Feasibility study of an ice slurry-cooling coil for HVAC and R systems in a tropical building

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Abstract

The study applies the software Transient Systems Simulation Program (TRNSYS) to estimate the air conditions as well as the energy consumption of a typical library located in a tropical climate country. The simulation uses Typical Meteorological Year (TMY) weather data for Kuala Lumpur as the research site, which is the Dentistry library in University of Malaya located in Kuala Lumpur. The current HVAC and R systems of the library are found to be inherently energy-inefficient and potentially improvable with an ice slurry-cooling coil. The impact on energy consumption and space thermal comfort of a custom-built AHU with an ice slurry-cooling coil incorporated in the HVAC and R systems is simulated in comparison to the baseline system.

Based on this investigation, it is recommended that an ice slurry-cooling coil could be incorporated in tropical climate HVAC and R systems for improvement in energy management and dehumidification enhancement.

1. Introduction

In hot and humid tropical climate countries such as Malaysia, Singapore, Brunei and Indonesia, heating, ventilating and air-conditioning (HVAC) systems are required to be installed in most of the commercial buildings to produce a comfortable, artificial environment especially during mid-day where the ambient temperature is about 32 °C.

The energy demand in Malaysia has shown fast growth in the past decade, and it is predicted to reach 108732GWh in the year 2011 [1]. Furthermore, it is found that commercial sectors in Malaysia use approximately 27% of the total electricity generation in Malaysia; which 64% from the commercial sector energy consumption is utilized for air-conditioning [2]. This means the projected electricity consumption for air-conditioning in commercial sectors will reach 18789GWh in year 2011. In 1 March 2009, Tenaga National Berhad (TNB) [3], the sole electricity provider in Peninsular Malaysia, had announced a new electricity tariff. The new tariff of electricity has shown energy cost reduction in all sectors due to the decrease in electricity generation costs. The direct implication of the new tariff is a 2.7% average reduction in the electricity bill in the commercial sector. According to the TNB electricity tariff C2 – medium voltage peak/off-peak commercial tariff, electricity rate during peak-hour, which starts from 0800 to 2200 h, is RM0.288 or USD0.05268 per kWh, while the tariff for off-peak hours is RM0.177 or USD0.05268 per kWh, which is approximately 40% lower than peak hour tariff.

In order to benefit from the lower electricity charge during night time, the ice thermal energy storage technology is an essential solution. With the application of the ice thermal storage systems, part of the electricity consumption during peak hours at mid-day could be shifted to off-peak hours at night time. Ice thermal storage system that depends on its operating strategies is economically significant as it results in the reduction in the operating costs of commercial buildings. Hasnain and Alabbadi [4] stated that ice thermal storage operating strategies are often classified as either full storage or partial storage. Partial storage systems can be sized for load-levelling or demand limiting operation. Since the investment in new power plants depends on the electricity consumption during peak hour, this will reduce the maximum demand for power plants in Malaysia.

Apart from the economic aspect, humidity control in the space is an important task for the air-conditioning system in tropical climate countries. If high relative humidity supply air is supplied from the air handling unit (AHU), this will lead to fungus growth in air-tight buildings and affect the health of the occupants.

Ice slurry thermal storage system is a dynamic solution to the problems mentioned above as the thermal conductivity of the ice slurry is higher than the thermal conductivity of conventional working fluids such as chilled water. Ice slurry is capable in responding quickly to the changes in cooling load. This is because the thermal conductivity of ice slurry determined by the latent heat of ice instead of the specific heat capacity of water. By using
ice slurry-cooling coil, it can produce cooler air before it passing through the heating coil; it thus produces air with lower relative humidity. Substantial researches have been conducted on ice slurry systems and their applications. Matsumoto and Suzuki [5] pointed out that ice slurry has good fluidity. Egolf and Kauffeld [6] find that ice slurry with ice fraction below 15-20% usually demonstrates Newtonian behavior. Thus, a large amount of cold energy can be transported with less pumping work in comparison to the conventional systems.

2. Background theory

2.1. Sick building syndrome (SBS)

Reports concerning buildings with indoor air quality (IAQ) problems began to surface rapidly since early 1970s. The sick building syndrome (SBS) is originated from the energy crisis that happened in the mid-1970s. Airtight envelope systems became popular for office building designs in order to reduce heating and cooling loads of air-conditioning systems. The responsibilities of maintaining allowable indoor air quality (IAQ) for such air-tight buildings depend on mechanical ventilation systems. Subsequently, poor operation of the ventilation system will cause ineffective removal of contaminated indoor air and display symptoms of ‘sick building syndrome’ (SBS), which generally describe the effects of poor IAQ. Studies have been conducted on sick buildings and found that SBS such as fungus growth in the space [7] and high carbon dioxide concentration [8] will give serious impact to occupants’ health. Their findings showed that the main prevailing symptoms were headache, lethargy and dryness in bodily mucous.

2.2. Cool thermal storage system

Cool thermal storage is the most preferred demand side energy management technology that shifts the cooling electrical demand from peak daytime periods to off-peak night time periods. Cool storage systems remove heat from a thermal storage medium during periods of low energy cost. The stored cooling capacity is subsequently used to meet the air-conditioning cooling load requirements. Basically cool storage technology can be divided into two main categories, sensible and latent energy storage [9]. The sensible energy cool storage systems, which are also known as chilled water storage systems, uses water as the storage medium because it has the high specific heat capacity in comparison to other common substances [9]. The ice thermal energy storage systems are categorized as latent energy storage systems, as they use the latent heat of ice to store cooling energy. In these systems, ice is generated during the charging cycle in off-peak hours, which result in the conversion of water to ice. According to ASHRAE handbook [9], ice thermal storage can be categorized into five techniques, which are external melt ice on coil, internal melt ice on coil, ice harvesting, encapsulated and slurry ice.

The operating strategy defines the overall method of controlling the thermal storage in order to achieve the design intent. Thermal storage operating strategies are often classified as either full storage or partial storage. Partial storage systems can be sized for load-levelling or demand limiting operation. For full storage operating strategy, the entire chiller capacity is shifted from peak hours to off-peak hours. The cool thermal storage systems supply the whole peak-cooling load while the chiller does not operate at all during peak hours. For a partial storage strategy, the chiller capacity is significantly less than the maximum capacity. The chiller only meets part of the peak-cooling load and the remaining load is supported by the cool thermal storage systems. Partial load storage strategies can be subdivided into load-levelling and demand limiting. For load-levelling partial storage system, the chiller typically operates at or near its full capacity for 24 h so that it could minimize the required chiller capacity and storage capacity. On the other hand, chiller in demand limiting partial storage system does not operate at constant capacity in comparison to load-levelling system. Instead, it operates during peak hours, at a reduced capacity at a fixed level.

2.3. Theory and fundamentals of ice slurry

Ice storage systems had been discovered since 1990 and grow rapidly in the recent years. The main purpose of using ice slurry is to utilize the high density of energy where latent heat of fusion is used. Stamatiou and Kawaji [10] reported that ice slurry is a mixture of fine ice crystals, water and carrier fluid such as glycol, salt or alcohol. Matsumoto et al. [11] mention that in a dynamic system, “which uses ice slurry with good fluidity as a thermal storage material, can transport a large amount of cold energy with less pumping work”. Moreover, this system can respond quickly to changes in the cooling load.

Researchers such as Cecilia Hägg [12] and Stamatiou et al. [10] found that the ice slurry density $\rho_{sl}$ was expressed in Eq. (2.1) where $\rho_{ice}$ is the density of the ice, $x_{ice}$ is the weight fraction of ice, and $\rho_{cf}$ is the density of the carrier fluid evaluated at the specified ice slurry temperature and initial carrier fluid concentration.
The total latent heat $Q_{\text{lat}}$ can be stored is equal to number of ITS tanks $N_{\text{tank}}$ used times capacity $C_{\text{cap}}$ of each tank, latent heat of ice $H_f$ and the maximum ice fraction $x_{\text{max}}$ of 20\%, which is showed in Eq. (A.2).

\begin{equation}
Q_{\text{lat}} = N_{\text{tank}} \times C_{\text{cap}} \times H_f \times x_{\text{max}}
\end{equation}

During off-peak period, ITS tanks are charged with cool energy. Ice generation rate, IGR could be determined by applying Eq. (A.3).

\begin{equation}
IGR = \frac{q_{\text{evaporator}} \times CF}{H_f}
\end{equation}

Where $q_{\text{evaporator}}$ represents the evaporator capacity and $CF$ is the cooling factor which assumes to be same with the cooling factor used for chiller for ITS tanks. During operation at day time, ice slurry is pumped to the AHUs. The total heat release $q_{\text{sl}}$ from storage tank can be determined by Eq. (A.4).

\begin{equation}
q_{\text{sl}} = m_{\text{sl}} \times x_{\text{ice,b}} \times H_f
\end{equation}

Where $m_{\text{sl}}$ is the mass flow rate of ice slurry and $x_{\text{ice,b}}$ is ice fraction of ice slurry which can be defined below.

\begin{equation}
x_{\text{ice,b}} = \frac{BIM}{N_{\text{tank}} \times C_{\text{cap}}}
\end{equation}

Therefore the total heat gain by ITS tank, $q_{\text{total}}$ is shown in Eq. (A.6) below.

\begin{equation}
q_{\text{total}} = q_{\text{sl}} + q_{\text{env}}
\end{equation}

Thus the rate that the ice is “burned” (ice burn rate) can be calculated by dividing $q_{\text{total}}$ to the latent heat of water as shown in Eq. (A.7).

\begin{equation}
IBR = \frac{q_{\text{total}}}{H_f}
\end{equation}

In order to determine the quantity of ice remaining in the storage tanks at the end of each time step, Eq. (A.8) is applicable.

\begin{equation}
FIM = BIM + (IGR - IBR) \Delta t
\end{equation}

where

- $BIM =$ Beginning ice mass (kg)
- $FIM =$ Final ice mass (kg)
- $\Delta t =$ time step

Final ice fraction of ice slurry $x_{\text{icef}}$ is determined by divided FIM with total capacity as shown in Eq. (A.9).

\begin{equation}
x_{\text{icef}} = \frac{FIM}{N_{\text{tank}} \times C_{\text{cap}}}
\end{equation}

### A.2. Modeling ice slurry-cooling coil

The main assignment for a cooling coil is to reduce the temperature of an air stream. At the same time, moisture content of air will be dehumidified due to water condensation on the coil. The process of cooling air stream is very complicated that involves heat and mass transfer. Stocker and Jones [18] indicate that successive process of an elementary cooling and dehumidifying coil are combined heat and mass transfer process from the air to the wetted surfaces, conduction through the water film and metal and finally convection to the refrigerant. A simplified non-standard component of ice slurry-cooling coil (Type 305) is built for TRNSYS simulation. It is developed based on the expression given by Stocker and Jones [18] shown in Eqs. (A.10–A.26).

For a differential area of coil, two equations are obtainable for the heat transfer rate, $dq$ for an element of area, $dA$.

\begin{equation}
dq = \frac{\alpha_c dA}{C_{\text{pm}}}(h_a - h_i)
\end{equation}

where

- $\alpha_c =$ convection coefficient
- $C_{\text{pm}} =$ specific heat of air mixture
- $h_a =$ enthalpy of air
- $h_i =$ enthalpy of saturated air at wetted surface temperature

The second equation expresses the rate of heat transfer to the refrigerant

\begin{equation}
dq = \alpha_r dA_i(t_r - t_i)
\end{equation}

where

- $t_r =$ temperature of refrigerant
- $t_i =$ temperature of wetter surface
- $dA_i =$ refrigerant side area
- $\alpha_r =$ combined conductance through wetted surface, metal and refrigerant

By equating both of the equations above, a constant ratio of $R$ can be computed.

\begin{equation}
R = \frac{\alpha_r}{C_{\text{pm}} \times \alpha_c} = \frac{t_r - t_i}{h_a - h_i}
\end{equation}

The enthalpy of saturated air $h_i$ is a function of the temperature of the wetted surface $t_i$ that could be related by the cubic equation as shown below.

\begin{equation}
h_i = 9.3625 + 1.7861 t_i + 0.01135 t_i^2 + 0.00098855 t_i^3
\end{equation}

If the inlet enthalpy of air stream $h_{a,1}$ and refrigerant temperature is given as an input in the simulation, $t_{i,1}$ and $h_{i,1}$ can be computed by equating Eqs. (A.12) and (A.13). It will form a nonlinear Eq. (A.14) and subsequently solves it by applying Newton Raphson method.

\begin{equation}
\frac{t_{i,1} - t_{r}}{R} - h_{a,1} + 9.3625 + 1.7861 t_{i,1} + 0.01135 t_{i,1}^2 + 0.00098855 t_{i,1}^3 = 0
\end{equation}

Outlet air stream condition can be predicted with a given coil area, refrigerant temperature and inlet condition using a stepwise solution. Cooling coil area is divided into several sections. The first section is $A_{1-2}$. Four equations below can then be written for the first increment of areas.

\begin{equation}
m_{\text{sl}}(h_{a,1} - h_{a,2}) = q
\end{equation}

\begin{equation}
\frac{\alpha_r A_{1-2}}{C_{\text{pm}}}(h_{a,1} + h_{a,2} - h_{i,1} + h_{i,2}) = q
\end{equation}

\begin{equation}
\frac{\alpha_r A_{1-2}}{C_{\text{pm}}}(t_{i,1} + t_{i,2} - t_i) = q
\end{equation}

\begin{equation}
h_{i,2} = f(t_{i,2})
\end{equation}

The four remaining unknown variables, $q$ heat transfer rate, enthalpy of outlet air $h_{a,2}$, enthalphy of outlet saturated air $h_{s,2}$ and outlet temperature of the wetted surface $t_{i,2}$ can be found from four simultaneous Eqs. A.19–A.22. It follows the steps below to form a nonlinear equation of $t_{i,2}$ and solves it by applying Newton Raphson technique.
\[ q = f(h_{2,2}) \quad \text{(A.19)} \]

\[ q = f(h_{2,1}, h_{1,2}) \quad \text{(A.20)} \] combine to create new equation

\[ q = f(h_2) \quad \text{New Eq.} \]

\[ h_{1,2} = f(t_{1,2}) \quad \text{New Eq.} \] combine to create new equation

\[ h_{2,2} = f(t_{2,2}) \quad \text{New Eq.} \] finally create a nonlinear equation of \( t_{2,2} \)

As a result, \( t_{2,2} \) can be determined by Newton Raphson method and other 3 variables can be computed accordingly. Later, the outlet dry bulb temperature \( t_{o,2} \) can be predicted using simultaneous equation which uses the sensible heat \( q_s \) relation. For area \( A_{1-2} \), the sensible heat transfers are shown below.

\[ q_s = m_{a}C_p(t_{o,1} - t_{o,1}) \quad \text{(A.23)} \]

\[ q_s = A_{1-2}q_c \left( \frac{t_{2,1} + t_{2,2}}{2} - \frac{t_{1,1} + t_{1,2}}{2} \right) \quad \text{(A.24)} \]

Finally, assuming that the heat transfer rate of \( q \) is the same for air, steam and ice slurry side, the outlet enthalpy of ice slurry can then be determined by using equations below.

\[ q = m_{d}A_{d} \Delta h_d \quad \text{(A.25)} \]

\[ h_{sl} = x_{ice}h_{ice} + (1 - x_{ice})h_{cf} \quad \text{(A.26)} \]

References


[9] ASHRAE applications handbook (SI), thermal storage; 2003, [chapter 34].


