DEVELOPMENT OF EVAPORATIVE COOLING TOWERS OF POWER PLANTS

P. GŐSI

Institute for Electrical Power Research
(VEIKI)

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Presented by Prof. Dr. I. SZABÓ

Summary

Evaporative cooling is very often used due to difficulties in water supply. The counterflow, natural draught cooling tower systems built for at least 500 MW turbine performance are the most economical. Due to the high initial costs and energy prices it is essential to improve the accuracy of the calculations and to develop the constructions.

The improved and approximative calculation of evaporative water cooling

Experimental studies are required for the calculations of cooling towers and the following problems have to be solved:

I. Measurements for the determination of the transfer factors of the packing structure (Task I).

II. The calculation of the "thermo-technical behaviour" at different operating conditions, i.e. determination of the so-called performance diagram of the structure with known transfer factors (Task II).

The simplified method worked out by Merkel [7] is still extensively used for the calculation of evaporative cooling. Recently, however, several authors have stated that the difference between the numbers of evaporation (or numbers of transfer units) as calculated without Merkel's neglections (in the following: by the improved method) and by Merkel's equations may be quite great in certain cases. Thus, it is useful for those who develop and design cooling towers to define numerically those limits of error with regard to the temperature of the cooled water that arise upon the application of Merkel's equations.

The following is presumed at the description of evaporative cooling:

— the surfaces of heat and mass transfer are equal
— heat flows only between the water and the air
— the dependence of specific heat upon temperature can be neglected

Table 1

	Measurements for the determination of Me and K_v numbers (Task I)								Calculation of the performance diagram (Task II)						
	Results of the measurements					Number of evaporation		$\frac{K_v - Me}{K_v} 100$	Data of the operation				Cooled water temperature		$\delta t_{w1} =$ $= \lvert t_{w1K_v} - t_{w1Me} \rvert$
	λ	Δt_w °C	t_{w1} °C	t_{G1} °C	x_1 kg/kg	Improved method K_v	Merkel's method Me	%	λ	Δt_w °C	t_{G1} °C	x_1 kg/kg	Improved method t_{w1K_v} °C	Merkel's method t_{w1Me} °C	
1	0.3	4.0	20.50	4.0	0.003589	1.255	0.855	31.8	0.3	20.0	35.0	0.025940	42.24	43.31	1.07
2	0.3	4.0	24.50	4.0	0.003589	0.513	0.421	17.9	0.3	20.0	35.0	0.025940	48.05	48.61	0.56
3	0.3	4.0	39.00	35.0	0.025940	0.406	0.362	10.8	0.3	4.0	4.0	0.003589	26.34	25.75	0.59
4	0.3	20.0	40.00	4.0	0.003589	0.780	0.615	21.2	0.3	20.0	35.0	0.025940	44.88	45.51	0.63
5	0.5	4.0	36.00	35.0	0.025940	0.552	0.502	9.1	0.5	20.0	4.0	0.003589	39.44	38.50	0.94
6	0.5	4.0	36.00	35.0	0.025940	0.552	0.502	9.1	0.5	20.0	35.0	0.025940	44.93	44.80	0.13
7	0.5	4.0	19.00	4.0	0.003589	0.615	0.545	11.4	0.5	20.0	4.0	0.003589	38.03	37.40	0.63
8	0.5	20.0	40.00	35.0	0.025940	1.007	0.897	10.9	0.5	20.0	4.0	0.003589	32.57	31.78	0.79
9	0.7	4.0	14.00	4.0	0.003589	0.923	0.851	7.8	0.7	20.0	4.0	0.003589	30.84	29.89	0.95
10	0.7	4.0	14.00	4.0	0.003589	0.923	0.851	7.8	0.7	20.0	35.0	0.025940	29.21	29.08	0.13
11	0.7	4.0	14.00	4.0	0.003589	0.923	0.851	7.8	0.7	4.0	35.0	0.025940	33.48	33.44	0.04
12	0.7	12.0	29.00	15.0	0.007584	0.905	0.813	10.1	0.7	12.0	4.0	0.003589	26.10	25.59	0.51
13	0.7	12.0	29.00	15.0	0.007584	0.905	0.813	10.1	0.7	12.0	26.0	0.015160	33.07	33.19	0.12
14	0.7	12.0	29.00	15.0	0.007584	0.905	0.813	10.1	0.7	12.0	35.0	0.025940	37.40	37.46	0.06
15	0.7	12.0	31.00	26.0	0.015160	1.228	1.116	9.1	0.7	12.0	4.0	0.003589	23.13	22.50	0.63
16	0.7	12.0	23.00	4.0	0.003589	1.246	1.055	15.3	0.7	12.0	15.0	0.007584	26.34	26.78	0.44
17	0.7	12.0	23.00	4.0	0.003589	1.246	1.055	15.3	0.7	12.0	26.0	0.015160	30.92	31.35	0.43
18	0.7	12.0	23.00	4.0	0.003589	1.246	1.055	15.3	0.7	12.0	35.0	0.025940	35.65	35.97	0.32
19	0.7	12.0	34.00	15.0	0.007584	0.563	0.506	10.1	0.7	12.0	4.0	0.003589	31.83	31.33	0.50
20	0.7	12.0	34.00	15.0	0.007584	0.563	0.506	10.1	0.7	12.0	26.0	0.015160	37.22	37.40	0.18
21	0.7	12.0	34.00	15.0	0.007584	0.563	0.506	10.1	0.7	12.0	35.0	0.025940	40.83	40.94	0.11
22	0.7	12.0	35.00	26.0	0.015160	0.714	0.653	8.6	0.7	12.0	4.0	0.003589	28.76	28.05	0.71
23	0.7	12.0	35.00	26.0	0.015160	0.714	0.653	8.6	0.7	12.0	15.0	0.007584	31.32	31.15	0.17
24	0.7	12.0	35.00	26.0	0.015160	0.714	0.653	8.6	0.7	12.0	35.0	0.025940	38.98	38.92	0.06
25	0.7	20.0	31.00	4.0	0.003589	0.909	0.776	14.7	0.7	20.0	26.0	0.015160	35.90	36.46	0.56
26	0.7	20.0	33.00	26.0	0.015160	1.285	1.162	9.5	0.7	20.0	4.0	0.003589	27.16	26.46	0.70
27	1.0	4.0	10.50	4.0	0.003589	1.260	1.191	5.4	1.0	20.0	4.0	0.003589	24.79	23.82	0.97
28	1.0	4.0	10.50	4.0	0.003589	1.260	1.191	5.4	1.0	20.0	35.0	0.025940	35.94	35.74	0.20
29	1.0	20.0	29.00	4.0	0.003589	0.901	0.787	12.6	1.0	4.0	35.0	0.025940	33.03	33.20	0.17
30	1.0	20.0	29.00	4.0	0.003589	0.901	0.787	12.6	1.0	4.0	4.0	0.003589	12.58	13.14	0.56
31	1.5	4.0	11.00	4.0	0.003589	0.959	0.911	5.0	1.5	20.0	4.0	0.003589	26.36	25.45	0.91
32	1.5	20.0	34.00	35.0	0.025940	1.521	1.406	7.5	1.5	20.0	4.0	0.003589	20.26	19.85	0.41
33	1.5	20.0	34.00	35.0	0.025940	1.521	1.406	7.5	1.5	4.0	35.0	0.025940	31.43	31.44	0.01

— the transfer resistance on the water's side can be neglected as compared to the air's side.

Choosing water temperature as the independent variable the following equation system is received for counterflow (see Fig. 1), if the air is unsaturated:

$$\frac{dh_G}{dt_w}=c_{pw}\frac{\dot{M}_w}{\dot{M}_G}\left\{1+\frac{c_{pw}t_w[x''(t_w)-x]}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pv}t_w)\cdot\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-c_{pw}t_w[x''(t_w)-x]}\right\} \tag{1}$$

$$\frac{dx}{dt_w}=c_{pw}\frac{\dot{M}_w}{\dot{M}_G}\frac{x''(t_w)-x}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pv}t_w)\cdot\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-c_{pw}t_w[x''(t_w)-x]} \tag{2}$$

$$d\dot{M}_w=\dot{M}_G\,dx \tag{3}$$

$$K_v=\int_0^1\frac{\beta_X A\,d\xi}{\dot{M}_w}=\int_{t_{w1}}^{t_{w2}}\frac{c_{pw}\,dt_w}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pv}t_w)\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-c_{pw}t_w[x''(t_w)-x]} \tag{4}$$

While in the region of saturated air:

$$\frac{dh_G}{dt_w}=c_{pw}\frac{\dot{M}_w}{\dot{M}_G}\left\{1+\right.$$

$$\left.+\frac{c_{pw}t_w[x''(t_w)-x]}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pv}t_w)\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-\right.$$

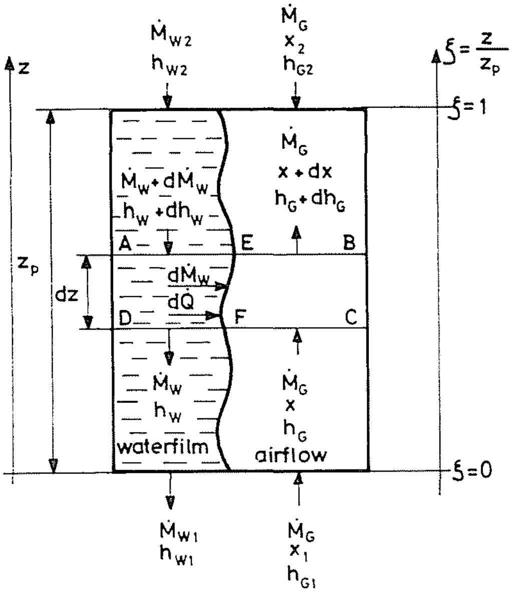

Fig. 1. Counterflow water cooling

$$\leftarrow \overline{\qquad - \frac{\alpha}{\beta_X c_{pm}}[x-x''(t_G)]\,[r_0-t_w(c_{pw}-c_{pv})]-c_{pw}t_w[x''(t_w)-x]} \Bigg\} \tag{5}$$

$$\frac{\mathrm{d}x}{\mathrm{d}t_w}=c_{pw}\frac{\dot{M}_w}{\dot{M}_G}\frac{x''(t_w)-x}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pv}t_w)\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-} \rightarrow$$

$$\leftarrow \overline{\qquad - \frac{\alpha}{\beta_X c_{pm}}[x-x''(t_G)]\,[r_0-t_w(c_{pw}-c_{pv})]-c_{pw}t_w[x''(t_w)-x]} \Bigg\} \tag{6}$$

$$\mathrm{d}\dot{M}_w=\dot{M}_G\,\mathrm{d}x \tag{7}$$

$$K_v=\int_0^1 \frac{\beta_X A\,\mathrm{d}\xi}{\dot{M}_w}=$$

$$=\int_{t_{w1}}^{t_{w2}}\left\{\frac{c_{pw}}{\frac{\alpha}{\beta_X c_{pm}}(c_{pG}t_w-h_G)+(r_0+c_{pw}t_w)\left[x''(t_w)-x\left(1-\frac{\alpha}{\beta_X c_{pm}}\right)\right]-}\right. \rightarrow$$

$$\left. \leftarrow \frac{}{- \dfrac{\alpha}{\beta_X c_{pm}} [x - x''(t_G)] [r_0 - t_w(c_{pw} - c_{pv})] - c_{pw} t_w [x''(t_w) - x]} \right\} \mathrm{d}t_w \tag{8}$$

In order to determine factor $\alpha/\beta_x c_{pm}$ containing the relationship of heat and mass transfer Bosnjakovic's [1] and Poppe's [9] equations can be used in the unsaturated and saturated regions, respectively.

In addition to the above presumptions Merkel neglected the variation of the water mass flow and calculated with the value of $\alpha/\beta_x c_{pm} = 1$. Thus, he received the following simplified energy balance:

$$\dot{M}_w c_{pw} \, \mathrm{d}t_w = \dot{M}_w \, \mathrm{d}h_w = \dot{M}_G \, \mathrm{d}h_G \tag{9}$$

The basic equation for the determination of Merkel-number is:

$$Me = \int_0^1 \frac{\beta_X A \, \mathrm{d}\xi}{\dot{M}_w} = \frac{\beta_x A}{\dot{M}_w} = \int_{t_{w1}}^{t_{w2}} \frac{c_{pw} \, \mathrm{d}t_w}{h''(t_w) - h_G} \tag{10}$$

The state of the leaving air and fog development cannot be calculated by Merkel's method, so the leaving air is usually considered to be saturated.

The solution of the greatly simplified equations, however, in several cases approximates sufficiently that of the equation system without simplifications.

In order to determine the limits of error of Merkel's method a lot of cooling tasks were calculated in the region relevant for evaporative cooling towers. In the course of examinations the air temperature varied between $t_{G1} = 4$ °C and 35 °C, the cooling range varied between $\Delta t_w = 4$ °C and 20 °C. The region limited by λ and by the packing characteristic curves written in the form $Me = C\lambda^k$ can be seen in Fig. 2. The atmospheric pressure is 10^5 Pa in every case. The differential equation systems were solved by a fourth order accurate predictor-corrector method. At the simplified calculations the value of Merkel's integral was determined by Simpson's formula. Analysing the results the most characteristic examples were chosen and are demonstrated in Table 1. Using Merkel's procedure in the cases of the two tasks (Task I and Task II) instead of the improved method errors will always be made, when the data of the two tasks are not exactly the same, i.e. when on the basis of the numbers of evaporation (K_v) the water temperatures are determined at different states of operation. It can be seen that error δt_{w1} remains below 1.1 °C, and values around 0.5 °C are quite frequent.

It can be stated that Merkel's equations can well be used at a considerable part of the practical cooling tasks. In tasks of great significance, however, differences of a few tenth of a degree become more relevant, and on the other hand, the state of the leaving air, the density of the air leaving the packing, fog formation, environmental impact etc. can reliably be calculated only by the application of the improved method. The case is similar at the wide examination of the processes taking place in the cooling towers (e.g. the study of the phenomena taking place in certain parts of the cooling towers, investigating the dependence of evaporation number on the temperature, etc.). In these cases it is practical to apply the improved method by all means. In certain cases Merkel-number may be by 30% less than evaporation number K_r as determined by the improved method.

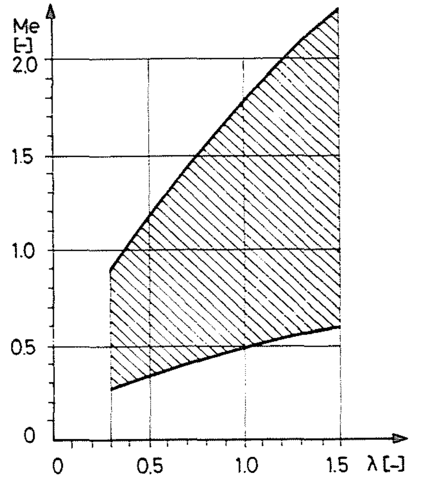

Fig. 2. The examined range

One-dimensional counterflow cooling tower model

At the one-dimensional cooling tower model the following is added to the simplifying assumptions described in the previous paragraph:

— air velocity, the mass flow density of the water and the construction of the packing structure do not vary radially;
— water cooling in the region under the packing can be neglected.

At the calculation and at the acceptance and control measurements the previously described double tasks should be solved in addition to the related fluid mechanical questions.

Two-dimensional counterflow cooling tower model

The methods generally used for the calculations of counterflow cooling towers are one-dimensional, where the heat and mass transfer taking place under the packing is neglected or considered as a security reserve, and the velocity of air along the radius is considered to be constant. As the dimensions increase, however, the radial changes of the water temperature and air state become also significant. These phenomena can be followed only by a two-dimensional (cylindrically symmetric) method of calculation. Although one can read quite often about the above problems in the literature only a few papers can be found about two-dimensional models [2], [4], [5], [6].

The essence of the elaborated method is that the differential equation systems describing the mixed-flow heat and mass transfer taking place in the air distribution region and the counterflow transfer processes connected to this region by an "intermediate edge" are solved without Merkel's neglections, taking the necessary edge conditions into consideration. The air streamlines necessary for the calculations are determined from model or in situ measurements or from calculations based on these measurements. (If the velocity profiles at the packing and at the entrance opening—i.e. the end points of the streamlines—are known from measurements, the streamlines can be calculated by a simple method, too—frictionless flow, neglection of the rain curtain—at the heat technical calculations this caused only insignificant error). The density distribution of the air leaving the packing is received from the two-dimensional heat and mass transfer calculations, and therefrom the average density of the mixed air filling the shell is determined. The highly turbulent flow developing in the slim, hyperbolic shell above the packing is considered to be one-dimensional, thus, the air mass flow is determined from the "draught equation".

Figure 3 shows the bottom part of a counterflow tower. Knowing the flow pattern the differential equation systems describing heat and mass transfer in the range limited by the heavy line have to be solved under the indicated boundary conditions.

Counterflow processes have already been dealt with in paragraph 1. The equations for mixed flow are as follows (see Fig. 4):

For unsaturated air region:

$$\frac{\partial h_G}{\partial s} = \frac{\beta_{xv}}{\rho_G v_G}\left\{\frac{\alpha}{\beta_X c_{pm}}(c_{pG} t_w - h_G) + (r_0 + c_{pv} t_w)\left[x''(t_w) - x\left(1 - \frac{\alpha}{\beta_X c_{pm}}\right)\right]\right\} \tag{11}$$

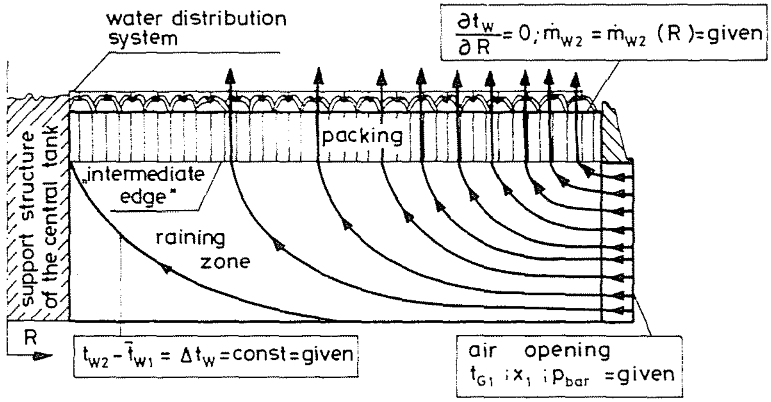

Fig. 3. Two-dimensional calculation of a counterflow cooling tower

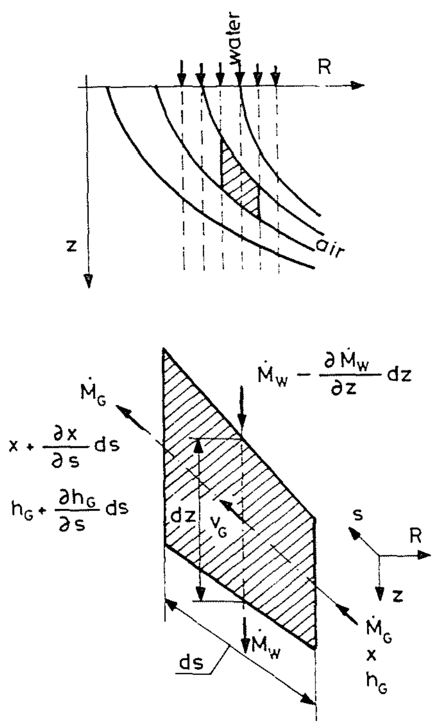

Fig. 4. Mixed-flow water cooling

$$\frac{\partial t_w}{\partial z} = -\frac{\rho_G v_G}{\dot{m}_w c_{pw}} \frac{\partial h_G}{\partial s} - \frac{t_w}{\dot{m}_w} \frac{\partial \dot{m}_w}{\partial z} \tag{12}$$

$$\frac{\partial \dot{m}_w}{\partial z} = -\beta_{xv} [x''(t_w) - x] \tag{13}$$

$$\frac{\partial x}{\partial s} = \frac{\beta_{xv}}{\rho_G v_G} [x''(t_w) - x] \tag{14}$$

The temperature of the air can be determined from the enthalpy and the degree of humidity.

The equations of the fog range are as follows.

Equations (12), (13) and (14) are valid here, too. The equation concerning enthalpy is:

$$\frac{\partial h_G}{\partial s} = \frac{\beta_{xv}}{\rho_G v_G} \left\{ \frac{\alpha}{\beta_X c_{pm}} (c_{pG} t_w - h_G) + (r_0 + c_{pv} t_w) \left[x''(t_w) - \rightarrow \right.\right.$$

$$\left.\left. \leftarrow - x \left(1 - \frac{\alpha}{\beta_X c_{pm}} \right) \right] - \frac{\alpha}{\beta_X c_{pm}} [x - x''(t_G)] \right.$$

$$\left. [r_0 - t_w (c_{pw} - c_{pv})] \right\} \tag{15}$$

In order to solve the equations of mixed flow a second order accurate finite difference method was used. At the counterflow part the predictor-corrector algorythm was fourth order accurate. Iteration loops were applied to meet the edge conditions. The time requirement of an average task on an R–40 computer is roughly 240 sec.

In the following a numerical example is given for the demonstration of the calculation method. The parameters of a cooling tower are the following:

Diameter at the level of the packing:	103.8 m
Height from ground level:	150 m
Height of the air-distribution space:	10.75 m
Height of the air-inlet opening:	8.25 m
Depth of the packing:	1.32 m
Characteristic curve of cooling taking place in the region between the nozzles and the bottom of the packing[1]:	$\mathrm{NTU} = 1.2(1/\lambda)^{-0.4}$

[1] The relationship has been determined for the given packing geometry by the reviewed method of calculation on the basis of laboratory measurements.

$$\mathrm{NTU} = \int_0^1 \frac{\beta_x A \, d\xi}{\dot{M}_G} = \int_{t_{w1}}^{t_{w2}} \frac{\frac{dx}{dt_w}}{x''(t_w) - x} \, dt_w$$

Characteristic curve concerning the cooling of the water drops and spouts under the packing[2]:	$\beta_{xr} = 0.031\lambda^{0.1}\dot{m}_w$
Resistance coefficient of the packing:	22
Temperature of the ambient air:	15 °C
Humidity of the ambient air:	0.0074 kg/kg
Air pressure:	10^5 Pa
Mass flow of the warm water:	15000 kg/s
Cooling range:	10 °C

The flow pattern of the air was determined on a 1:2.5 "model" at the Inota Thermal Power Station whose cooling tower was renewed by plastic packing developed by VEIKI. In the course of the control test in 1981 the velocity distributions above the packing and in the section of the entrance opening were measured. The air streamlines were set on the basis of these velocity profiles and the approximating velocity directions in the raining zone. The velocity distribution measured above the packing is shown in Fig. 5. The results of a laboratory model measurements by the Department of Fluid Mechanics of the Technical University of Budapest is also shown; it corresponds well to the in situ measurements.

The results of calculation are shown in Fig. 6. In an ideal case the temperature of the cold water and the state of the air above the packing are constant along the radius. It can be seen in Fig. 6 that the water temperature and the air state distributions are advantageous. However, it could not be

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For the calculation of dx/dt_w see Eq. (2) and (6). The value of NTU containing the product of $\beta_x A$ can be calculated by the aid of the formula on the right side by the measurements of water temperatures, entering air state and mass flows without knowing surface A that is very difficult to determine.

For a given packing NTU depends primarily upon the air and water mass flow. (β_x mass transfer factor is determined primarily by the Re-number of the air and the water film, thus, by $\dot{m}_G$ and $\dot{m}_w$. Since A does not only mean the wet surface of the packing, but the surface of the droplets formed at the entrance of the water and inside the packing, $A = f(\dot{m}_w)$). Despite the complexity of the process, in most of the cases it is sufficient to use only λ mass flow ratio instead of two mass flows in an exponential function approximating the measurements data in a relatively narrow region.

[2] An approximate relationship determined by cooling tower measurements. Volume mass transfer factor $\beta_{xr} = \beta_x \, dA/dV$ depends upon mass transfer factor β_x and dA/dV, the surface of droplets and water spouts in the unit of volume. Considering the rain curtain as a special cooling packing β_{xr} was determined similarly to NTU. The exponents of λ and $\dot{m}_G$ are small, because the air-side Re-number affecting greatly β_x alone is determined together by the velocity of fall of the droplets and the relatively small velocity of the flowing air.

realized by a constant, only by an outwardly decreasing λ-distribution along the radius. The changes along the radius come to the fore primarily in towers built for at least 500 MW turbine performance. This question will later be touched upon.

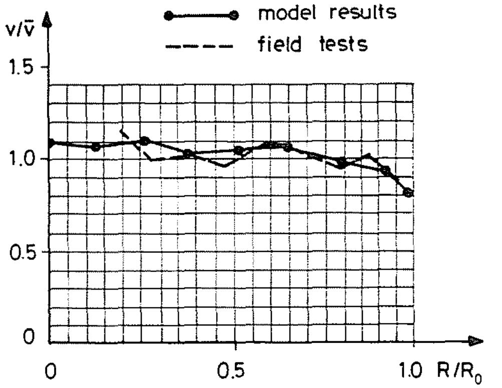

Fig. 5. Velocity distribution above the packing

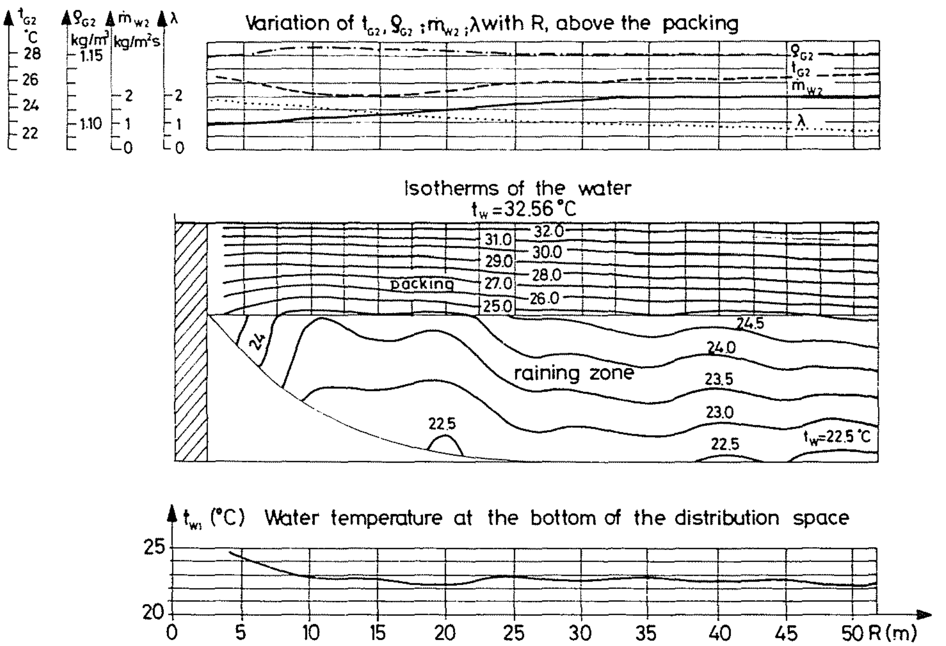

Fig. 6. Results of calculation

Increasing the cooling capacity of cooling towers

Packing development

Packing has a determining role in the price of cooling systems. According to our measurements the cooling effect of the normal plane asbestos-cement sheets can be increased significantly without increasing the amount of the material if the plates are placed in several layers one above the other about 20 mm from each other. The evaporation number of a packing of six layers whose height is 2500 mm is about 15% greater than that of the packing of the unbroken tables, while its resistance is greater by only 5% at the most. The amount of gridlike plastic packings to be built in can also be decreased if the ratio of the drop-type and film-type transfer processes and the shape of the air channels are optimized.

The effect of water distribution

By our measurements the temperature of the cooled water can be decreased by more than 1 °C on the average purely by the careful setting and regular upkeep of the water distribution system, beside unchanged cooling range.

The improvement of the distribution of water temperature and air state

This problem can be analyzed by the two-dimensional method of calculation. Let us return to the calculation results shown in Fig. 6. In order to achieve an advantageous distribution of the water temperature and that of the air state along the radius an outwardly decreasing λ-distribution had to be applied. The value of $\dot{m}_{w2}$ is not constant either, but outwardly increasing. A deleterious region of high water temperature would develop in the middle of the tower not only if $\lambda = \text{const.}$, but also if $\dot{m}_{w2}(R) = \text{const.}$

Making use of the fluid mechanical model measurements published in the literature several calculations were performed by different velocity profiles. On the basis of our examinations the following conclusions can be drawn. In cooling towers of several thousand m^2 area with the same type of packing in the full cross section the radially increasing λ mass flow relationship is very disadvantageous and causes a rather considerable impairment of the cooling effect as compared to the $\lambda = \text{const.}$ state. The ideal solution is a decreasing λ toward the wall of the shell, the most advantageous distribution should be determined individually. The radial distribution of water temperature and the air state can be influenced not only by the water mass flow density (and thus of λ), but also by packing development changing along the radius. In most of the cases, however, it is easier to modify $\dot{m}_{w2}$ distribution, it can be applied to already operating towers, thus, in the end cooling capacity can be improved without additional investment costs. The most advantageous $\dot{m}_{w2}$ distribution, as determined by calculations, can well be approached in practice by three regions of different specific water load. The advantageous effect of this so-called differentiated water distribution is proven e.g. by Soviet industrial experience.

Fluid mechanical questions

An effective, yet not generally used method of decreasing pressure loss on the air's side is the fluid mechanically favourable shaping of the upper edge of the air inlet opening. The evolving flow pattern may be significantly modified and distorted by the landmarks (buildings, other cooling towers etc.) around the tower. It could be seen from the calculations carried out by the two-dimensional model that the radially disadvantageous mass flow relationship leads to a rise in water temperature. Thus, in addition to the decreased amount of cooling air the impairment of cooling effect due to wind is caused by this phenomenon, too [8]. Accordingly, in order to decrease wind effect the cold air break in at the mouth of the shell and the velocity profile deformations developing in the air distribution space and in the cross section of the packing should be restricted. (The former can be realized by a confusor-like shaping of the upper part of the shell, while the latter by a laying where the opening for the entering air is sheltered from the wind).

Non-traditional designs

The cooling capacity of counterflow cooling towers can in certain cases be economically increased by supplementary exchange surfaces, too. It is a possible solution that a mixed flow packing is built in the air distribution space, and a cross-flow cooling surface is put around the entry opening of the air. By the aid of this cross flow part the form of the entry opening will become more favourable fluid mechanically, on the one hand, and the distributions of the water temperature and that of the air state can this way be simply improved, on the other hand. The two-dimensional method of calculation together with the appropriate fluid mechanical model tests may help to develop similar constructions, too. However, because of the complexity of the real process both model and field control tests are essential.

Conclusions

The main results and conclusions are as follows:

— If the calculations of the evaporative cooling towers in the usual range are carried out by the numerical integration of Merkel's simplified basic equation rather than by the improved method, the error concerning the cooled water will remain below 1.1 °C, and errors around 0.5 °C are frequent. Thus, Merkel's method can well be used at a considerable part of the practical cooling problems. At tasks of great significance, however, or at the thorough analysis of the processes (analysis of the temperature dependence on NTU, two-dimensional calculation of counterflow towers, etc.) it is advisable to use the improved method. The Me-number may in certain cases be by 30% less than K_v number of evaporation as calculated by the improved method.

— The results obtained by the two-dimensional (cylindrically symmetric) method of calculation are greatly influenced by the air velocity distribution at the packing. Therefore, it is advisable to determine it by in situ measurements or fluid mechanical model tests. The in situ measurements of a cooling tower of 1350 m^2 area showed that the results of "cold" fluid mechanical model measurements can well be used in the practice. The processes taking place in certain parts of a cooling tower can appropriately be followed by the presented model; however, control measurements are essential.

— The evaporation number of plane asbestos cement sheet packing can economically be increased without increasing the amount of the material if the sheets are placed in several layers one above the other.

— It can be concluded from our calculations that in cooling towers of several thousand m^2 area with uniform packing in their full cross section, it is advisable to realize outwardly decreasing λ mass flow ratio instead of a constant λ along the radius. Increasing λ values toward the wall of the shell cause a rise in water temperature. In practice the ideal λ-distribution—that has to be calculated separately for each tower type—can be approached by three regions at the most with different specific water load. The favourable effect of this differentiated water distribution is proven e.g. by the Soviet industrial experience. On the basis of the analysis of the λ-distributions it can also be stated that the cooling effect impairment due to the wind is greatly caused by the deformation of the velocity distributions at the packing, in addition to the decrease of air mass flow in the cooling tower. Furthermore, the λ values that decrease toward the wall of the shell are more favourable from the aspects of wind effect, possible breakdown of the nozzles and winter operation than the constant distribution.

Nomenclature

R, z	cylindrical coordinates;
z_p	depth of counterflow packing;
ξ	nondimensional depth of counterflow packing;
R_0	radius of the shell at the packing;
s	coordinate in the directions of the air streamlines;
A	transfer surface;
V	volume of packing;
$\dot{M}_G$	massflow of dry air;
$\dot{M}_w$	mass water flow;
$\dot{m}_w$	mass water flow per unit plane area of packing (mass water flow density);
$\lambda = \dot{M}_G/\dot{M}_w$	air to water ratio;
v	air velocity inside the shell, just above the packing;
$\bar{v}$	mean air velocity inside the shell, just above the packing;
v_G	air velocity (see Fig. 4);
$\dot{Q}$	heat flux;
t_w	water temperature;
Δt_w	cooling range;
$\bar{t}_{w1}$	average temperature of cold water in the basin;
h_w	enthalpy of water;
h_v	enthalpy of water vapour;
h_G	enthalpy of air-water vapour mixture;
x	absolute humidity of air;
$x''(t)$	absolute humidity of air at temperature t, at the state of saturation
t_{w1K_v}	temperature of the cooled water calculated by the improved method;
$t_{w1_{Me}}$	temperature of the cooled water calculated by the numerical integration of Merkel's equation;
p_{bar}	atmospheric pressure;
ρ_G	air density;
c_{pG}	dry air specific heat at constant pressure;
c_{pw}	water specific heat at constant pressure;
c_{pv}	water vapour specific heat at constant pressure;
c_{pm}	specific heat of air-water vapour mixture at constant pressure;
r_0	evaporation heat of water at 0 °C;
α	coefficient of heat transfer;
β_x	coefficient of mass transfer defined in terms of difference in absolute humidity;
$\beta_{xv} = \beta_x \, dA/dV$	volume coefficient of mass transfer;
NTU	number of transfer units;
K_v	number of evaporation;
Me	Merkel number;
C	constant;
k	exponent;

Subscripts

1, 2	see Fig. 1, 3, 6.

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