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Experimental and theoretical analysis of mechanical draft counterflow wet cooling towers

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Wet cooling towers have high thermal efficiency and are cost-effective industrial cooling units, used in factories, in energy plants, and in the HVAC market. The working principle of the cooling towers is based on cooling of the hot process water with cold atmospheric air via cooling fills and water distribution systems. This study investigates the pressure losses and thermal performance of mechanical draft counterflow water cooling towers having different cooling fill heights. During the thermal performance tests, the pressure loss through cooling fill was separately measured for each condition. The variation of pressure losses with air velocity was plotted through cooling fills at different loading heights. Performance curves were obtained and compared as graphs relating the liquid–gas ratio (*L/G*) to the Merkel number for cooling fills at different heights between 1500 mm and 2400 mm. Although the lowest and the highest flow resistances were measured at fill heights of 1500 mm and 2400 mm, the lowest and the highest performances for the cooling tower were calculated at 1500 mm and 2400 mm, respectively.

Introduction

Cooling towers are industrial units that cool water through heat and mass transfer. They are generally used in heavy industry power plants, chemical plants, and iron and steel factories, as well as in the HVAC market. In power plants, cooling towers generally work with turbine and electric generator systems, while in the HVAC market, they work with water-cooled chillers. They can also be used wherever there is a need for cooling water. Their simple working principle is: The hot water is sprayed onto the fill zone with a distribution system, and cools there with an extended surface area and atmospheric air. The atmospheric air used is provided by mechanical or natural draft systems. Mechanical draft systems have fans, motors, and other drive equipment and they use electrical energy, but natural draft systems only have a shell to create airflow. In both systems, a small portion of the process flow is evaporated because of the inlet air's relative humidity and direct evaporative cooling principle. Air evaporates water until the saturation moisture limit at the beginning of heat transfer. Afterward, water consumption continues according to direct evaporative cooling principles until the process ends (Kröger 2004). The working principle for the mechanical draft counterflow open circuit cooling tower is shown schematically in Figure 1.

In the literature, there are many methods for calculating the open-circuit wet cooling towers' cooling capacity. The most well-known of these are the Merkel, Poppe, and effectiveness (ε)–NTU methods in the heat transfer algorithm. The first basic theory was proposed by Walker et al. in 1923 (Walker, Lewis, and McAdams 1923). Merkel improved the first practical approach in 1925. Merkel's performance calculation method provides the most widely accepted theory. In this theory, heat and mass transfer equations for water vapor are combined. The enthalpy differences for humid air between air temperature and the local water bulk temperature are used in the analysis. Two main assumptions are made in this theory: Water loss by evaporation is ignored, and the Lewis number is considered to be 1 for the whole system. In this study, the Merkel method is used for numerical analysis (Merkel 1925). The ε-NTU method for cooling tower calculation was developed by Jaber et al. The main difference from Merkel's theory is a different integration procedure between the air and bulk water (Jaber and Webb 1989). A third model, the Poppe model, was developed by Poppe et al. in 1991. It is not a simplifying method like Merkel's theory. The Lewis factor is not assumed to be unity but is calculated in each case. The Poppe model contains different equations for unsaturated and supersaturated outlet moist air (Poppe and Rögener 1991).

Theoretical and experimental studies have been analyzed for collecting thermal performance and air-side pressure drop values.

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Theoretical studies were used for creating a thermal calculation algorithm, and experimental studies considered installation of a cooling tower test unit. The Merkel, Poppe, and ε-NTU methods were examined in detail in the PhD thesis of Johannes Christian Kloppers, published in 2003 (Kloppers 2003). Experimental parameters were collected from a counterflow natural draft cooling tower. A software package that can perform calculations with respect to all three of these methods was developed at the end of this study. The calculation algorithms that were used for analyzing thermal performance of a cooling tower are from the MSc thesis of Andre Alexis Dreyer, published in 1988 (Dreyer 1988). Similar to Kloppers's study, the thesis included these three types of analysis methods. Also, evaporative-type condensers were examined in this study.

Many studies have examined setting up a test cooling tower in a facility. The study by Stefanovic et al. examined specifying the locations of temperature transmitters on the test cooling tower in order to obtain the same input and output values in experimental tests (Stefanović et al. 2000). A thermal modeling graphical analysis made with 26 temperature transmitter measurements was performed by Stefanovic et al. in 2000. Measurements were recorded at different air and water flows. Heat and mass transfer were modeled in this study. Performance curves for different cooling fills were obtained, and it was found that heat and mass transfer coefficients changed according to design conditions. A performance test unit was designed and manufactured by Dvořák et al. in 2006 (Vitkovič et al. 2006). The air-side pressure drop value was measured by four pressure transmitters on this test unit. Air-side pressure drop measurement factors were examined. Air-side pressure drop equations were developed by Vitkovič et al. in 2006.

In the literature, most of the studies described mainly calculation methods and effects on performance from different water loading, air velocity, or type of fill. In this study, the cooling performance of a mechanical draft wet cooling tower is analyzed experimentally based on a theoretical approach. The fill height varies between 1500 mm and 2400 mm, and each fill height option has its own test results in terms of performance and air pressure drop. In this study, all of the calculation methods were analyzed and the Merkel method was selected as perferable, because it is more basic than the others and more suitable for use in a software package for calculating the performance of thousands of different products with one command. It is used to evaluate numerical results, and the Chebyshev theorem is used to solve equations. A similar pressure loss measuring system was used in the present study, but only the pressure loss related to cooling fill was measured.

Experimental setup and method

Experimental tests were used to evaluate the performance and construct the pressure-drop curves for cooling fill. A performance test station in İstanbul was used to record data for the performance and pressure loss curves of the cooling tower. The test facility has a boiler system that can generate up to 500 kW heat load for tests, with two burners to provide a flexible heat load. One of the boilers had two stages and the other one had a single stage. The hot process water circulates between the boiler and a plate-type heat exchanger through a hot water pump. The plate-type heat exchanger within the test facility is divided into a boiler line and a cooling water line, because these have different water quality requirements. The test cooling tower, cold water pump, flow meter, and measurement devices are placed along the cold water line on the circuit. The test facility pipeline and other equipment are shown schematically in Figure 2.

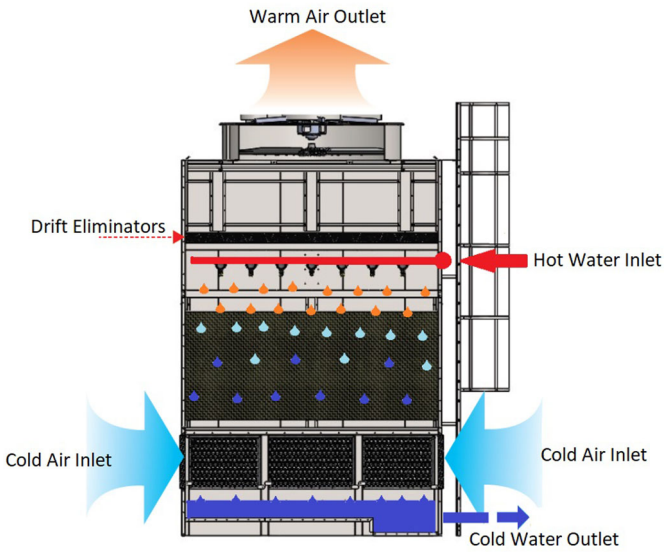

Fig. 1. Working principle of mechanical draft counterflow open-circuit cooling tower.

Two PT100 temperature transmitters at the inlet side and two of them at the outlet side were evaluated. The measurement deviation values of these transmitters had a maximum of ±0.5 °C. An electromagnetic flowmeter was used for measuring the water flow rate, and its maximum error limit was 0.2%. The pressure loss across the cooling fill was measured by a differential pressure transmitter. Its measurement range was from 0 to 1000 Pa and its accuracy was ±0.05%. The transmitter was used with a Pitot tube, and each different cooling fill height was measured at its own measurement location. In the cold water line, the water flow can be adjusted between 0 and 100 m^3/h through the electrical frequency drive system. Different water flows result in different pressure values. A pressure/flow diagram of the spray nozzle was prepared with a pressure indicator.

The cooling tower used in the performance tests was designed for special tests and can allow testing of different types of cooling fill usage. Specifications of the test cooling tower are shown in Table 1.

The test cooling tower is a counterflow type. Therefore, the hot water inlet is at the up side and the cold air inlet is at the down side of the cooling tower. A photographic view of the test cooling tower is given in Figure 3.

Analytical methods

There are two main kinds of transfer equations in cooling tower theoretical analysis: mass transfer and heat transfer.

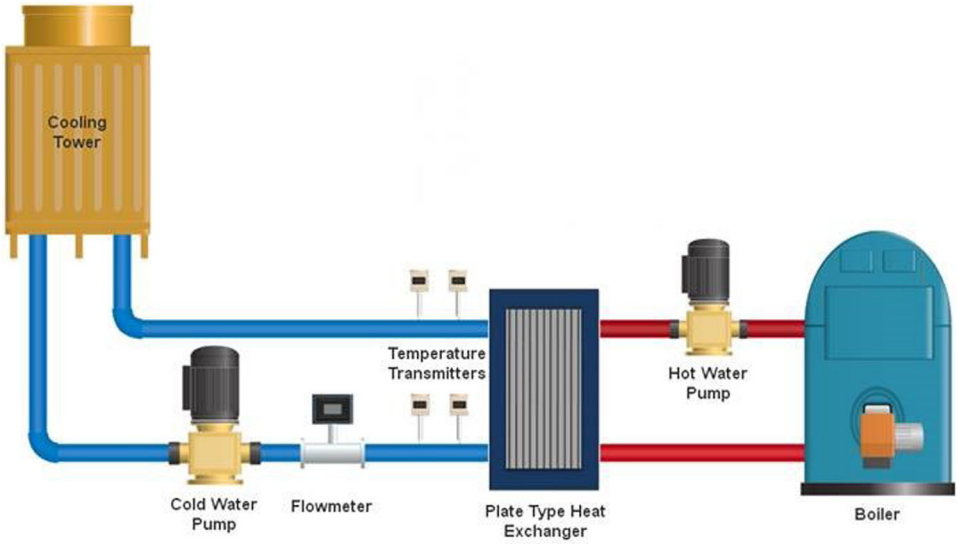

Fig. 2. Schematic view of the test facility pipeline and equipment.

Table 1. Specifications of the test cooling tower.

Equipment name	Information	Unit
Casing material	FRP	—
Fan type	Axial type	—
Draft type	Induced	—
Drive type	Direct driven	—
Fan blade material	Aluminum	—
Number of fan blades	4	—
Fan diameter	1200	mm
Fan blade profile	Air foil	—
Motor nameplate power	5.5	kW
Motor speed	960	RPM
Cooling fill type	PVC film type	—
Cooling fill wave height	19	mm
Orientation	60°, cross fluted	—
Cooling fill height	1500, 1800, 2100, 2400	mm
Water distribution system	1 main distribution line (DN100) 6 side distribution lines (DN50) 12 low-pressure spray nozzles	—
Cooling tower internal length	1850	mm
Cooling tower internal width	1850	mm
Cooling tower total height	7500	mm

Mass transfer

Mass transfer occurs between atmospheric air and hot process water. In fact, this phenomenon is purely psychrometric. This process is expressed in Equations 1 and 2:

$$G(1+w)+\left(L+\frac{dL}{dz}dz\right)=G\left[1+\left(w+\frac{dw}{dz}dz\right)\right]+L \quad (1)$$

$$\frac{dL}{dz}=G\frac{dw}{dz} \quad (2)$$

A typical psychrometric chart of the cooling tower process is provided in Figure 4. As can be seen from Figure 4, all psychrometric values of air increase. However, these psychrometric values can be changed by environmental effects.

When the relative humidity of the atmospheric inlet air is too low, in some cases the outlet dry bulb temperature may be lower than the inlet dry bulb temperature. However, the wet bulb temperature always increases in this cooling process, because the enthalpy of air depends on the wet bulb temperature. Psychrometric values of the air are given in Table 2.

Heat transfer

There are two types of fluid in wet cooling towers: atmospheric air and process water. The process water thermophysical properties do not change by more than 1% throughout the cycle because temperature differences are under 20 K in

general design cases. The total cooling capacity can be calculated using temperature differences between the inlet and outlet and the specific heat value at the mean temperature of the water. On the air side, enthalpy difference is generally used, because there are too many parameters changing the thermophysical properties, such as dry bulb, wet bulb, humidity, and others. The main energy equation of the cooling tower process is Equation 3:

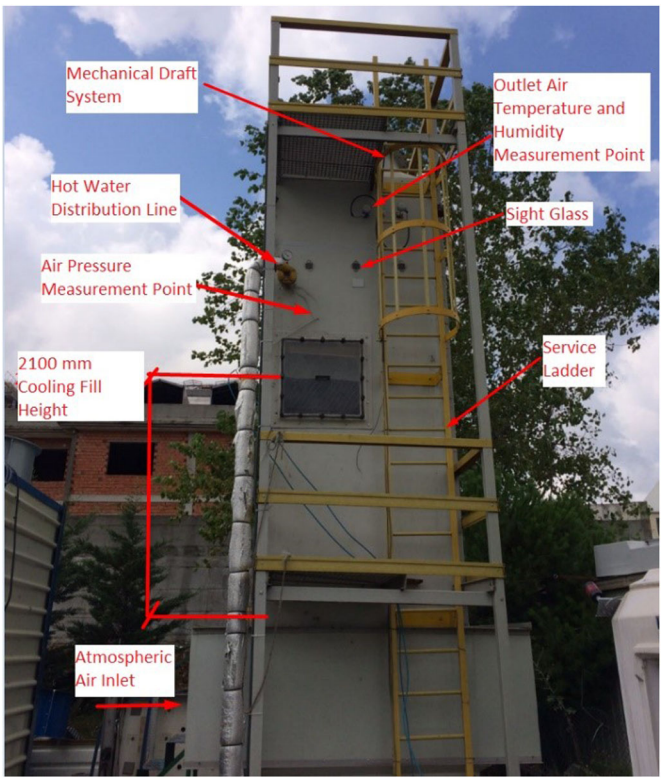

Fig. 3. A photographic view of the test cooling tower.

$$Lc_{p_w}\frac{dT_w}{dz} + c_{p_w}T_w\frac{dL}{dz} = G\frac{di_{ma}}{dz} \tag{3}$$

Sensible and latent heat transfers in cooling towers vary due to the design conditions and thermophysical properties of atmospheric air. The total heat transfer equations is Equation 4:

$$dQ = dQ_m + dQ_c \tag{4}$$

Heat transfer based on evaporating water using mass transfer is expressed in Equation 5:

$$\frac{dL}{dz}d_z = K(w_{sw} - w)_a \; dA \tag{5}$$

Heat transfer based on evaporation of water molecules is expressed in Equation 6:

$$dQ_m = i_v\frac{dL}{dz} = i_vK(w_{sw} - w)dA \tag{6}$$

The sensible heat transfer is described in Equation 7:

$$dQ_c = h(T_w - T_a)dA \tag{7}$$

The Lewis factor is the ratio of mass and heat transfer for the whole process. Bosnjakovic explained the Lewis factor with an expression given here as Equation 8 (Bosnjakovic 1965):

Table 2. Psychrometric values of the air inside cooling tower.

Values	Unit	P_1 point (air inlet)	P_2 point (air outlet)
Dry bulb temperature	°C	28.0	35.0
Wet bulb temperature	°C	20.0	35.0
Dew point	°C	16.1	35.0
Relative humidity	%	48.5	100.0
Humidity ratio	kg/kg	0.0115	0.036
Enthalpy	kJ/kg	57.4	129.0
Density	kg/m^3	1.164	1.121

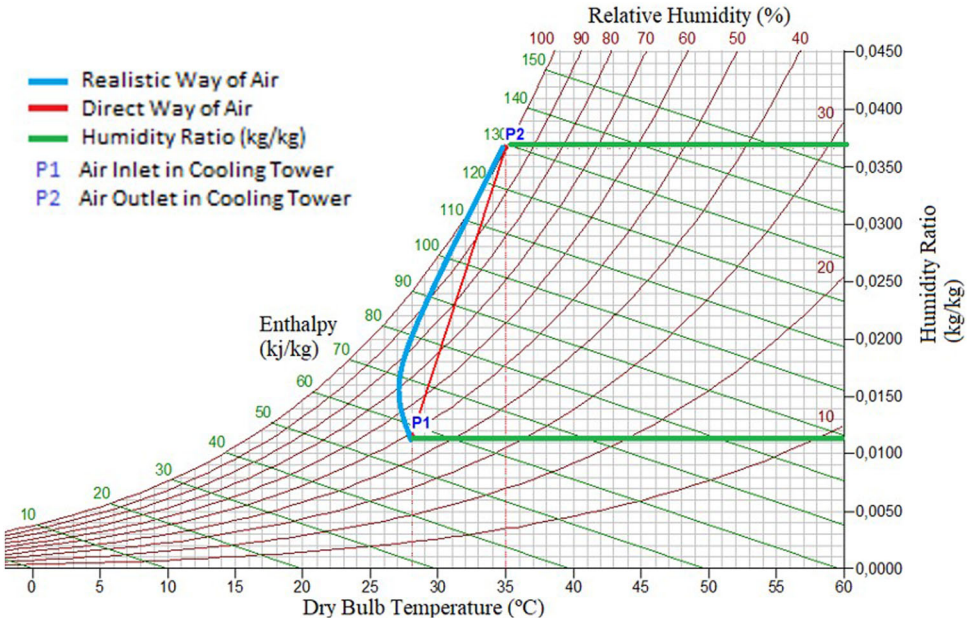

Fig. 4. A typical psychrometric chart of cooling tower process.

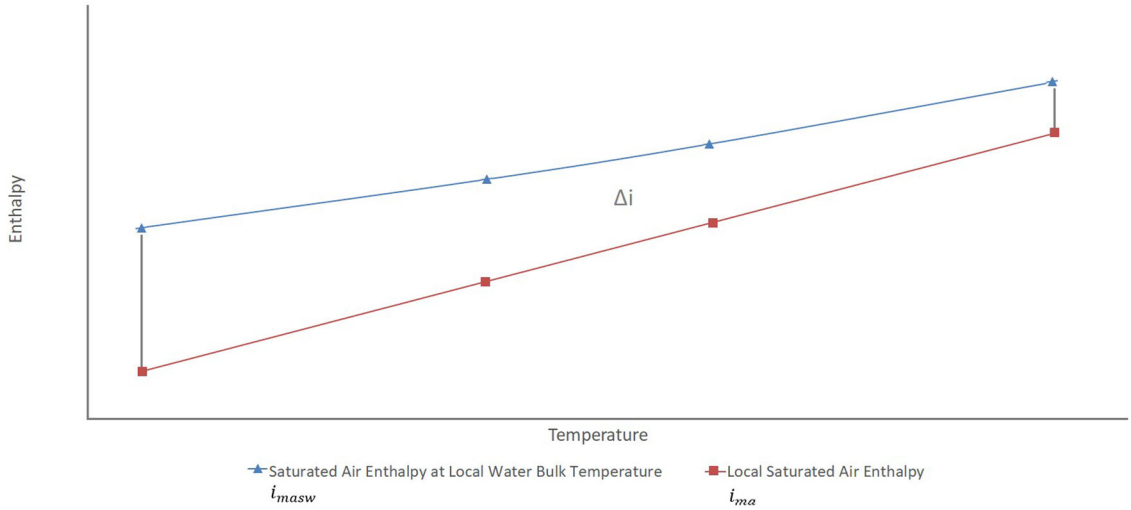

Fig. 5. Enthalpy–temperature diagram for Merkel method.

Table 3. Description of calculation steps.

Water side		Air side		
Water temperatures (°C)	Enthalpy of the saturated air at local water temperature (kJ/kg)	Enthalpy of moist air (kJ/kg)	Enthalpy difference	
T_w	i_{masw}	i_{ma}	i_{masw}-i_{ma}	$1/ (i_{masw}$-$i_{ma})$
$T_{wo} + 0.1(T_{wi} - T_{wo})$	$i_{masw,1.}$	$i_{ma,inlet} + \frac{LT_{wo}+0,1(T_{wi}-T_{wo})c_{p_{w,mean}}}{G}$	$i_{masw,1.}$- $i_{ma.1.}$	$1/(i_{masw,1.}$- $i_{ma.1.})$
$T_{wo} + 0.4(T_{wi} - T_{wo})$	$i_{masw,2.}$	$i_{ma,inlet} + \frac{LT_{wo}+0,4(T_{wi}-T_{wo})c_{p_{w,mean}}}{G}$	$i_{masw,2.}$- $i_{ma.2.}$	$1/(i_{masw,2.}$- $i_{ma.2.})$
$T_{wo} + 0.6(T_{wi} - T_{wo})$	$i_{masw,3.}$	$i_{ma,inlet} + \frac{LT_{wo}+0,6(T_{wi}-T_{wo})c_{p_{w,mean}}}{G}$	$i_{masw,3.}$- $i_{ma.3.}$	$1/(i_{masw,3.}$- $i_{ma.3.})$
$T_{wo} + 0.9(T_{wi} - T_{wo})$	$i_{masw,4.}$	$i_{ma,inlet} + \frac{LT_{wo}+0,9(T_{wi}-T_{wo})c_{p_{w,mean}}}{G}$	$i_{masw,4.}$- $i_{ma.4.}$	$1/(i_{masw,4.}$- $i_{ma.4.})$
Merkel number				$\frac{c_{pwm}(T_{wi}-T_{wo})}{4}\sum \Delta i$

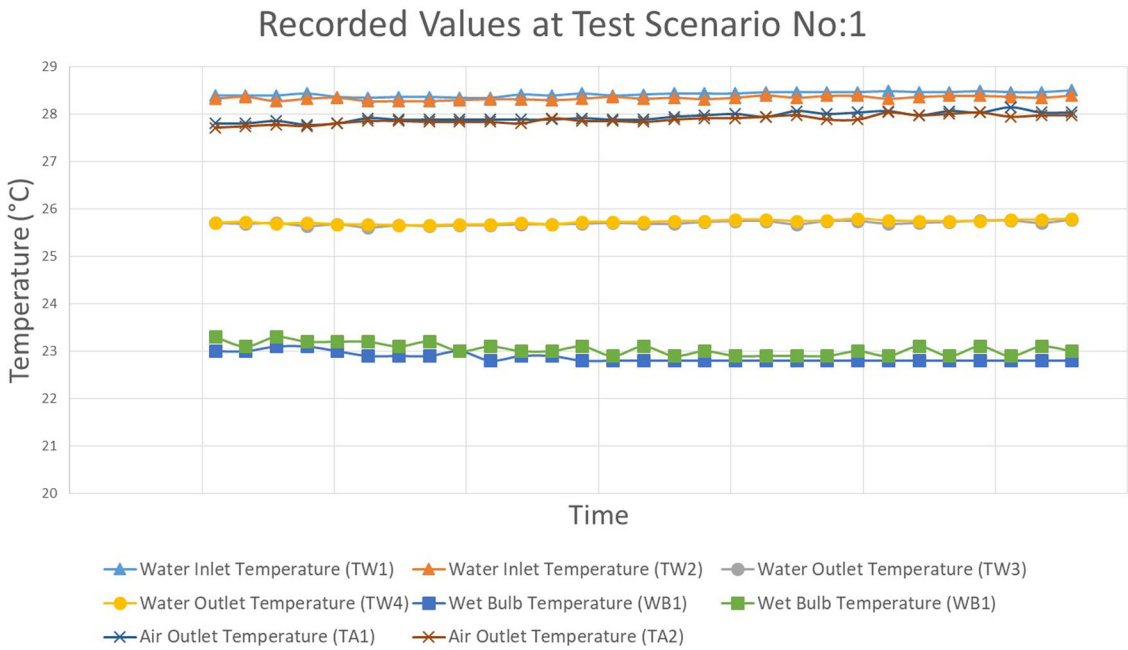

Fig. 6. Recorded values for test scenario sample 1.

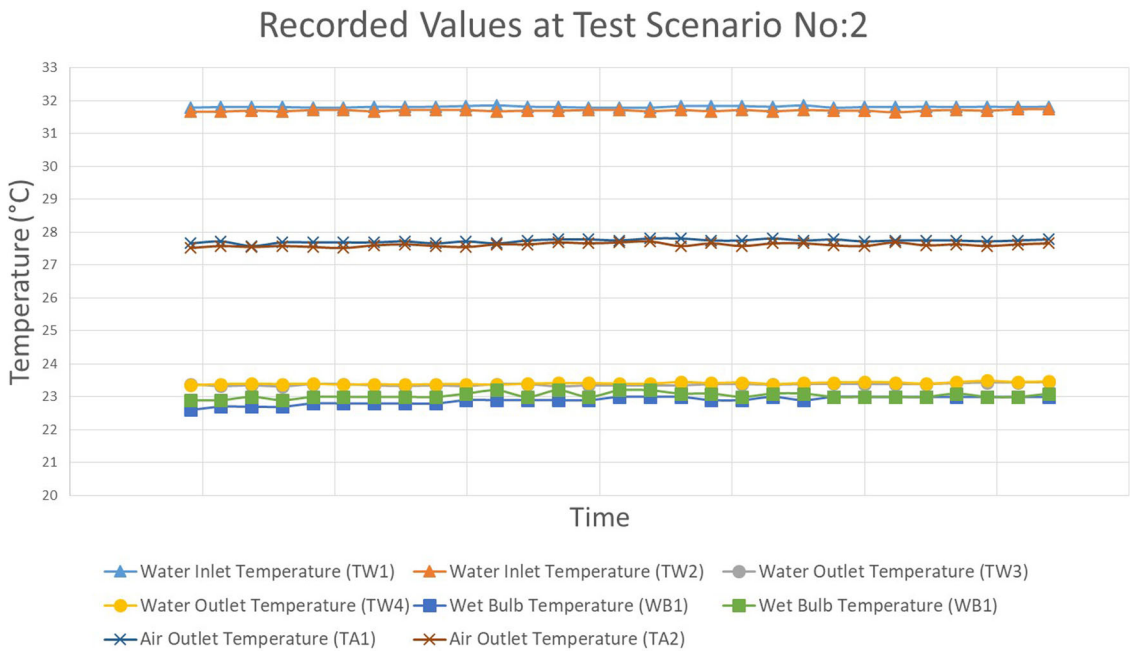

Fig. 7. Recorded values for test scenario sample 2.

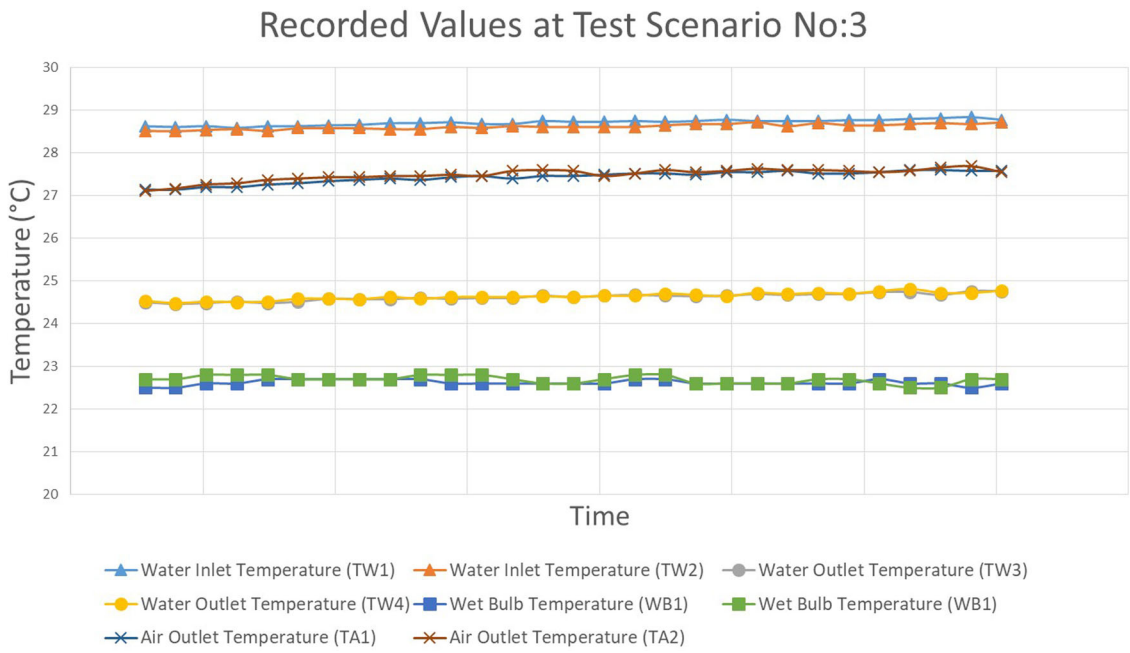

Fig. 8. Recorded values for test scenario sample 2.

$$Le_f = \frac{0,866^{0,667}\left(\frac{w_{sw}+0,622}{w+0,622}-1\right)}{\ln\left(\frac{w_{sw}+0,622}{w+0,622}\right)} \tag{8}$$

In Merkel's approach, the Lewis factor is accepted as equal to 1. It was reported that the Lewis factor was almost without effect on the obtained results. Equation 9 is regulated by Merkel's approach. It is impossible to obtain outlet values of air by conventional methods. Temperature and humidity of the air must also be known.

$$\frac{KA}{L} = \frac{KaA_{fr}l_{fi}}{L} = \int_{T_{wo}}^{T_{wi}} \frac{c_{pw}dT_w}{(i_{masw}-i_{ma})} \tag{9}$$

The basic and reported formula for the Merkel approach is given in Equation 10:

$$Me = \frac{KaV}{L} \tag{10}$$

Me is also known as the Merkel number. It is a nondimensional performance coefficient. The conventional method to

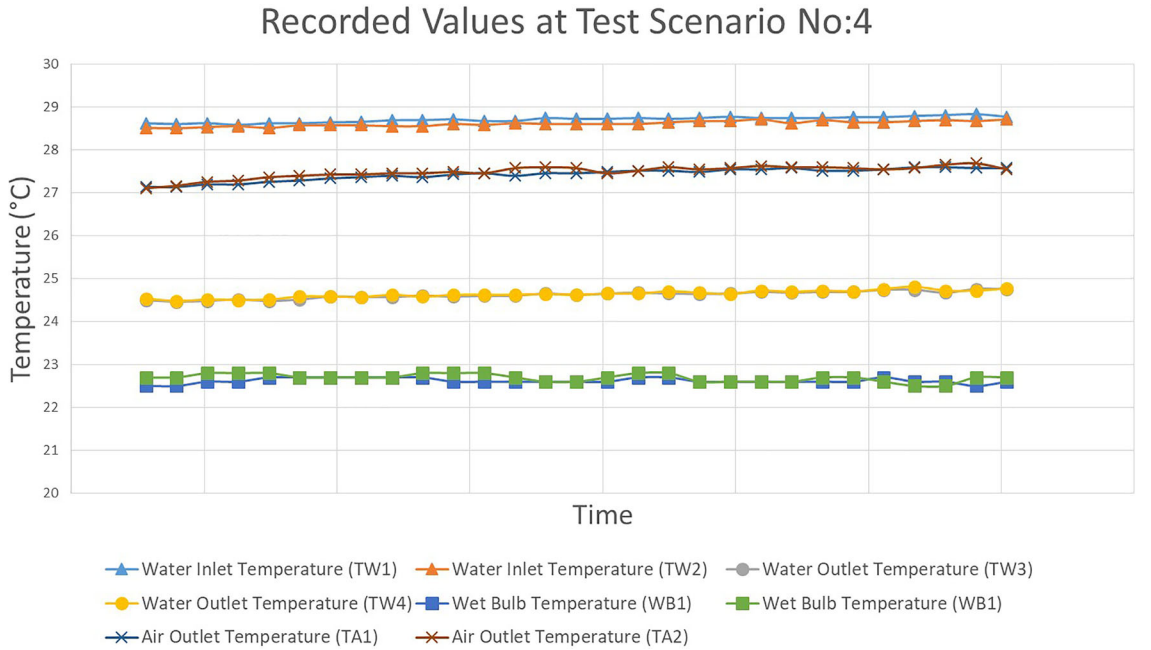

Fig. 9. Recorded values for test scenario sample 4.

Table 4. Test scenario sample 1.

Item	Unit	Value
Water flow	m^3/h	52.00
Fan motor frequency	Hz	40.00
Fill height	mm	1500

Table 5. Test scenario sample 2.

Item	Unit	Value
Water flow	m^3/h	17.00
Fan motor frequency	Hz	50.00
Fill height	mm	1500

Table 6. Test scenario sample 3.

Item	Unit	Value
Water flow	m^3/h	67.00
Fan motor frequency	Hz	35.00
Fill height	mm	1500

Table 7. Test scenario sample 4.

Item	Unit	Value
Water flow	m^3/h	35.00
Fan motor frequency	Hz	40.00
Fill height	mm	1500

calculate the Merkel integral (*KaV/L*) makes use of an enthalpy–temperature diagram, as shown in Figure 5 (Shah et al. 2015).

The water temperature and air enthalpy are changed along the fill zone, and the Merkel relation can only be applied to a small element of the heat transfer surface. In Figure 5, i_{masw} describes the water side and i_{ma} describes the air side. Enthalpy of the saturated air may be calculated for every different point, but Equation 3 must be used to calculate i_{ma}, defined in Equation 11. Evaporation loss is ignored in this equation because of calculation algorithm. This is a pre-iteration formula, and real evaporation loss value can be calculated after the *L/G* value has been determined.

$$i_{ma} = i_{ma,inlet} + \frac{L\frac{dT_w}{dz}c_{p_{w,mean}}}{G} \tag{11}$$

This enthalpy difference value gives information about *KaV/L*, in other words, the Merkel number. As this value decreases, there is an increase in *KaV/L*. Then the required airflow for cooling will also increase. This means that the *L/G* ratio depends on *KaV/L*, as well as air and water conditions.

Performance algorithm improvement with Chebyshev theorem at Merkel method

Many different methods have been used to reach a result from the Merkel approach. The most well-known method uses the four-point Chebyshev theorem in the analysis of integrals. Li and Priddy (Li and Priddy 1985) used a 13-point Chebyshev theorem in their studies, while Mills (Mills 1995) used a seven-point analysis method.

The method used here is according to British Standard 4485 (British Standard 4485 1988). A four-point Chebyshev theorem with Merkel's equation is given in Equation 12:

$$\int_{T_{wo}}^{T_{wi}} f(x)dx \approx \frac{b-a}{4}\left[f(x)_1 + f(x)_2 + f(x)_3 + f(x)_4\right] \tag{12}$$

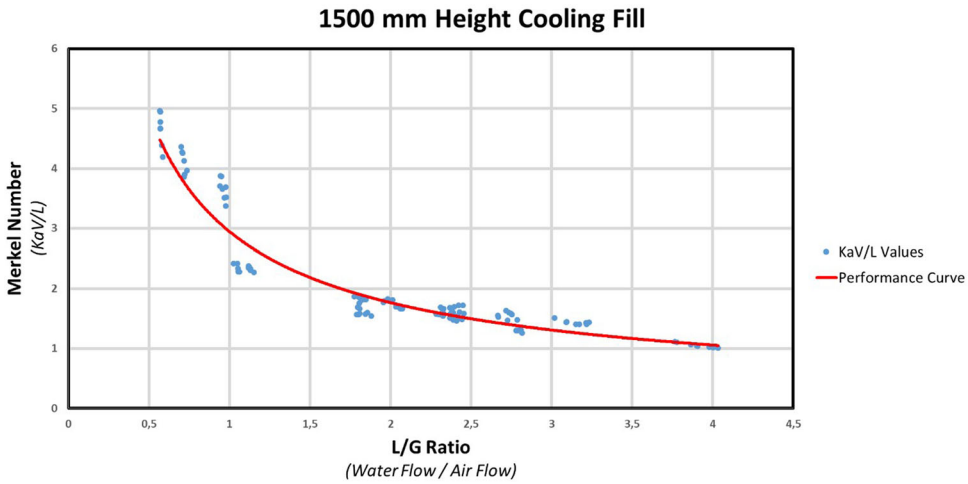

Fig. 10. Performance curve for 1500 mm height cooling fill.

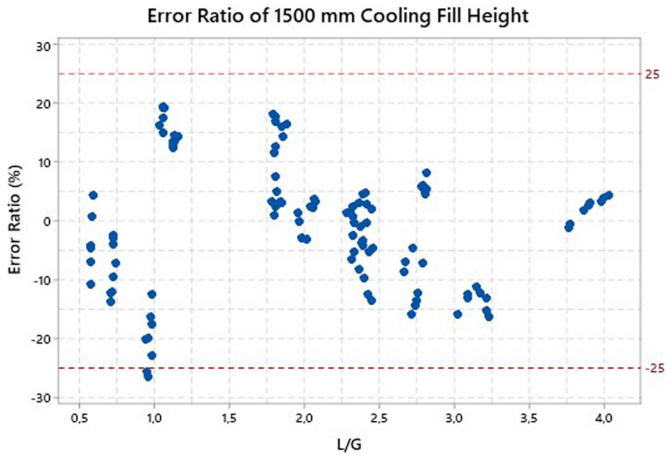

Fig. 11. Error analysis for 1500 mm height cooling fill.

According to Chebyshev, the next formula step from Equation 12 is given as Equation 13:

$$\begin{aligned} f(x_1) &\approx T_{wo} + 0.1(T_{wi} - T_{wo}) \\ f(x_2) &\approx T_{wo} + 0.4(T_{wi} - T_{wo}) \\ f(x_3) &\approx T_{wo} + 0.6(T_{wi} - T_{wo}) \\ f(x_4) &\approx T_{wo} + 0.9(T_{wi} - T_{wo}) \end{aligned} \tag{13}$$

The next stage of the algorithm is shown in Equation 14. This equation is the final step in calculating the Merkel number:

$$\begin{aligned} \frac{KaV}{L} &\approx \frac{(T_{wi} - T_{wo})}{4}\left[\frac{c_{pw1}}{\Delta i_1} + \frac{c_{pw2}}{\Delta i_2} + \frac{c_{pw3}}{\Delta i_3} + \frac{c_{pw4}}{\Delta i_4}\right] \\ &\approx \frac{c_{pwm}(T_{wi} - T_{wo})}{4}\left[\frac{1}{\Delta i_1} + \frac{1}{\Delta i_2} + \frac{1}{\Delta i_3} + \frac{1}{\Delta i_4}\right] \end{aligned} \tag{14}$$

In the end, many equations can be used for calculating the Merkel number. The equations and their usage areas are provided in Table 3.

Empirical approach correlations

The Merkel approach allows the calculation of heat transfer capacity by ignoring unknown parameters and using some empirical equations. In this calculation algorithm, the design airflow, *L/G* value, and outlet air temperature were ignored at the beginning. These were calculated after the *L/G* value was determined. There are many empirical correlations to use in calculating the Merkel number, depending on the type of tower and the design conditions. The most commonly used correlation in the literature and market is given in Equation 15 (Kröger 2004). Reuter et al. studied a performance comparison of the crossflow cooling tower fill, and they used the performance equation that is given here as Equation 15 (Reuter, Erens, and Dreyer 1992):

$$Me = \frac{KaV}{L} = c\left(\frac{L}{\mathrm{G}}\right)^n \tag{15}$$

An empirical formula option used in cooling tower performance calculation by Lowe and Christie is shown as Equation 16 (Lowe and Christie 1961):

$$Me = \frac{KaV}{L} = c_1\left(\frac{L}{\mathrm{G}}\right) + c_2 \tag{16}$$

An empirical formula option used by Majumdar in cooling tower performance calculation is provided as Equation 17 (Majumdar 1983):

$$Me = \frac{KaV}{L} = c_1 L^{c_2} G^{c_3} \tag{17}$$

According to Johnson, the empirical formula option in Equation 18 can be used in cooling tower performance calculation (Johnson 1989):

$$Me = \frac{KaV}{L} = c_1 L^{c_2} G^{c_3} l_{fi}^{c_4} \tag{18}$$

In this study, the Merkel number for cooling fill is calculated using Equation 15. The system algorithm was set between Equation 14 and Equation 15 to calculate this system, and the *L/G* value was iterated until system equalized, as shown in Equation 19:

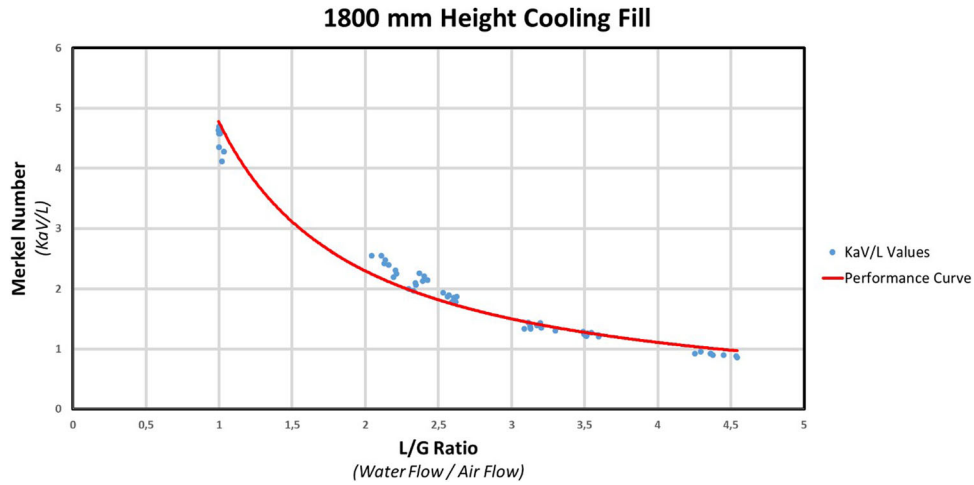

Fig. 12. Performance curve for 1800 mm height cooling fill.

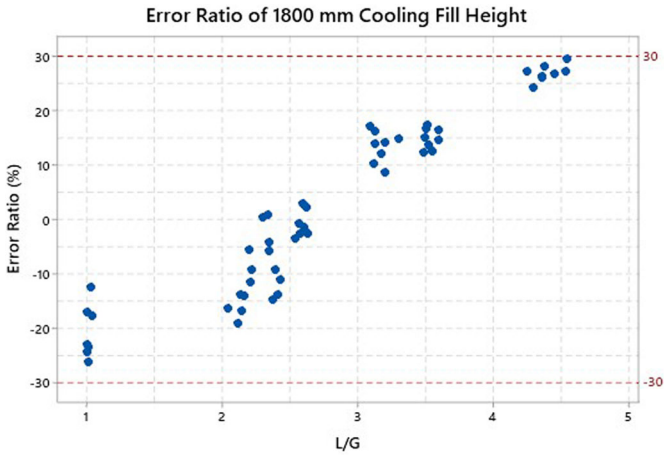

Fig. 13. Error analysis for 1800 mm height cooling fill.

$$c\left(\frac{L}{G}\right)^{n} \approx \frac{c_{pwm}(T_{wi}-T_{wo})}{4}\left[\frac{1}{\Delta i_1}+\frac{1}{\Delta i_2}+\frac{1}{\Delta i_3}+\frac{1}{\Delta i_4}\right] \quad (19)$$

Pressure loss calculations

Cooling towers are industrial equipment and need fresh atmospheric air when they cool water. Airflow is required for heat transfer, and it loses pressure in some sections of the cooling tower. These sections are the air inlet louver, rain zone, fill zone, spray zone, drift eliminator area, and fan stack entrance. Such pressure loss makes it difficult for the fans to move the air, which leads motors to work harder and consume more electricity. More motor work means increased electricity consumption. A general pressure loss formula is shown as Equation 20 (Kloppers and Kröger Detlev 2003):

$$\Delta p_{fi} = \frac{k_{fi}\rho_a v_a^2}{2} \quad (20)$$

The k_{fi} value used as a variable in Equation 20 is calculated for different geometric structures and parameters to find the fill pressure loss. Due to the increasing effect of air velocity on pressure loss, the velocity of air is selected to be as low as possible in cooling tower designs (Lemouari, Boumaza, and Kaabi 2011)

Experimental results

The air-side pressure loss values were measured in the cooling tower performance tests, and the thermal performance parameters were evaluated. General performance test procedures of Cenk Endüstri Tesisleri İmalat (CTI) and other institutions were followed. Measurement data were collected after a thermal balance was reached. Every test condition was recorded for 1 hour after the system had been stabilized. Many parameters, such as water flow, air velocity, and heat load, and others, were changed to establish a curve for a cooling fill type. In this study, more than 20 performance tests were performed to establish these curves. These tests and procedures were restarted for each different fill height option. At the end of the experimental study, performance coefficients and pressure loss graphs for four different heights of cooling fill had been obtained. The values of the input parameters and output parameters corresponding to these inputs are indicated in Tables 4–7 and Figures 6–9, respectively.

The process sides of the test cooling tower were evaluated by measuring the thermophysical properties. Air inlet enthalpy values were assessed using a wet bulb temperature transmitter that has its own fan and water supply system for measurement at 100% relative humidity. Outlet temperature was measured using a standard PT100 temperature transmitter. The relative humidity of the outlet air was found to be 100% at multiple outlet air points during the performance tests. A hand-held humidity sensor, with accuracy ±2.5%, was used to measure relative humidity. From the recorded values, inlet and outlet enthalpy of air could be calculated. On the other hand,

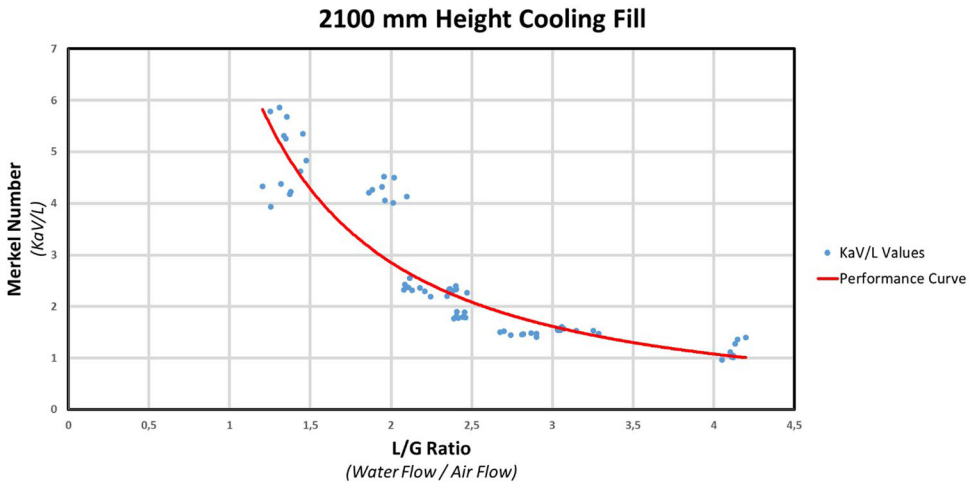

Fig. 14. Performance curve for 2100 mm height cooling fill.

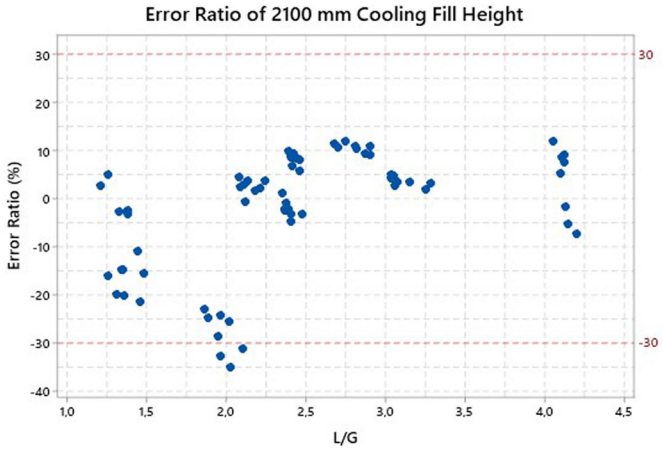

Fig. 15. Error analysis for 2100 mm height cooling fill.

the air mass flow rate could be calculated from Equation 3.

There are more conventional methods to measure air speed in cooling towers by using anemometers. However, our experimental performance tests were made in open air. The winds could distort the measurement of air speed. For more accurate results, differential enthalpy was calculated for this process.

Analyses of thermal performance test results

The *L/G* term is the ratio of liquid flow to gas flow. Performance graphics for cooling fill were created from the *L/G* and Merkel number values. The frequencies of the circulation pump and fan motor were changed for different flow options. In this study, five different airflow options and a minimum of four different water flow options were measured.

According to Merkel, it is necessary to know performance coefficients of the cooling fill in order to design to the thermal capacity of the cooling tower. One of the main objectives of this study is to obtain these performance coefficients experimentally. Mainly Equation 19 and others were used for this analysis. Cooling fill performance curves for different heights and their error ratio charts are shown from in Figures 10– to 17.

Performance curves for different slopes were obtained under different conditions for performance test results. Performance curves and their coefficients were used to create an exponential function. These coefficients had different exponents in the first analysis. To create better curves, exponent values were equalized to −0.74. All the *L/G* and *KaV/L* values were recalculated using this exponent and an average *c* coefficient was selected for related cooling fill. Corrected performance curves for all cooling fill heights are given in Figure 18.

Coefficients were determined using Equation 15 for every type of cooling fill. Calculated *c* and *n* values are shown in Table 8.

Pressure loss test results and analyses

The air-side pressure loss at cooling fill generally is generally affected by cooling fill type and its height, water loading, and air velocity. Water load or water loading is the value of the water flow rate for a unit area. While analyzing the thermal values of the cooling tower, pressure loss values were recorded simultaneously, and these were added to results of the analysis. Pressure loss values for different cooling fills, velocities, and variable water load options are shown in Figures 19 to –22.

Values were generally recorded in the summer. Wet bulb temperatures for the area were between 22 °C and 23 °C and altitude was 80 m. Thus, these values are valid for approximately 1.168 kg/m^3 air density.

Cooling fill height increase affected pressure loss positively. Similarly, the increase in air velocity and water load increased the pressure loss too.

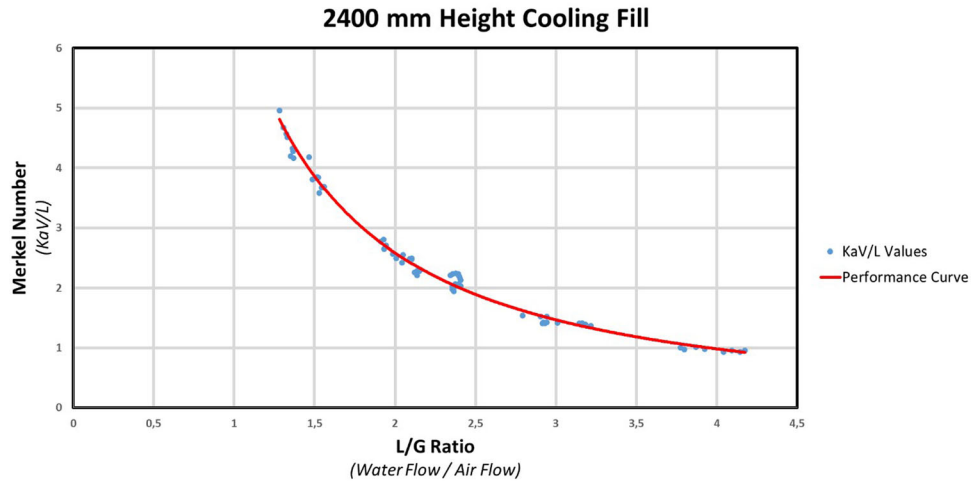

Fig. 16. Performance curve for 2400 mm height cooling fill.

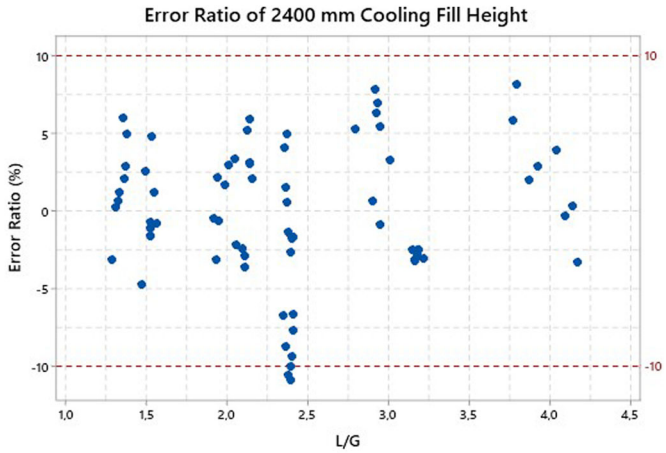

Fig. 17. Error analysis for 2400 mm height cooling fill.

Table 3 shows variations for design conditions. Iteration is solved with mathematical methods and the *L/G* value obtained is 2.54. This means that 1 kg/s of airflow is needed for cooling 2.54 kg/s of water flow in this condition and with this cooling fill. There are many factors affecting thermal performance in a cooling tower (e.g., water and air distribution inside it). But for the calculation in Table 10, all other parameters are ignored.

In the end, the requested airflow is calculated from the *L/G* ratio and design water flow. It is 389.1 kg/s of mass airflow. Designers then can select cooling tower dimensions, calculate air speed through the cooling fill section, and specify its fans and electrical motors for finalizing the thermal design side.

Sample thermal design calculation

Airflow design values for different cooling fills were examined and calculated. The requested design airflow can be reached by this method.

First, thermal and mechanical data are required for cooling tower calculation. Thermal information includes the water inlet, water outlet, and wet bulb temperature values and the mass flow rate of the water. Only the cooling fill performance coefficients are needed for deriving the Merkel number. Design parameters are given in Table 9.

For this algorithm, water flow is known but airflow is unknown. The *L/G* value is needed to iterate toward acquiring the airflow rate. The performance coefficient for 1800 mm fill height can be found in Table 8. The values for the sample calculation are shown in Table 10.

Different cooling fill height performance comparison for different design cases

At the end of this study, a counterflow wet cooling tower with dimensions 10 m × 10 m was modeled for comparing the performance of four different fill heights. Three different design conditions were used for comparison. Design conditions are shown in Table 11.

In first design condition, the 1500 mm cooling fill height requires 17% more airflow than for 2400 mm height. This ratio is 10% at the second and 7% at the third design condition. These ratio differences are clearly shown in Figure 23.

Air side pressure drop values for different design condition are calculated for every cooling fill. All other pressure drop sources are ignored, and the fan static efficiency was considered to be 60% for every design. The required electrical consumption ratio values for different cooling fill heights are shown in Figure 24.

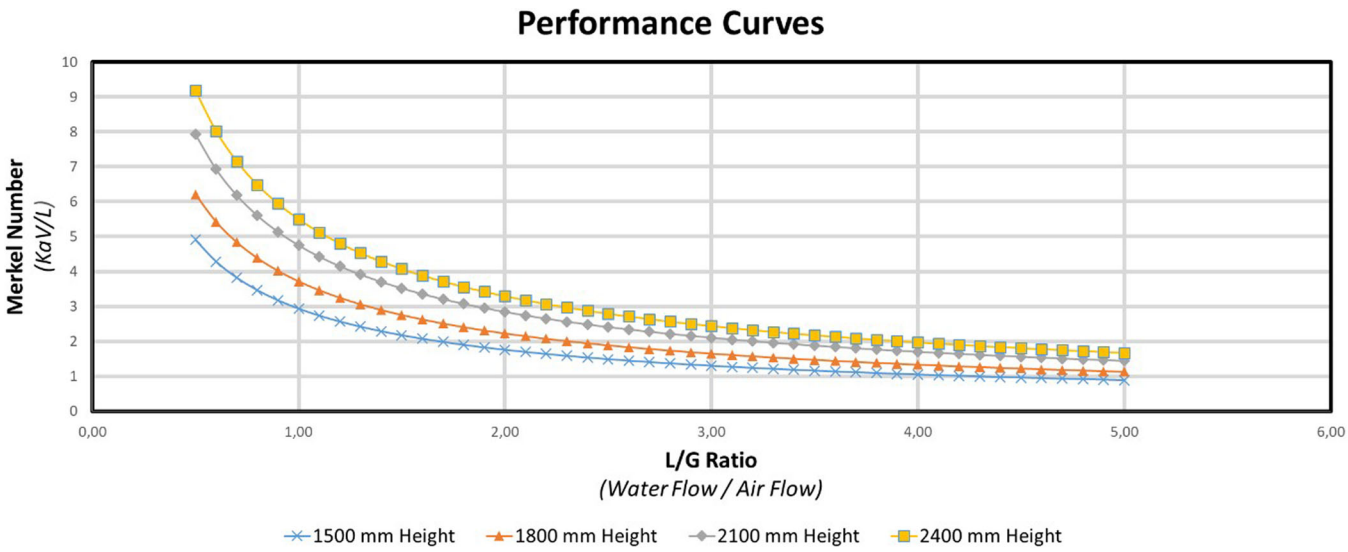

Fig. 18. Performance curves for cooling fills at different heights.

Table 8. Performance coefficients according to Equation 11.

Cooling fill height (mm)	*c* coefficient	*n* coefficient
1500	2.95	–0.74
1800	3.72	–0.74
2100	4.75	–0.74
2400	5.50	–0.74

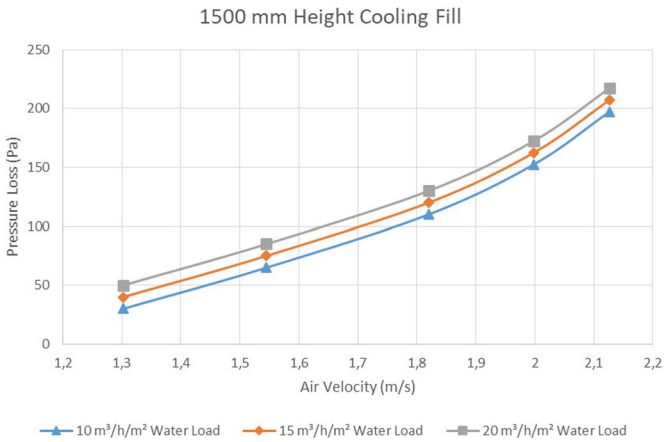

Fig. 19. Pressure loss curves for 1500 mm height cooling fill.

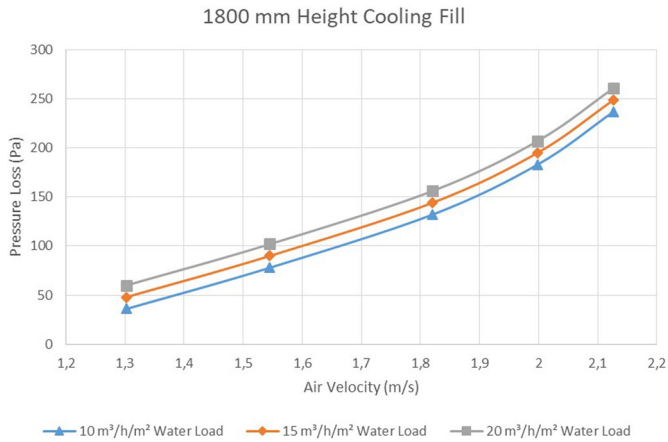

Fig. 20. Pressure loss curves for 1800 mm height cooling fill.

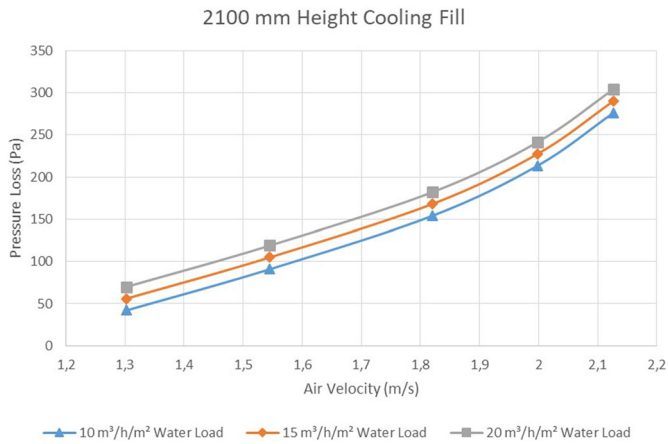

Fig. 21. Pressure loss curves for 2100 mm height cooling fill.

Conclusion

Throughout all the tests and analyses, cross-fluted type cooling fills with different heights were tested experimentally. At least 20 different thermal performance and pressure loss tests were recorded and analyzed according to the equations tailored for each cooling fill height. As the final step, thermal performance curves and coefficients and also pressure-loss curves were obtained to contribute to designs for new cooling towers. The efficient cooling fill height changes under different design conditions. According to Figure 24, in the first design condition the most efficient cooling fill height is 2100 mm, and this is also the most efficient cooling

fill for the second design condition. In third design condition, 1500 mm is the most efficient cooling fill height. Thus, cooling fill height and air velocity need to be specified with care.

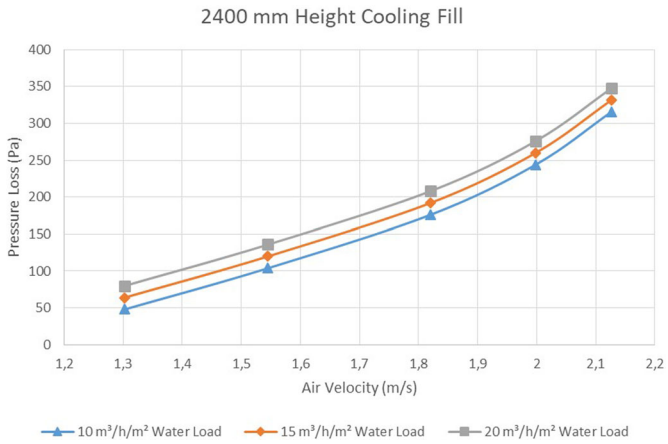

Fig. 22. Pressure loss curves for 2400 mm height cooling fill.

Table 9. Thermal design parameters for sample calculation.

Design parameter	Value	Unit
Water flow rate	1000	kg/s
Water inlet temperature	35	°C
Water outlet temperature	30	°C
Wet bulb temperature	24	°C
Cooling fill height	1800	mm

The experimental results of this study confirmed that increasing the cooling fill height affects thermal performance positively. Footprints of a cooling tower can decrease by up

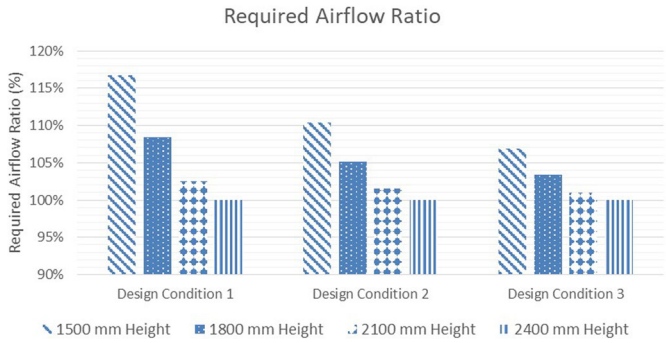

Fig. 23. Required airflow ratio for different cooling fill heights.

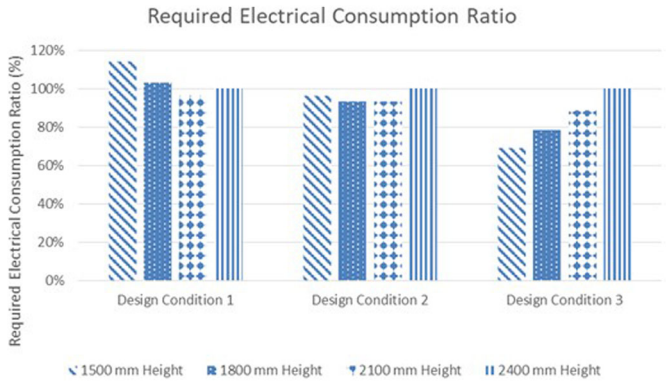

Fig. 24. Required electrical consumption ratio for different cooling fill heights.

Table 10. Sample calculation values.

Water side		Air side	Enthalpy difference	
Water temperatures (°C) T_w	Enthalpy of the saturated air at local water temperature (kJ/kg) i_{masw}	Enthalpy of moist air (kJ/kg) i_{ma}	$i_{masw} - i_{ma}$	$1/(i_{masw} - i_{ma})$
30.5	*102.31*	$72.16 + 0.1\ (L/G)\ (35–30)$	$i_{masw.1.}$- $i_{ma.1.}$	$1/(i_{masw.1.}$- $i_{ma.1.})$
32	*110.59*	$72.16 + 0.4\ (L/G)\ (35–30)$	$i_{masw.2.}$- $i_{ma.2.}$	$1/(i_{masw.2.}$- $i_{ma.2.})$
33	*116.44*	$72.16 + 0.6\ (L/G)\ (35–30)$	$i_{masw.3.}$- $i_{ma.3.}$	$1/(i_{masw.3.}$- $i_{ma.3.})$
34.5	*125.73*	$72.16 + 0.9\ (L/G)\ (35–30)$	$i_{masw.4.}$- $i_{ma.4.}$	$1/(i_{masw.4.}$- $i_{ma.4.})$
$3.72\frac{L}{G}^{-0.74}$				$\frac{4.179(35-30)}{4}\sum \Delta i$

Table 11. Thermal design parameters for sample calculation.

Design condition	3K approach (high air velocity design)	6K approach (intermediate air velocity design)	9K approach (low air velocity design)
Water flow (kg/s)	500	500	500
Water inlet temperature (°C)	32	35	40
Water outlet temperature (°C)	27	30	35
Wet bulb temperature (°C)	24	24	24

to 17% when using a high cooling fill height option, but this increases pressure loss, too. If operational costs can be ignored, smaller cooling towers are economic solutions for facilities.

Symbols

m_a	Air mass flow rate	(kg/s)
w	Humidity ratio	(kg/kg)
m_w	Water mass flow rate	(kg/s)
i_{ma}	Enthalpy of moist air	(kJ/kg)
i_{masw}	Enthalpy of saturated air at local water temperature	(kJ/kg)
c_{pw}	Specific heat of water	(kJ/kg-K)
T_w	Temperature of water	(°C)
Q	Total heat transfer capacity	(kW)
Q_m	Latent heat transfer capacity	(kW)
Q_c	Sensible heat transfer capacity	(kW)
w_{sw}	Humidity ratio of air at water temperature	(kg/kg)
K	Mass transfer coefficient	(kg/m^2-s)
A	Total heat transfer surface area	(m^2)
i_v	Enthalpy of water vapor	(kJ/kg)
Le_f	Lewis factor	(-)
a	Surface area density of cooling fill	(m^2/m^3)
A_{fr}	Frontal section area	(m^2)
l_{fi}	Air travel distance at cooling fill	(m)
Me	Merkel number(-)aSurface area density	(m^2/m^3)
V	Volume of the cooling fill ($A_{fr}l_{fi}$)	(m^3)
L	Mass water flow rate	(kg/s)
c_n	Performance coefficient	(—)
k_{fi}	Pressure drop coefficient of cooling fill	(—)
ρ_a	Density of air(kg/m^3)v_aVelocity of air	(m/s)
h	Heat transfer coefficient	(W/m^{2}K)

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