Numerical Study on the Performance Analysis of Plume Abatement Cooling Tower with Dry Type Heat Exchanger

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Key words: Cooling tower, Plume abatement, Dry type heat exchanger, Effectiveness-NTU method

ABSTRACT: This study treats the numerical analysis of performance and design for plume abatement wet/dry cooling tower with a dry type heat exchanger. A two-dimensional analysis is performed using the finite volume method for mechanical draft counterflow and crossflow tower. For a coupling problem between water and air system, a turbulent two phase flow is considered. The Effectiveness-NTU method is used for modeling of the dry type heat exchanger. The parametric simulations such as the relative flowrate of air and attachment length of an air mixer are performed to examine the effect on plume abatement. It is found that if the relative air flowrate ratio and the adequate air mixer type are chosen well in addition to the ratio of water to air flowrate, the loss of cooling capacity and the additional cost are reduced and the plume is abated.

Nomenclature

a : area of transfer surface per unit volume [m^2/m^3]
C : heat capacity [J/kg℃]
f : resistance to air flow [N/m^3]
moisture fraction of air
h : enthalpy of air [J/kg]
J : Jacobian of the coordinate transform
K : mass transfer coefficient [kg/m^2s]
m : mass of water vapor of liquid [kg]
P : pressure [Pa]
$\dot{q}$: rate of heat transfer [W]
T : temperature of air [℃]
t : temperature of water [℃]
U : velocity [m/s]
overall heat transfer coefficient [W/mK]
x, y : horizontal and vertical direction in Cartesian coordinate
x^i : non-orthogonal coordinates
y_i : Cartesian coordinates

Greek symbols

α : Cartesian component of the contravariant base vector
Γ : diffusion coefficient
δ : Kronecker's delta
ρ : density [kg/m^3]

Superscripts

$'''$: per unit volume

Subscripts

AM : air mixer
amb : ambient

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db : dry bulb
dry : dry section
eff : effective
F : water
G : moist air
in : inlet position
max : maximum value
min : minimum value
out : outlet position
v : vapor

1. Introduction

In the refrigerator and heat exchanger systems, efficient heat diffusion is indispensable. There are two major different types of diffusion as following; one is the direct diffusion from the refrigerator to ambient by natural and/or forced convection, and the other method, which is preferable, is to recirculate cooling water through the system whose heat is transferred to ambient air. A cooling tower, using this heat transfer mechanism, has a powerful function to release heat efficiently and to prevent environmental pollution significantly. Recently the cooling towers have been applied to many areas, such as petrochemistry, steel industry, food industry, air conditioning and refrigerating systems. Consequently it is necessary to analyze the performance of cooling towers accurately.

As the concern of environmental protection increases, the most important thing of design is related to antipollution. The plume, which is brought about in the cooling towers, is considered to be a sort of environmental pollution. This plume is visible when water vapor, exhausted through the exit of the cooling tower, condenses in contact with ambient air. Note that the water droplets, evaporated from the cooling process, are "pure" water, in contrast to the very small percentage of drift droplets or water blow out of the air inlets. Strictly speaking, the plume is not an environmental pollution but just an obstacle of a visual field. However, under certain conditions the plume from the cooling tower may give rise to hazards such as fogging or icing and sometimes causes serious problems at the following specific areas; airports, highways, as well as residential buildings. In these reasons most researchers have tried to predict the plume abatement. Buss[1] solved the analytical plume abatement conditions, Campbell[2] used a graphical method of the psychrometric chart in order to obtain the plume abatement conditions for cooling tower with a annular finned heat exchanger. Lately, Miura[3] proposed a NWD (Novel wet-dry) cooling tower and showed the remarkable conditions to reduce plume. All the analysis of those researchers have been done under the assumption that the mass transfer between water and vapor was negligible, which was proposed by Merkel.[4] They have relied on a graphical method using the psychrometric chart to find the plume abatement conditions, too.

In this study, we use the Finite Volume Method and two phase turbulent flow model with considering water evaporation in order to analyze the flow and temperature fields in the cooling tower that has a dry type heat exchanger for plume abatement. A standard $k-\varepsilon$ turbulent model is applied to predict the turbulent flow fields. The moisture fraction differential equation and the enthalpy equation are solved simultaneously as well as the momentum equation, to consider heat and mass transfer between water and vapor. For the analysis of the dry type heat exchanger, the effectiveness-NTU method is applied. To validate our numerical results, they are compared with the experimental and commercial software data, and the efficient and appropriate design conditions to abate plume are obtained.

2. Theory

Assuming that both air and water flow fields are continuous, the general coordinate system is applied to solve the time averaged governing equations. Heat and mass transfer from water to air are considered as the source terms in enthalpy and moisture fraction equation.

2.1 Governing equations

The general equations which have conserved form for air mass, momentum, enthalpy, moisture fraction and water mass and enthalpy are constructed as below. Detail meanings of each term in Eq. (1)~(6) are described in Ref. 5 as well as the standard $k-\varepsilon$ turbulence model constants.

2.1.1 Governing equations for air[5]

- mass conservation

$$\frac{1}{J}\frac{\partial}{\partial x^j}[J\alpha_m^j(\rho U_m)] = \dot{m}_v''' \quad (1)$$

- momentum conservation

$$\frac{1}{J}\frac{\partial}{\partial x^j}[J\alpha_m^j(\rho U_m U_i - \tau_{mi} + P\delta_{mi})] = -f_i - (\rho - \rho_{amb})g\delta_{1i} \quad (2)$$

- enthalpy conservation

$$\frac{1}{J}\frac{\partial}{\partial x^j}\left[J\alpha_m^j\left(\rho U_m h_G - \Gamma_{eff}\frac{\partial h_G}{\partial x^n}\alpha_m^n\right)\right] = \dot{q}''' \quad (3)$$

- moisture fraction conservation

$$\frac{1}{J}\frac{\partial}{\partial x^j}\left[J\alpha_m^j\left(\rho U_m f_G - \Gamma_{eff}\frac{\partial f_G}{\partial x^n}\alpha_m^n\right)\right] = \dot{m}_v''' \quad (4)$$

2.1.2 Governing equations for water[5]

- mass conservation

$$\frac{1}{J}\frac{\partial}{\partial x^1}[J\alpha_1^1(\rho_F U_F)] = -\dot{m}_v''' \quad (5)$$

- enthalpy conservation

$$\frac{1}{J}\frac{\partial}{\partial x^1}[J\alpha_1^1(\rho_F U_F h_F)] = -\dot{q}''' \quad (6)$$

2.2 Concept of plume abatement

For prevention of the plume, the operating line in the psychrometric chart of cooling tower must not cross the saturation curve. Figure 1 shows a typical type of cooling tower with a heat exchanger. Heating the ambient air without increase of its absolute humidity is accomplished when it passes over the dry section (■ → +) and is mixed with wetted air which comes through the wet section (■ → ●) in the cooling tower. Finally, the unsaturated mixed air (×) is exhausted, so the operating line doesn't cross the saturation curve as shown in Fig. 2. As a result, the possibility of plume generation could be decreased.

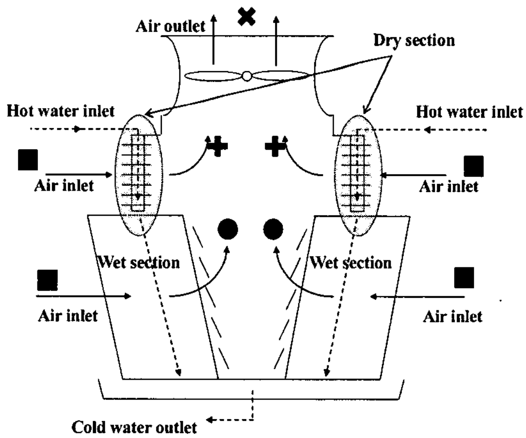

Fig. 1 Schematic diagram of crossflow wet/dry cooling tower.

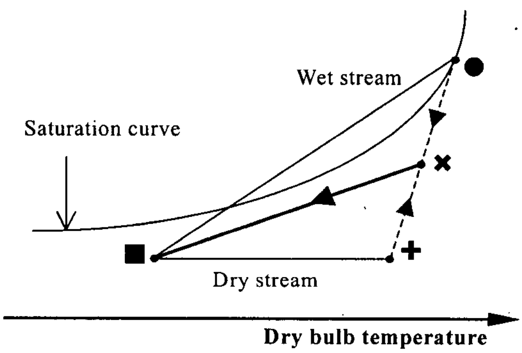

Fig. 2 A skill for plume abatement.

Table 1 The specifications of the cooling tower (unit except inclined angle: m)

Counterflow tower		Crossflow tower	
Length at the base	3.10	Length at the base	9.70
Mean half width	2.66	Mean half width	10.97
Radius at the exit	0.90	Radius at the exit	3.35
Tower height	5.00	Tower height	16.98
Wet section inlet port height	0.67	Fill height	11.98
Distance between wet inlet and fill	0.11	Fill width	5.15
Fill height	0.78	Inclined angle of the inlet louver	8°
Spray height	0.37		
Distance between spray and dry inlet	0.30		
Dry section inlet height	0.50		

2.3 Heat transfer in the dry section

A dry type heat exchanger is composed of vertical pipe arrays with annular fins. Hot water from the condenser runs inside of these pipes and air flows outside. Because the direction of two flows is perpendicular, the effectiveness-NTU method[6] is applied in order to obtain the outlet temperature of two flows.

$$q_{\max} = C_{\min}(t_{in} - T_{db,in}) \tag{7}$$

$$\mathrm{NTU} = UA_{dry}/C_{\min} \tag{8}$$

The effectiveness is expressed as a function of NTU and heat capacity rates. At the outlet of dry section, temperature of air and water, and enthalpy of air may be determined as

$$t_{out} = t_{in} - q_{dry}/C_{water} \tag{9}$$

$$T_{db,out} = T_{db,in} + q_{dry}/C_{air} \tag{10}$$

$$h_{out} = h_{amb} + q_{dry}/G \tag{11}$$

These results are used as the inlet conditions at dry section entrance in numerical calculation for internal region of the cooling tower.

2.4 Heat and mass transfer in the fill region

In order to model the heat and mass transfer between water and air in the fill region, the Merkel relation is employed as below.[4]

$$\dot{q}''' = Ka(h_F - h_G) \tag{12}$$

$$\dot{m}''' = Ka(f_F - f_G) \tag{13}$$

Using the experimental relations,[5] Ka is determined for each type of fill in counterflow and crossflow tower since it is not easy to find K and a separately in various types of fill and tower.

3. Numerical analysis

3.1 Geometric shape and grid system

We have established geometric shapes of the two types of cooling tower—counterflow and crossflow towers. The shape of counterflow tower is based on the BACT-TIF 80 model (Bumyang Air-conditioning Co. Ltd) and that of crossflow tower is based on the Ref. 7. The shapes and dimensions of each tower are shown in Fig. 3, Fig. 1, and Table 1, respectively. The two-dimensional non-orthogonal, non-staggered grid systems are constructed which have 138×55, 80×60 meshes for the counterflow and crossflow type towers, respectively. A SIMPLE (Semi Implicit Method for Pressure Linked Equation) algorithm is applied to link the pres-

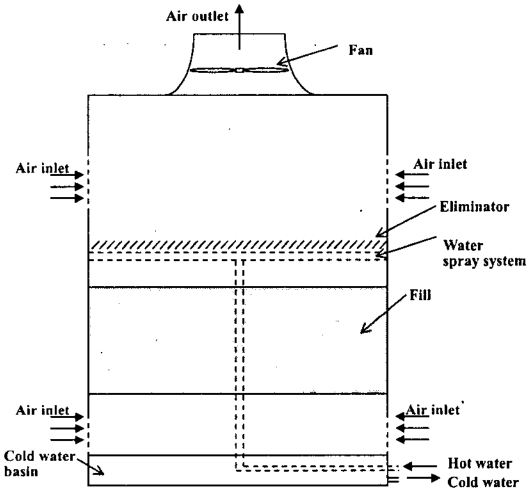

Fig. 3 Forced draft counterflow wet/dry tower.

sure and momentum. Enthalpy and moisture fraction change of air due to evaporation are considered as the source terms in enthalpy equation and moisture fraction equation.

3.2 Boundary conditions

3.2.1 Inlet boundary condition

From the state of ambient air, enthalpy and moisture fraction of inlet air at wet section are determined. The thermodynamic properties of inlet air through the dry section are determined by the effectiveness-NTU method described in section 2.3. Inlet velocities of air at both wet and dry section are assumed as uniform.

3.2.2 Outlet boundary condition

To keep the mass conservation law, a mass flux of air through the tower exit is matched with the inlet mass flux of air added to evaporation mass in the tower due to mass transfer.

3.2.3 Symmetric and wall boundary conditions

At the symmetric plane, a gradient along the normal direction is zero for every dependent property. At the side walls, no-slip condition is applied for velocity condition, and isothermal condition is applied to the other scalar variables.

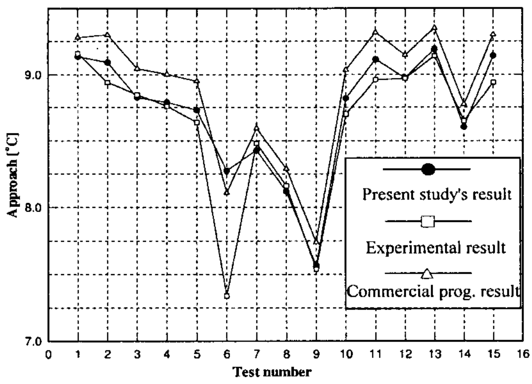

Fig. 4 Comparison of present result with experimental[7] and commercial program results.[8]

4. Results

4.1 Validation of results

To validate the simulation results, they are compared with the experimental[7] and commercial program[8] results. Since there are no useful results with plume abatement tower for comparison, we have obtained the results for a conventional crossflow tower and compared these with above references as shown in Fig. 4. The results showed good agreement with each other and we could develop the result for the plume abatement tower with this validated code.

4.2 Forced draft counterflow cooling tower

Dimensions and operating conditions of counterflow tower are summarized in Table 1 and Table 2, respectively. A fill in wet section has the shape of corrugated asbestos sheets. As a result, a mean temperature of outlet water, cooling range, approach, cooling tower effectiveness and evaporation rate are manifested in Table 3. The cooling range is relatively small, because the operating conditions are suitable

Table 2 Operating conditions for the present study

	Counterflow tower	Crossflow tower
Dry bulb temperature [℃]	18.00	15.00
Wet bulb temperature [℃]	17.50	14.00
Ambient pressure [kPa]	101.325	101.325
Inlet water temperature [℃]	37.00	37.40
Water mass flow rate [kg/s]	20.00	265.00
Air mass flow rate [kg/s]	8.00	375.67
Air mass flow rate with wet section inlet [kg/s]	6.40	252.536

Table 3 Calculation result for the counterflow tower

Average water outlet temperature [℃]	33.03
Range in dry heat exchanger [℃]	0.37
Range in wet section [℃]	3.60
Total range in tower [℃]	3.97
Approach [℃]	15.53
Evaporation rate [%]	0.31
Cooling tower effectiveness	0.20

for light cooling load states. Figure 5 shows the streamline of air. In this figure, horizontal and vertical axis depicts the height and width of tower in meter, respectively. The density of air decreases as the air stream approaches the center region of the tower. A secondary flow is created at the upper region of wet section, and causes many undesired effects, such as wall corrosion, noise, vibration, and specially, decrease of cooling capacity. Also the same effects result from a large secondary flow near the plenum corner. The isothermal contour of the air is shown in Fig. 6. Obviously, the temperature of wet air is rapidly increased when the air passes over the fill. There are apparent two layers at the air exit which are made by unmixed wet and dry air stream. That is, the outlet air stream near the center region is almost saturated except near the wall region which has very low humidity. Figure 7 shows the isothermal contour of water. Sprayed cooling water maintains almost uniform temperature until it meets the fill, and its temperature decreases as it passes over the fill region.

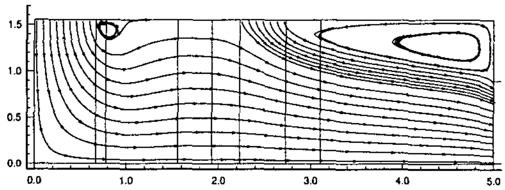

Fig. 5 Streamline of air for counterflow tower.

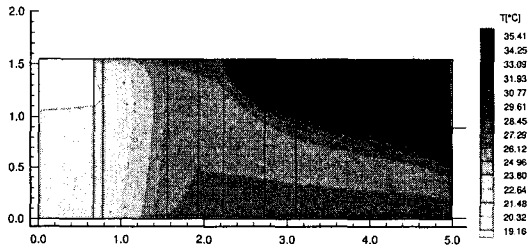

Fig. 6 Isothermal contour of air for counterflow tower.

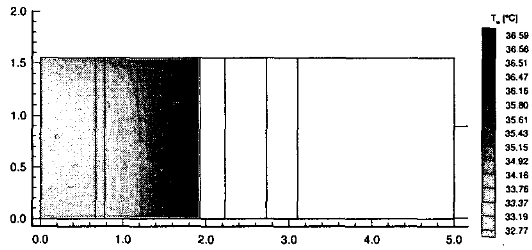

Fig. 7 Isothermal contour of water for counterflow tower.

Figure 8 illustrates the operating line under the above conditions on the psychrometric chart. As you can see in the figure, the mean temperature of the air at the exit shows plume abatement because the operating line does not cross the saturation curve. However, the air at

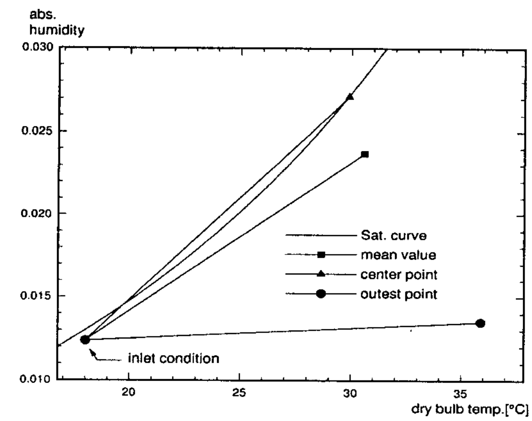

Fig. 8 Operating line on psychrometric chart.

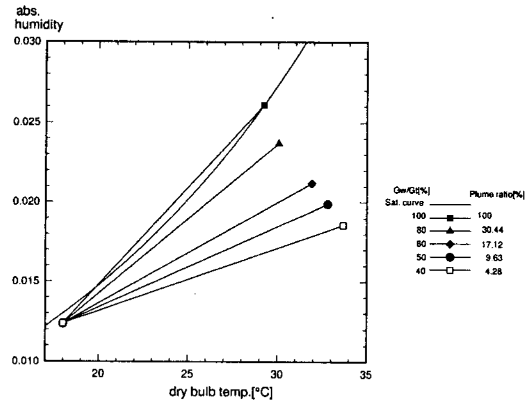

Fig. 9 Operating line with varying wet air flowrate.

the center exit region is nearly saturated in spite of induced dry air into the tower, the air through the outer region of exit has relatively low humidity, and the plume is partially generated. Under the operating conditions, as mentioned in Table 2, a plume area ratio (the ratio of plume area to total exit area) is calculated as 30.44%. The reason why the partial plume is generated is that the unmixed two air streams which have large temperature differences are exhausted through the exit. Since this partial plume causes serious problems[3] under the certain conditions, we should setup some devices in cooling tower which enhance the mixing between two air streams to reduce the partial plume.

One of the convenient method to enhance mixing is adjustment of mass flow rates between wet and dry air. For this purpose, under the same operating conditions, we have estimated the plume area ratio and cooling range with variation of relative mass flow rate of air (ratio of mass flow rates of wet air to total air), G_w/G_t, as 100%, 80%, 60%, 50% and 40 %, respectively. As shown in Fig. 9, the plume is abated for all of these cases except totally wet cases. As the mass flow rate of wet air has increased, a possibility of plume formation has decreased. However, the partial plume has not been eliminated perfectly. For each case, the value of plume area ratio is given in Fig. 9. According to these results, the amount of plume has decreased with increase of dry air mass flow rate, but the range of cooling water is also dropped (Fig. 10). That is, for plume abatement, the cooling efficiency has decreased with the mass flow rate of wet air. As a result, it needs additional costs – a larger or in series towers.

For plume abatement without additional expenses, installation of an air mixer is considered. In this study, two shelf types of air mixer are applied. Figure 11 presents the schematic diagram of these mixers. Type 1, which has wide usages, is attached to the outer wall just below the dry section and directly extended to upper chamber, with slightly inclined shape. To examine the effect on plume abate-

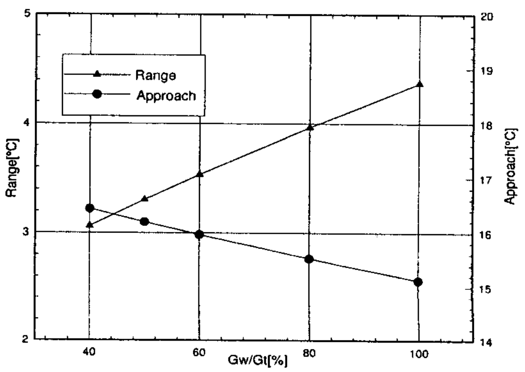

Fig. 10 The effect of wet air flowrate.

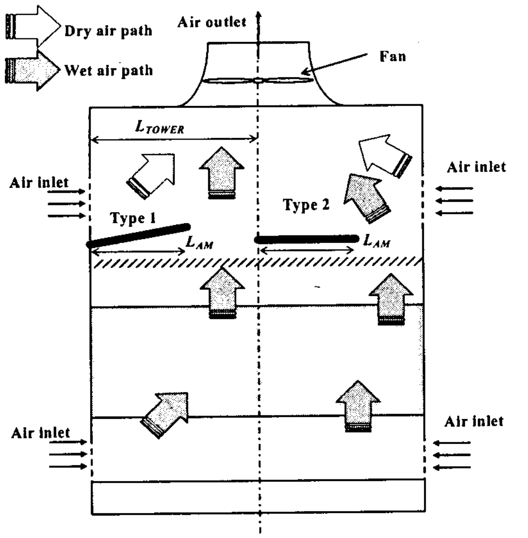

Fig. 11 Schematic diagram of air mixer, type 1 and type 2.

ment with variation of the air mixer length, the ratios of length to a tower half width – 0.2, 0.4, 0.6 and 0.8 are selected. The operating conditions are same as Table 2. The plume abatement ability increases with the ratio of length because the dry air is more induced to the core of the tower. Hence, the plume area ratio decreased to 20.66%, 18.37%, 5.97% when the ratio of length is 0.4, 0.6 and 0.8, respectively. For the case of the ratio 0.8, however the plume area ratio is sufficiently low as 5.97 %, and the partial plume is still formed. Therefore, as illustrated in Fig. 12, the unmixed two air streams which have large temperature difference between them are still exhausted through the exit. Even if the plume area has decreased, the plume has become denser and grown up to higher altitude than the case of without air mixer. There are no differences in the cooling efficiency whether the air mixer is installed. Pressure drop has increased with the ratio of length due to flow stagnation in the fill region, which was driven by installation of air mixer.

A different type of air mixer is proposed – type 2. This mixer is located horizontally in the space of chamber above the cooling water

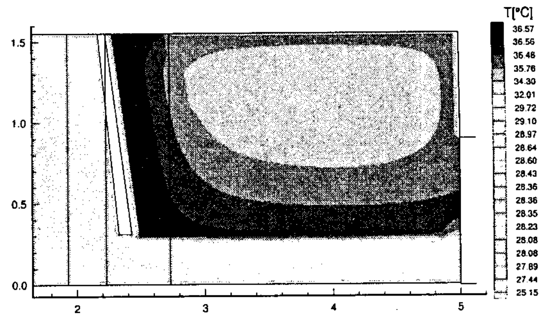

Fig. 12 Isothermal contour of air with type 1 at $L_{AM}/L_{TOWER}=0.8$.

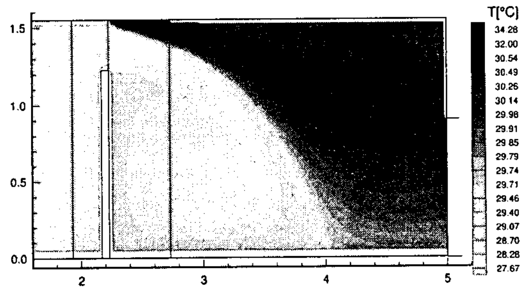

Fig. 13 Isothermal contour of air with type 2 at $L_{AM}/L_{TOWER}=0.8$.

sprayer and has the air passage between mixer and outer wall, as shown in Fig. 11. The operating conditions are also presented in Table 2 and 0.2, 0.4, 0.6 and 0.8 are selected as the ratio of mixer length to half width of tower. The plume area ratio is reduced with increase of the ratio of length – 19.75%, 16.01%, 11.11%, 0%, respectively. Hence plume abatement performance of the type 2 significantly is more improved than the type 1. Unlike the type 1, in which wet air is mixed with a portion of dry air, the type 2, makes wet air stream mix with dry air stream rapidly and effectively. As a result shown in Fig. 13, there are slightly low gradients of temperature and moisture fraction at the exit along the radial direction. Figure 14 presents the operating lines for dry bulb temperatures. In this figure, we confirm that almost uniform, unsaturated flow is exhausted to ambient through the tower exit. The cooling efficiency is independent of installing air mixer,

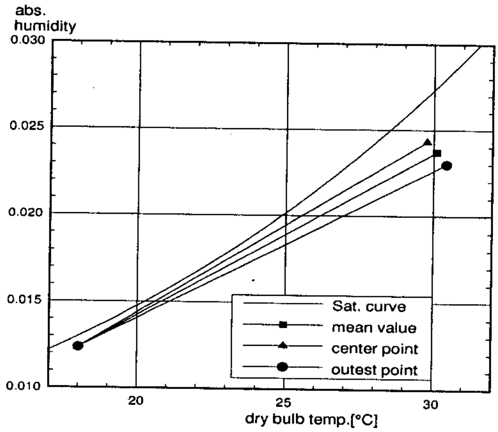

Fig. 14 Operating lines for type 2 at $L_{AM}/L_{TOWER}=0.8$.

same as the type 1. Although both types of air mixers show similar cooling efficiency, the present model gives a less plume formation. However, the type 2 may have some defects, such as installation, cleaning and causes vibration while the tower is working.

Briefly, for the above reasons, controls over relative mass flow rate of wet and dry air have an effect on the cooling efficiency, and appropriate installation of air mixer is directly associated with plume abatement.

4.3 Forced draft crossflow cooling tower

Dimensions and operating conditions of crossflow tower are summarized in Table 1 and Table 2, respectively. A fill in wet section has the shape of wedged plastic airfoil. Like a counterflow tower, mean temperature of outlet water, cooling range, approach, cooling tower effectiveness, and evaporation rate are manifested in Table 4. The cooling range is larger than that of counterflow tower and the operating conditions in the crossflow tower is matched with the heavy cooling load states.

Figures 15 and 16 present the streamline of air and isothermal contour of air and water in the cooling tower, respectively. Air temperature at the center region of tower is higher than

Table 4 Calculation result for the crossflow tower

Average water outlet temperature[℃]	24.44
Range in dry heat exchanger [℃]	1.28
Range in wet section [℃]	11.68
Total range in tower [℃]	12.96
Approach [℃]	10.44
Evaporation rate [%]	1.03
Cooling tower effectiveness	0.56

that at the near wall. Also, air temperature increases as the air approaches to water inlet region with same radial distance. In opposite to 1-dimensional temperature field in the fill region for counterflow tower, 2-dimensional temperature fields are typical features of the crossflow tower. Figure 17 shows the operating line on the psychrometric chart. On the basis of mean properties at the exit, the plume is abated. However, like counterflow tower, partial plume is formed as G_w/G_t increases.

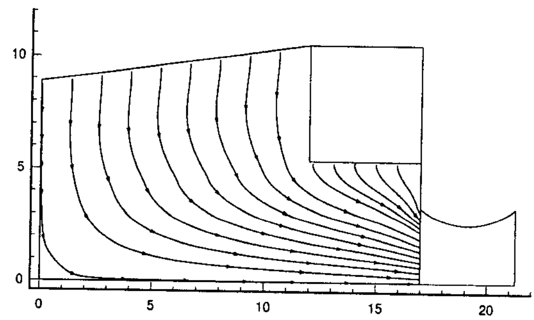

Fig. 15 Streamline of air for crossflow tower.

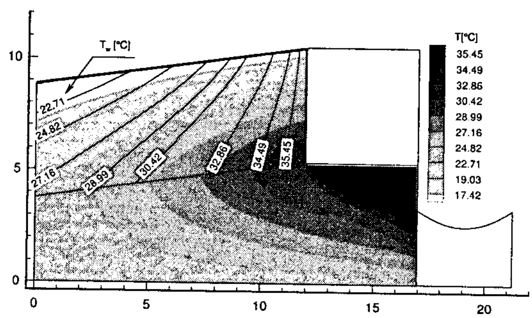

Fig. 16 Isothermal contour of air and water for crossflow tower.

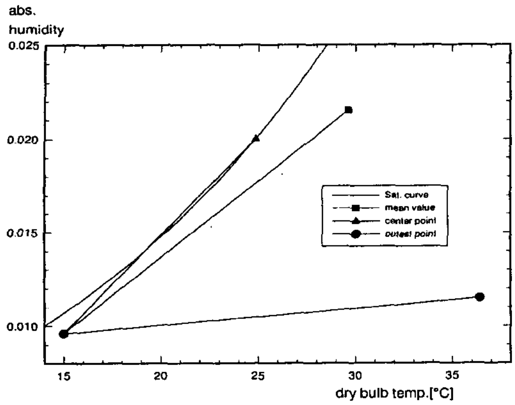

Fig. 17 Operating line on psychrometric chart.

For prevention of plume, relative mass flow rate of air is controlled as 100%, 80%, 60%, 50 % and 40% and for each relative mass flow rate, the plume area ratio is 100%, 19.78%, 5.01 %, 2.79% and 1.22%, relatively. In this results, one can know the partial plume is generated, just same as the counterflow tower. The possibility of plume generation decreases with increase of the mass flow rate of dry air, but the cooling capacity is reduced more rapidly. Hence we should build a tower whose cooling capacity is larger than the design value, and run it as a conventional wet tower except the operating condition has the possibilities of plume formation. For the crossflow tower, installation of air mixer is inadequate since there is not much available space in the tower, so we don't consider an effect on attaching the air mixer.

4. Conclusions

We have developed the software that could analyze the 2-dimensional velocity and temperature fields in cooling tower and predicted plume prevention from the tower with considering two phase turbulent flow by using a $k-\varepsilon$ turbulent model. The plume generation possibilities and cooling efficiencies are estimated with variation of relative mass flowrate of two air streams and with changing length and type of air mixer. As a result, an appropriate control over relative mass flow rate is related to cooling efficiencies, and establishment of the air mixer can reduce the plume formation. For the prevention of plume from the cooling tower, the advanced type of air mixer –type 2 is proposed and it gives an improvement in plume abatement.

Acknowledgement

The authors gratefully acknowledge the financial support from the Micro Thermal System Research Center sponsored by the Korean Science and Engineering Foundation.

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