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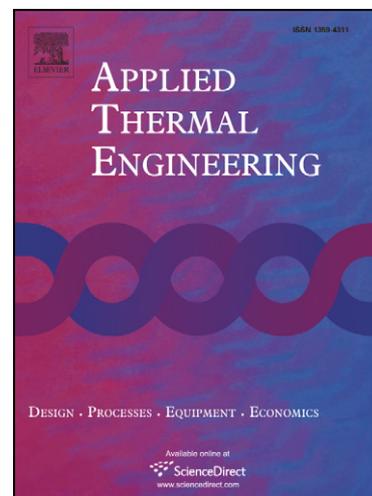
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Heat Transfer Correlations for Low Approach Evaporative Cooling Systems in Buildings

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Abstract

Meteorological enthalpy analysis of temperate and maritime climates above 45°N latitude suggests that the water-side evaporative cooling technique has considerable unrealised potential with contemporary "high temperature" building cooling systems - such as chilled ceilings. As low approach conditions are the key to fully exploiting the cooling potential of the ambient air, thermal performance at such conditions needs to be investigated. To address the research issues a test rig, based on an open cooling tower and plate heat exchanger and designed to maximize evaporative cooling potential, has been constructed. A combination of experimental measurement and analysis is used in the investigations. The thermal performance of open cooling towers, resulting from experimental research, is usually correlated, as a function of the water and air flow rate, in terms of the cooling tower coefficient, or number of transfer units (NTU) achieved. A new correlation has been developed for the low approach experimental tower, which shows a significant increase in the NTU level, at the lower water to air flow rate ratios of interest. As the cooling tower in this application is predominantly a mass transfer device in summer, the evaluation of the total volumetric heat and mass transfer coefficient ($\text{kg}_a \cdot \text{s}^{-1} \cdot \text{m}^{-3}$) is of particular interest. This coefficient has also been determined for the experimental tower and provides a key parameter for the design of this low energy form of heat dissipation in buildings.

Keywords:

Free Cooling, Low Energy Cooling, Evaporative Cooling, Cooling Towers, Chilled Ceilings, Thermal Performance, Heat Transfer.

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Nomenclature

Symbol

a	heat transfer area per unit volume ($\text{m}^2 \cdot \text{m}^{-3}$)
C_{pw}	specific heat of water at constant pressure ($\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)
C_T	cooling tower constant, dimensionless
Δh	elemental rise in the air enthalpy across a cooling tower element ($\text{kJ} \cdot \text{kg}^{-1}$)
Δt	drop in cooling tower water temperature across a cooling tower element (K)
G	air flow rate in the tower ($\text{kg} \cdot \text{s}^{-1}$)
G'	air flow rate flux in the tower ($\text{kg} \cdot \text{s}^{-1} \cdot \text{m}^{-2}$)
h_a	enthalpy of the bulk air stream in the cooling tower ($\text{kJ} \cdot \text{kg}^{-1}$)
h_{asw}	enthalpy of air in the cooling tower saturated at the bulk water temperature ($\text{kJ} \cdot \text{kg}^{-1}$)
K	total heat transfer coefficient ($\text{kg}_a \cdot \text{s}^{-1} \cdot \text{m}^{-2}$)
Ka	product of total heat transfer coefficient and heat transfer area ($\text{kg}_a \cdot \text{s}^{-1} \cdot \text{m}^{-3}$)
L	primary water flow rate in tower ($\text{kg} \cdot \text{s}^{-1}$)
L'	primary water flow rate flux in tower ($\text{kg} \cdot \text{s}^{-1} \cdot \text{m}^{-2}$)
n	exponent for G'
t	water temperature in the cooling tower ($^{\circ}\text{C}$)
T_{as}	ambient adiabatic saturation temperature (AST) ($^{\circ}\text{C}$)
T_{pa}	primary approach temperature (PAT) (K)
T_{pr}	primary loop return temperature ($^{\circ}\text{C}$)
T_{ps}	primary loop supply temperature ($^{\circ}\text{C}$)
T_{sa}	secondary approach temperature (SAT) (K)
T_{sr}	secondary loop return temperature ($^{\circ}\text{C}$)
T_{ss}	secondary loop supply temperature ($^{\circ}\text{C}$)
V	heat transfer volume in tower (m^3)
x	exponent for cooling tower L/G ratio in context of experimental correlations

Subscript

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

1. Introduction

Traditionally interest in evaporative cooling, as an effective cooling technique for buildings, has focused on hotter dry latitudes [18], where it was seen as being mainly applicable. Up to quite recently this focus has persisted [2]. Recent work however on air-side [13] and water-side [6] evaporative cooling, has demonstrated the considerable potential of the technique in temperate and maritime European regions. While the water-side evaporative cooling technique can be exploited with any water based building cooling system, the technique is particularly advantageous when used in conjunction with a chilled ceiling system, due to the higher cooling water temperatures (14-20°C) which are employed and hence the higher cooling water availability levels which result.

Figure 1 shows a simplified schematic of a water side indirect evaporative cooling system, with the key operating parameters indicated. The natural governing parameter is the adiabatic saturation temperature (AST), approximated by the psychrometric wet bulb temperature (WBT) of the ambient air. With indirect systems the significant performance parameter is the secondary approach temperature (SAT) which is equal to $T_{ss} - T_{as}$. It has been shown that cooling water availability levels heavily depend on the approach conditions achieved in European locations and that SATs as low as 3 K and primary approach temperatures (PATs) as low as 1 K are technically feasible and viable [7] with contemporary cooling tower packing surface densities of $200\text{m}^2.\text{m}^{-3}$ and low approach plate heat exchangers [5].

There are two basic approaches to this form of indirect cooling system (i) the closed wet cooling tower and (ii) the open tower with external plate heat exchanger. Each

arrangement has advantages in particular circumstances and locations [8]. While much research has been done on the closed tower in this application [10,12] there is a need to investigate the thermal performance of the open tower in operating conditions well outside those encountered in refrigeration condenser heat rejection, with range and approach conditions as low as 1-4 K, cooling water temperatures of 12-20°C and ambient conditions of < 20°C AST. These conditions result in much smaller levels of enthalpy difference, the key driving force in the tower, and therefore smaller associated heat and mass transfer rates with, crucially, resulting higher air and water flow rates than those encountered in more conventional applications.

To address these issues an experimental research facility has been developed at the Dublin Institute of Technology and is described in detail elsewhere [8]. 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 195m² of wave-form packing with a surface density of 200m².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. SATs as low as 2 K have been measured in the rig at an AST of 17°C and 20kW heat rejected.

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. 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:

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

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 [1] gives the following general correlation for cooling towers:

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

Bernier [3] measured the thermal performance of an experimental, semi-industrial scale cooling tower in a laboratory along the lines described in ASHRAE [1]. The resulting correlation was:

$$\frac{K aV}{L} = 1.42 \left(\frac{L}{G} \right)^{-0.43} \quad (3)$$

Kuehn et al. (1998) [14] gives a general correlation, based on the model studies of Braun et al. (1989) [4] and when $L < G$ as follows:

$$\frac{K aV}{L} = 1.3 \left(\frac{L}{G} \right)^{-0.6} \quad (4)$$

Historically the work of Lowe and Christe [15] 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 heat transfer coefficient, Ka ($\text{kg}_a \cdot \text{s}^{-1} \cdot \text{m}^{-3}$) can be determined from Equation 1 as $\text{NTU}(L/V)$. 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 ($16\text{-}26^\circ\text{C}$) in temperate climates is close to the water temperature ($12\text{-}20^\circ\text{C}$) and hence there is often little 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 3K). 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 [16], 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 determination of the NTU level is commonly undertaken by applying Merkel's equation in the following form:

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

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 0.1K). 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 [17]

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

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

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

therefore, for each element;

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

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

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 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 indicated in Figure 2.

Figure 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 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 clear also 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 [9] indicate a practical range of NTU level of 0.5 to 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, the tower water flow rate is 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 Figure 2 from the point of view of the approach conditions. The data contained in Figure 2 can be expressed in terms of the NTU required as a function of the approach condition achieved (see Figure 1). This is done in Figure 3 for L/G ratios ranging from 0.2 to 2.2.

It is clear from Figure 3 that the NTU level required has a low dependence on the approach condition while the approach is in excess of 4K 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 Figure 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 Figure 4. It seems that an L/G ratio in the range 0.2 to 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 1K 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 [3] 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. 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 ($>14^{\circ}\text{C}$) 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 $12\text{-}20^{\circ}\text{C}$ are required for chilled ceilings, 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 Figure 5. For the purpose of comparison also shown in Figure 5 are the results obtained when the general correlation of Kuehn et al. [14] and Bernier [3] is applied to the L/G ratios used in the tests. As indicated in Figure 5 the experimental results produced the following correlation:

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

The comparison with the experimental work of Bernier [3], carried out at 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 [14] is perhaps less appropriate, as it is based on model studies of Braun [4] and introduces some simplifications.

It must be borne in mind, that established correlations in this field relate 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.

The constant obtained in the experimental work (1.3) is equal to that given by Kuehn [14] and less than that given by Bernier (1.42) [3]. 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 capable of achieving exceptionally low approach conditions and 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.

At the low L/G levels of interest (< 1.0) 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 previously reported performance, for more

conventional towers as shown in Figure 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 ($200\text{m}^2\cdot\text{m}^{-3}$) and (ii) a low water flow rate flux, both of which combine to significantly increase the residence time of the water droplets in towers, 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. 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 ($\text{kg}_a\cdot\text{s}^{-1}\cdot\text{m}^{-3}$) is of particular interest. This coefficient is usually determined in terms of K_a , 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 [9]):

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

where G' and L' are the flow rate flux ($\text{kg}\cdot\text{s}^{-1}\cdot\text{m}^{-2}$). Coulson and Richardson [9] give the following general correlation for traditional industrial scale towers in conventional applications:

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

Other, more specific work by Goshayshi [11] with reference to experimental work on a model laboratory tower (however with a packing density of $200\text{m}^2\cdot\text{m}^{-3}$, similar to the semi-industrial scale test rig used in this work) resulted in a correlation of:

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

This indicates that Goshayshi [11] 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 Equation 11. For the work described in this paper Equation 9 can be re-written as follows:

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

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

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

The test cooling tower packing volume can be seen in terms of the packing volume (0.97 m³) or the total volume of the space between the nozzle layer and the water surface in the reservoir (1.52m³) i.e. the volume associated with the formal packing surface and total possible surface respectively. Hence in terms of the packing volume

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

The values of the exponents (0.77 and 0.23) are remarkably similar to those in Equation 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 correlation (Equation 15) can be expressed graphically as the variation in Ka with water flow rate flux for a series of air flow rates flux. Figure 6 shows these relationships. A comparison of the results of the experimental tests with the correlations described in Equations 11 and 12 is also shown. As expected the transfer coefficient (Ka) is less than that indicated by Equations 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.

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 “high” 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 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 should be in the range 0.2–1.0 for this application. As a low approach condition requires a relatively low range condition the water flow rate is itself 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 an increase of 30%-60%, over that indicated for traditional towers, at a 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 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.

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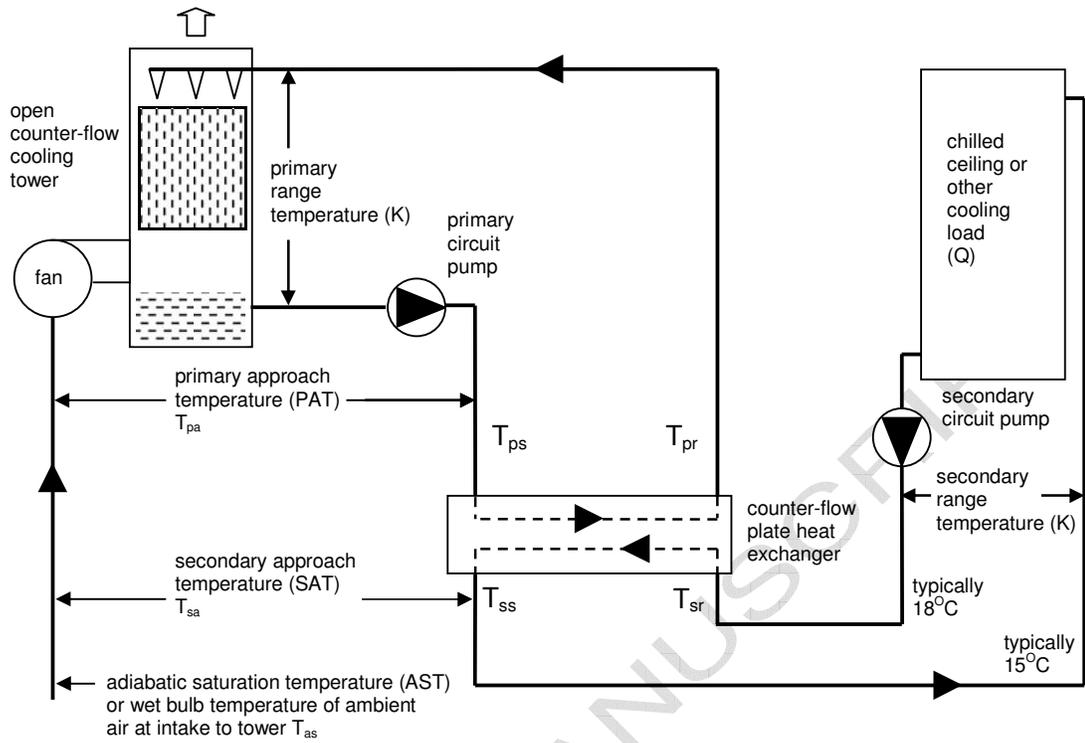


Figure 1: Simplified schematic of a water-side indirect evaporative cooling system

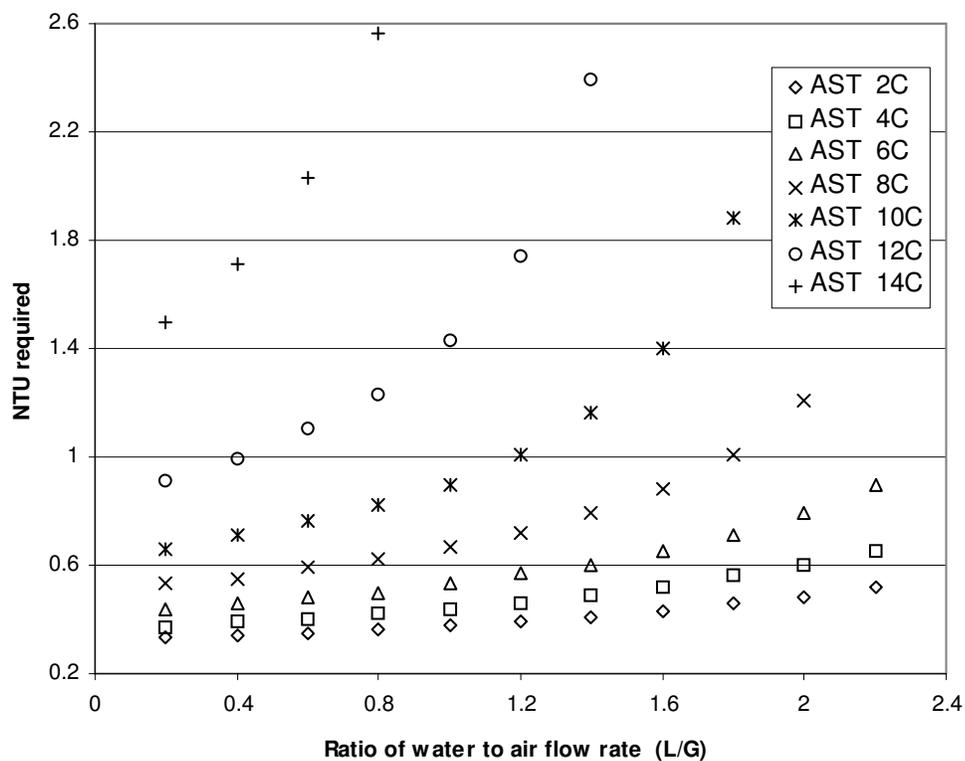


Figure 2. Required NTU levels for a range of L/G ratios and a series of ambient AST conditions. (based on an inlet water temperature of 19°C and an outlet temperature of 16°C).

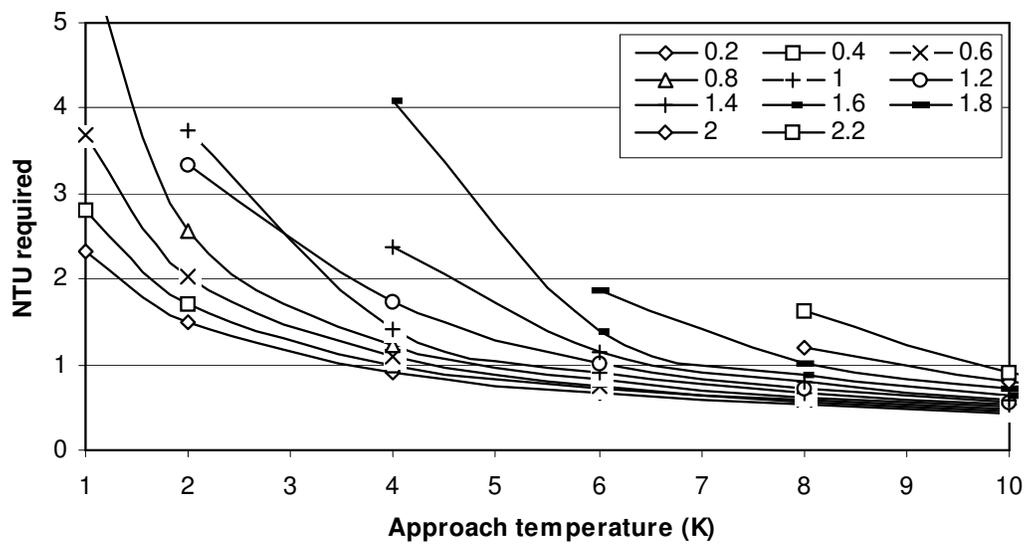


Figure 3. Required NTU levels at the low approach conditions of interest in this application with an L/G ratios ranging from 0.2 to 2.2. (based on an inlet water temperature of 19°C and an outlet temperature of 16°C).

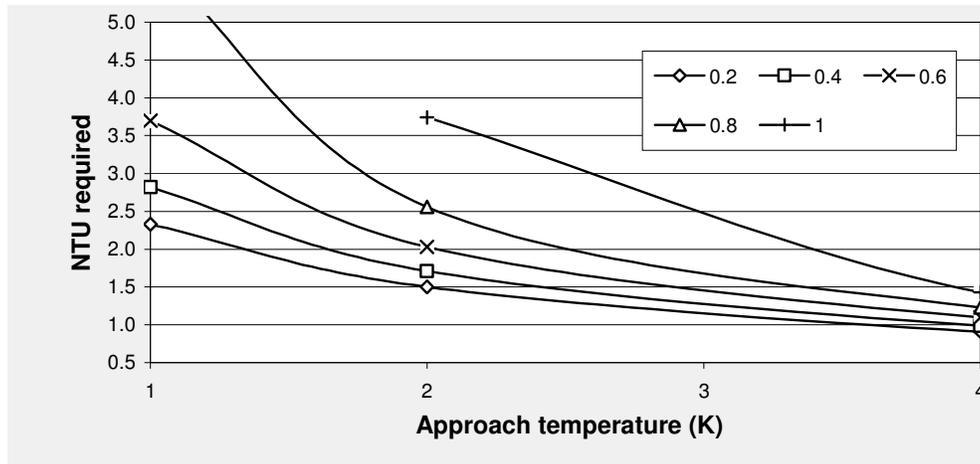


Figure 4. Required NTU levels at the low approach conditions of interest in this application with an L/G ratios ranging from 0.2 to 1.0.

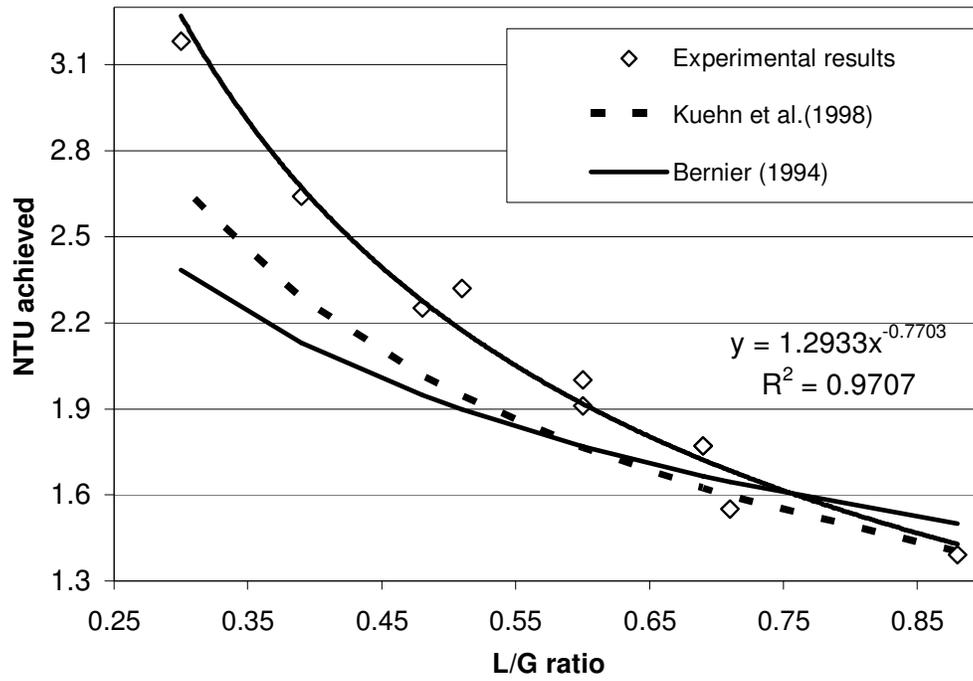


Figure 5. Comparison of NTU achieved in the tests with two of the established correlations for more conventional applications. Range of L/G ratio is the range of interest in this work.

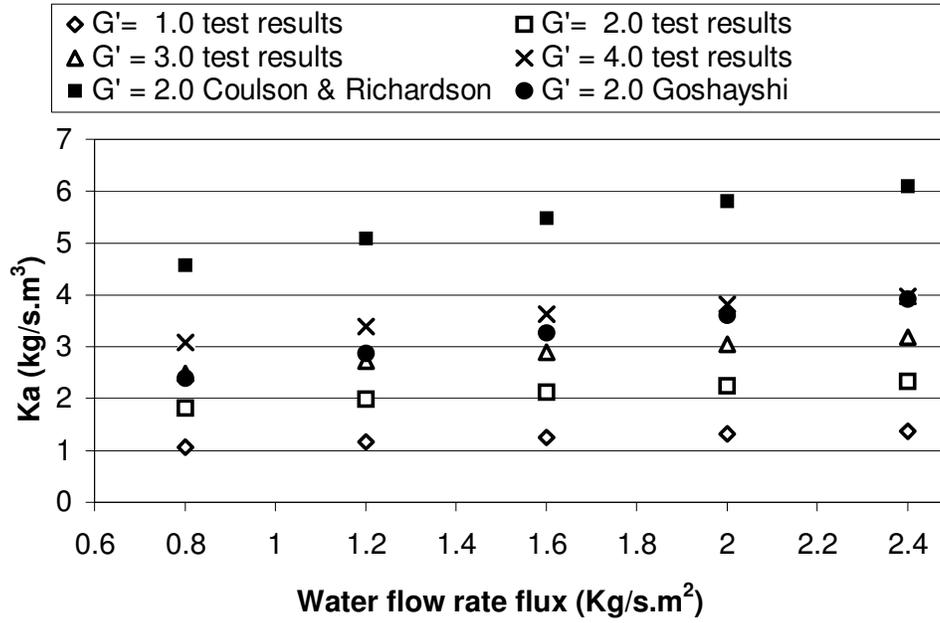


Figure 6 Volumetric total heat transfer coefficient for the test rig, as a function of water flow rate flux, for a series of air flow rate fluxes based on the experimental results. The comparison with the results of two other correlations is also shown.

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 x 0.1K (K.kg/kJ)	Accumulated inverse of enthalpy difference x 0.1K (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

Table 2: NTU level achieved in the tower for a series of test results. The inlet water temperature is within the range 15.4°C±0.8K and the AST is within the range 10.4 ±0.8K. The heat rejected is constant at 20kW for all tests.

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