New Journal and we have not received input yet 22 (2021) 100835

Contents lists available at ScienceDirect

Thermal Science and Engineering Progress

journal homepage: www.sciencedirect.com/journal/thermal-science-and-engineering-progress

Experimental investigation of fill pack impact on thermal-hydraulic performance of evaporative cooling tower

A.V. Dmitriev [a], I.N. Madyshev [b], V.V. Kharkov [b,*], O.S. Dmitrieva [b], V.E. Zinurov [a]

[a] *Kazan State Power Engineering University, 420066, Krasnoselskaya Street 51, Kazan, Russia*

[b] *Kazan National Research Technological University, 420015, Karl Marx Street 68, Kazan, Russia*

A R T I C L E I N F O

Keywords:
Evaporative cooling tower
Fill
Pressure drop
Heat and mass transfer coefficient
Merkel number

A B S T R A C T

The present work deals with an experimental study of the performance of an evaporative cooling tower (ECT) with a developed fill pack. The fill pack consists of inclined-corrugated contact elements (ICCE) made of the metal plates with perforations, providing a uniform distribution of interacting phases over the ECT cross-sectional area. This study investigates the effect of fill pack design on hydraulic and thermal characteristics of the ECT. Four empirical equations were found for the pressure drop through four types of dry fill packs. The detailed analysis includes two fill packs consisting of ICCE with 6 mm holes with and without a metal grid. Hydraulic operating regimes of the ECT and empirical equations for the wetting pressure drop through the fill packs had been defined. Based on experimental data and the method of transfer units, empirical relationships of the volumetric mass transfer coefficients were determined. The comparison between the obtained results and those found in the literature for other types of proves the high performance of the developed fill packs. Besides, using the Lewis relation, the dependencies of the volumetric heat transfer coefficient on the average air velocity were analyzed. The results indicate that maximum cooling efficiency of the studied fill packs was observed at the low wetting rates according to the Merkel number. The fill pack from ICCE with 6 mm holes and without the metal grid seems to be more efficient, as it provides relatively higher values of the heat and mass transfer coefficient and lower pressure drop.

1. Introduction

Cooling towers are devices widely used to dissipate unwanted thermal energy in power generation units, HVAC systems, petrochemical, and chemical industries. Cooling towers operation based on mass and thermal energy transfer from high-temperature water to coolant air. Although many technologies are available for the heat rejection process, the wet cooling towers or evaporative cooling towers (ECTs) are more attractive due to their flexibility in handling large heat loads. Also, they are relatively inexpensive and reliable. The fundamentals of the physical phenomena that occur in the cooling towers were reported by Walker, Merkel, Nottage and Poppe. Moreover, the Merkel method was recommended as a standard method in cooling tower performance research.

Inefficiency in the cooling process of ECTs results in a continuous loss of power generation or lower quality product. So, enhancing of thermal performance of cooling towers has always been of particular interest. Many factors, such as fill (or packing) types, a flow rate of air and water, inlet temperature of the process water, affect the ECT operation. Therefore, the cooling efficiency could be improved by obtaining the optimum values of these parameters. For this sake, numerous studies have been focused on the thermal performance of the ECTs under diverse operating conditions through experimental and theoretical analyses [1–9].

Furthermore, in the design of the new cooling towers, a crucial factor is to select the optimal design of the fill pack, which can intensify heat and mass transfer processes with minimal energy resources [10–14]. According to the previous studies, the factor influencing the ECT efficiency is mainly the thermal performance of the fill zone, because up to 70% of heat-dissipating capacity occurs in this zone [15]. Therefore, it is essential to study transport phenomena in the fill zone.

Goshayshi and Missenden [16] studied the effect of form and surface roughness of the corrugated fills on pressure drop and mass transfer in an experimental cooling tower. They showed that the mass transfer coefficient decreased with increasing pitch and distance between the fills. Gharagheizi et al. [17] experimentally investigated the cooling tower performance using two film fills. The results showed that the tower with vertical corrugated fills has higher efficiency than horizontal

* Corresponding author.
E-mail address: v.v.kharkov@gmail.com (V.V. Kharkov).

https://doi.org/10.1016/j.tsep.2020.100835
Received 28 October 2020; Received in revised form 27 December 2020; Accepted 27 December 2020
Available online 31 December 2020

Nomenclature

A, n	constants in Eq. (6)
A_1, a	hydraulic constants in Eq. (1)
b	exponent in Eq. (2)
c_p	specific heat at constant pressure, J/(kg·K)
D	diffusion coefficient, m^2/s
G_m/L_m	gas to liquid mass flow rate ratio
H	height of the fill pack, m
h	specific enthalpy, J/kg
K	constant in Eq. (9)
Me	Merkel number
ΔP_{dry}	pressure drop through the non-wetting (dry) fill pack, Pa
ΔP_{wet}	pressure drop through the wetting fill pack, Pa
Pr	Prandtl number
q	wetting rate
r	specific latent heat of evaporation, J/kg
Sc	Schmidt number
t	temperature, K
V	volume of the fill pack, m^3
W	gas velocity, m/s
x	moisture content of the air, kg/kg

Greek letters

α_v	volumetric heat transfer coefficient, W/(m^3·K)
β_v	volumetric mass transfer coefficient, kg/(m^3·s)
δ	relative error, %
λ	thermal conductivity, W/(m·K)
μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3

Subscripts

av	average
G	gas (air)
L	liquid (water)
m	mass
s	saturated
v	volumetric
1	inlet
2	outlet

corrugated fills. Also, Lemouari [18,19] performed an experimental analysis of the thermal performance for a wet cooling tower with a "VGA." (Vertical Grid Apparatus) type fill. They concluded that two defined operating regimes (a pellicular regime and a bubble dispersion regime) during the air and water contact inside the tower could determine the best way to promote the transport phenomena. Singla et al. [20] experimentally investigated the performance parameters in a forced draft cooling tower with an expanded wire mesh fill. They concluded that multiple combinations of the air and water flow rates satisfy a given Merkel number, which the operator can utilize for regulating desired conditions.

Rahmati et al. [21,22] have contributed with studies about the effect of fill stages and inlet parameters on the thermal performance of wet cooling towers. Their findings provided that the cooling efficiency is directly related to the stage numbers of fill, hot water temperature, and the air mass flow rate. In contrast, it decreases with increasing the water flow rate. Besides, in experimental work [23], the authors examined the influence of the arrangement and type (7, 9, and 18 ribs) of fills on the wet cooling tower performance. The results showed that the temperature drop of water and the cooling efficiency increase when adding ribs.

Rotational splash-type fill of a forced draft cooling tower was the subject of Lavasani's research [24]. The paper reports that this fill with more rotational velocity can significantly increase heat rejection from water. Recently, Amini et al. [25] experimentally investigated the thermal performance of the rotational splash type fill using nanofluids. The results showed that the performance depends on the type of nanofluids, the inlet temperature, and the concentration of nanofluid. Gao et al. [26] performed an experimental study on the thermal performance of the wet cooling towers with five kinds of layout patterns of fill. They pointed out that the non-uniform layout patterns can improve the thermal performance by 30% at maximum compared with uniform layouts.

Besides experimental works, many theoretical research papers have been developed to estimate the thermal performance of the ECTs. Milosabljevic and Heikkila [27] derived a mathematical model that can predict the thermal performance of different fill materials. Petruchik et al. [28] presented a mathematical model of evaporative cooling of water films using double-corrugated polyvinyl chloride sheets. Xia et al. [29] numerically studied a closed wet cooling tower, which consisted of two main parts: one heat and mass transfer unit and one heat transfer unit. They determined the heat and mass transfer coefficients and the effect of Lewis number on the ECT performance. Xie et al. [30] numerically investigated a model of the thermal-hydraulic performance of the closed ECTs with various fin tubes. The authors developed the correlations of the heat and mass transfer coefficients and the pressure drop for three cases (plain, oval, and longitudinal fin tube). Several other mathematical models deal with heat and mass transfer phenomena in the filling zone of the cooling towers, such as the works presented in [31–35].

In the previous study [36], we developed a new fill pack design with the inclined-corrugated contact elements (ICCE) for the evaporative cooling tower. Distinctive features of the proposed fill pack are as follows:

- high flow capacity in both the gas phase and liquid phases;
- uniform distribution of liquid over the entire cross-sectional area of the ECT with a simple water distribution system;
- entrainment of liquid droplets from the apparatus does not surpass 3–5% in the range of the average gas velocity from 1.5 to 2.4 m/s;
- reduced energy costs by eliminating the need for high-pressure atomizers;
- low-pressure drop;
- easy repair, maintenance, and operation of both the fill pack and auxiliary devices.

From the above brief review of research works, it follows that evaluating the pressure drop and heat and mass transfer coefficients of the fills is urgent for further energy-saving studies and engineering of new industrial types of the ECTs. Therefore, the paper's objective is to examine the effect of the design of the developed fill pack with the ICCE on hydraulic and thermal characteristics of the ECT.

The following tasks had to be fulfilled to achieve the objective of this research:

- determination of empirical equations of the pressure drop through the non-wetting and wetting developed fill packs of various types;
- evaluation of the volumetric mass transfer coefficient during cooling water in the fill packs consisting of the ICCE with 6 mm holes with and without the metal grid at the different wetting rates;
- comparative analysis of the volumetric mass transfer coefficients of the fill packs under study with other types of fill depending on the ratio of mass flow rates;
- determination of volumetric heat transfer coefficients of the developed fill packs;

- estimation of the cooling efficiency of the studied fill packs at the different wetting rates according to the Merkel number.

2. Experimental methodology

2.1. Description of the developed fill pack

The dimensions of the Plexiglas-made fill pack are 100 mm × 100 mm × 340 mm (L × W × H) (Fig. 1). It contains four contact elements installed at an angle of 45° to the walls, a tubular distributor of 12 mm diameter for hot water supply, and a lower tank for collecting liquid. The contact elements are corrugated metal plates with a thickness of 0.6 mm and a radius of curvature of 7.5 mm. When water flows around the corrugations, the turbulence in the films occurs even at a relatively low flow rate. The corrugated plates' side surfaces and peaks have round perforations with a 12 mm pitch for the liquid and gas flows. Besides, holes with a 3 mm diameter and a pitch of 10 mm are drilled in troughs of the corrugated plates.

2.2. Description of the experimental system

Fig. 2 illustrates an experimental setup of the ECT utilized in this study. Water and air are used as working fluids. The testing procedure is as follows.

Hot water is supplied to the central zone of the fill pack through the tubular distributor located between the first (upper) and the second ICCE. The tubular distributor's location in the experimental-system is caused by the need to create a space above it to prevent the entrainment of liquid droplets in the upward air flow from the fill pack at high average air rates (above 2.5 m/s).

The liquid flows along the surface of the plates in the following way. The main flow comes through the holes to the located-under plate, and some water flows down the pack walls. Air is blown by a fan, passing through the holes of the ICCE, and contacts the water. In this manner, the air flow sprays water in different directions along the entire volume of the fill pack. Liquid film flowing the ICCE surface is broken up due to facing formed jets and droplets.

Fig. 1. Real image of fill pack with inclined-corrugated contact elements.

Moreover, each located-above plate serves as a separable device since it prevents significant entrainment of the liquid in the upward air flow. So, this mechanism provides an advanced and continuously refreshed gas–liquid interface in the fill pack. The cooled water from the surface of the lower ICCE and the walls of the fill pack returns to the collecting tank.

The proposed mechanical-draft ECT using the perforated ICCE in the fill pack provides a uniform distribution of interacting phases over the cross-sectional area. Previously [37], we found complete mixing of liquid and gas flows across the tower's cross-section at relatively low air velocities. There is a strong case for forming the large interfacial area, which improves the ECT's performance. When designing industrial cooling towers, it is assumed to adjust the number of installed individual fill packs with the ICCE of the same size (100 × 100 mm) to suit the required capacity. This design approach can minimize the scaling up effect and allows developing devices of any given capacity without decreasing the cooling efficiency. It is worth noting that a single wetting point is sufficient since the developed fill pack's design provides the self-distribution of the liquid over its cross-sectional area. When using several fill packs (in industrial ECTs), a manifold-type collecting device supplies water from point sources.

The studied fill pack has a complex design with many geometric parameters, such as the curvature radius and the inclination angle of the plates, diameter and pitch of the holes, diameter of pipes, and their arrangement relative to each other). Besides, under actual operating conditions, it is difficult to implement similarity criteria for calculating the model cooling tower with other sizes. However, these studies' experimental results can help verify calculation methods, which enable calculating the proposed type cooling tower with different sizes or operating parameters.

One of the critical variables in ECT fill pack operation is the pressure drop. The effect of four different types of fill packs on the pressure difference has been experimentally investigated, and the most suitable fill packs were eventually introduced. A series of tests were carried out with the following types of the fill pack:

- Type 1: ICCE with holes of 5 mm diameter;
- Type 2: ICCE with holes of 6 mm diameter;
- Type 3: ICCE with holes of 6 mm in diameter and a metal grid. The grid is made of steel spiral wires vertically arranged in the pack of separate layers (Fig. 3).

This metal grid can increase the contact area available for mass and heat transfer in the fill pack. The measured void volume of the fill is from 0.98 to 0.99 m^3/m^3, which almost does not reduce the flow capacity. The selection of this type of metal grid is due to its high-performance indicators, such as the relatively high specific surface area (up to 250 m^2/m^3) and low hydraulic resistance.

- Type 4: a fill pack with a three-flow cooling circuit (Fig. 4).

This fill pack has a circuit of 30 copper tubes connected by silicone tubing. The copper tubes of diameter 8 mm were 125 mm in length. They were arranged across the ICCE, having holes 6 mm in diameter.

The experimental operating conditions and devices are shown in Table 1. An additional point to emphasize is that the anemometer is located inside a pipe designed to supply ambient air without changing temperature and humidity characteristics.

3. Results and discussions

3.1. Analysis of pressure drop

Fig. 5 shows the variations of dry pressure drop values (without water flow) against the average air velocity for four fill pack types in the ECT. It can be seen that the ICCE, with the small open cross-sectional

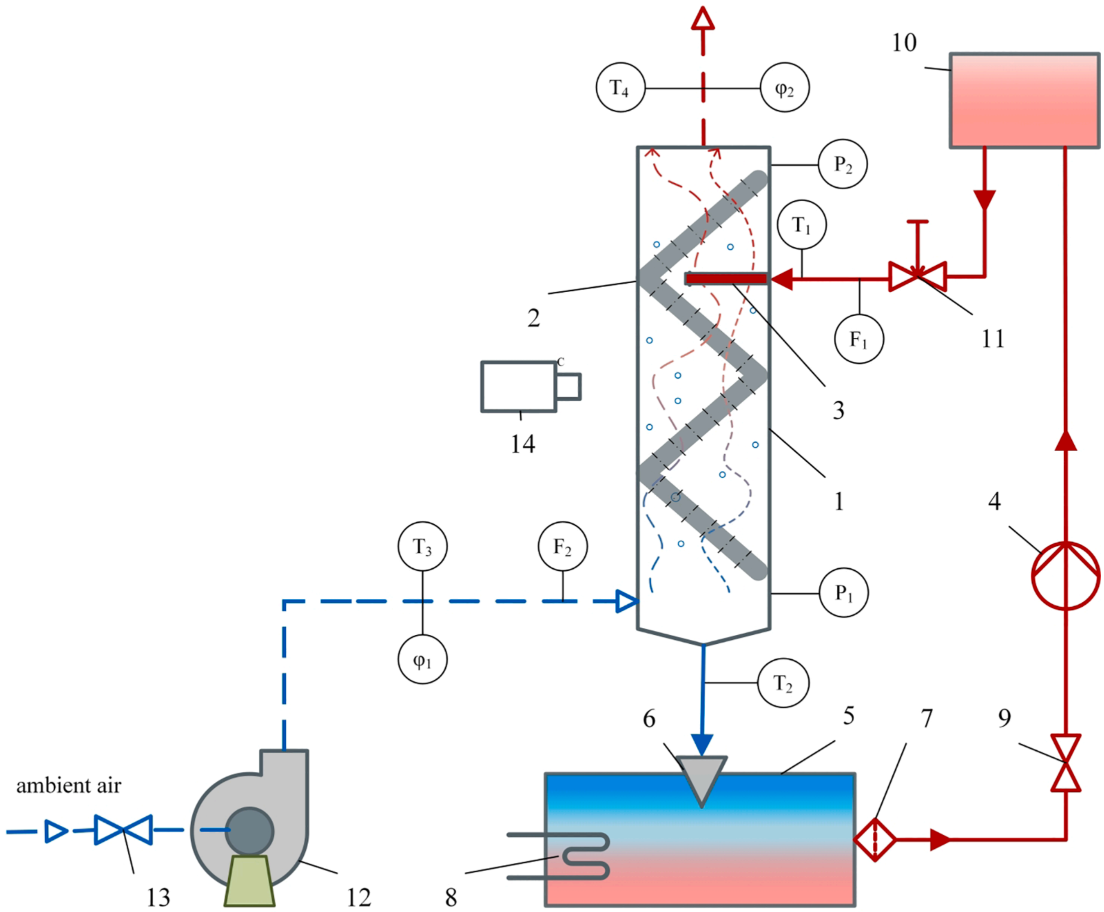

Fig. 2. Schematic diagram of experimental system: 1 – fill pack; 2 – inclined-corrugated contact elements; 3 – water distributor; 4 – pump; 5, 10 – water tank; 6 – funnel; 7 – liquid filter; 8 – heaters; 9, 11 – shutoff valves on the water supply line; 12 – fan; 13 – shutoff valves in the air supply line; 14 – camera.

Fig. 3. Real image of metal grid used in fill pack.

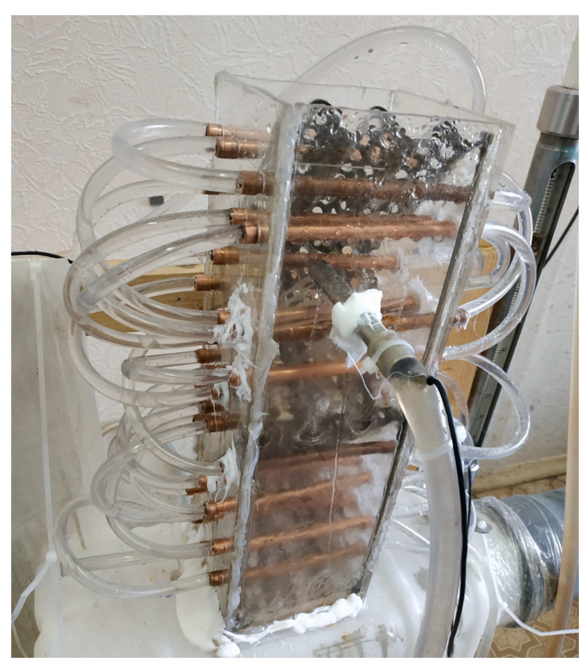

Fig. 4. Real image of fill pack with three-flow cooling circuit.

area available for air flow, in particular with a hole diameter of 5 mm (type 1), had the maximum pressure difference. Pressure drop values of other design designs of the fill pack (types 2–4) under non-wetting conditions are at the same level. The presence of the metal grid (type 3) increases the pressure drop of the dry fill pack by only 3.8% on

Table 1
Measuring and control devices specifications.

Parameter	Sensor	Range	Accuracy
Average velocity of colling air	Hot-wire anemometer TESTO 405i	1.5–2.8 m/s	±(0.1 m/s + 5% of mean value) (0–2 m/s) ±(0.3 m/s + 5% of mean value) (2–15 m/s)
Coolant air temperature	Thermohygrometer TESTO 605i	32.1–32.3 °C	±0.5 °C
Relative humidity of coolant air		34.0–34.2%	±3.0%
Water temperature	Meter-regulator OWEN 2TRM1	35.1–41.9 °C	±0.5%
Wetting rate	Rotameter LZB-VA10-15F	11.8–36.8 $m^3/(m^2{\cdot}h)$	±1.5%
Differential pressure	Differential manometer TESTO 510i	90–905 Pa	±5.0% (0–100 Pa) ±3.5% (100–1000 Pa)

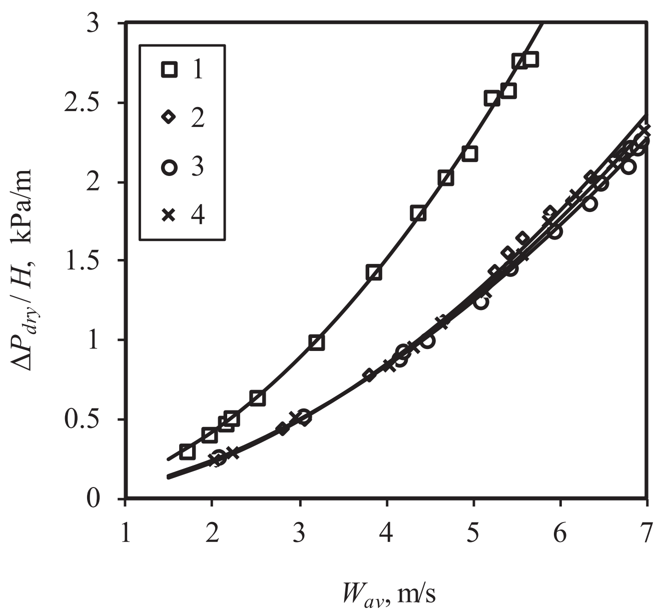

Fig. 5. Pressure drop in different types of dry fill packs: 1 – ICCE with 5 mm holes; 2 – ICCE with 6 mm holes; 3 – ICCE with 6 mm holes and metal grid; 4 – with the three-flow cooling circuit; dots – experimental points, solid lines – according to Eq. (1) (Table 2).

average (compared to type 2) at the average gas velocity up to 3 m/s. Use of the three-flow cooling circuit (type 4) does not increase ΔP_{dry}, since the maximum change is 1.76% (compared to type 2).

The least-squares method has been used in experimental data analysis. We obtained equations for determining the pressure difference for various dry fill packs (Table 2) based on the following formula:

$$\Delta P_{dry}/H = A_1\, W_{av}^{a} \qquad (1)$$

where ΔP_{dry} is the pressure drop of the dry fill pack, Pa; H is fill pack height, m; A_1, a are the hydraulic constants for each particular fill pack, determined experimentally; W_{av} is the average gas velocity, m/s.

The pressure drop of the wetting fill pack of type 3 is higher than that of the dry fill pack, as indicated in the characteristic diagram (Fig. 6). In particular, the pressure difference increases with growing the wetting rate. Loading points (lines I-I) correspond to the average gas velocity over a range from 1.5 to 2.4 m/s depending on the wetting rate. As the air rate rises, flooding points (lines II-II) are reached where the curve

Table 2
Empirical equations of pressure drop through non-wetting fill pack with relative errors.

Type of fill pack	Equation	Coefficient of determination	Maximum relative error, %
1	$\Delta P_{dry}/H = 0.1141\, W_{av}^{1.859}$	0.9995	3.32
2	$\Delta P_{dry}/H = 0.063\, W_{av}^{1.876}$	0.9990	4.63
3	$\Delta P_{dry}/H = 0.0701\, W_{av}^{1.786}$	0.9993	2.55
4	$\Delta P_{dry}/H = 0.065\, W_{av}^{1.842}$	0.9994	4.30

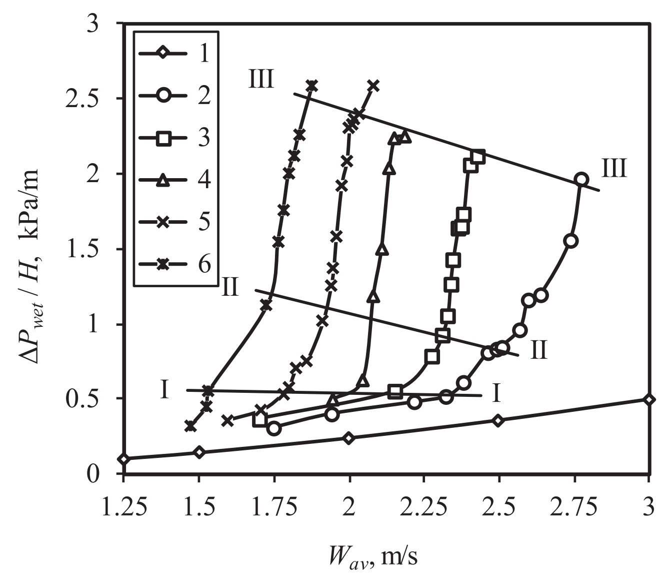

Fig. 6. Hydraulic characteristics of wetting fill pack (type 3) at different wetting rates q_v, $m^3/(m^2{\cdot}h)$: 1 – 0; 2 – 11.8; 3 – 18.0; 4 – 24.3; 5 – 30.6; 6 – 36.8; I-I – loading points; II-II – flooding points; III-III – superflooding points.

slope becomes very steep, causing a significant increase in the pressure drop. Above the line III-III, there is a condition beyond flooding (or superflooding) with droplet entrainment out the fill pack.

Two fill packs consisting of ICCE with 6 mm holes with (type 3) and without the metal grid (type 2) were selected for the following detailed analysis. Fig. 7 shows a comparative analysis of the pressure drop in the wetting fill packs under study.

It can be observed that the metal grid leads to an increase in the average pressure difference from 13 to 34%, depending on the wetting rate. Besides, it is possible to increase the average flow velocity of the gas, and correspondingly the flow capacity of cooling towers from 9.3 to 13.3% (according to the displacement of the flooding points in Fig. 7) by increasing the void fraction of the fill pack.

The equation of the pressure drop in the wetting fill can be given by:

$$\Delta P_{wet}/\Delta P_{dry} = 10^{bq_v}, \qquad (2)$$

where ΔP_{wet} is the pressure drop of the wetting fill pack, Pa; b is the empirical coefficient depending on the fill type; q_v is the volumetric wetting rate, $m^3/(m^2{\cdot}h)$.

The equation of ΔP_{dry} for the ICCE with 6 mm holes (Table 2, type 2) to Eq. (2) gives

$$\Delta P_{wet}/H = 0.063\, W_{av}^{1.876}\cdot 10^{bq_v}, \qquad (3)$$

and with the metal grid (Table 2, type 3)

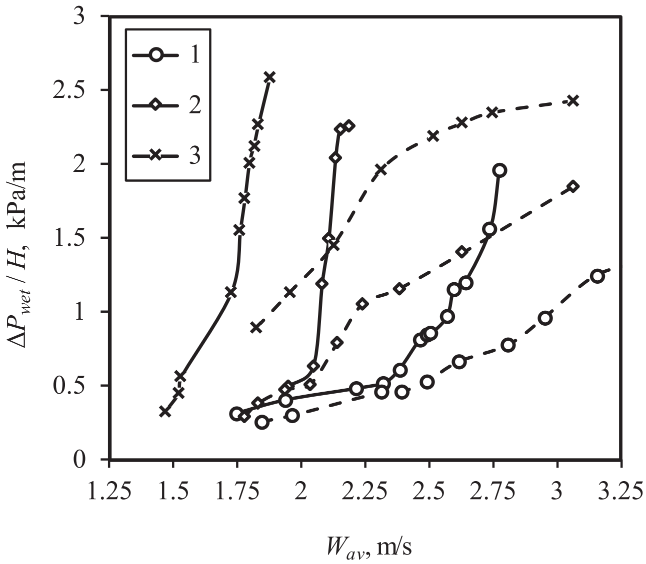

Fig. 7. Pressure drop in wetting fill pack vs. average gas velocity at different wetting rates q_v, $m^3/(m^2 \cdot h)$: 1 – 11.8; 2 – 24.3; 3 – 36.8; solid lines – type 3 (with metal grid); dashed lines – type 2.

$$\Delta P_{wet}/H = 0.0701\, W_{av}^{1.786} \cdot 10^{bq_v}. \tag{4}$$

By solving Eqs. (3) and (4) for b, we defined the empirical coefficient values for the studied fill packs, operating under various wetting conditions. Calculation results of value b with average relative errors are summarized in Table 3.

From Table 3, note that the empirical coefficient b increases with the transition to the more energy-intensive, as a rule, the high-efficiency operating regime of the fill pack. On the other hand, for the ICCE with 6 mm holes and the metal grid (type 3), an increase in the wetting rate of more than 20 $m^3/(m^2 \cdot h)$ reduces the average empirical coefficient b. For instance, b values were decreased by 28.8% and 25.3% in hold-up and loading zones, respectively, and by 9.01% in the flooding zone.

It has also been analyzed the empirical coefficient b of the ICCE with 6 mm holes with (type 3) and without the metal grid (type 2). According to Table 3, the metal grid increases in the average value of the coefficient b. For example, at q_v below 20 $m^3/(m^2 \cdot h)$, b increased by 72% in the hold-up zone, 56.3% in the loading zone, and 92.4% in the flooding zone. As expected, the fill pack with the metal grid (type 3) showed a higher difference in pressure.

In sum, the obtained Eqs. (3) and (4) provide a valuable tool for determining the pressure drop of the developed fill pack under wetting conditions. This tool is useful in engineering estimates of the pressure loss when air pumping through the ICCE with holes of 6 mm diameter with satisfactory accuracy.

Table 3
Comparison of experimentally determined b values with relative errors.

Wetting rate q_v, $m^3/(m^2 \cdot h)$	Hold up zone (below line I-I)		Loading zone (from line I-I to line II-II)		Flooding zone (from line II-II to the line III-III)	
	b	δ_{av}, %	b	δ_{av}, %	b	δ_{av}, %
Type 2						
Under 32	0.0111	11.45	0.0169	10.80	0.0215	9.60
Over 32	0.0186	3.44	0.0207	1.71	0.0206	5.20
Type 3						
Under 20	0.01900	6.30	0.0264	11.20	0.04140	8.20
Over 20	0.01355	8.30	0.0240	7.40	0.03095	9.20

3.2. Analysis of volumetric heat and mass transfer coefficient

Based on the method of transfer units, which makes it possible to estimate the height of the cooling towers, the volumetric mass transfer coefficient β_v [$kg/(m^3 \cdot s)$] can be written as

$$\beta_v = \frac{L_m\, c_{p,L}}{V} \frac{t_{L,1} - t_{L,2}}{\Delta h_{av}} \tag{5}$$

where L_m is the mass flow rate of water, kg/s; $c_{p,L}$ is the specific heat of water at constant pressure, J/(kg·K); V is the fill pack volume, m^3; $t_{L,1}$ is the inlet water temperature, °C; $t_{L,2}$ is the outlet water temperature, °C; Δh_{av} is the average enthalpy change, J/kg.

The volumetric mass transfer coefficient is typically correlated with the following general form:

$$\beta_v = A\, q_m\, (G_m/L_m)^n \tag{6}$$

where A, n are the constants specific to a particular tower fill; q_m is the mass wetting rate, $kg/(m^2 \cdot s)$; G_m is the mass flow rate of air, kg/s.

Many studies on constants A and n are available in the literature for many industrial and experimental cooling tower fill packs.

The average enthalpy change Δh_{av} is defined by the numerical integration as below.

$$\Delta h_{av} = \frac{t_{L,1} - t_{L,2}}{\int_{t_{L,2}}^{t_{L,1}} \frac{dt}{h_s - h}} \tag{7}$$

where h_s is the specific enthalpy of the saturated air, J/kg; h is the specific enthalpy of the air at the cross-sectional area of the fill pack, J/kg.

The enthalpy of the air in saturation h_s at the water temperature t_L is defined in Eq. (8).

$$h_s = 0.24\, t_L + x_s(595 + 0.47\, t_L) \tag{8}$$

where x_s is the moisture content of the saturated air, kg/kg.

The air enthalpy for each cross-sectional area of the fill pack h is calculated by the following expression:

$$h = h_1 + \frac{t_L - t_{L,2}}{K} \frac{L_m}{G_m} \tag{9}$$

where h_1 is the enthalpy of the inlet air in the specific cross-sectional area, J/kg; K is the constant, expressed as Eq. (10)

$$K = 1 - \frac{c_{p,L}\, t_{L,2}}{r} \tag{10}$$

where r is the specific latent heat of evaporation, J/kg.

Combining Eq. (6) and experimental data provides the empirical relationships for the volumetric mass transfer coefficient during cooling water in the fill pack consisted of the ICCE with 6 mm holes (type 2)

$$\beta_v = 2.50\, q_m\, (G_m/L_m)^{0.90} \tag{11}$$

and with the metal grid (type 3)

$$\beta_v = 2.48\, q_m\, (G_m/L_m)^{0.94} \tag{12.1}$$

$$\beta_v = 2.53\, q_m (G_m/L_m)^{0.90} \tag{12.2}$$

With the help of Eqs. (12.1) and (12.2), estimating the volumetric mass transfer coefficient values for the developed fill pack of the ECT at the different wetting rates is possible. On the one hand, Eq. (12.1) makes possible to approximate experimental data at the wetting rates from 3 to 6 $kg/(m^2 \cdot s)$ with an average relative error of 1%, and on the other hand, Eq. (12.2) is recommended at the wetting rates from 6.0 to 10.5 $kg/(m^2 \cdot s)$, while the maximum relative error is 2.4%, and the average relative error is less than 1.1%.

As shown in Fig. 8, the results indicate that the volumetric mass transfer coefficient increases directly to the average gas velocity. The greater air rate suggests an increase both in the turbulence intensity and

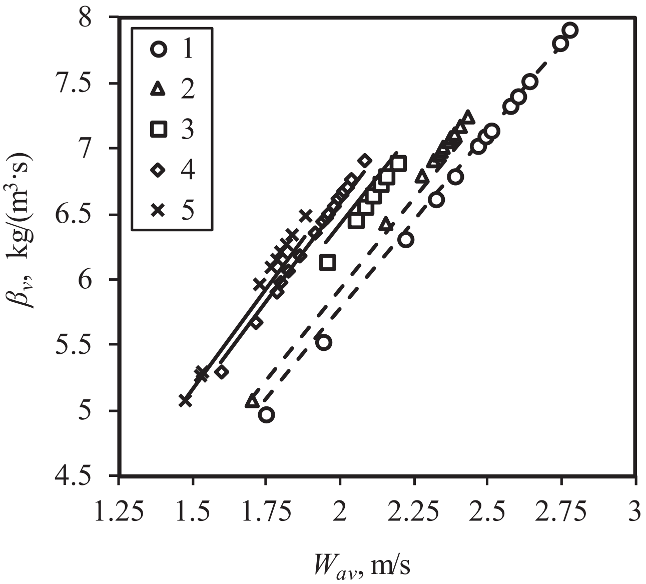

Fig. 8. Volumetric mass transfer coefficient vs. average gas velocity for fill pack (type 3) at various wetting rates q_m, kg/(m^2·s): 1 – 3.25; 2 – 4.97; 3 – 6.70; 4 – 8.42; 5 – 10.15; dots – experimental points, dashed lines – according to Eq. (12.1), solid lines – according to Eq. (12.2).

interfacial shear stresses. Moreover, a rise in the wetting rate from 3.25 to 10.15 kg/(m^2·s) increases the volumetric mass transfer coefficient by 14.9% on average.

The analysis of efficiency in using the metal grid shows a small dip in the volumetric mass transfer coefficients (Fig. 9). Part of the heat flux is likely spent on heat transfer and post-heating of the metal grid. In this case, the decrease in the volumetric mass transfer coefficients lies within measuring devices' relative error and is only 3.2%.

Fig. 10 represents a comparative analysis of the volumetric mass transfer coefficients depending on the mass flow rate ratio of air to water for the studied fill packs and other types of fills. Calculations based on the following operating conditions of the ECT: the wetting rate is 100 m^2, the mass flow rate of water is 896 t/h, the mass flow rate of the air varies from 239.2 to 2240 t/h.

We revealed that the maximum value of the volumetric mass transfer

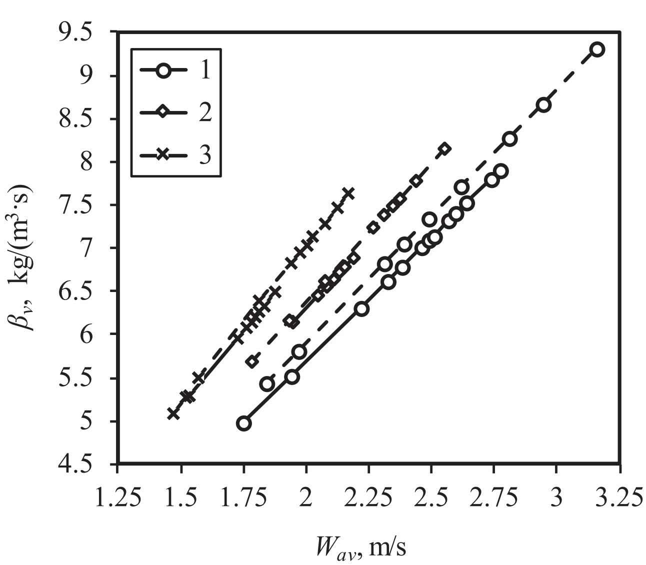

Fig. 9. Volumetric mass transfer coefficient vs. average gas velocity at various wetting rates q_m, kg/(m^2·s): 1 – 3.25; 2 – 6.70; 3 – 10.15; solid lines – type 3 (with metal grid); dashed lines – type 2.

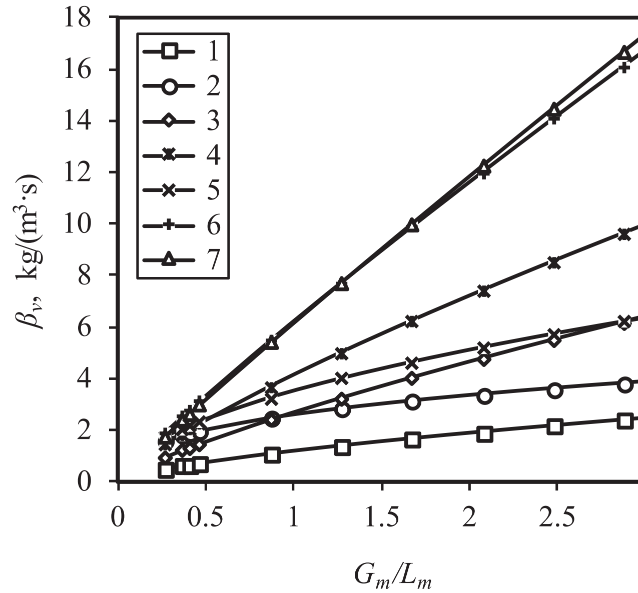

Fig. 10. Volumetric mass transfer coefficient against the air to water mass flow rate ratio for different fill packs of the cooling tower: 1 – film asbestos-cement fill (A = 0.479; n = 0.66); 2 – PR50 prism fill [38] (A = 1.05; n = 0.36); 3 – screen fill [39] (according to the equation $\beta_v = 1.04\,q_m^{1.04}(G_m/L_m)^{0.79}$); 4 – jet-film fill [40] (A = 1.66; n = 0.8); 5 – Balcke-Duerr lattice fill [41] (A = 1.41; n = 0.54); 6 – ICCE with 6 mm holes (type 2) (A = 2.5; n = 0.90); 7 – ICCE with 6 mm holes and metal grid (type 3) (A = 2.48; n = 0.94).

coefficient occurs in the proposed filler packs as the coolant air flow rate increases. For example, at $G_m/L_m < 0.5$ the volumetric mass transfer coefficient of the fill pack (type 3) is on average 38% higher than β_v of the jet-film fill [40] and at $G_m/L_m > 0.5$ a change reaches 61.2%. It indicates the significant performance potential of the developed fill pack design for efficient cooling the circulating water.

The volumetric heat transfer coefficient α_v [W/(m^3·K)] is usually estimated from the revised Lewis relation:

$$\alpha_v/\beta_v = c_{p,G}(Sc_G/Pr_G)^{0.5}, \tag{13}$$

where $c_{p,G}$ is the air specific heat at constant pressure, J/(kg·K); Sc_G is the Schmidt number, $Sc_G = \mu_G\rho_G^{-1}D_G^{-1}$; μ_G is the dynamic viscosity of the air, Pa·s; ρ_G is the density of the air, kg/m^3; D_G is the diffusion coefficient, m^2/s; Pr_G is the Prandtl number, $Pr_G = c_{p,G}\mu_G\lambda^{-1}$; λ is the thermal conductivity of the air, W/(m·K).

Here $(Sc_G/Pr_G)^{0.5} \approx 1$ for the air, so the volumetric heat transfer coefficient could be written as follows

$$\alpha_v = c_{p,G}\,\beta_v. \tag{14}$$

Thus, the volumetric heat and mass transfer coefficients are directly proportional. Fig. 11 reveals that at the average air velocity of 3.15 m/s and the wetting rate of 3.25 kg/(m^2·s), the volumetric heat transfer coefficient for the developed fill pack (type 2) reaches 9687.1 W/(m^3·K).

Finally, as one of the evaluation indicators of the ECTs, the Merkel number Me reflects the cooling efficiency of the used fill pack, and it is expressed as:

$$Me = \frac{\beta_v\,H}{q_m} = \frac{\Delta t_L\,c_{p,L}}{K\,\Delta h_{av}} \approx A\,H\left(\frac{G_m}{L_m}\right)^n \tag{15}$$

where Δt_L is the water temperature difference, °C.

It follows from Eq. (15) that the Merkel number is directly proportional to the volumetric mass transfer coefficient. Therefore, the results of the cooling efficiency of the ICCE with 6 mm holes with (type 3) and without the metal grid (type 2) are similar to the obtained dependencies from Fig. 9. The maximum cooling efficiency of the studied fill packs is

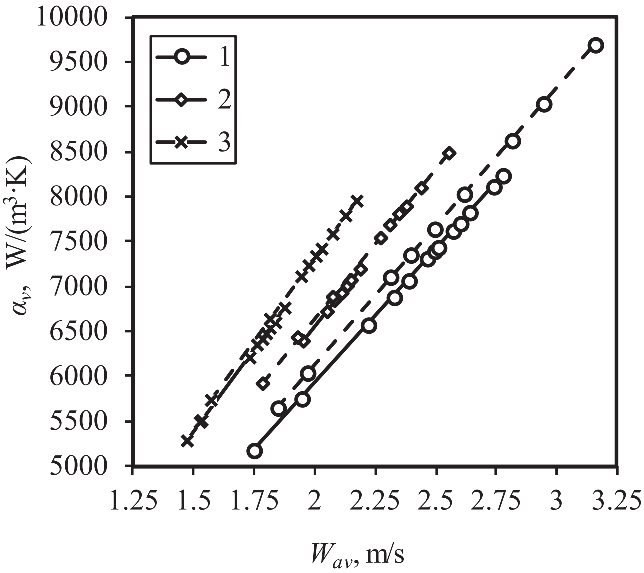

Fig. 11. Volumetric heat transfer coefficient vs. average gas velocity at various wetting rates q_m, kg/(m^2·s): 1 – 3.25; 2 – 6.70; 3 – 10.15; solid lines – type 3 (with metal grid); dashed lines – type 2.

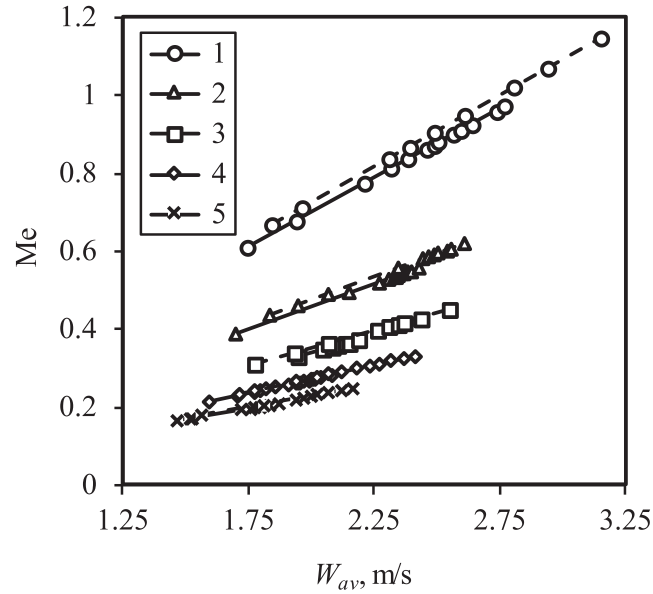

Fig. 12. Variations of Merkel number with average gas velocity at various wetting rates q_m, kg/(m^2·s): 1 – 3.25; 2 – 4.97; 3 – 6.70; 4 – 8.42; 5 – 10.15; solid lines – type 3 (with metal grid); dashed lines – type 2.

observed at the low wetting rates, as shown in Fig. 12.

For instance, when the wetting rate decreases from 4.97 kg/(m^2·s) to 3.25 kg/(m^2·s), the Merkel number changes from 52.6 to 56.4%. The reason is that at the low liquid flow rates, the actual contact area between the gas and liquid phases increases considerably with the activation of dead zones. Besides, the proposed design of the fill pack with the ICCE ensures the uniform distribution of liquid over the cross-sectional area of the cooling tower.

4. Conclusions

This experimental study has enabled the investigation of the effect of the fill pack design on the evaporative cooling tower's thermal–hydraulic performance. The following conclusions can be drawn from the present research:

- the empirical relationships of the pressure drop through the non-wetting developed fill pack of various types from the average air velocity were determined;
- the defined empirical relationships of the pressure drop through the wetting fill pack consisted of ICCE with 6 mm holes with and without the metal grid operating under various hydraulic regimes were analyzed;
- based on experiments and the method of transfer units, the empirical relationships of the volumetric mass transfer coefficient for the fill pack were determined. It was shown that the use of the metal grid decreases the volumetric mass transfer coefficient by 3.2% compared to the same fill pack without the metal grid.
- the analysis of the volumetric mass transfer coefficients depending on the mass flow rate ratios established that the developed fill packs had the maximum cooling efficiency in comparison to other types of fills with the increase in the air flow rate;
- based on the Lewis relation, the dependencies of the volumetric heat transfer coefficient on the average air velocity were constructed for the fill packs at the various wetting rates;
- the maximum cooling efficiency of the studied fill packs was observed at the low wetting rates according to the Merkel method.

The best heat and mass transfer characteristics at the lowest pressure drop were achieved in the developed fill pack consisting of ICCE with 6 mm holes and without the metal grid (type 2). The obtained empirical relationships are recommended as a guide for reaching optimum operating conditions for the fill pack in the evaporative cooling towers.

CRediT authorship contribution statement

A.V. Dmitriev: Conceptualization, Resources, Investigation, Supervision. **I.N. Madyshev:** Conceptualization, Methodology, Data curation, Investigation. **V.V. Kharkov:** Validation, Writing - original draft, Writing - review & editing. **O.S. Dmitrieva:** Software, Project administration. **V.E. Zinurov:** Formal analysis, Visualization, Data curation.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgement

This paper was supported by the grant of the President of the Russian Federation, project number MK–616.2020.8.

References

[1] J.C. Kloppers, D.G. Kröger, A critical investigation into the heat and mass transfer analysis of counterflow wet-cooling towers, Int. J. Heat Mass Transf. 48 (3-4) (2005) 765–777, https://doi.org/10.1016/j.ijheatmasstransfer.2004.09.004.
[2] M. Lucas, J. Ruiz, P.J. Martínez, A.S. Kaiser, A. Viedma, B. Zamora, Experimental study on the performance of a mechanical cooling tower fitted with different types of water distribution systems and drift eliminators, Appl. Therm. Eng. 50 (1) (2013) 282–292, https://doi.org/10.1016/j.applthermaleng.2012.06.030.
[3] B.K. Naik, P. Muthukumar, A novel approach for performance assessment of mechanical draft wet cooling towers, Appl. Therm. Eng. 121 (2017) 14–26, https://doi.org/10.1016/j.applthermaleng.2017.04.042.
[4] S. Saha A. Tikadar J. Khan T. Farouk Numerical Analysis on Evaporation Assisted Convective Cooling: Effect of Surface Morphology, in ASME Int. Mech. Eng. Congr. Expo. 2019 p. V008T09A051.
[5] H. El-Dessouky, Thermal and hydraulic performance of a three-phase fluidized bed cooling tower, Exp. Therm Fluid Sci. 6 (4) (1993) 417–426, https://doi.org/10.1016/0894-1777(93)90018-E.
[6] M. Llano-Restrepo, R. Monsalve-Reyes, Modeling and simulation of counterflow wet-cooling towers and the accurate calculation and correlation of mass transfer

coefficients for thermal performance prediction, Int. J. Refrig. 74 (2017) 47–72, https://doi.org/10.1016/j.ijrefrig.2016.10.018.
[7] S.P. Fisenko, A.A. Brin, A.I. Petruchik, Evaporative cooling of water in a mechanical draft cooling tower, Int. J. Heat Mass Transf. 47 (1) (2004) 165–177, https://doi.org/10.1016/S0017-9310(03)00409-5.
[8] C. Ren An Analytical Approach to the Heat and Mass Transfer Processes in Counterflow Cooling Towers 128 11 2006 1142 1148 10.1115/1.2352780.
[9] O.S. Dmitrieva, A.V. Dmitriev, I.N. Madyshev, A.N. Nikolaev, Flow dynamics of mass exchangers with jet-bubbling contact devices, Chem. Petrol. Eng. 53 (1-2) (2017) 130–134, https://doi.org/10.1007/s10556-017-0308-8.
[10] K. Singh, R. Das, An improved constrained inverse optimization method for mechanical draft cooling towers, Appl. Therm. Eng. 114 (2017) 573–582, https://doi.org/10.1016/j.applthermaleng.2016.12.002.
[11] L.D. Blackburn, J.F. Tuttle, K.M. Powell, Real-time optimization of multi-cell industrial evaporative cooling towers using machine learning and particle swarm optimization, J. Cleaner Prod. 271 (2020) 122175, https://doi.org/10.1016/j.jclepro.2020.122175.
[12] M.S. Söylemez, On the optimum sizing of cooling towers, Energy Convers. Manage. 42 (7) (2001) 783–789, https://doi.org/10.1016/S0196-8904(00)00148-5.
[13] A.B. Golovanchikov, V.A. Balashov, N.A. Merentsov, The filtration equation for packing material, Chem. Petrol. Eng. 53 (1-2) (2017) 10–13, https://doi.org/10.1007/s10556-017-0285-y.
[14] A.V. Dmitriev, O.S. Dmitrieva, I.N. Madyshev, Optimal designing of mass transfer apparatuses with jet-film contact devices, Chem. Petrol. Eng. 53 (7-8) (2017) 430–434, https://doi.org/10.1007/s10556-017-0358-y.
[15] N. Williamson, M. Behnia, S. Armfield, Comparison of a 2D axisymmetric CFD model of a natural draft wet cooling tower and a 1D model, Int. J. Heat Mass Transf. 51 (9-10) (2008) 2227–2236, https://doi.org/10.1016/j.ijheatmasstransfer.2007.11.008.
[16] H.R. Goshayshi, J.F. Missenden, The investigation of cooling tower packing in various arrangements, Appl. Therm. Eng. 20 (1) (2000) 69–80, https://doi.org/10.1016/S1359-4311(99)00011-3.
[17] F. Gharagheizi, R. Hayati, S. Fatemi, Experimental study on the performance of mechanical cooling tower with two types of film packing, Energy Convers. Manage. 48 (1) (2007) 277–280, https://doi.org/10.1016/j.enconman.2006.04.002.
[18] M. Lemouari, M. Boumaza, A. Kaabi, Experimental analysis of heat and mass transfer phenomena in a direct contact evaporative cooling tower, Energy Convers. Manage. 50 (6) (2009) 1610–1617, https://doi.org/10.1016/j.enconman.2009.02.002.
[19] M. Lemouari, M. Boumaza, Experimental investigation of the performance characteristics of a counterflow wet cooling tower, Int. J. Therm. Sci. 49 (10) (2010) 2049–2056, https://doi.org/10.1016/j.ijthermalsci.2010.05.012.
[20] R.K. Singla, K. Singh, R. Das, Tower characteristics correlation and parameter retrieval in wet-cooling tower with expanded wire mesh packing, Appl. Therm. Eng. 96 (2016) 240–249, https://doi.org/10.1016/j.applthermaleng.2015.11.063.
[21] M. Rahmati, S.R. Alavi, M.R. Tavakoli, Experimental investigation on performance enhancement of forced draft wet cooling towers with special emphasis on the role of stage numbers, Energy Convers. Manage. 126 (2016) 971–981, https://doi.org/10.1016/j.enconman.2016.08.059.
[22] M. Rahmati, S.R. Alavi, M.R. Tavakoli, Investigation of heat transfer in mechanical draft wet cooling towers using infrared thermal images: An experimental study, Int. J. Refrig 88 (2018) 229–238, https://doi.org/10.1016/j.ijrefrig.2017.11.031.
[23] P. Shahali, M. Rahmati, S.R. Alavi, A. Sedaghat, Experimental study on improving operating conditions of wet cooling towers using various rib numbers of packing, Int. J. Refrig 65 (2016) 80–91, https://doi.org/10.1016/j.ijrefrig.2015.12.004.
[24] A. Mirabdolah Lavasani, Z. Namdar Baboli, M. Zamanizadeh, M. Zareh, Experimental study on the thermal performance of mechanical cooling tower with rotational splash type packing, Energy Convers. Manage. 87 (2014) 530–538, https://doi.org/10.1016/j.enconman.2014.07.036.
[25] M. Amini, M. Zareh, S. Maleki, Thermal performance analysis of mechanical draft cooling tower filled with rotational splash type packing by using nanofluids, Appl. Therm. Eng. 175 (2020) 115268, https://doi.org/10.1016/j.applthermaleng.2020.115268.
[26] M. Gao, L. Zhang, N.-n. Wang, Y.-T. Shi, F.-Z. Sun, Influence of non-uniform layout fillings on thermal performance for wet cooling tower, Appl. Therm. Eng. 93 (2016) 549–555, https://doi.org/10.1016/j.applthermaleng.2015.09.054.
[27] N. Milosavljevic, P. Heikkilä, A comprehensive approach to cooling tower design, Appl. Therm. Eng. 21 (9) (2001) 899–915, https://doi.org/10.1016/S1359-4311(00)00078-8.
[28] A.I. Petruchik, A.D. Solodukhin, S.P. Fisenko, Evaporative cooling of water in complex-configuration film spray zones, J. Eng. Phys. Thermophy. 81 (1) (2008) 182–187, https://doi.org/10.1007/s10891-008-0017-4.
[29] Z.Z. Xia, C.J. Chen, R.Z. Wang, Numerical simulation of a closed wet cooling tower with novel design, Int. J. Heat Mass Transf. 54 (11-12) (2011) 2367–2374, https://doi.org/10.1016/j.ijheatmasstransfer.2011.02.025.
[30] X. Xie, C. He, T. Xu, B. Zhang, M. Pan, Q. Chen, Deciphering the thermal and hydraulic performances of closed wet cooling towers with plain, oval and longitudinal fin tubes, Appl. Therm. Eng. 120 (2017) 203–218, https://doi.org/10.1016/j.applthermaleng.2017.03.138.
[31] M. Prasad, Economic upgradation and optimal use of multi-cell crossflow evaporative water cooling tower through modular performance appraisal, Appl. Therm. Eng. 24 (4) (2004) 579–593, https://doi.org/10.1016/j.applthermaleng.2003.10.012.
[32] B.A. Qureshi, S.M. Zubair, A complete model of wet cooling towers with fouling in fills, Appl. Therm. Eng. 26 (16) (2006) 1982–1989, https://doi.org/10.1016/j.applthermaleng.2006.01.010.
[33] D. Lyu, F. Sun, Y. Zhao, Impact mechanism of different fill layout patterns on the cooling performance of the wet cooling tower with water collecting devices, Appl. Therm. Eng. 110 (2017) 1389–1400, https://doi.org/10.1016/j.applthermaleng.2016.08.190.
[34] G.V.S. Sesha Girish, A. Mani, Numerical simulation of forced convective evaporation system for tannery effluent, Int. J. Heat Mass Transf. 47 (6-7) (2004) 1335–1346, https://doi.org/10.1016/j.ijheatmasstransfer.2003.10.001.
[35] A.B. Golovanchikov, N.A. Merentsov, V.A. Balashov, Modeling and analysis of a mechanical-draft cooling tower with wire packing and drip irrigation, Chem. Petrol. Eng. 48 (9-10) (2013) 595–601, https://doi.org/10.1007/s10556-013-9663-2.
[36] A.V. Dmitriev, O.S. Dmitrieva, I.N. Madyshev, A.I. Khafizova, A.A. Sagdeev, A.N. Nikolaev, G.S. Sagdeeva, Mechanical draft cooling tower with self-distribution of liquid, 193253, 2019.
[37] L.N. Madyshev, A.I. Khafizova, O.S. Dmitrieva, S. Bratan, The study of gas-liquid flow dynamics in the inclined-corrugated elements of cooling tower filler unit, E3S Web Conf. 126 (2019) 00031, https://doi.org/10.1051/e3sconf/201912600031.
[38] Y.I. Ponomarenko, V.S. Arefiev, Cooling towers of industrial and power enterprises, Energoatomizdat, Moscow, 1998.
[39] A.G. Laptev, V.A. Danilov, I.V. Vishnyakova, Evaluating the effectiveness of circulating water cooling in a cooling tower, Therm. Eng. 51 (2004) 661–665.
[40] O.S. Dmitrieva, I.N. Madyshev, A.V. Dmitriev, Determination of the heat and mass transfer efficiency at the contact stage of a jet-film facility, J. Eng. Phys. Thermophy. 90 (3) (2017) 651–656, https://doi.org/10.1007/s10891-017-1612-z.
[41] A.G. Laptev, I.A. Vedgaeva, Design and calculation of industrial cooling towers, KGEU, Kazan, 2004.