The study proposes an open-cycle adsorption air-conditioning system, which consists of fixed-bed-type dehumidification units, a plate heat exchanger, and the Maisotsenko cycle (M-cycle) cooling unit. The system produces cool, dry air for air-conditioning purposes, and it can be driven by low-grade thermal energy, such as waste hot water, or solar thermal energy. The paper describes numerical modeling of the dehumidification unit as well as the M-cycle cooling unit, and the performance of the system is analyzed based on the transitional simulation.

**KEY WORDS:** desiccant air conditioning, evaporative cooling, Maisotsenko cycle, natural refrigerant

**I. INTRODUCTION**

Air-conditioning technologies provide us comfortable daily life as well as efficient working environment. On the other hand, houses and buildings consume large amounts of energy in exchange for the mechanically controlled air condition. In the context of global-scale energy and environmental issues, it is imperative to reduce fossil fuel consumption for air conditioning, as well as to replace conventional refrigerants, such as CFCs, HCFCs, and HFCs, with natural or low global warming potential (GWP) refrigerants.

Use of low-grade thermal energy such as waste hot water and solar thermal energy for cooling applications would have a great impact on energy savings. One of the reasons is that solar energy is enormously, in general, when cooling demand is large. Another
reason is cogeneration. Total energy efficiency of cogeneration becomes high, provided that waste heat from prime movers is effectively utilized. In the regions with seasonal variation of climate, it is essential to use waste heat not only for heating applications but also for cooling applications so that annual energy consumption of the overall system is minimized.

Liquid and solid sorption heat pump/refrigeration systems are studied worldwide for the purpose of thermal energy utilization. Among them, solid sorption technologies, such as adsorption heat pumps and solid desiccant systems, are intensively explored by many researchers as solar or low-grade waste-heat-driven air-conditioning systems [Demir et al. (2008), La et al. (2010), Hassan and Mohamad (2012)]. Commercialized adsorption refrigeration systems use silica-gel or zeolite as adsorbent and water as refrigerant. The system produces chilled water of around 10–15°C with a driving heat source temperature of 60–80°C. It is environmentally friendly because neither the adsorbent nor the refriger-
ant are toxic or environmentally harmful, and it recovers useful energy from waste heat. As pointed out by Demir et al. (2008), however, the large volume of the system and the requirement of high-vacuum conditions are major disadvantages of the adsorption systems, which impedes the use of this technology in residential or commercial buildings.

Solid desiccant systems also use adsorbent for removal of moisture from air. With the combination of desiccant dehumidification and evaporative cooling, the system will work similarly to an adsorption heat pump cycle in that the cooling effect is produced by evaporation of water at low partial pressure. Water is the refrigerant of the desiccant-evaporative cooling system, but there is no vacuum issue. Investigations on thermally driven desiccant-evaporative cooling systems can be found in precedent studies. Jain et al. (1995) examined various configurations of desiccant-evaporative cooling cycles by a psychrometric analysis. Their results showed that the performance of the system could be improved by the use of a wet-surface heat exchanger. Goldsworthy and White (2011) analyzed the optimum design of the desiccant cooling system with evaporative cooling, taking into consideration the energy consumption of fans. They showed that the electricity-based COP could be over 20 with the optimized condition under a regeneration temperature of 70°C. A desiccant system with the M-cycle-type indirect evaporative cooler was analyzed by Worek et al. (2012). The advantages of the M-cycle combined with desiccant dehumidification were discussed in their paper.

Although the advantages of the desiccant systems with high-performance evaporative coolers were revealed by these researchers, most of the studies were based on the rotor-type desiccant dehumidifiers. The desiccant rotor has an advantage of no fluctuation of outlet air condition, while the flexibility of the design is less because of the round shape and rotation mechanism. On the other hand, our study focuses on the block-type desiccant unit because of its architectural flexibility and the possibility of integration with various architectural structures. A major disadvantage of the desiccant block unit is that the system requires the batch processing of dehumidification (adsorption) and regeneration (desorption), which causes the periodic variation of the outlet air condition. Therefore, the transitional simulation model of the system was built and the effect of the switching time of dehumidification and regeneration on the system performance was investigated in our study.

II. THEORETICAL ANALYSIS OF OPEN-CYCLE ADSORPTION AIR CONDITIONING

A. Description of the System

The system consists of two desiccant dehumidification units, an M-cycle evaporative cooler, a plate heat exchanger, and a heating coil. The purpose of the system is to control the outdoor air condition before it is supplied to the space for ventilation. The schematic of the system is illustrated in Fig. 1. The outdoor air is dehumidified by the desiccant
unit at first, and then the temperature of the dehumidified air is controlled by the plate heat exchanger and the M-cycle evaporative cooler. On the other hand, the outdoor air in the regeneration airstream is preheated by the exit air from the dehumidification side of the desiccant unit. Although the M-cycle evaporative cooler is capable of cooling a high-temperature dehumidified air toward its dew point, the heat exchange between the dehumidified air and the regeneration air is worthwhile to reduce the heating load of the heating coil. By using waste hot water or hot water from the solar collector, the regeneration air is heated to a required temperature, and after the regeneration of the desiccant unit, it is released to the atmosphere.

B. Theoretical Performance

The performance of the proposed system can be predicted on the psychrometric chart. Figure 2 shows an example of the changes of the air condition on the psychrometric chart. In this case, the temperature of the regeneration air is 80°C, while the temperature of the supply air is 15°C. The ratio of the supply air to the total air of the M-cycle is 0.7, and the heat exchange effectiveness of the plate heat exchanger is 0.8. The cooling capacity of the supply air $q_{SA}$ is expressed as the enthalpy difference between the outdoor air and the supply air as given in Eq. (1):

$$q_{SA} = \gamma m_{DA}(h_{OA} - h_{SA})$$

where $h$ denotes enthalpy, and $m$ denotes the mass flow rate. $\gamma$ is the ratio of the mass flow rate of the supply air to that of the total air of the M-cycle.

On the other hand, the heat input to the heater is given by the enthalpy difference between the regeneration air and the outlet air from the plate heat exchanger, as shown in Eq. (2):

$$q_{HT} = m_{RA}(h_{RA} - h_{HT})$$

The COP of this system is defined as $COP = q_{SA}/q_{HT}$. The calculated COP value is 0.76 in the case of the conditions shown in Figure 2.

**FIG. 1:** Schematic of the open-cycle adsorption air-conditioning system
Figure 3 shows the theoretical COP as a function of the regeneration temperature. It was shown that the theoretical COP increased with lower regeneration temperature, and it achieved COP = 1 with the regeneration temperature of 70°C. This result showed a promising potential of the heat-driven air-conditioning system. To attain this high performance, however, extremely long flow channel length of the desiccant unit would be required, and it might not be feasible from an engineering point of view. Therefore, the practical performance of the system should be evaluated by a detailed simulation.
III. THE MATHEMATICAL MODEL

The performance analysis of the fixed-type desiccant unit requires transitional simulation, because the condition of outlet air from the desiccant unit changes with time due to the variation in water adsorption quantity of the desiccant material. Therefore, the mathematical models of the desiccant unit and the M-cycle evaporative cooler were developed for simulation analysis.

A. Model of the Desiccant Unit

The mathematical model of the desiccant unit is similar to the one used by Elsayed et al. (2006). The model used the approximation to a flat plate for a honeycomb structure of a desiccant rotor. The same approximated model is valid for our purpose, because we assumed a stack of desiccant sheets as a desiccant unit as depicted in Figure 4.

The heat and mass balances of air and desiccant material are formulated as follows. Equations (3) and (4) show the heat and mass balances of the air side. The heat balance of the desiccant material is given by Eq. (5), while the adsorption kinetics is expressed by Eq. (6):

\[
\frac{\partial T_a}{\partial t} = -u_a \frac{\partial T_a}{\partial z} + \frac{\alpha_a}{\rho_a c_a a_a} (T_b - T_a)
\]

\[
\frac{\partial X_a}{\partial t} = -u_a \frac{\partial X_a}{\partial z} - \frac{\rho_b a_b \partial w_b}{\rho_a a_a} \frac{\partial t}{\partial t}
\]

\[
\frac{\partial T_b}{\partial t} = \frac{\alpha_a}{\rho_b c_b a_b} (T_a - T_b) + \frac{Q_b}{c_b} \frac{\partial w_b}{\partial t}
\]

\[
\frac{\partial w_b}{\partial t} = \frac{\alpha_b}{a_b} (w_b^* - w_b)
\]

where \( T, X, \) and \( w \) denote temperature, humidity ratio, and adsorption quantity, respectively. \( t \) is time and \( z \) is position in flow direction. \( \rho \) and \( c \) are density and specific heat. The subscripts \( a \) and \( b \) represent air and desiccant material, respectively. \( a_a \) and \( a_b \) are

FIG. 4: Illustrations of the desiccant unit and the M-cycle unit
the half thickness of the air layer and the desiccant layer, respectively, and \( u_a \) is the air velocity in the channel. \( \alpha_a \) gives the convection heat-transfer coefficient between the air and the surface of the desiccant sheet, and \( \alpha'_b \) gives the equivalent mass-transfer coefficient. \( Q_b \) denotes the adsorption heat. \( w^* \) represents the equilibrium adsorption, and it is given as a function of the relative humidity as Eq. (7) (Hamamoto et al., 2002),

\[
w^* = \frac{1.7(p_w/p_s)}{1 + 8.5(p_w/p_s)}
\]

where \( p_w \) denotes the partial pressure of the water vapor, and \( p_s \) denotes the saturation pressure of water at desiccant material temperature.

The same mathematical models on a variety of desiccant materials were used by the authors’ previous work, and it was shown that the model reasonably agreed with the time variations of the experimental results (Miyazaki et al., 2009).

B. Model of the M-cycle Evaporative Cooler

The mathematical model of the counterflow-type M-cycle evaporative cooler was developed based on similar heat and mass balance considerations as a desiccant model. The temperature and humidity changes in a minute section of the dry and wet channels were modeled as shown in Figure 5.

\[
\frac{\partial T_d}{\partial t} = -u_d \frac{\partial T_d}{\partial z} + \frac{\alpha_d}{\rho_d c_d a_d} (T_p - T_d)
\]

**FIG. 5:** Heat and mass transfer in a minute section of the counterflow-type M-cycle
\[ \frac{\partial T_p}{\partial t} = \frac{\alpha_d}{\rho_p c_p a_p} (T_d - T_p) + \frac{k_e}{\rho_p c_p a_p} (T_s - T_p) \] (9)

\[ \frac{\partial T_s}{\partial t} = \frac{k_e}{\rho_s c_s a_s} (T_p - T_s) + \frac{\alpha_w}{\rho_s c_s a_s} (T_w - T_s) - \frac{\rho_w h_w r}{\rho_s c_s a_s} (X_w^* - X_w) \] (10)

\[ \frac{\partial T_w}{\partial t} = -u_w \frac{\partial T_w}{\partial z} + \frac{\alpha_w}{\rho_w c_w a_w} (T_s - T_w) \] (11)

\[ \frac{\partial X_w}{\partial t} = -u_w \frac{\partial X_w}{\partial z} + \frac{\alpha_w}{\alpha_e} (X_w^* - X_w) \] (12)

where the subscripts \(d, p, s\), and \(w\) indicate dry channel, plate, water-absorbing sheet, and wet channel; \(a_e\) and \(k_e\) represent total thickness and the equivalent thermal conductivity through the plate and the water absorbing sheet, respectively, defined as \(a_e = a_p + a_s\), \(a_e/k_e = a_p/k_p + a_s/k_s\); and \(X^*\) is the humidity ratio in saturation at the water-absorbing sheet temperature.

The convection heat-transfer coefficients in dry and wet channels were obtained by the Nusselt number with uniform heat flux for both the thermal entrance region and the fully developed region using Eq. (13) (JSME, 2009),

\[
\text{Nu}(z) = 5.364 \left[ 1 + z^* - \frac{10}{\pi} \right]^{\frac{3}{10}} \left[ 1 + \left( \frac{\pi/(115.2z^*)}{[1 + \left( \frac{\text{Pr}}{0.0207} \right)^{\frac{1}{2}}]^{\frac{1}{2}} [1 + z^* - \frac{10}{\pi}]^{\frac{1}{2}}} \right)^{\frac{2}{3}} \right]^{\frac{3}{10}}
\] (13)

where, \(z^* = (z/d)/(\text{Re Pr})\) and \(z^* = 220 z^*/\pi\).

The mass-transfer coefficient in the wet channel was calculated based on the assumption of \(\text{Le} = 1\).

**C. Comparison between Simulation and Experiment of the M-Cycle**

The cooling performance of the M-cycle evaporative cooler predicted by simulation was compared with that obtained by experiment. The counterflow-type M-cycle experimental unit consisted of a stuck of aluminum plates (100 x 640 mm, 0.5 mm thickness) and spacers (5 x 10 mm rectangular-shape aluminum bar). The geometries of the dry and wet channels are depicted in Figure 6. For the wet channel, a sheet of water-absorbing cloth was attached on the aluminum plate, and water was supplied through vinyl tubes on both sides.

Figure 7 shows the experimental apparatus. The air flow rate was measured at the inlet and exit of the dry channel by KEYENCE FD-A250 (±1% of F.S. at 25°C), and the humidity ratio was measured at the same points by Hygroclip (±0.3 K in temperature, ±1.5% in relative humidity). The temperatures inside the dry and wet channels were measured by using alumel-chromel wire thermocouples.
The temperature variations of the air inside the dry channel and inside the wet channels are depicted in Figure 8. Four graphs illustrate the difference of the air flow ratio γ. The horizontal axis is normalized by the flow channel length, and the vertical axis is normalized by the maximum cooling, which is equal to the temperature difference between the inlet air to the M-cycle and the dew point of the inlet air ($T_{MC} - T_{dp}$).

The results showed discrepancies between the simulation and the measurement, which would be caused by incompleteness of the experimental apparatus in terms of air leakage and insulation. The simulation predicted, however, a similar temperature fall by the M-cycle.

Figure 9 shows a comparison of the dew-point effectiveness as a function of the supply air flow ratio between the experiment and the simulation. The dew-point effectiveness was defined as
The supply air temperature of the experiment was assumed as an average value of the dry channel and wet channel at $z/L = 1$. The dew-point effectiveness was mitigated with increase in supply air flow ratio. Both experimental and simulation results showed a similar trend, even though the simulation predicted slightly better dew-point effectiveness. The difference between the experiment and simulation would be due to the leakage.
of the air, heat losses and nonuniform flow distribution in channels. It is fair to say that
the simulation predicted the performance of the M-cycle with reasonable agreement.

IV. SIMULATION ANALYSIS

A practical performance of the proposed system was predicted by the simulation de-
scribed in the previous section. The geometry of the system and the operating conditions
were summarized as shown in Tables 1 and 2.

A. Transitional Characteristics

The time variations of the temperature and humidity ratio are given in Figure 10. In this
case, the desiccant units changed their operation between dehumidification and regener-
ation with every 1200 s. Therefore the temperature and humidity ratio changed cyclically
with the period of 1200 s. It was observed that the humidity ratio of the dehumidified
air was gradually approached to supply air humidity of around 10 g/kg. The temperature
of the desiccant material was still near the regeneration temperature just after the regen-
eration mode, which caused the low dehumidification performance in the beginning of
the dehumidification mode. The supply air humidity ratio of approximately 10 g/kg was
maintained after 200 s. The temperature of the dehumidified air was raised to about 60°C
because of the dehumidification, and it was cooled to around 40°C by the plate heat ex-
changer. Even though the variations of temperature and humidity ratio were significant

<table>
<thead>
<tr>
<th>TABLE 1: Geometry of the system</th>
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<tbody>
<tr>
<td>Desiccant unit</td>
</tr>
<tr>
<td>Desiccant material thickness</td>
</tr>
<tr>
<td>The number of flow paths</td>
</tr>
<tr>
<td>The number of desiccant units</td>
</tr>
<tr>
<td>M-cycle unit</td>
</tr>
<tr>
<td>Length, width, and height of the dry channel</td>
</tr>
<tr>
<td>Length, width, and height of the wet channel</td>
</tr>
<tr>
<td>The number of flow paths</td>
</tr>
<tr>
<td>Supply air flow ratio</td>
</tr>
<tr>
<td>Flow type</td>
</tr>
<tr>
<td>Plate HEX</td>
</tr>
<tr>
<td>Overall heat conductance</td>
</tr>
<tr>
<td>Length, width, and height of air flow channel</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TABLE 2: The operating conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outdoor air 32°C, RH 60%</td>
</tr>
<tr>
<td>Volume flow rate 80 m³/h</td>
</tr>
<tr>
<td>Regeneration temperature 80°C</td>
</tr>
<tr>
<td>Switching time of desiccant unit 1200 s (600–3000 s)</td>
</tr>
</tbody>
</table>
in the beginning of the dehumidification period (up to 200 s), the supply air temperature was fairly stable due to a sufficient flow length of the M-cycle for cooling the inlet temperature to near its dew point.

B. Effect of the Switching Time

Figure 11 shows the variation of the humidity ratio of the supply air with the switching time of 600, 1200, 1800, 2400, and 3000 s. It was found that the minimum humidity ratio during the dehumidification period became lower with longer switching time. That was because the desiccant unit was in a dryer condition with a long regeneration period. On the other hand, the variation of the humidity ratio was larger with longer switching time, which would be unfavorable in terms of supply air stability.

FIG. 10: Time variations of the temperature and humidity ratio with the switching time of 1200 s: (a) temperature and (b) humidity ratio

FIG. 11: Effect of switching time on the variation of the humidity ratio of the supply air.
TABLE 3: Thermodynamic COP of the system

<table>
<thead>
<tr>
<th>Switching time</th>
<th>600 s</th>
<th>1200 s</th>
<th>1800 s</th>
<th>2400 s</th>
<th>3000 s</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP</td>
<td>0.69</td>
<td>0.74</td>
<td>0.75</td>
<td>0.74</td>
<td>0.72</td>
</tr>
</tbody>
</table>

The thermodynamic COP was calculated from the average value of the air condition during the cyclic operation. The results are given in Table 3. The COP was low when the switching time was short, because the influence of the low dehumidification period (~200 s) would be dominant on the performance, while the COP was also decreased with extremely long switching time due to the limitation of the dehumidification capacity of the desiccant unit. Consequently, the optimum switching time to maximize the COP existed, which was around 1800 s under the geometry and operating conditions of our case. Along with the COP value predicted by the theoretical analysis described in Section 2, the transitional simulation results confirmed that the thermodynamic COP of the system could be higher than 0.7 with the regeneration temperature of 80°C.

V. CONCLUSIONS

The study proposes an open-cycle adsorption air-conditioning system, which consists of a desiccant dehumidification unit and the M-cycle evaporative cooler as main components. The performance of the system was investigated by the transitional simulation. The results revealed the influence of the switching time on the variation of the supply air condition as well as the thermodynamic COP. It was found that the thermodynamic COP was maximized with the optimum switching time. The predicted COP was as high as 0.75 with a regeneration temperature of 80°C. It was concluded that the proposed system had a promising COP value as a waste-heat-driven air-conditioning system. Future works are the optimization of the geometry of the dehumidification unit and the M-cycle evaporative cooling unit, taking into account the pressure drop inside the flow channels so that the total system COP including fans could be maximized.

REFERENCES


