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Thermal Performance of Low Approach Evaporative Cooling Systems in Buildings

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Abstract

Meteorological enthalpy analysis of temperate and maritime climates above 45˚N 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 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 maximise evaporative cooling potential, has been constructed at DIT. A combination of experimental measurement and analysis is used in the investigations.

The 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 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, the evaluation of the total volumetric heat and mass transfer coefficient (kg/sm^3) is of particular interest. This coefficient has been determined for the experimental tower and provides a key parameter for the design of this form of heat dissipation in buildings.

Nomenclature

|	
T_{ps}	primary loop supply temp. (˚C)
T_{pr}	primary loop return temp. (˚C)
T_{ss}	secondary loop supply temp. (˚C)
T_{sr}	secondary loop return temp. (˚C)
T_{as}	amb. adiabatic sat. temp. (AST) (˚C)
T_{pa}	primary approach temp. (PAT) (K)
T_{sa}	secondary approach temp. (SAT) (K)
K	total heat transfer coefficient
Ka	product of total heat transfer coefficient and heat transfer area ($kg/s.m^3$)
a	heat transfer area per unit volume m^2/m^3
V	heat transfer volume in tower (m^3)
L	primary water flow rate in tower
L'	primary water flow rate flux in tower ($kg/s.m^2$)
G	air flow rate in the tower (kg/s)
G'	air flow rate flux in the tower
C_T	cooling tower constant,
x	exponent for cooling tower L/G ratio in context of experimental
n	exponent for G'
Subscripts	
ps	primary supply
pr	primary return
ss	secondary supply
sr	secondary return
as	adiabatic saturation
pa	primary approach
sa	secondary approach
T	relating to the cooling tower

1. Introduction and Background

Traditionally interest in evaporative cooling, as an effective cooling technique for buildings, was focus on hotter dry latitudes (Watt, 1986), where it was seen as being mainly applicable. Up to quite recently this focus has persisted (Bom et al., 1999). Recent work however on air-side (IEA, 2001), and water-side

(Costelloe and Finn, 2003) 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 (14-18°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 3K are technically feasible with contemporary cooling tower packing surface densities of $200m^2/m^3$ and low approach plate heat exchangers (Costelloe and Finn, 2003).
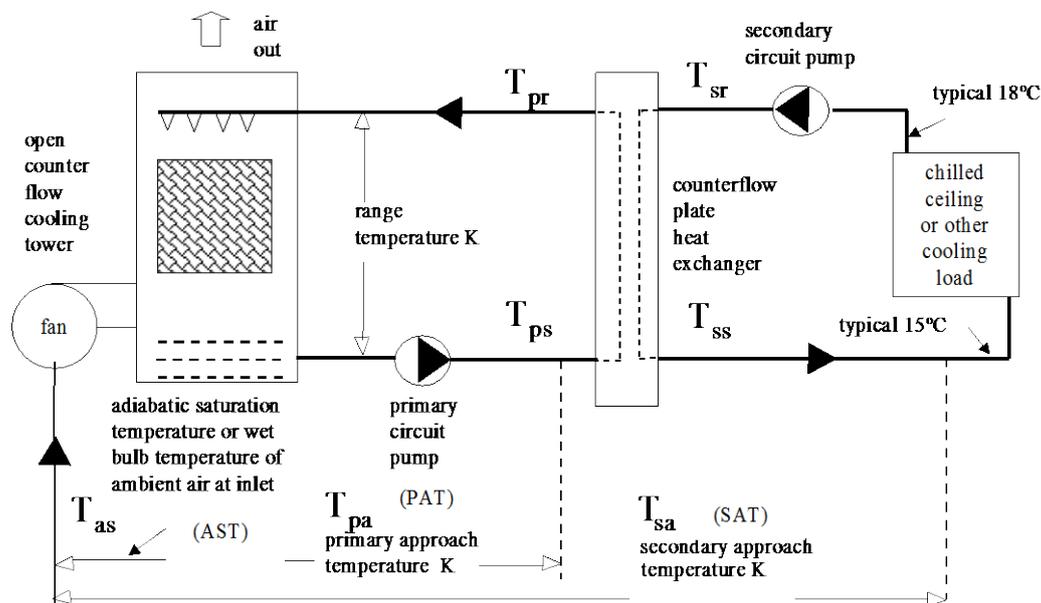
Figure 1: Simplified schematic of a water-side indirect evaporative cooling system

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 (Costelloe and Finn, 2000). While much research has been done on the closed tower in this application (Facao and Oliveira, 2000; Hasan and Siren, 2004) 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 14-18°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. To address these issues

an experimental research facility has been developed at the Dublin Institute of Technology and is described in detail elsewhere (Costelloe and Finn, 2000).

The key measure of open cooling tower performance is the cooling tower coefficient (KaV/L) or NTU level achieved. The performance of open cooling towers, resulting from experimental research is typically correlated in terms of the L/G ratio as follows:

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

2. Results and Discussion of 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 (Merkel, 1925). The results of this analysis for ten selected tests is shown in Table 1.

These results can also be expressed graphically, as shown in Figure 2. For the purpose of comparison also shown in Figure 2 are the results obtained when the correlation of Kuehn et al. (1998) and Bernier (1994) is applied to the L/G ratios used in the tests.

As indicated in Figure 2 the experimental results produced the following correlation:

$$\frac{KaV}{L} = 1.3 \left(\frac{L}{G} \right)^{-0.77} \quad \text{(Eq. 2)}$$

The comparison with the experimental work of Bernier, 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 correlation is perhaps less

appropriate, as it is based on the model studies of Braun et al. (1989) and introduces some simplifications

Inlet water temp ˚C	Exit water temp ˚C	Amb. AST ˚C	L/G ratio for test	NTU level
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

Table 1: 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 range10.4 +/-0.8K. The heat rejected is constant at 20kW.
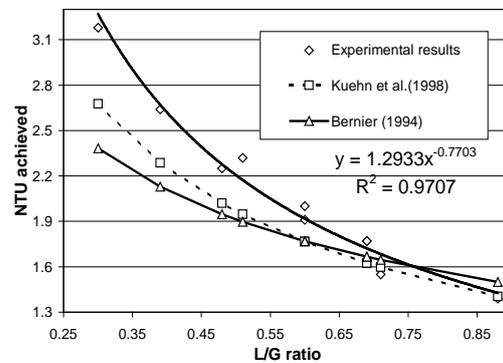
Figure 2: Comparison of NTU achieved in the tests with two of the established correlations. Range of L/G ratio is the range of interest in this work.

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.

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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 and less than that given by Bernier (1.42). 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 (< 0.7) in low approach evaporative cooling, in maritime temperate climates the test rig gives significantly higher NTU levels than Kuehn and considerably higher levels than Bernier. In fact at the low limit L/G ratio of 0.25, the test rig NTU level of 4.5 represents a 30 % increase on Kuehn and a 60% increase on Bernier. 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 design decisions (i) the use of high area of packing fill ($200m^2/m^3$) 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 vapour in martime ambient conditions and thereby maintain an enthalpy difference driving force. Hence a lower rate of water flow rate per unit of air flow rate is essential in these climates.

3. 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 ($kg/s.m^3$) 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.

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, is considerably larger than in traditional applications. As the ratio of heat rejected to volume of packing is low, it would therfore be expected that the volumetric heat transfer coefficient is also low in comparison with more conventional applications such as 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 (Coulson and Richardson, 2002):

$$Ka \propto (G')^n (L')^{(1-n)} \qquad \text{(Eq. 3)}$$

where G' and L' are the flow rate flux ($kg/s.m^2$). Coluson and Richardson (2002) give the following correlation for traditional industrial scale towers:

$$Ka = 2.95(G')^{0.72}(L')^{0.26} \quad \text{(Eq. 4)}$$

Other work by Goshayshi (1999) with reference to experimental work on a model laboratory tower (however with a packing density of $200m^2/m^3$, similar to the semi-industrial scale test rig used in this work) resulted in a correlation of :

$$Ka = 1.75(G')^{0.6}(L')^{0.45} \qquad \text{(Eq. 5)}$$

This indicates that Goshayshi 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 behaviour described by Coulson .

For this work Equation 2 can be re-written as follows:

$$Ka = \frac{1.3}{V}(G)^{0.77}(L)^{0.23} \quad \text{(Eq. 6)}$$

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:

$$Ka = \frac{1.091}{V}(G')^{0.77}(L')^{0.23} \qquad \text{(Eq. 7)}$$

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

$$Ka = 1.12(G')^{0.77}(L')^{0.23} \quad (Eq.8)$$

The values of the exponents (0.77 and 0.23) are remarkably similar to those quoted by Coulson and Richardson in Equation 4 (0.72 and 0.26), which is probably explained on the basis that both are semi-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 8) can be expressed graphically as the variation in Ka with water flow rate flux for a series of air flow rates flux. Figure 2 shows these relationships. A comparison of the results of the experimental tests with those of Goshayshi and Coulson & Richardson is also shown. As expected the transfer coefficient (Ka) is less than that indicated by the two other researchers and approximately half that indicated by Coulson. 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.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.
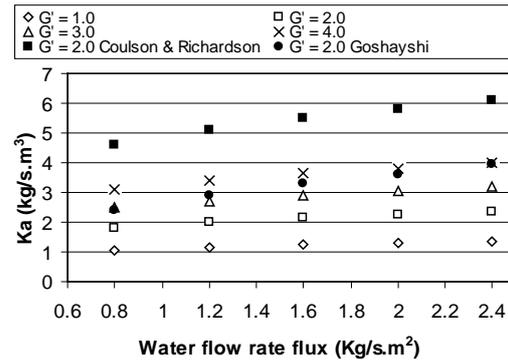
Figure 3 Volumetric total heat transfer coefficient for the test rig, as a function of water flow rate flux, for a series of air flow rate fluxes based on the experimental results. The comparison with the results of Goshayshi and Coulson & Richardson is also shown.

4. Conclusions

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 temperasture cooling, has been 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 correlation for the cooling tower coefficient in this work was:

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

At the low L/G ratios of interest (<0.7) 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.

2. 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 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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