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Thermal Efficiency Characteristics of Indirect Evaporative Cooling Systems

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Thermal efficiency characteristics of indirect evaporative cooling systems

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ABSTRACT

Recent developments in enhancing heat transfer in cooling towers, together with the success of chilled ceilings, have prompted a review of the evaporative cooling technique. in temperate maritime climates. The thermal efficiency of such systems is a key parameter, as a measure of the degree to which the system has succeeded in exploiting the cooling potential of the ambient air. This paper presents the results of experimental research into the thermal efficiency of a water-side open indirect evaporative cooling test rig designed to achieve low (1-4 K) approach conditions. Secondary efficiencies in the range 0.24-0.76 have been achieved.

NOMENCLATURE

T_{pf}	primary loop supply temp. (˚C)
T_{pr}	primary loop return temp. (˚C)
T_{sf}	secondary loop supply temp. (˚C)
T_{sr}	secondary loop return temp. (˚C)
T_{as}	ambient adiabatic sat. temp. (AST) (˚C)
T_{pa}	primary approach temp. (PAT)(K)
T_{sa}	secondary approach temp. (SAT)(K)
η_t	thermal efficiency
Subscripts	
ps	primary supply
pr	primary return
ss	secondary supply
sr	secondary return
as	adiabatic saturation
pa	primary approach
sa	secondary approach

1. INTRODUCTION

For many years, 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, 2003a) 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. The natural governing parameter in evaporative cooling is the wet bulb temperature (WBT) of the ambient air. The difference between the adiabatic saturation temperature (AST) and WBT is generally less than 0.25 K where the wet bulb depression is less than 11 K (Kuehn et al., 1998). The AST is used in this paper, in preference to the WBT as it is a fundamental property which can be determined without using empirical quantities.

Figure 1 shows a simplified schematic of a water side indirect evaporative cooling system, with the key operating parameters indicated. An important performance parameter is the primary approach temperature (PAT) which is equal to T_{ps} - T_{as}. This aspect is complicated by the

Table 1: European and Middle Eastern cities with similar design wet bulb temperatures (WBT) but different design dry bulb temperatures (DBT). Table in ascending order of the 1% design WBT (ASHRAE, 1997)

City	1% DBT ˚C	1% WBT ˚C	2% WBT ˚C
Dublin	20.6	17.1	16.3
Uppsala	23.7	17.2	16.2
Copenhagen	23.2	17.4	16.5
Oslo(Fornebu)	24.8	17.4	16.5
Helsinki	24.1	17.6	16.7
Birmingham	23.9	17.6	16.7
Plymouth	22.1	17.6	17.0
Stockholm (Bromma)	24.2	17.7	16.7
Al Jawf	39.7	17.7	17.3
Hof	25.0	17.8	16.8
Ankara	30.2	17.8	17.0
Bristol	24.5	18.2	17.3
Khamis Mushayt	30.6	18.2	17.6
Gdansk	24.8	18.3	17.2
Luxembourg	26.1	18.5	17.6
Brest	23.5	18.6	17.7
Salamanca	32.0	18.6	17.8
Prague	26.8	18.7	17.8
London (Heathrow)	25.7	18.7	17.8
Hamburg	25.9	18.8	17.9
Oostende	23.0	18.8	18.0
Munich	27.1	18.8	18.1
Zurich	26.4	18.9	18.1
Abha	29.9	19.0	18.3
Salzburg	27.9	19.1	18.2
Leipzig	27.6	19.2	18.4
Amsterdam	24.8	19.2	18.4
Koln	27.7	19.4	18.3
Geneva	28.5	19.4	18.6
Moscow	26.0	19.5	18.6
Vienna (Schwechat)	28.4	19.6	18.9

requirement, in contemporary applications, to separate the tower water circuit from the building cooling circuit with a heat exchanger. Hence the significant performance parameter becomes 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, 2003a). Hence when chilled ceiling systems are used, with typical cooling water supply temperatures of 14-18˚C considerable levels of cooling water availability are possible in many European (Costelloe and Finn, 2003a) and some Middle Eastern cities, as indicated in Table 1. These cities have similar design WBTs (the variation range is +/- 1.3 K) but have significantly different and in some cases widely different dry bulb temperatures (DBT) and locations.

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) 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 - 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 elsewhere (Costelloe and Finn, 2000). The thermal efficiency (η_t) of the process is defined as *the cooling achieved as a fraction of the maximum possible cooling which could have been achieved in the ambient conditions pertaining.* As such it is a key performance parameter and is a suitable means of assessing the thermal characteristics of the system. For the secondary circuit this parameter is defined by equations (1), similar equations define the primary circuit:

$$\eta_{ts} = \frac{T_{sr} - T_{ss}}{T_{sr} - T_{as}} = \frac{T_{sr} - T_{ss}}{\left(T_{sr} - T_{ss}\right) + \left(T_{ss} - T_{as}\right)}$$

$$= \frac{Sec.Range}{\left(Sec.Range\right) + \left(Sec.Approach\right)} \tag{1}$$

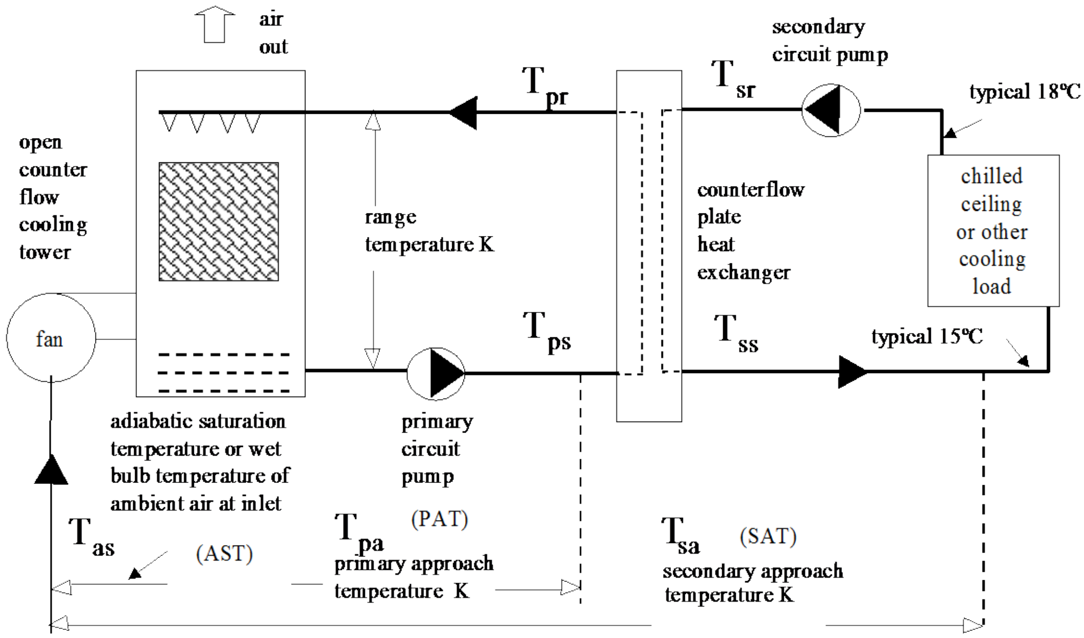

Figure 1: Simplified schematic of a water-side indirect evaporative cooling system

2. RESULTS AND DISCUSSION OF TESTS

Tests were conducted to investigate the impact of a range of operating variables on the thermal efficiency achieved. These variables include the cooling load imposed, the ambient AST, the primary and secondary circuit water flow rate and the cooling tower air flow rate. The parameter being examined was varied while the other test rig variables were maintained constant. As there is no control over the ambient AST a larger number of tests were conducted and those tests with near similar AST selected. Generally the criterion used is that the AST should not vary within the selected test group by more than +/- 0.9 K.

2.1 Inlet Water Temperature Variation

It would be expected that tower return water temperature has no significant bearing on the primary thermal efficiency (PTE) achieved when primary water flow rate, load and cooling tower air flow rate are maintained constant. Holding these variables constant maintains the tower range temperature constant and as the AST is also approximately constant (and therefore the approach is constant) the thermal efficiency is maintained approximately constant. By the same reasoning, for the secondary circuit, the secondary thermal efficiency (STE) is also maintained approximately constant. Figure 2 shows the results of the tests conducted to verify this aspect. It is seen that the tower return water temperature has no impact on the thermal efficiency achieved in these tests.

2.2 Cooling Load Variation

Tests were conducted to investigate the impact of load variation on the PTE and STE. In these tests as the imposed cooling load changes the ranges temperatures change in direct proportion, as the cooling water flow rates remain constant. Table 2 shows the results of these tests. The results clearly show that the thermal efficiency is not affected by changes in load. This implies, as shown in Equation (2), that the proportional change in the approach condition (F_a) must be approximately equal to the proportional change in the range condition (F_r), as the load is varied. As the change in the range condition is linear this implies a near linear correlation, for the rig between the load and the approach temperature.

$$\eta_{tp} = \frac{(T_{pr} - T_{ps})(F_r)}{(T_{pr} - T_{ps})(F_r) + (T_{ps} - T_{as})(F_a)} \qquad (2)$$

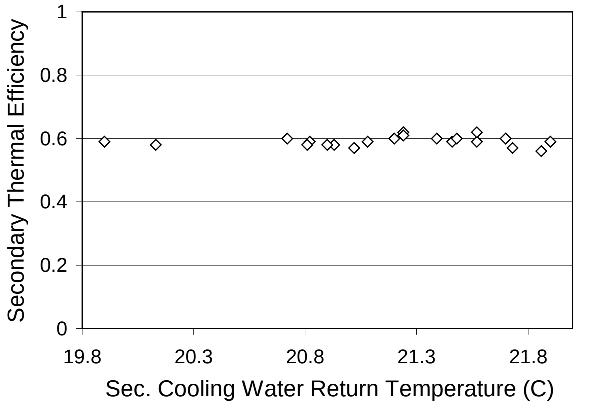

Figure 2: Constant secondary thermal efficiency with variation in secondary cooling water return temperature (load 20kw; AST 15.7 ˚C +/- 0.5 K; flow rates: primary 2.3 kg/s, secondary 1.6 kg/s, air 3.3 kg/s)

2.3 Primary Water Flow Rate Variation

A large series of tests were conducted to investigate the dependence of the thermal efficiency on the primary water flow rate. These tests were conducted in three groups. For each group the air flow rate and secondary water flow rate was maintained constant while the primary water flow rate was varied. For each group therefore there is a constant air to secondary water flow rate ratio (ASWR). A summary of the results of these tests is shown in Figure 3. The results show that the dependence of the secondary thermal efficiency (STE) on the primary water flow rate is generally not particularly strong.

Table 2: Variation in thermal efficiency with load (flow rates: primary 2.3kg/s, secondary 1.6 kg/s, air 3.3 kg/s)

Cooling load kW	Adiabatic saturation temp. ˚C	Primary thermal efficiency	Secondary thermal efficiency
24	8.9	0.52	0.50
24	9.2	0.53	0.51
20	8.5	0.56	0.52
20	9.1	0.52	0.51
15	8.7	0.56	0.52
15	9.3	0.50	0.50
15	9.8	0.50	0.50
9	9.2	0.56	0.51

It is strongest at the low ASWR of 1.9 and weakest at the high ASWR of 5.5, with dependence generally falling as the ASWR increases. While the primary water flow rate has a minimal impact on the STE it has a considerable impact on the energy performance of the process as measured by the coefficient of performance (COP) achieved. The energy performance of the test rig has been described elsewhere (Costelloe and Finn, 2003b). As, in general, the evaporative cooling system should operate at COPs above those achievable with refrigeration, this limits primary water flow to a maximum of 1.4 kg/s. Hence in the series of tests, conducted to investigate the impact of the air flow rate and secondary water flow rate variation, the primary water flow rate was maintained at 1.4 kg/s.

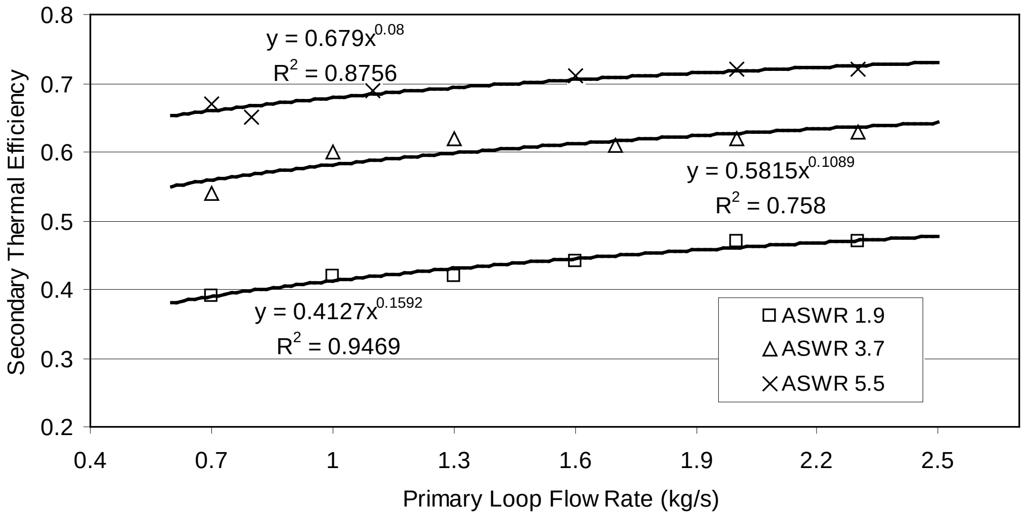

Figure 3: Variation in secondary thermal efficiency with primary loop flow rate (load 20kW, AST 10.4 ˚C +/- 0.8 K)

2.4 Air and Secondary Water Flow Variation

Figure 4 shows the impact of air flow rate on the STE for a series of 4 secondary water flow rates (SWFR) - reducing in four equal steps of 25%. It is seen that the impact of both of these variables is highly significant. These are also the two variables which are controlled in an actual chilled ceiling installation. The room cooling load is typically controlled by an energy efficient two port valve (ASHRAE, 2000), which results in a variable secondary water flow rate at the heat exchanger. Cooling tower air flow rate can also be efficiently controlled by using a fan motor inverter to maintain a constant secondary supply water temperature, as ambient AST varies (Costelloe and Finn 2003b). However it is seen that the highest levels of STE (for the test rig 76%) are obtained when the ambient AST is high and the room load is low, a combination which is infrequent in practice, in narrow plan buildings. It is also seen from the results that approximately the same efficiency is obtained when both the air flow rate and secondary water flow rate are maximum (47%) and when both are minimum (46%). This indicates that a control strategy, such as described above, maintains a near constant efficiency as air flow rate and water flow rate is reduced in tandem, when ambient AST falls in the off peak cooling season.

2.5 Ambient AST Variation

To examine this aspect a large series of tests were conducted with ambient AST varying from 2-18ºC. For these tests the rig was maintained at maximum air and water flow rate capacity. The results of these tests which are summarised in Figure 5 indicate that both the PTE and STE are significantly affected by the ambient AST, with PTE being marginally more affected than STE. The STE increases at a rate of approximately 1.3% per degree rise in ambient AST across the 16ºC range of the tests. This is comparable with but larger than the variation of 8% in a different range of 10–20ºC WBT reported for the closed tower (Facao and Oliveira, 2000). Hence these results demonstrate that efficiency is inherently greater when the external component of the cooling load is higher in Summer. This strengthens the case for water-side evaporative cooling in buildings.

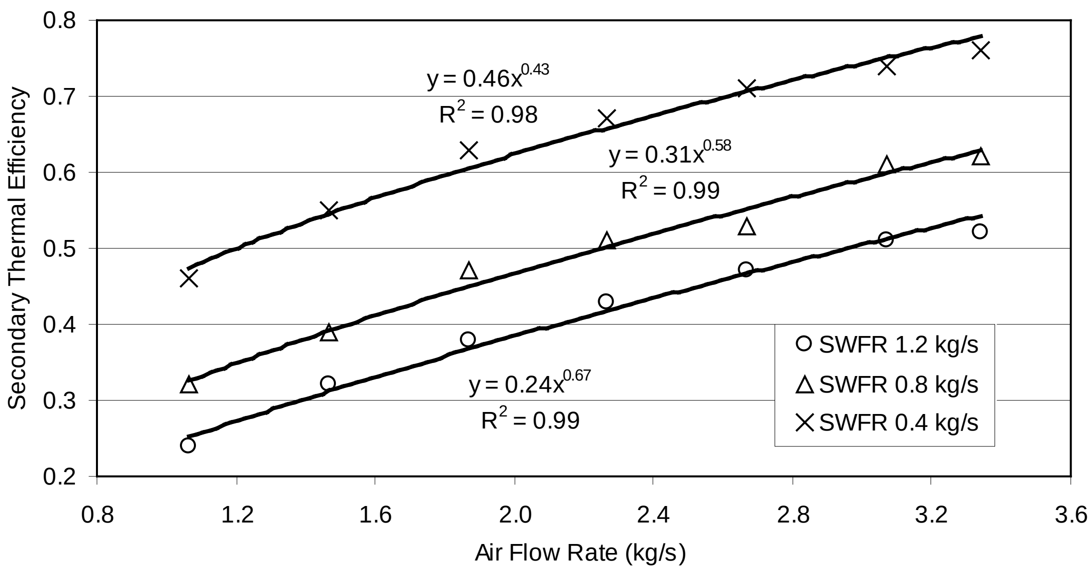

Figure 4.: Variation in secondary thermal efficiency with air and secondary water loop flow rate (load 20kw, AST 8.4 C +/- 0.9 K)

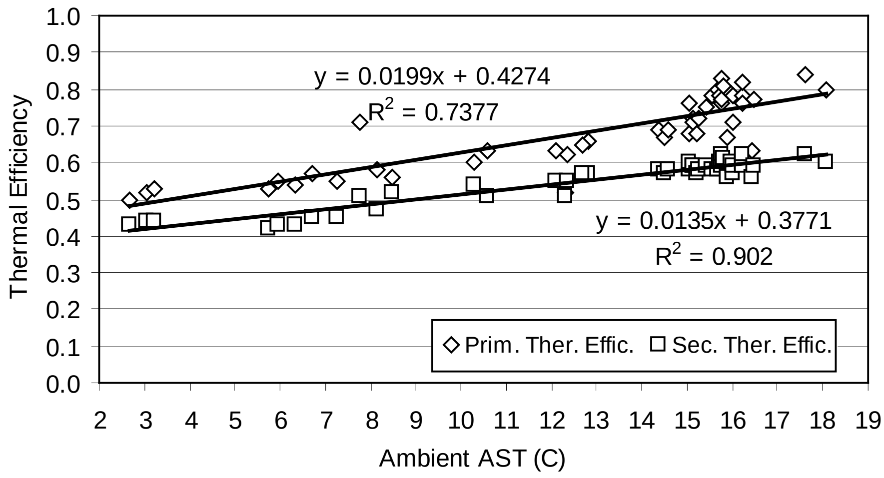

Figure 5: Variation in thermal efficiency with annual range of AST in Dublin (load 20kW, flow rates: primary 2.3 kg/s, secondary 1.6 kg/s, air 3.3 kg/s)

3. CONCLUSIONS

The thermal efficiency of an evaporative cooling test rig has been investigated under conditions of varying load, air and water loop flow rates and ambient adiabatic saturation temperature (AST). A range of secondary thermal efficiency (STE) of 0.18-0.76 has been found with primary thermal efficiency (PTE) in the range 0.26-0.82. The following specific conclusions can be drawn:

- Primary and secondary water return temperatures and the imposed cooling load have no significant effect on the efficiency achieved (Fig. 2, Table 2).
- Primary water flow rate has minimal impact on the STE achieved. The impact is stronger but not dominant at lower air to secondary water flow rate ratios (Fig. 3).
- Air flow rate has considerable impact on the STE achieved with efficiency decreasing with reducing air flow rate. However STE increases as the secondary water flow rate is reduced. A control strategy, therefore, which uses a variable volume air and water flow rate to respond to varying ambient and load conditions tends to maintain a near constant thermal efficiency (Fig. 4).
- Results indicate that the ambient AST has a substantial impact on the PTE and a marginally lower impact on the STE. The STE increases at a rate of approximately 1.3% per degree rise in ambient AST across the 16^{o}C range of the tests (Fig. 5).

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