The design of underground cooling towers†

A. Whillier B.Sc. (Eng.) (Rand) Sc.D. (M.I.T.) (Fellow),*

SYNOPSIS

Recommendations are given for the design of vertical cooling towers for use underground in mines and a method is given for predicting the entering and leaving water temperatures when a given amount of heat has to be dissipated.

NOMENCLATURE

A	cross-section area of cooling tower
B	constant in equation 11
BPF	Barenbrug performance factor$=\eta_a+\eta_w$
C_p	specific heat capacity of liquid water
F_o	basic efficiency factor (figure 3)
F_v	factor correcting for air velocity
F_{wb}	factor correcting for wet-bulb temperature
F_s	factor correcting for the number of screens in the tower
G	mass flow rate of air in tower per unit area, dry basis
H	height of spray-filled portion of tower, m
L	mass flow rate of water entering the tower, per unit area
N	number of screens
Q	rate of heat removal from circulating water
R	capacity factor, equal to ratio of actual water-air ratio to reference water-air ratio
S	sigma heat of air, per unit mass
t_w	water temperature
t_a	wet-bulb temperature of air
v	specific volume of air, dry basis
V	velocity of air up tower $V=G\times v$
w	density of air, kg/m³
L/G	water air ratio (equation 2)
WAR_o	reference water-air ratio (equation 3)
η_a	air efficiency (equation 6)
η_w	water efficiency (equations 5 and 10)
$\triangle p$	pressure drop across tower, mbar
$\triangle t_w$	drop in temperature of the water passing through the tower $\triangle t_w=t_{wi}-t_{wo}=$'range' in computer program.

Subscripts:

w	water
a	air
i	in
o	out

INTRODUCTION

The capacity of underground refrigeration plant installed in South African gold mines during the past few years has grown by about 10 per cent per year, and there is every indication that with mines going deeper this rate will be maintained in the future. An important component of any underground refrigeration plant is the cooling tower which is used to transfer heat rejected by the condenser cooling water into the upcast air. The efficient operation of the cooling tower plays a vital role in the economic operation of the whole plant.

Under typical conditions encountered in underground refrigeration plant, a reduction in the temperature of the condenser by 1 °C results in a reduction in refrigerator power costs of between 2½ and 3 per cent. It is therefore highly desirable that the cooling tower should be designed to cool the condenser cooling water to the lowest temperature possible.

The efficiency of any particular tower depends to a large extent on the degree to which certain rather obvious practical conditions are satisfied. These are simply that the air distribution and the water distribution over the cross sectional area of the tower must be uniform, and that the water droplets should be small, should remain small and should be kept in contact with the air for as long as possible.

Most cooling towers used in mines are of the forced draught vertical counterflow type in which the water droplets fall down through an upward-rising air stream. Internal packing is not used, although several horizontal wire mesh screens are often placed across the tower in order to obtain more uniform air flow. Only this type of tower is considered here.

Generally speaking in cooling tower design a known amount of heat has to be dissipated into a known amount of air entering the tower at a known temperature and humidity. The problem is to predict the entering and leaving water temperatures for a range of assumed water flow rates and assumed tower cross-sectional areas. A method for doing these calculations is given later in the paper but first it is necessary to discuss the many factors that have to be taken into account, and to make specific recommendations for the construction of cooling towers.

DISCUSSION OF PHYSICAL FACTORS INFLUENCING TOWER PERFORMANCE

A uniform distribution of the water in a tower can be assured by arranging the water inlet sprays uniformly across the tower, and by ensuring good air distribution.

*Chief, Applied Mechanics Division, Mining Research Laboratory, Chamber of Mines of South Africa.

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Experience indicates that the water tends to channel if the water loading of the tower exceeds about 10 litre/sec per m^2 of tower cross-section (12 gal/min per ft^2). Hence, towers should not be operated at water loadings exceeding this value.

The most difficult problem in tower design is to ensure that the air flow up the tower is uniform and without excessive turbulence. This can be achieved by ensuring uniform entry of the air into the tower around its entire periphery with as little eddy formation (turbulence) as possible, and by providing wire screens across the tower to assist in maintaining a uniform air distribution and in suppressing eddies or turbulence. A small conical skirt at the bottom of the tower will prevent the formation of a vena contracta, and will hence ensure more uniform flow of air up the tower.

The retention time of the water droplets in the tower can be increased by ensuring that the size of the water droplets leaving the sprays is reasonably small, usually about 2 mm in diameter. An additional benefit of having screens in the tower is that the free-fall of the droplets is interrupted and, hence, the retention time is increased further. The screens also help to break up the water into smaller droplets when localized cascading occurs.

A small but important increase in the retention time of one to two seconds can be achieved by arranging the nozzles to spray upwards rather than downwards.

It is desirable that proven non-clogging spray nozzles be used. A pressure at the sprays of 0.3 to 0.5 bar is usually adequate (depending on the type of spray nozzle used), this pressure being independent of whether the sprays point upwards or downwards since the pressure serves primarily to break the water stream into small droplets. Spray nozzles having holes less than 10 mm in diameter have been found to be easily blocked.

Since the terminal speed, relative to the rising air, of falling droplets of given size is independent of the air speed it follows that increasing the air velocity will keep the drops in suspension in the tower for a longer time. Experience suggests an upper limit of 8 to 9 m/sec above which carry-over of the droplets becomes large(1). Pressure losses at such high air speeds would also be high, so that in practice a somewhat lower air speed, perhaps 6 m/s, is desirable. It is usual to make towers larger than initially required so that should it subsequently become necessary to increase its duty and to increase the air quantity the air speed will still not be much greater than 6 m/s.

The carry-over of water in towers operating at relatively high air speeds can be prevented by providing a somewhat enlarged tower cross-section above the level of the water sprays. The enlarged zone should be sufficient to reduce the air velocity to about one half its value below the sprays. This is to ensure that carry-over will not become a problem at a later stage if it should be decided to increase the quantity of air passing through the tower.

DESIGN RECOMMENDATIONS

The following specific recommendations for the construction of cooling towers are based upon the considerations discussed earlier, and should be followed as closely as possible. Numerical calculations that are needed in the selection of sizes and in the prediction of performance are considered in the next section:

Water loading:

The water loading should be as low as possible, and preferably not more than 10 litre/sec per m^2 of tower area (12 gpm/ft^2). Loadings down to one-third of this value are commonly used.

Air velocity

The air velocity must never exceed 9 m/sec, and should be of the order of 6 m/sec (1 200 ft/min). Speeds between 0.5 and 1.4 of the latter value are commonly used.

Water-air ratio

Generally speaking, as much air as possible should be sent through the tower. For a given air quantity the lowest condenser temperature will be obtained if the actual water-air ratio is about equal to the reference water-air ratio, ie, if the capacity factor R is about equal to 1. Actual water-air ratios, L/G, found in existing underground towers vary from 0.5 to 2.5.

Diameter of tower

The diameter of the tower should be chosen so as to keep the water loading and air velocity less than the maximum values given above. Figure 1 has been prepared to assist in the selection of the tower size. New

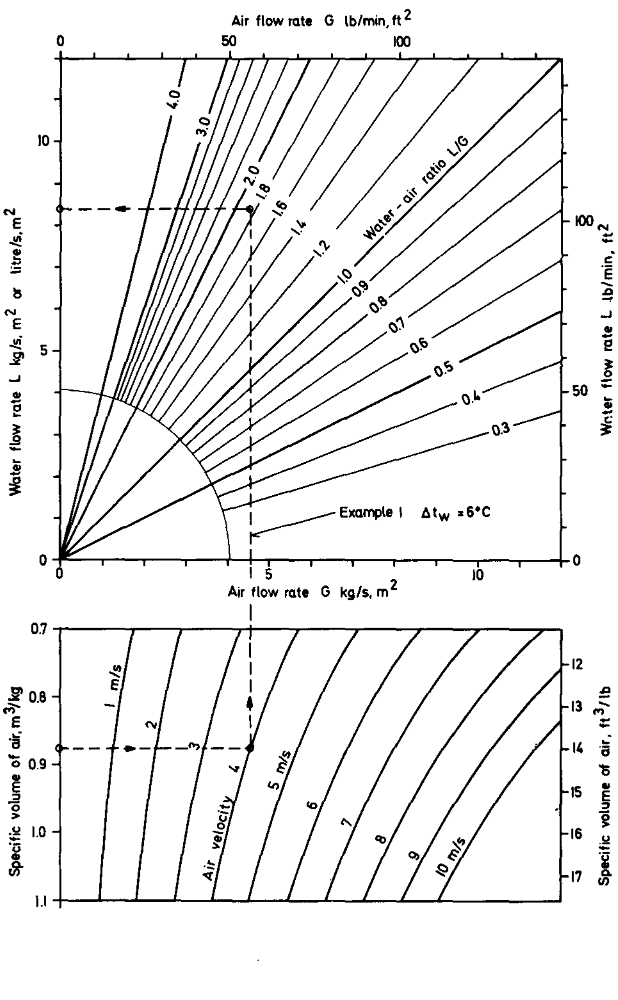

Fig 1—Water and air flow relationships in cooling towers

towers are usually oversized in order to allow for possible later increases in duty.

Height of tower from sump water level to spray level

Usually 4 to 6 diameters. Minimum 12 m, maximum 25 m.

Sprays

Non-clogging spray nozzles of proven design should be used, and should not have holes smaller than 10 mm diameter. The sprays should face upwards, and should be positioned so as to ensure uniform distribution of the water. The pressure at the sprays will depend on the type of spray used. For example, one type of spray nozzle requires a pressure of 0.3 to 0.5 bar.

Tower above spray level

The tower should extend above the sprays for at least 2 tower diameters. The diameter of this portion of the tower should be at least 40 per cent greater than that of the main part of the tower.

Air entry to the tower

The air should enter the tower at sump water level, uniformly around the full perimeter of the tower. The radial entry velocity should be about 4 m/s. A short conical skirt will prevent formation of the unwanted vena contracta.

Screens across the tower

Two or three wire screens are needed across the tower. The lower screen should be 3 to 5 metres above the sump water level, while the top screen should be about 2 metres below the water sprays (to serve as a working platform, if needed). Intermediate screens should be equi-spaced between the top and bottom screens. The screens should be of stainless steel with 3 mm diameter wires spaced about 20 mm apart. There should be no internal stairways, although a simple ladderway is desirable to facilitate periodic inspection, and occasional removal of debris from the screens.

Water temperature range, $\triangle t_w$

Towers are usually designed so as to cool the condenser water by 7 to 9 °C. The water flow rate then depends on the amount of heat to be transferred.

THERMODYNAMIC FACTORS IN COOLING TOWER PERFORMANCE

In vertical counterflow cooling towers the hot water is introduced near the top of the tower through suitable spray nozzles, and falls through the rising air stream. Between 80 and 90 per cent of the cooling of the water droplets is the result of evaporation, with convective cooling being relatively unimportant.

The laws of thermodynamics impose the following limitations on this cooling process:

First law of thermodynamics

Conservation of energy requires that the energy lost by the water must equal the energy gained by the air. with due allowance for the evaporation which takes place.

This can be expressed algebraically as follows:

$$Q = L\,A\,C_p\,\triangle t_w = G\,A\,\triangle S_a \quad . . . \quad (1)$$

where

A = cross-sectional area of the tower

C_p = specific heat capacity of the water

G = air mass flow rate, on a dry basis, per unit area of tower

L = water mass flow rate entering the tower per unit area of tower

$\triangle S_a$ = increase in sigma heat per unit mass of the air passing through the tower

$\triangle t_w$ = decrease in temperature of the water

Q = rate of heat rejection by the water.

In equation (1) only partial allowance is made for evaporation, a small additive correction term on the right-hand side, the magnitude of which is 3 to 4 per cent, being omitted. The considerable complication in the calculations that results from its inclusion is not warranted. Enthalpy rather than sigma heat may be used in equation (1), but the corrective term is then negative and has a magnitude of between 7 and 8 per cent.

Rearranging equation (1) leads to an expression for the water-air ratio, L/G:

$$L/G = \triangle S_a / C_p\,\triangle t_w \quad . . . . \quad (2)$$

Second law of thermodynamics

The limitations imposed by the second law are that the water cannot leave the tower at a temperature lower than the wet-bulb temperature of the entering air, and the wet-bulb temperature of the air leaving the tower cannot exceed the temperature at which the water enters the tower.

Reference water-air ratio

The combination of the limitations imposed by the first and second laws leads to the concept of the reference water-air ratio, $(WAR)_o$, a number which has an important role in determining the performance of cooling towers. This is defined as that water-air ratio at which, heat and mass transfer permitting, the water would be cooled to the entering wet-bulb temperature of the air, while the air would be heated to a wet-bulb temperature equal to that of the entering water.

Thus:

$$(WAR)_o = (S_{wi} - S_{ai})/C_p\,(t_{wi} - t_{ai}) \quad . . . . \quad (3)$$

where

$(WAR)_o$ = reference water-air ratio

t_{ai} = wet-bulb temperature of the inlet air

t_{wi} = water inlet temperature

S_{wi} = sigma heat of air at a wet-bulb temperature equal to t_{wi}

S_{ai} = sigma heat of the entering air.

It is important to note that the reference water-air ratio depends only on the temperatures of the air and water entering the tower, t_{ai} and t_{wi}, and on the barometric pressure. It is independent of the type of tower, the efficiency of the tower, or the actual water and air flow rates.

Since the prediction of the performance of any cooling tower requires that its value be known, curves of reference water-air ratios at various inlet air wet-bulb temperatures are shown in Figure 2 for six typical barometric pressures that are encountered underground, and for various values of $(t_{wi} - t_{ai})$. Experience has shown that most mine cooling towers operate at temperatures such that the reference water-air ratio, $(WAR)_o$, is between 1.45 and 1.65.

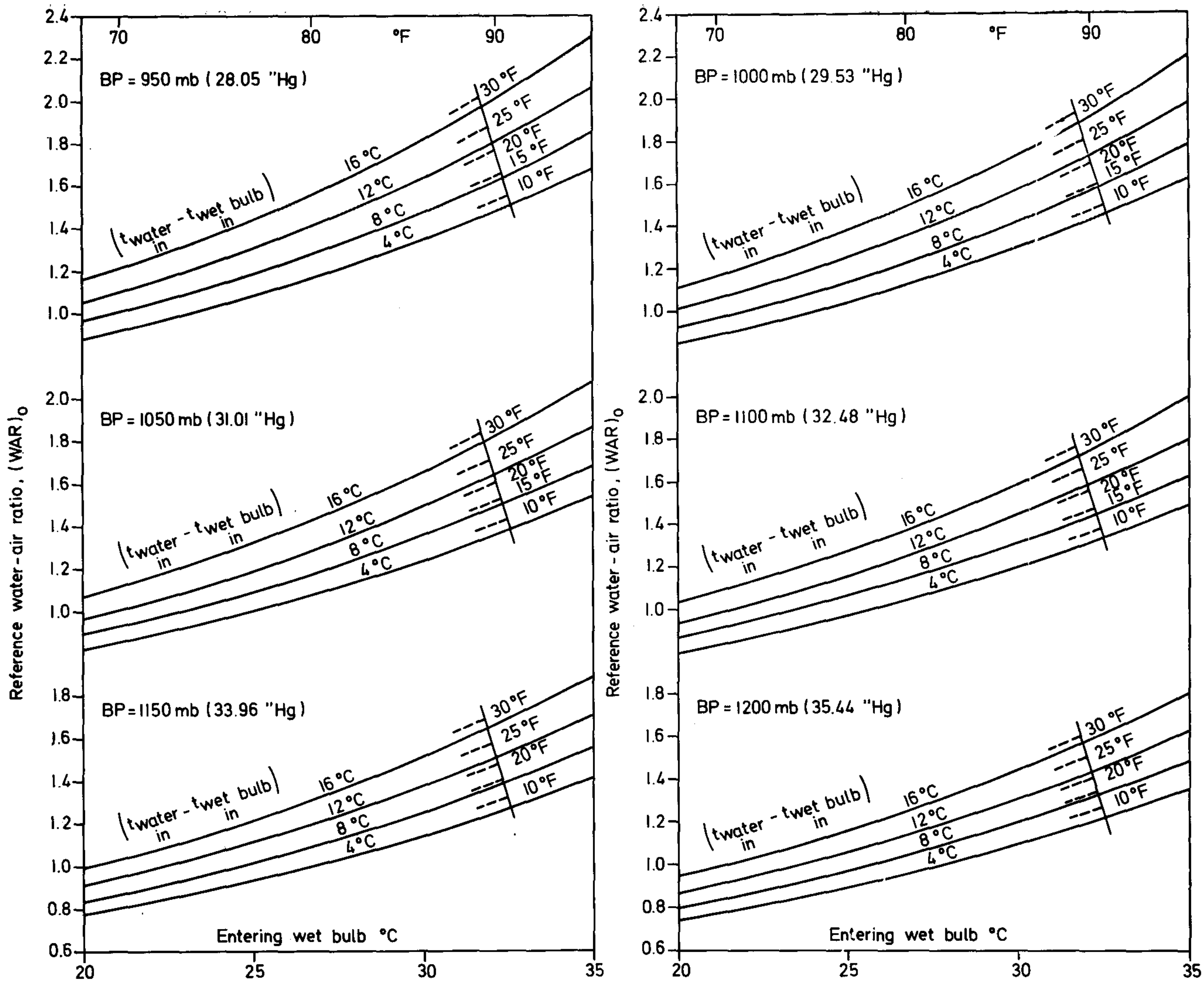

Fig 2—Reference water-air ratio, $(WAR)_o$

Effect of deviations from the reference water-air ratio

At a water-air ratio, L/G, less than the reference value, $(WAR)_o$, the air will not be fully utilized in that it will leave the tower at a temperature considerably lower than its maximum possible wet-bulb temperature. Similarly, at a water-air ratio, L/G, greater than $(WAR)_o$ the air will approach its maximum possible temperature rapidly while the water will leave the tower much hotter than its minimum possible temperature. Thus, in either case, one or other of the fluid streams will not be utilized to its maximum effectiveness. For most economical refrigeration performance it is desirable that the condenser water be as cold as possible, so that water-air ratios less than $(WAR)_o$ are to be preferred.

A very important parameter in cooling towers is the capacity factor, R, being the ratio of the actual water-air ratio, L/G, to the reference water-air ratio, $(WAR)_o$. Thus:

$$R = (L/G)/(WAR)_o \qquad \ldots\ldots\ldots \quad (4)$$

Water and air efficiencies of the cooling tower

The performance of a cooling tower is usually defined in terms of its water efficiency and its air efficiency.

The water efficiency, η_w, is defined as the ratio of the actual drop in temperature of the water passing through the tower, to the maximum possible drop in temperature as imposed by the second law. Thus:

$$\eta_w = (t_{wi}-t_{wo})/(t_{wi}-t_{ai}) = \triangle t_w/(t_{wi}-t_{ai}) \quad . \quad (5)$$

The maximum possible value of the water efficiency is 1. However, when the ratio R is greater than 1, the maximum possible water efficiency is always less than 1, being equal to $1/R$.

The air efficiency, η_a, is defined in terms of sigma heats, being the ratio of the actual increase in heat content of the air to the maximum possible increase in heat content as imposed by the second law. Thus:

$$\eta_a = (S_{ao}-S_{ai})/(S_{wi}-S_{ai}) \qquad \ldots\ldots \quad (6)$$

The maximum possible value of the air efficiency is equal to R when R is less than 1, and is equal to 1 if R is greater than 1.

With the aid of equations (2) and (3) the relationship between the air and water efficiencies is easily shown to be:

$$\eta_a = R\,\eta_w \qquad \ldots\ldots\ldots \quad (7)$$

An implied limitation on equation (7) is that neither η_a nor η_w can be greater than 1.0.

Other tower performance factors

Barenbrug[2] has suggested that a more meaningful measure of the performance of a cooling tower is to take the sum of the air and water efficiencies to yield a performance factor, *BPF*. Thus:

$$BPF = \eta_w + \eta_a \quad . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

With the aid of equation (7) this can also be written as:

$$BPF = \eta_w (1+R) \quad . \; . \; . \; . \; . \; . \; . \; . \quad (9)$$

The maximum possible value of the Barenbrug performance factor is equal to 2, although this can occur only when R=1.

Griesel[3] has proposed a somewhat different performance factor, the numerical value of which is always 4 to 8 per cent larger than Barenbrug's performance factor. It is interesting to note that the Griesel and Barenbrug factors would be identical if the relationship between sigma heat and wet-bulb temperature were linear. The use of Griesel's factor involves additional calculations and offers no particular advantages so it will not be referred to again.

A modified form of the water efficiency, called the 'tower' efficiency is sometimes used, its merit being that its maximum possible value is 1 at all values of R. At values of R less than 1, the tower efficiency is equal to the water efficiency. At values of R greater than 1, the tower efficiency is equal to R times the water efficiency, that is, it is equal to the air efficiency. Because it is seldom used in practice, the tower efficiency will not be referred to again; it is mentioned only for the sake of completeness.

PREDICTING THE PERFORMANCE OF COOLING TOWERS

From the above equations it will be appreciated that for a known amount of heat to be dissipated (with known air flow rate, assumed water temperature range $\triangle t_w$, and known inlet air conditions), the prediction of the entering and leaving water temperatures requires that the water efficiency of the tower be known. The problem therefore is to be able to predict the water efficiency in advance, once the physical shape of the cooling tower has been chosen. The method given below has been derived from a study of the performance of a large number of existing mine cooling towers, and represents a prediction of the efficiency that would be attained (and even exceeded in some instances) provided that the cooling tower is constructed according to the recommendations given earlier.

The water efficiency, η_w, depends primarily on the capacity factor R, and to a lesser extent on the air velocity, the entering air wet-bulb temperature, and the number of screens. Its value can be predicted from equation (10):

$$\eta_w = F_o \, . \, F_v \, . \, F_{wb} \, . \, F_s \quad . \; . \; . \; . \; . \; . \quad (10)$$

where

F_o = basic factor corresponding to the actual water-air ratio

F_v = factor correcting for the air velocity

F_{wb} = factor allowing for the wet-bulb temperature of the entering air

F_s = factor allowing for the number of screens in the tower.

Numerical values of these factors may be read from the graphs in Figure 3.

The basic factor, F_o, is by far the most significant. In order to determine F_o it is necessary to use Figure 2 to make a preliminary estimate of $(WAR)_o$ in order to calculate R.

Having calculated the water efficiency from equation (10), the temperature of the water entering the tower can then be determined from equation (5), since $\triangle t_w$ has been fixed in advance, by assumption.

With all temperatures known it is now possible to determine more exactly the reference water-air ratio, $(WAR)_o$, using Figure 2 or equation (3), and to repeat the calculation in order to determine more precise values of the water temperatures.

Usually it will be desirable to repeat the entire calculation for a range of assumed values of air speed and of $\triangle t_w$, in order to provide a basis for the selection of the area of the cooling tower and of the value of $\triangle t_w$ and, hence, of the total water flow rate through the condenser. In the Appendix a listing is given of a computer program which enables these calculations to be done rapidly for a range of conditions. An example of the computer print-out is also given.

It is desirable to design the cooling system so as to yield the lowest possible average water temperature, $(t_{wi}+t_{wo})/2$, since this will yield the lowest possible condenser temperature for the refrigeration plant. Other constraints, such as the power costs for water circulation and for overcoming the pressure drop of the air flowing through the tower must of course also be taken into account in choosing the optimum operating condition.

Pressure drop across cooling towers

The air pressure drop across the cooling tower is partly due to friction, partly to the losses that occur whenever air changes direction or speed, and partly to the falling water droplets. The following equation may be used for estimating this pressure loss:

$$\triangle p = [B+(2\times N)]\,(w/1.2)\,(V/12.9)^2 + 0.1\,(L\times H)/(12-V) \quad . \; . \; . \; . \; . \quad (11)$$

where

$\triangle p$ = pressure drop, mbar

V = air speed in tower, m/s

H = height of spray-filled portion of tower, m

L = water loading, kg/s, m²

w = density of air, kg/m³

N = number of screens or obstructions in tower

B = constant depending on the tower layout.

The constant B varies considerably from one tower to another, a typical value being 10. It can be as low as 5 or as high as 20. Its numerical value is best determined by measurement on actual towers.

Equation 11 may be used when it is necessary to determine the fan pressure and fan power needed to drive the air through the cooling tower.

In the computer program given in the Appendix the pressure drop across the tower is computed on the assumption that B=10 and H=25 m, these being typical

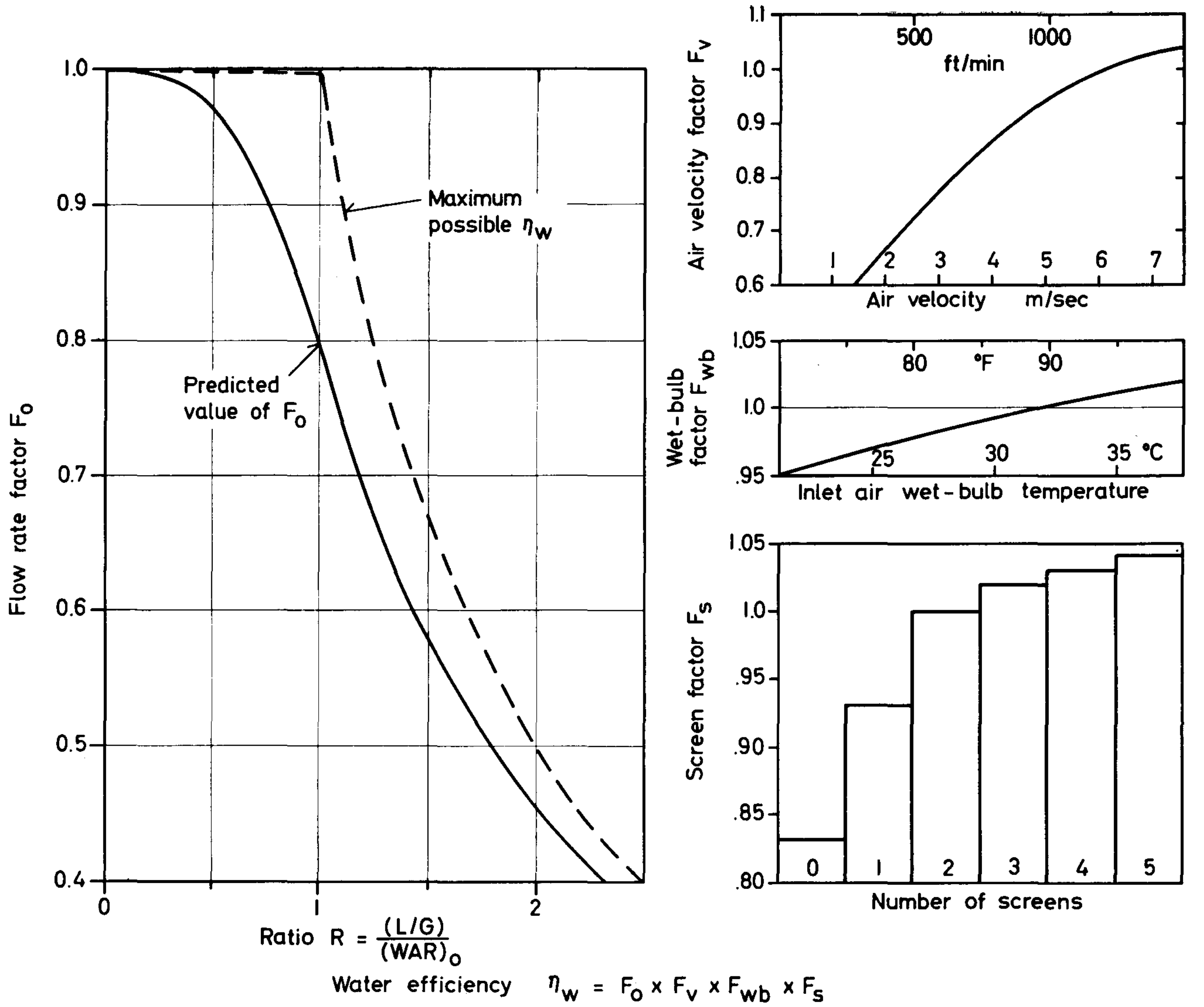

Fig 3—Curves for predicting water efficiency

values for underground cooling towers in South African gold mines.

EXAMPLE

In order to illustrate the method of calculation, the case will be considered of a nominal 3 500 kW (1 000 ice ton) capacity refrigeration plant, the condenser of which must reject Q=4 220 kW of heat. Three screens are to be installed in the tower.

Air is available at 31 °C wet-bulb and 33 °C dry-bulb. The barometric pressure is 1 050 mb and the maximum quantity of air available is 80 m^3/sec at a specific volume of 0.872 m^3/kg.

There is some latitude in the choice of the cross-sectional area of the tower, although to allow for possible future expansion of the refrigeration plant it is usual to choose a tower somewhat larger than the minimum that is required initially. In this instance the area will be chosen so as to give an air velocity in the tower of 4 m/s, necessitating a tower of cross-sectional area 20 m^2.

Since the water flow rate is not specified it is usual to repeat the calculations for several assumed values so chosen as to give a drop in temperature of the water in the range 6 °C to 10 °C.

The reference water-air ratio, $(WAR)_o$, can be estimated from Figure 2, since the barometric pressure and entering wet-bulb temperature of the air are known. At an assumed difference of 10 °C between the entering water temperature and entering wet-bulb temperature of the air, ie, $t_{wi}-t_{ai}=10$ the value of $(WAR)_o$ will be about 1.5. (A trial and error solution is needed in order to determine a more accurate value of $(WAR)_o$. Usually one iteration is sufficient.)

The steps in the calculation leading to the water entering and leaving temperatures are shown in Table 1. The calculations are straight forward and require no explanation.

This same example has been calculated using the computer listed in the appendix, the print-out also being given in the appendix.

ACKNOWLEDGEMENT

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CONVERSION FACTORS FOR SI UNITS

Energy	1 kJ=0.9478 Btu=1 kW/sec
Volume flow	1 m^3/s=2 118 cfm
Velocity	1 m/s=196.86 ft/min
Pressure	1 mbar=0.02953 inch Hg
	1 mbar=0.4015 inch wg
	33.864 mbar=1 inch Hg
Sigma heat	1kJ/kg=0.4299 Btu/lb
Specific heat	1 kJ/kg C=0.2388 Btu/lb F
Mass flow	1 kg/s m^2=12.3 lb/min ft^2
Density	1 kg/m^3=0.06243 lb/ft^3
Refrigeration	1 ice ton=3 516.9 W cooling rate at evaporator.

TABLE 1

EXAMPLE OF PREDICTION OF WATER TEMPERATURES

Heat 4220 kW, Inlet air 31 °C/33 °C/1050 mbar
Air quantity 80 m^3/s, Tower area 20m^2

Assumed	$\triangle t_w$	°C	6	8	10
Equation 1	Water rate L	kg/s,m^2	8.40	6.30	5.04
80/.872 x 20	Air rate G	kg/s,m^2	4.59	4.59	4.59
	L/G	—	1.831	1.373	1.098
Assumed	$(WAR)_o$		1.5	1.5	1.5
Equation 4	R		1.221	.915	.732
Figure 3	F_o		.68	.84	.91
Figure 3	F_v		.87	.87	.87
Figure 3	F_{wb}		.996	.996	.996
Figure 3	F_s		1.02	1.02	1.02
Equation 10	η_w		.601	.742	.804
Equation 5	$(t_{wi}—t_{ai})$	°C	9.98	10.78	12.44
Repeat using better value of $(WAR)o$ from Fig 2 or equation 3					
Figure 2	(WAR_o)		1.49	1.53	1.59
Equation 4	R		1.23	.898	.691
Figure 3	F_o		.68	.85	.925
Equation 10	η_w		.601	.751	.818
Equation 5	$t_{wi}—t_{ai}$	°C	9.98	10.65	12.22
	t_{wi}	°C	40.98	41.65	43.22
	t_{wo}	°C	34.98	33.65	33.22

Assumed			Predicted		Flow rates			Reference	LG / WARO	Predicted using Fig 3		Equation 10	Equation 6	Equation 8	Estimated using Equation 11		
AREA SQ M	VEL M/S	RANGE $\triangle t_w$ °C	TWI C	TWO C	Water L KG/SEC, SQ M	Air G	Water-air ratio LG	water-air ratio WARO	R	FO	FV	WATEFF	AIREFF	BPF	PRESSURE DROP MBAR AIR	WATER	TOTAL
40.00	2.0	6	43.572	37.572	4.200	2.293	1.832	1.5832	1.161	0.704	0.667	0.4773	0.5540	1.0313	0.37	1.05	1.42
40.00	2.0	7	43.754	36.754	3.600	2.293	1.570	1.5900	0.981	0.810	0.667	0.5489	0.5385	1.0874	0.37	0.90	1.27
40.00	2.0	8	44.534	36.534	3.150	2.293	1.374	1.6194	0.847	0.872	0.667	0.5911	0.5005	1.0916	0.37	0.79	1.15
40.00	2.0	9	45.542	36.542	2.800	2.293	1.221	1.6584	0.734	0.913	0.667	0.6189	0.4540	1.0729	0.37	0.70	1.07
40.00	2.0	10	46.705	36.705	2.520	2.293	1.099	1.7049	0.641	0.940	0.667	0.6368	0.4084	1.0451	0.37	0.63	1.00
20.00	4.0	6	41.010	35.010	8.399	4.586	1.832	1.4919	1.224	0.674	0.875	0.5994	0.7337	1.3331	1.47	2.62	4.09
20.00	4.0	7	41.217	34.217	7.199	4.586	1.570	1.4990	1.044	0.770	0.875	0.6852	0.7155	1.4006	1.47	2.25	3.72
20.00	4.0	8	41.665	33.665	6.299	4.586	1.374	1.5146	0.913	0.844	0.875	0.7501	0.6850	1.4351	1.47	1.97	3.44
20.00	4.0	9	42.348	33.348	5.599	4.586	1.221	1.5387	0.796	0.892	0.875	0.7931	0.6315	1.4246	1.47	1.75	3.22
20.00	4.0	10	43.180	33.180	5.039	4.586	1.099	1.5688	0.701	0.923	0.875	0.8210	0.5758	1.3968	1.47	1.57	3.04
13.33	6.0	6	39.949	33.949	12.599	6.879	1.832	1.4560	1.254	0.661	0.999	0.6705	0.8409	1.5114	3.30	5.25	8.55
13.33	6.0	7	40.143	33.143	10.799	6.879	1.570	1.4625	1.070	0.754	0.999	0.7656	0.8192	1.5848	3.30	4.50	7.80
13.33	6.0	8	40.492	32.492	9.449	6.879	1.374	1.4742	0.940	0.831	0.999	0.8428	0.7926	1.6354	3.30	3.94	7.24
13.33	6.0	9	41.054	32.054	8.399	6.879	1.221	1.4934	0.822	0.882	0.999	0.8951	0.7359	1.6310	3.30	3.50	6.80
13.33	6.0	10	41.762	31.762	7.559	6.879	1.099	1.5180	0.726	0.916	0.999	0.9292	0.6748	1.6039	3.30	3.15	6.45
10.00	8.0	6	39.662	33.662	16.798	9.171	1.832	1.4465	1.262	0.657	1.038	0.6927	0.8745	1.5671	5.87	10.50	16.37
10.00	8.0	7	39.853	32.853	14.398	9.171	1.570	1.4528	1.077	0.750	1.038	0.7907	0.8517	1.6424	5.87	9.00	14.87
10.00	8.0	8	40.178	32.178	12.599	9.171	1.374	1.4637	0.948	0.827	1.038	0.8716	0.8261	1.6977	5.87	7.87	13.74
10.00	8.0	9	40.710	31.710	11.199	9.171	1.221	1.4816	0.829	0.879	1.038	0.9269	0.7685	1.6954	5.87	7.00	12.87
10.00	8.0	10	41.385	31.385	10.079	9.171	1.099	1.5048	0.733	0.914	1.038	0.9629	0.7057	1.6686	5.87	6.30	12.17

EXAMPLE OF COMPUTER PRINT-OUT SHOWING PREDICTED WATER INLET AND OUTLET TEMPERATURES, AND PRESSURE DROP ACROSS TOWER, FOR A RANGE OF ASSUMED TOWER AREAS AND WATER FLOW RATES.