Cpwf(Tspw[r])/smspw;
    xxa := (xx[r+1]+xx[r])/2.0;
    Tdbsa := (Tdbsa[r+1]+Tdbsa[r])/2.0;
    Tspwa := (Tspw[r]+Tspw[r+1])/2.0;
  end;
end;

  Startcond;

end;
end;
IF Single and Inputfile THEN
begin
  write('      ',Tspw[r+1]:7:4);
  FOR i:=Pirows DOWNT0 1 DO write('      ',(Tspw[i+1]-Tspw[i]):6:3);
  writeln('      ',Tspw[1]:7:4);
end
ELSE IF Single THEN writeln('      ',Tspw[r+1]:7:4,'      ',Tspw[1]:7:4)
ELSE write('.');

condm := Tspw[r+1] - Tspwi;
IF (Abs(condm)>10.0) THEN
  Tspw[1] := Tspw[1]-condm/3.0
ELSE IF (Abs(condm)>0.0500) THEN
  Tspw[1] := Tspw[1]-condm/1.7
ELSE
begin
  IF (Abs((condm+cond1)/condm)<0.5) or (not First) THEN
  begin
    Tspw[1] := Tspw[1]-condm/10.0;
    IF not First THEN First := False;
  end
  ELSE Tspw[1] := Tspw[1]-condm/3.0;
  cond1 := condm;
end;
end;
Inc(step);
until (Abs(condm)<1.0E-3) or (step>16);
IF First THEN Tspw[1] := Tspw[1]+condm/3.0 ELSE Tspw[1] := Tspw[1]+condm/10.0;

FOR r := 1 TO Pirows DO QT := QT+Qst[r];
IF Up THEN MsdT := Msd[Pirows]*(Picol-0.5) ELSE MsdT := Msd[1]*(Picol-0.5);
xi := airhumidity(Tdbsa[i],Twbsa[i],Psa*1000);
xo := airhumidity(Tdbsa[Pirows+1],Twbsa[Pirows+1],Psa*1000);
Mevst := (xo-xi)*smsa*(Picol-0.5);

QT := QT*(Picol-0.5)/1000.0;
QTc := (Tspw[Pirows+1]-Tspw[1])*smspw*(Picol-0.5)*Cpwf((Tspw[Pirows+1]+Tspw[1])/2.0)/1000.0;
Qe := smspw*(Picol-0.5)*Cpwf((Tspwi+Tspwoe)/2.0)*(Tspwi-Tspwoe)/1000.0;

IF (Single)or(not Inputfile) THEN
begin
  Output;
  error := 1;
end
ELSE Save;
end;
end;

```

```
until (error<>0);
IF not Single THEN close(spray);
window(1,1,80,25);
ClrScr;
Item[1] := 'R - Return to Main Menu!';
Item[2] := 'Q - Quit!';
Title := '';
FirstPos := 1;
DISPLAYMENU(2,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    spsl := spsl*1000.0;
    spss := spss*1000.0;
    Fsh := Fsh*1000.0;
    Fss := Fss*1000.0;
    Fst := Fst*1000.0;
    Pist := Pist*1000.0;
    smsswi := smsswi*3600;
    smspw := smspw*(Picol-0.5);
    smsa := smsa*(Picol-0.5);
  end;
  2 : begin
    Choice := true;
  end;
end; { case }

until choice;
ClrScr;
end.
```

```

Program   : Wet.pas (Vertical air flow cooler - Enthalpy model)
Programmer : D.E. Kriel
Date      : 30/07/90

```

```
Program Wet;
```

```
{$N+,E+}           {use 8087 if present}
```

```
uses Crt,UTIL;
```

```
type
```

```
  mat = array [0..4,0..4] of double;
  col = array [0..50] of double;
```

```
var
```

```

Item                                     : StrArr;
Aw, isai, isao, Aa, Cpa, dI1, dI2, W, Ft,
ksa, xi, xo, Twbsai, Tspwi, smspw, sksf,
smsa, Pist, Prsa, Tspwoe, Feff, hxsl, Q,
Tswav, spsl, spss, Fsh, Fss, Fst, hspf,
cdsh, Tdbsaoe, Twbsaoe, Tsswi, Tdbsao,
skspiw, Pisl, Psa, U, Qe, densai, smsswi,
Tsaav, Qa, Twbsao, Tdbsai, Tspwo, hsd      : double;
r, Pirows, Picol, Selected, FirstPos      : integer;
InputFile                                  : boolean;
Default, DefDir, readfile, Title, Chosen  : string;
wethx, inpf                                : text;

```

```
(* * * Subroutine to initialize starting values * * *)
```

```
Procedure Start;
```

```
begin
```

```

Pirows      := 4;      { Number of pipe rows }
Picol       := 31;     { Number of pipe coloms }
Fsh         := 9.5;    { Fin height [mm]}
Fss        := 2.5;    { Fin spacing [mm]}
Fst         := 0.48;   { Fin thickness [mm]}
smspwi     := 10.0;   { Process water mass flow [kg/s]}
smsa       := 5.0;    { Air mass flow [kg/s]}
smsswi     := 400.0;  { Spray water mass flow [kg/m²h]}
Tspwi      := 50.0;   { Temperature of process water [°C]}
Tsswi      := 15.0;   { Temperature of spray water [°C]}
Tdbsai     := 25.0;   { Drybulb temperature of air [°C]}
Twbsai     := 19.5;   { Wetbulb temperature of air [°C]}
spsl       := 36.3;   { Pipe diameter - long axis [mm]}
spss       := 13.5;   { Pipe diameter - short axis [mm]}
skspiw     := 43.0;   { Thermal conductivity of pipe [W/m/°C]}
sksf       := 52.42;  { Thermal conductivity of fins [W/m/°C]}
Pisl       := 1.4;    { Pipe length [m]}
Pist       := 1.8;    { Pipe thickness [mm]}
W          := 1.1;    { Width of H.X. [m]}
hxsl       := 0.218;  { Length of H.X. [m]}
Psa        := 101.325; { Atmospheric Pressure [kPa]}
hsd        := 3000;   { Film heat transfer coefficient [W/m²/°C]}
default    := 'LMID'; { Name of output file}
end;
```

```
(* * * Subroutine to capture input data * * *)
```

```
Procedure Input;
```

```
var
```

```
  smspw0, smspw1 : double;
```

begin

```

repeat
  window(1,1,80,25);
  clrscr;
  IF InputFile THEN
    Item[1] := 'I - Input from data file..... '+Chosen
  ELSE
    Item[1] := 'I - User input data from keyboard';
  Item[2] := 'S - Cooler configuration: ';
  Item[3] := ' - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';
  Item[4] := ' - Number of Pipe coloms        = '+ITS(Picol,3)+' [-]';
  Item[5] := ' - Pipe length                   = '+RTS(Pisl,5,3)+' [m]';
  Item[6] := ' - Film heat transfer coeff.     = '+RTS(hsd,5,1)+' [W/m2/°C]';
  Item[7] := 'M - Massflow: ';
  Item[8] := ' - Massflow of Process water          = '+RTS(smspw,8,3)+' [kg/s]';
  Item[9] := ' - Mass flow of Air                   = '+RTS(smsa,8,3)+' [kg/s]';
  Item[10] := ' - Mass flow of Spray water             = '+RTS(smsswi,8,3)+' [kg/m3h]';
  Item[11] := 'T - Temperatures: ';
  Item[12] := ' - Temperature of Process water         = '+RTS(Tspwi,8,3)+' [°C]';
  Item[13] := ' - Temperature of Spray water          = '+RTS(Tsswi,8,3)+' [°C]';
  Item[14] := ' - Drybulb Temperature of air          = '+RTS(Tdbsai,8,3)+' [°C]';
  Item[15] := ' - Wetbulb Temperature of air          = '+RTS(Twbsai,8,3)+' [°C]';
  Item[16] := 'P - Absolute air Pressure              = '+RTS(Psa,8,3)+' [kPa]';
  Item[17] := 'O - Output file name is .....'+default;
  Item[18] := 'C - Continue';

  Title := 'VERTICAL AIR COOLED H.X.';
  FirstPos:= 18;
  DISPLAYMENU(18,FirstPos,Item,Title,Selected);
  CASE Selected OF
    1 : begin
      ClrScr;
      IF InputFile THEN
        InputFile := false
      ELSE
        begin
          CHOOSEFILE(DefDir,'*.DAT',Chosen);
          if Chosen='' then InputFile:=False else InputFile:=True;
          readfile:=DefDir+Chosen;
        end;
      IF InputFile THEN
        begin
          assign(inpf,readfile);
          reset(inpf);
          read(inpf,Psa);
          read(inpf,Tspwi);
          read(inpf,Tspwoe);
          read(inpf,Tsswi);
          read(inpf,Tdbsai);
          read(inpf,Twbsai);
          read(inpf,Tdbsaoe);
          read(inpf,Twbsaoe);
          read(inpf,smsa);
          read(inpf,smspw0);
          read(inpf,smspw1);
          read(inpf,smsswi);
          IF smspw0>smspw1 THEN smspw := smspw0 ELSE smspw := smspw1;
          Psa := Psa/1000.0;
          smsswi := smsswi*3600/(Pisl*W);
          close(inpf);
        end;
      end;
    2 : begin
      repeat;
        window(1,1,80,25);
        ClrScr;
        Title := 'COOLER CONFIGURATION';
        FirstPos := 5;
        Item[1] := 'R - Number of Pipe rows           = '+ITS(Pirows,3)+' [-]';
        Item[2] := 'P - Number of Pipe coloms        = '+ITS(Picol,3)+' [-]';
        Item[3] := 'L - Pipe length                   = '+RTS(Pisl,5,3)+' [m]';

```

```

Item[4] := 'H - Film Heat transfer coefficient = '+RTS(hsd,5,1)+' [W/m2/°C]';
Item[5] := 'C - Continue';
DISPLAYMENU(5,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the number of Pipe rows ');
      Pirows:=readint;
      until (Pirows>=0) and (Pirows<=100);
    end;
  2 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the number of Pipe coloms ');
      Picol:=readint;
      until (Picol>=0) and (Picol<=100);
    end;
  3 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the length of the Pipes [m] ');
      Pisl:=readreal;
      until (Pisl<5) and (Pisl>0);
    end;
  4 : begin
    repeat
      WINDOWBOX(10,18,60,1,'');
      write(' Give the film heat transfer coefficient [W/m2/°C] ');
      hsd:=readreal;
      until (hsd<10000) and (hsd>100);
    end;
  end;
until Selected=5;
end;
3 : begin
  repeat;
  window(1,1,80,25);
  ClrScr;
  Title := 'MASSFLOW';
  FirstPos := 4;
  Item[1] := 'P - Massflow of Process water = '+RTS(smspw,8,3)+' [kg/s]';
  Item[2] := 'A - Mass flow of Air = '+RTS(smsa,8,3)+' [kg/s]';
  Item[3] := 'S - Massflow of Spray water = '+RTS(smsswi,8,3)+' [kg/m2h]';
  Item[4] := 'C - Continue';
  DISPLAYMENU(4,FirstPos,Item,Title,Selected);
  CASE Selected OF
    1 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflow of Process water [kg/s] ');
        smspw:=readreal;
        until (smspw<100) and (smspw>0);
      end;
    2 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflow of Air [kg/s] ');
        smsa:=readreal;
        until (smsa>=0) and (smsa<100);
      end;
    3 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the massflow of Spray water [kg/m2h] ');
        smsswi:=readreal;
        until (smsswi<500) and (smsswi>0);
      end;
    end;
  until Selected=4;
end;
4 : begin

```



```

repeat;
  window(1,1,80,25);
  clrscr;
  Title := 'TEMPERATURES ';
  FirstPos := 5;
  Item[1] := 'P - Temperature of Process water = '+RTS(Tspwi,8,3)+' [°C] ' ;
  Item[2] := 'S - Temperature of Spray water = '+RTS(Tsswi,8,3)+' [°C] ' ;
  Item[3] := 'D - Drybulb Temperature of air = '+RTS(Tdbsai,8,3)+' [°C] ' ;
  Item[4] := 'W - Wetbulb Temperature of air = '+RTS(Twbsai,8,3)+' [°C] ' ;
  Item[5] := 'C - Continue';
  DISPLAYMENU(5,FirstPos,Item,Title,Selected);
  CASE Selected OF
    1 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the temperature of Process water [°C] ');
        Tspwi:=readreal;
        until (Tspwi<80) and (Tspwi>20);
      end;
    2 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the temperature of Spray water [°C] ');
        Tsswi:=readreal;
        until (Tsswi<40) and (Tsswi>0);
      end;
    3 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the drybulb temperature of the Air [°C] ');
        Tdbsai:=readreal;
        until (Tdbsai<50) and (Tdbsai>-10);
      end;
    4 : begin
      repeat
        WINDOWBOX(10,18,60,1,'');
        write(' Give the wetbulb temperature of the Air [°C] ');
        Twbsai:=readreal;
        until (Twbsai>=-10) and (Twbsai<=50);
      end;
    end;
  until Selected=5;
end;
5 : begin
  repeat
    WINDOWBOX(15,23,50,1,'');
    write(' Give the Air pressure [kPa] ');
    Psa:=readreal;
    until (Psa<103) and (Psa>97);
  end;
6 : begin
  WINDOWBOX(15,23,50,1,'');
  write(' Give the new file name ... ');
  default:=Readfilename;
end;
end; {case}
until Selected=7;
end;

(* * * Calculation of the starting conditions and constants* * *)

var
  Afr,Tni,smsswi,
  isaii,Ac,Ast,Asf,Awav : Double;

Procedure Initial;

var
  Ist,C,xii,cond,
  Isti,Tl,Tr : Double;

begin

```

```

Aw := 2.0*Pi*sqrt(0.5*(sqrt(spsl/2.0-Pist)+sqrt(spss/2.0-Pist))*Pisl*(Picol-0.5)*Pirows;
Awav := 2.0*Pi*sqrt(0.5*(sqrt((spsl-Pist)/2.0)+sqrt((spss-Pist)/2.0))*Pisl*(Picol-0.5)*Pirows;
Ast := 2.0*Pi*sqrt(0.5*(sqrt(spsl/2.0)+sqrt(spss/2.0))*Pisl*(Picol-1)*Pirows;
Asf := 2.0*Pi*((spsl/2.0+Fsh)*(spss/2.0+Fsh)-(spsl/2.0)*(spss/2.0))*Int(Pisl/Fss)*(Picol-1)*Pirows;
Aa := Ast+Asf;
Afr := W*Pisl;
Ac := Afr-((Picol-1)*Pisl*spss)-(Picol-1)*(Pisl/Fss*Fsh*Fst);
cdsh := 4.0*(Ac/Afr)*hxsl;
xi := airhumidity(Tdbsai,Twbsai,Psa*1000);
isai := enthalpy(Tdbsai,Twbsai,Psa*1000)*1000.0;

Ist := smsa*isai+smsswi*Cpwf(Tsswi)*(Tsswi-0.01);
IF Tsswi>Twbsai THEN
  begin
    IF Tsswi>Tdbsai THEN
      begin
        Tl := Twbsai;
        Tr := Tsswi;
      end
    ELSE
      begin
        Tl := Twbsai;
        Tr := Tdbsai;
      end;
    end
  end
ELSE
  begin
    Tl := Tsswi;
    Tr := Tdbsai;
  end;
end;
repeat
  Tni := (Tl+Tr)/2.0;
  xii := Airhumidity(Tni,Tni,Psa*1000.0);
  smsswii := smsswi-smsa*(xii-xi);
  isaii := Enthalpy(Tni,Tni,Psa*1000.0)*1000.0;
  Isti := smsa*isaii+smsswii*Cpwf(Tni)*(Tni-0.01);
  cond := Isti-Ist;
  IF cond>0.0 THEN Tr:=Tni ELSE Tl:=Tni;
until Abs(cond)<0.1;
end;

(* * * Calculation of the starting conditions and constants* * *)

Procedure Constants;

var
  fd,Resw,de,Prsw,ass,ha,
  ksw,Transfp,Resa,hes,bs : double;

begin
  Cpa := Cpaf(Tsaav);
  ksa := Airconductivity(Tsaav);
  Prsa := AirPrandtl(Tsaav);
  Transfp := 1000.0*(Enthalpy(Tspwi,Tspwi,Psa*1000.0)-Enthalpy(Tdbsai,Tdbsai,Psa*1000.0))
    /(Cpa*(Tspwi-Tdbsai));
  Resa := cdsh*smsa/Ac/Airviscosity(Tsaav);
  ha := 4.82262*P(Resa,-0.526995)/P(Prsa,(2.0/3.0))*Cpa*smsa/Ac;
  hes := 1/(1/hsd+1/(ha*Transfp));
  bs := sqrt(2.0*hes/(sksf*Fst))*Fsh;
  Feff := ((Exp(bs)-Exp(-1*bs))/(Exp(bs)+Exp(-1*bs)))/bs;
  ass := (Ast+Asf*Feff)/Aa;
  de := 4.0*(spsl/2.0-Pist)*(spss/2.0-Pist)/(2.0*sqrt(0.5*((sqrt(spsl/2.0-Pist)+sqrt(spss/2.0-Pist)))));
  Resw := de*smspw/(WaterViscosity(Tswav)*Pi*(spsl/2.0-Pist)*(spss/2.0-Pist)*Picol);
  fd := P((1.82*Log10(Resw)-1.64),-2);
  Prsw := WaterPrandtl(Tswav);
  ksw := Waterconductivity(Tswav);
  hspf := ksw/de*(fd/8.0*(Resw-1000)*Prsw*(1+P(de/Pisl,0.67)))
    /(1.0+12.7*sqrt(fd/8)*(P(Prsw,0.67)-1.0));
  U := 1/(Transfp*(Aa/(hspf*Aw)+Pist*Aa/(sksf*Aw)+1/(hes*ass)));
end;

```

```
(* * * Subroutine to find the geometry coefficient * * *)
```

```
Procedure Coeff;
```

```
var
```

```
  a           : mat;
  phish,phisl,phiscf : double;
  i,k         : integer;
```

```
begin
```

```
  a[1,1] := -0.339;
  a[2,1] := 0.0277;
  a[3,1] := 0.179;
  a[4,1] := -0.0199;
  a[1,2] := 2.38;
  a[2,2] := -0.0999;
  a[3,2] := -1.21;
  a[4,2] := 0.04;
  a[1,3] := -5.26;
  a[2,3] := 0.0904;
  a[3,3] := 2.62;
  a[4,3] := 0.0494;
  a[1,4] := 3.9;
  a[2,4] := -0.000845;
  a[3,4] := -1.81;
  a[4,4] := -0.0981;
```

```
  phish := (Tspwi-Tspwo)/(Tspwi-Tni);
  phisl := (Tdbso-Tni)/(Tspwi-Tni);
  phiscf := (phish-phisl)/(ln(Abs((1.0-phisl)/(1.0-phish))));
  Ft := 0.0;
```

```
  FOR i:=1 TO 4 DO
```

```
  begin
```

```
    FOR k:=1 TO 4 DO Ft := Ft+a[i,k]*P((1-phiscf),k)*Sin(2*i*(ArcTan(phish/phisl)));
```

```
  end;
```

```
  Ft := 1-Ft;
```

```
  IF Ft>1.0 THEN Ft := 1.0;
```

```
end;
```

```
var
```

```
  LMID,ispwi,ispwo : double;
```

```
(* * * Subroutine to calculate the new heat transfer capacity * * *)
```

```
Procedure Calculate;
```

```
begin
```

```
  Tspwo := Tspwi-Qa/(smsspw*Cpwf(Tswav));
```

```
  Coeff;
```

```
  ispwi := Cpwf(Tspwi)*(Tspwi-0.01);
```

```
  ispwo := Cpwf(Tspwo)*(Tspwo-0.01);
```

```
  di1 := ispwi-isao;
```

```
  di2 := ispwo-isaii;
```

```
  LMID := (di2-di1)/(ln(Abs(di2/di1)));
```

```
  Qa := Ft*LMID*U*Aa/Cpa;
```

```
end;
```

```
var
```

```
  is           : string;
  swevap,smsswif : Double;
```

```
(* * * Subroutine to print the final results on the screen * * *)
```

```
Procedure Output;
```

begin

```

IF Inputfile THEN Qe := smspw*Cpwf((Tspwi+Tspwoe)/2.0)*(Tspwi-Tspwoe)/1000.0;
Q := Q/1000.0;
Qa := Qa/1000.0;
swevap := smsa*(xo-xi);
smsswi := smsswi*3600.0/(W*Pisl);
IF Inputfile THEN OUTLINE(' RESULTS OF '+chosen) ELSE OUTLINE(' RESULTS');
writeln;
writeln('      For Vertical Air Flow Cooler with Spray density of ',smsswi:6:2,' kg/m^h');
writeln;
writeln(' Inlet Temperature of the Process water is      : ',Tspwi:7:4,' °C');
IF Inputfile THEN
  writeln(' Outlet Temperature of the Process water is    : ',Tspwo:7:4,' (',Tspwoe:7:4,') °C ')
ELSE
  writeln(' Outlet Temperature of the Process water is    : ',Tspwo:7:4,' °C');
writeln(' Inlet Temperature of the Spray water is          : ',Tsswi:7:4,' °C');
writeln;
IF Inputfile THEN
  writeln(' Capacity of the cooler is (cal. LMID)              : ',Q:7:2,' (',Qe:7:2,') kW ')
ELSE
  writeln(' Capacity of the cooler is (cal. LMID)              : ',Q:7:2,' kW');
writeln(' Capacity of the cooler is (cal. Air enthalpy): ',Qa:7:2,' kW');
writeln;
writeln(' Mass flow of the air is                            : ',smsa:6:3,' kg/s');
writeln(' Inlet condition of the Air is                      : ',Tdb sai:6:3,' °C (Drybulb)');
writeln('                                                    : ',Twb sai:6:3,' °C (Wetbulb)');
IF Inputfile THEN
begin
  writeln(' Outlet condition of the Air is                    : ',Tdb sai:6:3,
    ' (',Tdb sae:6:3,') °C (Drybulb)');
  writeln('                                                    : ',Twb sai:6:3,
    ' (',Twb sae:6:3,') °C (Wetbulb)');
end
ELSE
begin
  writeln(' Outlet condition of the Air is                    : ',Tdb sai:6:3,' °C (Drybulb)');
  writeln('                                                    : ',Twb sai:6:3,' °C (Wetbulb)');
end;
writeln;
writeln(' Amount of water on heat exchanger is              : ',smsswi:6:5,' kg/s');
writeln(' Amount of water that evaporates is                : ',swevap:6:5,' kg/s');
WINDOWBOX(25,22,30,1,'');
write(' Press spacebar to continue');
end;

```

```
(* * * Subroutine to write the final results to file * * *)
```

Procedure Save;

begin

```

assign(wethx,DefDir+default+is+'.out');
rewrite(wethx);
writeln(wethx);
IF Inputfile THEN writeln(wethx,'          THE RESULTS '+chosen)
ELSE writeln(wethx,'          THE RESULTS');
writeln(wethx,'          -----');
writeln(wethx);
writeln(wethx);
writeln(wethx,'      For Vertical Air Flow Cooler with Spray density of ',smsswi:6:2,' kg/m^h');
writeln(wethx);
writeln(wethx,' Inlet Temperature of the Process water is      : ',Tspwi:7:4,' °C');
IF Inputfile THEN
  writeln(wethx,' Outlet Temperature of the Process water is    : ',Tspwo:7:4,' °C (',Tspwoe:7:4,')')
ELSE
  writeln(wethx,' Outlet Temperature of the Process water is    : ',Tspwo:7:4,' °C');
writeln(wethx,' Inlet Temperature of the Spray water is          : ',Tsswi:7:4,' °C');
writeln(wethx);
IF Inputfile THEN
  writeln(wethx,' Capacity of the cooler is (cal. LMID)          : ',Q:7:2,' kW (',Qe:7:2,')')
ELSE

```

```

    writeln(wethx,' Capacity of the cooler is (cal. LMID)           : ',Q:7:2,' kW');
    writeln(wethx,' Capacity of the cooler is (cal. Air enthalpy): ',Qa:7:2,' kW');
    writeln(wethx);
    writeln(wethx,' Mass flow of the air is                       : ',smsa:6:3,' kg/s');
    writeln(wethx,' Inlet condition of the Air is                          : ',Tdb sai:6:3,' °C (Drybulb)');
    writeln(wethx,'                               : ',Tws ai:6:3,' °C (Wetbulb)');
    IF Inputfile THEN
    begin
        writeln(wethx,' Outlet condition of the Air is          : ',Tdb sao:6:3,
            ' °C (Drybulb) ('',Tdb saoe:6:3,'')');
        writeln(wethx,'                               : ',Tws sao:6:3,
            ' °C (Wetbulb) ('',Tws saoe:6:3,'')');
    end
    ELSE
    begin
        writeln(wethx,' Outlet condition of the Air is          : ',Tdb sao:6:3,' °C (Drybulb)');
        writeln(wethx,'                               : ',Tws sao:6:3,' °C (Wetbulb)');
    end;
    writeln(wethx);
    writeln(wethx,' Amount of water on heat exchanger is          : ',smsswi:6:5,' kg/s');
    writeln(wethx,' Amount of water that evaporates is              : ',swevap:6:5,' kg/s');
    close(wethx);
end;

var
    xo2,dt,cond,Tr,
    incr,Tl2,Tr2,Tl    : double;
    i,j,k,error        : integer;
    choice,change      : boolean;
    answer,character    : char;

```

```
(* * * MAIN PROGRAM * * *)
```

```

begin
    GetDir(0,DefDir);
    IF copy(DefDir,length(DefDir),1)<>'\' THEN DefDir:=DefDir+'\'
    Start;

    repeat
        ( * Start of main loop * )

        InputFile := False;
        Input;

        i:=0;
        repeat
            inc(i);
            Str(i,is);
            assign(wethx,DefDir+default+is+'.out');
            {$I-}
            reset(wethx);
            {$I+}
            error:=IOresult;
            if error=0 then close(wethx);
        until error<>0;

        window(1,1,80,25);
        ClrScr;
        choice := false;
        spsl := spsl/1000.0;
        spss := spss/1000.0;
        Pist := Pist/1000.0;
        Fsh := Fsh/1000.0;
        Fss := Fss/1000.0;
        Fst := Fst/1000.0;
        smsswi := smsswi*W*Pisl/3600.0;
        Isaav := (Tws ai+Tspwi)/2.0;
        Tswav := Isaav;

        Initial;

```

```

FOR i:=1 TO 2 DO
begin
  Tl := Tni;
  Tr := Tspwi;
  Constants;

  FOR r:=1 TO 4 DO
  begin
    incr := Tr-Tl;
    change := false;
    FOR k:=1 TO 19 DO
    begin
      IF not Change THEN
      begin
        Twbsao := Tl+k*incr/20.0;
        Tdbsao := Twbsao;
        isao := Enthalpy(Tdbsao,Twbsao,Psa*1000.0)*1000.0;
        Qa := smsa*(isao-isaii);
        Q := Qa;

        Calculate;

        cond := Q-Qa;
        IF cond>0.0 THEN
        begin
          Tr2 := Twbsao;
          Tl2 := Twbsao-incr/20.0;
          Change := True;
        end;
      end;
    end;
    IF Change THEN
    begin
      Tl := Tl2;
      Tr := Tr2;
    end
    ELSE IF (not Change) AND (r<4) THEN
    begin
      Tl := Tl-incr;
      Tr := Tr+incr;
      r := r-1;
    end
    ELSE Tl := Tr;
  end;
  Twbsao := (Tl+Tr)/2.0;
  Tdbsao := Twbsao;
  Tsaav := (Tni+Twbsao)/2.0;
  Tswav := (Tspwi+Tspwo)/2.0;
end;

xo := Airhumidity(Tdbsao,Twbsao,Psa*1000.0);

isao := enthalpy(Tdbsao,Twbsao,Psa*1000)*1000.0;
Qa := smsa*(isao-isaii);
Tspwo := Tspwi-Qa/(smspw*Cpwf(Tswav));

Coeff;

ispwi := Cpwf(Tspwi)*(Tspwi-0.01);
ispwo := Cpwf(Tspwo)*(Tspwo-0.01);
dI1 := ispw-i-sao;
dI2 := ispw-i-saii;
LMID := (dI2-dI1)/(ln(Abs(dI2/dI1)));
Q := Ft*LMID*U*Aa/Cpa;

Output;

repeat
  until Keypressed;

k := 0;

```

```
window(1,1,80,25);
ClrScr;
Item[1] := 'S - Save results and continue';
Item[2] := 'C - Continue without saving the results';
Item[3] := 'R - Save results and quit';
Item[4] := 'Q - Quit without saving the results';
Title := '';
FirstPos := 1;
DISPLAYMENU(4,FirstPos,Item,Title,Selected);
CASE Selected OF
  1 : begin
    Save;
    choice := false;
    spsl := spsl*1000.0;
    spss := spss*1000.0;
    Pist := Pist*1000.0;
    Fsh := Fsh*1000.0;
    Fss := Fss*1000.0;
    Fst := Fst*1000.0;
    smsswi := smsswi*3600/(W*Pisl);
  end;
  2 : begin
    choice := false;
    spsl := spsl*1000.0;
    spss := spss*1000.0;
    Pist := Pist*1000.0;
    Fsh := Fsh*1000.0;
    Fss := Fss*1000.0;
    Fst := Fst*1000.0;
    smsswi := smsswi*3600/(W*Pisl);
  end;
  3 : begin
    Save;
    choice := true;
  end;
  4 : choice := true;
end; { case }

until choice;

end.
```

```

Program   : UTIL.PAS - Unit for SPRAY PROGRAMS
Programmer : A.A.Dreyer/D.E.Kriel
Date      : 11/06/1990

```

```

unit UTIL;
interface
( Global variables )
type
  str1    = string[1];
  str10   = string[10];
  str80   = string[80];
  str150  = string[150];
  StrArr  = array[1..20] of string[60];
  BoxRec  = record
    UL,   UR   : char;
    LL,   LR   : char;
    Horiz, Vert : char;
  end;
const
  SingleBox : BoxRec = (UL   : '┌'; UR   : '┐';
                       LL   : '└'; LR   : '┘';
                       Horiz : '-'; Vert : '|');
  DoubleBox : BoxRec = (UL   : '┌'; UR   : '┐';
                       LL   : '└'; LR   : '┘';
                       Horiz : '-'; Vert : '|');

( Global procedures and functions )

function RTS(x:double;width,decimals:integer):string;
function ITS(x,width:integer):string;
procedure DISPLAYMENU(NumEntr,FirstPos:integer;Item:StrArr;Title:string;var Selected:integer);
procedure CHOOSEFILE(var Dir:string;Mask:string;var Chosen:string);
function Readfilename:string;
function READINT:integer;
function READREAL:double;
procedure DRAWBOX(X,Y,Width,Height,BorderAtt,TitleAtt:integer;title:str80);
procedure WINDOWBOX(x1,y1,width,height:integer;title:str80);
procedure OUTLINE(title:str80);
function P(aa,bb:double):double;
function Htan(aa:double):double;
function LOG10(aa:double):double;
function ATPRESSURE(h,t:double):double;
function SATVAPPRES(t:double):double;
function CPVF(t:double):double;
function CPAF(t:double):double;
function CPWF(t:double):double;
function AIRHUMIDITY(t1,t2,patm:double):double;
function SATAIRHUMIDITY(t1,patm:double):double;
function ENTHALPY(t1,t2,patm:double):double;
function SATENTHALPY(t1,patm:double):double;
function AIRVISCOSITY(t:double):double;
function WATERVISCOSITY(t:double):double;
function VAPOURVISCOSITY(t:double):double;
function AIRVAPMIXVISCOSITY(t,wa,patm:double):double;
function WATERDENSITY(t:double):double;
function HFGWATER(t:double):double;
function AIRDENSITY(t,patm:double):double;
function AIRVAPMIXDENSITY(t,wa,patm:double):double;
function AIRCONDUCTIVITY(t:double):double;
function WATERCONDUCTIVITY(t:double):double;
function AIRPRANDTL(t:double):double;
function WATERPRANDTL(t:double):double;
implementation

uses Crt,Dos,Graph;

( Function to convert a double number into its string representation )

```



```

function RTS(x:double;width,decimals:integer):string;
var s : string;
begin
  Str(x:width:decimals,s);
  RTS:=s
end; { of function RTS }

{ Function to convert an integer number into its string representation }
function ITS(x,width:integer):string;
var s : string;
begin
  Str(x:width,s);
  ITS:=s
end; { of function ITS }

{ Procedure to display a standard menu }
procedure DISPLAYMENU(NumEntr,FirstPos:integer;Item:StrArr;Title:string;var Selected:integer);
var
  xsize,yysize,y1,y2,i,j,x1,x2 : integer;
  Position,OldPosition,OldAttr : integer;
  NumSkip,NumDead,MaxLength : integer;
  SpecialKey,DirectChoice : boolean;
  DeadOption : array[1..20] of integer;
  FirstChars : array[1..20] of char;
  BlankString : array[1..20] of string[60];
  Character : char;

{ Procedure to determine which options are not valid }
procedure SearchDeadOptions;
var i,j,NumBlanks : integer;
begin
  NumDead:=0;
  for i:=1 to NumEntr do
  begin
    if Item[i][1]=' ' then
    begin
      inc(NumDead);
      DeadOption[NumDead]:=i;
    end;
    NumBlanks:=(MaxLength-length(Item[i]))-2;
    BlankString[i]:='';
    for j:=1 to NumBlanks do BlankString[i]:=BlankString[i]+' ';
  end;
end; { of procedure SearchDeadOptions }

{ procedure to write the firstcharacter of a string in a bright colour }
procedure WriteString(MainStr:string);
var
  First : string[1];
  Rest : string;
begin
  First:=Copy(MainStr,1,1);
  Rest:=Copy(MainStr,2,length(MainStr));
  HighVideo;
  write(First);
  LowVideo;
  write(Rest);
end; { of procedure WriteString }

{ Procedure to determine highlight each selected option }
procedure Highlight;
begin
  { Clear previous position }
  GotoXY(x1+2,y1+OldPosition);
  TextAttr:=OldAttr;
  WriteString(Item[OldPosition]+BlankString[OldPosition]);
  { Highlight new position }
  GotoXY(x1+2,y1+Position);
  TextAttr:=2*16+7;
  WriteString(Item[Position]+BlankString[Position]);

```

```

    TextAttr:=OldAttr;
    GotoXY(x1+2,y1+Position);
end; ( of procedure HighLight )

( Function to determine if a given position is valid or not )
function In_Dead_Array(i:integer):boolean;
var j : integer;
begin
    In_Dead_Array:=False;
    for j:=1 to NumDead do
        if (i = DeadOption[j]) then In_Dead_Array:=True;
    end; ( of function In_Dead_Array )

( Function to determine if a given character is valid )
function In_Valid_Array(cc:char):boolean;
var i : integer;
begin
    In_Valid_Array:=False;
    for i:=1 to NumEntr do
        if (UpCase(Item[i][1])=cc) then In_Valid_Array:=True;
    end; ( of function In_Valid_Array )

( Main procedure )
begin
    OldAttr:=TextAttr;
    TextAttr:=10;
    ( Draw a block around the entries )
    xsize:=Lo(WindMax)-Lo(WindMin);
    ysize:=Hi(WindMax)-Hi(WindMin);
    y1:=(ysize-NumEntr) div 4;
    y2:=y1+NumEntr+1;
    MaxLength:=0;
    for i:=1 to NumEntr do
        if (length(Item[i])>MaxLength) then MaxLength:=length(Item[i]);
    MaxLength:=MaxLength+2;
    x1:=(xsize-MaxLength) div 2;
    x2:=x1+MaxLength+1;
    with DoubleBox do
    begin
        GotoXY(x1,y1);
        write(UL);
        for i:=x1+1 to x2-1 do write(Horiz);
        write(UR);
        for i:=y1+1 to y2-1 do
            begin
                GotoXY(x1,i);write(Vert);
                GotoXY(x2,i);write(Vert);
            end;
        GotoXY(x1,y2);
        write(LL);
        for i:=x1+1 to x2-1 do write(Horiz);
        write(LR);
    end; ( with )
    ( Center title on top of box )
    if Title <> '' then
    begin
        GotoXY(x1 + ((x2-x1)-length(Title)) div 2, y1);
        TextAttr := 31;
        write(' ', Title, ' ');
    end;
    TextAttr:=OldAttr;
    ( Write the entries inside the block )
    for i:=1 to NumEntr do
    begin
        GotoXY(x1+1,y1+i);
        write(' ');
        WriteString(Item[i]);
    end;
    ( Search options which are regarded as dead )
    SearchDeadOptions;
    ( Check if the default position is valid )
    if In_Dead_Array(FirstPos) then

```

```

begin
  i:=0;
  repeat
    inc(i);
  until (i=NumEntr) or (not In_Dead_Array(i));
  FirstPos:=i;
end;
{ Set up and display first position }
Position:=FirstPos;
OldPosition:=Position;
HIGHLIGHT;
repeat
  OldPosition:=Position;
  SpecialKey:=False;
  repeat
    Character:=UpCase(ReadKey);
  until ((Character in [#0,#13]) or In_Valid_Array(Character));
  if Character=#0 then
    begin
      SpecialKey:=True;
      Character:=UpCase(ReadKey);
      case Character of
        #72 : Position:=Position-1;   { Up arrow }
        #80 : Position:=Position+1;   { Down arrow }
      end; { case }
    end;
    DirectChoice:=In_Valid_Array(Character) and (not SpecialKey);
    while In_Dead_Array(Position) do
      begin
        case Character of
          #72 : Position:=Position-1;   { Up arrow }
          #80 : Position:=Position+1;   { Down arrow }
        end; { case }
      end;
      if (Position>NumEntr) then      { scrolling past the bottom }
        begin
          i:=0;
          repeat
            inc(i);
          until (i=NumEntr) or (not In_Dead_Array(i));
          Position:=i;
        end;
      if (Position<1) then           { scrolling past the top }
        begin
          i:=NumEntr+1;
          repeat
            dec(i);
          until (i=0) or (not In_Dead_Array(i));
          Position:=i;
        end;
      HIGHLIGHT;
    until (Character=#13) or DirectChoice;
    { Determine the selected number }
    if DirectChoice then
      begin
        NumSkip:=0;
        for i:=1 to NumEntr do
          begin
            if In_Dead_Array(i) then inc(NumSkip);
            if Character=UpCase(Item[i][1]) then Selected:=i-NumSkip;
          end;
        end
      else
        begin
          NumSkip:=0;
          for i:=1 to Position do
            if In_Dead_Array(i) then inc(NumSkip);
          Selected:=Position-NumSkip;
        end;
        TextAttr:=OldAttr;
      end; { of procedure DISPLAYMENU }

```

```

{ Procedure to choose a file from a given directory }
procedure CHOOSEFILE(var Dir:string;Mask:string;var Chosen:string);
type
  BoxRec = record
    UL,   UR   : char;
    LL,   LR   : char;
    Horiz, Vert : char;
  end;
const
  MaxNumFiles = 400;  { Maximum number of files }
  MaxCol = 4;        { Number of columns }
  FirstLine = 4;     { Position of window from top of page }
  MaxRowDisplay = 16; { Maximum number of lines in the window }
  MaxPerPage=MaxRowDisplay*MaxCol;
  MaxPages=(MaxNumFiles div MaxPerPage)+1;
  DoubleBox : BoxRec = (UL   : '┌'; UR   : '┐';
                        LL   : '└'; LR   : '┘';
                        Horiz : '-'; Vert : '|');
var
  x1,y1           : integer;
  OldRow,OldCol   : integer;
  Row,Col,NumFiles : integer;
  Page            : integer;
  NormalAttr,TitleAttr : integer;
  MenuAttr,BrightAttr : integer;
  HighlightAttr,OldAttr : integer;
  FileName        : array[1..MaxNumFiles] of string[12];

{ Procedure to turn cursor off }
procedure CURSOROFF;
var Regs : Registers;
begin
  with Regs do
    begin
      AH:=$01;
      CH:=$20;
      Intr($10,Regs);
    end;
end; { of procedure CURSOROFF }

{ Procedure to turn cursor on }
procedure CURSORON;
var Regs : Registers;
begin
  with Regs do
    begin
      AH:=$01;
      CH:=6;
      CL:=7;
      Intr($10,Regs);
    end;
end; { of procedure CURSORON }

{ Procedure to read all the filenames/directories in a directory }
procedure READDIR(Dir,Mask:string);
var
  DirInfo : SearchRec;
  LowerDir : boolean;
begin
  { Find all the normal files in the specified directory }
  FindFirst(Dir+Mask,0,DirInfo);
  NumFiles:=0;
  while (DosError=0) do
    begin
      inc(NumFiles);
      FileName[NumFiles]:=DirInfo.Name;
      FindNext(DirInfo);
    end;
  { Find all the subdirectories in the specified directory }
  LowerDir:=False;
  FindFirst(Dir+'*.*',Directory,DirInfo);
  while DosError=0 do

```

```

begin
  if ((DirInfo.Name<>'.' and (DirInfo.Attr=16)) then
    begin
      if (DirInfo.Name[1]='.') then LowerDir:=True
      else
        begin
          inc(NumFiles);
          FileName[NumFiles]:=DirInfo.Name+'\';
        end;
      end;
      FindNext(DirInfo);
    end; ( while )
    if LowerDir then
      begin
        inc(NumFiles);
        FileName[NumFiles]:='..\';
      end;
    end; ( of procedure READDIR )

( Procedure to determine the blank string to fill a file name to 12 chars )
function BLANKSTRING(TempFile:string):string;
var
  j,NumBlanks : integer;
  Blank       : string;
begin
  NumBlanks:=(12-length(TempFile))+1;
  Blank:='';
  for j:=1 to NumBlanks do Blank:=Blank+' ';
  BlankString:=Blank;
end; ( of function BLANKSTRING )

( Procedure to determine highlight the currently selected file )
procedure HIGHLIGHT;
var OldNumber,Number : integer;
begin
  ( Clear previous position )
  OldNumber:=(Page-1)*MaxPerPage+MaxCol*(OldRow-1)+OldCol;
  GotoXY(x1+1+(OldCol-1)*14,y1+OldRow);
  TextAttr:=MenuAttr;
  write(' ',FileName[OldNumber]+BLANKSTRING(FileName[OldNumber]));
  ( Highlight new position )
  Number:=(Page-1)*MaxPerPage+MaxCol*(Row-1)+Col;
  GotoXY(x1+1+(Col-1)*14,y1+Row);
  TextAttr:=HighLightAttr;      (2*16+7)
  write(' ',FileName[Number]+BLANKSTRING(FileName[Number]));
  TextAttr:=MenuAttr;
end; ( of procedure HIGHLIGHT )

( Main procedure )
var
  Character           : char;
  Title,PageStr      : string;
  SpecialKey,ChangePage : boolean;
  i,j,k,NumFilesLastPage : integer;
  xsize,ysize,x2,y2  : integer;
  MaxLength,CurrentNumFiles : integer;
  MaxRow,NumPages     : integer;
  FilePage           : array[1..MaxPages,1..MaxPerPage] of string[12];
begin
  OldAttr:=TextAttr;
  NormalAttr:=12;
  TitleAttr:=120;
  MenuAttr:=120;
  HighlightAttr:=15;
  BrightAttr:=14;
  CURSOROFF;
  repeat
    GotoXY(1,25);
    TextAttr:=BrightAttr;
    write(' < Use arrow keys and RETURN to select a result file (ESC → no file) >');
    TextAttr:=NormalAttr;

```

```

READDIR(Dir,Mask);
if (NumFiles=0) then      ( empty directory )
begin
  CurrentNumFiles:=1;
  TextAttr:=MenuAttr;
  ( draw a box around the entries )
  ysize:=Hi(WindMax)-Hi(WindMin);
  y1:=FirstLine;
  y2:=y1+2;
  xsize:=Lo(WindMax)-Lo(WindMin);
  MaxLength:=36;
  x1:=(xsize-MaxLength) div 2;
  x2:=x1+MaxLength+1;
  with DoubleBox do
  begin
    GotoXY(x1,y1);
    write(UL);
    for i:=x1+1 to x2-1 do write(Horiz);
    write(UR);
    for i:=y1+1 to y2-1 do
    begin
      GotoXY(x1,i);write(Vert);
      GotoXY(x2,i);write(Vert);
    end;
    GotoXY(x1,y2);
    write(LL);
    for i:=x1+1 to x2-1 do write(Horiz);
    write(LR);
  end; ( with )
  ( Center title on top of box )
  Title:=Dir+Mask;
  GotoXY(x1 + ((x2-x1)-length(Title)) div 2, y1);
  TextAttr := TitleAttr;
  write(' ', Title, ' ');
  TextAttr:=MenuAttr;
  GotoXY(x1+1,y1+1);
  write('      No matching files found      ');
  Chosen:='';
  repeat until KeyPressed;
end
else
begin
  Page:=1;
  NumPages:=(NumFiles div MaxPerPage)+1;
  ( Copy files into the two dimensional array for display )
  i:=1;
  j:=0;
  for k:=1 to NumFiles do
  begin
    inc(j);
    FilePage[i,j]:= FileName[k];
    if (k=(i*MaxPerPage)) then
    begin
      inc(i);
      j:=0;
    end;
  end;
  NumFilesLastPage:=j;
  OldRow:=1;
  OldCol:=1;
  repeat
    if (Page=NumPages) then CurrentNumFiles:=NumFilesLastPage
    else CurrentNumFiles:=MaxPerPage;
    ( Display files in window )
    TextAttr:=MenuAttr;
    ( draw a box around the entries )
    ysize:=Hi(WindMax)-Hi(WindMin);
    y1:=FirstLine;
    y2:=y1+(CurrentNumFiles div MaxCol)+2;
    if ((CurrentNumFiles mod MaxCol)=0) then y2:=y2-1;
    xsize:=Lo(WindMax)-Lo(WindMin);
    MaxLength:=MaxCol*12+(MaxCol-1)*2+2;

```

```

x1:=(xsize-MaxLength) div 2;
x2:=x1+MaxLength+1;
with DoubleBox do
begin
  GotoXY(x1,y1);
  write(UL);
  for i:=x1+1 to x2-1 do write(Horiz);
  write(UR);
  for i:=y1+1 to y2-1 do
  begin
    GotoXY(x1,i);write(Vert);
    GotoXY(x2,i);write(Vert);
  end;
  GotoXY(x1,y2);
  write(LL);
  for i:=x1+1 to x2-1 do write(Horiz);
  write(LR);
end; { with }
{ Center title on top of box }
if (NumPages>1) then
  Title:=Dir+Mask+' (Page '+ITS(Page,1)+' of '+ITS(NumPages,1)+' )'
else
  Title:=Dir+Mask;
GotoXY(x1 + ((x2-x1)-length(Title)) div 2, y1);
TextAttr := TitleAttr;
write(' ', Title, ' ');
TextAttr:=MenuAttr;
{ Center page number at bottom of box }
if (NumPages<>1) then
begin
  if (Page>1) then
  begin
    if (Page=NumPages) then
      PageStr:='More files above'
    else
      PageStr:='More files above and below';
  end
  else PageStr:='More files below';
  GotoXY(x1 + ((x2-x1)-length(PageStr)) div 2, y2);
  TextAttr := TitleAttr;
  write(' ', PageStr, ' ');
  TextAttr:=MenuAttr;
end;
{ Write the entries inside the block }
Row:=1;
Col:=1;
TextAttr:=MenuAttr;
for i:=1 to CurrentNumFiles do
begin
  GotoXY(x1+1+(Col-1)*14,y1+Row);
  write(' ',FilePage[Page,i]+BLANKSTRING(FilePage[Page,i]));
  inc(Col);
  if ((Col>MaxCol) and (i<>CurrentNumFiles)) then
  begin
    inc(Row);
    Col:=1;
  end;
end;
MaxRow:=Row;
{ Rewrite the blank entries in the last row }
if ((MaxCol*MaxRow)>CurrentNumFiles) then
begin
  for i:=1 to ((MaxCol*MaxRow)-CurrentNumFiles) do
  begin
    Row:=MaxRow;
    Col:=(MaxCol-((MaxCol*MaxRow)-CurrentNumFiles))+i;
    GotoXY(x1+1+(Col-1)*14,y1+Row);
    write(' ');
  end;
end;
{ Place highlight on first file }
Row:=OldRow;

```

```

Col:=OldCol;
HIGHLIGHT;
( Move cursor to required position )
repeat
  ChangePage:=False;
  OldRow:=Row;
  OldCol:=Col;
  SpecialKey:=False;
  repeat
    Character:=UpCase(ReadKey);
  until Character in [#0,#13,#27];
  if Character=#0 then
  begin
    SpecialKey:=True;
    Character:=UpCase(ReadKey);
    case Character of
      #72 : Row:=Row-1;   ( Up arrow )
      #75 : Col:=Col-1;   ( Left arrow )
      #77 : Col:=Col+1;   ( Right arrow )
      #80 : Row:=Row+1;   ( Down arrow )
      #73 : begin          ( Page up )
              if (Page>1) then
              begin
                ChangePage:=True;
                Page:=Page-1;
                OldRow:=Row;
                OldCol:=Col;
              end;
            end;
      #81 : begin          ( Page down )
              if (Page<NumPages) then
              begin
                ChangePage:=True;
                Page:=Page+1;
                if (Page=NumPages) then OldRow:=1 else OldRow:=Row;
                OldCol:=Col;
              end;
            end;
      #71: begin          ( Home )
              if (Page=1) then
              begin
                Row:=1;
                Col:=1;
              end
            else
              begin
                ChangePage:=True;
                Page:=1;
                OldCol:=1;
                OldRow:=1;
              end;
            end;
    end; ( case )
  end;
  if (Row>MaxRow) then    ( scolling past the bottom )
  begin
    if (Page<NumPages) then
    begin
      ChangePage:=True;
      Page:=Page+1;
      OldRow:=1;
      OldCol:=Col;
    end
    else Row:=Row-1;
  end;
  if (Row<1) then        ( scolling past the top )
  begin
    if (Page=1) then Row:=1
    else
    begin
      ChangePage:=True;
      Page:=Page-1;
    end;
  end;
end;

```



```

OldRow:=MaxRowDisplay;
OldCol:=Col;
end;
end;
if (Col>MaxCol) then          ( scrolling past the right )
begin
Row:=Row+1;
Col:=1;
if (Row>MaxRow) then
begin
if (Page=NumPages) then
begin
Col:=MaxCol;
Row:=Row-1;
end
else
begin
ChangePage:=True;
Page:=Page+1;
OldRow:=1;
OldCol:=1;
end;
end;
end;
end;
if (Col<1) then              ( scrolling past the left )
begin
Row:=Row-1;
Col:=MaxCol;
if (Row<1) then
begin
if (Page=1) then
begin
Row:=1;
Col:=1;
end
else
begin
ChangePage:=True;
Page:=Page-1;
OldRow:=MaxRowDisplay;
OldCol:=MaxCol;
end;
end;
end;
end;
if ((Row=MaxRow) and ((MaxCol*(Row-1)+Col)>CurrentNumFiles)) then
begin
if (Page=NumPages) then
begin
if Character=#80 then Row:=Row-1;
if Character=#77 then Col:=Col-1;
end
else          ( Page down )
begin
ChangePage:=True;
Page:=Page+1;
OldRow:=1;
OldCol:=Col;
end;
end;
end;
if ChangePage then
begin
TextAttr:=NormalAttr;
ClrScr;
GotoXY(1,25);
TextAttr:=BrightAttr;
write(' < Use arrow keys and RETURN to select a result file (ESC → no file) >');
TextAttr:=NormalAttr;
end
else HIGHLIGHT;
until (Character in [#13,#27]) or ChangePage;
until (not ChangePage);
if Character=#27 then Chosen:=''

```

```

else
begin
  Chosen:=FilePage[Page,(MaxCol*(Row-1)+Col)];
  if (copy(Chosen,length(Chosen),1)='\')then  ( Change directory )
  begin
    if (copy(Chosen,1,1)='.') then
    begin
      repeat
        delete(Dir,length(Dir),1);
      until copy(Dir,length(Dir),1)='\';
    end
    else Dir:=Dir+Chosen;
    TextAttr:=NormalAttr;
    ClrScr;
  end;
end;
until (copy(Chosen,length(Chosen),1)<>'\');
CURSORON;
TextAttr:=OldAttr;
ClrScr;
end; ( of procedure CHOOSEFILE )

```

```

( function to read a file name - only valid characters are accepted )
function readfilename:string;
const n=7;
var
  i,num      : integer;
  x,y        : byte;
  t          : array[1..(n+1)] of char;
  name       : string;
  test,return : boolean;
begin
  i:=1;
  name:='';
  x:=WhereX;
  y:=WhereY;
  ClrEol;
  repeat
    repeat
      t[i]:=UpCase(ReadKey);
      num:=ord(t[i]);
      if (i>1) then
        test:=((num>=65) and (num<=90)) or ((num>=48) and (num<=57)) or (num=8)
      else test:=((num>=65) and (num<=90));
      return:=(num=13);
    until test or return;
    if (num<>8) then
      begin
        write(t[i]);
        if (num<>13) then name:=concat(name,t[i]);
        i:=i+1
      end
    else
      begin
        i:=i-1;
        delete(name,length(name),1);
        GotoXY(x,y);
        ClrEol;
        write(name)
      end;
  until return or (i=n);
  ClrEol;
  if (return and (length(name)=0)) then name:='DEFAULT';
  readfilename:=name;
end; ( readfilename )

```

```

function READINT:integer;
const n=5;

```

```

var
  i,num,code,int : integer;
  x,y           : byte;
  t             : array[1..n] of char;
  value        : string;
  test,return   : boolean;
begin
  i:=1;
  value:='';
  x:=WhereX;
  y:=WhereY;
  ClrEol;
  repeat
    repeat
      t[i]:=UpCase(ReadKey);
      num:=ord(t[i]);
      test:=((num>=48) and (num<=57)) or ((num=8) and (length(value)>0));
      return:=(num=13);
    until test or return;
    if (num<>8) then
      begin
        write(t[i]);
        if (num<>13) then value:=concat(value,t[i]);
        i:=i+1
      end
    else
      begin
        i:=i-1;
        delete(value,length(value),1);
        GotoXY(x,y);
        ClrEol;
        write(value)
      end;
    until return or (i=n);
    val(value,int,code);
    readint:=int;
end; ( of function READINT )

```

```

function READREAL:double;
const n=20;
var
  test1,test2,return : boolean;
  dot                : char;
  t                  : array[1..n] of char;
  valuestring       : string;
  x,y               : byte;
  i,num,code        : integer;
  realvalue         : double;
begin
  i:=1;
  dot:='N';
  valuestring:='';
  x:=WhereX;
  y:=WhereY;
  ClrEol;
  repeat
    repeat
      t[i]:=UpCase(ReadKey);
      num:=ord(t[i]);
      test1:=((num>=48) and (num<=57)) or ((num=8) and (length(valuestring)>0));
      test2:=((num=45) and (i=1)) or ((num=46) and (dot='N'));
      return:=(num=13);
    until test1 or test2 or return;
    if (num=46) then dot:='Y';
    if (num<>8) then
      begin
        write(t[i]);
        if (num<>13) then valuestring:=concat(valuestring,t[i]);
        i:=i+1
      end
    else

```

```

begin
  i:=i-1;
  if (copy(valuestring,length(valuestring),1)='.') then dot:='N';
  delete(valuestring,length(valuestring),1);
  GotoXY(x,y);
  ClrEol;
  write(valuestring)
end;
until return or (i=n);
ClrEol;
val(valuestring,realvalue,code);
readreal:=realvalue;
end; { of function READREAL }

{ Procedure to draw a box AROUND (outside) the window coordinates it is }
{ given. It starts drawing a box at (x - 1, y - 1). The boxes dimensions }
{ are width + 2 wide and height + 2 high. }
procedure DRAWBOX(X,Y,Width,Height,BorderAtt,TitleAtt:integer;title:str80);
var
  I,OldColor : integer;
  S           : string[80];
  SLen       : byte absolute S;
type BoxRec = record
  UL,  UR   : char;
  LL,  LR   : char;
  Horiz, Vert : char;
  LT,  RT   : char;
  TT,  BT   : char;
end;
const SingleBox : BoxRec = (UL   : '┌'; UR   : '┐';
                             LL   : '└'; LR   : '┘';
                             Horiz : '-'; Vert : '|';
                             LT   : '├'; RT   : '┤';
                             TT   : '┴'; BT   : '┬');
begin
  Window(1, 1, 80, 25);
  OldColor := TextAttr;
  with SingleBox do
  begin
    FillChar(S, SizeOf(S), Horiz); { fill string with horiz. chars }
    SLen := Width;
    X := Pred(X);
    Y := Pred(Y);
    Width := Succ(Width);
    Height := Succ(Height);
    TextAttr := BorderAtt;
    GoToXY(X, Y); { upper left }
    Write(UL, S, UR);
    for I := 1 to Height do { sides }
    begin
      GoToXY(X, Y + I);
      Write(Vert);
      GoToXY(X + Width, Y + I);
      Write(Vert);
    end;
    GoToXY(X, Y + Height); { lower left }
    Write(LL, S, LR);
    { Center title on top of box }
    if title <> '' then
    begin
      GoToXY(X + Pred(Width - Ord(title[0])) shr 1, Y);
      TextAttr := TitleAtt;
      Write(' ', title, ' ');
    end;
  end; { with }
  TextAttr := OldColor;
end; { of procedure DRAWBOX }

{ Procedure to open a window and draw a titled box around it }
procedure WINDOWBOX(x1,y1,width,height:integer;title:str80);

```

```

begin
  DRAWBOX(x1,y1,width,height,6,120,title);
  Window(x1,y1,x1+width-1,y1+height-1);
  ClrScr;
end; { of procedure WINDOWBOX }

{ Procedure to draw a line around the full screen }
procedure OUTLINE(title:str80);
begin
  WINDOWBOX(2,2,78,22,title);
end; { of procedure OUTLINE }

function P(aa,bb:double):double;
begin
  P:=exp(bb*ln(aa));
end;

function Htan(aa:double):double;
begin
  Htan:=(exp(aa)-exp(-1.0*aa))/(exp(aa)+exp(-1.0*aa));
end;

function LOG10(aa:double):double;
begin
  LOG10:=(ln(aa))/ln(10);
end;

{ Function to determine atmospheric pressure }
function ATPRESSURE(h,t:double):double;
var rho : double;
begin
  rho:=13545.87/(1+(18115E-8)*(t-20)+(0.8E-8)*sqr(t-20));
  atmpressure:=rho*9.7962*h/1000;
end;

{ Function to calculate the saturation vapour-pressure of water }
function satvappres(t:double):double;
var a,b,c,d,e,f,g,x,z,z1,z2,z3,TT:double;
begin
  TT:=t+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/TT;
  z1:=a*(1-x)+b*(ln(x)/ln(10));
  z2:=c*(1-P(10,(d*((1/x)-1)))));
  z3:=e*(P(10,(f*(1-x))-1)+g);
  z:=z1+z2+z3;
  satvappres:=P(10,z);
end;

{ Function to calculate the specific heat of water-vapour }
function cpvf(t:double):double;
var a,b,c,d,TT :double;
begin
  TT:=t+273.16;
  a:=1.3605E3;
  b:=2.31334;
  c:=-2.46784E-10;
  d:=5.91332E-13;
  cpvf:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;

```

```
{ Function to calculate the specific heat of air }
```

```
function cpaf(t:double):double;
var a,b,c,d,TT :double;
begin
  TT:=t+273.16;
  a:=1.045356E3;
  b:=-3.161783E-1;
  c:=7.083814E-4;
  d:=-2.705209E-7;
  cpaf:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;
```

```
{ Function to calculate the specific heat of water }
```

```
function cpwf(t:double):double;
var a,b,c,d,TT :double;
begin
  TT:=t+273.16;
  a:=8.15599E3;
  b:=-2.80627E1;
  c:=5.11283E-2;
  d:=-2.17582E-13;
  cpwf:=a+b*TT+c*TT*TT+d*TT*TT*TT*TT*TT*TT;
end;
```

```
{ Function to calculate the humidity of air }
```

```
function airhumidity(t1,t2,patm:double):double;
var cpa,cpv,cpw,ps,wa,q0,q1,q2 :double;
begin
  ps:=satvappres(t2);
  wa:=(0.62198*ps)/(patm-(1.005*ps));
  q0:=(2501.6-2.3263*t2)*wa;
  q1:=1.00416*(t1-t2);
  q2:=(2501.6+(1.8577*t1)-(4.184*t2));
  airhumidity:=(q0-q1)/q2;
end;
```

```
{ Function to calculate the humidity of saturated air (5-70°C,99-102kPa)}
```

```
function Satairhumidity(t1,patm:double):double;
var x1,x2,x3 : double;
begin
  x1 := -1.513124e-3*t1-10.4260667e-6*t1*t1;
  x2 := -1.01703527e-6*t1*t1*t1+18.10209*1/(100.0-t1);
  x3 := 0.057415e-6*patm+0.1055343*t1*t1*1/patm;
  satairhumidity := x1+x2+x3-0.18305;
end;
```

```
{ Function to calculate the enthalpy of air using the wb and db temps. }
```

```
function enthalpy(t1,t2,patm:double):double;
var cpv,cpa,wa,ivap,ida :double;
begin
  cpv:=1857.7; {cpvf(t1);}
  cpa:=1004.16; {cpaf(t1);}
  wa:=airhumidity(t1,t2,patm);
  ivap:=wa*(2501.6+((cpv*t1)/(1000)));
  ida:=(cpa*t1)/(1000);
  enthalpy:=ida+ivap;
end;
```

```
{ Function to calculate the enthalpy of saturated air using the wb and db temps. }
```

```
function Satenthalpy(t1,patm:double):double;
var cpv,cpa,wa,ivap,ida :double;
begin
  cpv:=1857.7; {cpvf(t1);}
  cpa:=1004.16; {cpaf(t1);}
  wa:=Satairhumidity(t1,patm);
  ivap:=wa*(2501.6+((cpv*t1)/(1000)));
  ida:=(cpa*t1)/(1000);
```

```
satenthalpy:=ida+ivap;
end;
```

```
{ Function to calculate the dynamic viscosity of air }
function airviscosity(t:double):double;
var a,b,c,d,TT      :double;
begin
  TT:=t+273.16;
  a:=2.287973E-6;
  b:=6.259793E-8;
  c:=-3.131956E-11;
  d:=8.15038E-15;
  airviscosity:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;
```

```
{ Function to calculate the dynamic viscosity of water }
function watervisosity(t:double):double;
var a,b,c,TT        :double;
begin
  TT:=t+273.16;
  a:=2.414E-5;
  b:=247.8;
  c:=140;
  watervisosity:=a*P(10,(b/(TT-c)));
end;
```

```
{ Function to calculate the dynamic viscosity of water vapour }
function vapourviscosity(t:double):double;
var a,b,c,d,TT      :double;
begin
  TT:=t+273.16;
  a:=2.562435E-6;
  b:=1.816683E-8;
  c:=2.579066E-11;
  d:=-1.067299E-14;
  vapourviscosity:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;
```

```
{ Function to calculate the dynamic viscosity of air/water vapour mix }
function airvapmixviscosity(t,wa,patm:double):double;
var xa,xv,mua,muv    :double;
begin
  xa:=5.3824/(1+1.608*wa);
  xv:=wa*4.2445/(wa+0.622);
  mua:=airviscosity(t);
  muv:=vapourviscosity(t);
  airvapmixviscosity:=(xa*mua+xv*muv)/(xa+xv);
end;
```

```
{ Function to calculate water-density }
function waterdensity(t:double):double;
var a,b,c,d,TT      :double;
begin
  TT:=t+273.16;
  a:=1.49343E-3;
  b:=-3.7164E-6;
  c:=7.09782E-9;
  d:=-1.90321E-20;
  waterdensity:=1/(a+b*TT+c*TT*TT+d*TT*TT*TT*TT*TT*TT);
end;
```

```
{ Function to calculate evaporation enthalpy of water }
function HFGWATER(t:double):double;
var a,b,c,d,TT      :double;
begin
  TT := t+273.16;
```

```

a := 3.4831814E6;
b := -5.8627703E3;
c := 12.139568;
d := -1.40290431E-2;
HfgWater := a+b*TT+c*sqr(TT)+d*TT*TT*TT;
end;

```

```

{ Function to calculate air-density }
function airdensity(t,patm:double):double;
var TT : double;
begin
  TT:=t+273.16;
  airdensity:=patm/(287.08*TT);
end;

```

```

{ Function to calculate the density of an air/water vapour mix }
function airvapmixturedensity(t,wa,patm:double):double;
var rhosa :double;
begin
  rhosa:=airdensity(t,patm);
  airvapmixturedensity:=(1.0+wa)*(1.0-wa/(wa+0.62198))*rhosa;
end;

```

```

{ Function to calculate the conductivity of air }
function airconductivity(t:double):double;
var a,b,c,d,TT :double;
begin
  TT:=t+273.16;
  a:=-4.937787E-4;
  b:=1.018087E-4;
  c:=-4.627937E-8;
  d:=1.250603E-11;
  airconductivity:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;

```

```

{ Function to calculate the conductivity of water }
function waterconductivity(t:double):double;
var a,b,c,d,TT :double;
begin
  TT:=t+273.16;
  a:=-6.14255E-1;
  b:=6.9962E-3;
  c:=-1.01075E-5;
  d:=4.74737E-12;
  waterconductivity:=a+b*TT+c*TT*TT+d*TT*TT*TT;
end;

```

```

{ Function to calculate the Prandtl-number of air }
function airprandtl(t:double):double;
var ka,mua,cpa :double;
begin
  ka:=airconductivity(t);
  mua:=airviscosity(t);
  cpa:=cpaf(t);
  airprandtl:=cpa*mua/ka;
end;

```

```

{ Function to calculate the Prandtl-number of water }
function waterprandtl(t:double):double;
var kw,muw,cpw :double;
begin
  kw:=waterconductivity(t);
  muw:=watervisosity(t);
  cpw:=cpwf(t);
  waterprandtl:=cpw*muw/kw;
end;

```