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Energy Performance of Indirect Evaporative Cooling in Chilled Ceiling Applications in Maritime Temperate Climates

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ENERGY PERFORMANCE OF INDIRECT EVAPORATIVE COOLING IN CHILLED CEILING APPLICATIONS IN MARITIME TEMPERATE CLIMATES

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ABSTRACT

Evaporative cooling has had limited application in maritime, temperate, climates due to the low levels of cooling water availability which result, when low temperature (5 to 8˚C), convection based, building cooling systems are used. However, the success of "high temperature" radiant cooling, in the form of chilled ceilings, has prompted a review of evaporative cooling in maritime, temperate, conditions. In order to maximize evaporative cooling availability, however, in this application, it is necessary to achieve low wet bulb temperature approach conditions, at viable levels of primary energy consumption. This paper presents the results of experimental research into the energy performance of an evaporative cooling test rig, designed to maximize cooling water availability at the temperatures required for chilled ceilings (14 to 18°C). Results are compared with typical energy efficiencies of conventional, refrigeration based, building cooling systems. A significant potential for improved energy performance, is shown.

INTRODUCTION

Concern with environmental emissions and global energy consumption has lead to the development of low energy cooling technologies, among them evaporative cooling, as an alternative to the use of refrigeration in buildings for general sensible cooling (Tassou, 1998). Cooling of buildings by means of water evaporation has traditionally been seen as appropriate mainly in hot and arid climates (Watt, 1986). The technique has had very limited application in maritime and temperate climates where the ambient relative humidity is often high. However, the recent success of chilled ceilings, as a effective means of cooling buildings has prompted a review of the evaporative cooling technique as an effective supplement to or even a substitute for, refrigeration based sensible cooling.

While the evaporative cooling technique can be exploited with any water based building cooling system, such as the commonly used fan coil system, the technique is particularly advantageous when a chilled ceiling system is used, due to the higher cooling water temperatures which are employed. Chilled ceiling systems generally require cooling water at 14 to 18˚C. A typical design arrangement would be a supply temperature of 15˚C with a 3˚C rise, returning at 18˚C. A research project (CIBSE, 1998), which investigated thermal comfort conditions maintained in a test room, with internal heat gains of $60W/m^2$ and served by a chilled ceiling, established that cooling water, supplied to the ceiling, at a temperature as high as 18°C could maintain a satisfactory comfort index (a maximum dry resultant temperature of 25.2°C and a maximum predicted mean vote of +0.8) in the space. This maximum condition was measured between 16:00 and 17:00 hours with lower conditions at other times. This was considered acceptable, as it implies a predicted percentage dissatisfied of no greater than 20% for short periods. A temperature of 18°C would seem to be the maximum water temperature which can be used with sensible cooling systems, of this type, although this largely depends on the comfort criteria considered acceptable for each project and whether an adaptive approach to comfort conditions could be employed. Two recent research papers state that chilled ceiling radiant panels can operate with a supply water temperature as high as 18 to 20°C (Gan *et al*., 2000) and (Facao and Oliveira, 2000). However, a supply water temperature of 20˚C would involve an adaptive approach to comfort conditions.

Figure 1, which shows a simplified schematic of a typical indirect evaporative cooling system, indicates the relevant design parameters. An important performance parameter is the primary approach temperature (PAT) which is equal to T_{pf} - T_{wb}. This aspect is complicated by the requirement, in contemporary applications, to separate the tower water circuit from the building cooling circuit with a heat exchanger. Physical and chemical tower side contaminants are therefore isolated from the secondary side, which can now be closed and pressurised. This action greatly reduces maintenance and water treatment in the extensive secondary circuit, allows different static pressures to be used in each stream, and gives more freedom in siting the tower. Hence the significant performance parameter becomes the secondary approach temperature (SAT) which is equal to T_{sf} - T_{wb}. The importance of the SAT, for evaporative cooling systems, is demonstrated in Figure 2, which shows the impact of reducing the SAT on the percentage total annual availability of cooling water for the possible range of cooling water temperatures, generated by evaporation, in Dublin (Costelloe and Finn, 2003). Examining this figure it can be seen, for example, that at a secondary flow ($T_{sf,}$ see Figure 1) cooling water temperature of 16°C, the total annual availability increases from 50% at 8 K SAT to 88% at 3 K SAT. While the relationship between SAT and percentage annual availability differs for each location the impact of reducing the SAT on availability is generally considerable, in European locations. In Milan, for example, a similar reduction in SAT, at 16˚C cooling water temperature, will increase total availability from 37% to 57% (Costelloe and Finn, 2003). Similar charts can be produced for other locations.

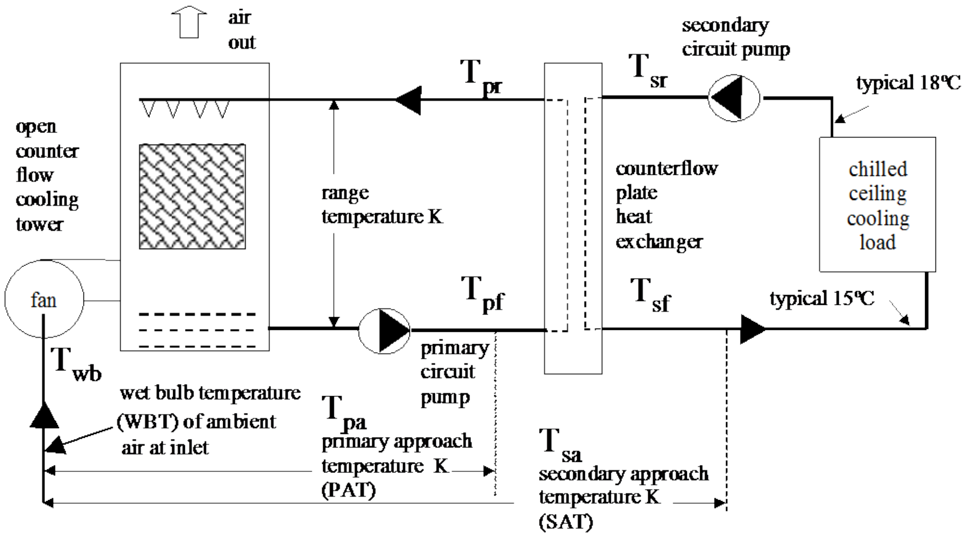

Figure 1: Simplified schematic of indirect evaporative cooling system with an open tower and chilled ceiling load.

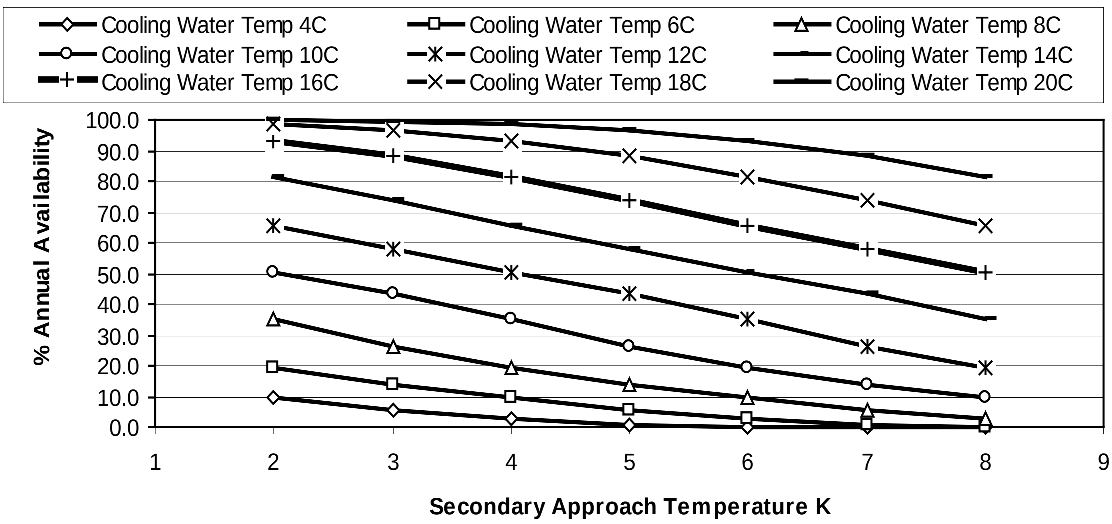

Figure 2: Impact of secondary approach temperature on percentage total annual availability of cooling water, in Dublin, for a range of cooling water temperatures (°C), (Costelloe and Finn, 2003).

Cooling water at 18˚C and 3K SAT is statistically unavailable for just 320 hours out of a possible 8760 hours per annum, in Dublin (Costelloe and Finn, 2003). The extent of availability, therefore, is so wide that it comes very close to providing year round cooling. Table 1, in ascending order of the wet bulb temperature (WBT), shows the 1% external design conditions for some cities in north western Europe with temperate and maritime climates (ASHRAE 1997). All these cities have similar external design WBTs, but different dry bulb temperatures (DBT) and would, therefore, have approximately similar levels of cooling water availability, in summer, at similar SATs.

Table 1: One percent design conditions for European cities with temperate, maritime climates (ASHRAE, 1997)

City	1% DBT ˚C	Mean coincident WBT˚C (coincident with DBT)	1% WBT ˚C	1% humidity ratio g/kg	1% dew point ˚C
Glasgow	21.6	16.0	16.7	10.7	15.0
Belfast	20.7	15.9	16.8	11.1	15.5
Dublin	20.6	16.3	17.1	11.4	15.9
Copenhagen	23.2	16.4	17.4	11.0	15.5
Oslo (Fernebo)	24.8	16.4	17.4	11.2	15.8
Manchester	23.1	16.4	17.4	11.3	15.7
Helsinki	24.1	16.3	17.6	11.4	15.9
Plymouth	22.1	16.6	17.6	11.9	16.6
Stockholm (Bromma)	24.2	16.2	17.7	11.5	16.1
Tallinn	23.3	16.9	17.9	11.7	16.3

This raises the prospect that the evaporative cooling technique, either acting alone or in conjunction with other low energy cooling strategies (such as night ventilation or fabric thermal storage), can supplant rather than simply supplement the requirement for refrigeration based sensible cooling, in these locations. For this to occur the evaporative cooling system must be used in conjunction a "high temperature" building cooling system, such as a chilled ceiling, and a low approach condition must be achieved in the cooling tower and heat exchanger, particularly during the warmer months. This, in turn, requires a high air and water mass flow rate, per unit of load, in the cooling tower, which itself, has the potential to raise the energy consumption of the process. The success of this strategy, therefore, largely depends on achieving low approach conditions, in the heat rejection system, at viable levels of primary energy consumption. In order to investigate this issue an automated experimental research facility has been designed and constructed as shown in Figure 3.

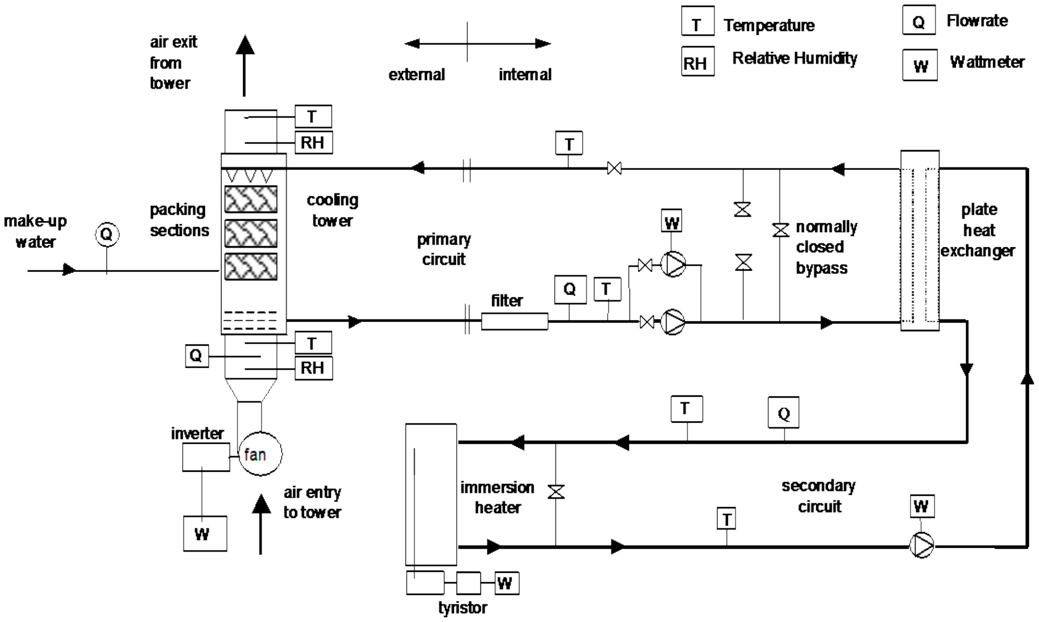

Figure 3: Schematic diagram of evaporative cooling experimental test rig with open tower

The test rig includes a prototype cooling tower, with three packing sections and a laboratory test rig with automatic data logging of key variables. The design of the open counter-flow cooling tower and the plate heat exchanger is optimized for close approach conditions. The electrical energy consumption of all power consuming equipment is individually measured and recorded. Modulated speed control of the cooling tower fan is achieved by inverter control of the fan motor. The cooling load is provided by an in-line electric immersion heater with modulated thyristor control. This enables the imposed cooling load on the tower to be accurately measured and controlled. The design of the rig is described in detail elsewhere (Costelloe and Finn, 2000).

1 ENERGY PERFORMANCE TESTS

The energy performance of the process can be assessed in terms of the energy coefficient of performance (COP) or the cooling energy produced by the evaporation process, per unit of electrical energy input to the cooling tower fan and primary circuit pump. For the purpose of the research tests this parameter can be defined as:

$$\mathbf{COP = Q/(P_f + P_p)} \qquad (1)$$

where

Q = accumulated cooling energy effect over the test period (kWh).

P_f = accumulated energy input to cooling tower fan over the test period (kWh).

P_p = accumulated energy input to primary circuit pump over the test period (kWh).

In general, chilled ceiling systems, require cooling water at a constant design temperature through out the year. The cooling water temperature cannot be allowed to fall, if condensation on the chilled ceiling, is to be avoided. Furthermore, chilled ceilings have also been shown to have a significant degree of self regulation (Butler, 1998), with an increased cooling output as room conditions rise at a constant design cooling water supply temperature. This feature enables the energy performance to be improved at lower ambient WBT, as the cooling tower air volume flow rate can be reduced, which increases the approach condition, but maintains the required design cooling water temperature. The objective of the tests, therefore, was to measure the energy performance of the heat rejection process, not only at minimum or design approach conditions but also across a typical range of 3 to 10 K SAT. The energy performance needs to be assessed, in the first instance, at a constant full cooling load of 24 kW at various approach conditions (tests 1-5). Secondly, in order to simulate the situation in which a partial cooling load was imposed on the tower, the same series of tests were repeated at a constant partial load of 17kW, or 70% of the full load (tests 6-11). The test results are summarised in Table 2. Ambient WBT ranged from 8 to 14°C during the tests.

Table 2: Summary of results of energy performance tests on evaporative cooling test rig

1	2	3	4	5	6	7	8
Test no.	Cooling tower fan power	Accumulated cooling energy effect (load rejected)	Accumulated energy input to tower fan	Accumulated energy input to primary pump	Average PAT achieved	Average SAT achieved	Test COP
	%	kWh	kWh	kWh	K	K	
1	100	48.2	5.0	1.4	1.2	4.4	7.5
2	75	45.3	3.8	1.3	1.3	4.4	8.9
3	60	47.9	2.9	1.3	1.7	4.8	11.4
4	32	47.0	1.6	1.3	4.4	7.4	16.2
5	20	47.0	1.0	1.3	6.2	9.4	20.4
6	100	34.4	5.1	1.3	1.0	3.0	5.4
7	75	35.0	3.8	1.2	1.0	2.9	7.0
8	60	35.0	3.0	1.3	1.5	3.8	8.1
9	32	33.9	1.6	1.3	2.5	4.7	11.7
10	20	34.1	1.1	1.3	4.1	6.5	14.2
11	16	34.1	0.8	1.3	5.5	7.8	16.2

2 DISCUSSION OF RESULTS

Figure 4 shows the dependence of the SAT on the cooling tower fan power. The rising SAT (and hence the reduction in the potential for cooling water generation) which results from reducing the fan power is clearly demonstrated. For both loads, the initial 40% reduction, results in a small 1 K rise in SAT. Hence significant energy benefits can be gained, initially, for small reductions in cooling potential. However, subsequent reductions in power input result in a steep rise in the approach condition and hence a significant decline in cooling availability.

Figure 5 expresses the relationship between the measured fan power input and the test COP, as defined in equation 1. As shown also in Table 3 the possible COP ranged from 5.4 to 20.4. These values are comparable with those reported by Hasan and Siren (2002), in a computational modeling study on closed wet towers for which a possible range of COP of 3 to 20 was indicated. For the 24 kW full load the COP rises from a minimum of 7.5 at a SAT of 4.4 K to 20.4 at an approach of 9.4 K. This performance can be compared with the COP levels which can be achieved with standard vapour compression systems, which range from 2.8 for small air cooled screw machines to the very best values of 7.0 which are reported for large water cooled centrifugal machines, operating at full load and producing chilled water at conventional temperatures of 5 to 8°C, as quoted in a review by Davis *et al*., (1999). In this review the COP level indicated is based on the efficiency of converting input energy into output cooling, within the machine; it does not take account of energy used outside the chiller, such as in pumps and cooling tower fans, in water cooled machines. While some modern vapour compression machines (particularly water cooled screw and centrifugal) can display a significant improvement in energy efficiency, at part load, the 17 kW part load maximum COP of the evaporative cooling process, at 16.2 is three times the initial COP at 5.4 (for a rise of 4.8 K in SAT) and is at least twice the best part load COP, which can be achieved, with current vapour compression systems. The significantly smaller fan power inputs required under higher approach partial load conditions makes possible the achievement high levels of COP, at lower loads in the non-summer months, when inverter control of the cooling tower fan is employed. The results measured are also comparable with the maximum COP of 16.6, reported by Tassou (1998) and achieved with wet bulb temperatures below 19˚C for supply air cooling using a combination of indirect air cooling followed by direct evaporative adiabatic cooling, in an air stream. This analysis demonstrates, therefore, that even with the inclusion of the primary water pump power in the COP assessment, favourable levels of COP can be achieved, particularly in the off-peak cooling season. This is significant, as low approach evaporative cooling requires a lower temperature difference between tower water inlet and outlet (the range temperature) and hence primary circuit pumps are relatively large for the load rejected.

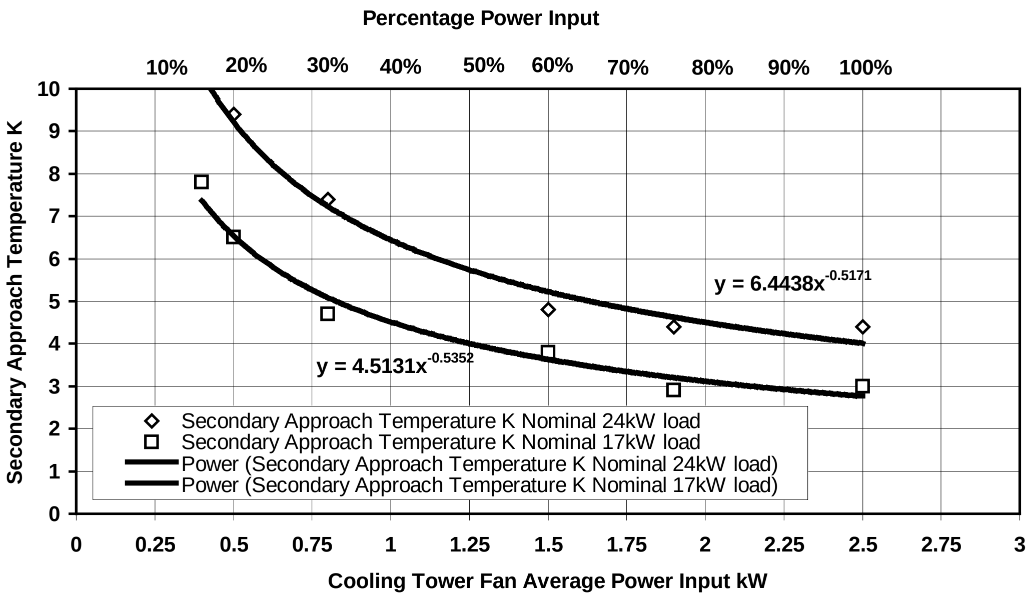

Figure 4: Dependence of the secondary approach temperature on fan power input. Measured data points, associated power law regression lines and regression line equations are shown.

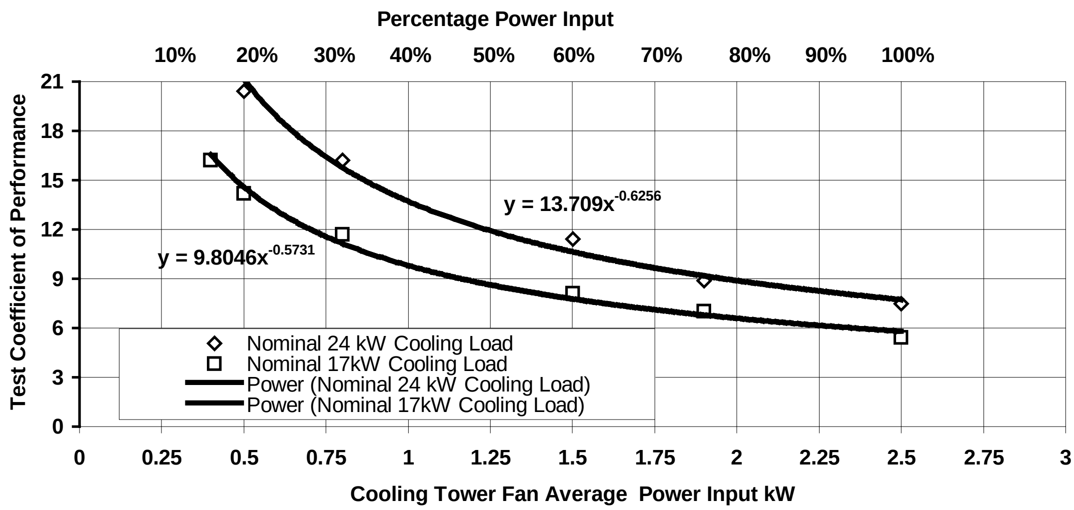

Figure 5: Relationship between fan power input and COP for the test rig. Measured data points, associated power law regression lines and regression line equations are shown.

3 ASSESSMENT OF RANGE OF ANNUAL ENERGY PERFORMANCE

Knowing the annual statistical frequency of occurrence of each WBT, (Costelloe and Finn, 2003) it is possible to determine an annualised COP for a specific cooling water temperature for each location and hence the range of the annual energy performance can be assessed. Hence, as shown in Table 3 it is possible to determine (Costelloe and Finn, 2001) the required series of SAT and their associated availability through out the year, for any given cooling water temperature. Knowing the relationship between the fan power input and the SAT achieved, for each load, from the regression line equation fitted to the experimental data (see Figure 4), the fan power required at each SAT can be predicted. For example, for the 17kw load, the correlation between the SAT and the fan power input is given by the power law regression equation:

$$\mathbf{T_{sa} = 4.5131\ W_f^{\,-0.5352}} \qquad (2)$$

where

T_{sa} = secondary approach temperature (K) and W_f = fan power input (kW).

Alternatively, where W_f is required from a known T_{sa}, the rearranged form of this equation can be used as follows:

$$\mathbf{W_f = 10^{-[(\log(Tsa/4.5131))/0.5352]}} \qquad (3)$$

These correlations define the relationship between column 5 and 6 of Table 3. Table 3 shows an analysis for a 15°C supply water temperature at a constant 17 kW cooling load and shows that in that portion of a full year (7137 hours), for which it is, statistically, feasible to generate 15°C cooling water in Dublin, on the basis of the test results, a primary energy consumption of 9,130 kWh is required to reject a total of 121,329 kWh. This results in an effective annual COP of 13.3, using the definition of COP stated in equation 1. On the same basis, cooling water generated at 18°C, at a constant load of 24kW has an effective annual COP 18.7.

These effective annual COPs would apply, however, only in a building with a constant steady year round, sensible, cooling load, such as occurs in data processing centres. In most buildings, subject to seasonal and diurnal variations in load the annualised COP would be less than these values. The use of this assessment, therefore, is to define the range and limits of the annual energy performance, at a particular design cooling water temperature, when a variable approach temperature control strategy is employed, with an inverter controlled cooling tower fan.

Table 3: Computation of annual energy performance for a constant 15°C cooling water generation and 17kW load.

1	2	3	4	5	6	7	8	9	10
Ambient WBT (Dublin)	Number of annual hours less than or equal to WBT	Number of annual hours at WBT range of WBT-0.5C	% annual hours less than or equal to WBT	SAT required for 15° C cooling water	Tower fan power required at this SAT	Primary pump power required at this SAT	Total primary energy fan & pump	Cooling energy effect in this number hours	Effective COP in this number cooling hours
°C	hours	hours	%	K	kW	kW	kWh	kWh	
7 or less	3778	3778	43.1	8.0	0.34	0.65	3752	64226	17.1
7.5	4103	325	46.8	7.5	0.39	0.65	337	5525	16.4
8.0	4414	311	50.4	7.0	0.44	0.65	339	5287	15.6
8.5	4710	296	53.8	6.5	0.51	0.65	342	5032	14.7
9.0	5058	348	57.7	6.0	0.59	0.65	431	5916	13.7
9.5	5446	388	62.2	5.5	0.69	0.65	520	6596	12.7
10.0	5760	314	65.8	5.0	0.83	0.65	463	5338	11.5
10.5	6115	355	69.8	4.5	1.01	0.65	588	6035	10.3
11.0	6461	346	73.8	4.0	1.25	0.65	658	5882	8.9
11.5	6816	355	77.8	3.5	1.61	0.65	802	6035	7.5
12.0	7137	321	81.5	3.0	2.14	0.65	897	5457	6.1
						Total	9130	121329	
						Effective annual COP			13.3

CONCLUSIONS

The results of experimental research into the energy performance of cooling water generation by indirect evaporative means, at temperatures suitable for chilled ceiling applications, under low and variable approach conditions, in an open cooling tower test rig have been presented and discussed. At low approach conditions the research indicates that the energy performance is still significantly better than that of the typical vapour compression, air cooled alternative but, approaches the COP levels which are reported for large high efficiency, water cooled, vapour compression plant. The following specific conclusions can be drawn:

At a full test rig load of 24 kW, and with a secondary approach temperature ranging from 4.4 to 4.8 K, a COP ranging from 7.5 to 11.4 was achieved, with primary circuit pump energy included in the COP calculation. These COP values are above the very best values of 7.0 which are reported for large vapour compression water cooled centrifugal machines and are considerably better than the standard reciprocating air cooled machine for which COP values of the order of 4.0 are typical, at these conditions. At these approach conditions the total annual availability of cooling water at 15°C is approximately 80% in Dublin and 72% in London.

At a partial load of 17 kW and with a SAT ranging from 2.9 to 3.8 K, a COP ranging from 5.4 to 8.1 was achieved, when primary pump energy is included in the COP calculation. Hence at very low approach and partial load conditions, the COP levels are comparable with the values reported for modern large high efficiency water cooled vapour compression plants. The COP achieved in this work are also comparable with those reported recently in a computational modelling study on closed wet towers, in this application (see discussion of results).

When advantage is taken of a falling ambient WBT, while generating a constant temperature cooling water, through out the year, the analysis of the results indicate that considerable reductions can be achieved in the annual energy consumption. For a constant 18°C cooling water generation, in Dublin, at a steady load of 24 kW, the annualised (over the 7949 hours available) COP is 18.8 and for a constant 15°C cooling water generation, at a steady load of 17 kW, the annualised (over the 7137 hours available) COP is 13.3. These values indicate the limits of the process, rather than typical values, which could be achieved. To achieve high levels of annual COP it is, therefore, necessary to incorporate accurate control of the cooling tower fan speed, in a manner which automatically tracks the ambient enthalpy condition. Similar and interlocked control of the primary pump is also desirable. Further research is required to develop a control optimisation analysis for fan and primary pump control strategy to minimise total energy input.

The high levels of COP which can be achieved in the off peak cooling season implies that the chilled ceiling system, supplied by evaporatively generated cooling water, is particularly suited to applications with long cooling seasons and steady sensible internal cooling loads. Such load profiles increasingly occur in the current generation of commercial buildings, constructed with a high performance envelope.

NOMENCLATURE

			Subscripts	
T_{pf}	primary loop flow temperature	˚C	pf	primary flow
T_{pr}	primary loop return temperature	˚C	pr	primary return
T_{sf}	secondary loop flow temperature	˚C	sf	secondary flow
T_{sr}	secondary loop return temperature	˚C	sr	secondary return
T_{wb}	ambient wet bulb temperature (WBT)	˚C	wb	wet bulb
T_{pa}	primary approach temperature (PAT)	K	pa	primary approach
T_{sa}	secondary approach temperature (SAT)	K	sa	secondary approach
Q	accumulated cooling energy in test period	kWh		
P_f	accumulated fan energy input in test period	kWh		
P_p	accumulated pump energy input in test period	kWh	f	fan
W_f	fan power input	KW	p	pump

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ABSTRAIT

Evaporative refroidissant a eu l'application limitée dans les climats maritimes, modérés, grâce aux les niveaux bas de disponibilité d'eau refroidissent qui résulte, quand les systemes de refroidissement pour les batiments, qui a base sur la convection et température basse (5 à 8^{o}C), sont utilisés. Cependant, le succès " d'haute température" rayonnante fraîche, sous forme d'a refroidi des plafonds, a incité une revue de evaporative frais dans les conditions maritimes et modérées. Dans l'ordre pour maximiser la disponibilité de evaporative refroidissant, cependant, dans cette application, c'est nécessaire d'atteindre les conditions d'approche de température d'ampoule mouillées basses, aux niveaux viables de consommation d'énergie primaire. Ce papier présente les résultats de recherche expérimentale dans l'exécution d'énergie d'un evaporative l'équipement de test refroidissant, a conçu pour maximiser la disponibilité d'eau refroidissant aux températures exigées pour les plafonds refroidis (14 à 18^{o}C). Les résultats sont comparé aux efficacités d'énergie typiques de conventionnel, refroidissement basees réfrigération systèmes pour les batiments. Un significatif potentiel pour l'exécution améliorée d'énergie, est montré.