A study on energy saving of cooling/reheating system using compact heat exchanger †

Seong-Yeon Yoo[1], Jin-Hyuck Kim[2,*] and Myoung-Seok Jie[3]

[1]BK21 Mechatronics Group, Chungnam National University, Daejeon 305-764, Korea
[2]Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, Daejeon 305-701, Korea
[3]Ajou Motor College, Boryeong, Chungcheongnamdo, 355-769, Korea

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Abstract

When circulated air passes through the cooling coil in an air-conditioning system, the air is over-cooled to eliminate the moisture and decrease the temperature. The cooled air is then reheated to recover the temperature. The purpose of the present study was to evaluate the performance of a cooling/reheating system with regard to both cooling and reheating energy savings affected by exchanging heat between the cooled air and the reheated air with a compact heat exchanger. The thermal and dehumidification behaviors of the system were evaluated experimentally and then compared with simulation data. The results show that the energy saving rate was as high as 50% under the present experimental conditions and was affected by the face velocity of the heat exchanger, the inlet temperature, the inlet humidity ratio, and the effectiveness of heat exchanger. Furthermore, the experimental data were found to be in fairly good agreement with the simulation data.

Keywords: Cooing/reheating system; Energy saving; Compact heat exchanger

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1. Introduction

In order to obtain the desired indoor air temperature and humidity, not only the latent heat load but also sensible have to be controlled. Conventional air-conditioning systems, in controlling indoor temperature and moisture under high-temperature and high-humidity conditions, normally utilize an over-cooled cooling coil and an additional reheating heater. The system modified in the present study can save both cooling energy and reheating energy by means of a compact heat exchanger.

Aluminum-plate heat exchangers are widely used in air-conditioning systems for heat recovery, but nonmetallic materials also have been adopted for the same purpose. Plastic is perhaps the most promising material for this purpose, owing to its low cost, light weight, and ease of processing. Therefore, the present research tested the performance of a cooling/reheating system equipped with a plastic heat exchanger.

Yoo et al. [1, 2] developed and tested five different plastic (polypropylene)-plate heat exchanger prototypes. The test results showed that in comparison with the flat-plate type at Reynolds number 2500, the overall performance of the corrugate type, the turbulent promoter type, and the dimple type, in which both the heat transfer coefficient and pressure drop were considered, increased by 43, 14 and 33%, respectively. Recently, surfaces imprinted with dimples and other elements have been researched extensively. One of the early investigations was conducted by Afansayev et al. [3], who investigated the effect of a shallow-dimple flat-plate application on overall heat transfer and pressure drop. Significant heat transfer augmentation (30-40%) for negligible pressure drop was reported.

Saman and Alizadeh [4, 5] researched the thermal and dehumidification behaviors of a standard cross-flow-type dimpled-surface plastic-plate heat exchanger intended for use as a dehumidifier/cooler. The results showed that the thermal effectiveness and dehumidification efficiency of the heat exchanger increase with increasing inlet temperatures and humidity ratios.

Much research has been conducted with regard to air-conditioning system relevant control methods under various conditions. Zhang et al. [6, 7] used temperature-humidity control strategies in the analysis of the stability of the heating and ventilation control system of a building, and found that an upper temperature limit to deactivate the heater for temperature-humidity control should be devised; otherwise instability would occur when the weather is both warm and humid condition.
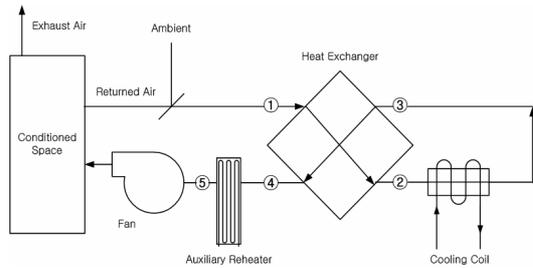

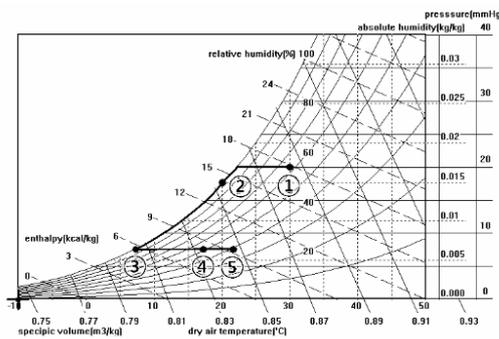Fig. 1. Schematic of cooling/reheating system using plastic heat exchanger.

Fig. 2. Progress of cooling/reheating system on psychometric chart.

Chen et al. [8] reported the results for field measurements of an independent dehumidification system, showing that the system was able to produce a comfortable indoor environment, even in very humid weather. Comparisons of independent and conventional systems indicate that the two types have equivalent distribution power consumptions, but that the system incorporating a heat exchanger can save 30% in cooling-source costs.

Huh et al. [9] described research into the optimal operation of HVAC systems, focusing on both temperature and humidity control. Over the years, moreover, there have been many additional studies focusing on the dehumidification performance of the HVAC system [10-12].

Recently, Mazzei et al. [13] conducted a critical review regarding the thermal comfort of the HVAC dehumidification system.

In this paper, a novel air-conditioning system, specifically a cooling/reheating system incorporating a plastic heat exchanger, is proposed. Thermodynamic modeling and performance experiments were performed. The experimental results were compared with the simulation data, and the system's energy saving rate was considered.

2. System description and mathematical modeling

2.1 System description

The A schematic of the air-conditioning system modeled in this work is shown in Fig. 1, while the system's psychometrics are illustrated in Fig. 2. In this system, a plastic heat exchanger is used to pre-cool and dehumidify hot and humid outdoor air

(point 1) before it is pumped to a cooling coil and reaches point 2. Subsequently the air is cooled and dehumidified completely by the cooling coil at point 3 before being returned to the heat exchanger at point 4. After the air is reheated, it is supplied to the indoors. However, in the case that the reheating does not sufficiently heat the air, the auxiliary reheater is used to make room temperature relevant at point 5.

2.2 Component modeling

The energy performance and conditions of the air at each point of the system were simulated by thermodynamic equations. The conditions of the air can be calculated with reference to temperature, relative humidity, the humidity ratio, enthalpy, and other parameters.

The condition of the air at point 2, after pre cooling and moisture removal, was calculated according to Eqs. (1), (2) and (3) below. Because the effectiveness of the heat exchanger varies with velocity, if, at point 2, the enthalpy is lower than the saturation enthalpy, the relative humidity will reach 100%. If, however, it fails to reach the saturated condition, the specific humidity at point 2 is the same as at point 1.

$$\varepsilon_{H.E.} = f(V) = \frac{\dot{m}(h_1 - h_2)}{\dot{m}C_p(T_1 - T_3)} \tag{1}$$

$$\phi_2 = 100\%(h_2 > h_s) \tag{2}$$

$$W_2 = W_1(h_2 > h_s) \tag{3}$$

The condition of the air after passing through the cooling coil in an ideal cooling/reheating system is determined by a proper indoor humidity ratio. Therefore, as shown in Eq. (4), the humidity ratio at point 3 is the same as the desired condition. However, most systems cannot perfectly maintain a relevant indoor humidity ratio; the effectiveness of the cooling coil can be used to determine the condition of the air. In addition, as the cooling and dehumidification process is completed through the cooling coil according to the saturated water vapor curve, the condition at the outlet includes a relative humidity of 100%.

$$W_3 = W_d \tag{4}$$

$$\varepsilon_{C.C.} = f(V) = \frac{\dot{m}(h_2 - h_3)}{\dot{m}C_p(T_2 - T_{C.C.})} \tag{5}$$

$$\phi_3 = 100\% \tag{6}$$

The enthalpy after the over-cooled air passes through the heat exchanger can be obtained using an equation similar to that used for the pre-cooling process through the heat exchanger mentioned earlier. In addition, as there is no mass transfer, the air will have the same humidity ratio as the rear side of the cooling coil.

$$\varepsilon_{H.E.} = f(V) = \frac{\dot{m}(h_4 - h_3)}{\dot{m}C_p(T_1 - T_3)} \tag{7}$$
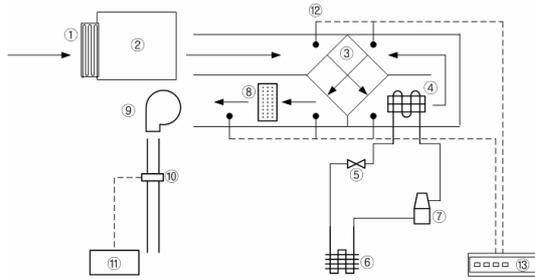
1: electric heater, 2: mixing chamber, 3: plastic heat exchanger,
4: cooling coil (evaporator), 5: expansion valve, 6: condenser,
7: compressor, 8: reheating coil (heater), 9: fan, 10: nozzle,
11: micro manometer, 12: temperature & humidity sensor,
13: data acquisition system

Fig. 3. Schematic of experimental apparatus.

$$W_4 = W_3 \qquad (8)$$

when reheating through the heat exchanger is not sufficient, the temperature is raised up to proper temperature by the auxiliary reheater. Furthermore, the humidity ratio at this time is the same as after the air passes through the heat exchanger.

$$T_5 = T_d \qquad (9)$$
$$W_5 = W_4 = W_d \qquad (10)$$

3. Experimental apparatus and procedure

3.1 Experimental apparatus

An experimental apparatus for the cooling/reheating system as designed and fabricated is shown in the Fig. 3. It consisted of a fan, ducts, a plastic-plate heat exchanger, a cooling coil, electric heaters, and a data acquisition system. The dimple-type plastic heat exchanger was employed. Two differently sized dimples, of 8 mm and 2 mm diameters, were alternately spaced at 25 mm intervals. The heat transfer surface had an area of 160 x 160 mm^2, and there was a 4 mm gap between the two plates.

The typical state of atmospheric air in summer was assigned as the general experimental condition. The electrical heater was installed to correct the temperature, and the air was to inflow after being mixed to uniformity in an acrylic chamber. The cooling coil used in the system was a six-column, pin-tube-type direct-expansion coil taken from a refrigerator. The capacities of the fan and cooling coil were varied by an inverter. A screen-mesh-type electric heater was used to help maintain the proper conditions.

A sonic nozzle installed at the outlet of the fan functioned as the flow meter. Calibrated T-type thermocouples and humidity sensors were inserted into the duct walls to measure the temperature and relative humidity.

3.2 Experimental procedure
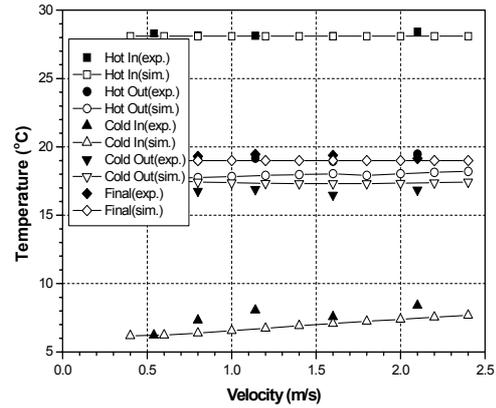
The experiments were carried out under general operation
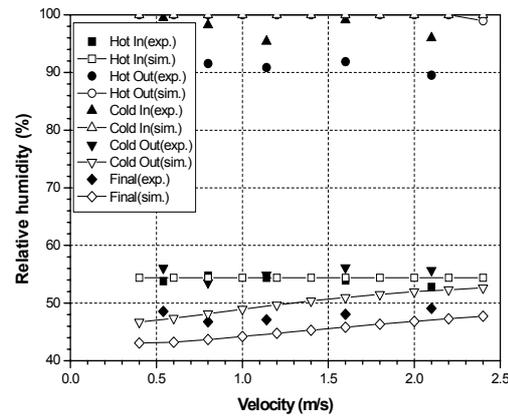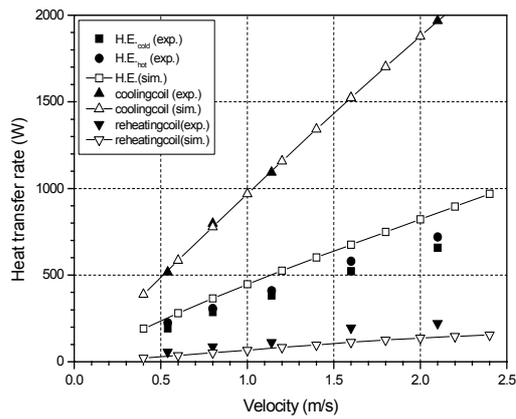

Fig. 4. Temperature with various velocities.


Fig. 5. Relative humidity with various velocities.

conditions, which were a dry bulb temperature of 28℃ and a wet bulb temperature of 23.5℃. In this case, the relative humidity was 68%, and the humidity ratio was 0.0164kg/kg. The face velocity of the heat exchanger was set at 2m/s. The desired indoor-supply condition was set to a temperature of 19℃ and a relative humidity of 50%. The capacity of the cooling coil was determined by the desired humidity ratio, and the compressor was partly adjusted so that a certain condition could be maintained. When reheating through the heat exchanger was not sufficient, the temperature was raised using an auxiliary reheater.

4. Results and discussion

Fig. 4 shows the temperature distribution of the system with the various velocities. In the legend, Hot In represents point 1, Hot Out point 2, Cold In point 3, Cold Out point 4, and Final (the final condition) point 5. According to the mathematical modeling mentioned earlier, when the desired final condition was constant, the temperature at the rear of the cooling coil was determined based on that final condition. In this study then, when the proper indoor-supply condition was 19℃ and 50% relative humidity, the temperature passing through the cooling coil should have been constantly maintained at 8℃. However, as the compressor capacity of the cooling device
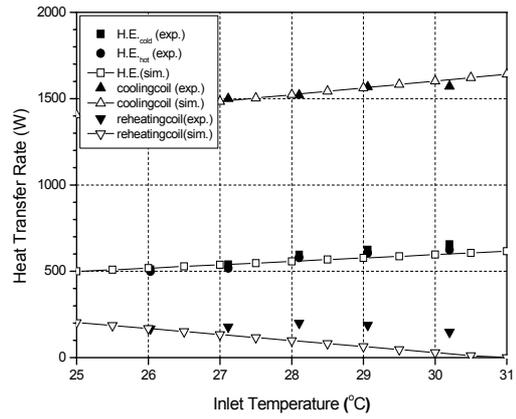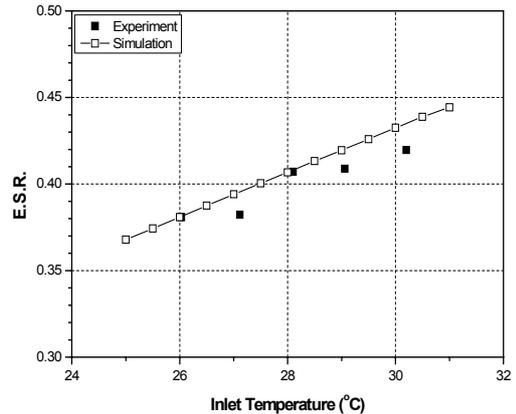
(a) Heat transfer rate

(a) Heat transfer rate
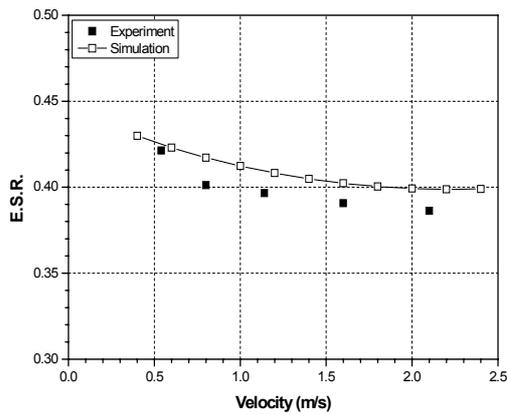
(b) Energy saving rate

Fig. 6. Heat transfer rate and E.S.R. with various velocities.

(b) Energy saving rate

Fig. 7. Heat transfer rate and E.S.R. with various inlet temperatures.

could not be controlled sufficiently to meet the desired level, the appropriate temperature value was not maintained. It was found that as the face velocity at the front of heat exchanger increased, the effectiveness of the heat exchanger decreased and the temperature at the rear of the cooling coil decreased, due to insufficient capacity. Therefore, temperature differences between the hot and the cold side of the heat exchanger decreased with increasing velocities.

Fig. 5 depicts the relative humidity distribution of the system with increasing velocities. According to the simulation data, the temperature of the air passing through the heat exchanger should have been lower than the saturation temperature for all of the humidity ranges in the experiment. However, according to the experimental results, the relative humidity was shown to be between 90 and 95%. The internal temperature in the plastic heat exchanger was not constant, but showed locally different temperature distributions. As the entire surface of the heat exchanger did not reach condensation temperatures, some air was not condensed and, consequently, was passed through. This phenomenon is thought to be similar to the bypass effect in the cooling coil. As shown in the Figure, the relative humidity of the air passing through the cooling coil was lower than the saturation humidity. In the case of the six-column pin-coiled heat exchanger used in this

research, the bypassed air amount was given as approximately 5% [14].

In order to evaluate the performance of the system, Fig. 6 shows the heat transfer rate and the energy saving rate. The heat transfer rates in the heat exchanger cooling coil and the auxiliary reheating coil are given in the following equation.

$$\dot{Q}_h = \dot{m}(h_1 - h_2) \tag{11}$$

$$\dot{Q}_{C.C.} = \dot{m}(h_2 - h_3) \tag{12}$$

$$\dot{Q}_c = \dot{m}(h_4 - h_3) \tag{13}$$

$$\dot{Q}_{re} = \dot{m}(h_5 - h_4) \tag{14}$$

In addition, the ratio of the heat transfer of heat exchanger to that of the entire system can be shown in the following equation.

$$E.S.R. = \frac{\dot{Q}_{H.E.}}{\dot{Q}_{System}} = \frac{\dot{Q}_h + \dot{Q}_c}{\dot{Q}_h + \dot{Q}_{C.C.} + \dot{Q}_c + \dot{Q}_{re}} \tag{15}$$
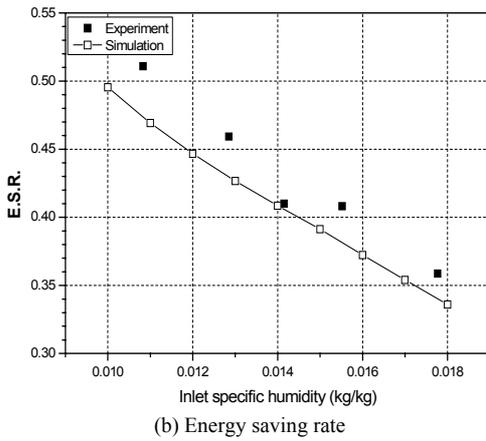
As the velocity increased, the effectiveness of the heat exchanger was reduced and the heat transfer rate from the heat exchanger decreased. The heat exchanger undertakes a great amount of cooling and reheating. This necessitates an incre-
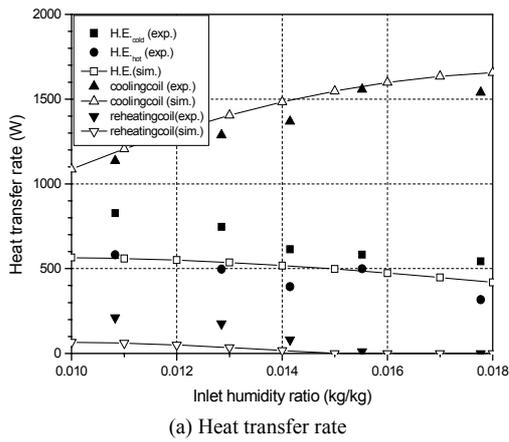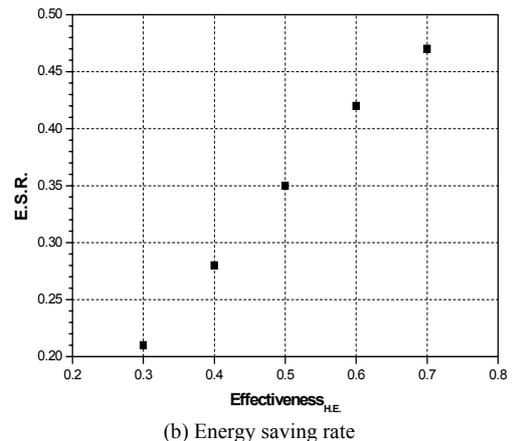
(a) Heat transfer rate
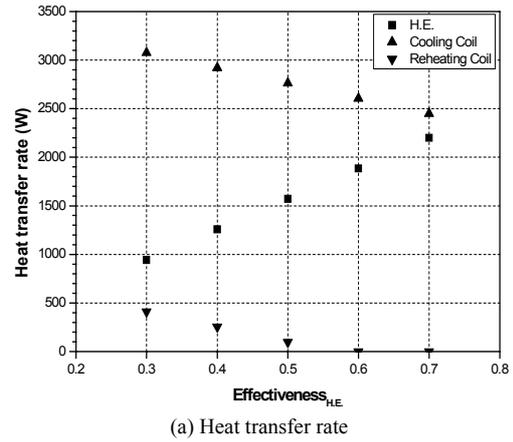
(b) Energy saving rate

Fig. 8. Heat transfer rate and E.S.R. with various inlet humidity ratios.

(a) Heat transfer rate

(b) Energy saving rate

Fig. 9. Heat transfer rate and E.S.R. with various effectiveness of heat exchanger.

ment of heat transfer through the cooling coil as the velocity increases. If, therefore, the final temperature was below the desired condition, the temperature was raised through the auxiliary reheater, and the auxiliary reheating gradually increased in accordance with the increase in velocity. The energy saving rate, not surprisingly then, decreased with the increasing velocity: the energy saving rate at the face velocity of 2 m/s was about 39%. As there was not enough insulation, and as the auxiliary reheating was little more than the appropriate value, the measured data figures were slightly lower than those for the simulation data.

According to the results shown in Fig. 7, the heat transfer rate increased as the air passed through both the cooling coil and the heat exchanger. As the increment through the heat exchanger was larger than through the cooling coil and reheating diminished, the energy saving rate decreased from 42 to 38%. Significantly, the experimental data showed fairly good agreement with the simulation data.

Fig. 8 plots the performance changes of the system as the inlet humidity ratio increased from 0.01 to 0.018kg/kg. As the demand for dehumidification increased due to increment of inlet humidity, the heat transfer rate through the cooling coil was shown to have increased and the temperature at the rear to have been lowered. Consequently, the temperature difference

between the hot and cold sides of the heat exchanger decreased and the heat transfer rate through the heat exchanger was reduced. Furthermore, the energy saving rate was suddenly reduced from 51 to 36%. It can be concluded that the performance of the cooling/reheating system incorporating a plastic heat exchanger is affected mainly by the inlet humidity ratio.

Fig. 9 shows the performance of the system with the change of the effectiveness of the heat exchanger, in the simulation only. As the effectiveness of the plastic heat exchanger increased, the heat transfer rate through the cooling coil was significantly reduced and that through the heat exchanger was increased. Because the heat exchanger assumes the lion's share of the cooling and reheating task, as mentioned previously, the system can save more energy with a highly effective heat exchanger like that shown in Fig. 9(b).

5. Conclusion

In this study, the thermal and dehumidification performance of a cooling/reheating system incorporating a compact heat exchanger was determined under various conditions. Furthermore, the energy saving rate was considered. The energy saving rate was shown to be as high as 50% under the present

conditions, and was clearly affected by the face velocity at the front of the heat exchanger, the inlet temperature, the inlet humidity ratio, and the effectiveness of the heat exchanger. When the velocity increased, the energy saving rate was reduced due to decease in the effectiveness of the heat exchanger. The energy saving rate increased with increasing inlet temperature as the temperature differences between the hot and the cold sides of the fluid increased and the demand for auxiliary reheating decreased. As the demand for dehumidification increased, the energy saving rate decreased with increasing humidity. Furthermore, the energy saving rate in the simulation was significantly influenced by the effectiveness of the heat exchanger. All the measured data under the three conditions were fairly consistent with the simulation data.

Acknowledgment

Nomenclature————————————————

C_p	:	Specific Heat
$E.S.R.$	:	Energy saving rate
h	:	Enthalpy
$\dot{m}$	:	Mass flow rate
$\dot{Q}_c$	:	Heat transfer rate through cold side of heat exchanger
$\dot{Q}_{C.C.}$	:	Heat transfer rate through cooling coil
$\dot{Q}_h$	:	Heat transfer rate through hot side of heat exchanger
$\dot{Q}_{re}$	:	Heat transfer rate through reheating device
W	:	Humidity ratio

Greek symbols

ε	:	Effectiveness
ϕ	:	Relative humidity

References

[1] S. Y. Yoo, M. H. Jung, K. H. Kim and Y. M. Lee, An Experimental Study on the Performance of Plastic Plate Heat Exchanger, *Korean Journal of Air-Conditioning and Refrigeration Engineering*, 17 (2) (2005) 117-124.

[2] S. Y. Yoo, M. H. Jung and Y. M. Lee, A Study on the Factors Affecting the Performance of Plastic Plate Heat Exchanger, *Korean Journal of Air-Conditioning and Refrigeration Engineering*, 17 (9) (2005) 839-848.

[3] V. N. Afanasyev, Ya. P. Chudnovsky, A. I. Leontiev and P. S. Roganov, Turbulent flow friction and heat transfer characteristics for spherical cavities on a flat plate, *Exp. Thermal Fluid Science*, 7 (1) (1993) 1-7.

[4] W. Y. Saman and S. Alizadeh, Modelling and Performance Analysis of a Cross-flow Type Plate Heat Exchanger for Dehumidification/cooling, *Solar Energy*, 70 (4) (2001) 361-372.

[5] W. Y. Saman and S. Alizadeh, An Experimental Study of a Cross-flow Type Plate Heat Exchanger for Dehumidification/cooling, *Solar Energy*, 73 (1) (2002) 59-71.

[6] L. Zhang, D. Zhu, X. Deng and B. Hua, Thermodynamic Modeling of a Novel Air Dehumidification System, *Energy and Building*, 37 (3) (2005) 279-286.

[7] L. Zhang, Energy Performance of Independent Air Dehumidification Systems with Energy Recovery Measures, *Energy*, 31 (8) (2006) 1228-1242.

[8] X. Chen and Y. Jiang, Field Study on Independent Dehumidification Air-Conditioning System-II: Performance of the Whole System, *ASHRAE Trans.*, 111 (2) (2005) 277-284.

[9] J. H. Huh and M. J. Brandemuehl, Optimization of air-conditioning system operating strategies for hot and humid climates, *Energy and Buildings*, 40 (7) (2008) 1202-1213.

[10] D. B. Shirey, Demonstration of efficient humidity control techniques at an art museum, *ASHRAE Trans.*, 99 (1) (1993) 93-102.

[11] S. A. Mumma, Designing dedicated outdoor air systems, *ASHRAE Journal*, 43 (5) (2001) 28-31.

[12] J. Murphy, Dehumidification performance of HVAC systems, *ASHRAE Journal*, 44 (3) (2002) 23-28.

[13] P. Mazzei, F. Minichiello and D. Palma, HVAC dehumidification systems for thermal comport: a critical review, *Applied Thermal Engineering*, 25 (2005) 677-707.

[14] The society if heating, air conditioning and sanitary engineers of Japan, *Handbook*, 12th ed. (1995) 274-281.

Seong-Yeon Yoo received his B.S. degree from Seoul National University, Korea, in 1977. He obtained his M.S. and Ph.D. degrees in Mechanical Engineering from Korea Advanced Institute of Science and Technology in 1979 and 1989, respectively. Dr. Yoo is currently a Professor at Chungnam National University, Korea, and serves as a Vice-president in KSME. His research interest is thermal system design such as heat exchanger, air-conditioning system, ventilation system, etc.

Jin-Hyuck Kim received his B.S., M.S. and Ph.D. degree in Mechanical Design Engineering from Chungnam National University, Korea, in 2004, 2006 and 2009, respectively. Dr. Kim is currently working in Mechanical Engineering Research Institute of Korea Advanced Institute of Science and Technology as a Post Dr. He is a member of ASHRAE, KSME and SAREK. His research interest is designing and performance analyzing the thermal system.