EXPERIMENTAL STUDY ON CASCADE HEAT PUMP DRYER WITH A SOLAR COLLECTOR UNDER LOW TEMPERATURE OUTDOOR AIR ENVIRONMENT

by

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A heat pump dryer can save more energy than other dryers since its drying efficiency is 2-3 times higher than that of other types of dryers. However, the lower bound of evaporating temperature for an R134a heat pump cycle ranges from 5 to 10 °C, when the outdoor air temperature closely approaches the evaporating temperature, it experiences reduced efficiency and ultimately becomes inoperable. To address this issue, a cascade heat pump dryer equipped with a solar collector was considered in order to examine the operability and efficiency of the heat pump cycle, depending on changes in the outdoor air temperature in wintertime. The changes in cascade cycles, depending on the temperature in a drying chamber, were also observed. The results showed that the average coefficient of performance (COP) of the cascade heat pump dryer was approximately 2.6 under the temperature range of –10 to 20 °C. An electrical heater whose COP is less than one should be used in that temperature range. It was also found that COP of the dryer increased by approximately 35% when using a solar collector under a low outdoor air temperature environment.

Key words: *heat pump dryer, cascade cycle, solar collector, COP, energy efficiency*

Introduction

As energy exhaustion and carbon emission issues are exacerbated globally, the importance of improving the energy efficiency with environmental problems has been more important than ever. Drying processes account for a sizable amount of energy consumption as a key process in many industries, including agriculture, wood, chemical, textile, and paper industries.

A hot air drying method is generally used for industrial dryers [1]. A heat pump dryer can save a significant amount of energy compared to other types of dryers, due to its higher drying efficiency [2]. However, the lower bound of evaporating temperature of R134a heat pump dryer ranges from 5 to 10 °C. When the outdoor air temperature falls below the evaporating temperature, the dryer is not operable [3].

Because of this problem, the drying chamber is heated first to a certain temperature, say 30 °C, using an electric heater at the initial stage of the drying process [4] before operat-

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ing the heat pump dryer. The related electrical power consumption results in reduced energy efficiency. However, if the evaporating temperature is lower than the outdoor air temperature, the heat pump dryer can be operated. There are four methods to achieve a low evaporating temperature: a multi-stage compression cycle [5], a double-stage expansion cycle [6], a cascade cycle [7, 8], and multi-temperature heat pump [9]. For the multi-stage compression cycle and double-stage expansion cycle, *COP* decreases in the process of achieving lower evaporating temperature since the compression ratio increases. On the other hand, the cascade cycle employing a binary refrigerant is able to achieve condensing temperature ranging from 40 to 90 °C and evaporating temperature from –75 to –40 °C with a low compression ratio [10, 11]. This is a result of the cascade cycle using two compressors, employing refrigerants such as R134a and R744, R600a, *etc.* [12, 13] for the HC (high-stage cycle) while using R410a, R404a, and R717, *etc.* [14] for the LC (low-stage cycle). Meanwhile, an NH_3-CO_2 cascade system shows a higher efficiency compared to other refrigerants cascade system. However, it has not been considered in this study since it is mainly used as a refrigeration system.

Thus, in this study, the low-temperature operability and *COP* of the heat pump dryer will be greatly improved by adopting a solar collector and a cascade system of R404a and R134a. Furthermore, a switching system from a cascade cycle to a single-stage cycle will be also developed to enhance the *COP*. Although a feasibility study on a cascade heat pump cycle operable under low temperature was previously conducted [15], an experimental study for validation has not been performed yet. Thus, experiments with a cascade heat pump dryer were conducted to evaluate the changes in operability and performance of the heat pump dryer depending on the actual outdoor air temperature in wintertime and drying chamber temperature. A solar collector was also used to improve the performance of the heat pump dryer under the low outdoor air temperature environment. Thus, the low-temperature operability and *COP* of the cascade heat pump dryer have been greatly improved in this study.

Experimental methods

A cascade heat pump cycle with a solar collector was considered to improve the low-temperature operability and *COP* of the dryer. In the equipment, two single-stage cycles were connected via the cascade heat exchanger. The Copeland compressors, ZB15KQE, were adopted. It is a scroll type compressor and the isentropic efficiency varies from 0.5 to 0.7 as the pressure ratio and the condensing temperature change from 2 to 8 and from 30 to 70 °C, respectively. The maximum isentropic efficiency is about 0.7 at the pressure ratio of 3 and the condensing temperature of 55 °C. Using an electronic expansion valve, the opening ratio of the HC was set at 75% while that of the LC was set at 50%. The drying chamber was made of acrylic with dimensions of 700 mm × 700 mm × 700 mm. As for the refrigerants, R134a was used for the HC and R404a for the LC. The configuration of the experimental equipment is shown in fig. 1. The cascade system can be run as a single-stage system by opening SV1 and closing SV2. Evaporator 1 only operates as an evaporator for a single-stage system. We do this since the single-stage system has higher efficiency than the cascade system. So we switched the cascade system to the single-stage system when the drying chamber temperature reaches 32 °C.

As for the experimental conditions, the data were measured from –10 to 20 °C with an interval of 5 °C. Data measurement was conducted to get steady-state values after about one hour from the start.

Pressure sensors were used to measure the high- and low-pressures in the HC and the LC to gauge the pressure. A T-type thermocouple was adopted to measure the various places. The mass-flow rate was measured using a metal tube flow meter, while the compressor power consumption was measured by a digital power meter. The error factor of measuring devices is ±0.5% for a pressure sensor, ±0.05% for T-type thermocouple, ±0.1% for the mass-flow meter, ±0.05% for a power meter.

For data processing, EES [16] was utilized for P-h diagrams and other theoretical calculations. Summary of equations for performance variables is given in eqs. (1)-(6), where $\dot{m}_1$ is the mass-flow rate of the HC and $\dot{m}_2$ is the mass-flow rate of the LC. The locations from h_1 to h_8 are illustrated in fig. 1. Condenser heating capacity, evaporator cooling capacity, compressor power consumption, *COP,* and solar collector capacity of the cascade heat pump dryer, is calculated:

$$\dot{Q}_{\text{cond1}} = \dot{m}_1 (h_2 - h_3) \tag{1}$$

$$\dot{Q}_{\text{evap2}} = \dot{m}_2 (h_9 - h_8) \tag{2}$$

$$\dot{W}_{\text{comp1}} = \dot{m}_1 (h_2 - h_1) \tag{3}$$

$$\dot{W}_{\text{comp2}} = \dot{m}_2 (h_6 - h_5) \tag{4}$$

$$COP = \frac{\dot{Q}_{\text{cond1}}}{\dot{W}_{\text{comp1}} + \dot{W}_{\text{comp2}}} \tag{5}$$

$$\dot{Q}_{\text{chamber}} = \dot{m}_{\text{air}} C_p (T_{\text{ch}} - T_{\text{amb}}) \tag{6}$$

In addition, a solar collector was used to improve the *COP* and operability of the heat pump dryer. In South Korea, horizontal global radiation per day is 0.97 kW/m^2 in summer and 0.49 kW/m^2 in winter [17]. The heat collecting efficiency of solar collectors in the market is around 0.75 [18], this means that it is possible to obtain 1.5 kW of heat in summer and 0.7 kW in winter when using a solar collector with an area of 2 m^2. In this study, a solar collector was replaced by a plate-type heat exchanger with a constant-temperature water bath. The heat transfer rate in the heat exchanger was adjusted between 0 kW and 1.7 kW by changing the mass-flow rate of 40 °C water.

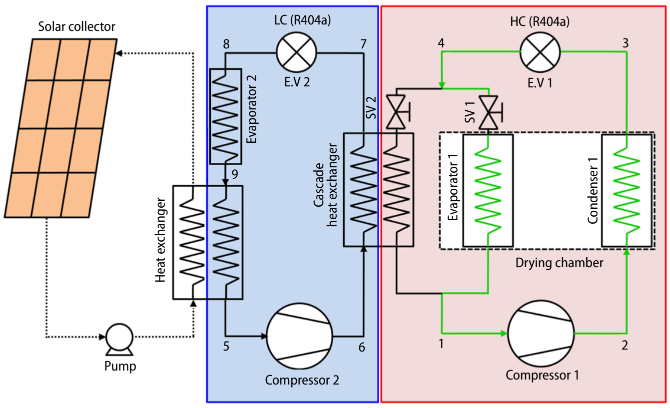

Figure 1. Schematic of the cascade heat pump dryer

Result and discussion

Cascade heat pump dryer performance according to outdoor air temperature

To investigate changes in the cycle performance by outdoor air temperature, the P-h diagram is shown in fig. 2. If the outdoor air temperature rises, the evaporation pressure at the low stage and compression pressure at the high stage increase proportionately. At the outdoor temperature of –10 °C, the evaporation and condensation pressures were 166 kPa and 1401 kPa, respectively. Since the evaporating temperature is approximately –20.3 °C, which is 10 °C lower than the outdoor temperature, the evaporator can absorb the heat from the outdoor air serving as a heat source. Figure 3 presents evaporating temperatures according to outdoor air temperatures. If the outdoor air temperature increases from –10 to 20 °C, the evaporating temperature rises from –20.3 to –10.1 °C, with the evaporating temperature being relatively lower than the outdoor air temperature. Therefore, operating the cascade heat pump dryer is found to be viable for low-temperature outdoor air.

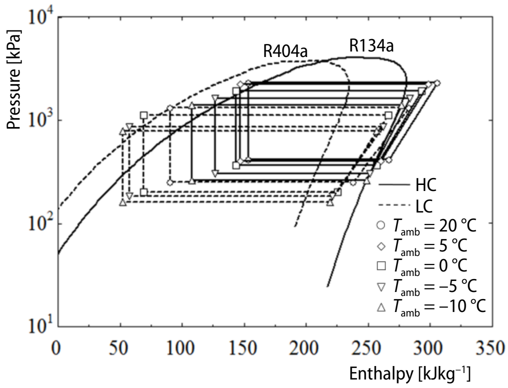

Figure 2. The P-h diagram of cascade cycle by outdoor air temperature

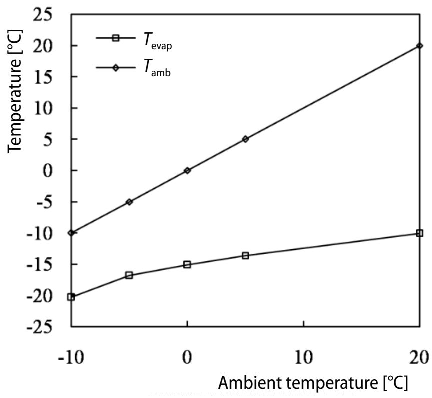

Figure 3. Variation of evaporating temperature with outdoor air temperature

Figure 4 shows the variation of suction gas temperature and superheat with the outdoor air temperature. If the outdoor temperature rises, suction gas temperature and superheat also increase, which means that the *COP* relatively decreases. When the outdoor air temperature is –10 °C, the superheat of the LC is 24.4 °C. This means that the superheat is sufficient under the low outdoor air temperature. From this data, it can be inferred that a stable cycle can be guaranteed without a liquid back phenomenon at the compressor, thanks to the sufficient cooling capacity from the low-temperature outdoor air.

Figure 5 is the mass-flow rate of refrigerants according to outdoor air temperature. If the outdoor air temperature increases from –10 to 20 °C, the mass-flow rate of the HC rises from 0.051 kg/s to 0.066 kg/s, and that of the LC increases from 0.028 kg/s to 0.039 kg/s. This is because the evaporating pressure and refrigerant density at the inlet of the compressor are increased as the outdoor air temperature rises. The increased mass-flow rate of refrigerant causes the heating capacity of the condenser and the cooling capacity of the evaporator to increase, as illustrated in fig. 6. On the other hand, when the outdoor air temperature rises from –10 to 20 °C, the *COP* decreases from 3.2 to 2.3 since the increase in compressor power consumption is larger than that in heating capacity of the condenser. Thus, the cascade cycle can achieve higher energy efficiency than the conventional dryers when the outdoor air temperature is low. As the ambient temperature decreases, the pressure ratio of condensing pressure and evaporating pressure for the single-stage cycle increases causing the energy efficiency to drop rapidly.

Performance of cascade heat pump cycle according to drying chamber temperature

The changes in the performance of the cascade heat pump dryer by the drying chamber temperature were also investigated. Figure 7 shows the mass-flow rate according to the dry-

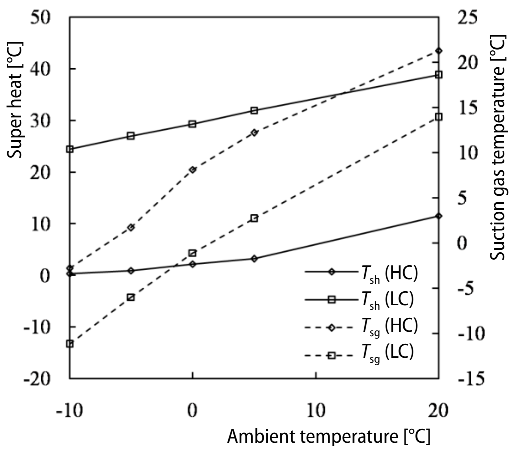

Figure 4. Variation of superheat and suction gas temperature with outdoor air temperature

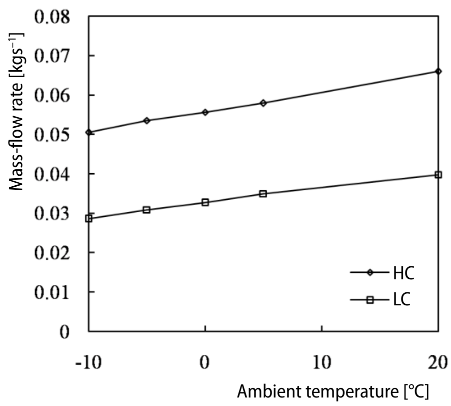

Figure 5. Variation of mass-flow rate of cascade cycle with outdoor air temperature

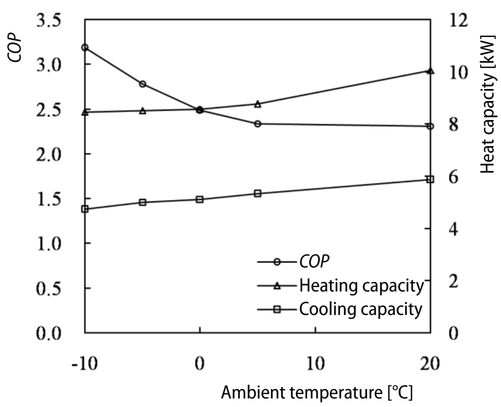

Figure 6. Variation of COP with outdoor air temperature

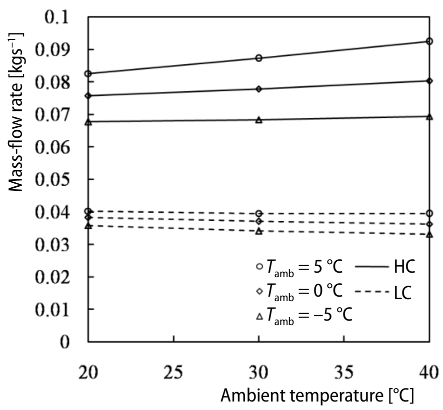

Figure 7. Variation of mass-flow rate with drying chamber temperature

ing chamber temperature. If the temperature of the drying chamber increases from 20 to 40 °C, the mass-flow rate of the HC increases from 0.067 kg/s to 0.07 kg/s under outdoor air temperature of -5 °C. On the contrary, the mass-flow rate of the LC decreases from 0.035 kg/s to 0.033 kg/s under outdoor air temperature of -5 °C. This is because the heat exchange of the cascade heat exchanger is increased due to the increased mass-flow rate of the HC as the temperature of the drying chamber increases. As a result, the evaporation pressure and the refrigerant density of the LC coming from the cascade heat exchanger decrease, resulting in decreased mass-flow rate of the LC.

Figure 8 shows the variation of heating capacity and cooling capacity with drying chamber temperature. When the outdoor air temperature is 5 °C, the heating capacity increased from approximately 9.1 kW to 10 kW as the drying chamber temperature rose from 20 to 40 °C. The cooling capacity, on the other hand, is decreased from 6 kW to 5.5 kW. As described earlier, this is because the mass-flow rate of the LC is reduced. When the drying chamber temperature rises, the heating capacity increases while that of cooling capacity remains almost unchanged. The cooling capacity is influenced heavily by the changes in the drying chamber temperature, while the heating capacity is relatively more influenced by the outdoor air temperature. This is because the condenser is located inside the drying chamber whereas the evaporator is exposed to the outdoor air.

Figure 9 is the variation of *COP* according to the drying chamber temperature. Under outdoor air temperature of –5 °C, the *COP* decreased from approximately 2.9 to 2.5 as the drying chamber temperature increased from 20 to 40 °C. This is because the condensing pressure and the power consumption of the compressor are increased as the drying chamber temperature rises. Therefore, the cascade cycle should be used to the point at which the single-stage heat pump dryer is operable in terms of energy efficiency.

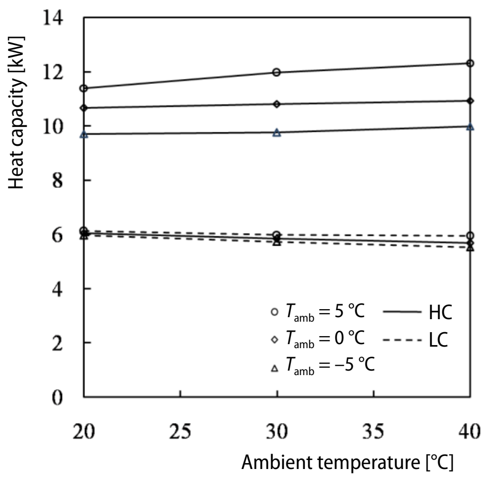

Figure 8. Variation of heating capacity and cooling capacity with drying chamber temperature

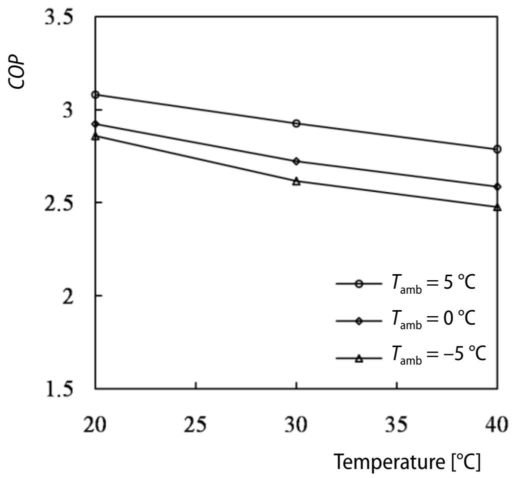

Figure 9. Variation of COP with drying chamber temperature

Performance of cascade heat pump cycle using a solar collector

A solar collector was used to enhance the operability and *COP* of the cascade heat pump dryer under the low-temperature outdoor air. The changes in the cascade heat pump cycle were observed by changing the capacity of the solar collector. Figure 10 shows P-h diagrams with the solar collector. In order to reproduce the situation where the evaporator does not absorb enough heat from the low-temperature outdoor air, the evaporator capacity was reduced. When the solar collector is not used, the superheat of the LC is 0.8 K, close to the saturated gas state of R404a. When the outdoor air temperature decreases further, the compressor experiences the liquid back phenomenon, causing the operation of the heat pump dryer to be failed. When the solar collector is used, the evaporating pressure increases from 141 kPa to 301 kPa, and the condensation pressure from 1221 kPa to 1321 kPa. The compression ratio decreases because the increase of evaporation pressure is greater than that of the con-

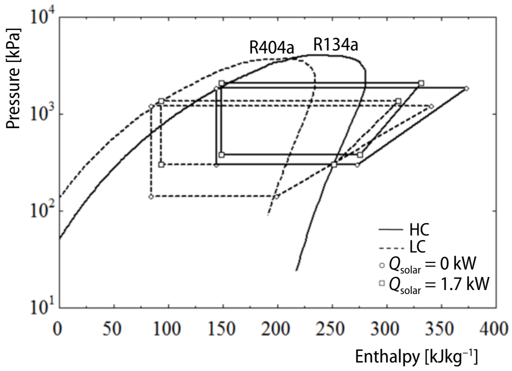

Figure 10. Variation of P-h diagrams with solar collector capacity

densation pressure. As a result, the temperature of refrigerant coming from the compressor decreased and the isentropic efficiency increased, leading to reduced power consumption of the compressor.

Figure 11 shows the variation of mass-flow rate with solar collector capacity. As for the LC, the mass-flow rate of the refrigerants relatively linearly increased with solar collector capacity. In the case of the HC, however, the mass-flow rate starts to significantly increase from 1 kW of solar collector capacity. This might be caused by the fact that the low-stage flow passage of the cascade heat exchanger is not fully filled with refrigerant at a low mass-flow rate. The heating capacity of the condenser is more likely to increase, as the capacity of the solar collector increases, due to the greater mass-flow rate. The performance of the dryer improves as well if the heating capacity of the condenser increases.

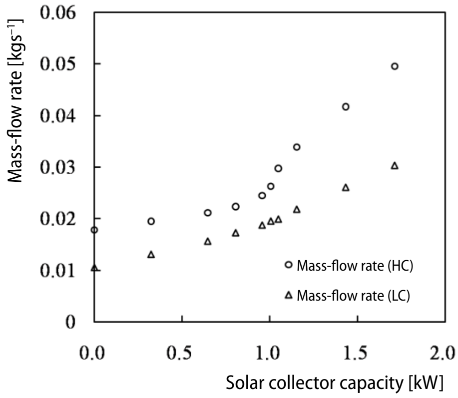

Figure 11. Variation of mass-flow rate with solar collector capacity

Figure 12 shows the variation of heating capacity and cooling capacity with solar collector capacity. As the capacity rises, the heating capacity increases from 4.2 kW to 9.1 kW and the cooling capacity of 1.2 kW to 4.8 kW. This demonstrates a similar trend as shown in experiments on mass-flow rate as in fig. 11, the heating capacity dramatically increases from the point where the evaporator capacity reduces to the solar collector capacity, while the cooling capacity rises relatively linearly.

Figure 13 shows the variation of *COP* with solar collector capacity. When the capacity of the solar collector increases, the *COP* rises from 1.3 to 2 because the solar collector capacity is a *free* energy from the environment. A greater solar collector capacity results in greater mass--flow rate which increases the heating capacity of the condenser. In addition, the compressor power decreases since the evaporating pressure increases. As a consequence, the isentropic efficiency increases and the refrigerant temperature at the compressor inlet decrease. Therefore,

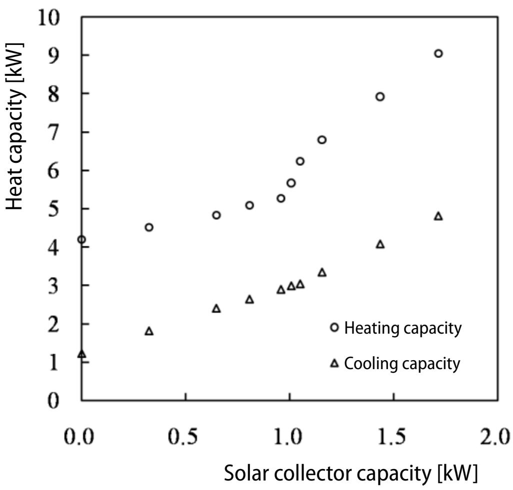

Figure 12. Variation of heating capacity and cooling capacity with solar collector capacity

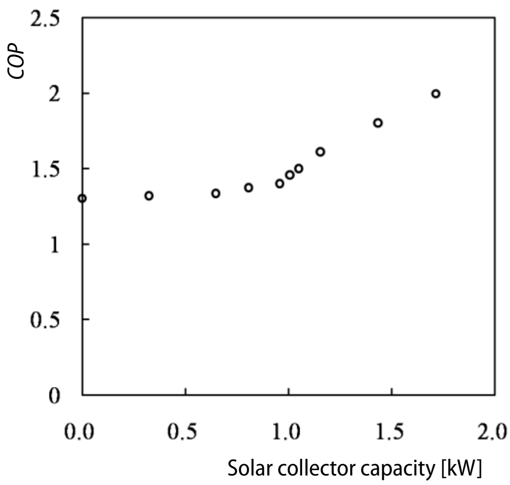

Figure 13. Variation of COP with solar collector capacity

it is found that the operational performance under low temperature is improved by the solar collector if the outdoor air temperature is similar to the evaporating temperature, due to the increased amount of cooling capacity. The *COP* is also enhanced by up to 35% when compared to the case where the solar collector is not used. Further study needs for verification of drying performance improvement for the cascade heat pump dryer with a solar collector, particularly in a preheating phase.

Conclusions

This study conducted a set of experiments focusing on a cascade heat pump dryer equipped with a solar collector with the aim of improving the operability and performance of the dryer under low temperature. The low-temperature operability of the dryer as well as the changes in the dryer performance with drying chamber temperature and the solar collector capacity was evaluated. The conclusions of this study are as follows.

- The cascade heat pump dryer can operate under outdoor air temperature as low as –10 °C, the lower bound temperature considered in this study. In addition, the lower the outdoor air temperature was, the higher the *COP* of the cascade heat pump dryer was.
- The average *COP* is around 2.6 under an outdoor air temperature range of –10 to 20 °C; this means the energy efficiency of the cascade cycle is more than 2.6 times higher than that of the electric heater used in the conventional single-stage cycle. The energy efficiency of the cascade cycle without a solar collector showed *COP* of 1.84-2.11 [4].
- The heating capacity of the condenser increases while the cooling capacity of the evaporator remains almost unchanged as the temperature of the drying chamber rises. Therefore, the heating capacity is more influenced by the changes in the temperature of the drying chamber while the cooling capacity is relatively more influenced by the outdoor air temperatures.
- In addition, it is more efficient to use this cascade cycle for preheating the single-stage heat pump dryer to the operable temperature, since the *COP* decreases as the temperature of the drying chamber rise.
- The solar collector increases the heating capacity of the condenser and the cooling capacity of the evaporator resulting from the greater mass-flow rate of the refrigerants. The *COP* of the heat pump dryer consequently increases by up to 35%, when compared to the case employing no solar collector.

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Nomenclature

C_p – specific heat, [$\mathrm{kJkg^{-1}K^{-1}}$]
h – enthalpy, [$\mathrm{kJkg^{-1}}$]
$\dot{m}_{\mathrm{air}}$ – mass-flow rate of air, [$\mathrm{kgs^{-1}}$]
$\dot{m}_1$ – mass-flow rate of high-stage cycle, [$\mathrm{kgs^{-1}}$]
$\dot{m}_2$ – mass-flow rate of low-stage cycle, [$\mathrm{kgs^{-1}}$]
$\dot{Q}_{\mathrm{cond}}$ – heating capacity, [kW]
Q_{evap} – cooling capacity, [kW]
$\dot{Q}_{\mathrm{solar}}$ – solar collector capacity, [kW]
T – temperature, [°C]
T_{amb} – ambient temperature, [°C]
T_{ch} – drying chamber temperature, [°C]
T_{sg} – suction gas temperature, [°C]
T_{sh} – super heat, [°C]

$\dot{W}_{comp1}$ – power consumption of high-stage cycle, [kW]
$\dot{W}_{comp2}$ – power consumption of low-stage cycle, [kW]

Subscripts

COP – coefficient of performance
HC – high-stage cycle
LC – low-stage cycle

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