PERFORMANCE ANALYSIS OF A RE-CIRCULATING HEAT PUMP DRYER

by

Nurettin YAMANKARADENIZ [a], K. Furkan SOKMEN [d], Salih COSKUN [a*], Omer KAYNAKLI [b], and Bilsay PASTAKKAYA [c]

[a] Air Conditioning and Refrigeration Technology Program, Electric and Energy Department, Vocational School of Technical Science, Uludag University, Gorukle, Bursa, Turkey
[b] Mechanical Engineering Department, Engineering and Architecture Faculty, Uludag University, Gorukle, Bursa, Turkey
[c] Machinery Program, Orhangazi Vocational School, Uludag University, Orhangazi, Bursa, Turkey
[d] Mechanical Engineering Department, Institute of Science, Uludag University, Gorukle, Bursa, Turkey

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A re-circulating heat pump dryer (HPD) system was designed, constructed and tested at steady-state and transient conditions. Refrigerant 134a was used as a refrigerant in this system. The tests were performed to observe behavior of HPD system. So, changes of temperature and relative humidity of drying air through the dryer and heat pump operating temperatures were observed during the drying process and effects of by-pass air ratio (BAR) on the system's parameters as system performance and specific moisture extracted ratio (SMER) at steady-state were investigated. The HPD system was also tested to investigate temperatures and relative humidity changes of drying air during drying process on the system's parameters depend on time. Air flow rate circulated through the HPD system was 554 m^3/h during the all tests. According to test results, the system's parameters did not change up to 40% of BAR. Then the coefficient at performance (COP) and SMER values were decreased after 40% of BAR. While SMER values changed between 1.2 and 1.4, COP_{sys} changed between 2.8 and 3.3 depend on BAR. As well as during the drying process, the COP and SMER values were also affected and decreased depend on time.

Key words: *drying, dryer, heat pump, by-pass air ratio, heat pump assisted dryer*

Introduction

A drying system that is both energy efficiency and preserves product quality is desired. Some products need to be dried at low temperature (30-35 °C) for product quality. The HPD systems are suitable for some sensitive products need to be dried at low temperature. Re-circulating HPD especially are more efficient than conventional dryers. In the re-circulating HPD, both sensible and latent heat can be recovered from dryer exhaust air, improving the overall thermal performance. There are a lot of studies in literature about heat pump assisted drying systems [1-7]. In the drying process, the moisture of the air must be regulated. Firstly, in order to control the

* Corresponding author; e-mail: scoskun1968@gmail.com

dehumidification rate during the whole drying process in terms of productivity and the quality of the product. Because the evaporator is the heat exchanger responsible for the recovery of latent heat from the air, changing the mass flow through the evaporator will impact the performance of the heat pump [8]. Secondly, in order to mix the air leaving the evaporator at 80-90% relative humidity with the air leaving the dryer. This supplies a sufficient air flow rate through the condenser to avoid excessive compressor discharge pressures and temperatures. At the same time, the mixing process allows the condenser to reach the desired air supply temperature without over-designing the heat transfer surface. The heat pump evaporator may also be provided with a variable speed fan to improve the control of the material drying rate. The combined action of the bypassing air and the fan variable speed provides optimum air flow rate through the evaporator, independently to the air flow rate through the condenser [9]. There are some studies about *BAR* effect on the system's parameters in the literature. Achariyaviriya *et al*. [10] reported that *COP* decreased when the fraction of *BAR* through the evaporator increased. Oktay [11] explored the parameters affected by the performance of HPD. According to Oktay [11], *BAR*, the total air flow rate and the exhaust flow rate are the key parameters that affect the system's performance. According to Chua and Chou [12], since the evaporator is the heat exchanger responsible for the recovery of latent heat from air, changing the mass flow through the evaporator will impact the performance of the heat pump. The *BAR* to evaporator was increased beyond a level, the total heat recovered at the evaporators reduced. For every 20% increment in *BAR* over the 40% mark, the drop in heat recovered ranged between 0.6-0.8 kW. In the study by Qi-Long *et al*. [13], *BAR*, and air velocity on the drying characteristics of horse mackerel were studied. According to Teeboonma *et al*. [14], the most important factors, while examining the optimum conditions for HPD and for minimizing the HPD cost, were the recycled air ratio, *BAR*, air flow rates, and the drying air temperature. The results illustrated that the optimum conditions of each test material are not similar, especially the optimum air flow rate and *BAR*. The physical properties of the test material significantly affect the optimum airflow rate and *BAR*. In another study, Clements *et al.* [15] have recommended that *BAR* should be in the range of 60-70% to obtain the optimum ratio for *SMER*. In this study, a re-circulating HPD system was designed, constructed and tested. Experimental analysis was performed on this system to determine the optimum working conditions at steady-state and transient conditions.

Experimental set-up and measurements

Experimental set-up

The experimental set-up was designed and constructed at the Air Conditioning and Refrigeration Laboratory, Vocational School of Technical Science, Uludag University, Bursa, Turkey as seen from in fig. 1. It consisted of two main parts: heat pump, and drying chamber. The air was heated by a heat pump system including a reciprocating compressor, a condenser (internal and external ones), a thermostatic expansion valve, and an evaporator. Refrigerant 134a was used as a refrigerant in this system. Drying air was circulated by a radial fan in a duct system and fresh air was not used during the drying process. There are a bypass and a flow damper to adjust bypass air flow rate. There are two condensers, internal and external. When there was an increase at discharge pressure and temperature of the heat pump system. The external condenser was cut in to decrease the discharge pressure and temperature.

In this study, a humidification unit was installed in the drying chamber instead of the product. Since the aim of this study was to investigate HPD unit and dryer system performance rather than the product. Thus, a large degree of flexibility, reliability, and modularity have been

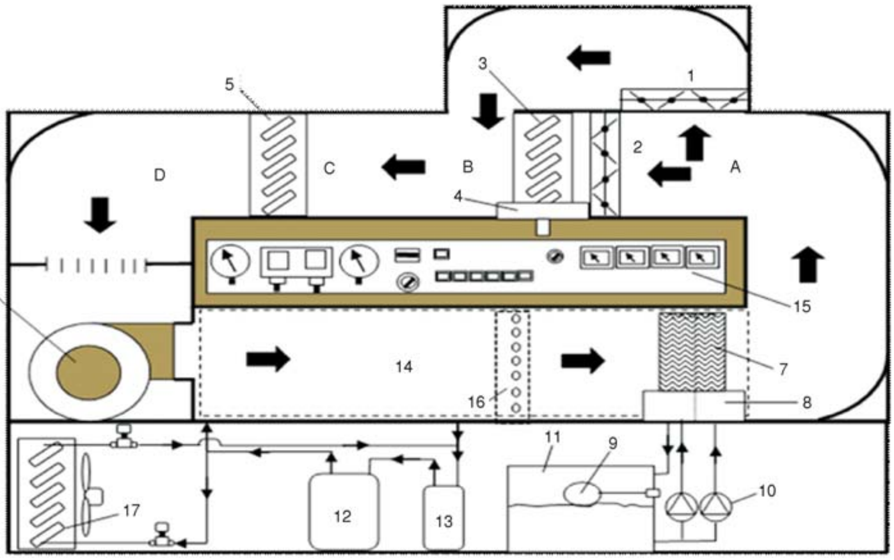

Figure 1. Schematic description of HPD system
1 – bypass damper, 2 – flow damper, 3 – evaporator, 4 – drain pan, 5 – internal condenser, 6 – radial fan, 7 – evaporative cellulose pad, 8 – reservoir, 9 – float, 10 – water circulating pumps, 11 – water tank, 12 – compressor, 13 – accumulator, 14 – drying chamber, 15 – control panel, 16 – velocity measurement points, 17 – external condenser

provided. The humidification unit consists of an evaporative cellulose pad, a reservoir, a water storage tank, and two circulating pumps. Water is pumped to the evaporative cellulose pad from the water storage tank at 20 °C to obtain humid air conditions at the outlet of the drying chamber constantly during the experiment.

During the sizing of HPD system components, appropriate correlation between the initial mass and moisture content, the moisture migration and extraction rate of each dried product and the heat pump dehumidification capacity has to be provided for any drying process. Matching the dewatering capacity of the dried material with the heat pump dehumidification capacity is a major issue for any heat pump dryer [16]. So, humidity extracted from product should be determined. In this study, after predetermined air mass flow rate, air inlet, and outlet conditions (shown as A, B, C, and D points) through the dryer and then humidity content sprayed into the air were determined by means of psychometric diagram. The heat pump dehumidification capacity depends on the quantity of water to be removed from the dried material. The ratio between the heat pump dehumidification capacity and the heat pump input power has to be as high as possible. Then capacities of heat pump components (*i. e*. evaporator, condenser, and compressor) were specified by using heat transfer equations between air and refrigerant. Condensing and evaporating design temperatures was selected as about 50 °C and 7.5 °C, respectively. The condensing temperature was not selected higher temperature than 50 °C. Because higher condensing temperature leads higher compressor outlet temperature and consequently, compressor damaged.

In this experimental set-up, the re-circulating drying air primarily enters the condenser. While the temperature of the drying air increases, the relative humidity decreases throughout the condenser by absorbing heat from the refrigerant. The refrigerant also condenses in the condenser coils by giving heat to the circulating air. The circulating air leaving the con-

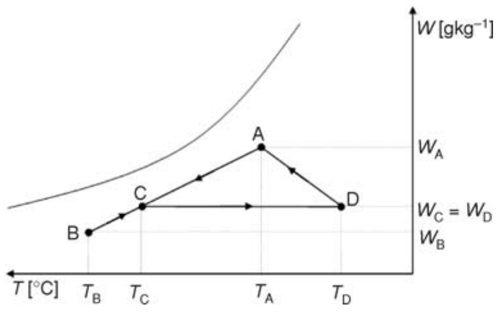

Figure 2. Change of the air properties during the drying process on the psychometric diagram

denser is delivered into the drying chamber, where the warm air is humidified then the moist air with increased humidity enters the evaporator. The air flow rate passing and bypassing the evaporator is controlled during the experimental analysis by setting up the flow and the bypass damper. One part of the air is passed to the evaporator, and the other part is bypassed. Warm and humid air passing through the evaporator provides heat to the refrigerant in the coils which cool down as condensation occurs over their surface. Air passing and bypassing from the evaporator is mixed at the outlet of the evaporator and is sent to the condenser. Air passing to the condenser absorbs the heat from the refrigerant in the condenser coils and leaves the condenser warmer. Finally, the exhaust air is sent to the drying chamber by means of the radial fan. Change of the drying air during the drying process was shown on the psychometric diagram in fig. 2. Technical specifications of the main components of HPD system were given in tab. 1.

Table 1. Specification of main components of HPD unit

Main components	Technical specifications
Fan	Radial fan, 1080 rpm for 50 Hz, 1.5 A max., 3 stage
Compressor	2875 rpm, 1 ½ HP, displacement 32.7 cm^3
Evaporator	Number of vertical pipe: 12, number of horizontal pipe: 3, number of fin: 196 Dimension of exchanger: 380 mm high × 85 mm width × 450 mm length, pipe outside diameter: 0.9525 m
Internal condenser	Number of vertical pipe: 12, number of horizontal pipe: 3, number of fin: 196 Dimension of exchanger: 380 mm high × 85 mm width × 450 mm length, pipe outside diameter: 0.9525 m
External condenser	Number of vertical pipe: 12, number of horizontal pipe: 2, number of fin: 117 Dimension of exchanger: 360 mm high × 145 mm width × 410 mm length, pipe outside diameter: 0.9525 m

Measurements

Some measurements were achieved during the experiment. Air velocity was measured carefully according to Tchebycheff method to calculate air flow rate correctly [16]. The air velocity was measured by a hotwire anemometer. The velocity measurement was conducted. The duct cross-section is re-ctangular and it is 530 mm width and 385 mm height. Using the log-Tchebycheff method, the duct is divided into re-ctangular areas, which are further adjusted in size to account for effects of the duct wall on the airflow. Thirty points were measured in order to get a good average. Mean air velocity was measured as 0.77 m/s and mean air flow rate was calculated as 554 m^3/h air flow rate. The re-circulating air temperature and relative humidity values were also measured by multipurpose data acquisition modules from points A, B, C, and D as shown in fig. 1. Temperature and humidity probes were placed in a hole on the duct wall at these points. Operating temperatures in the heat pump system were measured from pipe surface by means of thermocouples. Condensed water through the evaporator was collected in a container and measured by a sensitive scale and power consumption of the fan and the compressor were also measured by an electric meter during the experiment. The tests were carried out on HPD system under steady-state conditions to determine *BAR* system performance. All measured data were recorded by a data logger and transferred to the computer. The accuracies of the measurement devices are given in tab. 2.

Table 2. The accuracies of the measurement devices

Measured quantity	Measurement device	Measurement range	Accuracy
Total power consumption (fan and compressor)	Digital three phase electric meter	0-3000 W	±0.02%
Moisture extraction ratio	Digital weighing scale	0-55 kg	±0.01%
Air temperature	Data sheet SHT71 humidity and temperature sensor	–40-123 °C	±0.4%
Air humidity	Data sheet SHT71 humidity and temperature sensor	0-100% RH	±3%
Air velocity	TESTO 0635 1047 hot wire NTC	0-20 m/s	±4%

Uncertainty analysis

Uncertainty is considered as a possible perturbation of a given quantity around its nominal value. The experimental uncertainties in this study were calculated using standard uncertainty analysis methods reported by Moffat [17]. Uncertainties of the calculated parameters were determined by using accuracies of related measured variables and calculated total uncertainties. The estimated uncertainties of the COP_{hp}, COP_{sys}, moisture extraction rate (*MER*) and *SMER* were the within the range of ±1.2%, ±1.2%, ±2.1%, and ±1% of the nominal values, respectively. A summary of the representative mean values of the measured parameters and experimental results are given in tab. 3.

Calculations

For the de-humidification efficiency of a heat pump, *COP* is not a satisfactory yardstick. In the drying processes the goal is always to withdraw humidity and therefore a performance factor can be defined that relates the evaporation enthalpy of 1 kg water to the energy necessary to remove it. This make sense, since a heat pump cooling and heating the air streams at a small temperature differences may have an excellent *COP* without removing any water [18]. Therefore, *SMER* was also investigated as the system parameters in addition to *COP*. The *COP* value is defined in two ways. Firstly, if only the compressor power consumption is taken into account, the *COP* value is expressed as COP_{hp} and defined:

$$COP_{\mathrm{hp}} = \frac{\text{heat given the drying air through the condenser}}{\text{compressor power consumption}} = \frac{\dot{Q}_{\mathrm{cd}}}{\dot{W}_{\mathrm{comp}}} \tag{1}$$

Here, total power consumption is equal to sum of the fan and compressor power consumptions. Second, if the total power consumption is taken into account, the *COP* value is expressed as COP_{sys} and defined:

$$COP_{\mathrm{sys}} = \frac{\text{heat given the drying air through the condenser}}{\text{total power consumption}} = \frac{\dot{Q}_{\mathrm{cd}}}{\dot{W}_{\mathrm{tot}}} \tag{2}$$

The heat given the drying air through the condenser is calculated from the equation:

$$\dot{Q}_{\mathrm{cd}} = \dot{V}_{\mathrm{a}} \rho_{\mathrm{a}} c_{\mathrm{pa}} (T_{\mathrm{D}} - T_{\mathrm{C}}) \tag{3}$$

Table 3. Representative mean values of measured and calculated parameters and experimental results

Measured parameters	Symbol	Value	Unit	Total uncertainty [%]
Average air velocity	V	0.77	m/s	±4
Drying air temperature at A point (drying chamber outlet)	T_A	28.6	°C	±0.4
Drying air temperature at B point (evaporator outlet)	T_B	17.8	°C	±0.4
Drying air temperature at C point (condenser inlet)	T_C	19.5	°C	±0.4
Drying air temperature at D point (drying chamber inlet)	T_D	43.8	°C	±0.4
Drying air relative humidity at A point	Φ_A	73.5	%	±3
Drying air relative humidity at B point	Φ_B	95.8	%	±3
Drying air relative humidity at C point	Φ_C	92.4	%	±3
Drying air relative humidity at D point	Φ_D	24.3	%	±3
Total power consumption (compressor and fan)	$\dot{W}_{tot}$	1362	W	±0.2
Fan power consumption	$\dot{W}_f$	159	W	±0.2
Compressor power consumption	$\dot{W}_{comp}$	1203	W	±0.2
Heat pump suction pressure (absolute)	P_{ev}	3.8	Bar	±0.5
Heat pump discharge pressure (absolute)	P_{cd}	13.9	Bar	±0.5
Refrigerant evaporating temperature	T_{ev}	7.5	°C	±0.3
Refrigerant condensing temperature	T_{cd}	52	°C	±0.3
Refrigerant temperature at the compressor inlet	T_1	10.7	°C	±0.3
Refrigerant temperature at the compressor outlet	T_2	74.8	°C	±0.3
Refrigerant temperature at the condenser outlet	T_3	44.9	°C	±0.3
Refrigerant temperature at the evaporator inlet	T_4	7.5	°C	±0.3
Calculated parameters				
Volumetric air flow rate	$\dot{V}_a$	554	m^3/h	±5.2
Heat given the drying air through the condenser	$\dot{Q}_{cd}$	4490	W	±5.2
Moisture extraction rate	MER	2.120	kg/h	±2.1
Performance coefficient of the heat pump	COP_{hp}	3.74		±1.2
Performance coefficient of all system	COP_{sys}	3.30		±1.2
Specific moisture extraction ratio	$SMER$	1.56	kg/kWh	±1

The *MER* is the amount of water removed from the product per hour and *SMER* is the energy required to remove a unit mass of water and defined in eq. (4). In this study, *MER* was obtained from evaporator by measuring condensed water as kg per hour.

$$SMER = \frac{\text{moisture extraction rate}}{\text{total power consumption}} = \frac{MER}{\dot{W}_{tot}} \tag{4}$$

The *BAR* represents the amount of air without contact with evaporator surface. The *BAR* is calculated as a function of the temperatures:

$$BAR = \frac{T_C - T_B}{T_A - T_B} \tag{5}$$

Result and discussion

This study can be evaluated as two parts. First part is steady-state analysis. In this section, the experiments were conducted to observe temperature and relative humidity changes of drying air, change of heat pump operating temperatures and capacities, and also investigate effects of *BAR* on the system's parameters (*MER*, *SMER*, COP_{hp}, and COP_{sys}). Second part is transient analysis. In this section, HPD system was tested to investigate behavior depend on time. So, temperatures and relative humidity changes of drying air during drying process on the system's parameters were investigated. Air flow rate circulated through HPD system was 554 m^3/h during the all tests.

Steady-state analysis

The HPD system was operated after obtaining the steady-state conditions. It was tested for various *BAR* value which adjusted by two damper, one of them was called as *flow damper* placed in front of the evaporator, the other was called as *by-pass damper* which by-passed the humid air from the evaporator. During the setting of *BAR*, as the flow damper was throttled, by-pass damper was turned on. The HPD system was operated and tested for one hour for every *BAR* settings and in the meantime, constant humidity was supplied by two water circulating pumps instead of product. Although the by-pass damper was closed tightly, because of leakage, the minimum *BAR* was calculated as about 18%.

The changes of the drying air temperatures and relative humidifies at HPD system depend on *BAR* are shown in fig. 3. Heat transfer between air and refrigerant at the evaporator decreased with increasing *BAR* because of less air flow rate through the evaporator. Condenser capacity was also affected and decreased depend on the evaporator capacity. With increasing *BAR*, the highest temperature falling was observed at the outlet of the evaporator and this caused to increase air temperature at the condenser inlet. Because, the drying air at the condenser inlet is the mixing of cooled air which passed the evaporator and by-passed air.

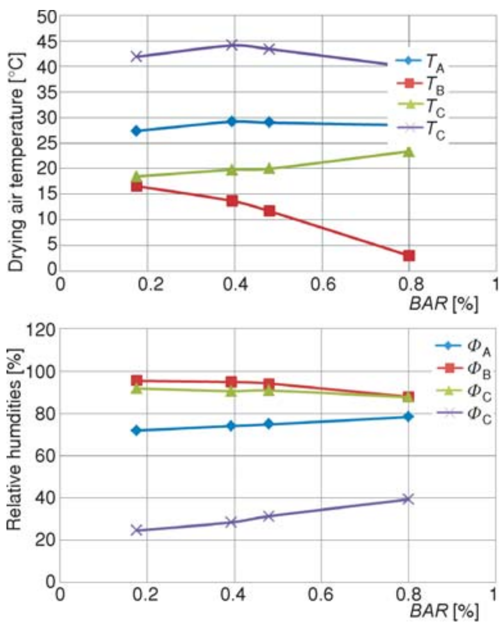

Figure 3. Changes in drying air temperatures and relative humilities depend on *BAR*

The temperature and relative humidity of the drying air entered the drying chamber (D) was about 41.5 °C and 25% as dampers were closed. They increased and reached about 45 °C and 30% at 40% of *BAR*, owing to increasing the drying air temperature at the condenser inlet and then drying air temperature at drying chamber inlet (D) decreased while relative humidity increased with increasing *BAR*.

As shown in fig. 4, heat pump operating temperatures and capacities affected with *BAR*. Decreasing cooling and dehumidifying capacity of the evaporator affected condenser capacity and other operating temperatures because of lessen air flow rate through the evaporator. All operating temperatures such as compressor inlet, outlet gas temperatures, condensing and evaporating temperature, *etc*. also decreased after 40% of *BAR*.

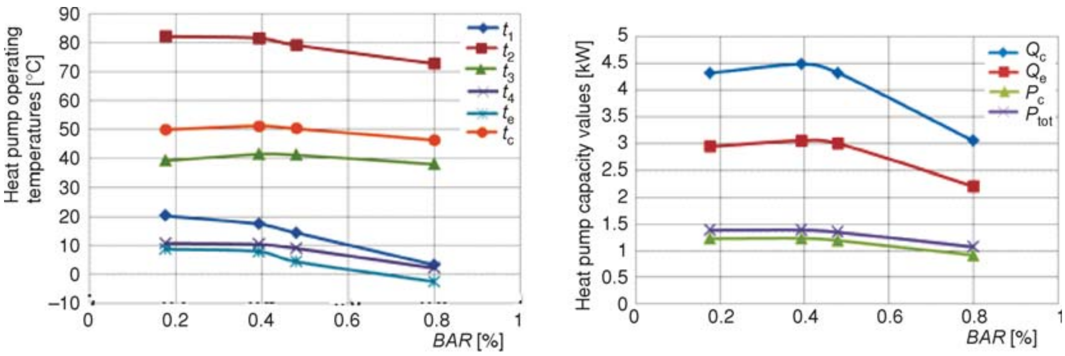

Figure 4. Changes in heat pump operating temperatures and capacities depend on *BAR*

The *MER*, *SMER*, COP_{hp}, and COP_{sys} values depend on *BAR* value were obtained as shown in fig. 5. All of these parameters decreased after about 40% of *BAR*. While *SMER* values changed between 1.2 and 1.4, COP_{sys} changed between 2.8 and 3.3 depend on *BAR*. According to Jolly *et al.* [19], the *SMER* of HPD systems were in the range of 1.0-4.0 kg/kWh, which confirms the results of the present study. Jia *et al.* [20] also found that the *SMER* increased with increasing of *BAR* up to the optimal *BAR* and then it decreased. But, air around the evaporator could only improve *SMER* by approximately 20% and the maximum *SMER* was not very sensitive to the *BAR*. According to Chua *et al.* [1], *SMER* was significantly affected by the *BAR*. A higher *BAR* would reduce *SMER* when the amount of bypass air was increased. Both the heat transfer and transfer coefficient in the evaporator would then decrease as a result of lower air flow rate and lower air velocity through the evaporator. Therefore, by employing the by-pass air across the evaporator method to control the air humidity, the resulting consequence would be a reduction in the heat pump dryer performance. Tai *et al.* [21] reported that the maximum *SMER* was obtained when *COP* is at maximum. However, later studies presented by Jia *et al.* [20] and Prasertsan and Sean-Saby [22] reported that maximum *COP* and *SMER* did not necessarily occur under the same working conditions. According to Oktay [11], this phenomenon was explained by the interaction between the process air in the dryer system and the refrigeration in the heat pump. Because of this interaction, the

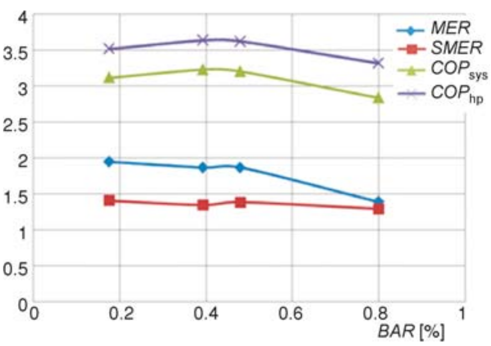

Figure 5. The *MER*, *SMER*, COP_{hp}, COP_{sys} values depend on *BAR*

heat pump and dryer should be evaluated together.

Transient analysis

System was also tested depend on time to investigate temperatures and relative humidifies of drying air through the dryer and the effects of the drying air relative humidity changes at the drying chamber outlet on the system parameters such as *MER*, *SMER*, COP_{hp}, and COP_{sys}. For this, before starting test, evaporative cellulose was humidified completely and then water circulating pumps were stopped. So, humidity evaporative cellulose pad was allowed to dry. For this, firstly two circulating pumps were operated. Second, only one circulating pump was operated and finally, all circulating pumps were cut out. During the drying process, bypass damper was closed tightly.

As seen from fig. 6, during the drying process, temperature and relative humidity of drying air at the dryer inlet was changed about 43 °C and 22% to 46 °C and 18%, while temperature and relative humidity of drying air at the dryer outlet changed from about 30 °C and 70% to

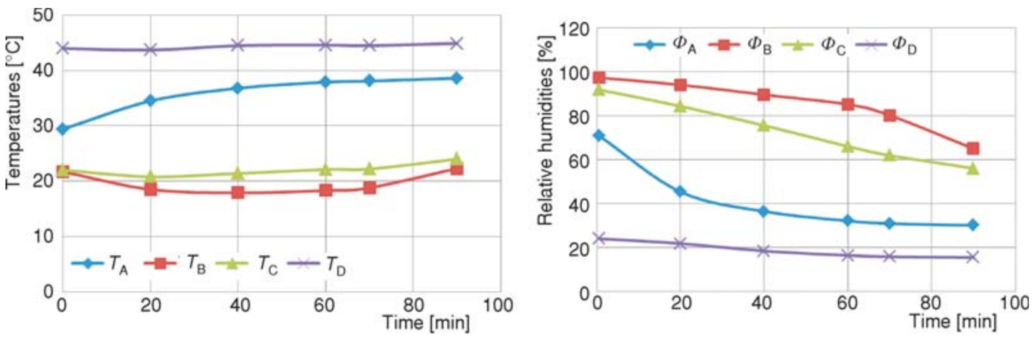

Figure 6. Drying air temperatures and relative humilities in the dryer depend on time

38 °C and 30% depend on time.

The HPD system was tested depend on changing of relative humidifies at the drying chamber outlet. As seen from fig. 7, when the relative humidity at the drying chamber outlet was reduced, all system parameters also decreased depend on time.

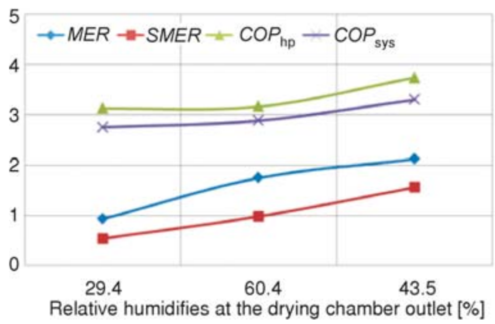

Figure 7. Changes of system parameter's depend on the relative humidify changes at the drying chamber outlet

Conclusions

In this study, a re-circulating heat pump assisted dryer was tested to investigate effects of *BAR* on the system parameters such as *MER*, *SMER*, COP_{hp}, and COP_{sys} and observe temperature and relative humidity changes of air and refrigerant temperature change through HPD system at steady-state condition. Moreover, the effects of the drying air relative humidity changes at the drying chamber outlet on the system parameters depend on time. At the end of the experiments, the following conclusions may be drawn from the present experimental study.

- According to Jolly *et al*. [20], the *SMER* of HPD is in the range of 1.0-4.0 kg/kWh, whereas *SMER* of a conventional dryer is in the range of 0.2-0.6 kg/kWh. The *SMER* value is obtained as about 1.5 kg/kWh in this study. So, the re-circulating HPD system should be preferred instead of conventional dryer.

- When *BAR* is used in order to control the dehumidification rate during the drying process or to mix the air leaving the evaporator with the air leaving the drying chamber to avoid excessive compressor discharge pressures and temperatures, *BAR* should not exceed 40%, otherwise system's performance will be affected negatively and *COP* and *SMER* values will decrease.
- When the relative humidity of drying air at the drying chamber outlet decreased during the drying process depend on time, *MER*, *SMER*, and *COP* values also decreased. However, the discharge pressure and temperature values increased depend on time and so this increment should be considered at design stage. So, the external condenser should be mounted.

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