A General Approach for the Rating of Evaporative Closed Circuit Coolers

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Numerous procedures to predict the performance of closed circuit evaporative coolers have been reported in the past with varying degrees of approximation in the thermodynamic and heat transfer equations. While these methods are generally found to be adequate for most industrial design applications, they have shortcomings when linking them to natural draught situations where a crossflow unit is preferred and the outlet air density is critical in determining the air flow rate. This article describes a more rigorous method based on the method of Poppe [18], Bourillot [16] and Evers and Dreyer [23] as applied to conventional cooling towers.

Nomenclature

a	– air/water surface per unit volume	[m²/m³]
A	– area	[m²]
c_p	– specific heat	[J/kgK]
d	– diameter	[m]
h_c,h	– heat transfer coefficient	[W/m₂K]
H	– enthalpy of air	[J/kg]
H_v (0°)	– enthalpy of evaporation of water at 0 °C	[J/kg]
k_t	– thermal conductivity	[W/mK]
K	– mass transfer coefficient	[kg/m²K]
L	– length of cooler	[m]
N_{Le}	– Lewis factor	
n	– number of elements along length of tube	
N_{Re}	– Reynolds number	
P_{atm}	– atmospheric pressure	[Pa]
t	– temperature	[°C]
U	– overall heat transfer coefficient	[W/m²K]
V	– velocity	[m/s]
w	– massflow rate	[kg/s]
W	– width of cooler	[m]
Y	– humidity ratio	[kg water/kg dry air]
Z	– height of cooler	[m]
Γ	– recirculating water mass flow rate over one half of tube per unit length	[kg/m.s]
μ	– dynamic viscosity	[kg/m.s]
ρ	– density	[kg/m³]

Subscripts

a	– air
db	– dry bulb
f	– fouling
i	– inside, inlet
m	– mean
mc	– minimum cross section
o	– outside, outlet
p	– process water
sa	– saturated air at t_w
sw	– saturated air at t_w
v	– vapour
w	– recirculating water
wb	– wet bulb

Introduction

The analysis of closed circuit evaporative cooler performance presents considerably more problems than does the analysis of much simpler heat transfer devices such as finned tube heat exchangers or direct contact cooling towers. This is due to the three-fluid character of the device which includes heat and mass transfer phenomena.

A number of methods of analysing evaporative coolers and condensers have been presented in the past and they have in many instances been found to be perfectly adequate for general industrial design purposes. However, these methods are limited to cases where the air is in counterflow with the recirculating (or spray) water and most of them include a number of approximations to simplify calculations. The most frequently used approximations are incorporated in the Merkel method which accounts for the combined heat and mass transfer effect at the air-water interface and the assumption of zero water loss in the recirculating water. Most of the methods also use a one dimensional approach, making an analytical solution possible and in some cases a constant average recirculating water temperature has been assumed.

The solutions or predictions obtained using the various approximations mentioned above are generally satisfactory for the design of compact forced draught counterflow units, but fail to give satisfactory results in larger units, particularly when employed in natural draught situations. In the latter case the air is not necessarily completely saturated as it leaves the tube banks and the determination of the exact air density is of prime importance when predicting the performance of the whole natural draught unit. In such cases it is often necessary from a practical point of view to use relatively long tube-runs with not many pipe rows, making a one-dimensional approach less desirable. In addition one-dimensional models are inadequate when applied to crossflow situations.

The approach described below makes none of the approximations used by most of the previous authors and is a general numerical procedure which can be applied to virtually any geometry, although different procedures might be required to achieve convergence. It is also possible to predict accurately the state of the outlet air making it particularly suitable for natural draught applications.

Clearly, because of the three dimensional nature of the approach, computing times are generally much longer, but in view of the applications envisaged this is in some instances justified. Whereas the one dimensional methods can generally be executed in acceptable run times on a desk top computer, this is not so in the present case.

Literature Review

Evaporative coolers

One of the earliest useful analytical treatments of close circuit evaporative coolers was due to Parker and Treybal [1]. The method was derived before low cost computing facilities were generally available and used Merkel's approximation for the

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heat-mass transfer process. One of the most significant features of this work is that the recirculating water termerature was not assumed constant and it was therefore the first solution which described the variation of this temperature as the water ran over the tubes. In addition the enthalpy of the saturated air was assumed to be a linear function of temperature, making it possible to integrate the simultaneous differential equations over the height of the coil.

Mizushina and Miyashita [2] using a similar approach to Parker and Treybal integrated their equations numerically using a computer. At the same time the above two authors [3] carried out some useful experiments to determine the applicable heat and mass transfer coefficients in a smooth tube core with triangular spacing.

Perez-Blanco and Bird [4] did an analysis on the performance of a rather idealised vertical counterflow evaporative cooling unit, but used the correct thermodynamic equations without any approximations.

Kreid et al [5] presented an approximate method of analysing deluged heat exchangers with fins using an effective overall heat transfer coefficient based on log mean enthalphy difference which is analogous to the LMTD in a dry cooler. They practically demonstrated the method to predict heat transfer rates within 5% of the actual values.

Leidenfrost and Korenic [6, 7] presented a rigorous analysis of finned tube evaporative condensers which could be applied to cross or counterflow devices and did not make use of Merkel's approximation or a Lewis factor of unity. Their analysis can even accomodate partially dry heat exchangers. They proposed the used of a stepwise integration process using a graphic method originally derived by Bosnjakovic [8] to determine the exit state of the air and water leaving an element which amounts to a computerisation of that method.

Some useful software was developed by Webb and Villacres [9] to approximate the performance of various types of evaporative cooling devices using a unified approach for the air side. It is stated that the prediction accuracy is in the order of 3% of the manufacturer's data on several devices. However, the evaporative cooler program is limited to vertical counterflow equipment.

Cooling Towers

The number of articles on conventional cooling towers using Merkel's method are too numerous to mention and are not particularly relevant to the subject under discussion.

Various researchers such as Mesarovic [10], Yadigaroglu and Pastor [11] Nahavandi et al [12], Nahavandi and Oelinger [13] and Sutherland [14] have shown that differences of between 12 and 15% in predicted performance of cooling towers can be obtained when using the correct thermodynamic heat and mass transfer and mass conservation equations as opposed to the approximate Merkel method. However it is pointed out by some of these authors that greater accuracy is obtained with the Merkel method if the transfer coefficients determined with that method are used. This point is clearly illustrated by the work of Webb and Villacres [9].

More recently some very comprehensive programs have been developed by Majumdar et al [15], Bourillot [16] and Park et al [17] using a more basic approach first suggested by Poppe [18]. This method uses the basic thermodynamic equations together with the equations for heat and mass transfer as well as the conservation of mass equation. It also accounts for the possibility of oversaturation in the air where mist formation occurs.

The present method uses the same basic approach as Bourillot [16] but the equations contain additional terms relating to the heat transfer between process fluid and the recirculating water. It does not consider the possibility of dry patches as in the case of Leidenfrost and Korenic [6, 7] and also differs from their approach in that a fourth order Runge-Kutta integration procedure is employed.

Basic Theory

A typical element from an evaporative cooler as seen in figure 1 is considered. The elements are chosen according to the physical dimensions of the cooler. Each element is thus chosen as an

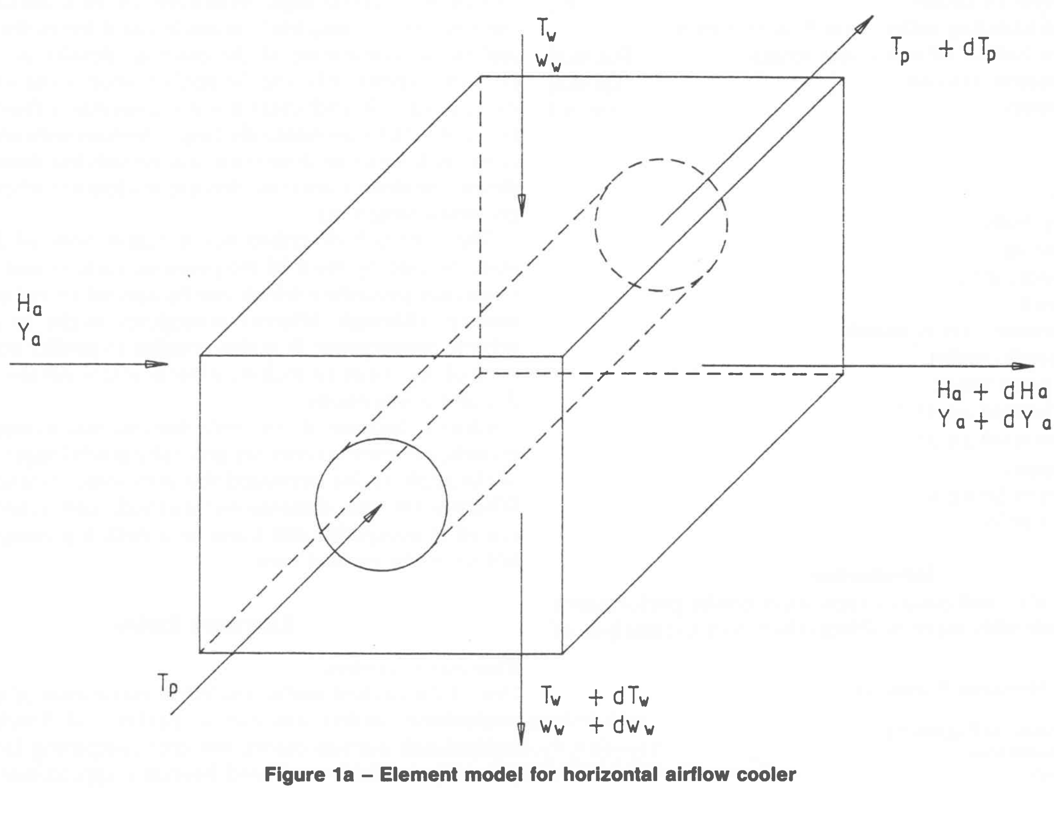

Figure 1a – Element model for horizontal airflow cooler

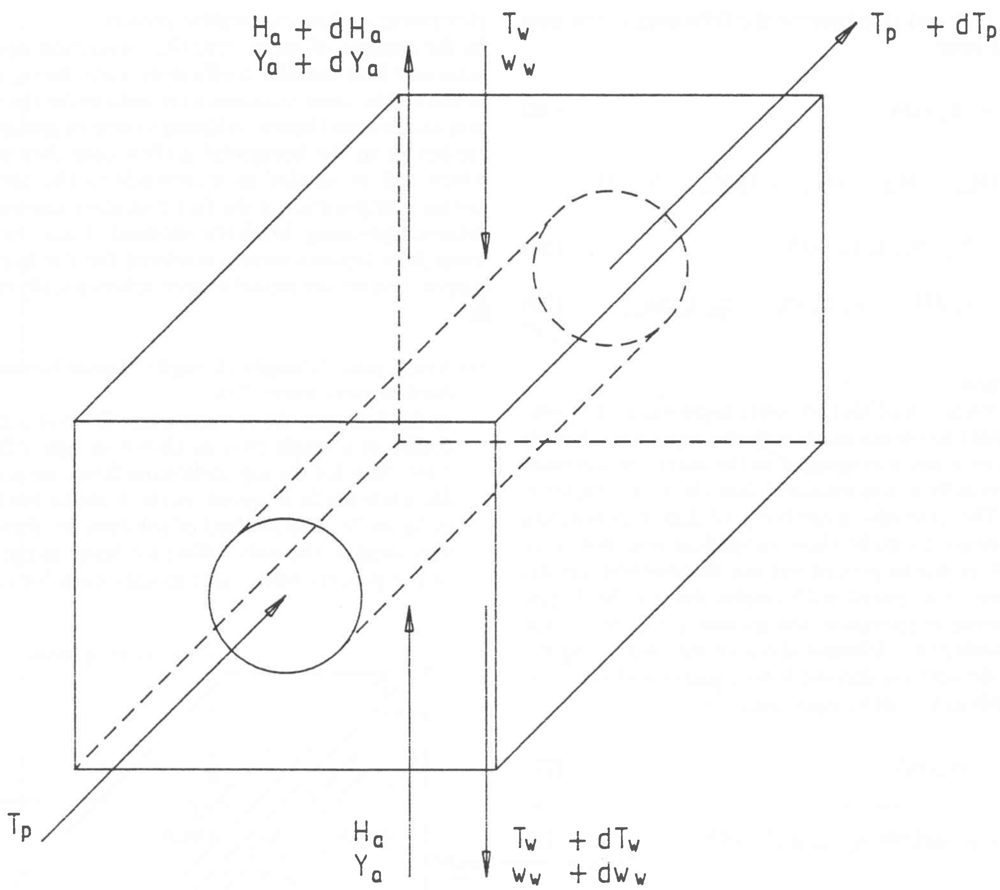

Figure 1b – Element model for vertical airflow cooler

imaginary block around a section of tube with an outer tube surface of area dA.

Each element is influenced by three energy streams, process water flowing inside the tube, gravity driven recirculating water flowing from the top down and air flowing either horizontally or vertically upwards.

The following assumptions are used in the analysis:

(i) U-bends at the ends of the exchanger are insulated against heat transfer.

(ii) The recirculating water is distributed evenly along the tubes.

(iii) The air/water surface area outside tube area.

Poppe formulation using exact thermodynamic analysis

The mass balance for the element gives:

$$dw_w = -w_a \, dY_a \tag{1}$$

and the energy balance gives:

$$w_w \, c_{pw} \, dt_w = -w_a \, dH_a - w_p \, c_{pp} \, dt_p - c_{pw} \, t_w \, dw_w \tag{2}$$

The evaporation law of Dalton gives the flow rate of recirculating water evaporating into non-saturated air from the element as

$$dw_w = -K \, (Y_{sw} - Y_a) \, dA \tag{3}$$

For the mist zone Dalton's evaporation law states

$$dw_w = -K \, (Y_{sw} - Y_{sa}) \, dA \tag{4}$$

The change of enthalpy of the air is given by

$$w_a \, dH_a = -H_v \, dw_w + h_c(t_w - t_a) \, dA \tag{5}$$

The enthalpies of water vapour and water liquid are given by the following relations

$$H_v = H_v \, (O°) + c_{pv} \, t_w \tag{6}$$

$$H_w = c_{pw} \, t_w \tag{7}$$

For non-saturated air the enthalpy is given by

$$H_a = Y_a \, H_v \, (O°) + c_{pm} \, t_a \tag{8}$$

$$\text{with } c_{pm} = c_{pa} + Y_a \, c_{pv} \tag{9}$$

In the mist zone the following holds

$$H_a = c_{pm} \, t_a + Y_{sa} \, H_v \, (O°) \tag{10}$$

$$\text{with } c_{pm} = c_{pa} + Y_{sa} \, c_{pv} + (Y_a - Y_{sa}) \, c_{pw} \tag{11}$$

For non-saturated air the following five equations can be derived from equations (1) to (9) to fully describe the processes that takes place within each element.

$$dw_w = -w_a \, dY_a \tag{13}$$

$$dY_a = \frac{K}{w_a} (Y_{sw} - Y_a) \, dA \tag{14}$$

$$dH_a = \frac{K}{w_a} [N_{Le} \, (H_{sw} - H_a) - (N_{Le} - 1)(Y_{s\dot{w}} - Y_a) \, H_u] \, dA \tag{15}$$

$$dt_w = \frac{1}{w_w \, c_{pw}} [-w_a \, dH_a - w_p \, c_{pp} \, dt_p - c_{pw} \, t_w \, dw_w] \tag{16}$$

$$dt_p = \frac{-U}{w_p \, c_{pp}} (t_p - t_w) \, dA \tag{17}$$

Equations (14), (15) and (16) become the following in the mist (supersaturated) zone.

$$dY_a = \frac{K}{w_a}(Y_{sw} - Y_{sa})\, dA \tag{18}$$

$$dH_a = \frac{K}{w_a}[N_{Le}(H_{sw} - H_a) - (N_{Le} - 1)(Y_{sw} - Y_{sa})\, H_v + (Y_a - Y_{sa})\, N_{Le}\, t_w\, c_{pw}]\, dA \tag{19}$$

$$dt_w = \frac{1}{w_w\, c_{pw}}[-w_a\, dH_a - w_p\, c_{pp}\, dt_p - c_{pw}\, t_w\, dw_w] \tag{20}$$

Merkel formulation

The classical formulation of Merkel contained a number of simplifications in order to obtain easily solvable equations. Firstly the evaporation of water was ignored in the mass conservation equation and secondly it was assumed that the Lewis factor is equal to unity. The extensive availibility of digital computers makes it unnecessary to make these simplifications, but since most of the existing design procedures use this method, results thus obtained were compared with results due to the Poppe method. By ignoring evaporation and assuming a Lewis factor of unity the following two differential equations controlling the changes in each element are derived from equations (13) to (16) and used in combination with equation (17).

$$dH_a = \frac{K}{w_a}(H_{sw} - H_a)\, dA \tag{21}$$

$$dt_w = \frac{1}{w_w\, c_{pw}}[-w_a\, dH_a - w_p\, c_{pp}\, dt_p] \tag{22}$$

The state of the air in each element cannot be fully determined because only the enthalpy of the air is known. Since the air massflow through a cooling tower is determined by the difference between the air density inside and outside the tower it is very important to determine the density of the air inside the tower with a high degree of accuracy. The Merkel method however has a shortcoming in this regard since only one property of the air is known and the density of the outlet air cannot be determined accurately. Usually the outlet air from the cooling tower pack is assumed to be saturated in order to obtain a value for the air density. This is adequate for counterflow cooling towers, but in the case of a crossflow cooling tower the outlet air enthalpy varies with height and the assumption of saturated outlet air may lead to inaccuracies.

Singham [22] proposed the following equation to determine the outlet humidity ratio of the air when the Merkel formulation is used

$$dY_a = \frac{Y_{sw} - Y_a}{1 - Y_{sw}}\frac{K}{w_a}\, dA \tag{26}$$

If the equation is employed together with the three Merkel equations the state of the air in each element can be determined, thus the exit density of the air can be determined without the additional assumption that the exit air is saturated. The amount of recirculating water evaporating from each element can also be determined when employing the equation above. It should be noted that when employing the above equation the simplifying assumptions of Merkel are still used and that an estimation is made of the humidity of the air and the amount of water evaporated.

Solution Methods for various Geometries

In all the geometries considered the process fluid is assumed to flow through rows of tubes connected in parallel.

Horizontal airflow evaporative coolers

In the absence of more suitable correlation equations for the mass and heat transfer coefficients from the recirculating water to the air the same relations were used as for the vertical airflow case as discussed below. Although these equations are not really applicable to the horizontal airflow case they will give results which will be similar in magnitude to the correct values. A further complication is the fact that most known data has been determined using Merkel's method. Four different process water flow layouts were considered for the horizontal airflow evaporative cooler model as seen schematically in Figure 2 (a to d).

(i) *Single pass (straight through), top-to-bottom and front-to-back process water flow*

In the first case the process water flows straight through the cooler in a single pass as shown in figure 2a. The process water flow for the top-to-bottom flow case is shown in figure 2b, while the flow layout for the front-to-back case is shown in figure 2c. The method of solution for these three cases is very similar, the only difference being in the determination of the process water inlet temperature for each element.

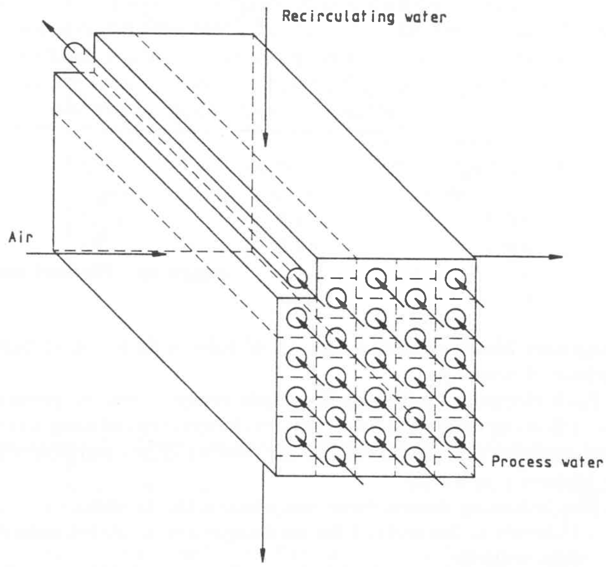

Figure 2a – Single pass, horizontal airflow

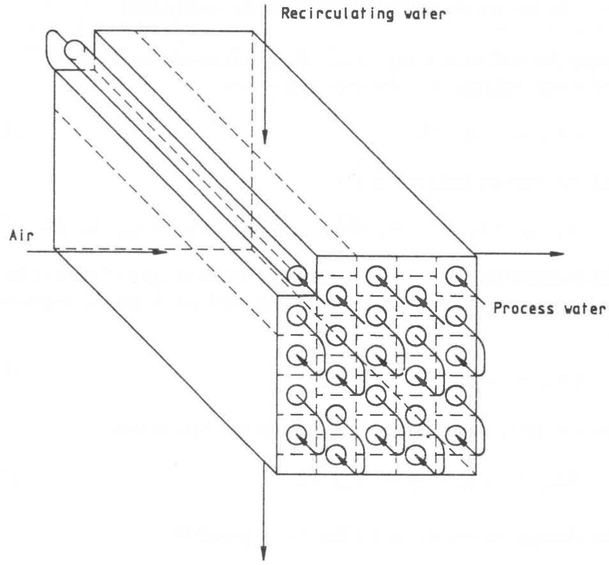

Figure 2b – Multiple pass Top-to-Bottom, horizontal airflow

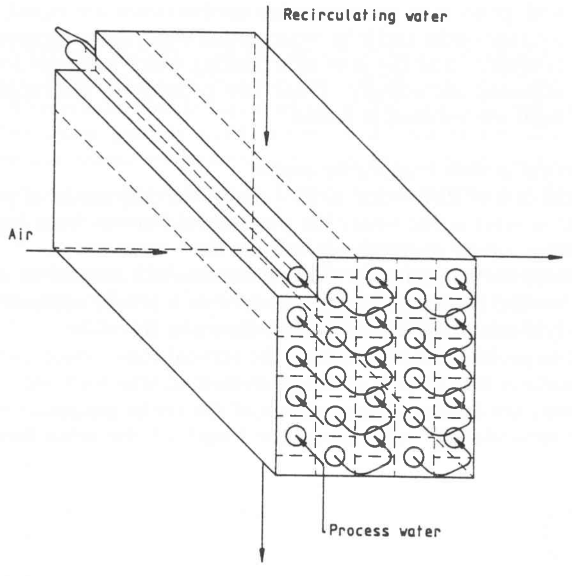

Figure 2c – Multiple pass Front-to-Back, horizontal airflow

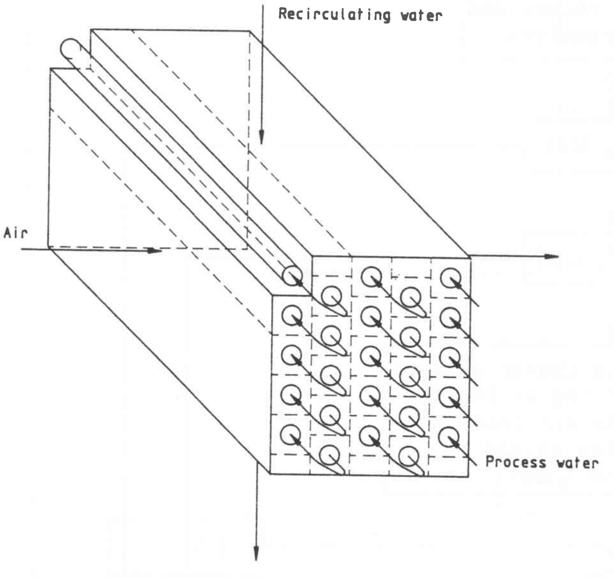

Figure 2d – Multiple pass Back-to-Front, horizontal airflow

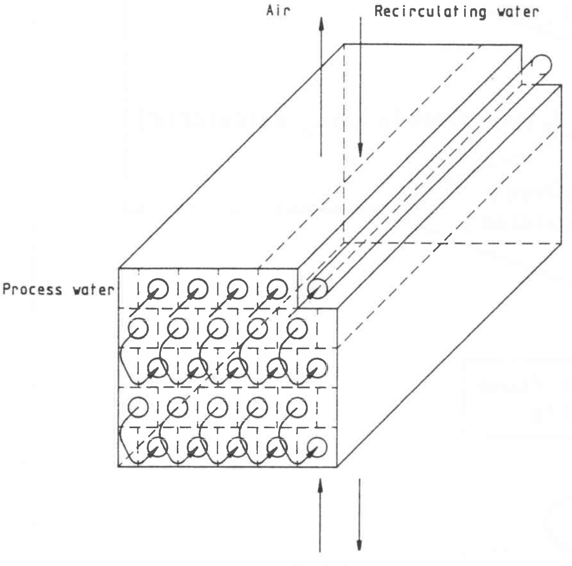

Figure 2e – Multiple pass Top-to-Bottom, vertical airflow

Since the recirculating water system is a closed loop this implies that the temperature at the inlet will be equal to the average mixing temperature at the outlet. In order to obtain a solution for the operating point of the cooler an inlet temperature for the recirculating water is assumed and later corrected, if necessary. Since all the inlet conditions are now known for the top element at the air inlet side the calculations will proceed from here. The exit conditions for this element are determined by solving the controlling differential equations (either Poppe or Merkel formulation) simultaneously by employing a fourth order Runge-Kutta method. When the exit conditions for the first element have been determined the inlet conditions for the next element (k) will be known and the exit conditions for this element can be determined until all the elements in the top row have been evaluated. The evaluation of the second line of elements proceeds in a similar fashion. As soon as all the elements in the first layer facing the inlet air have been evaluated the second layer is evaluated and so on. The average outlet recirculation temperature can now be determined (by adding the enthalpy flows of these various elements) and comparing its value to the inlet recirculating water temperature. If the inlet temperature differs from the average outlet recirculating water temperature a new inlet temperature is assumed and the element-by-element evaluation of the cooler is repeated until the inlet and average outlet recirculating water temperature are the same, giving the operating point of the cooler. A simple flow chart showing the calculation procedure for the three cases discussed above is given in figure 3.

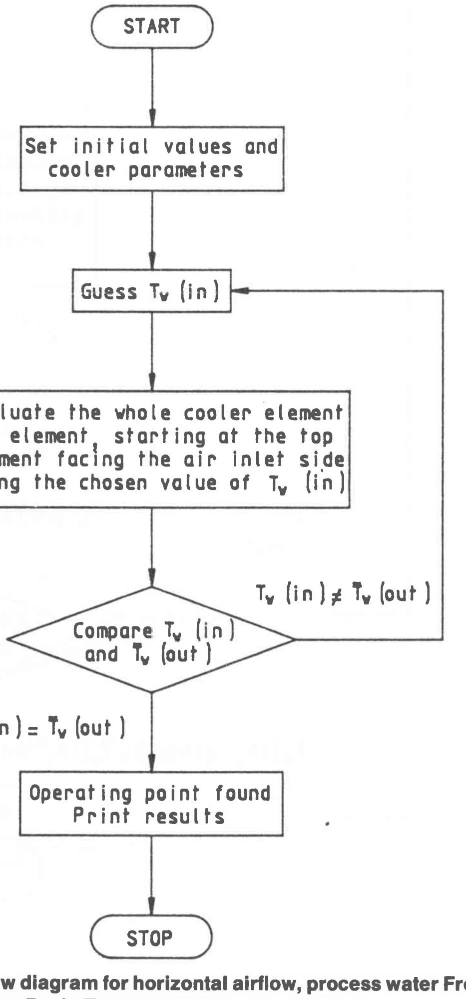

Figure 3 – Flow diagram for horizontal airflow, process water Front to Back, Top to Bottom or Single Pass

(ii) *Process water flow from the back to the front of the cooler*
In this case there is no element for which all the inlet conditions are known even after an initial guess for the recirculating water inlet temperature. An iterative solution method has to be used for the evaluation of a cooler with this process water flow layout. The solution procedure is shown in figure 4. Firstly a value for the recirculating water inlet temperature is chosen and then an outlet process water temperature for the elements facing the inlet air stream. The average inlet process water temperature is determined by evaluating the cooler in a similar manner to the case of the cooler with a front-to-back process water flow layout. If the average calculated inlet process water temperature differs from the given inlet process water inlet temperature, the corresponding outlet process water temperature is changed for each element by half the difference between the calculated and the given inlet process water temperature. Once the calculated and given inlet process water temperatures are equal, the average outlet and inlet recirculating water temperatures are compared and the inlet recirculating water temperatures is adjusted accordingly. These two procedures are repeated until the solution is found.

Vertical airflow evaporative cooler
In the case of the vertical airflow cooler the only model of practical interest is one where the process fluid moves from top to bottom, i.e. in counterflow with the air.

Since the vertical airflow cooler consists of a number of similar vertical elements alongside each other it is only necessary to analyse one of these elements as shown in figure 2e.

The problem when analysing the vertical cooler three-dimensionally is that there is no single element in which all inlet conditions are known. The variation of the outlet temperature of the recirculating water along the length of the tubes further

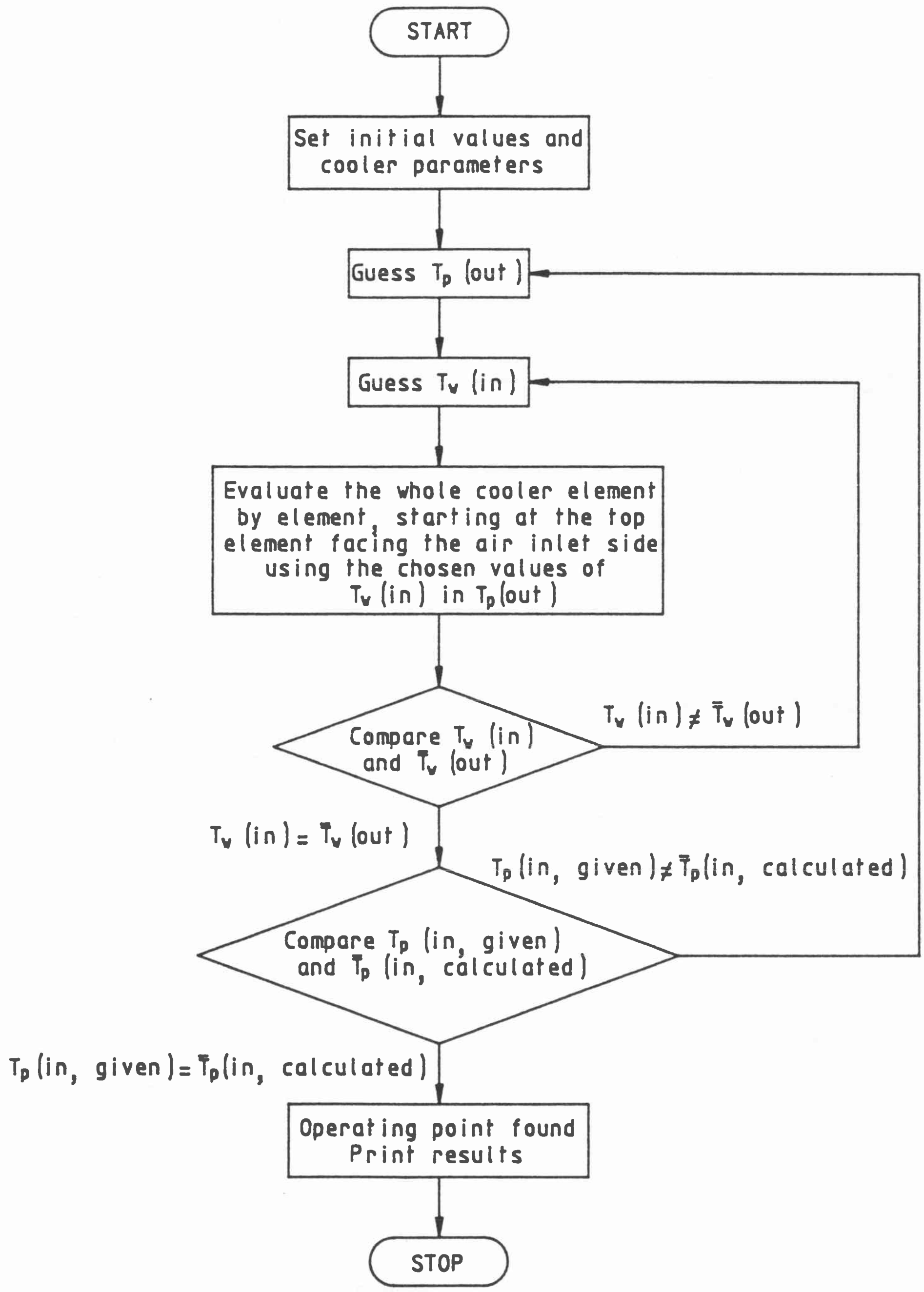

Figure 4 – Flow diagram for horizontal airflow, process water Back to Front

complicates matters as this temperature has to be guessed with a high degree of accuracy before integrating upwards through the tower.

To obtain a solution the process water outlet temperature is chosen and then the recirculating water outlet temperature found by an iterative procedure, after which the calculated inlet process temperature is compared to the acutal value. A new outlet process temperature is then chosen and the whole process repeated until the solution is found out. The solution procedure is shown in figure 5.

Heat and Mass Transfer Correlations and Properties of air, water and air/water mixtures

Empirical relations for the mass transfer coefficient in vertical airflow evaporative coolers with triangularly spaced tubes and a pitch of two diameters are given by Parker and Treybal [1] and Mizushina et al [3].

In the present case Mizushina's correlation for the mass transfer coefficient has been used.

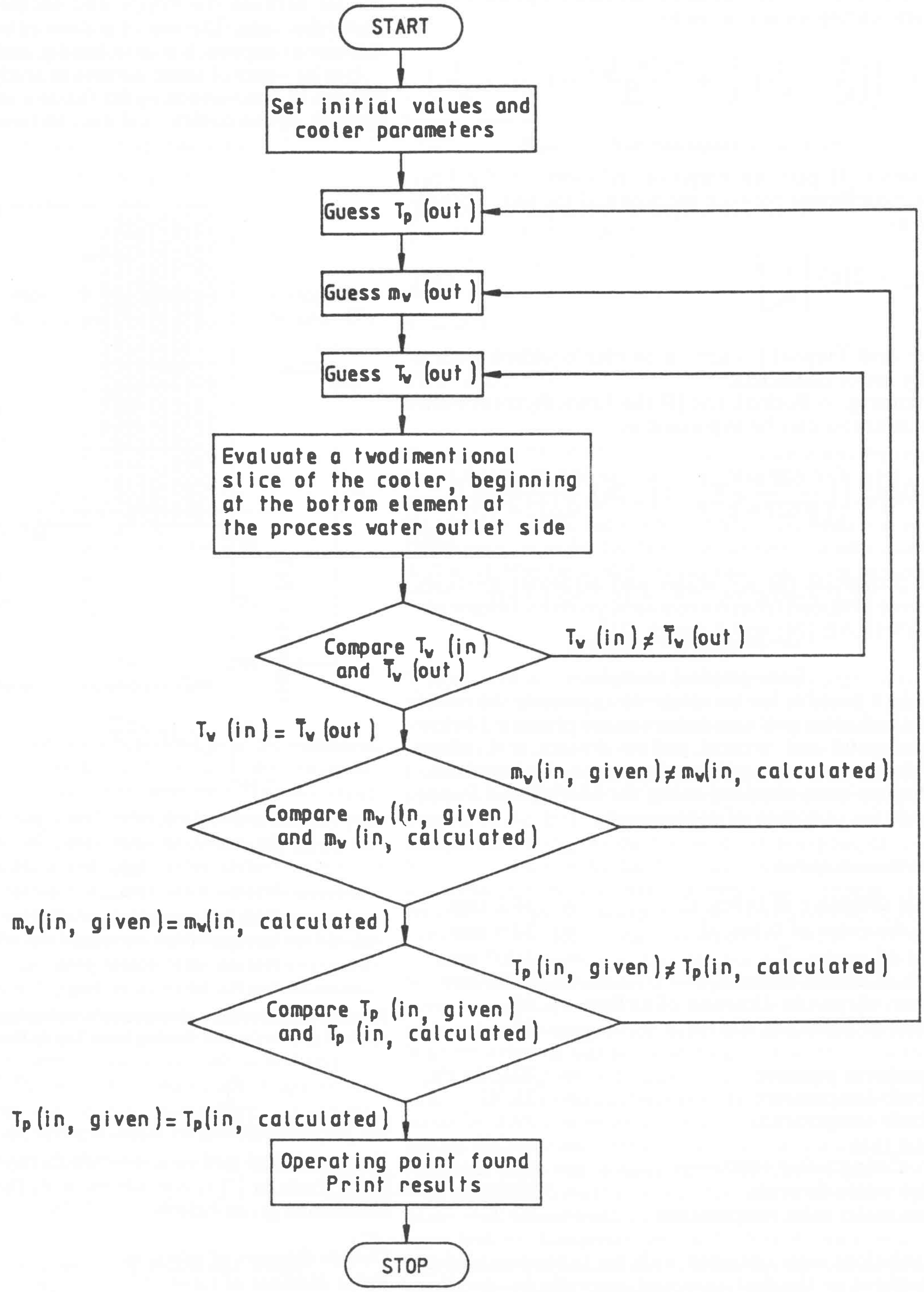

Figure 5 – Flow diagram for vertical cooler with process water Top to Bottom of cooler

$$Ka = 5{,}0278 \times 10^{-8} \, (N^{Rea})^{0{,}9} \, (N_{Rew})^{0.15} \, (d_o)^{-2{,}6} \qquad (25)$$

where the Reynolds number of the air flow is given by

$$N_{Rea} = \frac{\rho_a \, V_{a,mc} \, d_{mc}}{\mu_a} \qquad (26)$$

and the Reynolds number of the recirculating water is given by

$$N_{Rew} = \frac{4\Gamma}{\mu_w} \qquad (27)$$

The overall heat transfer coefficient between the process water and recirculating water is given by

$$U = 1 \Bigg/ \left(\left(\frac{1}{h_p} + \frac{1}{h_{f,i}} \right) \frac{d_o}{d_i} + \frac{d_o \ln [d_o/d_i]}{2k_t} + \frac{1}{h_w} + \frac{1}{h_{f,o}} \right) \qquad (28)$$

Mizushina [3] gave an empirical relation for the heat transfer coefficient between the pipe and the recirculating water as

$$h_w = 2103 \left[\frac{\Gamma}{d_o} \right]^{\frac{1}{3}} \qquad (29)$$

Parker and Treybal [1] used a similar equation with a slightly lower coefficient.

According to Bosjnakovic [8] the Lewis factor for air/water mixtures can be expressed as

$$N_{Le} = 0{,}865^{\frac{2}{3}} \left(\left(\frac{0.622 + Y_{sw}}{0.622 + Y_a} \right) - 1 \right) \Bigg/ \ln \left(\frac{0{,}622 + Y_{sw}}{0{,}622 + Y_a} \right) \qquad (30)$$

Other properties for air, water and air/water mixtures used were obtained from correlations given by Johannsen [19], ASHRAE [20] and Schmidt [21].

Some practical examples

To make it possible for the reader to appreciate the use of the method some problem solutions are presented below for horizontal and vertical airflow devices and, where possible, compared to other solutions in the literature.

Solutions were obtained using the Merkel and Poppe methods for purposes of comparison.

(1) *Horizontal Airflow*

Outside diameter of tubes, d_o	=	38,1 mm
Inside diameter of tubes, d_i	=	34,9 mm
Height of cooler, Z	=	2 000 mm
Length of cooler tubes, L	=	2 000 mm
Number of rows in direction of airflow	=	10
Number of pipes per row	=	25

Atmospheric pressure	=	101 325 Pa
Dry bulb temperature	=	25 °C
Wet bulb temperature	=	19,1 °C
Airflow rate	=	14,573 kg/s
Recirculating water rate	=	3,33 kg/s
Process water flowrate	=	15 kg/s
Process water inlet temperature	=	50 °C

Two solutions were obtained with the tubes regarded as one element in the first case and secondly by dividing them into five elements length-wise.

The results obtained are given in table 1.

	n = 1			n = 5		
Method	t_{wi} °C	ρ_{out} kg/m³	Q kW	t_{wi} °C	ρ_{out} kg/m³	Q kW
Merkel	34,68	1,146	638,550	34,68	1,146	638,494
Poppe	34,48	1,144	639,932	34,49	1,144	639,877

It can be seen from the above results that there is little to choose between the Poppe and Merkel methods in this particular case. The use of a three rather than a two dimensional approach is also hardly justifiable.

It is however of some interest to study the temperature profiles of the various outlet fluids along the length and breadth of the cooler as shown in figure 6.

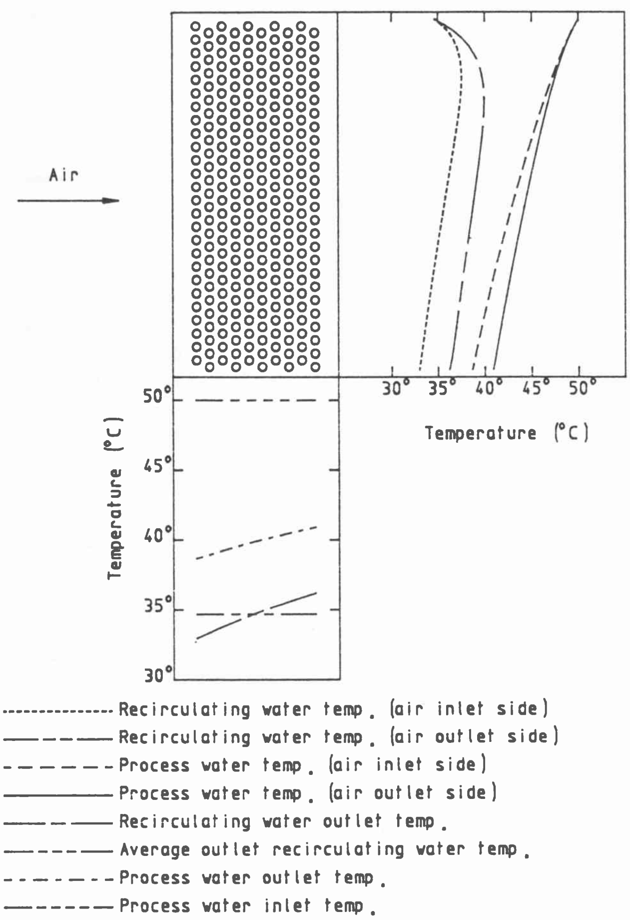

Figure 6 – Temperature distribution in horizontal airflow with process water flowing from Top to Bottom

(2) *Vertical cooler*

Here a vertical airflow cooler similar to the example cited by Mizushina [2] is considered with the dimensions and conditions given below.

Outside diameter of tubes, d_o	=	34,0 mm
Inside diameter of tubes, d_i	=	27,6 mm
Width of cooler, W	=	1,395 m

Length of cooler tubes, L	= 1,385 m
Number of tubes across width	= 20
Number of tubes rows along cooler height	= 13
Atmospheric pressure	= 101 325 Pa
Dry bulb temperature	= 28 °C
Wet bulb temperature	= 23,71 °C
Airflow rate	= 6,008 kg/s
Recirculating water flowrate	= 4,5833 kg/s
Process water flowrate	= 2,778 kg/s
Process water inlet temperature	= 50 °C

The results obtained for single element tubes and tubes divided into five elements are given in table 2.

	n = 1			n = 5		
Method	t_{wi}	ρ_o	Q	t_{wi}	ρ_o	Q
	°C	kg/m³	kW	°C	kg/m³	kW
Merkel	29,30	1,144	180,685	29,29	1,144	180,797
Poppe	29,51	1,143	178,232	29,51	1,143	178,343

Again there is little difference between the two solutions. The outlet recirculating water temperature distribution is given in figure 7 and it can be seen that there is hardly any variation in this particular case.

Mizushina obtained 174 kW for this example but did not state what fouling factor was used in his solution.

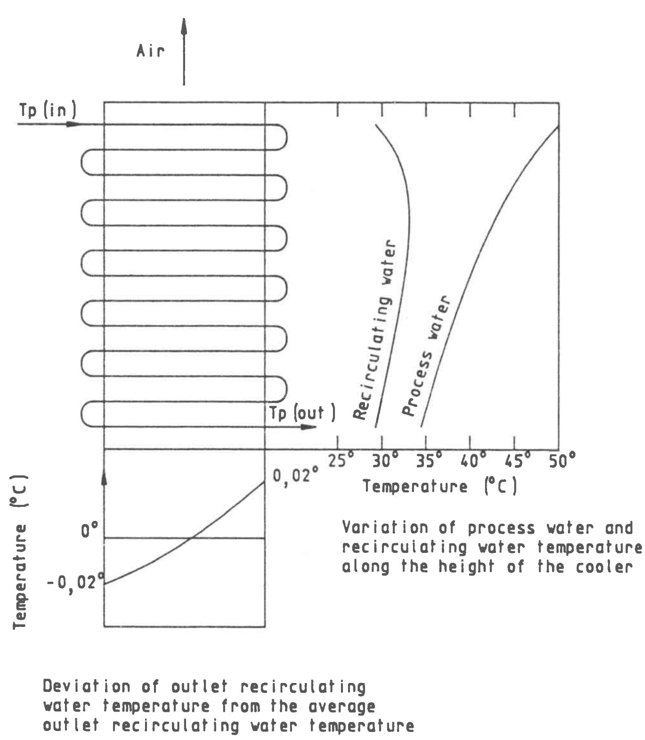

Figure 7 – Temperature distribution in vertical airflow cooler using two-dimensional analysis model

Application to natural draught tower

The method described here has been incorporated in a program to determine the performance of a natural draught tower using 18 meter high tube banks around the base of a large tower (figure 8).

In this particular case a cooling capacity of 490 kW compared to 900 kW for the open circuit tower under the same conditions was found.

Conclusion

An improved model for the analysis of all types of evaporative coolers using the correct thermodynamic and conservation of mass equation has been described. While the results obtained with this method differ marginally from those using a Merkel approach at the water-air interface the program can be used with confidence in any situation.

When analysing vertical airflow devices a one-dimensional approach is completely adequate for an accurate prediction of the performance of an evaporative cooler.

There is an acute need for data on mass transfer coefficients in all types of evaporative cooler. Such data can only be obtained experimentally.

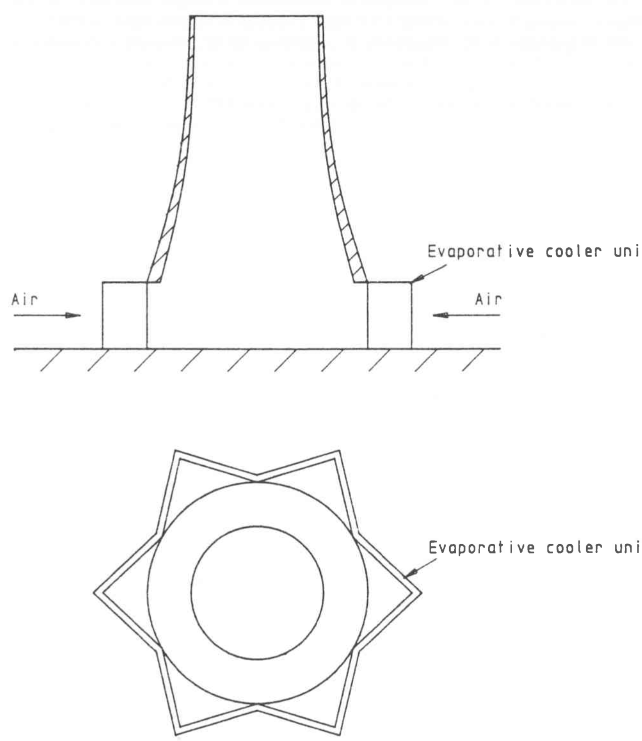

Figure 8 – Example of application of horizontal airflow model in large cooling tower

Acknowledgements

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