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A Study on The Counter-Flow Cooling Tower Performance Analysis and Experiments

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Key words : Thermal performance, Counter-flow cooling tower, Filler, Characteristic curve, Performance curve

Abstract

The thermal performance of cooling towers is affected by the temperature of inlet water, wet bulb temperature of entering air and water-air flow rate. In this study, the effects of these variables are simulated using NTU-method and experimentally investigated for the counter-flow cooling towers. The simulation program to evaluate these variables which affect the performance of cooling tower was developed. The maximum errors between the results of simulations and experiments were 3.8% under the standard design conditions and 5.4% under the other conditions. The performance was increased up to 46 ~ 50% as the water loading was increased from 6.8 $m^3/hr \cdot m^2$ to 15.9 $m^3/hr \cdot m^2$. The range was reduced up to 56~42% when the wet bulb temperature of the entering air was increased from 22℃ to 29℃.

Nomenclature

A : Heat transfer area [m^2]
a : Mass transfer area per unit volume [m^2/m^3]
C_L : Specific heat [kJ/kg · K]
G : Mass flow rate of air [kg/hr]
h : Enthalpy [kJ/kg]
K : Mass transfer coefficient [kg/hr · m^2]
L : Mass flow rate of water [kg/hr]
NTU : Number of transfer units
R : Cooling range [K]
S : Contact area of water and air [m^2]
T : Temperature [K]
V : Cooling tower filler volume [m^3]

Subscripts

1 : Inlet condition
2 : Oulet condition
a : Air

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s	: Saturated state
w	: Water
wb	: Dry bulb
db	: Wet bulb

1. Introduction

A cooling tower is a highly economical system to remove the waste heat by consuming minimum electric energy and it is one of excellent environmental facilities preventing water contamination and air pollution together. In brief, comparing with other refrigeration or air-conditioning systems, a cooling tower exchanges much more heat with small power consumption and it does not cause any environmental problems.

A cooling tower, a system supplying cooled water due to its sensible and latent evaporating heat by the direct heat exchange with ambient air, has been used in extracting the waste heat arisen from industrial processing. In recent, the optimally designed technology is crucially needed in a cooling tower to prevent over-designed systems.

Due to its complexities in the internal heat exchanging process of a cooling tower, analytical and computational methods are almost impossible to predict its performance correctly. Therefore, characteristic values and design parameters have been obtained from experiments.

Principles of the heat exchange between the hot inlet water and the suction air can be explained as two mechanisms; one is the sensible cooling of water caused by the temperature difference between air and water and the other is the latent cooling by the evaporating heat of water. Therefore, main factors presenting the performance of a cooling tower are the cooling range and approach of wet bulb temperature.(1,2)

Other main input parameters, which affect to the thermal performances, are temperature, humidity of air and water flow rate.

To enhance a cooling effect, in general, filler is installed inside the cooling tower. The heat-exchanging performance of filler has a great effect on the cooling tower's total performance. To predict heat-exchanging performance of filler, we need all the data of the mass transfer coefficient, contacting time, area between air and water, and pressure distribution of the filler.

However, the analytical process is too complicated to get all the local data. Therefore, most researches are relied on the experiments to get the performance of a cooling tower.

In this research, based on the experimental performance data computer simulation program was proposed by NTU method considering various operating parameters. In other word, this research was aimed to furnish the computer simulation program, which can be used in the database for optimum design.

2. Basic Theory

A simplified heat and mass transfer mechanism which is called potential driving force concept was assumed with following assumptions:(3)

(1) Neglect the water loss by evaporation.
(2) Thermal diffusion coefficient is equal to mass diffusion coefficient.
(3) Air at a contact area is a saturated state.
(4) Latent heat of vaporization is constant.

For the above assumption (2), Lewis number is defined to link the thermal diffusion and mass diffusion. When the temperature is 0 ℃

and the pressure is 760 mmHg, Lewis number is 0.866. At this situation, the Lewis value is close to 1 and the partial pressure of vapor is relatively small. Therefore, the assumption (2) is appropriate.

The total heat transfer rate is a function of the enthalpy difference, i.e., the difference between the enthalpy of saturated vapor at water temperature and the enthalpy of contact air as follows.

$$Q = KS(h_w - h_a) \tag{1}$$

In a counter-flow cooling tower, an energy balance equation can be expressed as

$$dq = K(h_w - h_a)dS = Gdh_a = LC_L dT_w \tag{2}$$

By integrating equation (2), we can obtain

$$\frac{KaV}{L} = \frac{G}{L}\int_{h_{a1}}^{h_{a2}} \frac{dh_a}{h_w - h_a} = C_L \int_{T_1}^{T_2} \frac{dT_w}{h_w - h_a} \tag{3}$$

In the above equation the term, KaV/L, is not the function of filler but the function of temperature and flow rate. This term is called NTU (Number of Transfer Unit) in other words.

Committee of Performance and Technology under CTI investigated many integration methods based on the Merkel's equation to find more precise calculation of equation (3). They concluded that the 4-point Tchebycheff integration method is the most accurate.

In 1967, CTI proposed various NTU values for 40 different kinds of wet bulb temperature, 21 different ranges and 35 different cases of contact temperature by the Tchebycheff integration method as follows.

$$NTU = \frac{KaV}{L} = \frac{G}{L}\int_{ha1}^{ha2} \frac{dh_a}{h_w - h_a} \tag{4}$$

Heat transfer characteristics of a cooling tower can be represented by a characteristic curve. A characteristic curve shows filler characteristic value, KaV/L, on changing the ratio of water-air flow rate at a logarithmic coordinate. At given filler, the mass flow rate of water, L, is a constant. Then, if the mass flow rate of air, G, decreases, the heat transfer capacity of filler decreases. By the increase of water-air flow rate, the characteristic value of filler tends to decrease. When a characteristic curve is developed, a water-air flow rate changes at the cooling tower being installed filler. The inlet/outlet enthalpy of the air around the water film and ambient air is measured with changing water-air flow rate of a cooling tower installed the filler. Then, using equation (4), the characteristic value can be obtained. Based on the experiment, the characteristic value of filler can be written by the following relation.

$$\frac{KaV}{L} = C_1\left(\frac{L}{G}\right)^{-m} \tag{5}$$

Here C_1 and m are empirical constants.

Since the characteristic value of a cooling tower is valuable information for manufactures, the characteristic value on the equation cannot be used in general. Therefore, a characteristic curve is mainly obtained by the experiment.

3. Performance analysis of a cooling tower

3.1 Algorithm of the capacity calculation

The purpose of performance prediction at a cooling tower is to calculate the cooling tower capacity on cooling water flow rate, inlet temperature, and air wet bulb temperature. Here, the algorithm is based on characteristic flow rate, inlet temperature of water, and air wet bulb temperature. The relation of dependent and independent variables can be written by

$$T_{w2} = f_1[\, T_{wb},\, T_{w1},\, G,\, L] \tag{6}$$

From equation (6), the dependent variable can be calculated by determining the independent variables. Then, the results can be utilized for calculating the capacity.

Performance calculation of a counter flow cooling tower can be obtain that a design point is determined at a point of intersection between performance curve of equation (4) and characteristic curve of equation (5).

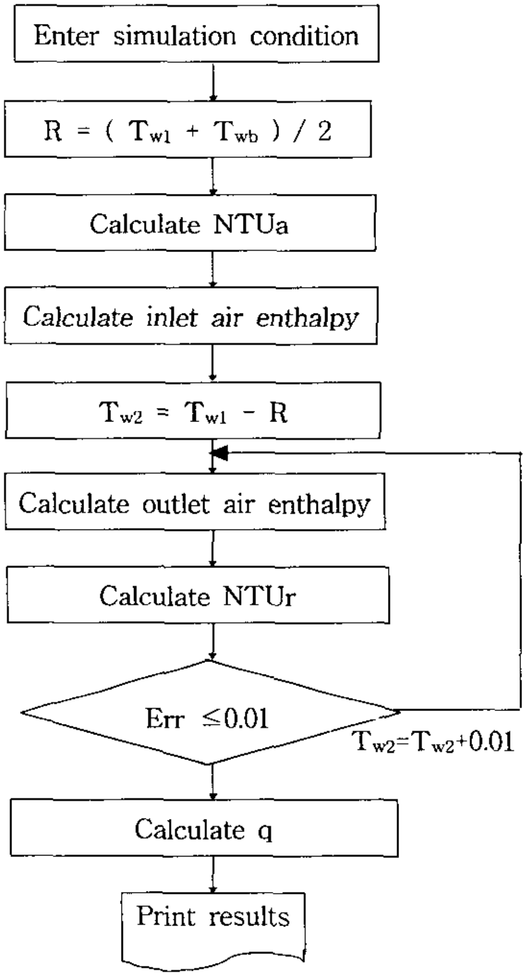

Fig. 1 Simulation program flow chart.

First, it can be assumed NTUr as a performance curve, and NTUa as a characteristic curve. Then, on the assumption of cooling range $R=(T_{w1}+T_{wb})/2$ the difference between NTUa and NTUr can be checked by using a bisection method. Finally, if equation (7) is satisfied, the program converses.

$$Err = \frac{NTUa - NTUr}{NTUa} \leq 0.01 \tag{7}$$

3.2 Analysis results

The standard design condition of a cooling tower is that the inlet and outlet temperature of cooling water is 37 ℃, respectively, and air wet bulb temperature is 27℃.

The results of the performance analysis are in the following. Fig. 2 shows the change of the range on the water loading varying the air velocity at the filler. As the air velocity increase, the range increases. This is reason for the increase of the heat transfer coefficient in the system. Also, as the water loading in-

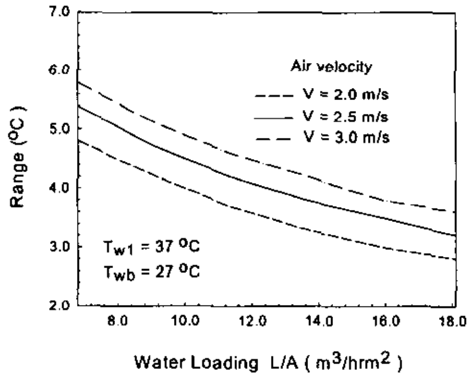

Fig. 2 Effect of air velocity and water loading on range.

crease from 6.8~18.1 $m^3/hr \cdot m^2$, the range decreases to 2.0~2.2 ℃. This is because as the water flow rate increase the liquid film on flowing along the surface of filler thickens. Consequently, the total heat transfer area does-not increase proportionally with the increase of film thickness. Fig. 3 shows the range of the change on the wet bulb temperature with varying the air velocities when the water loading is 9.1 $m^3/hr \cdot m^2$, the cooling water inlet temperature is 37℃, and the dry temperature of inlet air is 32℃. As the wet bulb temperature increase from 21~29℃, the range depends on the air velocity, and gradually decreases to 2.5~3.1 m/s. Fig. 4 shows the range on the change of the air velocity and the cooling inlet temperature when the air velocity is 2.5 m/s, the cooling water inlet temperature 37 ℃, and the dry bulb temperature of the inlet air 32 ℃. When the water loading increases the cooling range also decreases. Fig. 5 shows the results of the change of the range on the air velocity and the cooling inlet temperature when the water loading is 9.1 $m^3/hr \cdot m^2$, the wet bulb temperature of the inlet air is 27 ℃, the dry bulb temperature of the inlet air is 32 ℃.

4. Experimental apparatus and method

4.1 Experimental apparatus

The experimental apparatus is designed to analyze the change of a cooling performance considered important parameters on the thermal performance of a cooling tower. The experimental apparatus is manufactured by an acrylic and is installed the filler of 230×23×600 (Munter 19060 filler).

To distribute water equally on filler, the nozzle is installed. Air quantity flowing to the cooling tower is controlled air by using a turbo-fan and an inverter. Using the damper of the high temperature of a cooling tower exit, the outside air wet bulb temperature is properly controlled. To control dry bulb temperature, an electric heater of 2 kW capacities is installed, and constant temperature is kept by a feed back control system.

To control the temperature as a constant state, water is sprayed to the filler. To satisfy the experimental condition, an electric heater and a feed back control system is installed.

The water flow rate is measured by a rotermeter having a 2% error, and a hot wire

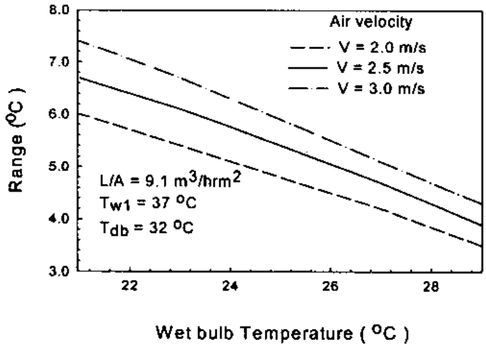

Fig. 3 Effect of air velocity and wet bulb temperature on range.

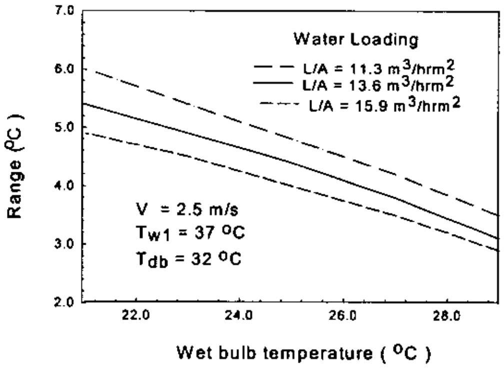

Fig. 4 Effect of water loading and wet bulb temperature on range.

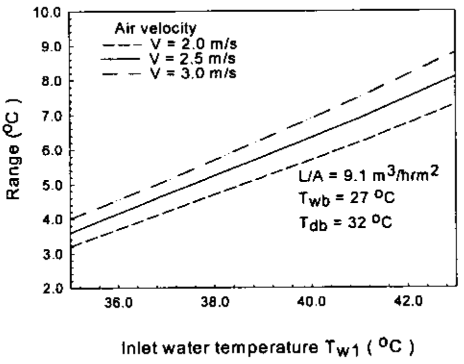

Fig. 5 Effect of inlet water temperature and air velocity on range.

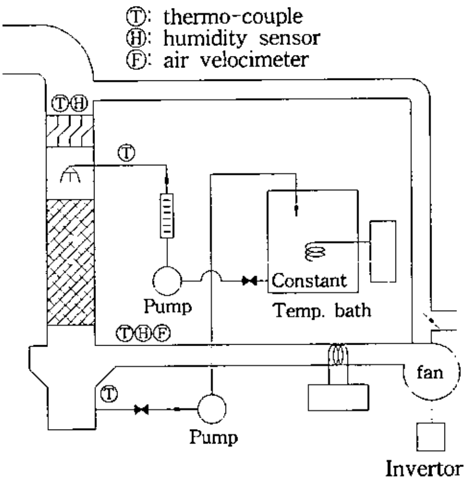

Fig. 6 Schematic of experimental apparatus.

current meter measures the air velocity. The temperature of water and dry bulb temperature of air are measured by T-type thermo-couple of a diameter 0.2 mm, and the test section is calibrated with an error of 0.1 ℃. Before the experiment, wet bulb temperature is measured by a wet sense with 2% error.

The outputs of each sense is recorded by NetDAQ and FLUKE and stored by a personal computer. Fig. 6 shows a schematic diagram of the experimental apparatus.

4.2 Experimental method

In this experiment, the inlet temperature of cooling water is 30~40 ℃, the wet bulb temperature of inlet air is 22~29 ℃, the dry bulb temperature of inlet air is 32 ℃, the air velocity is 2.0-3.0 m/s, and the flow rate of cooling water is 6-14 lpm.

Main procedure of the experiment is the control of a flow rate of suction air, and the flow rate and temperature of supplied cooling water. Then, after the experiment reaches to a steady state the data is measured.

A blower and inverter having maximum air flow rate of 17 m³/min keep a constant of the setting air flow rate, and exhausted air of high humidity though a return duct is supplied at the wet bulb temperature control part of the inlet air. Then, this air mixed with ambient air is controlled dry bulb temperature and wet bulb temperature to satisfy the experimental conditions. Extra exhausted air goes to an ambient, and this minimizes the change of inside environment.

Water circulated by a circulation pump is sent to a constant temperature bath. Then, at a constnat temperature bath water has a constant temperature, and then sprayed at a nozzle of the test section. The nozzle used at the experiment has a diameter of 2 mm, and it is positioned by a 3 rows. Each row has 10 nozzles. Totally, 30 nozzles are installed. This sprays water equally to filler.

4.3 Experimental results and discussions

Fig. 7 shows the change of the range on the air velocity and the water loading. By comparison two figures, Fig. 2, which is the results of the analysis program, and Fig. 7,

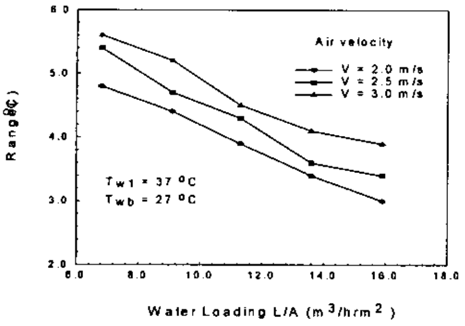

Fig. 7 Effects of air velocity and water loading.

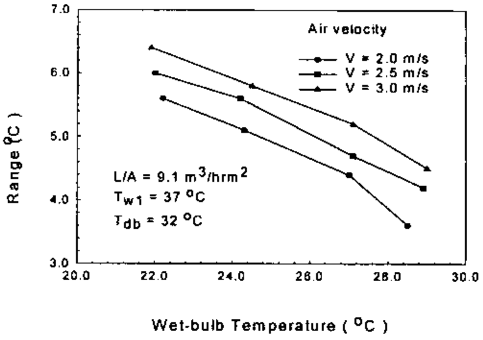

Fig. 8 Effects of air velocity and wet bulb temperature.

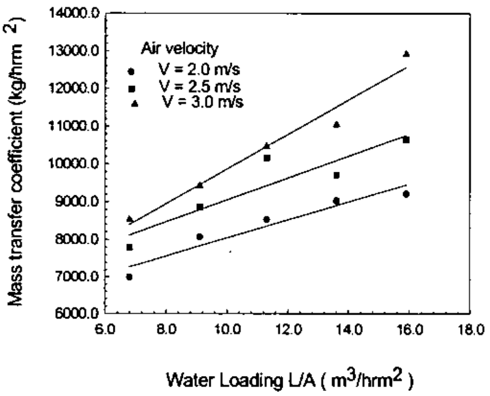

Fig. 9 Effects of water loading on mass transfer coefficient.

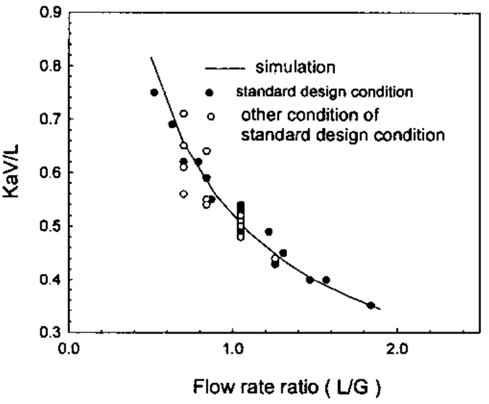

Fig. 10 Comparision chart between experimental and simulation data.

which is the results of the experiment, the maximum error is approximately 3.8%.

Fig. 8 shows the variation of the range on the change of the air velocity and the wet bulb temperature of the inlet air. As the wet bulb temperature increase, the range is decreasing on the evaporating quantity of water because of decreasing the ability of containing water. And, this figure shows a maximum error of 5.5% compared to the results of the analytical analysis.

The thermal performance of a cooling tower obtained from the experiment is used to analyze the characteristics of the water loading and the air velocity. Based on the experimental results, filler property, KaV/L, by integrating using the Tchebycheff method is can be given the overall mass transfer coefficient.

4.4 Error analysis

Fig. 10 presents the comparison between the characteristic of a cooling tower by the analysis program and experimental data. Two results are agrees well, and the reason of the discrepancy between two data is in the following.

(1) It is used the approximated equation on the thermal and mass transfer.

(2) The enthalpy of air is calculated by the relation of wet bulb temperature only.

(3) Evaporation is neglected.

(4) To determine NTU, a simple curve is used.

5 Conclusions

To analyze the characteristic performance of a cooling tower, the experiment and simulation is studied. This research has led to the following conclusions.

(1) It is developed the program, which can evaluate the performance on the parameters affecting the cooling tower performance. The error of the standard design condition is a maximum value of 3.8%. The error of other conditions is 5.4%.

(2) When the water loading increases from 6.8 $m^3/hr \cdot m^2$ to 15.9 $m^3/hr \cdot m^2$, the range is decreased to 30%~38%.

(3) As the wet bulb temperature of the inlet air increase to 22 ℃~29 ℃, the range decreases to 56%~42%. When the inlet temperature is 35 ℃~40 ℃, the range change 69%~46%.

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