CHAPTER 8

EXPERIMENTATION ON HEAT PUMP DRYER INTEGRATED WITH AIR CONDITIONING

In this chapter, the thermal performance of the heat pump convective drying system with air conditioning is carried out for hot and dry atmospheric conditions. For the hot atmospheric condition, people use the air conditioner for living comfort, which utilizes a large amount of electrical energy and contributes to environmental pollution. Furthermore, the dehumidification of air is not necessary for open-loop drying due to having low humidity for dry ambient conditions. Hence, it is possible to reduce energy consumption and environmental pollution by combining the heat pump drying and the air conditioning in one system for the hot and dry atmospheric conditions. The main advantage of the combined heat pump drying and air conditioning system is that it utilizes the waste heat of the condenser from the air conditioner for drying as the heat pump dryer has a better overall system performance coefficient because it combines the performance coefficient of the heat pump and the air conditioner. Thus in this study, the experimental investigation of the heat pump system for combined air conditioning and food product drying has been carried out. The new generation future refrigerant R1234yf is used in the HP cycle. The effects of drying time, airflow rate, and air inlet temperature in both evaporator and condenser on the COP of the heat pump system, COP of the air conditioning system, COP of the overall system, evaporator outlet air temperature, condenser outlet air temperature, SMER, and exergy destruction are investigated for the drying of radish in hot and dry atmospheric conditions. The economic comparison of the integrated heat pump system with the individual systems is presented as well.

8.1. Material and methods

The current investigation focuses on the design and fabrication of a convective HP dryer for the drying of the food products with air conditioning using a future refrigerant R1234yf in the HP cycle, and the experiment was carried out for radish drying to analyze the air conditioning system and dryer performance in hot and dry atmospheric conditions. Fig. 8.1 reveals the schematic presentation of the HP dryer with air-conditioning system. The basic components of the heat pump system are a new future refrigerant (R1234yf), semi-hermetic compressor, wavy fin cross-flow heat exchangers (condenser, evaporator), expansion device (capillary tube), two fans (one for condenser and one for evaporator) [details are provided in Chapter 3]. The fabricated experimental facility is shown in Fig. 8.2 for the combined HP drying and air conditioning system. To measure the refrigerant temperature in the HP cycle at the location of compressor input, compressor output, condenser output, and evaporator input, four resistant thermometers (PT100) are installed. The low temperature (evaporator) side pressure and the high temperature (condenser) side pressure is estimated by using the pressure gauges as shown in Fig. 8.2. Three energy meters are installed to estimate the electric energy requirement by two fans (one for evaporator and another for both condenser and dryer) and the compressor.

In the heat pump system with air conditioning, two separate fans are installed to control the airflow through the condenser and the evaporator to get the desired drying and cooling properties of air. The main advantage of this type of system is that it can be used for both purposes as a heat pump dryer and air conditioner with a very low global warming effect because it utilizes the waste heat of the condenser for drying. This system is having two fresh air inlets, one for the heat pump dryer (through the condenser) and the second for the air conditioner (through the evaporator). The convective drying cabin of the inside dimension of 0.35m ×0.4m ×0.7m and the drying chamber consisted of several trays inside it. The drying air flows in the HP drying system with the help of a fan (220/230V, speed = 1350rpm, 50Hz, power = 70W, sweep = 300mm). The resistant thermometers and pressure transducer were used in the system to estimate the drop in pressure and temperature of the air in the drying cabin and the whole system.

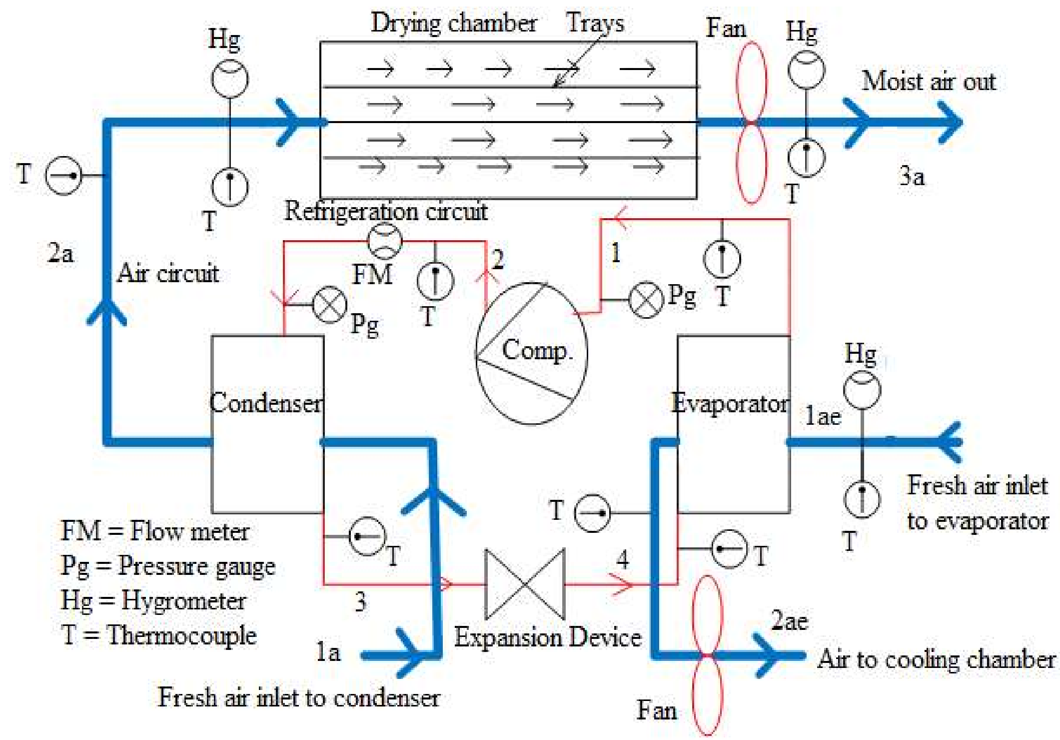

Fig. 8.1: Schematic of the presentation of HP dryer with cooling of air

The integrated heat pump system for both drying and air conditioning is designed and fabricated to investigate the performance of the air conditioner and HP dryer for the drying of the radish. The radishes are washed and sliced into 2 mm size. The moisture content (final and initial) of the radish chips is estimated by applying the oven method at 105 °C. 4.5kg of radish chips was taken for the experiment in the new system. The radish chips on the trays loaded inside the drying cabin. Four hygrometers (at inlet and outlet to drying cabin and evaporator) and the resistant thermometers (PT100) are used in the HP drying with an air conditioning system to measure the evaporator inlet and outlet, condenser outlet, and drying cabin outlet temperature which is also used to optimize the drying air temperature by controlling the airflow rate through the system.

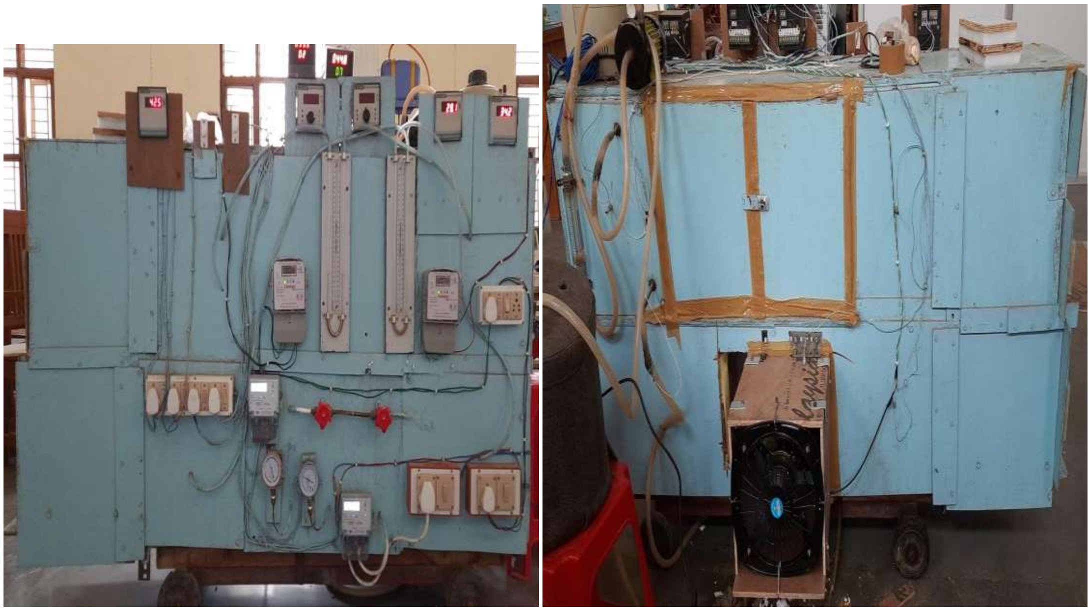

Fig. 8.2: Developed experimental facility of HP dryer with cooling of air

The experiment is carried out to evaluate the exergy, economic, and energy performance of the HP system at different airflow rates through the condenser and evaporator and at different atmospheric conditions (inlet to evaporator and condenser) for the radish chips drying in the system. The fans connected to the evaporator and condenser were switched on and fixed to the desired value of the airflow rate. After that, the R1234yf HP-system made switched on to reach the steady-state condition for the first fixed condition of inlet temperature and flow rate. Then the initial readings of the resistant thermometers, energy, and hygrometers meters were recorded, and during the experiment, the humidity at evaporator inlet and outlet, drying cabin inlet and outlet, and drying air temperature at outlet and inlet to the drying chamber, evaporator, and condenser was recorded after every 10minutes until the end experiment. The experiment is performed for the different velocities of air 1.0m/s, 1.5m/s, 2.0m/s, and 2.5m/s through the evaporator and condenser and at different inlet temperatures oar air to the HP drying and air conditioning system. For the current study, the system is operated in open mode (separate inlet for condenser and evaporator) without mixing of air. The decrease of moisture content from the product should be equal to the increase of drying air humidity in the drying cabin and also equal to the amount of moisture condensed over the evaporator for the closed system. The energy measuring devices are connected to measure the energy required for the fan (condenser and evaporator) and compressor.

8.2. Data extraction

The heat exchange in the HP condenser between refrigerant and air is given by,

$$Q_{cond} = \dot{m}_{a,cond} c_{pam} (T_{2a} - T_{1a}) = \dot{m}_r (h_2 - h_3) \tag{8.1}$$

The heat exchange in the HP evaporator is given by,

$$Q_{evap} = \dot{m}_{a,evap} (h_{1ae} - h_{2ae}) = \dot{m}_r (h_1 - h_3) \tag{8.2}$$

The performance of the heat pump is generally evaluated as heating COP. But, here, the new term overall heating coefficient of the performance is used for the performance evaluation of the heat pump and it is given as (Byrne and Ghoubali, 2019),

$$\text{OHCOP} = \frac{Q_{cond}}{W_{comp} + W_{fan,evap} + W_{fan,cond}} \tag{8.3}$$

The performance of the air conditioner (cooling effect) is evaluated in terms of cooling COP. But here, the new term overall cooling coefficient of the performance is used for the performance evaluation of air conditioners and it is given as,

$$\text{OCCOP} = \frac{Q_{evap}}{W_{comp} + W_{fan,evap} + W_{fan,cond}} \tag{8.4}$$

The overall system coefficient of performance (OSCOP) is used to evaluate the combined performance of the system (heat pump dryer + air conditioner), and it is estimated as the ratio of the amount of heat transfer in the HP-condenser and evaporator to the total fan and compressor energy inputs. The OSCOP is given as follows (Byrne and Ghoubali, 2019),

$$\text{OSCOP} = \frac{Q_{cond} + Q_{evap}}{W_{comp} + W_{fan,evap} + W_{fan,\text{cond}}} \tag{8.5}$$

The overall energy efficiency of the heat pump drying with the air conditioning system indicates the utilization of the amount of total energy supplied to the system. In this system, the drying and the air conditioning if performed simultaneously, then the energy efficiency of the overall heat pump drying with air conditioning system can be given by,

$$\eta_{en} = \frac{(m_w / t_d)\, h_{fg} + Q_{evap}}{W_{comp} + W_{fan,cond} + W_{fan,evap}} \tag{8.6}$$

In the exergy analysis of drying and air conditioning with HP, the exergy output, input, and destruction of exergy in the HP system for various components are figured out. Based on the exergy flow of refrigerant and humid air various components of the system, the component exergy destruction and exergy efficiency have been evaluated. The exergy analysis is carried out based on Eq. discussed in previous Chapters.

The economic analysis of the drying with an air conditioning system is related to the investment cost and operating cost of the drying system. The capital cost of the drying system includes the costs of heat exchangers (condenser, evaporator), compressor, capillary tube, fan, refrigerant, duct, base structure, system fabrication, and labor. The amount running cost of the drying system is estimated by (Yahya et al.,2018),

$$C_{RU} = C_{RM} + C_P + C_L + C_m \tag{8.7}$$

Where C_{RU} is the running cost (\$), C_{RM} is the cost of material (\$), C_p is the cost of energy requirement (\$), C_L is the labor cost (\$), C_m is the maintenance cost (\$). Where maintenance cost (C_m) is considered as 2 % of the total capital cost (Yahya et al., 2018), In the present economic analysis, the HPD or air-conditioning system is compared with the combined (heat pump dryer and air-conditioning) system. The raw material cost and the labor cost is the same for both the system and canceled out in comparison, thus the economic analysis can be carried out based on energy consumption cost and the maintenance cost only, and the equation can be written as,

$$C_{RU} = C_P + C_m \tag{8.8}$$

The total cost (C_{Total}) of the HP drying with an air conditioning system is given as,

$$C_{Total} = C_{IC} + C_{RU} \tag{8.9}$$

Where C_{IC} is the initial investment cost ($)

The annual profit by using the proposed integrated HP system for drying radish chips and air conditioning is the difference between the total cost to run the HP system separately for drying and air conditioning and the total cost to run the HP system for combined drying and air conditioning. Hence, annual profit is given as the following,

$$C_F = C_{RU,dryer} + C_{RU,airconditioning} - C_{RU,dryer+airconditoning} \tag{8.10}$$

Where C_F is the total profit ($)

8.3. Experimental results and discussion

The chips (radish) were dried in the HP dryer with an air conditioner for different atmospheric temperatures in hot and dry conditions, and the energetic and exergetic performances of the drying with the air conditioning system were considered for analysis. Performance of the cooling system, drying system, and the overall system was investigated for different airflow rates and atmospheric conditions. The performance of the HP and air-conditioning system depends on the atmospheric conditions and the airflow rate in the system. The human comfort zone and the various process of air on the psychrometric chart are shown in Fig. 8.3. Fig. 8.4 represents the variation of the evaporator outlet temperature (cooling temperature) with different evaporator inlet temperatures and airflow rate for fixed condenser inlet conditions and airflow rate. The cooling temperature is higher for the higher mass flow rate of the air through the evaporator for the same inlet temperature, and also, the cooling temperature increases with inlet atmospheric temperature at a fixed flow rate. The evaporator output temperature is lower for the lower mass flow rate of air and found to be in the comfort zone (temperature varies between 22 to 27°C, relative humidity between 40-60%) of cooling for the higher outside atmospheric temperature of 45°C in the current HP drying and air conditioning system.

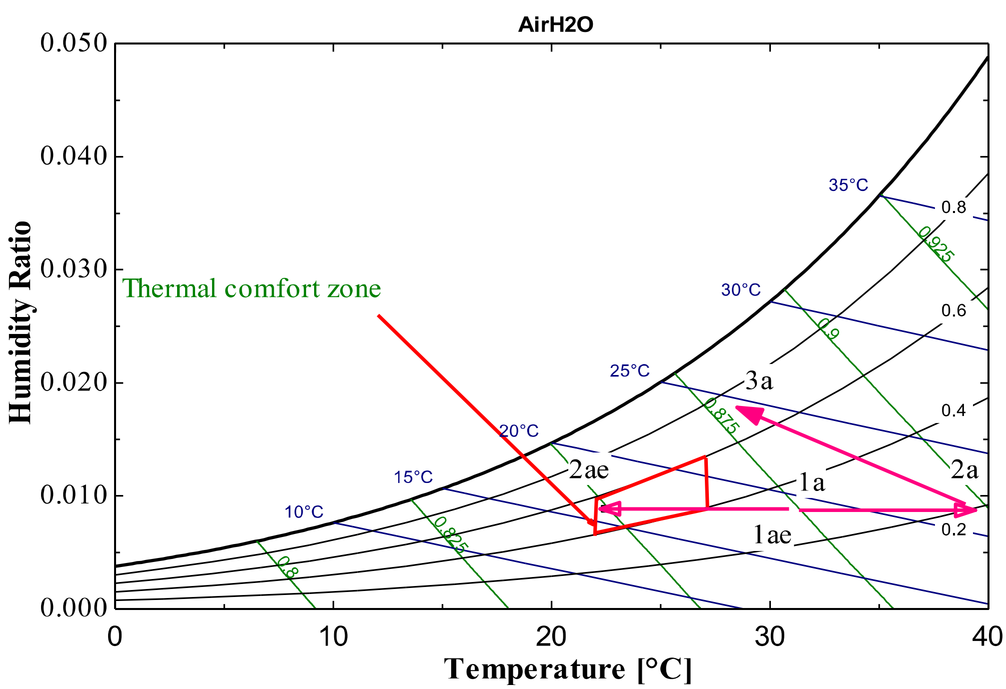

Fig. 8.3: Human thermal comfort zone and various air processes on a psychrometric chart

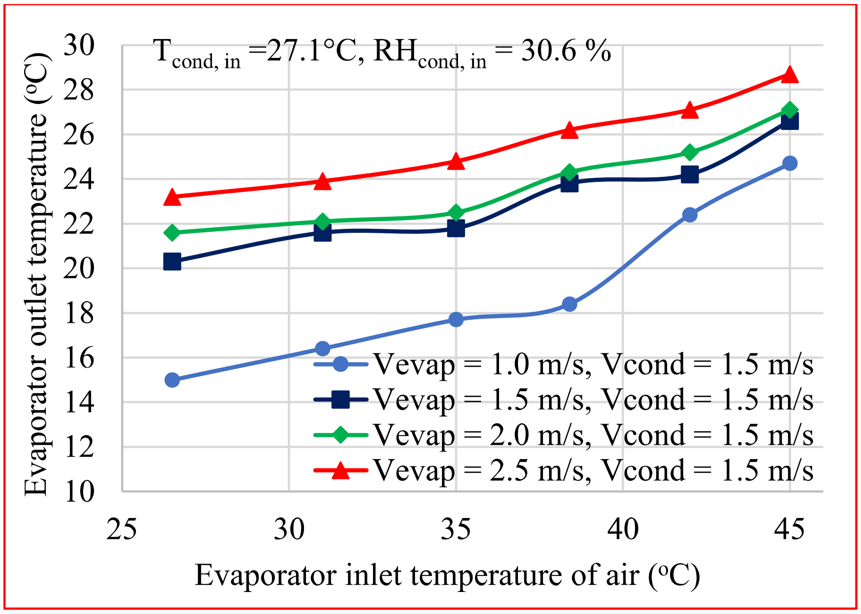

Fig. 8.4: Variation of evaporator outlet temperature according to atmospheric temperature

Fig. 8.5 indicates the effect of the different evaporator inlet (air) temperatures on the condenser outlet temperature that is drying temperature for the fixed airflow through the condenser and inlet temperature. The drying temperature is higher for the higher mass flow rate and inlet temperature of air through the evaporator. A higher drying temperature of 54.6°C is found for the evaporator inlet temperature of 45°C and air velocity of 2.5m/s through the evaporator. Fig. 8.6 represents the evaporator outlet relative humidity of the air for the different inlet conditions of the atmospheric for the fixed inlet condition of the condenser for the combined system. The evaporator outlet relative humidity (50.2%) of air is found in the comfort zone for the air velocity of 1.0m/s for both condenser and evaporator for the atmospheric temperature of 42°C. Thus the result from the experiment shows that the developed HP dying system can be used for the air conditioning and also for the atmospheric temperature of between 42-45°C in the hot and dry zone.

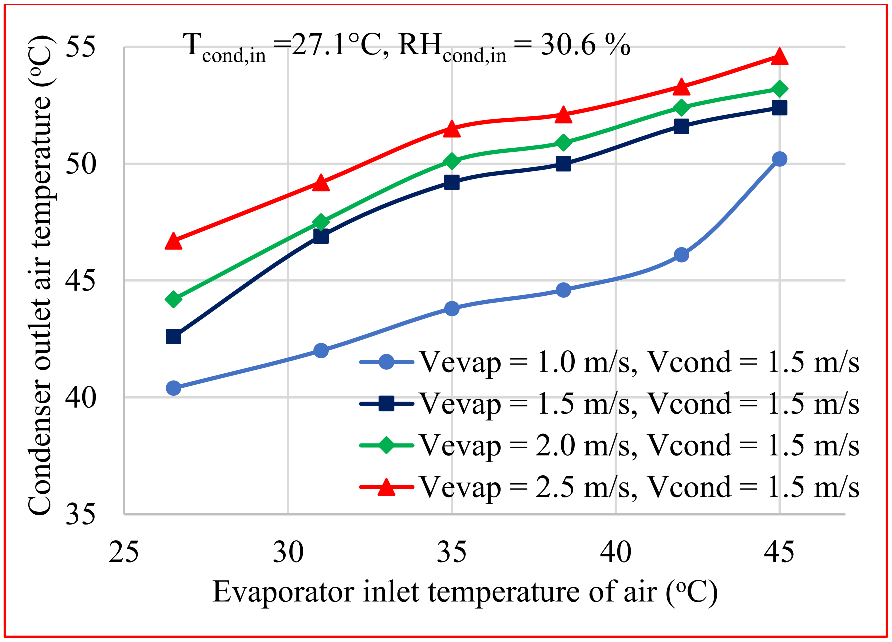

Fig. 8.5: Condenser outlet temperature variation with atmospheric temperature

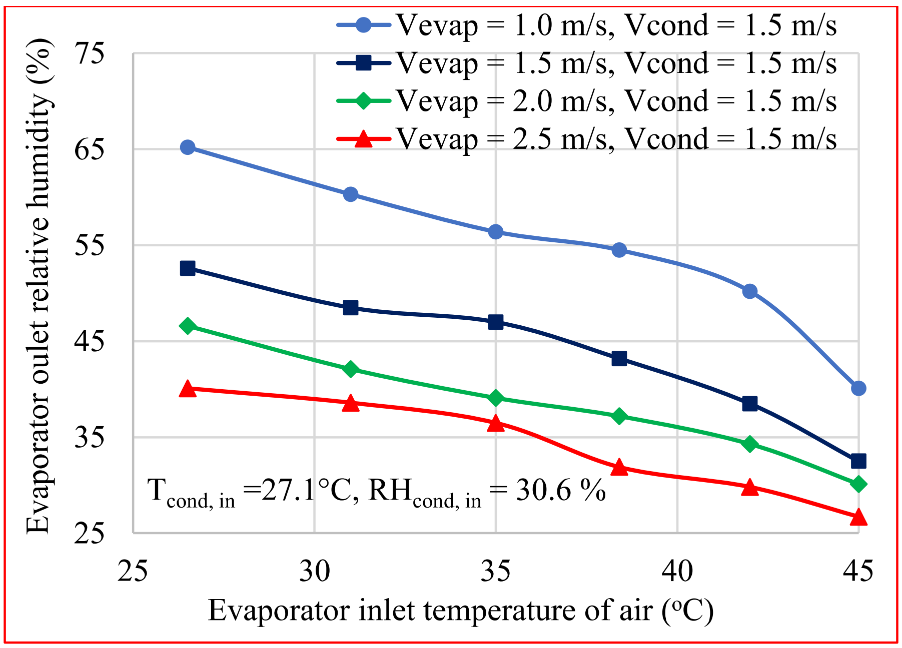

Fig. 8.6: Evaporator outlet relative humidity with atmospheric temperature

The COP of the HP system and air conditioning system was a little bit higher for the lower values of temperature and humidity input to the evaporator (the value of COP of HP and air conditioner decreases with an increase in evaporator inlet air temperature and humidity) as compared to the higher value of input air temperature and humidity. In the current study of the HP system and the air conditioning system, three types of performance coefficients were defined. The overall cooling coefficient of the performance (OCCOP) for the air condition system, the overall heating coefficient of the performance (OHCOP) for the HP system, and the overall system coefficient (OSCOP) for the combined system of the air conditioning and the heat pump. Fig. 8.7 indicates the effect of the evaporator inlet temperature on the OCCOP for the fixed inlet condition of the condenser at different air velocities through the evaporator. From the figure, it can be concluded that the OCCOP decreases with an increase in the evaporator inlet temperature as well as with air flow rate due to considering fresh atmospheric air inlet to the evaporator (Bellos and Tzivanidis, 2019). But the OCCOP of the system increases with the increase in the inside temperature of the cooling chamber (evaporator outlet temperature). The average OCCOP is found to be 3.227 for the velocity of 2.5m/s in the inlet temperature range of 27-45°C.

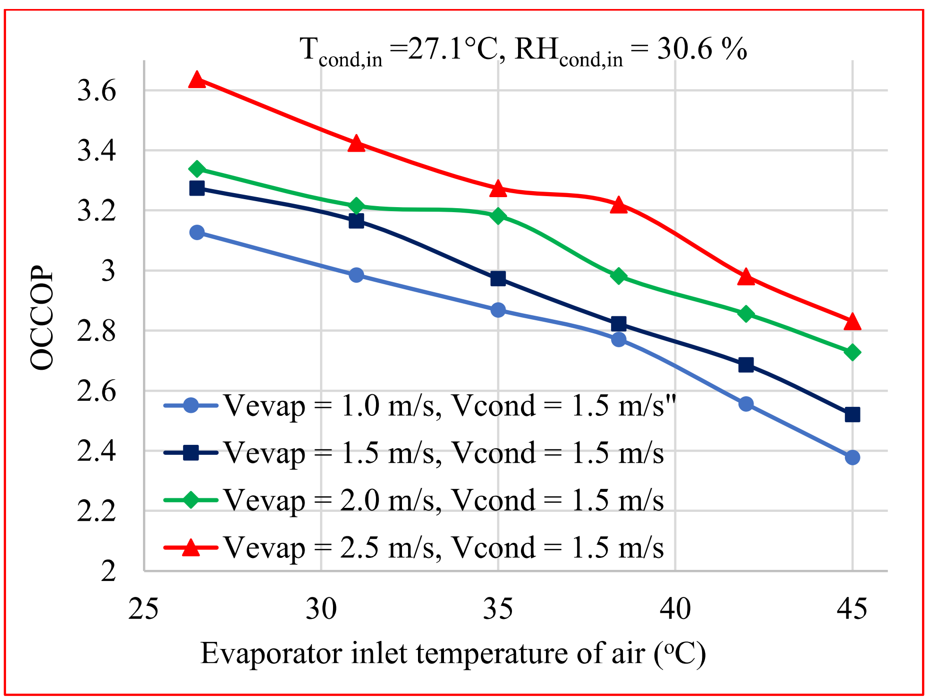

Fig. 8.7: OCCOP variation with evaporator inlet temperature

The most important performance parameter of the combined heat pump and the air conditioning system is the overall system coefficient of performance because it represents the combined effect of the cooling and heating on the system. OSCOP gives the combined efficiency of the system for the same energy consumed by the system. Fig. 8.8 shows the variation of OSCOP with evaporator input temperature, and the value of the OSCOP is found to be higher for the higher air velocity through the evaporator for the fixed input condition of the condenser and its value decreases with an increase in input temperature to the evaporator (Bellos and Tzivanidis, 2019). The average value OSCOP for the velocity of 2.5m/s in the temperature range of 26-45°C is 7.456.

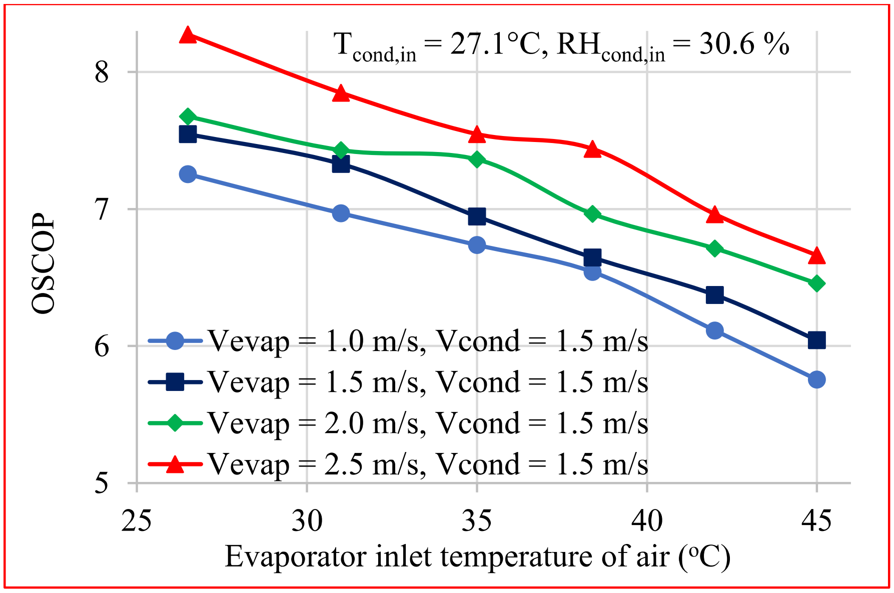

Fig. 8.8: OSCOP variation with evaporator inlet temperature

Fig. 8.9 depicts the outlet air temperature variation of both evaporator (for cooling) and condenser (for drying) with the atmospheric temperature at a fixed mass flow rate (V_{cond} = 1.0 m/s, V_{evap} = 1.0 m/s) through both condenser and evaporator. The cooling and the drying temperature of the air at the atmospheric temperature of between 42-45°C are found to be 22.6-24.7°C and 62.1-63.9°C, respectively. The cooling temperature and the humidity of the evaporator outlet were found to be in the comfort zone for the current studied combined system if a higher atmospheric temperature of between 42-45°C in hot and dry conditions.

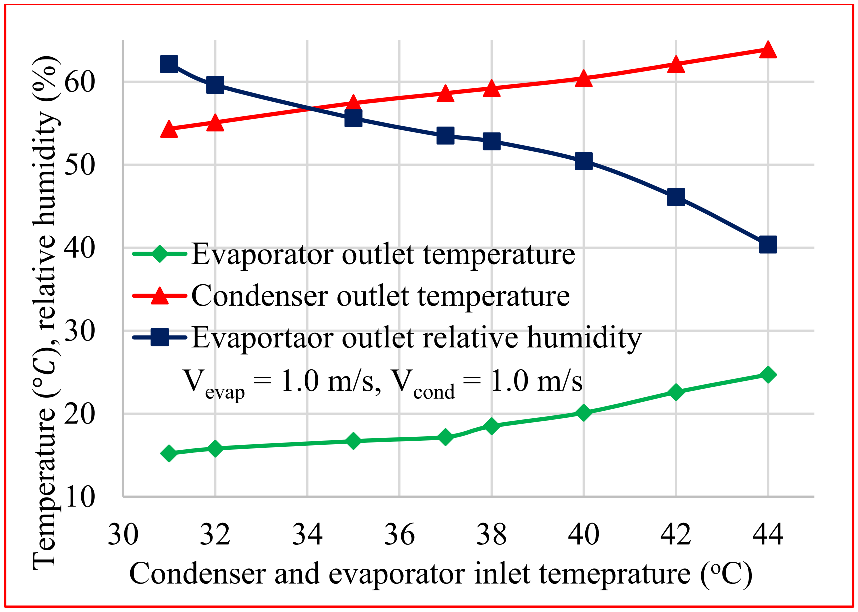

Fig. 8.9: Temperature and relative humidity variation with atmospheric temperature

The effect of the condenser inlet air condition on the cooling and the heating temperature for the fixed evaporator inlet condition and mass flow rate is presented in Fig. 8.10. For the figure, it can be concluded that there is no effect of the condenser inlet air temperature on the air outlet condition of the evaporator as refrigerant state points are the same if the other parameters are constant. The condenser outlet temperature only increases with an increase in condenser inlet temperature due to the same reason.

Fig. 8.11 indicates the variation of the SMER for the drying of the product at the mean inlet temperature of 38.4°C to the condenser and evaporator. In this system, there are two inlets, one for the condenser and one for the evaporator. The SMER first increases with drying time because initially product having higher moisture content, so moisture removal is also higher. But after some time, it decreases due to the loss of water content from the product. The average SMER is found to be 2.098kg/kWh.

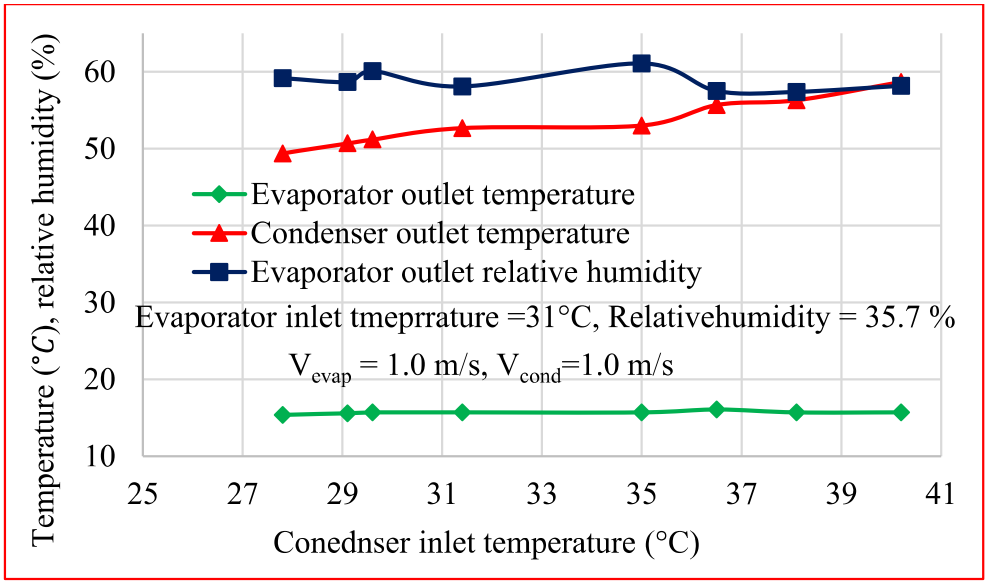

Fig. 8.10: Temperature and relative humidity variation with condenser inlet temperature

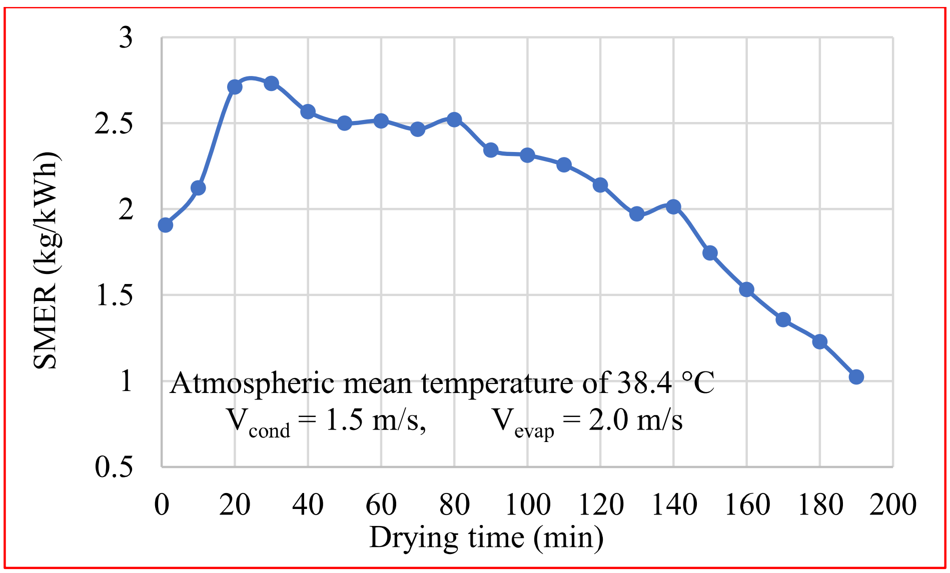

Fig. 8.11: SMER for drying at a mean atmospheric temperature of 38.4°C

Exergetic investigation is the most relevant methodology to evaluate the thermal performance and efficient energy utilization in any HP system. The irreversibility occurs in the evaporator due to the sensible cooling and the condensation of air and moisture over the cold finned surface. The value of exergy destruction and the exergy efficiencies are given in Table 8.1 for the mean inlet temperature of 38.4°C to the condenser and evaporator. The exergy destruction was lower for the expansion device due to the negligible heat exchange in the expansion device. The exergy (destruction) was higher for the higher inlet temperature of the condenser and evaporator in the HP and air conditioning system. The total values of exergy destruction for the different atmospheric conditions are presented in Fig. 8.12.

Table: 8.1. Component-wise irreversibility and exergy efficiency

System component	Irreversibility (kW)	Exergy efficiency
Compressor	0.203	0.631
Condenser	0.1705	0.912
Expansion device	0.0866	0.753
Evaporator	0.162	0.728
Drying chamber	0.1268	0.315

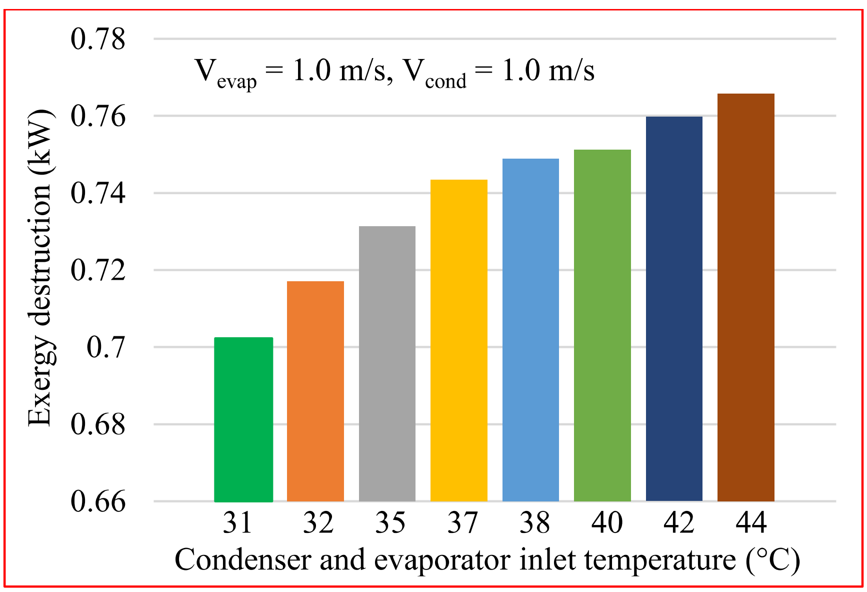

Fig. 8.12: Exergy destruction of the system at different atmospheric temperatures

The component-wise exergy destruction is shown in Fig. 8.13 for a mean atmospheric (evaporator and condenser inlet condition) temperature of 38.4°C. From the figure, it can be observed that the maximum exergy destruction occurs in the compressor and is minimum in the expansion device. In the compressor, the exergy destruction is higher due to the mechanical and electrical losses, and in the expansion device, exergy destruction is lower due to negligible heat exchange (isenthalpic expansion).

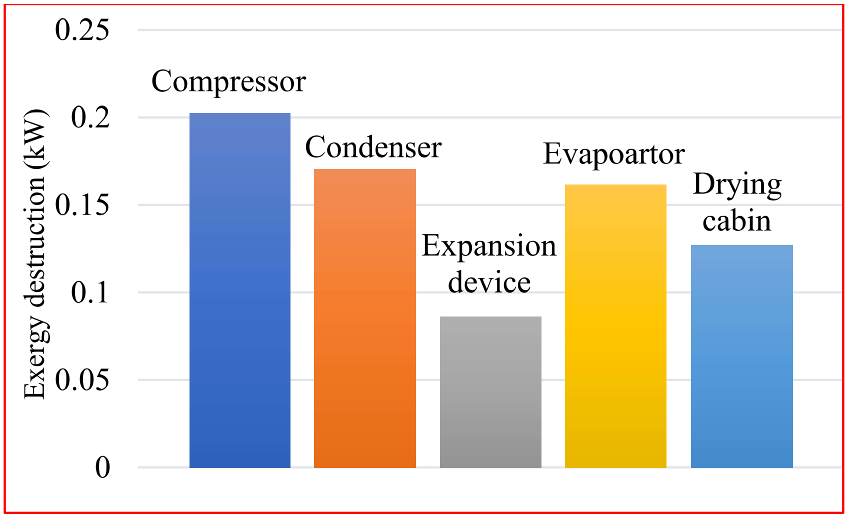

Fig. 8.13: Component wise exergy destruction at an atmospheric temperature of 38.4°C

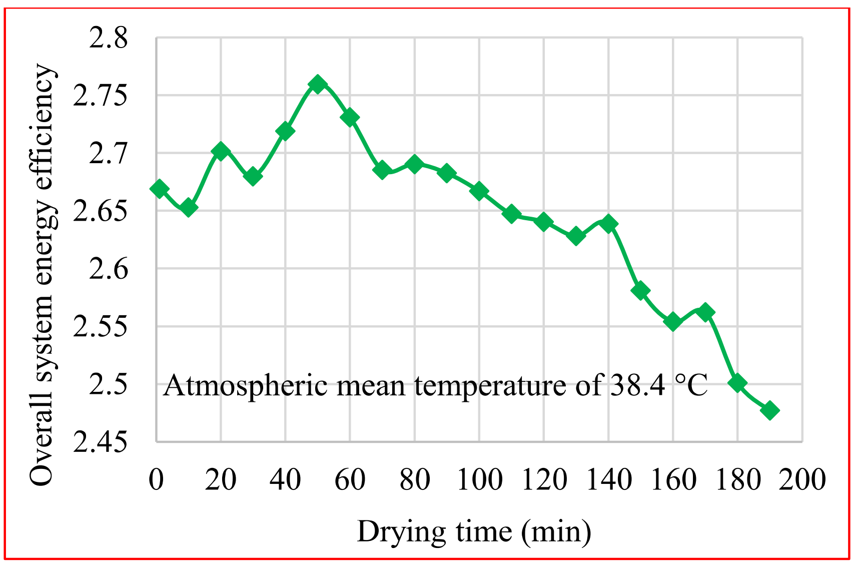

Fig. 8.14: Overall system energy efficiency at mean atmospheric temperature of 38.4°C

The overall energy efficiency represents the utilization of the input energy in the drying with an air conditioning system. in the present system, the drying and the cooling of the air were carried out simultaneously, so the energy utilization occurred in both cooling and drying (moisture removal from the product). Fig. 8.14 shows the variation of the overall system energy efficiency with drying time at atmospheric (condenser and evaporator inlet temperature) of 38.4°C for condenser and evaporator air velocity of

1.5m/s and 2.0m/s, respectively. The overall energy efficiency first increases with time due to an increase in moisture removal from the product, but after some drying time, its value reduces due to the reduction in moisture content of the product.

The economic investigation of the HP dryer with air conditioning for different input conditions to the system was considered for the total profit (annual) by saving the energy to dry product with air conditioning simultaneously, initial capital investment, total annual running cost, and payback period for HP drying with air conditioning at the atmospheric condition of 38.4°C. The annual maintenance, initial investment cost, effective running cost, total selling cost, net profit, the cost of the dryer, and the payback period are listed in Table 8.2. The annual (300 days) and daily running costs of the HP dryer and air conditioning (as a single system) and HP dryer with air conditioning (combined system) were calculated. The running cost includes the energy requirement, and product and labor costs. The energy requirement cost is the most significant parameter within the running costs for the air conditioning and heat pump drying systems. The initial cost of the HP dryer system and the HP drying with air conditioning are calculated as \$573.2 and \$577.3, respectively. The total (daily) running cost of both single system HP dryer and air conditioning is \$ 2.05/day and the running cost (daily) of the combined HP drying and the air condition system was \$1.02/day. The total annual running cost of both single system HP dryer and air conditioning is \$615/day and the running cost (daily) of the combined HP drying and the air condition system is \$306/day. The total annual profit gain by using the combined system (HP dryer + air conditioning) instead of the separate systems of HP dryer and air conditioning is \$309.

Table: 8.2. Economic analysis for heat pump drying with air conditioning

Component cost (\$)	Separate running	Combine running
Compressor (\$)	136.03	136.03

Condenser ($)	89.75	89.75
Evaporator ($)	72.92	72.92
Expansion device ($)	3.51	3.51
Drying cabin ($)	46.61	46.61
Refrigerant (R1234yf) ($)	24.68	24.68
Fan (Two) ($)	33.66	33.66
System fabrication cost ($)	170.14	170.14
Initial investment ($)	577.3	577.3
Energy consumption (kWh/day)	15.24	7.62
System running cost ($/day)	2.05	1.02
Maintenance cost (2 % of the initial cost), annual	11.546	11.546
System running cost ($/year)	626.546	317.546
Total profit ($/year)	....	309

8.4. Highlights

The HP drying system with air conditioning has been designed and fabricated in the laboratory and the thermal performance of the system was experimentally investigated for the different atmospheric temperatures and airflow through the system. A comparative investigation of different input temperatures to the system has been carried out (for drying of radish) at a different flow rate of air. The effect of the different input conditions on the combined system has been investigated. The conclusions are as follows:

- The COP of the HP system and air conditioning system decreases with increases in the input temperature to the evaporator (atmospheric air inlet) and its value is higher for the higher mass flow rate of air through the system.
- The overall system coefficient of performance (OSCOP) is found to be much higher than the COP of the single system (heat pump or air conditioner) with an average value of 7.456 in the input temperature range of 26-45°C because it gives the advantage of both systems with the single energy input source.
- The evaporator outlet air temperature and the humidity were between 22.6-24.7°C and 40.4-50.2 %, with a drying temperature of 62.1°C for the atmospheric temperature of 42°C. Thus this system can be used for both purposes as an HP dryer and air conditioner for the maximum atmospheric temperature of 45°C.
- Overall system performance mostly depends on the inlet condition and mass flow rate of air through the evaporator.
- Condenser inlet conditions is having a negligible effect on the system performance if the evaporator inlet conditions are fixed.
- The total exergy destruction is highest for compressor and lowest for expansion device.
- The total annual profit gain by using the combined system instead of the separate systems of HP drying or air conditioning is $309.
- Thus this system can be recommended for the application where both drying and air conditioning are required (such as hotels where both laundry and air conditioning are required) because it gives the advantage of both systems in single-unit energy saving (only single energy input for both systems).
- The energy savings can be achieved by using the combined heat pump dryer and air conditioning system instead of the individual system.