Economical and thermal optimization of possible options to control visible plume from wet cooling towers

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Economic study has been done for the heat pumps, biogas and solar collectors along with the phase change materials (PCM) storage and the results have been compared for various costs. The different combinations and options for the heating and cooling requirements, the heating capacities and the comparisons of different costs are studied in detail. It has been found that all the costs are much lesser for a solar collector system followed by the biogas plant and the heat pump systems. On the other hand, all the costs are found to be the highest in the case of an air cooled geothermal heat pump system, while it is reverse in case of the water cooled solar collector system.

\textbf{Keywords:} Phase change materials, Solar collector, Biogas, Heat pump, Water cooled chiller, Air cooled chiller, Visible plume, Commercial building, Wet cooling towers

1 Introduction

Cooling towers are enclosed boxes for cooling of water based on its evaporation by direct contact with the air, which is achieved partly by an exchange of latent heat from the water evaporation and partly by a transfer of sensible heat. The development related to the analysis of the cooling tower began when the first work carried out by Lewis\textsuperscript{1}, and was used by Robinson\textsuperscript{2} for the first time to establish the general principles. For several decades, many researchers have studied the convection phenomena occurring in cooling towers. Merkel\textsuperscript{3} used the enthalpy potential as the driving force for air-water exchange and assumed a similarity between heat and mass convective transfer by taking the Lewis number as unity. Baker and Shrylock\textsuperscript{4} developed a detailed explanation of the performance of a cooling tower, thus clarifying the assumptions and approximations used by Merkel. Sutherland\textsuperscript{5} showed that Merkel’s theory leads to an underestimation of tower size by 5-15\%.

During unfavorable weather conditions, the exhaust of the cooling tower remixes with the cooler ambient air and as it cools down the excess moisture condenses in small fog droplets, creating visible plume. Nowadays, the visible plume from huge buildings attracts public attention and is a matter of greater concern at many places including Hong Kong. Methods of reducing, preventing, removing and minimizing of visible plume have taken many forms, such as, heating the exhaust with natural gas burner, steam coils, installing precipitators, and spraying chemicals\textsuperscript{6-11}. However, such types of solutions, in general, are expensive and are not always effective.

A little can be done in a wet cooling tower to reduce/remove plume persistence. It is observed that the condition for the visible plume is the lower ambient temperature and the higher relative humidity. The wet cooling towers are economically cheap and technically simple, easy to build and need less power to operate as compared to other types of towers. However, they do not have any control over the visible plume. In recent years, the visible impact of releases to the ambient has become a matter of greater concern due to the awareness of environmental degradation and protection among the society. Very recently, some researchers have used the heating and cooling strategy by heat pumps\textsuperscript{12-13} and solar collectors\textsuperscript{14} to control the visible plume from wet cooling towers. In case of heat pumps, it was found that the combination of heating and cooling could be a better choice not only from the point of view of energy conservation but also from the point of view of economics.
The present work is a comparative economic study for the above mentioned application to utilize the heat pumps, biogas and solar collectors combined with the phase change materials (PCMs) based thermal energy storage to control the visible plume from wet cooling towers of the International Commerce Centre (Kowloon), Hong Kong.

2 Methodology

2.1 System description

The line diagram shown in Fig. 1(a) describes a chiller coupled with a building envelope (space to be cooled), a cooling tower (to extract heat from the chiller) and a heating system (biogas/heat pump/solar collector) to heat the exhaust of the cooling tower. The refrigerant absorbs heat from the evaporator and changes from saturated liquid state to saturated vapour state and enters the compressor, where it is compressed up to superheated state. At this state, the refrigerant enters the condenser and cools down up to point 3 by transferring the heat to the circulating water coming from the cooling tower. At this point 3, it enters the expansion valve and expands up to point 4 and again enters the evaporator, thereby, completing the chiller loop. The water enters the condenser at state point 8, absorbs the heat from the refrigerant and pumps back to the cooling tower at state point 6. The hot water at state 6 sprays over the fill packing comes in the contact of ambient air being circulated by the exhaust fan on the top of the cooling tower as shown in Fig. 1a. In this process the water cools down up to state point 8 by evaporative cooling and the heat is taken away by the ambient air. The air entering the cooling tower through side openings leaves the tower through the exhaust fan and duct at state point 7, completing the cooling tower loop. On the other hand, the circulating water leaving the evaporator at state point 10 enters the cooling coil where the secondary

Fig. 1 — Line diagrams of (a) the proposed system (b) the formation of plume (white fog) in a wet cooling tower, and (c) the potential of plume in a wet cooling tower at a typical weather condition in Hong Kong
fluid is circulated to cool down the building space and the resulting hot water is pumped back to the evaporator at state point 9, thus completing the building loop.

2.2 Formation and potential of visible plume

The air passing through the heat exchanger (fill) comes in contact with hot water, carries heat and moisture and exits in a partial superheated state. This plume then gradually remixes with the cooler ambient air. As it cools down, the excess moisture condenses into small fog droplets, creating visible plume as shown on the simplified psychrometric diagram of Fig. 1(b). During the winter season, the cold and humid ambient air at point 1 is warmed up to point 2 in the tower after absorbing the heat released by the hot water and then remixes with the ambient air along with line 2-1. Most of this mixing occurs in the superheated region as shown in Fig. 1c. In presence of larger moisture in the ambient air, the excess moisture cannot be held at equilibrium and so condensation occurs and a dense and persistent plume is formed. On the other hand, during hot and dry weather conditions, the exhaust air may get heated and remixes along the line 4-3, which falls wholly in the sub-saturated region and as a result, no visible plume is formed.

During unfavourable weather conditions, the exhaust of the cooling tower remixes with the cooler ambient air and as it cools down, the excess moisture condenses in small fog droplets, creating dense and persistence visible plume. Nowadays, the visible plume from huge buildings attracts public attention and is a matter of greater concern at some places like Hong Kong. As shown in Fig. 1(b), greater is the temperature difference between inlet and outlet air of the cooling tower, more intense will be the plume. The intensity of plume is more for lower ambient temperature and vice-versa. The area between the straight line and the saturation curve in the superheated region, predicts the possibility and potential of the plume. Thus by joining these two points by a straight line one can predict the possibility and the potential of the visible plume, based on the area of intersection. The area of intersection can be evaluated by standard mathematical formulation with the help of definite integral:

\[ A = 2 \int_{w_1}^{w_2} \sqrt{w} \, ds \]  

\[ \ldots (1) \]

where \( w_1 \) and \( w_2 \) represent the concentration of water in the air (g/kg) and \( \sqrt{w} \) represents the curve shown in Fig. 1c. The area of intersection given by Eq. (1), represents the magnitude of the potential of the visible plume, depending on the inlet and outlet air temperatures of the tower. Based on the temperature range of winter condition given in Table 1, the ambient temperature of 15°C has been chosen and the relative humidity of 95% for present case study. The optimum operating parameters, such as, the building load, cooling load, heating load, dry and wet bulb temperatures, inlet air temperature, relative humidity, water flow rate and air flow rate are similar to those given by earlier researchers\textsuperscript{12-14}. The building under consideration in this study is International Commerce Centre (Kowloon) Hong Kong. Under the most unfavourable combination of the ambient conditions, thermal load and topography, a plume can extend up to a few hundred meters and causes invisibility and darkness. The hourly variation of humidity and air temperature of Hong Kong for a typical year is given in Fig. 2. The most probable weather conditions have been taken for calculating the operating parameters for cooling towers (Table 1).

2.3 Heating and cooling strategy

Methods of reducing the visible plume have taken many forms, such as, heating the exhaust with natural gas burner, steam coils, installing precipitators, and spraying chemicals on the tower exhaust. However,
these solutions, in general, are expensive and are not always effective. Keeping all the possible options given in the literature, the new methods were applied by earlier researchers\textsuperscript{12-14}. The heating strategy by solar collectors and biogas, besides, the combined cooling and heating through heat pump systems are mentioned by Wang and Tyagi\textsuperscript{12}, Tyagi \textit{et al.}\textsuperscript{13} and Wang \textit{et al.}\textsuperscript{14}.

Tyagi \textit{et al.}\textsuperscript{13-14} showed that there are two ways to utilize the heat pump systems; firstly, as a heater and secondly, as the combination of cooler and heater. In the first method, the heating capacity of the heat pumps was utilized to heated up the exhaust of the wet cooling towers. In the second method, the cooling produced by the heat pumps on the evaporator side was utilized to cool down the exhaust of the towers. In that way a certain amount of moisture can be condensed besides, the reduction in the temperature, then it was heated in similar way as mentioned in the first method. In this way not only are the heating requirements reduced but the cooling produces by the heat pump is also utilized. Tyagi \textit{et al.}\textsuperscript{13} found that the latter is more effective provided the optimum distribution of the exhaust for cooling and heating is maintained. On the other hand, in the case of solar collectors and biogas systems, the heating method could be the only possible way.

2.3.1 Heating

It is clear from Fig. 3(a) that heating is carried out in a way that the exhaust of the tower while mixing with the ambient air along the line B to C remains in the sub saturated region Fig. 1(b). The area below the straight line AB, as shown in Fig. 3(a), represents the heating requirement and can be calculated by:

\[ Q_H = m_v C_p (T_B - T_A) \]  

where \( Q_H \) the heat required to reduce/control the visible plume, \( m_v \) and \( C_p \) are respectively, mass rate and specific heat of the exhaust air, while \( T_A \) and \( T_B \) are, respectively, the outlet and set point temperatures of the exhaust to be heated. All other parameters can either be measured directly from the system or can be calculated by the models given above. However, \( T_A \) and \( T_B \) need to be calculated based on the exhaust and inlet temperatures of the cooling tower by plotting on the psychrometric chart, shown in Fig. 3(a).

2.3.2 Cooling and heating

Figure 3(b) shows the combined cooling and heating carried out in a way that the exhaust of the tower while mixing with the ambient air along the line B to C remains in the sub saturated region as mentioned in Fig. 1(b). The area under the straight line A’A, parallel to the X-axis shows the requirement of cooling, while the area under the straight line AB, parallel to Y-axis shows the requirement of heating. The state points A’ and A are set for heating and cooling requirements based on the heating and cooling capacities of the heat pumps installed in the building for this purpose.

In other words, if there is no cooling option like, solar collector and biogas, then there is no way to cool down the exhaust of the towers. When heat pumps are used, there is a considerable amount of cooling (around 33% heating) produced by heat pumps\textsuperscript{12}. This is also known as the energy extracted from the heat source (cooling produced at the evaporator side) of the heat pump. This cool energy goes waste if not

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Fig. 3 — The heating and cooling requirements for a typical cooling tower (a) without cooling option (b) with cooling option
utilized for any purpose. At the same time, if this cool
energy is utilized to cool down the exhaust of the
tower, then the requirement of the heating is reduced
by 33%. This not only reduces the capacity of the heat
pump but also reduces the costs of the system, and
hence, both the heating and the cooling produced by
the heat pumps are fully utilized.

The cooling is carried out through line AC on the
saturation curve so that some moisture also reduces
because the condensation of the exhaust air takes
place. The requirement of the cooling is given by:

\[ Q_L = m_c C_p \left( T_{\alpha'} - T_{\alpha} \right) \]  \hspace{1cm} \text{...(3)}

While the heating is carried out through line AB at
constant moisture and the requirement of heating is
given by:

\[ Q_H = m_c C_p \left( T_{B} - T_{\alpha} \right) \]  \hspace{1cm} \text{...(4)}

where \( Q_L \) and \( Q_H \) are the heating and cooling required
to reduce/control the plume, while \( T_{\alpha'} \) is the outlet
temperature of the exhaust of the cooling towers and,
\( T_{\alpha} \) and \( T_{B} \) are the set points to adjust the heating
and cooling requirements of the exhaust in order to avoid
the visible plume. Based on the ambient air
conditions, the operating mode of cooling towers,
building cooling load and all other parameters can be
measured directly from the operating systems and/or
calculated by the simulation model. However, \( T_{\alpha} \) and
\( T_{B} \) are adjusted and calculated based on the total
heating and cooling requirements of the exhaust. For a
typical set of operating parameters, the outlet and inlet
temperatures of the towers, the heating and cooling
process are shown on the psychrometric chart, as can
be seen in Fig. 3(a-b).

2.4 Utility and importance of phase change materials (PCMs)

Phase change materials (PCMs) are latent heat
storage materials. They use chemical bonds to store
and release the heat. The thermal energy transfer
occurs when a material changes from solid phase to
liquid phase and vice-versa. This is called as change
in state or phase. Unlike conventional storage
materials, PCM absorbs and releases heat nearly at a
constant temperature. They store much more heat per
unit volume than sensible storage material such as
water, masonry or rock. Another key advantage with
the use of a PCMs is that the storage of heat/energy
and its recovery occurs isothermally, which makes
them ideal for space heating/cooling applications.

PCMs have been used for thermal storage in
conjunction with both passive and active energy
storage and a large number of PCMs are known in the
required temperature ranges. However, for
employment as latent heat storage materials, they
must exhibit certain desirable thermodynamic,
chemical, and kinetic properties.

In the present work, the utility and importance of
PCMs based thermal energy storage system for a
building has been studied. A part of the heat energy
produced from solar collectors is stored in the PCM
based energy storage tank and used in the evening
and/or in the late office hours. This not only avoids
the peak load but also reduces the consumption of
electricity, the running cost of the system and hence,
makes the system more economical. Further studies are
under way to store the heat/energy released from the
condenser of the chiller which can be used for heating
the exhaust of the cooling tower through PCM based
heat exchangers and/or other heating/cooling
applications.

3 Results and Discussion

The combined heating and cooling option is
possible only in case of heat pump systems and hence,
solar collector and biogas systems can not be used for
dual applications. Also, the heat pump is a
conventional system while the solar collector and the
biogas are non-conventional systems. They are two
different modes of operating of such applications and
therefore, the details are given separately. For
different operating conditions, the outlet temperatures
of the high and low speed tower are different and
basically depend on the number and speed of the
cooling towers. The potential of visible plume is
found to be higher for low speed towers, while it is
found to be reversed in the case of high speed towers.
This is because of the fact that the mixing of air (total
volume flow rate) in the high speed tower is much
higher than that of the low speed tower and therefore,
the exhaust temperature of a high speed tower is much
lesser than that of the low speed tower and vice-versa.
Due to this reason, there is a big difference between
the outlet temperature of the high speed and the low
speed cooling towers. In contrast, the exhaust
temperature of cooling tower depends on the number
of cooling towers in use and the mode of operation, at
a given ambient temperature.

Table 1 shows the weather conditions while, the
requirements of power, building load, outlet
temperatures of the cooling towers for different
modes of operation along with the number of cooling
Table 2 — Operating parameters of cooling towers and chillers for critical winter case (Building cooling load is 17236 kW)

<table>
<thead>
<tr>
<th>Number of cooling towers</th>
<th>Outlet Temp. (°C)</th>
<th>Heating required kW</th>
<th>Power consumption</th>
<th>Heat transfer rates</th>
<th>COP of chiller</th>
</tr>
</thead>
<tbody>
<tr>
<td>High speed</td>
<td>Low speed</td>
<td>a) High speed</td>
<td>b) Low speed</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>26.32</td>
<td>27.42</td>
<td>1988.8</td>
<td>270</td>
</tr>
<tr>
<td>5</td>
<td>-</td>
<td>23.70</td>
<td>-</td>
<td>1311.0</td>
<td>450</td>
</tr>
<tr>
<td>-</td>
<td>10</td>
<td>23.67</td>
<td>-</td>
<td>1311.0</td>
<td>300</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>20.30</td>
<td>21.04</td>
<td>655.5</td>
<td>600</td>
</tr>
<tr>
<td>10</td>
<td>-</td>
<td>19.50</td>
<td>-</td>
<td>655.5</td>
<td>900</td>
</tr>
</tbody>
</table>

Table 3 — Properties of the PCM used (CaCl₂·6H₂O)

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melting temperature (°C)</td>
<td>29.70</td>
</tr>
<tr>
<td>Latent heat of fusion (kJ/kg)</td>
<td>169.98</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td></td>
</tr>
<tr>
<td>Solid</td>
<td>1802</td>
</tr>
<tr>
<td>Liquid</td>
<td>1562</td>
</tr>
<tr>
<td>Specific heat (kJ/kg °C)</td>
<td></td>
</tr>
<tr>
<td>Solid</td>
<td>1.46</td>
</tr>
<tr>
<td>Liquid</td>
<td>2.13</td>
</tr>
<tr>
<td>Thermal Conductivity (W/m °C)</td>
<td></td>
</tr>
<tr>
<td>Solid</td>
<td>1.09</td>
</tr>
<tr>
<td>Liquid</td>
<td>0.54</td>
</tr>
</tbody>
</table>

towers is given in Table 2. The specification of the PCM is given in Table 3 and the specifications of solar collector are given in Table 4. Table 5 presents the cost of the PCM along with the amount of heat to be stored and the requirement of the tank size. The details of biogas are shown in Table 6 and the details of air and water cooled heat pumps are given in Table 7. The comparison between electric and geothermal heat pumps is presented in Table 8. On the other hand, the mass flow rate of water and air, the energy consumption, and cost of the collector along with the requirements of heating for PCM and tower exhaust are given in Tables 9 and 10, respectively. It is also important to note that the thermal energy storage system can store enough heat that can be utilized for another 3-4 h in the absence of solar energy during the afternoon/late office hours when the electricity consumption/ demand is higher due to peak load.

Using the inlet and outlet temperatures of the tower, the potential of plume is plotted on the psychrometric chart in Figs 4-7 for their respective parameters given in Table 2. It is clear from these plots that higher is the exhaust air temperature; higher will be the potential of the visible plume. Also higher is the potential of the plume, higher will be the requirements of heating. It has been seen from these figures that the potential of plume is higher when the number of cooling towers are less and vice-versa. These results are based on the calculations done for the critical weather conditions marked on Fig. 2 and
The potential of visible plume is highest when the number of (high speed) tower is the least as can be seen from Fig. 4a, while it is least in the case of highest (high speed) number of cooling towers (Fig. 7). At the same time, the COP of the chiller and the requirements for heating are found to be the lowest and the highest respectively. Apart from these two extreme cases, the potential of plume, the COP of the chillers and the heating requirements are found to be moderate and changes as the number of high and low speed towers are changed.

It is also important to note that the heat pump case is dual, so the heating requirement may vary unlike the solar collector and biogas. Also, there are two combinations, i.e. the heating as well as the cooling and heating options for the heat pump systems unlike the solar collector and biogas systems. That is why Tables 7 and 8, indicate more options than those given in Tables 6, 9 and 10. Also there are two types of heat pumps with two different cooling and heating modes, i.e. the water cooled and air cooled heat pump and the electric and geothermal heat pumps. Similarly, there are three different models of biogas plants viz. Janta (Public), Deenbandhu (poor people), and KVIC (Khadi & Village Industry Corporation). On the other hand, only one type of solar collector i.e. flat plate solar collector among four different types of solar collectors (i.e. flat plate, parabolic trough, table:}

| Heating Capacity (kW) | Consumption of Power (kW) | Different costs (in US$) | Fan power consumption depends on the speed and number of cooling towers in use.
|-----------------------|---------------------------|-------------------------|-----------------------------------------------------
|                       | Heat pump | Chiller | Fans* | Capital | Running | Other | Water | Air | Water | Air | Air | Water | Air | Water | Air | Water | Air |
| 1988.8                | 390.0     | 641.5   | 3486.5 | 270     | 119807  | 215653 | 531022 | 590025 |
| 1311.2                | 257.1     | 423.0   | 2651.4 | 450     | 78988   | 142178 | 350099 | 388999 |
| 963.6                 | 188.9     | 310.8   | 2834.1 | 270     | 58048   | 104487 | 257287 | 285875 |
| 917.7                 | 179.9     | 296.0   | 2633.7 | 300     | 55283   | 99510  | 245032 | 272258 |
| 786.6                 | 154.2     | 253.7   | 2651.4 | 450     | 47386   | 85294  | 210027 | 233364 |
| 655.5                 | 128.5     | 211.5   | 2462.7 | 600     | 39488   | 71078  | 175023 | 194470 |
| 563.7                 | 110.5     | 181.8   | 2404.2 | 900     | 33958   | 61124  | 150512 | 167235 |
| 491.7                 | 96.4      | 158.6   | 2462.7 | 600     | 29620   | 53317  | 131212 | 145875 |

1 Fan power consumption depends on the speed and number of cooling towers in use.
2 Running cost is calculated for 8 hrs per day.
3 Other cost is considered to be 3-5% (annually) of the capital cost.
The running cost is much lesser but the investment cost in some cases is found to be the highest when the biogas plant is compared with those of the heat pump and solar collector systems. The running cost for the solar collector is much less followed by the biogas system as compared to those of different heat pump systems and hence, the former case is the cheapest option from the point of view of economics as well as from the point of view of thermodynamics. As we know the solar energy is intermittent in nature and available only in the day time and the building is a commercial that is most of the working hours will be in the day time only. Thus, the application of solar

Table 9 — The mass flow rates in solar collector and storage systems

| Temp. Diff. (*°C*) | Number of Collector | $Q_H = 2734.60$ kW | | $Q_H = 1802.63$ kW | | $Q_H = 901.31$ kW |
|-------------------|---------------------|---------------------|-------------------|---------------------|-------------------|
|                   |                     | Mass flow rate (kg/s) | Mass flow rate (kg/s) | Mass flow rate (kg/s) |                     |
|                   |                     | Water | Air | Water | Air | Water | Air |
| 20                | 10                  | 3.26  | 13.59 | 2.15  | 8.96 | 1.07  | 4.48 |
| 15                | 4.34                | 18.12 |                | 2.86  | 11.95 | 1.43  | 5.97 |
| 10                | 6.51                | 27.18 |                | 4.29  | 17.92 | 2.15  | 8.96 |
| 5                 | 13.02               | 54.37 |                | 8.58  | 35.84 | 4.29  | 17.92 |
| 20                | 4.07                | 16.99 |                | 2.68  | 11.20 | 1.34  | 5.60 |
| 15                | 5.43                | 22.65 |                | 3.58  | 14.93 | 1.79  | 7.47 |
| 8                 | 10                  | 8.14  | 33.98 | 5.36  | 22.40 | 2.68  | 11.20 |
| 5                 | 16.28               | 67.96 |                | 10.73 | 44.80 | 5.36  | 22.40 |
| 20                | 5.43                | 22.65 |                | 3.58  | 14.93 | 1.79  | 7.47 |
| 15                | 7.23                | 30.20 |                | 4.77  | 19.91 | 2.38  | 9.95 |
| 6                 | 10                  | 10.85 | 45.30 | 7.15  | 29.86 | 3.58  | 14.93 |
| 5                 | 21.70               | 90.61 |                | 14.31 | 59.73 | 7.15  | 29.86 |
| 20                | 8.14                | 33.98 |                | 5.36  | 22.40 | 2.68  | 11.20 |
| 15                | 10.85               | 45.30 |                | 7.15  | 29.86 | 3.58  | 14.93 |
| 4                 | 10                  | 16.28 | 67.96 | 10.73 | 44.80 | 5.36  | 22.40 |
| 5                 | 32.55               | 135.91|                | 21.46 | 89.59 | 10.73 | 44.80 |
| 20                | 16.28               | 67.96 |                | 10.73 | 44.80 | 5.36  | 22.40 |
| 15                | 21.70               | 90.61 |                | 14.31 | 59.73 | 7.15  | 29.86 |
| 8                 | 10                  | 32.55 | 135.91| 21.46 | 89.59 | 10.73 | 44.80 |
| 5                 | 65.11               | 271.83|                | 42.92 | 179.19| 21.46 | 89.59 |

Table 10 — Details of different parameters for water and air cooled solar collectors

<table>
<thead>
<tr>
<th>Temp. Diff. (<em>°C</em>)</th>
<th>Consumption of Power (kWh)</th>
<th>Investment + maintenance cost (US$)</th>
<th>Running cost per day (US$)*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Water cooled</td>
<td>Air cooled</td>
<td>Water cooled</td>
</tr>
<tr>
<td>10</td>
<td>12.26</td>
<td>28.44</td>
<td>6000 (20)**</td>
</tr>
<tr>
<td>8</td>
<td>15.33</td>
<td>35.56</td>
<td>5425 (15)</td>
</tr>
<tr>
<td>6</td>
<td>20.44</td>
<td>47.41</td>
<td>3625 (10)</td>
</tr>
<tr>
<td>4</td>
<td>30.66</td>
<td>71.11</td>
<td>1500 (05)</td>
</tr>
<tr>
<td>2</td>
<td>61.32</td>
<td>142.22</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>8.08</td>
<td>18.75</td>
<td>6000 (20)</td>
</tr>
<tr>
<td>8</td>
<td>10.11</td>
<td>23.44</td>
<td>5425 (15)</td>
</tr>
<tr>
<td>6</td>
<td>13.47</td>
<td>31.25</td>
<td>3625 (10)</td>
</tr>
<tr>
<td>4</td>
<td>20.21</td>
<td>46.88</td>
<td>1500 (05)</td>
</tr>
<tr>
<td>2</td>
<td>40.42</td>
<td>93.75</td>
<td></td>
</tr>
</tbody>
</table>

*Power consumption is calculated for 8 hrs per day.

**Number of collectors

heliostats, and parabolic dish) has been considered due to the simplicity, cost and other related parameters. For solar collector system the fuel (the sun light) is free but the storage system (and PCM) is quite expensive.
collectors can be an alternative (if not a substitute) provided the weather condition is not a constraint. On the other hand, in the biogas system the bio fuel (such as cow dung and bio waste like food waste) is also very cheap and can be operated with lesser cost than those of heat pump systems. But the space and other related constraints are to be investigated thoroughly before implementing this idea in present building, especially, at the present place, where land price and space is also a constraint.

The specification of a typical PCM is given in Table 3 while the cost for three main options is given in Table 5. This PCM can be used in the storage system combined with the flat plate solar collector, biogas and heat pump systems. Tables 9 and 10, clearly show that for a large temperature difference either the number of collectors and/or the mass flow rate of the circulating (water/air) fluid is smaller while...
it is reverse in the case of small temperature difference. Also for the same specifications (Table 9), the mass flow rate of water is much lesser than the mass flow rate of the air, which is basically due to the large difference in the specific heats and other physical properties of the two fluids. Thus the water cooled collectors can give a better output provided the optimum configuration is used. Again, for water cooled collectors, the heat transfer process needs a heat exchanger and leads to the loss of energy and higher investment cost. On the other hand, the hot air from the collector can be remixed directly with the exhaust of the cooling tower, as there is no need to maintain the quality of air for such applications. So the extra cost in the form of heat exchanger and due to the energy loss can be saved. Similarly, for biogas system, the biogas can be fired and/or burned at state point 7 as already reported in Ref. (16). Also, the temperature of the refrigerant is much higher at the exit of the compressor i.e. at the entrance of the condenser. So a desuperheater can be used to utilize the heat produced by the condenser, especially, where there is no need of heating the building such as Hong Kong.

4 Conclusions

The detailed comparative economic study for the heat pump, solar collector and biogas systems to control the visible plume from wet cooling towers of a commercial building is presented in this paper. The study has been carried out while considering the possible constraints and the comparison has been given using the total cost, consists of investment and operational costs. For a biogas plant the running cost is much lower however, the investment cost in some cases is found to be very high as compared to those of the heat pump and solar collector systems. The running cost for the solar collector is found to be the least when compared to the biogas system and heat pump systems. The solar collector system has been found to be a better option from the point of view of economics as well as thermodynamics.

As we know that the solar energy is intermittent in nature and available only in the day time and the building is commercial one and most of the working hours fall in the day time only. So the application of solar collectors can be an alternative but may or may not be a substitute, provided weather condition is not a major constraint. On the other hand, in the biogas system the bio fuel (such as cow dung and bio waste like food waste) is also very cheap and can be operated with a low cost in comparison with the heat pump systems. However, the space and other related constraints should also be investigated thoroughly before implementing this idea in building heating systems especially, at the present place, where there is a lack of enough land and open space.

There is a possibility that the exhaust of air cooled system can be remixed with the exhaust of the cooling towers while it is not possible in the case of the water cooled system. In that way, we can also save the investment cost of the heat exchangers besides the energy requirement. Again, as the heat transfer process depends on the medium, so water is a better choice and preferred medium for heating and cooling applications. Also for water cooled systems a continuous circulation is needed. A water fouling treatment is required and hence, extra maintenance cost is incurred.

In biogas plants, the KVIC model is found to be more expensive followed by Janta and then the Deenbandhu. This is because of the structure and materials used in different plants. On the other hand, the running cost of all the three models is found to be the same because the consumption of fuel is equal.

The investment and other costs of an electric heat pump system is lesser than that of the geothermal heat pump system, while the converse is true for the running cost. The running and other costs are different for different systems and fluids. For example, the investment cost and operational costs of water cooled systems in both the cases is lower than those of the air cooled systems.
References
1 Lewis W K, Trans ASME, 44 (1922) 325.
3 Merkel F, VDI Forschungsarbeiten, No. 275, Berlin, Germany, 1925.
12 Wang S W & Tyagi S K, Report on the prediction, potential and control of plume from cooling towers of International Commerce Center, Kowloon, Hong Kong, The Hong Kong Polytechnic University, Hong Kong, 2006.

Appendix

System Simulation and Analysis

A.1. Chiller
The chiller model simulates the chiller performance under various working conditions on the base of impeller tip speed \( u_2 \), impeller exhaust area \( A \), impeller blades angle \( \beta \) and other coefficients/consts. This might be available from chiller technical data and/or can be identified by an associated preprocessor using chiller performance data under full load and partial load. The capacity of the chiller is assumed to be controlled by adjusting the inlet vane angle \( \theta \), the refrigeration cycle of a two-stage centrifugal chiller model was given by Wang et al.\(^\text{18}\). The compressor is modeled on the basis of mass conservation, Euler turbo-machine and energy balance. The Euler equation is modified by considering the impeller exit radial velocity \( c_{2r} \) distribution and given as:

\[
h_{\text{th}} = u_2 \left( 1 - \left( \frac{\pi^2}{8} \right) c_{2r} \right) \left( c_{2r} + B \cdot \frac{v_i}{v_j} \cdot \frac{1}{\cos \theta} \right)
\]

where, \( h_{\text{th}} \) is theoretical head, \( B \) is the ratio of impeller channel depth at intake to that at exhaust, \( v_i \) and \( v_j \) are specific volume at impeller intake and exhaust respectively. Energy balance equations are applied on two control volumes, i.e. compressor control volume and impeller control volume are given by:

\[
h_{\text{th}} = h_{\text{pol.comp}} + h_{\text{hyd.comp}}
\]

where, \( h_{\text{pol}} \) is polytrophic compression work, \( h_{\text{hyd}} \) is hydrodynamic losses, \( c_i \) is the vapour velocity at impeller exhaust. The hydrodynamic losses in two control volumes are considered to be composed of three elements, given by:

\[
h_{\text{pol.imp}} = h_{\text{hyd.imp}} + \frac{c_i^2}{2} \quad \ldots (A3)
\]

\[
h_{\text{pol.comp}} = \frac{1}{\cos \theta} \left( \psi_1 v_i + \psi_2 \frac{v_i}{v_j} \frac{1}{\cos \theta} + \psi_2 \frac{v_i}{v_j} \frac{1}{\cos \theta} \right) \left( \frac{v_i}{v_j} \right)^2 \left( \frac{v_i}{v_j} \right)^2 \quad \ldots (A4)
\]

\[
h_{\text{hyd.imp}} = \frac{1}{\cos \theta} \left( \psi_1 v_i + \psi_2 \frac{v_i}{v_j} \frac{1}{\cos \theta} \right) \left( \frac{v_i}{v_j} \right)^2 \left( \frac{v_i}{v_j} \right)^2 \quad \ldots (A5)
\]

where, \( \zeta \), \( \psi_1 \), \( \psi_2 \), \( \chi \) are the introduced constants. The friction losses are considered to be proportional to the square of the compressor volume flow rate and hence, proportional to the square of the impeller exit radial velocity \( c_{2r} \). The inlet losses are considered to be proportional to the square of the velocity through pre-rotation vanes channel. The incidence losses are considered to be proportional to the square of the shock velocity component. For given evaporator pressure, condenser pressure and position of inlet vanes, the compressor model can calculate the radial velocity and specific volume at impeller exhaust, and hence, the refrigerant mass flow rate and the internal
power consumption. The compressor capacity is controlled by the inlet vanes angle (θ) as given in Eqs (A4) and (A5). By considering the effects of water flow rate (M_{w,ev}, M_{w,cd}) and heat flux (Q_{ev}, Q_{cd}), the evaporator and condenser overall heat transfer coefficient-area products (UA_{ev}, UA_{cd}) are empirically represented by:

\[
UA_{ev} = \left[ C_1M_{w,ev}^{0.8} + C_2Q_{ev}^{-0.745} + C_3 \right]^{-1} \quad \text{(A6)}
\]

\[
UA_{cd} = \left[ C_4M_{w,cd}^{0.8} + C_5Q_{cd}^{0.333} + C_6 \right]^{-1} \quad \text{(A7)}
\]

where, \( C_1-C_6 \) are constants, the evaporation and condensation temperatures, and hence, the evaporator and condenser pressures are calculated, for the given chiller capacity (Q_{ev}), heat rejection (Q_{cd}), chilled and cooling water flow rates and inlet temperatures.

The power consumption of the chiller (W) is calculated on the basis of the compressor internal power consumption (W_{in}). With the consideration of mechanical and leakage losses in the compressor and the electrical and the mechanical losses of the motor, the power consumption of the chiller can be calculated as:

\[
W = \alpha \cdot W_{in} + W_1 \quad \text{(A8)}
\]

where, \( \alpha \) is a coefficient.

A.2 Cooling Tower

The cooling tower model was used to simulate the states of outlet air and outlet water of the cooling tower. In this simulation, the effectiveness model for cooling towers developed by Braun et al.\(^1\) has been used. Based on the steady-state energy and mass balance on an incremental volume, the following differential equations can be derived:

\[
dm_a = m_a d\omega_a \quad \text{(A9)}
\]

\[
dT_w = \frac{dh_a}{dV} - C_{pw} (T_w - T_{ref}) \frac{d\omega_a}{dV} \quad \text{(A10)}
\]

where, \( m_w \) is mass flow rate of water, \( m_a \) is mass flow rate of dry air, \( \omega_a \) is air humidity ratio, \( h_a \) is enthalpy of moist air per mass of dry air, \( C_{pa} \) is specific heat of water at constant pressure, \( T_{ref} \) is reference temperature for zero enthalpy of liquid water. The effectiveness of cooling tower is used to simulate the heat and mass transfer processes in the cooling tower given as:

\[
Q = \varepsilon_a m_a (h_{a,i} - h_{a,o}) \quad \text{(A11)}
\]

where \( \varepsilon_a \) is air side heat transfer effectiveness, \( h_{a,i} \) is the enthalpy of inlet moist air per mass of dry air, \( h_{a,o} \) is the saturation air enthalpy with respect to the inlet temperature of water surface. The outlet air state and water state can be determined through energy balance as given by:

\[
h_{a,o} = h_{a,i} + \varepsilon_a (h_{a,w,i} - h_{a,i}) \quad \text{(A12)}
\]

\[
\omega_{a,o} = \omega_{a,i} + (\omega_{a,i} - \omega_{a,w,o}) \cdot \exp(-NTU) \quad \text{(A13)}
\]

\[
T_{w,o} = T_{ref} + \frac{m_{w,1} (T_{w,i} - T_{ref}) C_{pw} - m_s (h_{a,o} - h_{a,i})}{m_{w,o} C_{pw}} \quad \text{(A14)}
\]

Further details about the heat pump and cooling tower simulations can be found in the concerned references\(^1-5,18-19\).