A Procedure for the Design or Rating of Counterflow Evaporative Cooler Cores

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A computerised method for determining the size of evaporative cooler cores based on the method of Mizushina is described. An additional program using the above procedure can be used to rate the performance capabilities of a given coil with specified inlet conditions. Block diagrams for both procedures are given and problems encountered in using the method are discussed. Some example calculations are also given. Although the principles of the method have previously been described in the literature the author has attempted to point out some of the problems and pitfalls experienced in developing a successful algorithm for the design or rating of evaporative coolers.

Nomenclature

		Units
a	area per unit volume of coil outside tubes	[m²/m³]
a′	area per unit volume of coil air-water interface	[m²/m³]
A	area inside tubes	[m²]
A′	area at air-water interface	[m²]
B	coil breadth	[m]
C	specific heat	[J/kg K]
D	tube diameter	[m]
f	fouling factor	[m² °C/W]
i	enthalpy	[J/kg]
K	mass transfer coefficient	[kg/s per J/kg]
L	length of coil	[m]
m	flowrate	[kg/s]
NH	number of rows, horizontally	
NV	number of rows, vertically	
p	pitch	[m]
q	heat transfer rate	[W]
R	Reynolds number	
T	temperature	[°C]
U_0	overall heat transfer coefficient	[W/m² °C]
v	specific volume	[m³/kg]
V	velocity	[m/s]
Z	coil height	[m]
α	heat transfer coefficient	[W/m² °C]
μ	viscosity	[kg/ms]

Subscripts

a	air-water vapour mixture
ai	inlet air (at bottom of tower)
apfo	saturation value corresponding to process fluid outlet temperature
arwo	saturation value corresponding to recirculating water outlet temperature
ao	outlet air
aos	outlet air, saturated condition
H	horizontal
V	vertical
p	preferred
pf	process fluid
pfi	process fluid inlet condition (top of coil)
pfo	process fluid outlet condition
rw	recirculating water
rwi	recirculating water inlet condition
rwo	recirculating water outlet condition
wbi	wet bulb at air inlet

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Introduction

The design of conventional evaporative cooling tower cores is at this stage fairly well documented and although sophisticated computer design techniques do exist, the methods used can be considered as text-book material. However, this is not yet the case for evaporative coolers and condensers, the reason being that it is difficult to produce an accurate closed form design solution for these devices.

For brevity evaporative coolers or condensers are seen as multi-tube heat exchangers with or without fins situated inside a wet cooling tower and serving the dual purpose of heat exchanger and fill-material.

There is a considerable lack of design information on this type of apparatus due to its relative complexity compared to the conventional heat exchanger or cooling tower. It is in fact a three-fluid heat transfer device utilising both heat and mass transfer mechanisms. Because of its complexity closed form solutions to determine outlet conditions or sizes are not possible. It is therefore essential to resort to rather complicated computer solutions, even when using approximations such as Merkel's theory on the air side.

Some of the methods found in the literature are discussed and a procedure to determine either core sizes or outlet conditions is described with particular reference to the problems experienced in finding a solution. Some practical examples are also given.

Although the method is restricted to counterflow coolers it could easily be extended to accommodate crossflow exchangers. Little effort would be required to modify the program to accommodate evaporative condensers.

Available design information

The design of an evaporative cooler/condenser incorporates the use of conventional heat transfer theory on the process fluid or refrigerant side, up to the air-water interface, where separate heat and mass transfer equations or the Merkel equation are applied.

The most applicable design methods for conventional tube or finned tube exchangers are those described by Leidenfrost and Korenic [1, 2], Mizushina et al [3] and Webb [4]. The former authors used a more rigorous approach, keeping the heat and mass transfer equations separate in their analysis. However, the assumption of a Lewis number of unity was employed. In contrast Mizushina employed the Merkel (see for example Stoecker and Jones [13]) approximation on the air side. The methods of both groups require the simultaneous numerical integration of energy equations for the air, process fluid and recirculating water to obtain a solution. In doing so the relevant heat or mass transfer equations are also employed.

Kreid et al [5], have proposed an approximate method whereby the heat and mass transfer equations are manipulated to produce an overall heat transfer equation related to enthalpy difference. While this method is possibly useful for a first order approximation it contains some drastic assumptions and is therefore not recommended for design purposes. A similar approach has also been suggested by Perez-Blanco and Linkous [6].

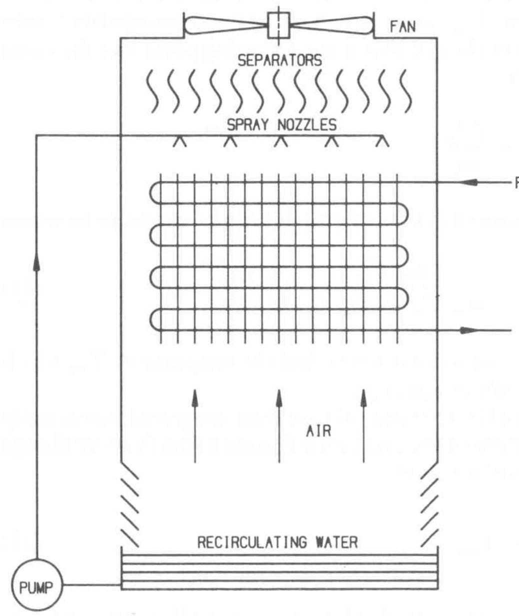
In this article a method similar to that of Mizushina et al [3] is described with a considerable relaxation of the conditions proposed by them. While it is perhaps less accurate than the method of Leidenfrost and Korenic [1, 2], it should be borne in mind that an integral number of pipe rows must usually be determined and the answers are unlikely to differ. However, when determining the capability of a given cooler of fixed dimensions it is very likely that Leidenfrost's method will produce a more accurate solution.

Theoretical background

While the previous authors have adequately described the theory behind evaporative condensers or coolers, a short resumé is given for clarity.

In order to analyse such a cooler, energy equations must be set up for all three fluids, together with the relevant heat or mass transfer equations. In this case a counterflow tower with air flowing upwards and both process fluid and recirculating water flowing downwards is considered (see Figure 1). As this procedure concerns the core design, it is irrelevant whether the tower is of the forced or induced draught type, although these options may well have definite practical implications. The method can be applied to either bare tube or finned tube cores.

A horizontal one-dimensional slice (Figure 2) taken through the tower is considered.



Figuur 1 – Counterflow evaporative cooling tower (induced draught type)

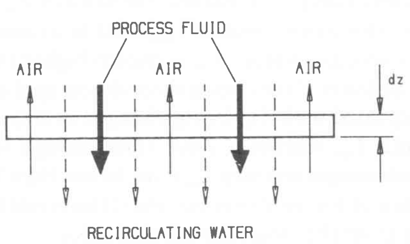


Figure 2 – Schematic incremental slice through coil

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

$$dq_{pf} = U_o.(T_{pf} - T_{rw}).dA \qquad (1)$$

with the overall heat transfer coefficient

$$U_o = 1/(D_o/D_i.\alpha_{pf} + 1/\alpha_{rw} + f) \qquad (2)$$

f is a fouling factor which can account for fouling on either side of the tube while α_{rw} is a heat transfer coefficient related to the outside tube area.

On the other hand the heat transfer at the air-water interface is given by:

$$dq_a = K.(i_{rw} - i_a)\, dA' \qquad (3)$$

Here the elemental area dA' is used since it may differ considerably from dA when fins are employed. However, in a particular cooler the ratio dA/dA' will be constant throughout the coil.

From the energy equation:

$$dq_{pf} = \dot{m}_{pf}.C_{pf}.dT_{pf} \qquad (4)$$

$$dq_a = \dot{m}_a.di_a \qquad (5)$$

$$dq_{rw} = \dot{m}_{rw}.C_{rw}.dT_{rw} \qquad (6)$$

The above three equations are in turn coupled by:

$$dq_{pf} = dq_{rw} + dq_a \qquad (7)$$

so that

$$\dot{m}_{pf}.C_{pf}.dT_{pf} = \dot{m}_{rw}.C_{rw}.dT_{rw} + \dot{m}_a.di_a \qquad (8)$$

In all the above equations, (4) to (8), the mass flow rates are considered constant, which is a reasonable approximation since evaporation is usually about 1% of the recirculating water flow rate. The temperature and enthalpy changes are also taken as positive from bottom to top.

The elemental areas can also be expressed in terms of the element volume and area per unit volume.

$$dA = L.\ B.\ a.dz \qquad (9)$$

$$dA' = L.\ B.\ a'.dz \qquad (10)$$

A further three equations are now derived for integration purposes.
From equations (1), (4) and (9)

$$\frac{dT_{pf}}{dz} = (U_o.a.\ B.\ L/\dot{m}_{pf}.C_{pf}).(T_{pf} - T_{rw}) \qquad (11)$$

from (3), (5) and (10)

$$\frac{di_a}{dz} = (K.a'.B.L/\dot{m}_a).(i_{rw} - i_a) \qquad (12)$$

and from (8)

$$\frac{dT_{rw}}{dz} = (\dot{m}_{pf}.C_{pf}/\dot{m}_{rw}.C_{rw})\frac{dT_{pf}}{dz} - (\dot{m}_a/\dot{m}_{rw}.C_{rw}).\frac{di_a}{dz} \qquad (13)$$

Design procedure

Practical Considerations

In establishing a feasible design approach certain practical aspects of the cooler must be considered. They are, amongst others, acceptable process fluid flow rates, recirculating water flow rates, tube sizes and air velocities. While Mizushina's [3] program allows for a number of tube sizes, it is restricted to triangular arrangements with a pitch of two diameters. In this program the tube size is chosen depending on availability of material.

Their recommended interior and exterior flow rates are also acceptable, but may be deviated from depending on the shape of cooler required (e.g. square or oblong). Such deviations affect the process fluid or recirculating water Reynolds numbers, and therefore the heat transfer coefficients, slightly. Inadequate velocities inside the tubes may cause fouling problems, in which case smaller diameter tubes can be selected.

A recirculating water flow rate of 150 to 200 kg/hr per meter of tube is usually sufficient to give adequate heat transfer coefficients without an excessive rate of evaporation in comparison to this rate. Excessive flow rates should be avoided as the recirculating water tends to inhibit airflow, increasing the required fan power.

While it is possible, using various restricting equations proposed by Mizushina [3], to calculate the air mass flow rate, the maximum practical air velocity must be taken into account. This is usually in the order of 2,5 to 3 m/s in most towers and is determined by the rate of droplet entrainment. In this program the air velocity is chosen by the user and a value of 2,5 m/s is recommended.

Procedure to design a core for given process requirements.

The following section describes the logic of the procedure, which is also summarised in the block diagram in Figure 3. When designing an evaporative cooler core, required inlet and outlet process fluid temperatures are usually given as well as the flow rate. Additional variables required are the environmental restraints in terms of wet and dry bulb temperatures as well as atmospheric pressure.

Tube size is also considered given, while the air velocity and preferred process fluid Reynolds number and water recirculating rate are fixed, but obviously can be changed if required.

In the event of the required process fluid outlet temperature being lower than wet bulb plus a specified limit (in this case 2° C) the program is aborted.

It is assumed that the process fluid is divided amongst a number of parallel tubes, the number of which is determined from the preferred Reynolds number, being the nearest integral value to

$$NH = 4.L/\pi.D_i.\mu_{pf}.R_p \tag{14}$$

The initial breadth of the coil is then found to be

$$B = NH.2.p_h \tag{15}$$

An initial air mass flow rate is calculated by making the length of the coil equal to its breadth.

$$\dot{m}_a = V_a.B^2/v_a \tag{16}$$

Using the tower capacity, as determined from the process conditions, the air outlet enthalpy is given by

$$i_{ao} = i_{ai} + \dot{m}_{pf}.C_{pf}.(T_{pfi} - T_{pfo})/\dot{m}_a \tag{17}$$

The outlet air enthalpy should preferably not exceed the enthalpy of saturated air at the inlet temperature of the recirculating water. Since the inlet and outlet temperatures of the recirculating water must be equal, this in effect means that the saturation temperature corresponding to i_{ao} must be lower than T_{rwo}, which in turn is lower than T_{pfo}. Should this requirement not be met, a new mass flow is chosen with the air outlet enthalpy equal to saturation enthalpy of air at T_{pfo}, thus

$$\dot{m}_a = \dot{m}_{pf}.C_{pf}.(T_{pfi} - T_{pfo})/(i_{apfo} - i_{ai}) \tag{18}$$

A new coil length is then calculated from

$$L = \dot{m}_a.v_a/V_a.B. \tag{19}$$

Should a coil of square cross-section be preferred, the number of tube rows is adjusted to give the same cross-sectional area as determined above, bearing in mind that the preferred Reynolds number is no longer retained. Should the deviation be too great, a smaller diameter tube can be chosen to compensate. If a manufacturer has certain preferred cross-sectional dimensions (e.g. 1 m × 1 m) the next convenient size uwpards is chosen. Either the air mass flow calculated above can be retained with slightly decreased velocity or a new slightly higher mass flow is determined using the same air velocity.

Having determined the cross-sectional dimensions, the applicable heat transfer and mass transfer coefficients can now be calculated using appropriate equations discussed below.

An appropriate recirculating water inlet (or outlet) temperature has now to be chosen and equations (11), (12) and (13) numerically integrated to find a solution. The choice of this value is fairly arbitrary with the knowledge that it must lie between T_{wbi} and T_{pfo}. However, an initial hand manipulated computer program showed that a good starting point was the value of T_{rwo} where

$$\frac{dT_{rw}}{di_a} = \frac{dT_{pf}}{di_a}$$

Using equations (8), (11) and (12) this can be shown to be where

$$\frac{T_{pfo} - T_{rwo}}{i_{arwo} - i_{ao}} = \frac{\dot{m}_{pf}.C_{pf}}{\dot{m}_{rw}.C_{rw} + \dot{m}_{pf}.C_{pf}}\,\frac{K}{U_o}\,\frac{dA'}{dA} \tag{21}$$

A search routine was written to find the temperature T_{rwo} which satisfied the above equation.

Equations (11), (12) and (13) are then integrated numerically from the bottom of the coil upward using a fourth order Runge – Kutta procedure until:

$$T_{rwi} \leq T_{rwo} \tag{22}$$

This program differs from Mizushina et al [3], in that they regarded equation (20) as the upper limiting condition for T_{rwo}, which is not always the case as will be discussed in the next section.

If, when condition (22) is reached, the value of T_{pfi} is not equal to the given value a new value of T_{rwo} must be assumed. If T_{pfo} is less than the specified value, T_{rwo} is chosen higher than the previous value or lower if the opposite condition applies. The integration is repeated with T_{rwo} being changed in steps of 1.0, 0,1, 0,01 etc. until T_{pfo} reaches a value close enough to the given value. At each integration step T_{rw} must be less than T_{pf} to satisfy the second law of thermodynamics and if this is not the case T_{rwo} must be made smaller than the current value.

It was found that T_{pfi} is extremely sensitive to small changes in

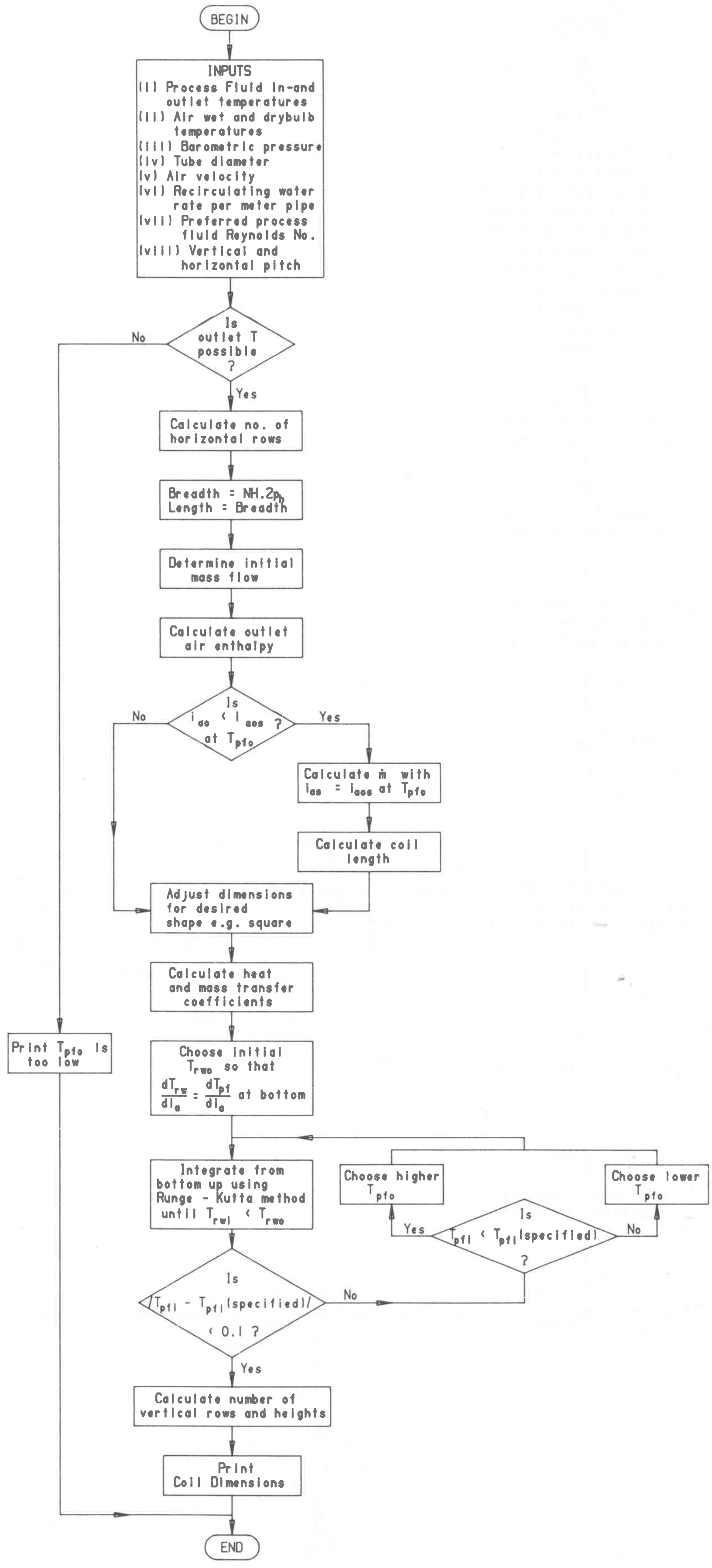


Figure 3 – Block diagram for program to determine evaporative cooler coil size

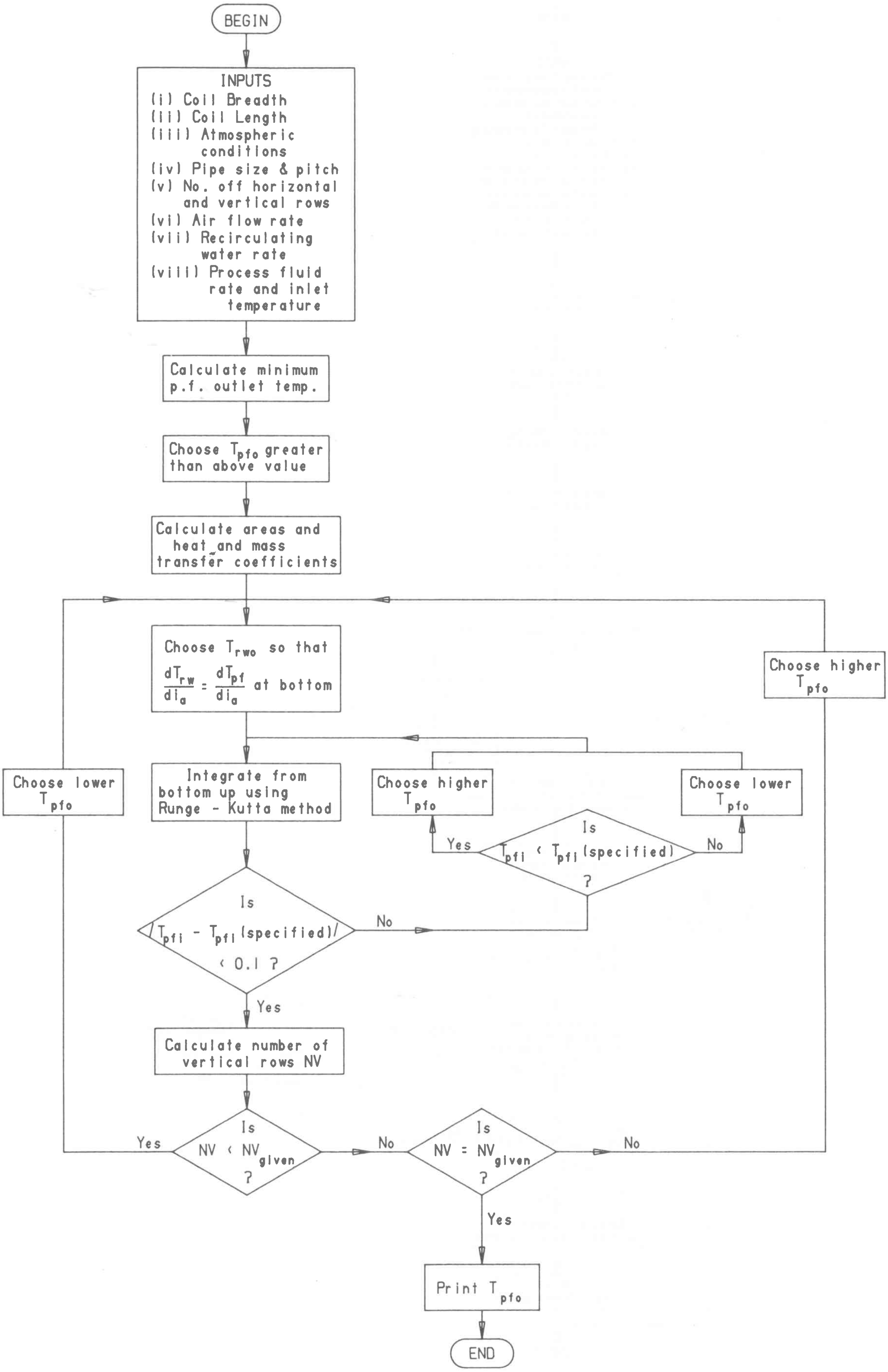


Figure 4 – Block diagram for coil rating program

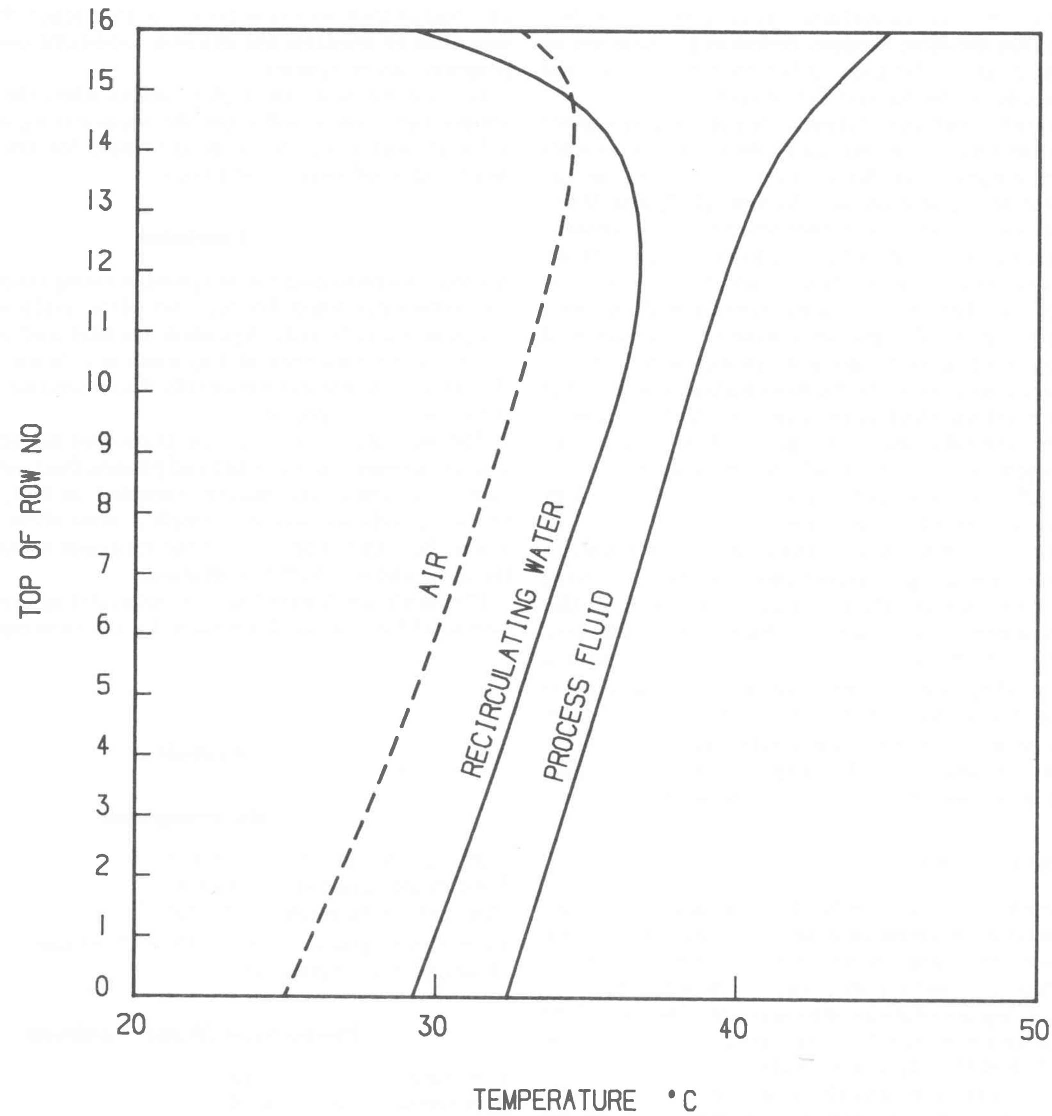


Figure 5 – Process fluid and recirculating water temperatures for example 1

T_{rwo} and it is essential to use double precision throughout the program.

Having determined the value of T_{rwo} which satisfies the process conditions the number of integration steps can be counted to determine the coil height, z. From this the number of vertical rows can be determined being the integer value higher than

$$NZ = z/p_v \tag{23}$$

Procedure to evaluate an existing cooler

The procedure described above has been used successfully to design an evaporative cooler core. However, a more frequent requirement is to determine the performance capability of an existing cooler with given air and process fluid inlet conditions. This is especially the case where a company has a series of cooler models where dimensions cannot easily be changed. Operation at other altitudes and environmental conditions is also obviously of interest.

In this case parts of the above program can be used with some changes to the sequence of calculations. Obviously the dimensions are fixed and the heat and mass transfer equations are calculated using the specified flow rates. The procedure described above to find the coil height is then used iteratively until an exchanger having the same number of vertical rows is found. The program logic is summarised in the block diagram shown in Figure 4.

First, a minimum possible process fluid outlet temperature is determined from an energy balance using the given air and process fluid inlet conditions and flow rates. This usually occurs when the outlet air is saturated at the process fluid outlet temperature. It is then necessary to determine a value of T_{pfo} corresponding to i_{aos} using the energy balance. This value would correspond to the condition of maximum (100%) effectiveness. An initial effectiveness of 80% is then arbitrarily chosen and the number of vertical rows determined for the corresponding value of T_{pfo}. If the number of rows is greater than the specified number a lower effectiveness is chosen and vice versa until the calculated number of rows equals the actual number.

Comment on Heat and Mass Transfer Equations and Fin Efficiencies

Heat transfer equations

The heat transfer equations for the process fluid inside the tubes are the same as for conventional heat exchangers. It is recommended here that the more modern Pethukov [7] equation or Gnielinski [8] equation be used rather than the traditional Dittus and Boelter or Sieder and Tate equations.

The heat transfer coefficient between the outside core surface and the air-water interface presents a problem since it is rather dependent on core geometry. A number of correlations such as those suggested by Leidenfrost and Korenic [1, 2] and Mizushina et al [3] exist, but they are rather limited in their application. Correlations for finned surfaces in particular are a problem since they are linked to a fin efficiency which is generally low for wetted surfaces. The obvious solution to this predicament is that values for a particular geometry have to be determined experimentally if a reasonably accurate answer is desired.

The same argument applies to the mass transfer coefficient at the air-water interface which is analogous to the heat transfer coefficients in multi-tube heat exchangers with or without fins. There are no general correlations which can be used to determine these coefficients although a heat-mass transfer analogy may be resorted to as a first approximation.

Mizushina et al [9] describe an experiment in which both α_{rw} and K are determined using measurements of temperature in the recirculating water film and the air. Since measurement of film surface temperatures presents some problems it would be possible to determine α_{rw} by subtraction if the cooler were operated as a normal cooling tower in one instance (without process fluid) and evaporative cooler in the other. The value of α_{rw} obtained could be based on the outside tube area avoiding the problem of fin efficiency. For the purpose of testing the program Mizushina's correlations for α_{rw} and K were employed.

Air Psychrometric Properties

Whereas Mizushina [3] recommended the use of a linear equation linking saturation temperature and enthalpy, this was considered an unnecessary approximation since it is possible, using a digital computer, to write a short subroutine which takes all variables including air pressure into account. This was done with equations recommended by Johannsen [10] which can also be found in ASHRAE [11] and Schmidt [12].

An additional subroutine was also written making it possible to determine saturation temperature from enthalpy using the routine referred to above.

Physical Properties of Air and Water

Subroutines were written to determine the various physical properties of air-water mixtures and water in terms of pressure and temperature.

Since the properties of the air or water are generally temperature dependent it is desirable that they be calculated at the temperatures applicable at the particular location in the cooler where the various coefficients have to be calculated. This requires that the coefficients be calculated repetitively during the integration process if high accuracy is required. In the examples discussed below properties were calculated at average cooler temperatures rather than local values. It is intended to modify the program to accommodate the latter possibility.

Some Practical Calculations

Example 1 gives the size of a cooler determined with the given process fluid and air conditions while Example 2 gives the result for a cooler of fixed dimensions and given inlet conditions. The tempereature distributions for the first example are shown in Figure 5. It should be noted that the outlet temperature on the graph, 32,3° C, is slightly lower than the specified value, 32,5° C, as an integral number of rows (vertically) has to be chosen. The number of vertical rows used in the above examples is considerably higher than is usually employed in practice. However, this was done to illustrate the extreme conditions over which the program can be applied.

It is also not usual to employ coolers where the wet bulb air temperature profile will cross the recirculating water profile, although such cases should be accounted for when a cooler is employed at off-design conditions.

Conclusion

A successful procedure for designing or rating evaporative coolers has been described. The accuracy of the results obtained with the program is heavily dependent on heat and mass transfer correlations for the recirculating water and the air. These values should be obtained experimentally if a reasonable degree of accuracy is to be expected.

The recirculating water temperature need not be restricted to a value between the air outlet and process fluid outlet temperatures, as can be clearly seen in example 1. In fact, the program for rating coils, as used for example 2, must allow for solutions where the inlet temperature of the recirculating water is below the air outlet wet bulb temperature.

Obviously the lowest limit for recirculating water temperature would be the wet bulb value for the entering air.

Example 1

Tube arrangement

Tube outside diameter = 15 mm
Tube inside diameter = 13 mm
Horizontal tube pitch = 30 mm
Vertical tube pitch = $\sqrt{3}.15$ = 25,98 mm
(Triangular arrangement)

Process Fluid (Water) Conditions

Flow rate = 5 kg/s
Inlet temperature = 45° C
Outlet temperature = 32,5° C

Air Conditions

Dry bulb temperature = 25° C
Wet bulb temperature = 18° C
Barometric pressure = 760 mm Hg

Results

(i) *Dimensions*

No. of rows across = 29
No. of vertical rows = 16
Width = 0,885 m
Length = 1,671 m
Height = 0,416 m

(ii) *Air*

Outlet temperature = 32,21° C (assuming saturation)
Flow rate = 4,134 kg/s.

(iii) *Recirculating Water*

Inlet and outlet temperature = 29,44° C
Flow rate = 2,355 kg/s

(iv) *Coil capacity* = 261,05 kW

Example 2

Coil Dimensions

Tube O.D.	= 15,0 mm
Tube I.D.	= 13,0 mm
No. of rows across	= 26
No. of rows vertically	= 10
Horizontal pitch	= 30 mm
Vertical pitch	= 25,98 mm
Coil Breadth	= Coil Length = 0,78 m

Process Fluid

Flow rate	= 4,5 kg/s
Inlet temperature	= 50° C

Air and Recirculating Water

Inlet dry bulb temperature	= 25,0° C
Inlet wet bulb temperature	= 18,0°C
Barometric pressure	= 760 mm
Air mass flow rate	= 1,85 kg/s
Recirculating water flow rate	= 2,5 kg/s

Results

Process fluid outlet temperature	= 42,1° C
Air outlet temperature	= 35,3° C
Capacity	= 149,2 kW
Recirculating water temperature In/Out	= 36,7°C

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