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Study on the Performance Improvement of the Heat Pump Dryer Using the Waste Heat Recovery System

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Abstract- Heat pump dryers are used in various industries, but heat pump operation is impossible when the drying chamber has a low temperature. To facilitate operation of the heat pump, the drying chamber must be adequately heated. This study seeks to improve drying chamber efficiency by applying a heat recovery system to send waste heat to a second drying chamber. EES, a commercial software, was used to analyze the heat pump cycle for 12 tons of frozen red pepper based on a 60HP compressor. Data for dehumidification level and relative humidity was used to estimate the actual amount of heat recovery. In addition, numerical analysis was performed to determine the optimal capacity of the heat exchanger. The proposed heat recovery system succeeded in improving energy efficiency by reducing the dryer heating time by approximately 28.8%.

Keywords- Dryer, Heat pump, Heat exchanger, Heat Recovery

1. Introduction

Dryers are used for drying products in various industries, such as agriculture, fishery, timber and clothing [1]. In particular, heat pump dryers have been significantly developed and applied in the timber and food industries since the 1970s [2]. Compared to existing convection dryers, heat pump dryers have improved efficiency in energy use [3]. This can be explained by their recovery of heat in heat pump cycles [4]. In addition to heat pump cycles installed in the drying chamber, heat pump dryers are comprised of condensers and evaporators. The former raises the internal temperature of the drying chamber, while the latter emits air and introduces external air through the air damper when high-temperature humid air is being dehumidified via the condensate. Drying occurs when this process is repeated. The recovery of high-temperature humid air helps to improve energy efficiency.

In the case of large dryers used for drying 12 tons of frozen red pepper, the air damper begins releasing heat and introducing external air to maintain a constant temperature when the drying temperature exceeds 60°C [5]. This release of heat can be seen as a significant energy loss. In addition, when the internal temperature of the drying chamber falls below 20°C during the winter, the condenser is subject to greater strain due to its interference with heat absorption [6]. Before operating the heat pump cycle, the drying chamber must be pre-heated to a certain temperature using an electric heater. While the heating time differs according to the type and amount of products to be dried, about 5 to 7 hours are needed to dry 12 tons of frozen red pepper using a 100 kW heater. The consumption of electric energy in this process is a main factor leading to the deteriorating performance of the dryer.

Many studies have examined methods for improving the performance of heat pump systems. Jang [7] examined performance improvement by installing an energy-saving cooling dehumidification drying device based on condensate waste heat. Lee [8] attached a supplementary external evaporator to reduce the time needed for raising the temperature, while Chung et al [9] studied solar-based drying systems.

This study introduced a heat exchanger to supply recovered waste heat to other drying chambers, thereby improving the efficiency of the drying process by shortening the heating time. A heat recovery system was applied to the heat pump drying chamber for 12 tons of frozen red pepper, and energy reduction was estimated based on the reduced heating time.

2. Research Methodology

2.1 System Overview

This study introduces a heat exchanger system in each drying chamber by applying a single-stage compression cycle and heat recovery system to the heat pump dryer. Fig. 1 shows the schematic diagram of the heat pump dryer applied to the heat recovery system. The internal temperature of the inactive drying chamber B is raised by absorbing heat from the active drying chamber A. To heat drying chamber B while maintaining the temperature of drying chamber A, an optimal capacity for the heat exchanger must be determined. The dryer used in this study was $L\ 12m \times W\ 7.5m \times H\ 3m$, and the compressor was 60HP.

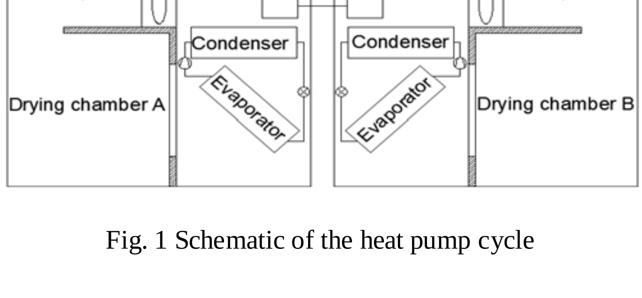

Fig. 1 Schematic of the heat pump cycle

Table 1 Cycle analysis conditions

Evaporating temperature	5°C
Condensing temperature	55°C
Superheated degree	10°C
Subcooled degree	8°C
Mass flow rate	0.828kg/s
Refrigerant	R134a
Efficiency of compressor	0.635

2.2 Theoretical Analysis

The commercial software EES [10] was used to determine the heat output of the heat pump dryer and the performance of the heat pump cycle. Table 1 shows the conditions of cycle analysis. The refrigerant was R134a, and the evaporating and condensing temperature were assumed to be 5°C and 55°C respectively. Other conditions were a superheated temperature of 10°C, a subcooled degree of 8°C, and a mass flow rate of 0.828 kg/s. The compressor efficiency was 0.635.

Meanwhile, the internal temperature of the drying chamber continuously increases during operation of the heat pump cycle. This quantity of heat is the same as that of the heat exchanger, thereby maintaining the temperature of drying chamber A and allowing the heating of drying chamber B before the drying process. The capacity of the heat exchanger was calculated as shown in Eq. 1.

$$\dot{Q}_{ex} = \left(\dot{Q}_{cond} - \dot{Q}_{evap}\right) - (\dot{m}_1 h_1 - \dot{m}_2 h_2)$$

Unlike other heating systems containing an internal condenser and external evaporator, the heat pump drying system has condensers and evaporators in the drying chamber. In Eq. 1, the difference in heat release by condensers and heat absorption by evaporators was used to determine the capacity of the heat exchanger. Here, $\dot{m}_1 h_1$ is the quantity of heat of the humid air before reaching the evaporators, and $\dot{m}_2 h_2$ is the quantity of heat of the humid air after the evaporators. The total quantity of heat needed to heat the drying chamber is given by Eq. 2.

$$\dot{Q}_{dc} = \dot{m}_{dc} C_p (T_{dc} - T_{dc0})$$

Here, $\dot{Q}_{dc}$ is the total quantity of heat needed in the drying chamber, $\dot{m}_{dc}$ is the air mass flow of the drying chamber, T_{dc} is the target heating temperature of the drying chamber, and T_{dc0} is the initial temperature of the drying chamber. Before operating the dryer, the internal temperature of the drying chamber was 0°C, and the target heating temperature was 30°C.

2.3 Numerical Analysis

The commercial software FLUENT [11] was used to analyze changes in temperature in drying chambers A and B, and a dual cell model was applied in the heat exchanger analysis. The air velocity of the drying chambers and mass flow rate of the cooling water were fixed. The heat transfer area was set as a variable in order to determine the capacity of the heat exchanger. The size of the heat exchanger was $W\ 2.8m \times H\ 1.2m \times D\ 0.4\ m$, and the analysis conditions are presented in Table 2.

Table 2 Analysis conditions of FLUENT

Temperature of drying chamber A	60°C
Temperature of drying chamber B	0°C
Water inlet temperature	10°C
Air velocity	2.5m/s
Water mass flow rate	1kg/s
Heat transfer area	100 ~ 500m^2

3. Results and Discussion

3.1 Performance Analysis of the Heat Pump Cycle

Fig. 2 shows the P-h diagram, based on the conditions applied to the analysis of the heat pump cycle. To derive results close to that of the single-stage compression cycle, heat loss occurring during compression was assumed to be 10%, and the pressure drop in the refrigerant pipe was 0.5°C. When using a 60HP compressor, the refrigerant was R134a. The capacities of the compressor and evaporator were 166.6kW and 125kW respectively, and the COP at this point was 2.8.

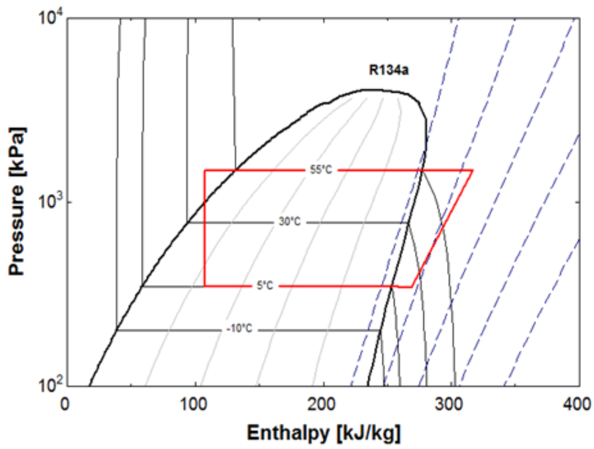

Fig. 2 P-h diagram of the heat pump cycle

3.2 Amount of Dehumidification and Heat Absorption

To examine changes in the amount of dehumidification and heat absorption by products in relation to drying time, a heat pump dryer for frozen red pepper was used [12]. The evaporator exhibited good heat absorption during the early stage of drying due to high dehumidification, but this dropped significantly after 32 hours. Then, heat output by the condenser became much larger than heat absorption, resulting in overheating of the drying chamber. This not only causes overheating of the compressor, but also lowers product quality. As such, the heat recovery system should be operated after 32 hours, when the heat recovery system produces any hardly condensate.

Table 3 shows the psychometric conditions before and after reaching the evaporators. The dry bulb temperature before and after the evaporators was 60°C and 55°C respectively. The relative humidity was set based on the results of Lee et al [12]. The difference in relative humidity before and after the evaporators was 23%, after 32 hours, and almost constant at 10% after 40 hours. This implies that there was hardly any heat loss due to dehumidification at the final

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stage of drying. When the difference in relative humidity was 10%, the dehumidification heat was only 1.5kW as shown in Fig. 3. Heat must be released to prevent overheating of the drying chambers. The amount of heat released was calculated by Eq.1 as 40.1kW.

Table 3 Psychometric conditions

Evaporator before	
Dry bulb temperature	60^{o}C
Wet bulb temperature	33.90^{o}C
Absolute humidity	0.0230 kg/kgDA
Relative humidity	18.1 %
Enthalpy	123.359 kJ/kg
Evaporator After	
Dry bulb temperature	55^{o}C
Wet bulb temperature	36.47^{o}C
Absolute humidity	0.0317 kg/kgDA
Relative humidity	31.2 %
Enthalpy	137.757 kJ/kg

3.3 Heat Exchanger Capacity for Optimization of the Heat Recovery System

If the amount of heat absorbed by drying chamber A is smaller than 40.1 kW, some heat remains in the condenser. The compressor becomes overheated when the temperature of the drying chamber exceeds 60^{o}C, thus causing failure of the heat pump cycle. Heat must be released via the air damper in order to maintain the drying chamber temperature. On the other hand, if the amount of heat absorbed by drying chamber B is greater than 40.1kW, a high amount of heat is radiated to drying chamber B. The rise in internal temperature of drying chamber B is accelerated, but the temperature of drying chamber A drops due to reduced heat output at the condenser, and this results in lower moisture content of the internal air. When air of less than 60^{o}C passes through the products, the amount of condensate decreases during dehumidification and drying efficiency deteriorates.

In other words, the heat exchanger should have a capacity of 40.1kW for system optimization. Fig. 4 shows heat rejection with heat transfer area in drying chambers A and B. From numerical analysis, we found that the optimum capacity was 40.1kW when assuming a heat transfer area of 200m^2 for 12 tons of frozen red pepper. When the heat recovery system is optimized and operated after 40 hours, it is possible to maintain the temperature of drying chamber A and heat up drying chamber B until drying is complete, even without using an air damper.

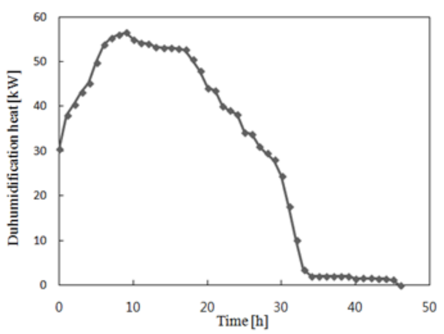

Fig. 3 Dehumidification heat with drying time

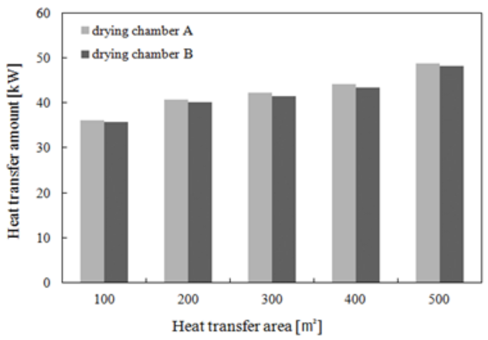

Fig. 4 Heat transfer amount with heat transfer area in drying chambers A and B

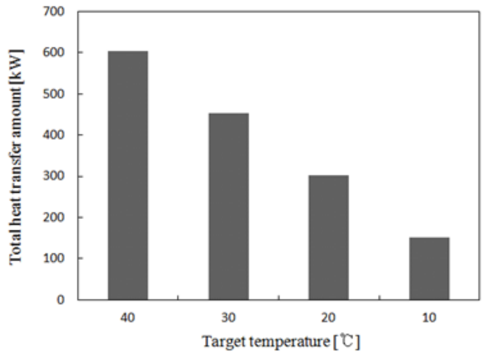

Fig. 5 Total heat transfer amount with target temperature

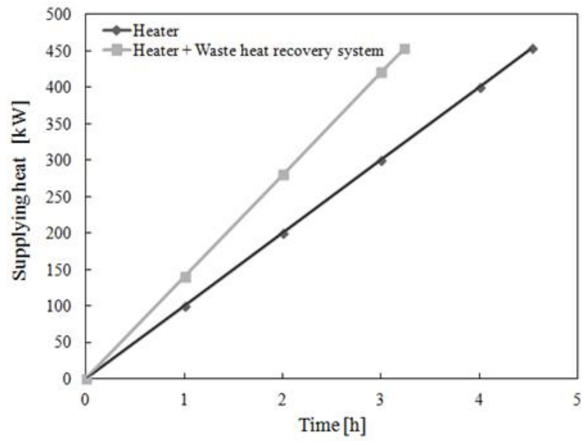

Fig. 6 Supplying heat with time for each system

3.4 Analysis of Reduced Heating Time in the Heat Recovery System

Fig. 5 shows the total heat transfer amount needed to reach the target temperature. There is a linear increase in the quantity of heat needed to heat the drying chamber up to the target temperature. A heat transfer amount of 150.9kW is needed to reach a target temperature of 10^{o}C, and 452.9kW is needed to achieve 30^{o}C. Fig. 6 shows the time taken to reach 30^{o}C using a 100kW heater, and the time taken to reach the target temperature using the heat recovery system. About 4.5

hours are needed when using the 100kW heater. If the heat recovery system is applied, the total heat supply becomes 140.1kW, and the heating time is reduced by 1.3 hours. As such, compared to existing heat pump drying systems, the proposed system improves energy efficiency by approximately 28.8%.

4. Conclusion

This study applied a heat recovery system to existing heat pump dryers to shorten dryer heating time, based on an analysis of cycle performance and level of dehumidification. The conclusions of this study are as follows.

1. The amount of heat needed during the dehumidification process after 33 hours of drying is 1.5kW, and approximately 40.1kW can be recovered.
2. The optimal heat transfer area was $200m^2$ for a heat exchanger, $W\,2.8m \times H\,1.2m \times D\,0.4\,m$. This corresponds to a pitch of 2.1mm for plate type wavy fins.
3. A 100kW electric heater required 4.5 hours to heat the drying chamber. The heating time was reduced by 1.3 hours using the heat recovery system, which is equivalent to a 28.8% improvement in energy efficiency.

5. Acknowledgement

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