RAISING EVAPORATIVE COOLING POTENTIALS USING COMBINED COOLED CEILING AND HIGH TEMPERATURE COOLING STORAGE

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ABSTRACT

Evaporative cooling is able to generate the cooling medium at a temperature approaching to the ambient wet bulb temperature. In this paper, a low-energy air conditioning strategy is proposed, which is a combination of cooled ceiling (CC), microencapsulated phase change material (MPCM) slurry storage and evaporative cooling technologies. The assessment of evaporative cooling availability and utilization is done for five representative climatic cities, including Hong Kong, Shanghai, Beijing, Lanzhou and Urumqi in China, and the energy saving potential of the proposed air conditioning system is analyzed by using a well validated building simulation code. The results indicate that the new system offers energy saving potential up to 80% under northwestern Chinese climate and up to 10% under southeastern Chinese climate. The optimal sizing method of the slurry storage tank is also proposed based on the slurry cooling storage behaviors and cooling demand variations of the ceiling panels.

INTRODUCTION

In the past decades, CC systems are increasingly used for cooling in the commercial office buildings. A displacement ventilation system is usually used to provide the fresh air at a minimum ventilation rate for a room when a CC system is applied for the cooling load removal. Since a CC system operates at relatively high cooling water temperature, usually around 14-20°C [Feustel and Stetiu 1995], depending on the areas of the ceiling panels, it offers the opportunities to use the cooling water generated by evaporative free cooling [Niu and v.d. Kooi 1995] and reduces the energy requirement of the vapor compression refrigeration system. By integrating the cooling tower model into a dynamic thermal analysis program, energy saving potentials of combined system of CC and evaporative cooling tower were evaluated for different climates by different researchers [Niu and v.d. Kooi 1995, Vangtook and Chirarattananon 2007], and the simulation results revealed significant energy saving potentials of combining evaporative free cooling with CC system for both the European countries like the Netherlands and the Asian countries like Thailand.

In recent years, a new kind of thermal energy storage medium was proposed, in which a phase change material (PCM) was microencapsulated with plastic shells and suspended into water to form solid-liquid MPCM slurry [Yamagushi et al 1996, 1999]. The heat capacity of the slurry is then significantly increased with the presence of MPCM particles. Due to the wide temperature availability of PCM, MPCM slurry also offers the flexibility of wide range of working temperatures. The purpose of this study is to investigate the energy saving potential of a hybrid system which is a combination of CC, MPCM slurry storage and evaporative cooling technologies. The MPCM slurry is used to store the cooling energy generated by the evaporative cooling system, and which is then for the sensible heat removal from the space via a CC system. Assuming proper values of the temperature approach, which is defined as the temperature difference of ambient wet bulb temperature and the cooling water produced by the evaporative cooling system, the energy saving potentials of the hybrid system in five representative Chinese cities are compared, and this is done using an hour-by-hour simulation program ACCURCY. Finally, the optimal sizing method of the slurry storage tank is proposed.

METHODOLOGY

Properties of MPCM slurries

The original MPCM slurry was prepared via a micro-encapsulation process and consisted of microencapsulated hexadecane ($C_{16}H_{34}$) particles and the pure water. Fig. 1 shows the appearance of MPCM slurry and SEM microscopic photograph of the MPCM particles. The MPCM particle used industrial-grade hexadecane, with a melting temperature, T_m=18.1 ^{o}C and latent heating of melting, ΔH_m= 224 kJ/kg, as the core material and Amino plastics as the shell material, respectively. The core-shell ratio was controlled to be about 7:1 by weight during the preparing process. The phase transition point and latent heat of hexadecane with and without micro-encapsulation were measured by a Differential Scanning Calorimeter (Perkin Elmer DSC7) at the heating/cooling rate of 5^{o}C/min. The size distribution of micro-particles was measured by a particle characterization system (Malvern Instrument Ltd., Malvern Mastersizer 2000). The volumetric average diameter of the particles was found to be 10.2 μm.

Fig.1. Appearance of the MPCM slurry and SEM microscopic image of the MPCM particles

The heat storage and heat transfer of MPCM slurry in the storage tank are associated with the following properties: density, thermal conductivity, specific heat capacity and latent heat of the PCM, shell material, its carrier fluid and the particle concentrations. The overall properties were calculated from the weighted fraction of the individual properties and are given in Table 1. The slurry with a particle mass fraction of 0.3 is used for the simulation in this study, which is the upper band of mass fraction that still has a relatively low viscosity [Wang et al 2007].

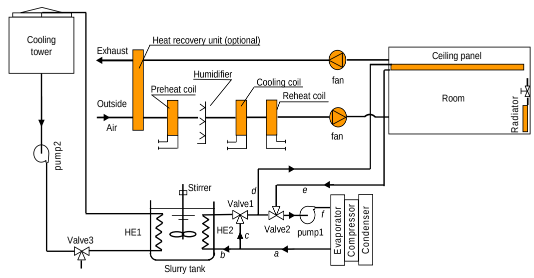

Fig.2. Schematic diagram of the hybrid system

System description

We proposed a hybrid system which is a combination of a CC system, a MPCM slurry storage tank and an evaporative cooling system. A conventional vapor compression refrigeration system is also used to provide auxiliary cooling water generation when the cooling energy stored in the MPCM slurry is not sufficient to provide the cooling for the ceiling panels. The schematic diagram of the hybrid system is given in Fig. 2. The water-type ceiling panels are installed to extract the sensible heating load in the room. The cooling water circulated in the ceiling panels is generated by either the MPCM slurry storage tank or the conventional vapor compression refrigeration. In combination, a conventional air handling unit (AHU) provides cooled and dehumidified minimum air for ventilation purposes, avoiding possible condensation on the ceiling panel surface at the same time.

Modeling the hybrid system

Thermal performance of the hybrid system is evaluated by a previously validated building energy simulation code ACCURACY [Niu and v.d. Kooi 1995, Chen and v.d. Kooi 1988]. It is based upon the room-energy-balance method, and the ceiling panels are treated as individual surfaces which exchange heat convectively with the room air and radiantly with other surfaces. Heat conduction within the ceiling panels is treated in a one dimensional way using a three-nodal-point model. The program calculates not only the cooling load, but also the required supply water for different panel installation areas. Adopting such building dynamic simulation model, the hour-by-hour data of heating/cooling loads, room air temperature and other components of the room with ceiling panels running with water are calculated [Niu and v.d. Kooi 1994, 1997].

The charging and discharging models are based on a simple simulation of the heat exchangers immersed in the slurry tank with the fixed heat transfer capability values and can be calculated by the log mean temperature difference (LMTD) method [Incropera and Dewitt 1996].

Charging equation

$$Q^*_{charging} = (UA)_{HE1} \frac{\left(T_{water1,in} - T_{water1,out}\right)}{\ln\left(\frac{T_m - T_{water1,out}}{T_m - T_{water1,in}}\right)} \quad (1)$$

$$Q^*_{charging} = \dot{m}_{water1} C_{p,w} \left(T_{water1,out} - T_{water1,in}\right) \quad (2)$$

where U_{HE1} is the overall heat transfer coefficient between the cooling water and the MPCM slurry of the heat exchanger 1, A_{HE1} the total heat transfer area of the heat exchanger 1. $T_{water1,out}$ outlet cooling water temperature of the cooling tower, $T_{water1,in}$ inlet cooling water temperature the cooling tower, $\dot{m}_{water1}$ mass flow rate of the cooling water, $C_{p,w}$ specific heat capacity of the cooling water.

Discharging equation

$$Q^*_{discharging} = (UA)_{HE2} \frac{\left(T_{water2,out} - T_{water2,in}\right)}{\ln\left(\frac{T_{water2,in} - T_m}{T_{water2,out} - T_m}\right)} \quad (3)$$

and

$$Q^*_{discharging} = \dot{m}_{water2} C_{p,w} \left(T_{water2,out} - T_{water2,in}\right) \quad (4)$$

where U_{HE2} is the overall heat transfer coefficient between the cooling water and the MPCM slurry of the heat exchanger 2, A_{HE2} the total heat transfer area of the heat exchanger 2, T_{water2}, outlet cooling water temperature of the ceiling panels, $T_{water2,in}$ inlet cooling water temperature of the ceiling panels. In this study, to simplify the calculation, it is assumed that the temperature difference between the inlet and outlet of the heat exchanger is 2 ^oC and the mass flow rate of the heat transfer fluid in the cooling tower is variable according to the cooling load of the heat exchanger 1.

Simulation considerations

To estimate the energy consumptions of the room with the hybrid system in five typical cities, a south-facing office room equipped with the same hybrid systems is simulated. The room is assumed to be situated in the intermediate story, with identical adjacent rooms, and above and below. The room is 5.1 m long, 3.6 m wide and 2.6 m high. The facing-south façade has a 2.88 m^2 double glazing area with center-of-glass U value of 1.31 W/ (m^2 ^oC). The window is equipped with Venetian blinds outside. The external wall consists of three layers: 180 mm thick concrete slab inside; 200 mm thick brick outside; 60 mm insulation in between. The combined heat transmission coefficient of the façade is 1.28 W/ (m^2 ^oC). The floor and ceiling are 320 mm thick concrete slab bases with 70 mm thick outside cement layers. The partition walls are made of 26 mm gypsum board. The south wall is the external wall and the north wall is adjacent to a corridor. It is determined that about 60 % of the ceiling is covered with cooled ceiling panels when cooled ceiling system is applied.

The building is occupied only in the working hours with a schedule from 9:00 to 18:00. In the working hours, constant internal sensible loads of 800 W of equipment and lighting are set for the simulation, or about 44 W/m^2 floor area. 37% of these loads are considered to be radiant. In the calculation of the cooling load, no air infiltration occurs. The operating hours of the ceiling panels and ventilation are also scheduled from 9:00 to 18:00.

The processed air is supplied at a rate of 100 m^3/h, which is about 2 ach in terms of the room volume. This flow rate will meet the ventilation requirement for three persons according to the ASHRAE standard(ASHRAE 2004). To keep the air dew point temperature below the ceiling panel surface temperature, the outside air is cooled and dehumidified to 15^{o}C The supply cooling water rate to the ceiling panels is about 0.119 kg/s, which results in a temperature rise of the cooling water of about 1.6 ^{o}C at the estimated peak load conditions. The required water supply temperature to the ceiling panels will be calculated based on the room energy balance method, and the minimum is set to be 18ºC. The overall control strategy of the combined system is that the air supply rate and temperature are kept constant, and further heating and cooling requirement are respectively met by a heating radiator and ceiling panels.

The reference year weather data of Hong Kong, Shanghai, Beijing, Lanzhou, Urumqi are used as outside conditions in the calculations. The energy performances of the primary equipment are modeled with certain constant indices [ASHRAE 2001]: Two COP values are used for the daytime and nighttime chiller operation, with a COP value of 4.0 when the evaporating temperature is set at 15^{o}C for MPCM charging, and 3.2 when the evaporating temperature is 7^{o}C for handling the ventilation air; the boiler efficiency is 0.75 and fan efficiency is 0.6; the pump efficiency is 0.70; the fan pressure rise is 1400 Pa for the air; the pump pressure is 0.2 bar for the heat exchanger in the slurry tank and 0.4 bar for the ceiling panels.

RESULTS AND ANALYSIS

Cooling water availability and the cooling tower APT

A crucial feature of the evaporative cooling technique is the achievement of a small temperature difference between the heat transfer fluid leaving the cooling tower and the ambient wet bulb temperature ($T_{w,b}$). Such a temperature is called the approach temperature (*APT*), which represents the capability of a cooling tower. The cooling water temperature from a cooling tower can be calculated according to the formula

$$T_{water,out} = T_{w,b} + APT \qquad (5)$$

For a given cooling tower, the *APT* is determined at the maximum cooling load condition, and the typical value is around 3 ºC. Using local meteorological reference year weather data, which consist of hourly weather data including dry bulb temperature, dew point temperature, atmospheric pressure, solar radiation and wind speed, the water temperature available from a cooling tower for any hours can be calculated from Eq. (5). The weather files were obtained based on a 1982-1997 period of record obtained from the Chinese National Climatic Data Center [US DoE]. The hourly wet bulb temperature is calculated by the subroutine for weather data conversion in ACCURACY, which applied the standard wet bulb temperature calculation equation in ASHRAE handbook [2001].

It is obvious that the influence of *APT* on the annual availability of the cooling water generation potential is significant for a specific evaporative cooling system, the lower the *APT* value, the higher the annual availability of the cooling water potential will be. However, the approach temperature for a specific evaporative cooling system is confined by the current manufacturing technique, recent research showed that it is possible to design a low approach evaporative cooling system to generate cooling water at a temperature of 3 ^{o}C above the ambient wet bulb temperature [Costelloe and Finn 2000], therefore, an *APT* of 3 ^{o}C is used for the evaporative cooling water generation in the following analysis.

Cooling energy storage of MPCM slurry by evaporative cooling

The potential cooling energy storage into an MPCM slurry tank from the evaporative cooling is determined by the cooling water temperature generated by the cooling tower, the intended storage temperature of the slurry, and the tank size and the heat transfer capability of the heat exchanger immersed in the MPCM slurry. Assuming an implicitly large tank size, the theoretical cooling storage (TCS) is limited by the overall heat transfer capacity of the heat exchanger according to the formula

$$Q_{Storage} = \sum (UA)_{HE1} \frac{\left(T_{water1,in} - T_{water1,out}\right)}{\ln\left(\frac{T_m - T_{water1,out}}{T_m - T_{water1,in}}\right)} \Delta\tau \qquad (7)$$

where Q_{stoage} is the TCS, $\Delta\tau$ is the time step during which the cooling water temperature is lower than the PCM melting temperature, typically, one hour is chosen for $\Delta\tau$, since the hourly weather data are used for the calculation. The monthly theoretical cooling energy storage can be obtained by the summation of the hourly energy storage in one month based on the hour by hour calculation. Likewise, the annual TCS of the MPCM slurry by the evaporative cooling is then calculated. By using chilled-ceiling, the required cooled-water temperature can be raised to 18 ºC and higher, which will raise the utilization of this natural cooling water.

Fig. 3 shows the annual theoretical cooling energy storage available from evaporative cooling at different $(UA)_{HE1}$ values for five typical cities in China. The annual TCS increases linearly with the $(UA)_{HE1}$ values for the five cases. At any given $(UA)_{HE1}$ value, Urumqi has the highest TCS and Hong Kong has the lowest TCS. The annual TCS is 798 kWh in Urumqi at $(UA)_{HE1}$ and 451 kWh in Hong Kong at $(UA)_{HE1}$ value 100 kW/^{o}C. Fig. 4 shows the monthly TCS by the evaporative cooling based on the $(UA)_{HE1}$ of 100 kW/^{o}C for five typical cities.

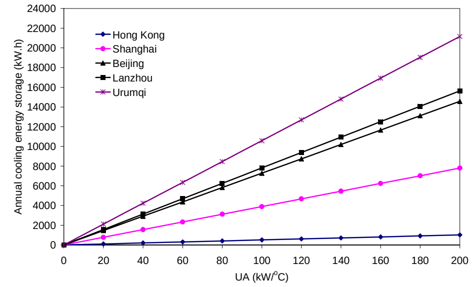

Fig.3. Annual cooling energy storage by the evaporative cooling at different UAs

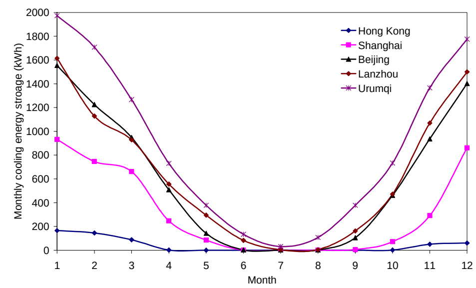

Fig.4. Monthly cooling energy storage by the evaporative cooling at UA=100 kW/^{o}C

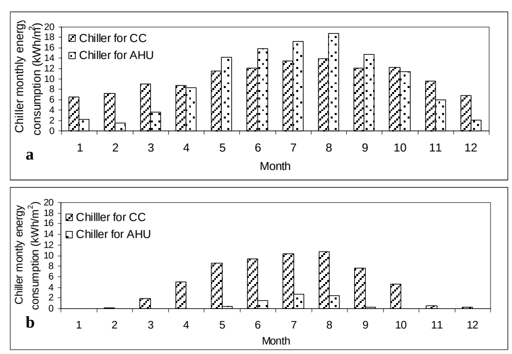

Fig.5. Monthly energy consumption of the chiller systems: (a) Hong Kong; (b) Urumqi

Monthly chiller energy consumption

The chiller energy consumption of the conventional CC without thermal energy storage can be divided into two parts. One is the chiller energy consumption for generating the cooling water for the cooling coil of air handing unit, and the other is the energy consumption for generating the cooling water for ceiling panels to remove the sensible heat. However, for the CC with thermal energy storage part of the energy consumption of the ceiling panel can be supplied by the cooling energy stored by the evaporative cooling, which provides the possibility of the energy saving.

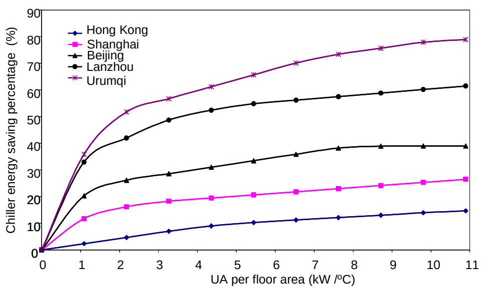

Fig.6. Chiller energy saving percentage with different heat exchanger sizes

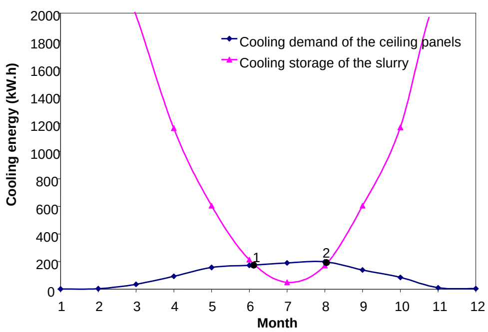

Fig.7. Comparison of the cooling energy storage and cooling demand of the ceiling panels in Urumqi

ACCURACY calculates the required heat extraction rates of the air and the ceiling panels from the room, together with inlet and outlet cooling water temperature across the ceiling panels. The chiller energy consumption for generating cooling water for ceiling panel are then calculated based on the heat extraction rates of the ceiling panels of the room. And the energy requirement for the air handing is predicted based on the psychometric process [5], which treats the hour by hour cooling load and heat extraction statistically in relation to the outdoor weather data. Fig. 5 shows the monthly chiller energy consumption for two identical office rooms equipped with conventional water-type CC without thermal energy storage located in Hong Kong and Urumqi, respectively. In the case of Hong Kong, the chiller energy consumption covers the whole year. Due to the hot and humid climate, more energy is consumed by the AHU than CC for air handing in the summer season. In Urumqi, however, except for the summer season, there is almost no need of the chiller energy consumption for air handing in other seasons.

Optimal design of the MPCM slurry storage tank

Fig. 6 shows the energy saving percentage of the room equipped with hybrid system at five typical Chinese cities at different $(UA)_{HE1}$ per floor area values based on the *APT* of 3°C. The chiller energy saving percentage increases with the increase of the $(UA)_{HE1}$ per floor area. Because of the weakening effect of the energy saving percentage at higher $(UA)_{HE1}$ value as can be observed in Figure 6, a $(UA)_{HE1}$ per floor area of 8.71 kW/(°C.m^2) is chosen for system design calculation. The optimal size of slurry storage tank is determined by the comparison of the cooling storage behaviors of the slurry tank and the variations of the monthly cooling demand of the ceiling panels. Fig. 7 shows the comparison of the monthly TCS of the MPCM slurry and cooling demand of the ceiling panel for the city of Urumqi. The crossing points 1 and 2 represent two points when the monthly TCS of the MPCM slurry just satisfies the cooling demand of the ceiling panel. The higher storage value of the two cross points can be used to size the slurry storage tank. Since the evaporative cooling could be stored on both Saturday and Sunday and delivered the cooling to ceiling panels next Monday, it is necessary to use the two-day storage cooling capacity to size the slurry storage tank. With the thermal properties of the MPCM slurry listed in Table 1, the slurry tank size with one-month cooling storage capacity is calculated. A summary of the design month and design size of the MPCM slurry tank at five typical Chinese cities is presented in Table 2.

CONCLUSIONS

A new design of a hybrid system is proposed, which is a combination of a CC system, a MPCM slurry storage tank and an evaporative cooling system. The essential feature of the system is that the thermal energy storage using MPCM slurry enables the evaporative cooling to be stored in MPCM slurry at 24-hour operation mode whenever the wet-bulb temperature reached the pre-set point. Using radiant ceiling panel, the pre-set point can be as high as 15 ^oC on the basis of an *APT* of 3^oC. The annual cooling energy storage of the MPCM slurry by the free evaporative cooling has been calculated for five typical cities, the results show that Hong Kong has the lowest cooling energy storage and Urumqi has the highest cooling energy storage, and the other three cities have the energy saving between the two cities. The chiller energy saving percentage has been obtained by using the energy simulation code ACCURACY based on hour by hour calculations. A storage tank sizing method is proposed and tested based on the monthly TCS and variations of the cooling demand of the ceiling panels, the design sizes of the slurry storage tanks for five typical cities are then calculated. The present hybrid system is recommended for climatic conditions where the weather is dry and the diurnal temperature difference is high.

ACKNOWLEDGEMENTS

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Table 1 Property of the MPCM slurry and its components

	Density	Specific heat	Thermal conductivity	Latent Heat
	$kg.m^{-3}$	$J.kg^{-1}.{}^{o}C^{-1}$	$W.m^{-1}.{}^{o}C^{-1}$	$kJ.kg^{-1}$
Hexadecane (solid) [13], [14]	780	1805	0.4	224
(liquid)	770	2221	0.21	
Urea-formaldehyde [15]	1490	1675	0.433	
Water (at 20 ^{o}C) [16]	998	4183	0.599	
MPCM particle (solid)	829	1789	0.382	196
(liquid)	819	2153	0.203	
MPCM Slurry (mass fraction)				
Φ=0.2	976	3707	0.551	39.2
Φ=0.3	933	3470	0.528	58.8

Table 2 Design sizes of the MPCM slurry storage tank in five typical cities

	Design month	Design size (m^3/m^2 floor area)
Hong Kong	February	0.0338
Shanghai	April	0.0294
Bejing	May	0.0452
Lanzhou	August	0.0474
Urumqi	August	0.0458