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A modular approach to study the performance of a two-stage heat pump system for drying

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Abstract

The performance of a two-stage evaporator heat pump drying system with chlorodifluoromethane (HCFC22) as the working refrigerant was experimentally studied. The system employed low- and high-pressure internal evaporators for cooling and dehumidification of the drying air. On the condenser side, two subcoolers were used for additional sensible heating. The system further incorporated a passive evaporator–economiser for pre-cooling of the air before the evaporators and pre-heating before the condenser. The system performance was evaluated in terms of the amount of heat recovered at the evaporators, the system COP and SMER. It was demonstrated in this paper that more heat, up to 35% more, could be recovered via the two-stage evaporator system in comparison to a single evaporator system. By varying the pressure level of the high-pressure evaporator, greater amount of heat recovered could be realised when the ratio of latent to total load at the two-stage evaporator was improved.

Keywords: Heat pump drying; Two-evaporator; Economiser; Subcoolers

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Nomenclature

COP	coefficient of performance
HP	high pressure
LP	low pressure
m	mass flow rate, $\mathrm{kg\,s^{-1}}$
$\dot{m}_w$	water condensed at the evaporator per unit time
Q	heat transfer rate, or coil capacity, W
SMER	specific moisture extraction rate, $\mathrm{kg\,kW\,h^{-1}}$
T	temperature, K
T_a	air temperature, K
T_s	superheated temperature of refrigerant, K
ΔW	change in absolute humidity of air, g/kg dry air
$\dot{W}_c$	power input to the compressor, kW

Subscripts

1	inlet
2	exit
a	air
cd	condenser
e	evaporator
hp	heat pump
off	air-off heat exchanger
on	air-on heat exchanger
r	refrigerant

1. Introduction

Commercial heat pump systems range in size from small domestic units with cooling rates of the order of hundred of watts to industrial systems that can deliver tens of megawatts of heating effect. Especially for large system, it is common knowledge that the energy efficiency of heat pumps can be significantly enhanced when more thermal energy from the heat source is made available for heat recovery. By incorporating the heat pump system to a chamber for drying applications, it is possible to recover significant amount of latent energy from the moisture-laden air. One possibility of enhancing the amount of heat recovered is via the employment of a two-stage evaporator system. According to Li and Su [1], a refrigerating system with two or more evaporators performs better than that with only one evaporator. From the thermodynamic viewpoint, a two-evaporator system is equipped with larger surface area for heat recovery, resulting in reduced compressor work to drive the cycle.

Brundrett [2] has depicted in a chart the energy efficiency of different heat pump configurations. The chart illustrates the improvement in the heat pump energy efficiency from a single to two-stage dehumidification system. Researchers such as Rose et al. [3], having tested domestic two-evaporator refrigerators with zeotropic refrigerant mixtures, concluded that improved power saving of up to 20% was possible compared with a system having only one evaporator. The computer simulation of a two-evaporator refrigerating system charged with pure and mixed refrigerants conducted by Jung and Radermacher [4] showed a significant increase in COP. Simons et al. [5] also found favourable energy saving potential after testing a two-evaporator refrigerant cycle with some bypasses to the compressor.

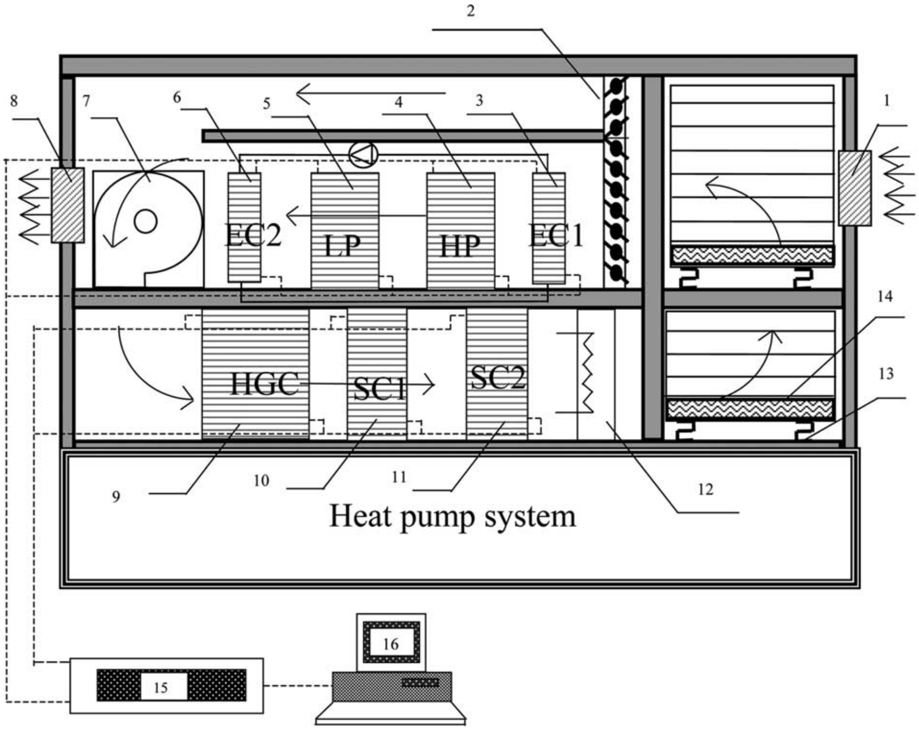
According to Invernizzi and Angelino [6], the most direct way to improve the efficiency of simple heat pump cycles and thereby enabling more heat recovery is via the introduction of multistage throttling and compression process. Heat pumps are known to be energy efficient devices when used in drying operations. Their ability to convert the latent heat of condensation into sensible heat at the hot gas condenser makes them unique in drying applications. Further, their ability to produce well-controlled drying conditions is particularly important to drying high-valued quality products. Many researchers have acknowledged the importance of producing a range of precise drying conditions to dry a wide range of products, particularly for high-valued agricultural products. In this paper, we study the thermal characteristics and performance of a two-stage heat pump drying system comprising a low-pressure evaporator (LPE) and a high-pressure evaporator (HPE) followed by two subcoolers installed in series with the condenser. The performance of the dryer will be studied under modular change from a basic heat pump cycle to a more complex one comprising a two-stage evaporator and one that further includes an economiser and two subcoolers. Performance of the heat pump will be evaluated in terms of the amount of heat recovered, COP (Coefficient of Performance) while performance of the dryer may be quantified in terms of the Specific Moisture Extraction Rate (SMER). Chlorodifluoromethane (HCFC22) was employed as the working medium in the present experimental set-up which is beginning to realise lesser usage for commercial heat pump systems. However, based on the recent work carried out by Rakhesh et al. [7], heat pump systems employing HFCs such as R407C and R407A have shown to perform better with higher COP and exergy efficiency compared to one using HCFC22 as its working medium. Further, Jung et al. [8] also demonstrated experimentally that heat pump systems employing HFC125 and HFC134a have 4–5% higher COP and capacity than one using HCFC22. Therefore, it is expected that a more energy efficient heat pump drying system can be realised if these mentioned HFCs were used as the working fluid.

The commercial/industrial implications of the present research would be the design of a highly energy-efficient heat pump drying system with good air conditioning control mechanism to produce high quality dried agricultural and marine products.

2. Experimentation

2.1. Test facility

A two-stage modular heat pump dryer designed and built at the department of Mechanical Engineering is shown in Fig. 1. This advanced facility was necessary so that the measurements of refrigerant and air states permitted the computation of essential heat pump characteristics as



Item #	Parts	Description	Item #	Parts	Description
1	-	Fresh air intake air louver	9	HGC	Hot gas condenser
2	-	Air damper	10	SC1	Subcooler 1
3	EC1	Economiser 1	11	SC2	Subcooler 2
4	HP	High pressure evaporator	12	-	Heating bank
5	LP	Low pressure evaporator	13	-	Load-cell
6	EC2	Economiser 2	14	-	Water tray
7	-	Centrifugal fan	15	-	Micromac data-logger
8	-	Exhaust air louver	16	-	Personal computer

Fig. 1. Schematic arrangement of the two-stage modular heat pump dryer.

it evolved from a simplified heat pump system to an advanced two-evaporator subcooler system. The heat pump employed a 3.73 kW scroll compressor having a swept volume of 13.8 m^3/h. The test facility comprised of two direct-expansion evaporators and three hot coils consisting of one hot gas condenser and two subcoolers as shown in Fig. 2. Two internal evaporators, one at high and one at lower pressure, interacted with the drying air for dehumidification purpose. The evaporators were of finned-tube coils consisting of eight rows and five circuits for the high-pressure evaporator and 10 rows and five circuits for the low-pressure evaporator. The coils had staggered configurations. Fig. 3 shows the refrigerant connection of the two-stage evaporator which

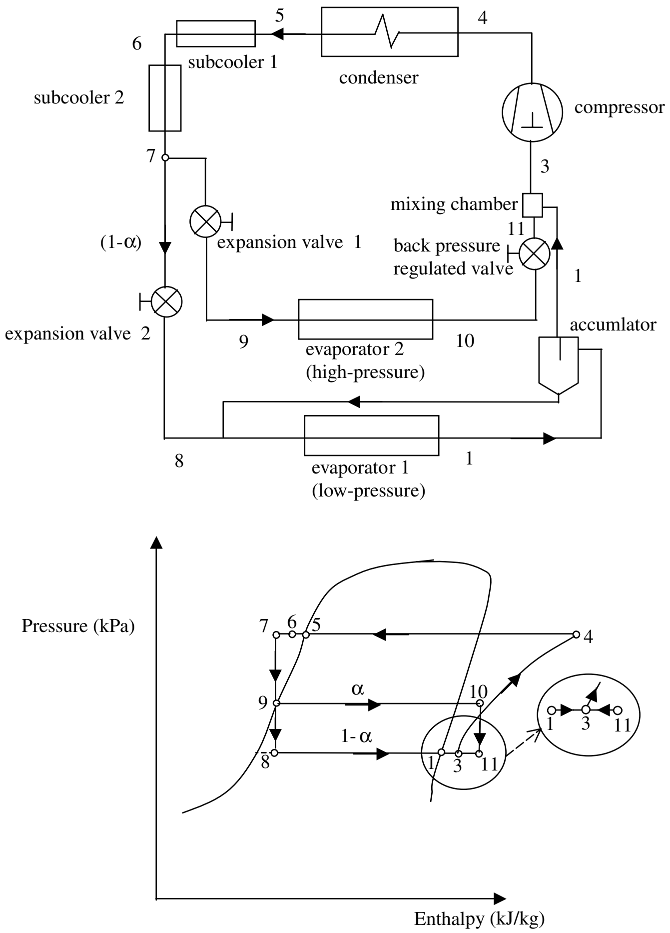


Fig. 2. Refrigerant cycle of the heat pump cycle with two evaporators and a single compression stage.

constitutes part of a heat pump drying cycle. The two evaporators were coupled by a back-pressure regulator to regulate the pressure from that of HPE to the LPE. Thermostatic expansion valves were used to control the degree of superheat at the evaporator outlet by regulating the flow rate of the refrigerant. Each expansion valve had a remote bulb with HCFC22 and an external equaliser that was connected to the suction line on the compressor side of the remote bulb. As the suction temperature varied with time, the vapour pressure in the remote bulb changed accordingly. The pressure signals were transmitted through the capillary tube to the expansion valve to control the refrigerant flow rate. The external equaliser served to overcome excessive pressure drop in both evaporators. It also helped to prevent excessive vapour superheat from occurring; preventing the significant loss of cooling capacity. A passive evaporator–economiser was also installed to recycle part of the sensible heat load of the evaporator, thus hoping to improve the latent/total cooling ratio of the evaporators. A description of key system refrigerating components for the present test facility is given in Table 1.

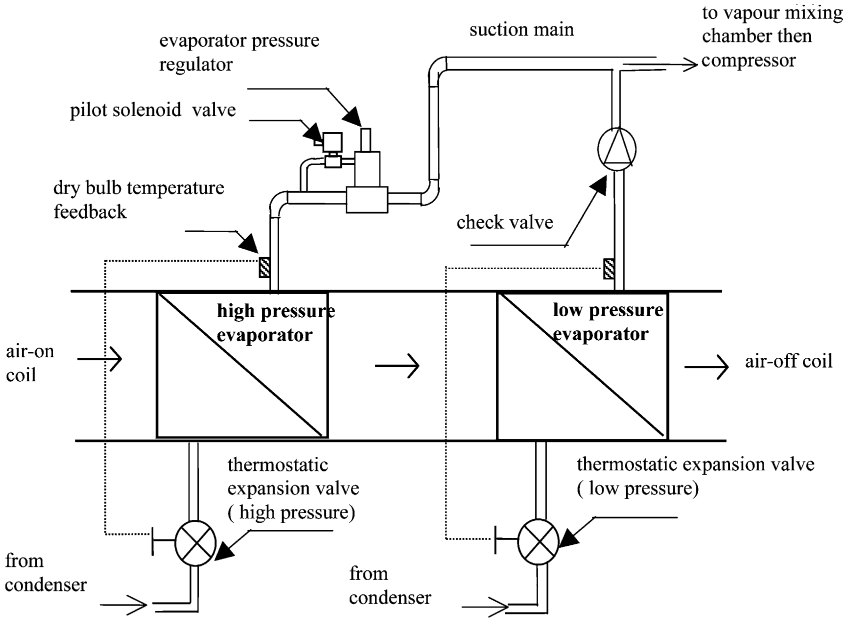


Fig. 3. Schematic diagram of the two-stage evaporator system.

2.2. Instrumentation, calibration and procedure

The basic instrumentation required in such a refrigerating facility for evaluating its performance must include measurements of flow rate, temperature, pressure and the electrical power input. To equip the test-facility with sensors for measurements, the practical installation of several sensors with desired measuring accuracy of less than ±5% was considered. Sensors were selected to relate the state conditions of the air and refrigerant at various localities. The air-cycle monitoring system comprised of 20 channels of dry-bulb and wet-bulb type 'T' thermocouples, four channels of relative humidity sensors and two channels of anemometer air flow sensors. The temperature sensors were located before and after each evaporator, and before and after the drying chamber and economiser. Refrigerant temperature and pressure measurements were made through class A-type RTDs (Resistive Temperature Detectors) and pressure transducers. They were placed at various tapped-out points on the heat pump cycle. Variable magnetic flowmeters, measuring up to an accuracy of ±0.5 l/h, were installed before evaporators to indicate the refrigerant flow rate into both evaporators. The power input to the compressor was monitored using a power clamp-on meter which is capable of measuring three-phase power up to 600 kW.

Two different data-logging systems were employed to record data from the entire system. An 8-bit data-logger with 20 channels was used to record data from the type 'T' thermocouples and relative humidity sensors. A 12-bit data logger was used to record all signals from RTDs and pressure transducers. All the required readings were logged at 20 s intervals. Readings were considered for system performance analysis when they have been maintained steady for at least 120 min. Each experimental run would take up to 6–8 h of data collection.

Table 1
Description of key system components

Equipment	Description	Type and dimension
Evaporator (low pressure)	Fin type	Aluminium ripple
	Fin pitch	3.2 mm
	Face area	$0.145\,m^2$
	Number of tube rows	10
Evaporator (high pressure)	Fin type	Aluminium ripple
	Fin pitch	3.2 mm
	Face area	$0.145\,m^2$
	Number of tube rows	8
Condenser	Fin type	Aluminium ripple
	Fin pitch	3.2 mm
	Face area	$0.145\,m^2$
	Number of tube rows	8
Subcooler 1 & 2	Fin type	Aluminium ripple
	Fin pitch	3.2 mm
	Face area	$0.145\,m^2$
	Number of tube rows	2
Economiser	Fin type	Aluminium ripple
	Fin pitch	3.2 mm
	Face area	$0.145\,m^2$
	Number of tube rows	2
Expansion valve	Danfoss TE2X2-2.3	
Back-pressure regulating valve	Danfoss KVP-15	
Three way modulating control valve for refrigerant	Staefa M3FB25LX	
Compressor	Hitachi Scroll 500DH	
Refrigerant	R22 (chlorodifluoromethane)	
Dry air mass flow rate across evaporators	0.15–0.45 kg/s	
Dry air mass flow rate across condenser and subcoolers	0.3–0.45 kg/s	

All type 'T' thermocouples and RTD sensors were calibrated before the experiment. The probes were immersed in the water bath to reduce the fin effect. The exposed surface of the probes was insulated to minimise heat leak to the surrounding. Relative humidity sensors were calibrated in the factory with an uncertainty of ±2.0%. They were re-calibrated in the laboratory with standard chemical salts of known water activities. The maximum calibration uncertainty obtained was ±2.3% for chemical salts with known water activities of 0.224, 0.65, 0.753, 0.88 and 0.973. The calibration for air-vane anemometer was obtained from the factory calibration chart with an uncertainty of ±1.0%. The voltage-signal pressure transducers were calibrated by the manufacturer with the specification of maximum uncertainty at ±0.5% at full scale linear. Table 2 summarises the estimated uncertainty for different measurements.

Table 2
List of measurement accuracies for the variables measured in the heat pump drying test facility

Measured property	Measuring sensor	Uncertainty	Range
Refrigerant flow rate	Magnetic variable-area flowmeter	±1.8%	35 to 350 l/h
Inlet and outlet refrigerant temperature	Class 'A' Resistive Temperature Detector	±2.1%	−10 to 90 °C
Inlet and outlet refrigerant pressure	Pressure transducer gauge	±0.5%	50 to 500 psi
Inlet and outlet air temperatures	Type 'T' thermocouple	±2.4%	2.5–90 °C
Inlet and outlet air humidity	Relative humidity sensor	±2.3%	22.4–97.3%
Air velocity	Rotating anemometer	±1.0%	0.5–30.0 m/s
Heat flux	Heat flux surface sensor	±3.0%	1.0–3.0 kW/m^2
Compressor power input	Power clamp-on meter	±0.5%	up to 600 kW

Artificial drying load was introduced to the air-cycle by placing trays of water in the drying chamber. The desired heat pump configuration was selected by activating or de-activating the various solenoid valves. The drying conditions in terms of temperature, humidity and air flow were selected by entering the parameters into the various PID controllers. The weight of the water in the trays was monitored with electronic load-cell balance. Each experimental run took approximately 15 min to reach steady-state condition. Data from both air and refrigerant sides were logged into separate data-loggers. These data were then downloaded for processing after 2 h of heat pump dehydration operation. For greater details of the control mechanism for the two-stage heat pump dryer, the reader is referred to Chou et al. [9].

2.3. Repeatability tests and data quality

To determine the stability of the test facility and examine the degree of fluctuation of the experimental data, four experiments of similar drying air conditions (temperature, humidity and air flow) and heat pump cycle settings were conducted. On the air-side measurements, comprising the wet and dry bulb temperature measurements, the relative humidity and the flow velocity, less than 6.5% of data variation was presented. On the refrigerant-side measurements, comprising the local temperature, pressure and volumetric flow rate, less than 6.3% of data variation was obtained.

Besides having measurements that were repeatable, it is equally important to analyse the quality of the data. The quality of the data would reveal the accuracy of the sensors, sensitivity and consistency of the measurements for establishing the performance of the two-stage evaporator and subsequent model validation [10]. As part of the dryer commissioning process, energy balance was performed on both evaporators based on the air and refrigerant cycle. The enthalpy change of the air-side was determined by considering the temperature difference after each evaporator and the mass flow of the air. The energy absorbed at the two evaporators from the inlet two-phase condition to the exit superheat condition was calculated by measuring the inlet and exit refrigerant pressure, temperature and the refrigerant flow rate. Both energy values were compared to evaluate the adequacy of the selected instrumentation for the test facility. The heat transfer rate at the air-side was calculated by

$$Q_a = m_a \cdot C_p \cdot (T_{a,2} - T_{a,1}) \tag{1}$$

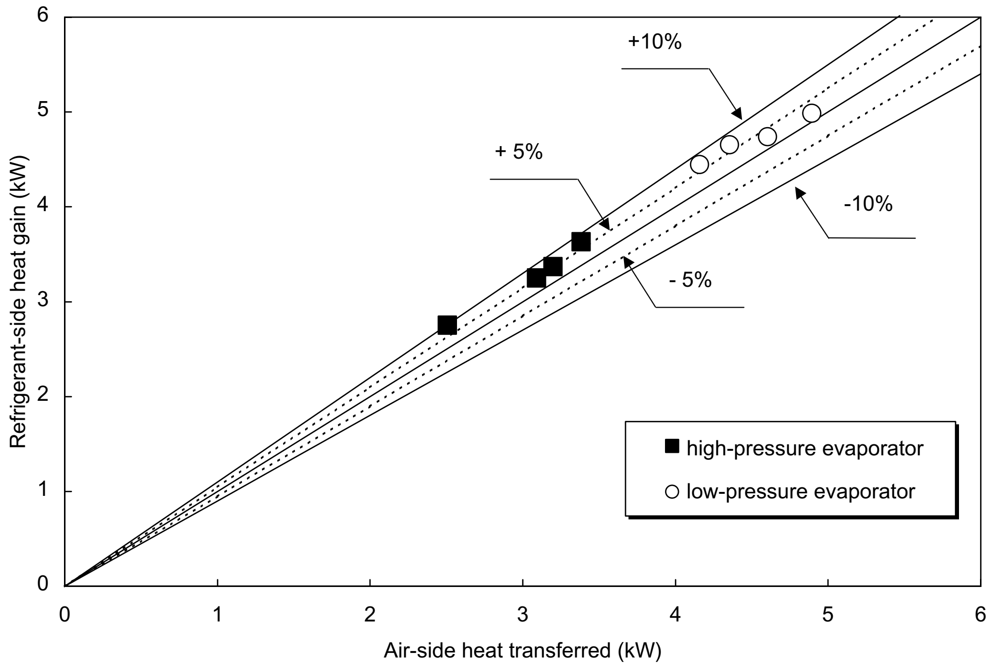


Fig. 4. Energy balance between air and refrigerant cycle.

and the enthalpy change of the refrigerant was calculated by

$$Q_e = m_r \cdot (h_2 - h_1) + m_r \cdot C_p \cdot (T_{s,2} - T_{s,1}) \tag{2}$$

Fig. 4 shows the results from four replicated experiments with similar operating conditions to evaluate the quality of the data. Eight measurements, four for each evaporator, were compared for enthalpy change at both air and refrigerant side. The maximum energy imbalance was found to be 8.9%. The heat transfer between the air duct, with 1.5 in thick of aero-flex insulation, and environment was measured with several heat flux sensors. Average measurements indicated the value to be approximately 28 W/m^2 for a temperature difference of 8 °C between air and ambient conditions. Taking into account the heat flux readings, the repeatability of the data was estimated to be 6.5%.

3. Evaluation of heat pump dryer performance

The overall performance of a heat pump drying system may be characterised by several criteria. Among them, the COP and SMER have been used by Jia et al. [11], Clements et al. [12] and Schmidt et al. [13]. Besides COP and SMER, the performance of the heat pump cycle may be examined in terms of the quantity of heat recovered at the evaporators. The heat recovered at each evaporator may be calculated from Eq. (2). The COP is defined as the ratio of the heat recovered at the condenser to the work required by the compressor;

$$\mathrm{COP}_{\mathrm{hp}} = \frac{\dot{Q}_{\mathrm{cd}}}{\dot{W}_{\mathrm{c}}} \tag{3}$$

where $\dot{Q}_{\mathrm{cd}}$ is the heat delivered to the condenser and subcoolers, and $\dot{W}_{\mathrm{c}}$ is the power input to the compressor.

The SMER can be defined as the energy required to remove 1 kg of water from the drying chamber and may be related to the power input to the compressor and may be calculated by

$$\mathrm{SMER}_{\mathrm{hp}} = \frac{\dot{m}_{\mathrm{w}}}{\dot{W}_{\mathrm{c}}} \tag{4}$$

where $\dot{m}_{\mathrm{w}}$ and $\dot{W}_{\mathrm{c}}$ are the water condensed at the evaporator per unit time and the power input to the compressor, respectively.

4. Results and discussion

4.1. Comparison between single and two-stage evaporator systems

According to Carrington et al. [14], the basic principal to improve heat recovery is to increase the proportion of latent to total heat load at the evaporators. For a given air-on (air entering the face of the heat exchanger) condition and fixed pressure level of the LPE, the mass flow of the refrigerant was either fully channelled to the LPE or part of the refrigerant was allowed to flow to the HPE by regulating the appropriate solenoid valve. Fig. 5 compares the amount of heat

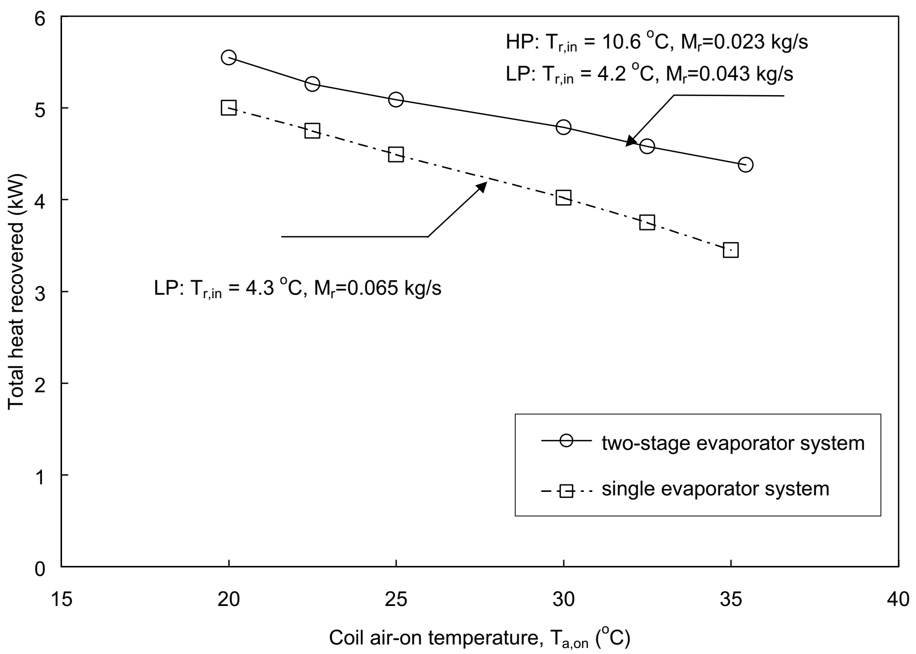


Fig. 5. Comparing heat recovered between single and two-stage evaporator system.

recovered from the system having a single evaporator and one consisting of two evaporators operating in tandem. It is clear that higher amount of heat, up to 35% more, can be recovered via the two-stage evaporator system. This improvement in heat recovery may be associated with the greater heat transfer area for thermal heat exchanging with the drying air. The high-pressure evaporator undertook part of the cooling duty to cool the air to dew-point temperature. In this way, the LPE could dedicate more of its surface for latent heat recovery. In a physical sense, the two-stage evaporator system may be viewed to have enlarged the mechanical boundaries for heat recovery in a heat pump cycle.

Having extended evaporators in a heat pump drying cycle is not the only available method to enhance heat recovery, other cycle arrangements or heat transfer augmentation methods that can capture more latent heat from the drying air would provide similar advantage in heat recovery improvement. At this juncture, it seems that multi-staging the evaporators improves heat recovery, an interesting question posed would be if there is a limitation to improving heat recovery through evaporator multi-staging. There could be the possibility of diminishing returns in heat recovery for greater number of evaporator staging. Firstly, for a given air-on face velocity, the third evaporator cooling capacity may not be fully utilised because of reduced heat transfer due to lower temperature difference between the evaporator's air-on and coil surface, as the air leaves the evaporator much higher than the refrigerant operating temperature. Secondly, as the temperature level of the low-pressure evaporator drops even lower to accommodate more multi-staging, the potential for air-side freezing exits. The build-up of frost lowers the capacity of the evaporator by adding an extra thermal resistance due to conduction through the frost layer and becomes a deterrent to improving heat recovery. Therefore, the number of stages of compression is often determined by economic as well as practical considerations [15]. Based on the readings obtained from the pressure gauges, pressure drops for HPE and LPE were in the range of 0.2–0.5 and 0.4–0.7 bar, respectively.

4.2. *Effect of high-pressure evaporator*

A major interest behind exploring the two-evaporator refrigerator system is that the two levels of evaporation would provide two different air streams with different temperature and humidity. Such a system would ensure the versatility of multiple drying chambers whereby two or more different products requiring different drying conditions can be dried with one heat pump system. Therefore, the ability of the two-stage evaporator to regulate the humidity of the drying air becomes crucial.

In this section, experiments were conducted to study the effect of regulating the two-phase refrigerant pressure level, and thus the surface temperature of the high-pressure evaporator on the dehumidification process and amount of heat recovered. This was accomplished by adjusting the spring knob of the back-pressure regulator to throttle the refrigerant flow to the high-pressure evaporator and thereby allowed the pressure of the inlet refrigerant to be changed. Fig. 6 shows that as the surface temperature of the evaporator dropped, the absolute humidity of the air gradually decreased. The reduction in absolute humidity was more significant at lower air-on temperatures. When the air-on temperature was low, a smaller portion of the evaporator was required to cool the air to dew-point temperature. This allowed a larger part of the evaporator to be dedicated to latent heat absorption. By regulating the pressure level of the HPE and thereby controlling its

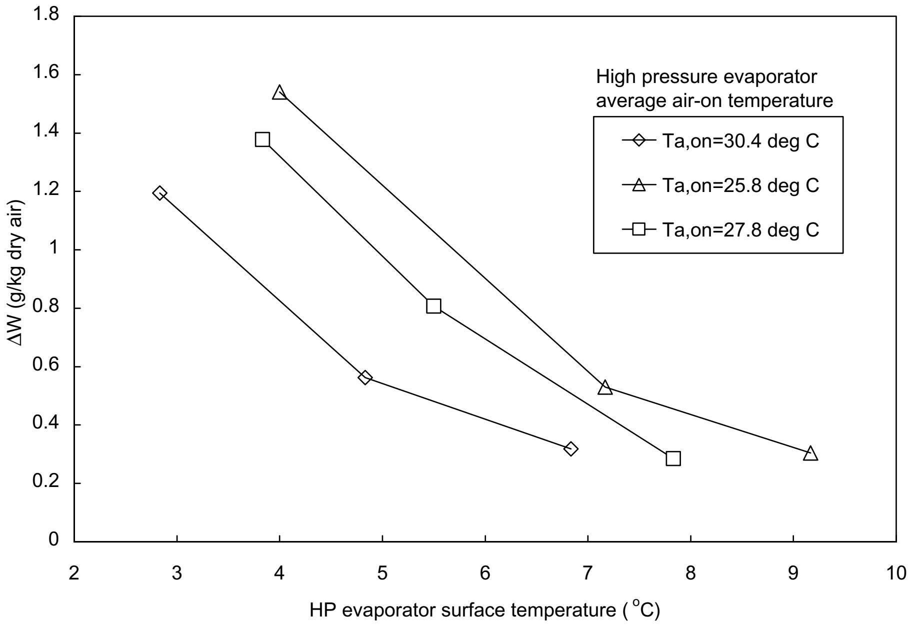


Fig. 6. Control of air humidity via regulating the pressure of high-pressure evaporator.

surface temperature, we were able to control the dehumidification process without condensate freezing occurring on the evaporator coil. The build-up of frost would lower the cooling potential of the evaporator by adding an extra thermal resistance due to conduction through the frost layer.

Fig. 7 shows the changes in latent heat absorption by the high-pressure evaporator as its surface temperature was regulated. This figure shows the general trend of increased heat recovered at the high-pressure evaporator and total heat recovered for the two-stage evaporator system as the temperature of the high-pressure evaporator was reduced. Up to 40% improvement in total heat recovered was possible. By regulating the surface temperature of the high-pressure evaporator, the amount of latent heat transfer between the air and the refrigerant could be adjusted. As the temperature of the refrigerant was lowered, the temperature gradient between the air and the refrigerant increased, resulting in the higher absorption of latent heat from the air once the air was sensibly cooled to below air-on dew-point temperature.

4.3. Effect of employing subcooler

Subcooling of the refrigerant at the exit of the condenser in a simple vapour-compression refrigeration system allows the refrigerant to enter the main cycle evaporator with low quality [16]. Thus, allowing the refrigerant to absorb more heat in the evaporator improves the COP of the refrigeration cycle. For the present system, the benefit of the subcoolers to enhance the dryer performance was tested by gradually introducing the first subcooler followed by the second subcooler to the heat pump cycle. The effect of using subcoolers to furnish additional sensible heat generally improves the SMER of a heat pump dehumidifier [14].

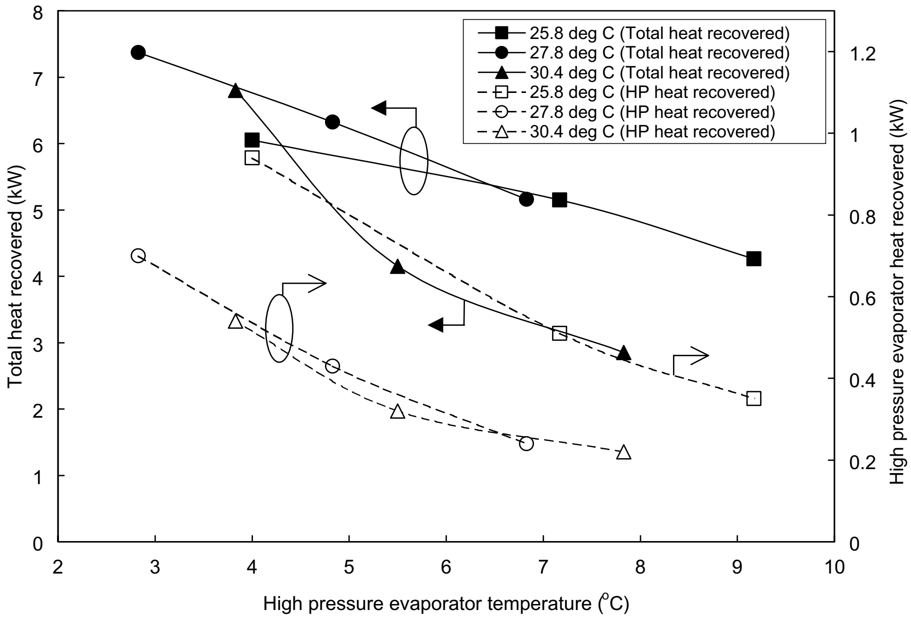


Fig. 7. Variation in heat recovered (HPE) and total heat recovered (two-stage evaporator) as the temperature of HPE changes.

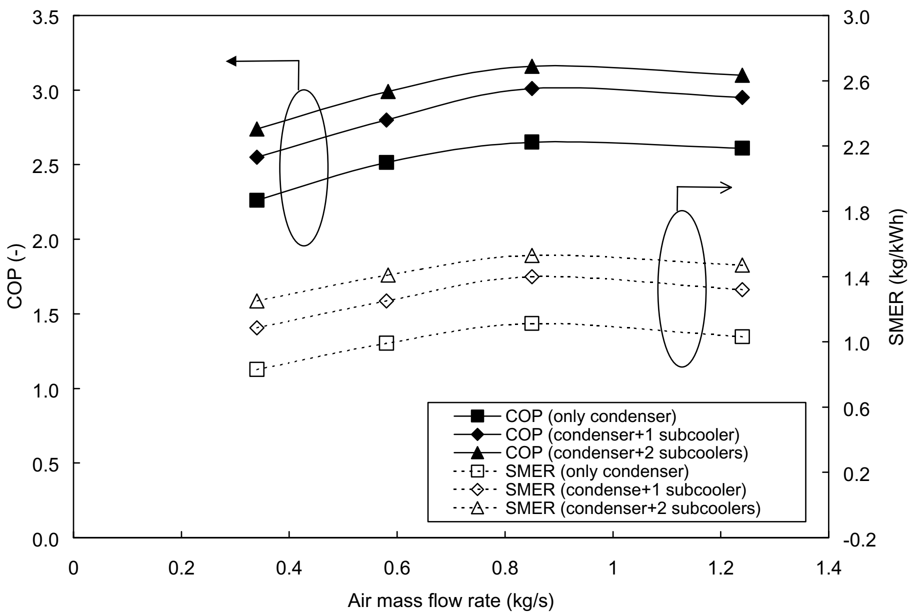


Fig. 8. Impact of subcoolers on the system performance in terms of COP and SMER.

Fig. 8 shows the system performance as the subcoolers were gradually activated. The addition of subcoolers improved the system performance in terms of COP and SMER. The introduction of the subcoolers not only allowed a lower quality refrigerant into the low-pressure evaporator but also provided additional sensible heating to the air without the incorporation of an auxiliary heater. The gradual activation of the subcoolers improved the COP and SMER in the range of 12–20% and 25–50%, respectively. These results were agreeable with those obtained from Carrington et al. [14] who showed that the use of subcoolers could improve the SMER by approximately 50% at 30% relative humidity at air dry bulb temperature of 50 °C. However, the additional advantage gained in terms of system performance by employing both subcoolers was less in comparison to the system when only the first subcooler was activated. This observation may be attributed to the finite rate of heat transfer in the heat exchangers as the air interacted with more heat exchangers [16].

4.4. Effect of employing economisers

In this present experimental heat pump set-up, an economiser was installed before and after the HPE and LPE coils. This economiser pre-cooled the air before it entered the evaporators and preheated the air before it entered the hot gas condenser. The use of the economiser to transfer heat from the air on one side of the evaporator to the air on the other side can increase the water extraction capacity of the heat pump dryer. Pre-cooling at the air-on side of the evaporators enabled the major part of the cooling coil capacity to be used to extract water from the air. The introduction of an economiser heat exchanger to a refrigerant cycle makes the process more efficient without altering its basic operation [17]. Fig. 9 shows the impact of employing the economiser to pre-cool and pre-heat the drying air before the drying air passed over the evaporators. The additional gains in the heat recovered for the same operating conditions were not large. The small

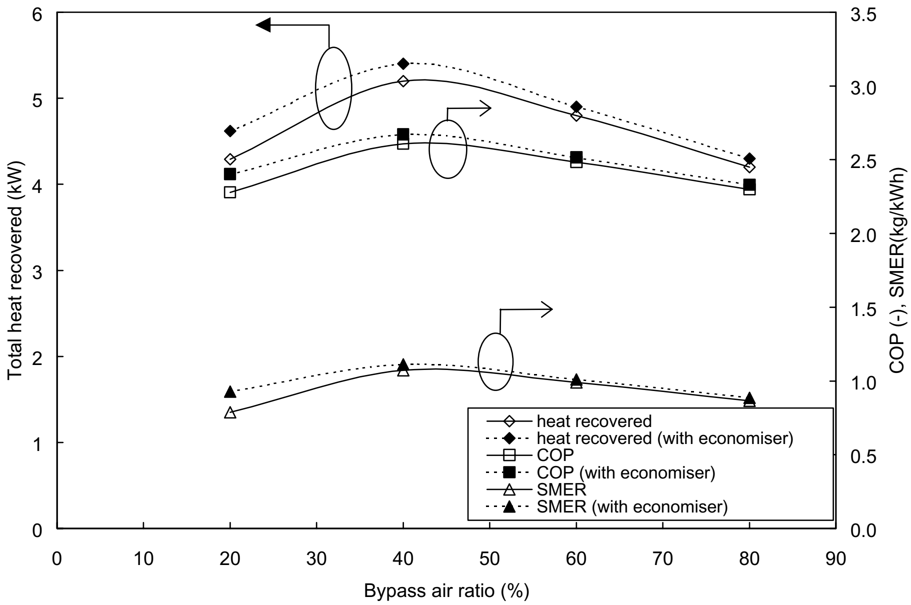


Fig. 9. Comparing system performance (COP and SMER) with and without economiser.

improvement in system performance could be due to its relative smaller total surface area which was about 1/4 that of the low-pressure evaporator. The correct choice of the economiser heat exchanger and its sizing is of vital importance in the design of an efficient and economic absorption cooler [17]. The required application will determine whether the economiser heat exchanger should be designed for extreme conditions or for typical operating conditions. For this present system, the proposed saving by pre-cooling the evaporator air-on and heating of the evaporator air-off before entering the condenser and subcoolers was not significant. The installation of the economiser resulted in the improvement in COP and SMER of about 3% and 4%, respectively. A more significant system improvement would be possible if a larger surface area economiser was employed. Having said that, it is noteworthy that economiser is a passive device that benefits the system performance without requiring additional input of electrical energy into the system.

4.5. Effect of air bypass over evaporators

For many convective air drying applications, it is necessary to regulate the humidity of the drying air to ensure premium quality dried products. One regularly method is to bypass the percentage of the drying air through the evaporator. The performance of a simple dehumidifier is sensitive to the evaporator air flow, an effect which can present itself as a controlling feature [18]. Since 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. Fig. 10 shows that as the bypass air ratio (BAR) to the evaporator was increased beyond a level, the total heat recovered at the evaporators reduced. For every 20% increase in bypass ratio over the 40% mark, the drop in heat recovered ranged between 0.6 and 0.8 kW. A smaller amount of heat recovered was

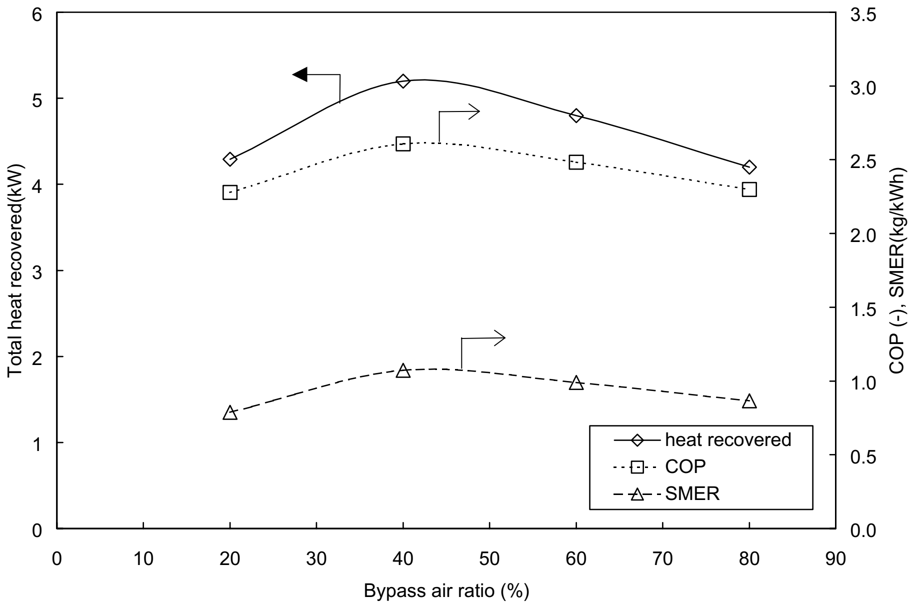


Fig. 10. Impact of regulating bypass air ratio on the total heat recovered at evaporators.

easily translated to lower COP and SMER. SMER could be affected by two factors. Firstly, higher compressor work was expected to provide for the required sensible heating at condenser. Secondly, a more humid drying air was translated to lower drying potential; resulting in less amount of moisture being absorbed. Therefore, based on the observed data, SMER was significantly affected by the BAR across the evaporators. The observations presented in Fig. 10 were consistent with those observed by Oktay [19] and Jia et al. [11]. Oktay [19] observed that when the BAR was increased from 20% to 40%, to 60% and to 80% the corresponding decrease in the heat pump performance indicators (COP/SMER) were 7%/29%, 20%/43%, 70%/52%, respectively. Based on the explanations provided by Oktay [19] and Jia et al. [11], higher BAR would reduce the SMER when the amount of by-pass air was increased. Both the heat transfer and heat transfer coefficient in the evaporator would then decrease as a result of lower flow rate and lower air velocity through the evaporator. Therefore, in employing the bypass air across the evaporator method to control the air humidity, the resultant penalty would be a drop in the heat pump dryer performance.

5. Conclusions

A prototype two-stage evaporator heat-pump-assisted mechanical drying system was designed, fabricated, and tested for enhancing heat recovery. The present test facility has confirmed the importance of improving heat recovery to improve the performance of heat-pump-assisted drying systems. Adopting the modular approach of increasing the complexity of the heat pump cycle, the following key conclusions may be drawn from the present experimental study of a two-stage heat pump dryer with HCFC22 as the refrigerant:

1. Up to 35% more heat can be recovered via a two-stage evaporator heat pump drying cycle in comparison to one having only a single evaporator.
2. Regulating the pressure of the high-pressure evaporator was found to be an effective method to regulate the humidity of the drying air while enhancing the amount of heat recovered from the drying air.
3. The addition of subcooler improved the system performance in terms of COP and SMER in the range of 12–20% and 25–50%, respectively. The introduction of the subcoolers not only allowed a lower quality refrigerant into the evaporators but also provided additional sensible heating to the air without the incorporation of an auxiliary heater.
4. Compared to other options, economiser added only marginal improvement to promoting heat recovery. This is because of its relative smaller surface area compared to the high- and low-pressure evaporators.
5. Increasing the bypass air ratio to the evaporators resulted in lower heat recovered. For every 20% increment in bypass ratio, a corresponding drop in heat recovered ranging between 0.6 and 0.8 kW was observed.

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