Articles

School of Civil and Structural Engineering (Former DIT)

2009-01-01

Heat Transfer Correlations for Low Approach Evaporative Cooling Systems in Buildings

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Heat transfer correlations for low approach evaporative cooling systems in buildings

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Abstract

The experimental performance of an open industrial scale cooling tower, utilising small approach temperature differences (1–3 K), for rejection of heat at the low water temperatures (11–20 °C) typical of chilled ceilings and other sensible air–water heat dissipation systems in buildings, is examined. The study was carried out under temperate maritime climatic conditions (3–18 °C wet-bulb temperature range). Initially a theoretical analysis of the process at typical conditions for this climate was conducted, which indicated that a water to air (L/G) mass flow rate ratio of less than 1.0 was required for effective operation. Consequently for these low L/G ratios, the thermal performance of the experimental tower was measured and correlated. A new correlation is proposed which shows a significant increase in the NTU level achieved, for the required L/G ratios (0.3–0.9). As the cooling tower in this application is predominantly a mass transfer device under summer conditions, the evaluation of the total volumetric heat and mass transfer coefficient (kg_a s^{-1} m^{-3}) is of particular relevance and is also determined.

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Keywords: Free cooling; Low energy cooling; Evaporative cooling; Cooling towers; Chilled ceilings; Thermal performance; Heat transfer

1. Introduction

In recent years, evaporative cooling has been acknowledged as a potential low energy cooling alternative to traditional vapour compression based air conditioning systems [1]. Although there is considerable evidence that evaporative cooling can in many climatic regions reduce energy consumption in commercial buildings, the technique has not been widely exploited to date [2]. This is especially true for maritime climates where evaporative cooling is often viewed as unsuitable. The temperate maritime climate, such as experienced in north western Europe, is regarded as cool and damp, the opposite to warm dry conditions which are generally perceived as the ideal climate for evaporative cooling. As a result, for many years, interest in evaporative cooling has been primarily confined to latitudes below 45°N, where it was seen as being most beneficially applied [3,4]. In so far as evaporative cooling has been considered in maritime locations, it has generally been used as a water-side indirect system. With this approach, cooling water is produced centrally in a cooling tower and distributed to local air–water terminals such as chilled ceilings, fan-coils or induction units at temperatures of 11–18 °C.

Two distinct approaches have been developed to water-side indirect evaporative cooling – closed (indirect) wet cooling towers in which the heat exchanger is located within the tower and open (direct) towers with external plate heat exchangers. Each arrangement has features, which suit particular circumstances and locations. The closed tower exhibits the following features; (i) the primary water circuit is simplified, being confined to the cooling tower, (ii) the heat exchange and evaporative cooling process are both integrated in the cooling tower tube bundle heat exchanger which takes the place of the cooling tower

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Nomenclature

Symbols

a	heat transfer area per unit volume ($m^2 \, m^{-3}$)
C_{pw}	specific heat of water at constant pressure ($kJ \, kg^{-1} \, K^{-1}$)
C_T	cooling tower constant, dimensionless
Δh	elemental rise in the air enthalpy across a cooling tower element ($kJ \, kg^{-1}$)
Δt	drop in cooling tower water temperature across a cooling tower element (K)
G	air flow rate in the tower ($kg \, s^{-1}$)
G'	air flow rate flux in the tower ($kg \, s^{-1} \, m^{-2}$)
h_a	enthalpy of the bulk air stream in the cooling tower ($kJ \, kg^{-1}$)
h_{asw}	enthalpy of air in the cooling tower saturated at the bulk water temperature ($kJ \, kg^{-1}$)
K	total heat (i.e. simultaneous heat and mass transfer) transfer coefficient ($kg_a \, s^{-1} \, m^{-2}$)
Ka	product of total heat transfer coefficient and heat transfer area ($kg_a \, s^{-1} \, m^{-3}$)
L	primary water flow rate in tower ($kg \, s^{-1}$)
L'	primary water flow rate flux in tower ($kg \, s^{-1} \, m^{-2}$)
n	exponent for G'
t	water temperature in the cooling tower (°C)
T_{as}	ambient adiabatic saturation temperature (AST) (°C)
T_{pa}	primary approach temperature (PAT) (K)
T_{pr}	primary loop return temperature (°C)
T_{ps}	primary loop supply temperature (°C)

T_{sa}	secondary approach temperature (SAT) (K)
T_{sr}	secondary loop return temperature (°C)
T_{ss}	secondary loop supply temperature (°C)
V	heat transfer volume in tower (m^3)
x	exponent for cooling tower L/G ratio in context of experimental correlations

Subscripts

a	with reference to the general air stream
as	adiabatic saturation
asw	air saturated at water temperature
i	with reference to an individual heat transfer element
N	final cooling tower heat transfer element
pa	primary approach
pr	primary return
ps	primary supply
sa	secondary approach
sr	secondary return
ss	secondary supply
T	with reference to the cooling tower

Abbreviations

AST	adiabatic saturation temperature
DBT	dry bulb temperature
NTU	number of transfer units
PAT	primary approach temperature
SAT	secondary approach temperature
WBT	psychrometric wet-bulb temperature

packing, (iii) the tube bundle in effect forms the final element of the secondary circuit, and (iv) the arrangement is suitable for locations with warmer year round ambient temperatures and more uniform load conditions. The open tower with external heat exchanger has the following distinct features: (i) the evaporative cooling function in the cooling tower and the heat transfer function in the heat exchanger are separated, (ii) an external heat exchanger is used which can be located indoors, hence for locations in which ambient frost conditions are common, the secondary circuit is protected from frost damage, (iii) for large non-uniform load conditions, multiple plate heat exchangers (connected to a single tower) and located within the building can provide secondary circuit flexibility, offering different secondary supply water temperatures or different secondary pressures; in multi-storey buildings a heat exchanger can be located at each floor, giving floor by floor independence; individual secondary circuits can be shut down when not in use, (iv) various forms and specifications of advanced heat exchanger technology and open cooling tower technology can be deployed in the design of the heat rejection system, and (v) it is possible to use the primary

circuit water at a lower approach directly in primary air plant cooling coils (such as for displacement ventilation) while simultaneously using the secondary circuit water in a more extensive secondary cooling system, thus improving the availability of cooling water generated by evaporation for the primary circuit.

The thermal performance of closed towers has been the subject of a number of recent research projects undertaken by European Commission supported consortia, from which there are numerous associated publications [5–11]. Facao and Oliveira describe the development of a prototype closed cooling tower, specially adapted for use with chilled ceilings [5]. The tower has a design cooling capacity of 10 kW, for an inlet water temperature of 21 °C and an air wet-bulb temperature of 16 °C. Its cross-section was 0.6×1.2 m with a height of 1.55 m. The tube bundle utilised in the tower had 228 staggered tubes each of 10 mm outside diameter, with a total transfer area of 8.6 m^2. In a subsequent paper, the same authors describe the correlations for heat and mass transfer coefficients obtained for this system [6]. The proposed correlations were compared with existing data associated with higher temperature rejec-

tion towers. Hasan and Siren describe the theoretical analysis and computational modelling of a similar closed cooling tower [7]. Experimental measurements of the performance of the prototype tower were used to define the tower transfer coefficients. This allowed tower flow rates and tube configurations to be optimised on the basis of the cooling load thereby maximising the coefficient of performance. In further studies, the same authors examined the performance of two evaporatively cooled heat exchanger designs for a specially constructed small scale closed wet tower [8]. The heat exchangers utilised were plain and plate-finned circular tube types which occupied the same volume. An increase in heat transfer was noted for the plate-finned tubes. In addition, the wet-finned surfaces showed lower fin efficiency compared with dry surfaces. In another paper, Hasan and Siren compared the performance of a closed tower which incorporated both circular tubes and oval tubes in its heat exchanger design [9]. They concluded from this study that the oval tube had a better combined thermal–hydraulic performance compared with the circular tube case. Gan and Riffat describe a numerical technique for evaluating the performance of a closed wet cooling tower for chilled ceiling systems [10]. The technique is based on the application of computational flow dynamics (CFD) for the two-phase flow of gas and water droplets. An Eulerian approach is used to model the gas phase flow and a Lagrangian approach is used for the water droplet phase flow, with two-way coupling between two phases. In another paper, the same authors describe further results using a CFD cool-

ing tower model [11]. The predicted thermal performance is compared with experimental measurement for a large industrial closed tower and a prototype cooling tower. The accuracy of the CFD modelling of the pressure loss for fluid flow over the heat exchanger is assessed for a range of flow velocities. The predicted pressure loss for single-phase flow of air over the heat exchanger was found to be in good agreement with the empirical equation for tube bundles.

The current work can be distinguished from the previously reported work [5–11] as it is based on an approach that exploits the open tower concept as its fundamental design principle. This was motivated by a number of potential issues associated with the deployment of an open tower in a maritime climate setting. They included (i) the feasibility of using a relatively high density packing ($200 \, m^2 \, m^{-3}$) in conjunction with a high packing volume, thereby enabling small approach temperatures to be achieved between the primary supply temperature generated and the ambient wet-bulb temperature and (ii) the ability to use a highly effective plate heat exchanger between the primary and secondary circuits thereby ensuring that small heat exchanger approach temperatures are obtained. The approach deployed in this work exploits an open counter-flow cooling tower and a counter-flow plate heat exchanger, both with enhanced heat transfer areas for the purpose of minimizing the approach conditions (see Fig. 1). The cooling tower has a high degree of inbuilt operating flexibility, giving a possible L/G mass flow rate ratio range of 0.25–3.0. Further details on this research are described else-
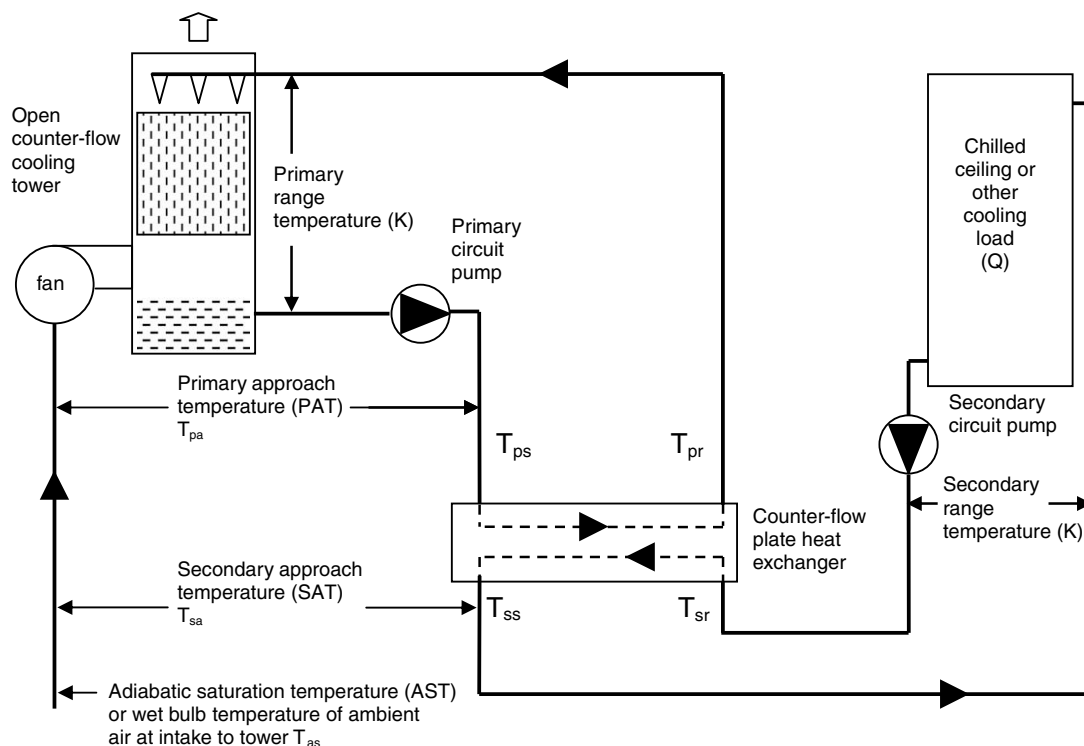

Fig. 1. Simplified schematic of a water-side indirect evaporative cooling system.

where [12–14]. To date however, there has been little work carried out on the deployment of open tower designs, for rejection of heat at low temperatures (11–20 °C) typical of chilled ceiling and other air–water systems, under maritime climatic conditions (3–18 °C AST). Naphon describes a study which reports both experimental and theoretical results of the heat transfer characteristics of an open bench-top cooling tower [15]. However, its operating bounds are significantly different from the current work, with typical air and water mass flow rates ranging between 0.01 and 0.07 kg s^{-1}, and between 0.04 and 0.08 kg s^{-1}, respectively. Tests were carried out for an air inlet temperature of 23 °C, and water inlet temperatures between 30 and 40 °C. A mathematical model was developed and solved by an iterative method to determine the heat transfer characteristics of the tower. There is reasonable agreement between the measured data and predicted results.

In the current application, the operating conditions are well outside those encountered in more conventional applications such as refrigeration condenser heat rejection. This is because the difference between the cooling tower water temperature and the cooling tower air temperature is small and consequentially there are much reduced levels of enthalpy difference, the key driving force in the tower. However, this can be compensated for by increasing the volume of packing within the tower. The corollary of this is that the associated volumetric (per unit volume of tower packing) heat and mass transfer rates in summer are smaller than with more conventional applications. The objectives of the current paper are (i) to analyse the NTU level required with low approach cooling towers operating at conditions encountered in dry air–water systems in temperate climates, (ii) to measure the thermal performance of the low approach cooling tower at a series of low (<1.0) water to air flow rate (L/G) ratios, (iii) to develop correlations for the tower thermal performance at these exceptional conditions, and (iv) to compare these results with those obtained with higher approach tower designs in more conventional heat rejection applications. Several issues distinguish this research from other work cited in the previous paragraphs. First, the current paper deals with an open tower design. Second, there is a lack of experimental work on open tower systems, particularly on industrial or semi-industrial scale test rigs which have been designed to achieve low approach conditions. Finally, the question arises as to the applicability of correlations available for other open tower configurations, which are subject to significantly different boundary conditions and to different tower design parameters from the current work.

2. Theoretical analysis applied to low approach evaporative cooling

The key measure of open cooling tower performance is the cooling tower coefficient (KaV/L) or number of transfer units (NTU) achieved. Ka (kg$_a$ s^{-1} m^{-3}) is the product of the total heat transfer coefficient K (kg$_a$ s^{-1} m^{-2}) and the

heat transfer area per unit volume a (m^2 m^{-3}). The total heat transfer refers to simultaneous sensible and latent transfer. This product is used because of the difficulty of separating the heat transfer coefficient from the heat transfer area. The total heat transfer area is not necessarily equal to the packing surface area as total heat may also be transferred from water droplets in suspension above, within or below the packing. The performance of open cooling towers, determined by experimental research is typically correlated in terms of the NTU level achieved as a function of the water to air flow rate ratio (the L/G ratio) as follows:

$$\frac{KaV}{L} = C_T \left(\frac{L}{G}\right)^{-x} \tag{1}$$

In this equation C_T (the cooling tower constant) and the exponent x are constant for a specific correlation and are determined from the experimental results. ASHRAE [16] gives the following general correlation for cooling towers:

$$\frac{KaV}{L} \propto \left(\frac{L}{G}\right)^{-0.6} \tag{2}$$

Bernier [17] measured the thermal performance of an experimental, semi-industrial scale cooling tower in a laboratory along the lines described in ASHRAE [16]. The resulting correlation was

$$\frac{KaV}{L} = 1.42 \left(\frac{L}{G}\right)^{-0.43} \tag{3}$$

Kuehn et al. [18] gives a general correlation, based on the model studies of Braun et al. [19] and when $L < G$ as follows:

$$\frac{KaV}{L} = 1.3 \left(\frac{L}{G}\right)^{-0.6} \tag{4}$$

Historically the work of Lowe and Christe [20] is considered seminal. The results of this work demonstrate that enhancing the fill arrangement by the use of corrugated sheets of various forms significantly improves the value of the tower constant over that obtained with flat sheets but has no significant impact on the value of the exponent. Hence the rationale underlying the use of corrugated fill in most modern packings.

The overall total heat transfer coefficient (simultaneous sensible heat and mass transfer), Ka (kg$_a$ s^{-1} m^{-3}) can be determined from Eq. (1) as NTU(L/V). The area "a" is included with K because of the difficulty of determining the area involved in isolation from the coefficient. In this application the dominant mode of heat transfer in the tower is latent as the ambient dry bulb temperature of the air in summer (16–26 °C) in temperate climates is close to the water temperature (11–20 °C) and hence there is often little net heat transfer by sensible means. While the water temperature required in this application with chilled ceilings is constant through out the year, the ambient dry bulb temperature falls in winter and hence the portion of

the total heat transferred which is sensible, increases. In summer, however, the process of the air through the tower is effectively isothermal in temperate climates when cooling water is being generated for chilled ceilings (the air dry bulb temperature variation in the tower does generally not exceed 3 K). This process can be contrasted with the traditional applications for cooling towers in building cooling systems, such as in water-cooled condenser heat rejection in which typically, sensible and latent heat rejection occurs in near equal measure, the air process following an approximate 45° diagonal on the psychrometric diagram. Hence in this particular application the correlation for Ka, the overall heat transfer coefficient, is particularly interesting and is a key parameter in the design of this form of heat dissipation in buildings.

The idea of seeing the heat and mass transfer in a cooling tower in terms of enthalpy potential is attributed to Merkel [21], who proposed that the total heat transfer taking place at any point in a cooling tower is proportional to the difference between the total heat of the air saturated at the water temperature at that point and the total heat of the unsaturated bulk air at that point. The total heat transfer coefficient K ($kg_a\ s^{-1}\ m^{-2}$) relates to the enthalpy difference between the saturated and unsaturated air streams and is not to be confused with the mass transfer coefficient which has the same units and is frequently denoted by the same symbol in the literature. The determination of the NTU level is commonly undertaken by applying Merkel's equation in the following form:

$$\frac{KaV}{L} = C_{pw} \sum_{i=1}^{i=N} \frac{(\Delta t)_i}{(h_{asw} - h_a)} \qquad (5)$$

The technique is based on dividing the counter-flow tower into a series of horizontal heat transfer elements, each with an equal incremental drop in the water temperature (typically 0.1 K). Numerical integration is used in the form of a summation of the discrete values prevailing in "N" sections of the tower (beginning at the tower known outlet water temperature and known inlet air adiabatic saturation temperature (AST) or wet-bulb temperature and working upwards). The enthalpy of the saturated air film (h_{asw}) at the mean water temperature (t) of the element is determined from the following equation attributed to Stoecker and Jones [22]

$$h_{asw} = 4.7926 + 2.568t - 0.029834t^2 + 0.0016657t^3 \qquad (6)$$

In each element, the heat lost by the water is equal to the heat gained by the air, hence:

$$G\Delta h = LC_{pw}\Delta t \qquad (7)$$

therefore, for each element,

$$\Delta h = (L/G)C_{pw}\Delta t \quad \text{or} \quad \Delta h = (L/G)0.418 \qquad (8)$$

Hence as the inlet air enthalpy is known, the enthalpy of the bulk air in the tower can be determined at each element by incrementing the rise in enthalpy across the element for a specific L/G ratio. It follows therefore that the NTU level is a function of the L/G ratio. An example of this procedure is shown in Table 1 for an inlet water temperature 19 °C, an exit temperature of 16 °C (a condition which would be applicable to a chilled ceiling), an ambient condition of 12 °C AST and an L/G ratio of 1.0. It is seen that the NTU level on the basis of this analysis is 1.43. The analysis is based on dividing the counter-flow tower into 30 horizontal elements each with drop of 0.1 K in the water temperature.

3. Required NTU analysis in this application

An experimental research facility has been developed for this research (see Fig. 1) and is described in detail elsewhere [12–14]. The test rig consists of an open counter-flow cooling tower and counter-flow plate heat exchanger, both with enhanced heat transfer areas for the purpose of minimizing the approach conditions. The tower has 195 m² of waveform packing with a surface density of 200 m² m⁻³, while the plate heat exchanger has a design overall heat transfer coefficient of 4691 W m⁻² K⁻¹. The cooling tower has a high degree of inbuilt operating flexibility with an air and water flow rate range of 0.8–2.8 m³ s⁻¹ and 0.8–2.4 L s⁻¹, respectively, giving a possible L/G mass flow rate ratio range of 0.25–3.0. The cooling tower fan motor is inverter controlled while the 24 kW electric cooling load heater is thyristor controlled. Secondary approach temperatures (SAT) as low as 2 K have been measured in the rig at an adiabatic saturation temperature (AST) of 17 °C and 20 kW heat rejected.

Cooling towers can operate over a wide range of water flow rates, air flow rates, and heat load rejection rates, with variation in the primary approach temperature. A distinction can be drawn between the achieved (or available) NTU level and the required NTU level. The NTU level required is a function only of the operating conditions. The NTU level achieved depends on the operating conditions and the performance of a particular tower and is obtained from the experimental results. For any particular tower the achieved NTU level must at least equal the required NTU level for the operating conditions if the tower is to perform successfully.

As the NTU level varies with the ambient AST and with the L/G ratio it can be determined, using the template contained in Table 1, for a wide range of L/G ratio and for the ambient ASTs commonly encountered in temperate climates. This analysis has been completed and the results are indicated in Fig. 2.

Fig. 2 indicates that the NTU level required is strongly dependent on the L/G ratio used at the higher ambient AST conditions – in excess of 8 °C. The NTU level required is always less than 1.0 at all L/G ratios while the AST remains at or below 7 °C. However the main interest in this work is in evaporative cooling performance at the higher AST temperatures – 10 °C and above. At these ambient temperatures the NTU level required rises sharply

Table 1
Cooling tower NTU analysis based on an inlet water temperature of 19 °C, an outlet temperature of 16 °C, an AST of 12 °C, an L/G ratio of 1.0 with 30 elements each raising the mean water temperature by 0.1 K

| Element reference number	Mean water temperature of element (°C)	Enthalpy of saturated air at mean. water temp. (kJ/kg)	Enthalpy unsaturated air at mid element (kJ/kg)	Inverse enthalpy difference × 0.1 K (K kg/kJ)	Accumulated inverse of enthalpy difference × 0.1 K (K kg/kJ)	NTU level
1	16.05	45.2142	34.6124	0.0094	0.0094	0.0394
2	16.15	45.5044	35.0304	0.0095	0.0190	0.0793
3	16.25	45.7957	35.4484	0.0097	0.0286	0.1197
4	16.35	46.0880	35.8664	0.0098	0.0384	0.1606
5	16.45	46.3814	36.2844	0.0099	0.0483	0.2020
6	16.55	46.6758	36.7024	0.0100	0.0584	0.2439
7	16.65	46.9712	37.1204	0.0102	0.0685	0.2864
8	16.75	47.2677	37.5384	0.0103	0.0788	0.3293
9	16.85	47.5653	37.9564	0.0104	0.0892	0.3728
10	16.95	47.8640	38.3744	0.0105	0.0997	0.4169
11	17.05	48.1638	38.7924	0.0107	0.1104	0.4615
12	17.15	48.4647	39.2104	0.0108	0.1212	0.5067
13	17.25	48.7667	39.6284	0.0109	0.1322	0.5524
14	17.35	49.0698	40.0464	0.0111	0.1432	0.5987
15	17.45	49.3741	40.4644	0.0112	0.1545	0.6456
16	17.55	49.6795	40.8824	0.0114	0.1658	0.6932
17	17.65	49.9861	41.3004	0.0115	0.1773	0.7413
18	17.75	50.2938	41.7184	0.0117	0.1890	0.7900
19	17.85	50.6027	42.1364	0.0118	0.2008	0.8394
20	17.95	50.9128	42.5544	0.0120	0.2128	0.8894
21	18.05	51.2241	42.9724	0.0121	0.2249	0.9401
22	18.15	51.5367	43.3904	0.0123	0.2372	0.9914
23	18.25	51.8504	43.8084	0.0124	0.2496	1.0434
24	18.35	52.1653	44.2264	0.0126	0.2622	1.0960
25	18.45	52.4815	44.6444	0.0128	0.2750	1.1493
26	18.55	52.7990	45.0624	0.0129	0.2879	1.2034
27	18.65	53.1177	45.4804	0.0131	0.3010	1.2581
28	18.75	53.4376	45.8984	0.0133	0.3142	1.3135
29	18.85	53.7589	46.3164	0.0134	0.3277	1.3697
30	18.95	54.0814	46.7344	0.0136	0.3413	1.4266
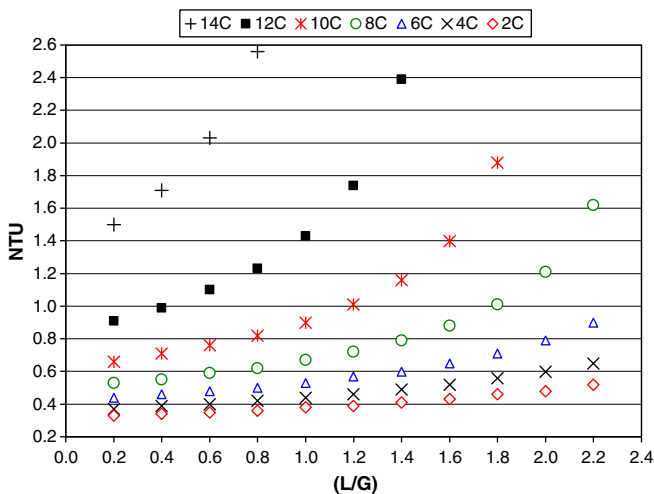
Fig. 2. Required NTU levels for a range of L/G ratios and a series of ambient AST conditions (14–2 °C) (based on an inlet water temperature of 19 °C and an outlet temperature of 16 °C).

with AST and with L/G ratio. At an L/G ratio of 1.0, for example, an NTU level of 1.0 is only sufficient while the ambient AST remains below 11 °C. At this ratio, an AST of 12 °C requires an NTU of 1.43 and at an AST of 14 °C requires an NTU of 3.74. It is also clear from this chart that the combination of a high ambient AST (>14 °C) and a high L/G ratio (>1.4) leads to levels of NTU required which are impractically large, being in excess of 4.0. It also seems that it is preferable to maintain the L/G level in the region of 1.0 or less, at the higher AST levels of interest, if NTU levels are to be kept within practical limits of approximately 4.0. Coulson and Richardson [23] indicate a practical range of NTU level of 0.5–2.5 for traditional applications. However for this work, in which a low approach is a key requirement, it might be expected that the NTU level would exceed 2.5. Given that the water flow rate is generally dictated by the load rejected and that a low range condition is required in low approach conditions (see Fig. 1), the tower water flow rate is necessarily relatively high in this application. If also the L/G ratio is maintained below 1.0 (i.e., $G > L$), the conclusion that the process inherently requires a relatively high air flow rate seems inescapable.

It is also of interest, in this context, to examine the conditions described in Fig. 2 from the point of view of the approach conditions. The data contained in Fig. 2 can be expressed in terms of the NTU required as a function of the approach condition achieved (see Fig. 1). This is done in Fig. 3 for L/G ratios ranging from 0.2 to 2.2.
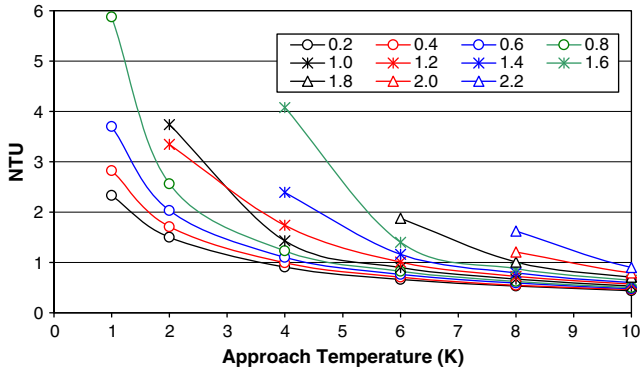
Fig. 3. Required NTU levels at the low approach conditions of interest in this application for L/G ratios from 0.2 to 2.2 (based on an inlet water temperature of 19 °C and an outlet temperature of 16 °C).
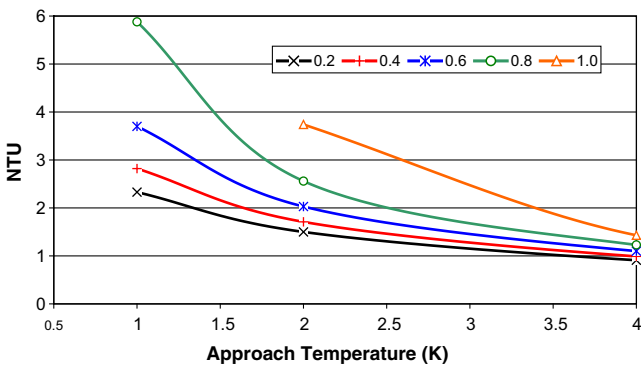

Fig. 4. Required NTU levels at the low approach conditions of interest in this application for L/G ratios ranging from 0.2 to 1.0.

It is clear from Fig. 3 that the NTU level required has a low dependence on the approach condition while the approach is in excess of 4 K and the L/G ratio is at or below 1.0. If, as reasoned above, the NTU level required has a practical limit of 4.0, then the L/G level is limited to a maximum of 1.2, if the approach is not to exceed 2 K at design conditions.

When the data presented in Fig. 3 is examined for an approach condition of 1–4 K, the implications of the low approach condition are brought into focus as shown in Fig. 4. It seems that an L/G ratio in the range 0.2–1.0 is, perhaps, an appropriate range in this application as it allows an approach of less than 2 K at an NTU level of 4 and makes very low approach conditions possible at more practical NTU levels of <4. Also on this basis if a very low approach condition of the order of 1 K is to be possible, ensuring full exploitation of the ambient cooling potential, then L/G ratios of less than 0.6 would seem to be required, to achieve this.

4. Results and discussion of experimental tests

Tests were conducted in which the following five cooling tower operating variables were measured: (i) the inlet water temperature, (ii) the exit water temperature, (iii) the ambient AST, (iv) the water flow rate, and (v) the air flow rate. These measurements enable the performance of the tower to be analysed by determining the difference in enthalpy between the saturated air film and the unsaturated air at each element of the tower in accordance with Merkel's method.

As there is some evidence from the work of Bernier [17] that the NTU level achieved may have some slight dependence on the ambient AST and the inlet water temperature for a particular tower, the tests are more accurate when selected at similar AST levels and similar inlet water temperatures as shown in Table 2. This process was carried out for 10 selected tests. Furthermore, it was established previously that the combination of a high L/G ratio (>1.0) and a high ambient AST (>14 °C) results in required NTU levels which are impractically large (>4.0) even for this low approach application. Also it was seen that the preferable range for the L/G ratio was 0.2–1.0. Hence tests were also selected with L/G ratios in this range and as tests with primary cooling water in the range 11–20 °C are required for dry air–water cooling applications, the tests selected are also within this range. The experimental results are shown in Table 2.

These results can also be expressed graphically, as shown in Fig. 5. For the purpose of comparison, the results obtained when the general correlation of Bernier [17] and Kuehn et al. [18] is applied to the L/G ratios used in the tests are also shown in Fig. 5. As indicated in Fig. 5, the experimental results produced the following correlation:

$$\frac{KaV}{L} = 1.3\left(\frac{L}{G}\right)^{-0.77} \tag{9}$$

The experimentally determined NTU values exhibit an average uncertainty of $\pm 3.4\%$ (with a standard deviation of 0.1%) over its range, with an associated average uncertainty of $\pm 4.43\%$ for the range of the L/G ratio (standard deviation of 0.14%) and are indicated by error bars in Fig. 5. The comparison with the experimental work of

Table 2
NTU level achieved in the tower for a series of test results

Inlet water temperature (°C)	Exit water temperature (°C)	Ambient AST (°C)	L/G ratio for test	NTU level achieved
15.06	12.89	9.75	0.88	1.39
16.21	13.37	10.12	0.71	1.55
14.92	13.02	11.15	0.69	1.77
14.61	12.69	10.80	0.69	1.77
14.93	12.70	10.84	0.60	1.91
15.30	13.00	11.17	0.60	2.00
15.11	12.35	10.60	0.48	2.25
15.16	12.36	10.61	0.51	2.32
15.55	12.35	10.91	0.39	2.64
15.97	11.91	10.69	0.30	3.18

The inlet water temperature is within the range 15.4 °C ± 0.8 and the AST is within the range 10.4 °C ± 0.8. The heat rejected is constant at 20 kW for all tests.
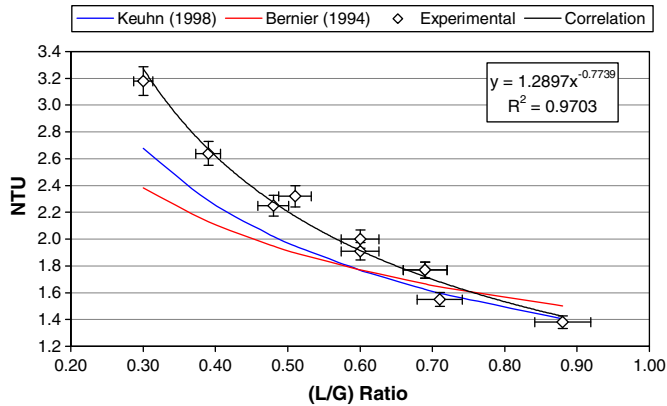
Fig. 5. Comparison of NTU achieved in the tests with two of the established correlations for more conventional applications. Range of *L/G* ratio is the range of interest in this work.

Bernier [17], carried out at WBTs of approx 16 °C and an approach condition of 5 K, is perhaps more appropriate to the current work. The comparison with Kuehn's work [18] is perhaps less appropriate, as it is based on model studies of Braun [19] and introduces some simplifications.

It must be borne in mind, that established correlations in this field (Eqs. (2)–(4)) relate to the primary objective of effective heat rejection, rather than, as in this work, a low approach and hence effective availability as the primary concern. Therefore, it is not necessarily to be expected that the correlation for this work will produce similar constants to established correlations. Nevertheless, in order to set this work in context, it is informative, to compare the measured results with those of other researchers in the general field.

While there is some scatter in the experimental data shown in Fig. 5, which is due to the experimental uncertainty, the coefficient of the regression (R^2) at 0.97 indicates a close fit. While differences exist between the level of the AST condition of this work (10.4 °C) and that of Bernier [17] (16 °C) and between the inlet water temperature of this work (15.4 °C) and that of Bernier (26 °C), the influence of these two parameters on the cooling tower performance is slight [17] and would not account for the large differences between the NTU levels achieved at *L/G* ratios of ⩽0.6 and which differences rise to a 60% increase in NTU level at an *L/G* level of 0.25. The test rig used in the work of Bernier was not designed for very low approach heat rejection (approach temperatures of 5 K are reported [17]) as has the tower used in this work. It is in this fact that the explanation of the differences in the NTU achieved at low *L/G* ratios lies rather than in any comparatively minor effects introduced by ambient and inlet water conditions. The impact of ambient AST on thermal effectiveness for this test rig has been examined in another phase of the research and is reported elsewhere [12].

The constant obtained in the experimental work (1.3) is equal to that given by Kuehn [18] and less than that given by Bernier (1.42) [17]. However the exponent (−0.77) is significantly less than that given by either Kuehn (−0.6) or Bernier (−0.43). Hence the NTU level, at the lower *L/G*

ratios of interest, is above that indicated by the existing correlations. This is to be expected as the experimental tower is designed for and is capable of achieving exceptionally low approach conditions of 2–3 K SAT and 1 K PAT [12–14]. As the NTU level achieved rises as the approach falls, a tower with a low design approach condition would be expected to achieve a high NTU level.

As discussed in Section 3, the results of a theoretical analysis of the NTU level required indicate that the *L/G* ratio in this application should be in the range of 0.2–1.0. Also if a very low approach condition is to be possible (of the order of 1 K), to guarantee full exploitation of ambient cooling potential, then an *L/G* ratio of ⩽0.6 would appear to be required. At these low *L/G* levels of interest in low approach evaporative cooling, in maritime temperate climates the test rig gives significantly higher NTU levels than conventional towers. In fact, at the low limit *L/G* ratio of 0.25, the test rig NTU level of 4.5 represents a 30–60% increase on the previously reported performance, for more conventional towers as shown in Fig. 5. This indicates that a building heat rejection system, designed on a similar basis to the test rig will have the ability to produce exceptionally low approach temperatures at a low water flow rate flux. This is due to two test rig design decisions: (i) the use of high area of packing fill (200 m² m⁻³) and (ii) a low water flow rate flux, both of which combine to significantly increase the residence time of the water droplets in the tower, and thereby decrease the approach, provided the air flow rate is at a level to absorb the water vapour in semi-humid to humid maritime ambient conditions and thereby maintain an enthalpy difference driving force at significant levels. Hence a lower rate of water flow rate per unit of air flow rate (a low *L/G* ratio) is essential in these climates.

5. Total heat transfer coefficient

As the cooling tower is predominantly a mass transfer device in this application, the evaluation of the volumetric total heat transfer coefficient ($kg_a\ s^{-1}\ m^{-3}$) is of particular interest. This coefficient is usually determined in terms of *Ka*, not *K* due to the difficulty of isolating the relevant area from the transfer coefficient. The heat transfer area is not necessarily equal to the packing surface area as heat and mass can also be transferred from water droplets in suspension in the air stream.

Due to the requirement in this work to achieve a low approach condition, the volume of the test rig tower packing for the cooling load rejected (20 kW) is considerably larger than in traditional applications. As the ratio of heat rejected to volume of packing is low, it would therefore be expected that the volumetric heat transfer coefficient is also low in comparison with more conventional applications such as in refrigeration condenser heat rejection, where the design approach condition is often a multiple of that required in this application.

The quantity Ka is usually correlated as follows (as described by Coulson and Richardson [23]):

$$Ka \propto (G')^n (L')^{(1-n)} \tag{10}$$

where G' and L' are the flow rate flux (kg s^{-1} m^{-2}). Coluson and Richardson [23] give the following general correlation for traditional industrial scale towers in conventional applications:

$$Ka = 2.95(G')^{0.72}(L')^{0.26} \tag{11}$$

Other, more specific work by Goshayshi [24] with reference to the experimental work on a model laboratory tower (however with a packing density of 200 m^2 m^{-3}, similar to the semi-industrial scale test rig used in this work) resulted in a correlation of

$$Ka = 1.75(G')^{0.6}(L')^{0.45} \tag{12}$$

This indicates that Goshayshi [24] found that the proportionality constant for the model tower was considerably lower but that the water flow rate has a greater impact and the air flow rate a lesser impact on heat transfer than with the industrial scale tower general behaviour described in Eq. (11). For the work described in this paper, Eq. (9) can be re-written as follows:

$$Ka = \frac{1.3}{V}(G)^{0.77}(L)^{0.23} \tag{13}$$

As the cross-sectional area of the tower is 0.84 m^2 it follows that $L = 0.84 (L')$ and $G = 0.84(G')$ and therefore:

$$Ka = \frac{1.091}{V}(G')^{0.77}(L')^{0.23} \tag{14}$$

The test cooling tower packing volume can be seen in terms of the packing volume (0.97 m^3) or the total volume of the space between the nozzle layer and the water surface in the reservoir (1.52 m^3): i.e. the volume associated with the formal packing surface and total possible surface, respectively. Hence in terms of the packing volume

$$Ka = 1.12(G')^{0.77}(L')^{0.23} \tag{15}$$

The values of the exponents (0.77 and 0.23) are remarkably similar to those in Eq. (11) (0.72 and 0.26), which is probably explained on the basis that both are industrial scale towers. The experimental results constant (1.12) however is very much lower at 40% of that quoted in this equation. The low value of Ka in comparison with the results obtained by the application of Eqs. (11) and (12) is explained by the design of the test rig cooling tower. For the volume of the tower fill at 0.97 m^3 the heat rejected (20 kW) is considerably lower than that would be the case in more conventional applications. As Ka is a volumetric total heat transfer coefficient (kg$_a$ s^{-1} m^{-3}), it is therefore to be expected that the total heat transfer per unit volume is considerably lower than more conventional applications, such as in refrigeration condenser heat rejection. The tower design with high packing volume per unit of heat rejected effectively produces a high NTU level and a low approach.
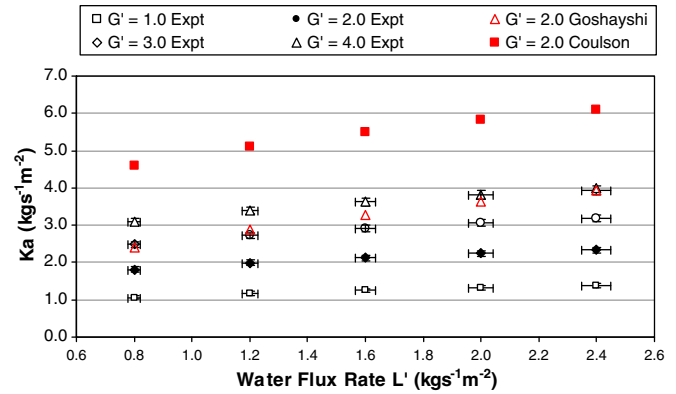

Fig. 6. Volumetric total heat transfer coefficient for the test rig, as a function of water flow rate flux, for a series of air flow rate fluxes based on the experimental results. The comparison with the results of two other correlations is also shown.

However the corollary of this is that the volumetric total heat transfer coefficient is low.

The correlation (Eq. (15)) can be expressed graphically as the variation in Ka with water flow rate flux for a series of air flow rates flux. Fig. 6 shows these relationships. For these data, the experimental uncertainty associated with the water flowrates varies from ±2.74% to ±2.09% at 0.8 kg s^{-1} m^{-2} and 2.4 kg s^{-1} m^{-2}, respectively. The uncertainty associated with the Ka values can be seen to vary from ±4.3% at 1.0 kg s^{-1} m^{-3} to ±2.9% at 4.0 kg s^{-1} m^{-3}. All uncertainty estimations are indicated by error bars in Fig. 6. A comparison of the results of the experimental tests with the correlations described in Eqs. (11) and (12) is also shown. As expected the transfer coefficient (Ka) is less than that indicated by Eqs. (11) and (12). In general the results of the tests indicate that the total volumetric heat transfer coefficient is strongly dependent on the air flow rate and with a weak, but not insignificant, dependence on the water flow rate. An increase of 1.0 kg s^{-1} m^{-2} in the air flow rate raises the transfer coefficient at all water flow rates by about 60% and raises it above that previously achieved, at all water flow rates, indicating the dominance of the air flow rate in effecting heat transfer. Hence the air flow rate is a far more crucial determinant of the heat transfer ability of the tower than the water flow rate.

The heat transfer coefficient can also be correlated as a function of the air Reynolds number as well as a function of the mass flow rates. However in the case of this application it is seen that while the dependence on the water flow rate is weaker than that of the air flow rate, nevertheless the dependence on the water flow rate is not insignificant or negligible. Therefore, the form of the correlation which includes the water flow rate is preferred. The correlation with respect to air Reynolds number is useful, however, when the inter-relationship between the heat transfer coefficient achieved and the packing pressure loss is to be examined, when for example the energy efficiency of the heat rejection process is to be optimised. However such an analysis is beyond the scope of the current work.

6. Conclusions

Initially an analysis of the NTU level required with low approach cooling towers operating at conditions typically encountered in chilled ceiling applications, in temperate climates, was conducted. Subsequently the thermal performance of an experimental open cooling tower, at a series of low water to air flow rate ratios, which are required in low approach water temperature cooling, was measured. The measured results have been analysed in terms of the tower coefficient achieved and a new correlation has been developed from this analysis which is applicable to low ($L/G < 1.0$) water to air flow rate ratios. Using this correlation a further correlation has been derived for the volumetric heat transfer coefficient, based on the air and water flow rate flux in the tower. Both correlations have been compared with established correlations in the literature for open towers in more traditional applications and have been found to differ considerably from existing correlations. The correlations proposed in this work provide a key parameter for the design of this form of heat dissipation in buildings. Specifically, the following conclusions can be drawn:

1. The results of the theoretical analysis indicate that the ratio of the water flow rate to the air flow rate in the tower needs to be in the range of 0.2–1.0 for this application. As a low approach condition requires a relatively low range condition it follows that the water flow rate itself is relatively high. Hence it can be concluded that a relatively high cooling tower air flow rate, per unit of load rejected, is required in this application. The correlation for the cooling tower coefficient in this work was

$$\frac{KaV}{L} = 1.3\left(\frac{L}{G}\right)^{-0.77}$$

2. At the low L/G ratios of interest (<1.0), the coefficient rises significantly as the L/G ratio falls with a maximum increase of 30–60%, over that indicated for traditional towers, at the lowest L/G ratio of 0.25. This indicates that building heat dissipation systems, designed on the same basis as the test rig, have an ability to produce very low approach temperatures (and hence higher availability levels) at low water to air flow rate ratios.

3. The correlation for the heat transfer coefficient was

$$Ka = 1.12(G')^{0.77}(L')^{0.23}$$

In this correlation, the values of the exponents are very similar to those quoted in the literature for industrial scale towers; however, the experimental results constant (1.12) is considerably less. This indicates that while the pattern of the variation in the volumetric heat transfer coefficient with air and water flow rate flux is similar to that for traditional towers the actual volumetric heat transfer coefficient achieved is relatively low, due to the high volume of cooling tower fill employed, per unit of heat rejected. There is a potential for further analysis to determine the total heat transfer coefficient as a function of air Reynolds number. While such an analysis is outside the scope of the current work, its utility is clear in the context of inverter controlled cooling tower fans as part of an energy efficient control strategy for this form of heat rejection in buildings.

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