Improving Cooling Tower Performance for Sustainable Refrigeration

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SUMMARY

Recent Health and Safety scares have made cooling towers less popular for heat rejection in cooling systems. This is unfortunate, as modern packings offer less scope for biological contamination and allow greater heat loads per unit volume than traditional designs.

This paper describes the system advantages of rejecting low grade condenser heat through latent means both in capital and energy terms. It also describes how tower packings may be evaluated for optimum behaviour in terms of heat transfer achieved and energy wasted. Experimental investigations are described which have led to new correlation techniques allowing packings to be modelled mathematically. The techniques described will allow building services engineers to analyse and design more sustainable systems for heat rejection.

INTRODUCTION

Sustainable refrigeration may be defined as that which is: adequate in cooling power to meet its duty, economic in balancing capital and operating costs through discounted cash flow, technically efficient in its Co-efficient of Performance and mechanical safety and reliability and environmentally acceptable in its various impacts. These impacts include ozone depletion, greenhouse effect contribution, materials use and recycling. The vital benefits of refrigeration should also be borne in mind, not only in improving comfort through air conditioning but also in the food chain where agricultural efficiency is vastly improved and in medicine where vaccines are delivered by it, in the context of WHO infant immunisation programs.

The process of refrigeration, pictured in the reverse Carnot cycle requires the cyclical absorption of heat at low temperature and rejection of heat at a higher temperature. Work must be done on the system, according to the Second Law of Thermodynamics. The Efficiency is expressed as the Co-efficient of Performance, (CoP), the ratio of cooling delivered to power consumed. This theoretical picture is modified by the use of real systems such as vapour compression or absorption. Here real fluids, or refrigerants, are used, with finite latent heats, heat transfer co-efficients, flow pressure drops and irreversibilities in compression and expansion. Real processes include resistances to flow and heat transfer, which increase the difference in temperature between source and sink and also increase the power required in compression. The process of rejecting heat to the sink is the subject of this paper.

The ability to reject waste heat from a system is of fundamental importance as the upper, rejection temperature, T_2 has a great impact on the power required for a given cooling duty. If a high resistance to heat loss is encountered at this stage, then the temperature difference and therefore the rejection temperature must rise proportionately, further reducing efficiency and increasing energy consumption. The rejection (or condensing) temperature T_2, must therefore be as near to the absorbing (or evaporating) temperature T_1, as technically possible. Because of the non-linearity of the division process, the CoP increases dramatically as the temperatures converge (Fig 1).

Evaporating Temperature (K)	278	278	278	278	278
Condensing Temperature (K)	278	288	298	338	378
Co-efficient of Performance	∞	27.8	13.9	4.63	2.78

Fig 1 Table of CoP as a Function of Evaporating and Condensing Temperatures

Actual CoP's achieved in practice are typically only about half of the theoretical values quoted, because of irreversibilities.

The limiting or lowest possible value of T_2 is clearly that of the environment or sink temperature T_a to which heat is rejected. Examples of this are surrounding air dry bulb temperature for dry coolers, river or sea water temperature for riparian or coastal sites, air wet bulb temperature for cooling towers, and ground water temperatures for high water table sites. The sink temperature is usually a worst case value, such as a summer time maximum. An additional factor to be taken into account is the thermal resistance, $\Delta T/Q$, where $\Delta T = T_2 - T_a$. The higher this resistance, e.g. through undersized, cheap condensers, the higher must rise both T_2 and the consequent compressor power**.**

PREVIOUS WORK

James et al (1993) reviewed evaporative cooling for refrigeration, comparing alternative forms of heat rejection. They reported that for an ambient state of 28°C DB, 21°C WB, typical refrigerant condensing temperatures would be, 43°C for a Dry Cooler, 38°C for a closed cooling tower, 36°C for an open cooling tower and 33°C for an evaporative condenser. A study was carried out which showed that evaporative condensers required about 25% of the air flow (and hence fan power, the pressure drop being similar) of dry coolers. Compressor power would be reduced by 23% and size by 7%, even in the least advantageous case.

Butler and Alamdari (1998) reviewed the opportunities for the use of "free" cooling (i.e. that avoiding refrigeration) with chilled ceilings and beams. They also reported results on tests at BRE using a mock-up room, which showed that chilled water temperatures as high as 18°C could still produce comfort conditions in the space. In this case, the three effects of solar radiation, internal gains and thermal storage of the structure were all built in to the test room. Test results showed that a maximum internal air temperature of 25.2°C occurred on a peak solar gain day. A Predicted Mean Vote (PMV) of 0.8 was calculated. This demonstrated the attractiveness of the system. It was also reported that ceiling panels could be improved if extended surface profiles (such as fins) were adopted. The limiting factor was the "Approach", or the difference in temperature of the evaporatively cooled water obtained above that of the air wet bulb temperature. This was accepted as 3 K, although other sources suggest that 1 K may be achieved.

Goshayshi (1999) and Goshayshi et al., (1999), examined the performance of modern cooling tower packings adopted by manufacturers in the last decade. The performance of these had previously remained unpublished because of commercial considerations. His study, involving an extensive program of tests of a generic range of packings showed that modern ribbed types improved mass transfer rates over those of traditional smooth packings by a factor of 1.45 to 1.83. In addition, these new packings gave improved performance for given air side pressure drops, making better use of the fan power expended.

SYSTEM PERFORMANCE

The performance of a heat transfer system depends largely on the temperatures (or enthalpies) between which it operates. Small heat exchangers using no more than natural convection will be quite satisfactory if source and sink temperatures are far apart. Conventional refrigeration plants are often designed this way to give low capital costs at the expense of energy and environment costs. A true cost comparison which an informed purchaser would make would be based on optimal life cost using discounted cash flow analysis of both first and running costs, such as that described by Tozer (1998).

In this case, the temperature limits are; the maximum allowable value within the conditioned space and the wet bulb level of the ambient air. Also, temperature differences must exist between these limiting values and those of the chilled water. The magnitude of these differences is dictated by the thermal effectiveness of the ceiling panels and the cooling tower. The specific cooling effect, Q-, (W/m^2), of ceiling panels was discussed by Dickson (1998) who developed an empirical equation ;

$$Q- = 0.2421\Delta T^2 + 7.5319\Delta T, \quad \text{for the range } 0 < \Delta T < 10 \text{ K} \qquad (1)$$

This expression accounts for both radiation and convection and implicit in this equation is the assumption that the heat transfer area is the same as the plan area of the tile. Rates within plant are enhanced by extending surfaces to include fins, corrugations and roughnesses. A treatment of the surface of the tile to give a more complex surface would increase the cooling power proportionately, provided that conduction between pipes and tiles was maintained. It is possible to double the area by adding fins having a projection 1.5 times their spacing. This would raise the cooling effect quoted at 10 K difference from 100 W/m^{2}to 200 W/m^2.

A limitation is imposed where displacement ventilation is also used. Geens (2000) reported that chilled ceilings may overturn air stratification if panel temperatures fall below the room air temperature. In practice this limit is equal to the air temperature at head height, 1.8 m above floor level, thus leaving stratified room air undisturbed in the region of the occupants.. Above head height, a large air temperature increase is noted and convective cooling is therefore much increased .

The cooling tower is the key to the system. It provides a compact low energy input device for cooling water by direct contact with ambient air. Because it involves combined heat and mass transfer, transferring both sensible and latent heat, the operation of the tower is limited by the wet bulb temperature of the air, rather than the dry bulb value. The tower may be analysed by Merkel's theory (see Goshayshi, 1999), using either the industrial Number of Transfer Unit (NTU) technique or Merkel's numerical integration.

Cooling Tower Performance Analysis

In order to discover the effect of improved packing, calculations were performed on the basis of a conventional tower, determining for a design case the heights of packing required. This same tower was then re-examined on the basis that the packing had been replaced with modern corrugated materials. The use of cooling towers for chilled ceilings requires conditions well outside of those normally encountered. The water temperatures are much lower, ranging from 14°C to above 18.5°C, compared to about 45°C for condenser cooling. This therefore involves very small driving forces and small heat transfer rates per unit area of packing. In this case it is proposed to examine the case of chilled ceiling panels operating at a mean water temperature of 21°C. For a surface co-efficient of 8 W/mK with air at 27°C, cooling of 48 W/m^2 could result.

The table below (Fig 2) examines three cases; that of conventional splash packing as described by Coulson and Richardson (1995) and the smooth and rough packings of Goshayshi et al (1999).

Packing Type	Splash	Smooth Ribbed	Rough Ribbed
Air Mass Flux, G' (kg/m^{2}s)	0.82	1.91	1.07
Water mass flux , L' (kg/m^{2}s)	0.26	1.78	1.78
Tower Co-efficient, Ka (/s)	0.24	9.2	9.5
Heat transfer rate (kW/m^2) (based on 2 K water diff.)	2.18	2.18	2.18
Air Enthalpy Change (kJ/kg)	2.66	1.14	2.04
Water Temperature drop (K)	2.0	0.3	0.3
Flow/return water temperatures (°C)	22./20.	21.15/20.85	21.15/20.85
Height of packing (Merkel) (m)	3.38	0.06	0.06
Ambient air state (all cases)	27°C DB, 19.8° C WB, 56.22 kJ/kg		

Fig 2. Comparison of Packed Tower Performances

The small differences, which the above table reveals, show how difficult it is to be precise in calculating performance at these extremes. However the important conclusions which may be drawn are that new packings are some 56 times more effective than traditional splash types and that very close approaches may be possible, leading to near isenthalpic conditions. Further, the cooling delivered by such high output towers will be very dependent upon ambient conditions. The evaporatively cooled water will be produced close to but above the wet bulb temperature of the ambient air. It will therefore be very unlikely to pose a condensation problem with a chilled ceiling.

The advantage of such a system is that cooling is delivered without refrigeration, the costs being those of the tower, its maintenance and fan and pump running costs. The low temperatures at which the towers operate would reduce the possibility of Legionella infestation and therefore the costs of complying with Health and Safety Legislation. Clearly, there is much scope for additional research in the laboratory and trial applications in the field to develop the ideas presented for commercial application.

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