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

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A B S T R A C T

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 is 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.

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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 USD0.08571 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 [11] 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 $\rho_{ic}$ is the density of the ice, $x_{cf}$ the weight fraction of ice, and $\rho_{cf}$ the density of the carrier fluid evaluated at the specified ice slurry temperature and initial carrier fluid concentration.
\[
\rho_d = \frac{1}{\frac{V_{in}}{V_{out}} + \frac{1 - x_{ice}}{V_{cf}}}
\]  
(2.1)

Under actual heat transfer conditions, the heat received by ice slurry is used for ice melting, for warming up solution and ice crystals and for changing the concentration of solution, which occurs in ice slurry. The heat transfer rate \( Q \) can be represented by the Eq. (2.2).

\[
\dot{Q} = \dot{m} \Delta h_{sl}
\]  
(2.2)

And the amount of melted ice is given by Eq. (2.3):

\[
\Delta h_{sl} = x_{ice, in} - x_{ice, out}
\]  
(2.3)

Enthalpy difference of ice slurry, \( \Delta h_{sl} \) which includes both sensible heat and latent heat of melting, is one of the most important factors in sizing an ice slurry system. The enthalpy of ice slurry can be expressed as:

\[
h_{sl} = x_{ice} h_{s} + (1 - x_{ice}) h_{cf}
\]  
(2.4)

where \( h_{s} \) is the concentration, \( h_{ice} \) is the enthalpy of ice and \( h_{cf} \) is the enthalpy of the concentrated solution, which can be represented as equation below.

\[
h_{cf} = \Delta h_{mix} + c_{cf}(T - T_{ref})
\]  
(2.5)

In the Eq. (2.5), \( \Delta h_{mix} \) is the enthalpy of mixing of the current solution, \( c_{cf} \) specific heat of carrier fluid, and \( T_{ref} \) = 273.15, reference temperature. The enthalpy of ice, \( h_{ice} \), is shown in Eq. (2.6).

\[
h_{ice} = -H_{f,ice} + c_{f,ice}(T - T_{ref})
\]  
(2.6)

where \( c_{f,ice} \) is the specific heat of ice and \( H_{f,ice} \) is the latent heat of fusion for ice, 332.4 kJ/kg. Eqs. (2.4), (2.5), (2.6) suggested by Niezgoda-Zelasko [13] are used to define the enthalpy of ice slurry.

Numerous experiments have been conducted to determine local and average heat transfer coefficients of ice slurry. Egolf et al. [14] suggested, the local heat transfer coefficient, \( \alpha_{sl} \), that for the ice slurry flow in a heat exchanger can be determined from the thermal conductivity of the fluid \( k_{sl} \), wall temperature gradient \( \frac{\partial T}{\partial y} \) at \( y \), and mean temperature difference between the peripheral mean wall temperature, \( T_{wall} \), and the fluid bulk temperature, \( T_{sl,mean} \). \( \alpha_{sl} \) can be described by Newton’s law of viscosity; as shown Eq. (2.8) where \( \gamma \) is the shear rate and \( \mu \) is the viscosity.

\[
\alpha_{sl} = \frac{k_{sl}}{(T_{wall} - T_{sl,mean})_{y} \left( \frac{\partial T}{\partial y} \right)_{y=0}}
\]  
(2.7)

Several experiments show that ice slurry behaves as a Newtonian fluid at low ice concentrations, and as a non-Newtonian fluid at high ice concentrations. The shear stress, \( \tau \), can be described by Newton’s law of viscosity; as shown Eq. (2.8) where \( \gamma \) is the shear rate and \( \mu \) is the viscosity.

\[
\tau = \mu \frac{\partial v_x}{\partial y} = \mu \cdot \gamma
\]  
(2.8)

3. Methodology

3.1. Field work

Field work study was conducted to the current air-conditioning system in BUA library, where the indoor air conditions were recorded. Additional information was acquired from the operation and maintenance manual, such as the design specifications and piping plans. Supply air conditions was measured by the thermo-hygro-anemometer, which was used to measure the dry bulb temperature, relative humidity and air velocity, at the centre of the round duct with the diffuser detached. The carbon dioxide concentration in the supply air was measured by indoor air quality meter CO₂ analyzer. Furthermore, space air conditions, such as dry bulk temperature and relative humidity, were measured and recorded by using the mentioned devices. These air conditions were tabulated for further analysis.

3.2. Simulation

The SimCad software for TRNSYS is a Computer Added Design (CAD) program for building modeling. It is available as a stand-alone program, does not require any licenses from other CAD packages. BUA library layout is done in SimCad. SimCad offers all commonly used CAD drawing tools to construct 3 dimension building structure from scratch. During the construction, wall materials, window types and thermal zone can be assigned. By simply saving the project, an intermediate data file (with an extension of .BUI) that contains the necessary information about the building envelope and oriented data structure. The intermediate data file will be used by TRNBuild for further processes.

Both programs, TRNBuild and SimCad are used in series to create the complete intermediate data file needed to simulate multi-zone (Type 56) buildings in TRNSYS. Using the TRNBuild program, non-architectural details like schedules, air ventilation rate, occupants, etc. are included inside the BUI file. The BUI file will be used to simulate the multi-zone (Type 56) component in TRNSYS.

TRNSYS simulation studio is a complete and extensible simulation environment for transient simulation of air-conditioning systems in multi-zone buildings. A multi-thermal zone building component (Type 56) was created to represent the BUA library. The TRNSYS library includes many components that are commonly found in HVAC systems such as fans, pumps, cooling coils and chillers. Since TRNSYS has an open and modular structure, it enables custom or non-standard component models to be added into it, by utilizing programming languages such as the C++ and FORTRAN.

3.3. Energy, economics and air quality analysis

The energy consumption current air-conditioning system was reviewed and total operating cost was calculated. Economic analysis was performed for new HVAC systems with ice slurry storage and ice thermal storage system to determine operating cost savings by applying the lower electricity tariff during off-peak hour offered by Tenaga Nasional Berhad (TNB). In addition, quality of air in the thermal zones for the new HVAC systems were analyzed and compared with the current systems.

4. Field work study

In this study, air-conditioning system in the library of Balai Un-Puk Aziz (BUA) was investigated and analyzed. BUA is located in the Faculty of Dentistry, University of Malaya, Kuala Lumpur, Malaysia. The BUA building is primarily air-conditioned by central chilled water air-conditioning system. The building consists of 3 levels where the library, computer room and the administration office is located in Level 2. The chiller plant located approximately 100 m from the main building and consists of 3 units of Carrier 30HXC190 chillers and 6 units of Aerofoil AE-80-32-D water pumps. There are 2 units of Nihon CTA225UN cooling towers at the back of the chiller plant, where the condensing water is pumped through the chiller condenser from the plant and cooled in these towers.

There are a total of 11 Carrier air handling units (AHU) in the main. Three out of the eleven AHUs in BUA are used to condition the air in level 2. The computer room air is conditioned by a Carrier 39G-1018 AHU (AHU C), while the library and administration office shared two AHUs, which are the Carrier 39G-2127 AHU (AHU A) and Carrier 39G-1118 AHU (AHU B), as shown in air-ducting layout.
in the Fig. 1. The blue\(^1\)-coloured ducts represent the supply air (SA) ducts from AHU while green-coloured ducts represent the return air (RA) ducts from each thermal zone that draws the RA back to the mixing room where it will be mixed with the outdoor air (OA).

Data on the conditions of SA and RA was collected from each diffuser from the three different AHUs in the library of BUA. The average dry bulb temperature (DBT) and relative humidity (RH) for all the AHUs are shown in Table 1.

Since AHU A and AHU B is used to supply cool air for the same thermal zone in the library, data for AHU A and AHU B is combined to generate the average DBT and RH by using the area-weighted method. As a result, the average RA DBT and RH for the library zone are 20.9\(^\circ\)C and 64.1\%. By applying the thermal comfort standard from ASHRAE standard, air conditions for both rooms in library and computer lab is out of the thermal comfort zone, which is shown in Fig. 2.

5. Baseline simulation in TRNSYS

The BUA library is separated into 2 zones; Library and Computer Lab. AHU unit 39G-1018 is used for computer lab, whereas 39G-1118 and 39G-2127 are used for Library. Outdoor air is mixed with return air from both zones in controlled flow mixers (Type 11c) which act as a mixing room. Chilled water is supplied from chiller (Type666) to three different cooling coils (Type 52b) inside AHU unit and return to chiller where Tee Piece (Type11 h) is used. Since TRNSYS has an open and modular structure, some non-standard models can be added by utilizing programming language, such as Chilled Water Flow Controller (Type311), which is used to control chilled water flow rate and supply air conditions.

5.1. Chiller cooling load

The simulated result from the baseline design of air-conditioning systems, which is illustrated in Fig. 3, indicates that the average chiller cooling loads are 874 MJ/h or 243 kW. The year round chiller cooling load is presented in Fig. 4. It shows that cooling demand is higher in the months of April to June in comparison to the rest of the months. The maximum chiller loads happened in Week 21, Day 147, 1500 h, which amounted to 1206 MJ/h or 335 kW. The average cooling load for day 147 is 1098 MJ/h, about 25% higher than the average cooling load.

Since the peak load occurred in day 147, the chiller load on that day is used for further analysis where chiller load profile on that day is shown in Fig. 5. It is noted that the cooling load demand started from 0800 h, at 1029 MJ/h, reached the peak load of 1206 MJ/h at 1500 h and subsequently dropped to 1039 MJ/h at 1800 h.

5.2. Zone indoor air condition

In order to simulate the actual conditions of BUA library, the library is separated into two thermal zones: library and computer lab, as mentioned before. Since the peak-cooling load fell on day 147, the indoor air conditions for week 21 will be reviewed. The simulation results of average dry bulb temperature (DBT) and relative humidity (RH) of the indoor air for both zones are tabulated in Table 2. The result showed that the current air-conditioning systems are capable of fulfilling the maximum cooling demand.

5.3. Power consumption

In this simulation, the power consumption of chillers, water pumps, AHUs fans and office equipments was analyzed. The power consumption for HVAC equipments is approximately 78% of the total power consumption of the library in this study. The HVAC equipments power consumption breakdown can be sorted to the

\( ^1\) For interpretation of color in Fig. 1, the reader is referred to the web version of this article.
consumption of chillers at 207 MJ/h, AHUs fan at 60 MJ/h, chilled water pumps at 43 MJ/h and condensing water pumps at 32 MJ/h. Office equipments such as television, photocopiers, computers, refrigerator and incandescent light consume a total power of 94 MJ/h. All the equipments are assumed to be fully functional during working hours of BUA library from 0800 to 1800 h, Monday to Saturday. Hence, according to Fig. 6, the maximum power consumption for various equipments on day 147 is 588 MJ/h and the annual cumulative energy consumption of the BUA library is 1504GJ.

5.4. Electricity charge

From the simulation result, the maximum monthly electricity cost of USD4690 is in May, which consists of usage charge of USD3144 and USD1546 for maximum demand charge. The minimum monthly charge fell on February with an amount of USD4120. The detailed electricity charge for every month is presented in Fig. 7. It is noted that the annual electricity cost of BUA library, based on TNB commercial tariff C2, is USD52918; where
68% is usage charge. In other words, the maximum power demand charge is contributed significantly to the total electricity charge.

6. Simulation layout for ice thermal storage systems

Ice thermal energy storage uses the latent heat of ice to store the cooling energy. Ice is generated during the charging cycle at off-peak hours and the stored ice absorbed heat from the building during daytime.

6.1. Ice tank thermal storage system

Ice thermal storage (ITS) tank is a proven technology, where it is capable of meeting the requirements of flexible energy management. In this study, the Ice-Cel Model TS205, an ice thermal storage tank from Dunham-Bush, is the ice thermal storage tank used for full and partial load storage systems. The Ice-Cel Model TS205 ITS tanks are charged by a 373 kW chiller Type666 in TRNSYS. The schematic diagram of the ice tank thermal storage system is presented in Fig. 8. The return brine is cooled by ITS tanks in this system; hence, a non-standard component Type 301 is created based on the performance data obtained from Dunham-Bush.

6.2. Ice slurry thermal storage system

Ice slurry thermal storage system is an alternative cool thermal storage technology. The technology of ice slurry is still undergoing the development stage. Ice slurry is generated from an ice slurry generator and a shell-and-tube heat exchanger that accepts 10% glycol solution. The ice slurry from generator drops into the storage tank directly and is pumped to one or more tanks located elsewhere. The non-standard components of TRNSYS were created for the simulation. Fig. 9 shows the schematic diagram of the ice slurry thermal storage systems. The cooling coil (Type52b) for baseline design is replaced by an ice slurry-cooling coil (Type305). A simplified non-standard component of ice slurry-cooling coil (Type305) was created for TRNSYS simulation. It was formulated based on the expression given by Stocker and Jones [18] shown in Eqs. (A.15), (A.16), (A.17), (A.18). By applying the Newton Raph-
The ice slurry generated during off-peak period will be stored in the ice slurry storage tank (Type 302). A simplified ice slurry storage system was built using the relevant theory and steps found in Appendix A1. Subsequently, the ice slurry will be supplied to three different cooling coils. Return ice slurries will be mixed in the mixer (Type 315) before it returns to the ice thermal storage system.

7. Ice thermal storage control strategies

In a conventional chilled water cooling system in BUA library, the chiller is operated during the peak hours from 0800 to 1800 h, from Monday to Saturday. In order to shift the electricity usage for chiller from the peak hours to off-peak hours, the application of cool thermal storage systems is recommended. Cool thermal storage systems could be operated in full loading or partial...
loading storage mode, where partial load storage is further categorized to demand limiting and load-levelling storage.

During off-peak hour, ice tanks thermal storage is charged by a 373 kW capacity chiller in the charging cycle. For an ice thermal storage system, the chiller produced brine at −3.3 °C and supplied to charge ice thermal storage (ITS) tanks, which will be used to meet the cooling energy demand during daytime. For the full load storage system, the chiller does not operate during peak hours and all the cool energy is supplied by the cool thermal storage. Brine that returned from AHU at 8.9 °C is cooled down to 2.2 °C when it is passing through ITS tanks and re-circulated to the AHU again.

On the other hand, the chiller for a partial load storage system functioned during daytime to provide part of the cooling energy instead of fully depending on the cool thermal storage system. Return brine at 8.9 °C is pre-cooled by the chiller to 5.6 °C before entering ITS tanks. As mention earlier, the partial load storage system is subdivided into two strategies. For the load-levelling partial load storage, one chiller is operated at half of the current capacity, which is 187 kW. This chiller will be operated 24 h daily to supply the cooling capacity, as well as charging the cool thermal storage during off-peak hours. However, the demand limiting partial load storage system utilized two chillers with lower capacities, which are at 125 kW each, during charging cycle. A comparison of the existing air-conditioning systems and cool thermal storage system operation strategies is given in Table 3.

### 8. Simulation results and discussion

#### 8.1. Chiller load

From the simulated result of ice thermal storage systems on day 147, as illustrated in Fig. 10, the maximum chiller load for baseline design is 1207 MJ/h during peak period. The partial load-levelling and demand limiting storage systems recorded peak loads of 585 MJ/h and 295 MJ/h respectively. The chillers in partial load storage systems operated from 0800 to 1800 h for BUA library daily operation. It is used to recharge the cool thermal storage during off-peak period from 2200 to 0800 h. As a result, it will be turned off for 3 h daily. On the other hand, there is no chiller load during peak hour for full storage systems as it is shifted to off-peak period, which indicated the peak chiller load of 1326 MJ/h.

A comparison of chiller loads between baseline design and cool thermal storage air-conditioning systems is given in Table 4. It showed that the total energy consumption for cool thermal storage systems is higher than baseline design air-conditioning systems because of the reduced suction temperature in ice making mode, as explained by Dunham-Bush [15].

#### 8.2. Power consumption

Fig. 6 showed the power consumption of the current air-conditioning system in BUA library is concentrated within the peak period from 0800 to 1800 h. It may be concluded that the most power consuming equipments are the chillers, followed by office appliances, AHUs fan, chilled water pumps and condensing water pumps.

By applying the full load thermal storage strategy, the power consumption of chiller and condensing water pump are shifted to the off-peak hours, as shown in Fig. 11. It is noted that the AHU fans and office equipments are the only components operating during the peak period. Meanwhile, the chilled water pump is operated from 2200 to 1800 h, pumping chilled water to charge the ITS tanks during off-peak period and discharges otherwise.

When the partial load thermal storage strategy is applied, chillers, chilled water and condensing water pump are operated from 2200 to 1800 h. Chillers are used to produce chilled water during peak hours and are used to charge ITS tanks during off-peak hours. Therefore, part of the chiller power consumption is shifted to off-peak hours, which are exhibited in Figs. 12 and 13. The difference between load-levelling and demand limiting storage systems is the

<table>
<thead>
<tr>
<th>Table 3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Comparison of baseline design and cool thermal storage air-conditioning systems operation strategies.</strong></td>
</tr>
<tr>
<td>Baseline design</td>
</tr>
<tr>
<td>Chiller size (MJ/h/kW)</td>
</tr>
<tr>
<td>Number of chiller</td>
</tr>
<tr>
<td>Number of chiller used during peak period (0800–1800)</td>
</tr>
<tr>
<td>Number of chiller used during off-peak period (2200–0800)</td>
</tr>
</tbody>
</table>

**Fig. 10.** Total power consumption profile on day 147.
percentage of the chiller load shifted to off-peak period. In addition, all the equipments under both strategies ceased operation from 1800 to 2200 h, due to the presence of a period between the off-working and off-peak hours.

Fig. 14 showed that the cumulative energy consumption for the BUA library existing air-conditioning systems is 1504 GJ. It is found that 20% higher energy consumption will occur if cool thermal storage systems are used to replace the existing system. The reasons are the increased chiller power consumption during night time due to the reduced suction temperature in ice making mode and the prolonged operation of the chilled and condensing water pump. Both water pumps are operated for 20 h daily from Monday to Saturday, instead of 10 h operation in the existing system.

8.3. Economic analysis

From the economic analysis point of view, ice thermal storage systems are considerable although the total energy consumption

<p>| Table 4                                                                 |
|-------------------------------------------------------------------------|-----------------|-----------------|-----------------|</p>
<table>
<thead>
<tr>
<th></th>
<th>Baseline design</th>
<th>Full load</th>
<th>Partial load (50%): load leveling</th>
<th>Partial load (67%): demand limiting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chiller size (MJ/h/kW)</td>
<td>(1343/373)</td>
<td>(1343/373)</td>
<td>(673/187)</td>
<td>(450/125)</td>
</tr>
<tr>
<td>Maximum chiller load at peak period (MJ/h)</td>
<td>1207</td>
<td>0</td>
<td>585</td>
<td>295</td>
</tr>
<tr>
<td>Maximum chiller load at off-peak period (MJ/h)</td>
<td>0</td>
<td>1326</td>
<td>663</td>
<td>884</td>
</tr>
<tr>
<td>Chiller energy Consumption during peak period (MJ)</td>
<td>12,086</td>
<td>0</td>
<td>6305</td>
<td>3525</td>
</tr>
<tr>
<td>Chiller energy Consumption during off-peak period (MJ)</td>
<td>0</td>
<td>13,264</td>
<td>6632</td>
<td>8843</td>
</tr>
<tr>
<td>Total energy consumption (MJ)</td>
<td>12,086</td>
<td>13,264</td>
<td>12,937</td>
<td>12,368</td>
</tr>
</tbody>
</table>

**Fig. 11.** Power consumption of full load storage systems on day 147.

**Fig. 12.** Power consumption of load leveling storage systems on day 147.
is higher in comparison to conventional air-conditioning systems. Simulation results showed that the total electricity costs for baseline design are USD52918. Ice thermal storage systems that shifted part of the power consuming operation to the off-peak period have lower rates. Electricity cost for full thermal storage systems is USD40297, which brings the highest energy cost saving about 24%, followed by partial load demand limiting systems that save 11% of energy costs and partial load-levelling systems that save 5% of energy costs, as shown in Fig. 15.

It is noted that the cost savings in usage charge are insignificant, which are about 7% of saving for full thermal storage systems. The electricity usage charge for both partial load storage systems is 4% higher than baseline design. On the other hand, the peak power consumption base on the tariff of RM35.60 or USD10.60 per kW during peak hours showed significant savings in all three types of cool thermal storage systems. The full load storage gives 60% of costs saving and followed by partial load demand limiting and load-levelling systems at 37% and 25% respectively.

8.4. Zone air quality

For modern air-tight buildings, occupants’ health and thermal comfort is related to the indoor air quality. Cool thermal storage systems are capable in producing supply air with a lower humidity ratio in comparison to conventional air-conditioning systems. Simulation result showed that a lower indoor air humidity ratio for both zones could be achieved by ice slurry storage systems in comparison to the existing system. The average humidity ratio for the computer laboratory on day 147 is reduced from 10.9 g/kg for the baseline design to 9.6 g/kg and 7.6 g/kg for ice tanks and ice slurry thermal storage systems. The library also exhibited the decrease in humidity ratio on the same day, which is from 11.5 to 10.4 g/kg and subsequently reached 8.1 g/kg. The average indoor air conditions of week 21 are tabulated in Table 5. Normally, the conventional chilled water supply temperature for a AHU is from 6 °C to 7 °C, ice tanks thermal storage systems supply chilled water temperature from 1 °C to 4 °C, while ice slurry storage system supply...
ice slurry from –2 °C to 0 °C. In addition, thermal conductivity of ice slurry is higher than chilled water and the overall cooling coil local heat transfer coefficient, a, is directly proportionate to the thermal conductivity of fluid $k_f$ and the wall temperature gradient,$\left( \frac{\partial T}{\partial y} \right)$[14]. As a result, ice slurry-cooling coil could produce coolest supply air to the thermal zones. Furthermore, indoor air psychrometric conditions for both ice thermal storage systems shown that the supply air is over-cooled. Therefore, it is feasible that AHU units to be sized down in new systems, according to Hasnain et al. [16], due to the increased air temperature difference over the AHUs.

9. Concluding summary

Field works have been carried out at the BUA library in University of Malaya and 1264 m$^2$ of air-conditioned zones is investigated for this study. It is revealed that the BUA library did not exhibit any sick building syndrome. However, the indoor air conditions for both zones are not within the comfort zone as suggested by ASHRAE, as shown in Fig. 2.

Ice thermal storage system could be operated in full load or partial load storage. The maximum chiller load on day 147 is 1207 MJ/h for baseline design, while partial load-levelling and demand limiting storage systems records peak loads of 585 MJ/h and 295 MJ/h respectively. The chiller load of full storage systems is shifted to off-peak period, resulting in zero chiller loading during the peak hours.

It is found that the cumulative energy consumption of cool thermal storage systems increased by 20% from the existing system. This is because the chillers consume more energy during charging cycle and the usage of water pump is longer in comparison to the current air-conditioning systems.

The total electricity costs for baseline design are USD52918, as shown in the simulation result. Electricity costs for full thermal storage systems are USD40927, which give the highest energy cost saving of about 24%, followed by partial load demand limiting systems and partial load-levelling systems. All three types of cool thermal storage systems showed significant cost savings from the peak power demand charges. Full load storage yielded 60% of cost savings and followed by partial load demand limiting and load-levelling at 37% and 25% respectively.

Simulation results also showed that a lower indoor air humidity ratio for both zones could be achieved by using the ice slurry storage systems. In addition, indoor air psychrometric conditions for both ice thermal storage systems showed that they were over-cooled. Therefore, it is feasible to downsize AHU units in the new systems.

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Appendix A

A.1. Modeling ice slurry thermal storage tank

By assuming the overall heat transfer coefficient for the tank wall, $U_t$, the same with the ITS tanks, the heat loss due to the environment $q_{env}$ could be calculated by using Eq. (A.1).
\[ q_{\text{env}} = U_i \times A_i \times (T_{\text{env}} - T_{\text{tank}}) \]  
(A.1)

The total latent heat \( Q_{\text{lat}} \) can be stored is equal to number of ITS tanks \( N_{\text{tank}} \) used times capacity \( C_{\text{p}} \) of each tank, latent heat of ice \( H_f \) and the maximum ice fraction \( x_{\text{max}} \) of 20%, which is shown in Eq. (A.2).

\[ Q_{\text{lat}} = N_{\text{tank}} \times C_{\text{p}} \times H_f \times x_{\text{max}} \]  
(A.2)

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

\[ IGR = \frac{q_{\text{evaporator}} \times CF}{H_f} \]  
(A.3)

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 \( q_d \) from storage tank can be determined by Eq. (A.4).

\[ q_d = \dot{m}_i A_{\text{ice, b}} \times H_f \]  
(A.4)

Where \( \dot{m}_i \) is the mass flow rate of ice slurry and \( x_{\text{ice, b}} \) is ice fraction of ice slurry which can be defined below.

\[ x_{\text{ice, b}} = \frac{\text{BIM}}{N_{\text{tank}} \times C_{\text{p}}} \]  
(A.5)

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

\[ q_{\text{total}} = q_d + q_{\text{env}} \]  
(A.6)

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).

\[ IGR = \frac{q_{\text{total}}}{H_f} \]  
(A.7)

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.

\[ FIM = BIM + (IGR - IGR) \Delta t \]  
(A.8)

where

\[ \begin{align*}
BIM & = \text{Beginning ice mass (kg)} \\
FIM & = \text{Final ice mass (kg)} \\
\Delta t & = \text{time step}
\end{align*} \]

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

\[ x_{\text{ice}} = \frac{FIM}{N_{\text{tank}} \times C_{\text{p}}} \]  
(A.9)

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, \( dA \).

\[ dq = \frac{\alpha_i dA}{C_{\text{pm}}} (h_i - h_1) \]  
(A.10)

where

\[ \alpha_i = \text{convection coefficient} \]

\[ C_{\text{pm}} = \text{specific heat of air mixture} \]

\[ h_i = \text{enthalpy of air} \]

\[ h_1 = \text{enthalpy of saturated air at wetted surface temperature} \]

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

\[ dq = \alpha_i dA \left( t_i - t_r \right) \]  
(A.11)

where

\[ t_i = \text{temperature of refrigerant} \]

\[ t_r = \text{temperature of wetter surface} \]

\[ dh_i = \text{refrigerant side area} \]

\[ \alpha_i = \text{combined conductance through wetted surface, metal and refrigerant} \]

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

\[ R = \frac{\alpha_i}{C_{\text{pm}} x_{\text{ice}}} \left( \frac{A}{A_i} \right) = \left( \frac{t_i - t_1}{h_i - h_1} \right) \]  
(A.12)

The enthalpy of saturated air \( h_1 \) is a function of the temperature of the wetted surface \( t_i \) that could be related by the cubic equation as shown below.

\[ h_1 = 9.3625 + 1.7861 t_1 + 0.01135 t_1^2 + 0.00098855 t_1^3 \]  
(A.13)

If the inlet enthalpy of air stream \( h_{i,1} \) and refrigerant temperature is given as an input in the simulation, \( t_1 \) and \( h_1 \) 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.

\[ \frac{t_1 - t_R}{R} - h_{i,1} + 9.3625 + 1.7861 t_1 + 0.01135 t_1^2 + 0.00098855 t_1^3 \]

\[ = 0 \]  
(A.14)

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 \( A_{i,1} \). Four equations below can then be written for the first increment of areas.

\[ m_i (h_{i,1} - h_{i,2}) = q \]  
(A.15)

\[ \frac{\alpha_i A_{i,1-2}}{C_{\text{pm}} x_{\text{ice}}} \left( \frac{h_{i,1} + h_{i,2} - h_{i,1} + h_{i,2}}{2} \right) = q \]  
(A.16)

\[ \frac{\alpha_i A_{i,1-2}}{A_i / h_i} \left( \frac{t_{i,1} + t_{i,2} - t_i}{2} \right) = q \]  
(A.17)

\[ h_{i,2} = f(t_{i,2}) \]  
(A.18)

The four remaining unknown variables, \( q \) heat transfer rate, enthalpy of outlet air \( h_{i,2} \), enthalpy of outlet saturated air \( h_{i,2} \) 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.
\[ q = f(h_{t_2}) \quad \text{(A.19)} \]  
combine to create new equation

\[ q = f(h_{t_2}, h_{t_1}) \quad \text{(A.20)} \]  

\[ q = f(h_{t_2}) \quad \text{New Eq.} \]

\[ h_{t_2} = f(t_{t_2}) \quad \text{New Eq.} \]

\[ h_{t_2} = f(t_{t_2}) \quad \text{(A.21)} \]

finally create a nonlinear equation of \( t_{t_2} \)

As a result, \( t_{t_2} \) can be determine 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 \( A_{t_1} \), the sensible heat transfers are shown below.

\[ q_s = m_{t_2}c_{pm}(t_{o_1} - t_{o_1}) \quad \text{(A.23)} \]

\[ q_s = A_{t_2}q_{t_2}\left(\frac{t_{t_2} + t_{t_2}}{2} - \frac{t_{t_1} + t_{t_2}}{2}\right) \quad \text{(A.24)} \]

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.

\[ q = m_{t_2}\Delta h_d \quad \text{(A.25)} \]

\[ h_{t_1} = x_{\text{ice}}h_{\text{ice}} + (1 - x_{\text{ice}})h_{\text{cf}} \quad \text{(A.26)} \]

References


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


