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Performance Evaluation of a Heat Pump System with Dual Condensers for Drying Herbs

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ABSTRACT

A dual condenser heat pump dryer (HPD) was utilised for herb drying, demonstrating its capability to provide a cost-effective drying solution via economical energy consumption, expedited drying processes, and improved system coefficient of performance (COP). The drying properties, exergy energy, and techno-economic feasibility were comprehensively evaluated. The HPD employed refrigerant R-32, operating within a temperature range of 6 °C for the evaporator and 47 °C for the condenser to dry kesum and pandan leaves. The experimental results demonstrate that the dryer required a modest amount of electricity, 0.012 and 0.143 kW/kg of fresh kesum and pandan, respectively. The dryer effectively removed moisture during its passage through the dual condensers and evaporator, yielding 16.85 kg and 3.39 kg of condensate water from the initial mass of 50 kg and 5.5 kg, respectively. The specific moisture extraction rate (SMER) for the HPD mode was calculated at 11.23 and 2.26 kg/kWh, and the average COP was 5.7 and 5.34 for kesum and pandan, respectively. The dual condenser improved the COP by approximately 11%. A maximum dryer productivity of about 0.75 kg/m²h is obtained at the air with a temperature of 47°C, velocity of 2.32 m/s and dryer surface load of 3.39 kg/m². The exergy efficiency of the thorough drying period varied from 56.85% to 96.33%. Moreover, the economic analysis determined a payback period of 3.24 months and 6.09 months for drying 24,000 kg and 2,640 kg of kesum and pandan leaves annually, with calculated annual incomes of 147,822.48 and 27,046.80 USD, respectively.

Keywords: Heat pump dryer; dual condenser; R32; COP; SMER; exergetic; techno-economic; herbs

INTRODUCTION

The drying process is an important factor of both product quality and economic feasibility in agricultural post-harvest systems (Gautam et al. 2023). To achieve high-quality dried products, an effective drying technique that minimises water activity levels is required, which suppresses fungal development and protects against uncontrolled thermal exposure. Concurrently, the cost of drying is controlled by a variety of parameters, including dryer layout and operational costs, which necessitate systematic evaluation and optimisation to ensure process efficiency (Tajudin et al. 2021). Passive and active solar drying technologies are known for their cost-effectiveness, as they are both quick and inexpensive. Sun drying is still prevalent in areas with continuously high ambient temperatures, such as Malaysia. However, the success of this procedure is dependent on elements like as the absorber's surface area, the amount of time it is exposed to sunlight, and the frequently suboptimal sanitary conditions that might emerge throughout the drying process (Cheng et al. 2022). This preference remains due to the unreasonably high costs of modern drying equipment such as freeze, vacuum, drum or steam dryers, which are frequently only available to large-scale commercial firms with substantial financial resources (M. Yahya et al. 2018). In typical drying applications, the process comprises simultaneous heat and mass transfer, in which liquid water in the product is vaporised, coupled with drying air, and mechanically removed. Forced drying uses air movement to provide the heat required for moisture evaporation. This heat, known as latent heat, is roughly 2560 kJ per kilogram of water, ensuring that moisture is effectively removed from the product (Zheng et al. 2022). The above-mentioned drying idea is used in the current study, with the drying method classified as air and contact drying at atmospheric pressure. It differs from energy-intensive methods such as vacuum drying and freeze drying, providing a more cost-effective alternative with lower energy requirements. Heat is given to the food during air and contact drying through hot air or direct contact with heated surfaces. This technique helps to evaporate moisture, and the generated water vapour is effectively eliminated from the drying air (Kumar et al. 2022).

To enhance drying effectiveness, current studies incorporate dual condenser configurations and inverter compressor technology, aiming to improve thermal control and energy efficiency during the drying process. The primary condenser transfers heat to the drying air, raising its temperature for effective moisture removal, whilst the secondary condenser recovers residual heat from the refrigerant, which can be used to preheat incoming air or improve refrigerant cycle stability. An experiment on a residential refrigerator compared the performance of a wire-on-tube condenser with single-loop flow to a dual-loop cycle flow condenser without fins. The data demonstrated that the adjustment to a dual-loop design resulted in turbulent flows, which improved heat dissipation within the tubes while boosting flow velocity (Ghanim 2016). This enhancement emphasises the potential benefits of improving condenser design for increased thermal efficiency (Ibrahim et al. 2011). Another study discovered that the temperature differential between the compressor outlet and the condenser outlet was lower in a dual-loop cycle flow condenser than in a wire-on-tube condenser with single-loop flow. This shows that the dual-loop design improves thermal performance by increasing heat dissipation and ensuring more consistent temperature levels throughout the system (Čurović et al. 2022). The experiment also indicated that, under all load situations, the temperature differential within the dual-loop condenser was consistently less than that of the conventional condenser. This result demonstrates the dual-loop design's improved efficiency in maintaining consistent thermal performance under various operational loads.

This study used a heat pump drier (HPD) as a forced drying device, utilising its inverter compressor to achieve much lower electricity usage than traditional dryers on the market. Furthermore, heat pump systems provide an energy-efficient solution by recovering heat from a variety of sources, allowing for broad application in residential, commercial, and industrial settings (Assadeg et al. 2020). Increasing the compressor's electrical frequency input from 34 to 66 Hz improves cooling capacity by 0.88 kW. However, this change reduces the coefficient of performance (COP) by 0.84 and the heat pump dryer's sensible heat ratio (SHR) by 0.11. These changes emphasise the trade-off between increasing cooling capacity and retaining energy efficiency and sensible heat recovery (Brandt et al. 2019; Muhammad Yahya et al. 2023). This discovery shows that the electrical demand falls in proportion to the cooling load, which is not typical of constant-speed compressors. (Rajamoorthy et al. 2018). As a result, the study concentrated on developing an experiment to evaluate the performance of a heat pump dryer (HPD) for drying herbs such as kesum (Persicaria odorata) and pandan (Pandanus amaryllifolius) leaves, with an emphasis on both performance and techno-economic evaluation. The dual-condenser heat pump drying system was examined, revealing that the heat pump's efficiency, as measured by its coefficient of performance (COP), varies depending on the indoor and outdoor environmental conditions (Colak et al. 2008). Furthermore, some uncertainty remains inherent in the overall measuring procedure, affecting the precision of the findings (Hu et al. 2022).

This study evaluated the efficacy of a low-cost drying machine designed to save energy, improve heat rejection at the condenser, and speed up drying times (Eng Sing et al. 2022). To accomplish this, a two-horsepower (HP) conventional air conditioner unit manufactured in Malaysia served as the foundation for a heat pump drier. A lab-scale prototype of a closed-system tray-feed drier was created to assess drying efficiency and conduct a techno-economic analysis. The results showed that the prototype may increase the drying capacity each loading cycle compared to commercially available systems while lowering the startup expenses by about 50%. The heat pump dryer used R-32 refrigerant and worked at temperatures ranging from 6 to 10°C for the evaporator and 44 to 47°C for the condenser. The technique featured warming the air as it travelled through the dual condensers, which improved the drying process. The design included connecting two condensers in series and integrating a copper piping network between the compressor, condensers, and evaporator to improve thermal performance (Sufiyan et al. 2020). The condensers were strategically positioned at the back of the drying chamber to ensure optimal heat exchange during the drying process. Meanwhile, the evaporator was placed in the front of the drying chamber to efficiently extract moisture from the items being dried. The moisture evaporated by the evaporator was then collected and measured to determine the system's performance.

METHODOLOGY

SPECIFICATION OF HEAT PUMP DRYER

The Heat Pump Dryer (HPD) is an innovative development that was built and tested at the Solar Thermal Research Laboratory of Universiti Kebangsaan Malaysia, located at 101.78°E longitude and 2.92°N latitude. The goal of this study was to create a drying machine designed specifically for herb drying, using an inverter-driven heat pump system with dual condensers. The system was designed primarily to run efficiently in tropical climates, taking into account the environmental difficulties and opportunities that such places present.

This study included the development, construction, and testing of a lab-scale prototype of a closed-system, tray-feed dryer. According to Yan, the closed-system design does not include auxiliary coolers or condensers and closely adheres to the principle of heat pump system engineering (Yan et al. 2023). Furthermore, the Heat Pump Dryer (HPD) is a fully closed drying system, with no external air intake or exhaust air discharge into the environment. This design maintains a high level of cleanliness throughout the drying process, making it ideal for luxury products such as saffron and reducing excessive heat loss from the system. When developing or analysing a dryer, it is critical to consider the fundamental concepts of heat, mass, and momentum transfer, as well as a thorough grasp of the material's quality characteristics (Mujumdar 2014). The heat pump dryer consists of several important components, including dual condensers, an evaporator, a compressor, an electronic expansion valve, and a drying chamber. To increase the condenser's capacity, an extra condenser unit with identical dimensions was added to the existing split-unit air conditioner system.

HERBS AND TOTAL HEAT CAPACITY

Kesum, pandan leaves, and aluminium have specific heat capacities of 3.38, 3.39, and 900 J/kg · K, respectively. Water has a vaporisation heat of 2387 kJ/kg at 48°C (steam table). The heat load was calculated using the original moisture content and the drying temperature. Equations 1 and 2 show the fundamental formulas used in this calculation.

$$Q = \dot{m} * C_p * \Delta T \quad \text{or} \tag{1}$$

$$Q = \dot{m} E_o \tag{2}$$

Where Q is the heat load (kW), E_0 is the water evaporation rate, $\dot{m}$ is the herb mass flow rate (kg/s), Cp is the herb's specific heat capacity (kJ/kg K), and Δt is the temperature change. If a 20% safety factor is utilised in this study, an air conditioner with a capacity of 18,000 Btuh or 5.28 kW is used as a heat pump dryer to operate the system and provide heat for drying herbs, as shown in Table 1.

Nomenclature			
AC	Alternating current	P	Work (W)
Am	Ante Meridiem	Pm	Post meridiem
Avg	Average	P-h	Pressure-enthalpy
COP	Coefficient of performance	Psia	Pound per square inch absolute
C_p	Specific heat (kJ/kg.K)	Psig	Pound square inch gage
Db	Dry bulb (ºC)	Q	Heat energy (W)
E	Energy (W)	Q_s	Sensible energy (kW)
Ex	Exergy rate (kW)	Q_a	Heat of heating air (J)
FOB	Free-on-board	q_v	Volume flow rate (m^3/s)
H	Enthalpy (kJ/kg)	RH	Relative humidity
h_{fg}	Specific enthalpy (kJ/kg)	USD	US dollar
HP	Heat pump	R-32	Difluoromethylene (CH_2F_2)
HPD	Heat pump dryer	SMER	Specific moisture extraction rate
LCCA	Life cycle cost analysis	T	Temperature (ºC)
K	Kelvin	t	Time (s)
M	Mass (kg)	TC	Thermocouple sensor
m_g	Mass flow rate of air (kgs^{-1})	T_1	Temperature of inlet chamber (ºC)
DR	Drying rate ($g_{water}/g_{wet\ matter*}h$) Air	T_{out}	Temperature of outlet chamber (ºC)
a		T_0	Ambient Temperature (ºC)
MER	Moisture extraction rate	ω	Moisture content (g/g)
η	Efficiency	Wb	Wet bulb (ºC)
γ	Thermal conductivity (Wm^{-1} ºK^{-1})	σ	Thickness (m)
ρ	Density (kg/m^3)	SHR	Sensible heat ratio

TABLE 1. Specification of heat pump dryer

Item	Specification
Outdoor model	RKG50CV1D8
Indoor model	3SYL18C3-AMDOK-061
Rated power supply/current	220-240 /~/50 / 6.22A
Refrigerant	R32
Compressor (horsepower)	2
Heating/ Cooling capacity	18000Btu/h(5.28kW)
Cabinet size (LWH) mm	2020 X 1805 X 1860
Chamber size (LWH) mm	1500 X 1010 X 1010
Tray size (LWH) mm	1495 X 997 X 38
No. of trays	10
Gap between tray (mm)	95

EXPERIMENTAL SETUP

Figure 1 illustrates data on air temperature, relative humidity, and velocity collected at five different locations. The Heat Pump Dryer's (HPD) drying process begins with the compressor pumping saturated R-32 refrigerant from a low pressure of 130 psig and a temperature of 6°C to a high pressure of 380 psig and 47°C. The refrigerant enters the condensers at high pressure and temperature, making the heated air hotter and less humid by the time it reaches Point 5. As the herbs' temperature rises, their surface moisture evaporates, raising air humidity and decreasing temperature at Point 1. At Point 2, a portion of the air goes through the evaporator, which reduces both the temperature and humidity by removing moisture. Simultaneously, the air balance bypasses the condenser, moving from Point 1 to Point 3. The air from Points 2 and 3 is combined and routed through Condenser 2, then Condenser 1. During this procedure, the air is reheated before being recirculated into the drying chamber, allowing for the next cycle of drying. This continuous action enables efficient moisture removal and consistent drying conditions.

Figure 2 shows the refrigeration design of the Heat Pump Dryer (HPD), which uses R-32 refrigerant at temperatures of 6 ºC for the evaporator and 47 ºC for the condenser. After leaving the condenser as a subcooled liquid, the refrigerant goes through an electronic expansion valve, where its pressure is reduced. The resulting cold liquid refrigerant circulates back into the evaporator, where it absorbs heat and moisture from the surrounding air. While the heat is regenerated inside the system, moisture is eliminated. The HPD system comprises essential components, including an inverter compressor, dual condensers with two fans, a fan coil, an electronic expansion valve, and an evaporator. At its heart, the process starts with an inverter-driven compressor that compresses the R-32 refrigerant gas from low pressure (130.0 psig) and temperature (6°C) to high pressure (380.0 psig) and temperature (47°C). The temperature rises as a result of heat absorbed from fresh items as well as heat created by the compressor. After compression, the R-32 refrigerant releases heat successively through the two condensers. This rejected heat is used for space heating within the dryer by the evaporator fan unit. The refrigerant completes the cycle by expanding in the expansion valve and returning to the evaporator at 6ºC. The warm air from the condensers is constantly circulated into the drying chamber by two axial fans and one evaporator fan unit. At the evaporator, the refrigerant absorbs heat and condenses moisture from the air, ensuring that moisture is removed from the system efficiently while enabling the next dry cycle.

To assess the heat pump cycle's performance, refrigerant temperatures were measured at important points: the compressor's suction and discharge, as well as the condenser's outlet pipe (Points 1, 2, and 3 in Figure 2). The refrigerant pressures at the suction and discharge sides were measured with a pressure gauge. Type-K thermocouples were used to provide accurate temperature measurements and ensure dependable data collection. The energy usage of the heat pump was tracked using a Fluke 1736 power logger, which provided insights into the system's operational efficiency. Additionally, sensors gathered and logged data on temperature and relative humidity, providing a comprehensive view of the drying environment.

Figure 3 shows the setup, which includes a closed dryer cabinet that houses critical components such as an air conditioner's interior evaporator and outdoor condenser units, a drying chamber, and ten trays to contain the items being dried. Meanwhile, Figure 4 depicts the dual condenser installation. These condensers are connected in series by copper pipes with a diameter of 12.7 mm, ensuring efficient heat transfer. Furthermore, service valves are included to improve operating flexibility and maintenance.

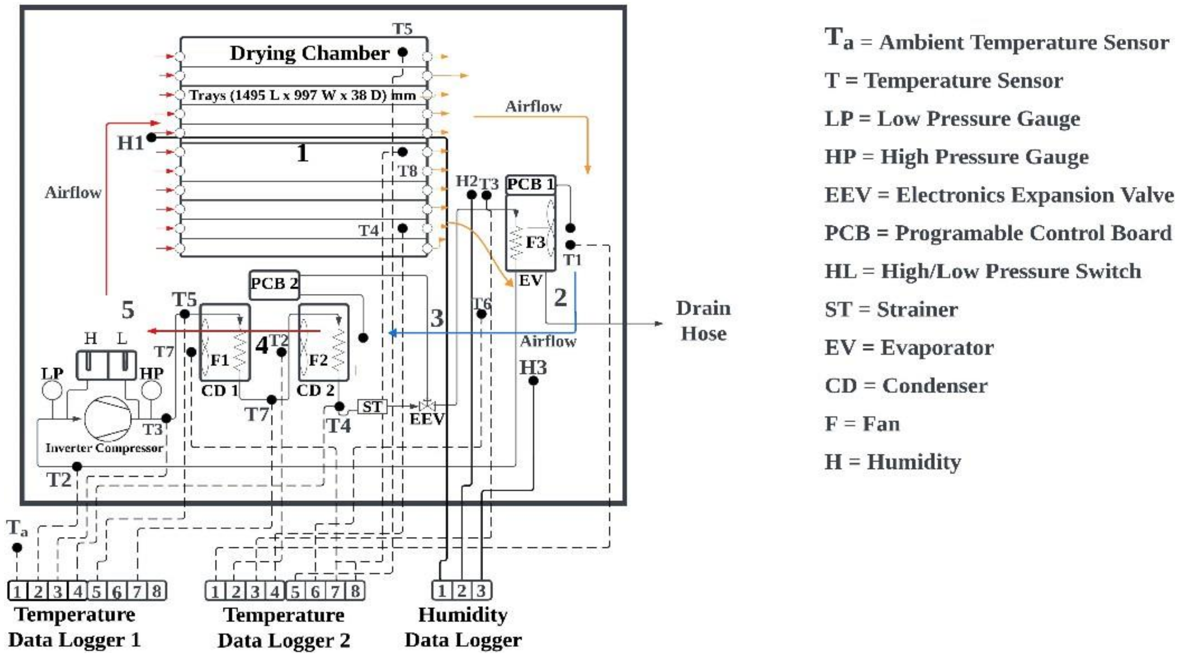

FIGURE 1. Heat Pump Dryer (HPD) system

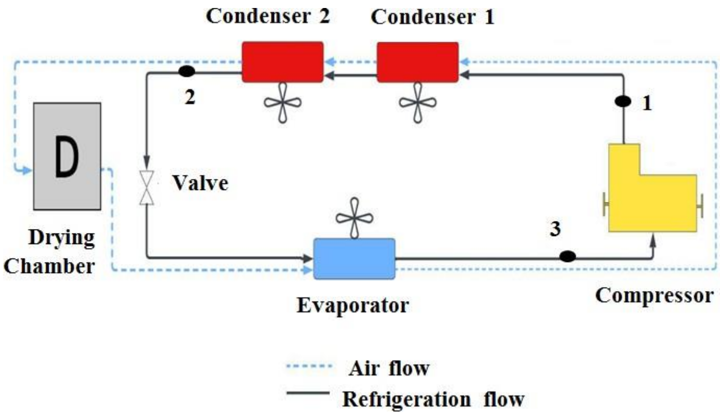

FIGURE 2. The refrigeration cycle of the heat pump dryer for a dual condenser system

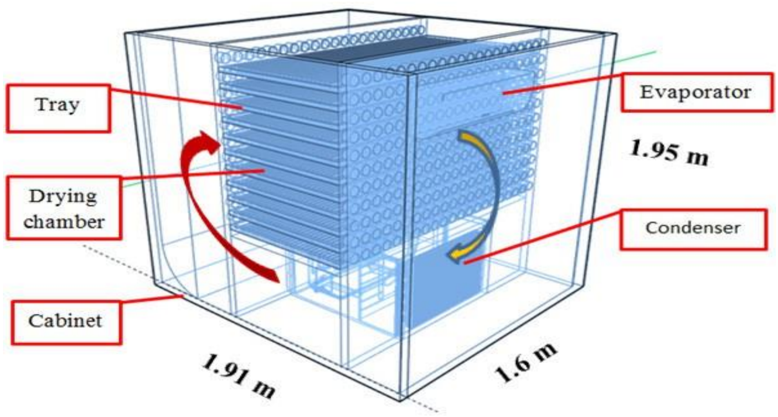

FIGURE 3. The arrangement of the HPD system.

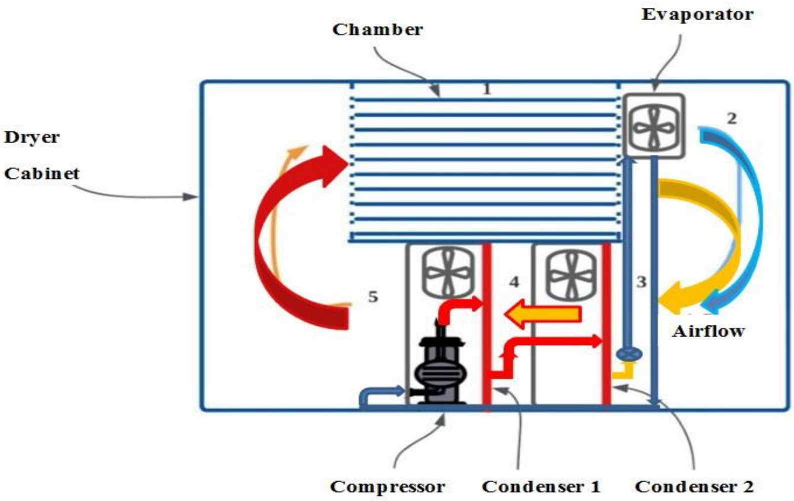

FIGURE 4. Dual condenser installation

EXPERIMENTAL PROCEDURES

This study included three different experimental setups, but the materials and techniques remained consistent. The first experiment employed a single condenser configuration, while the second investigated the efficacy of a dual condenser arrangement in drying 3.5 kg and 5.5 kg of fresh pandan leaves, respectively. The pandan leaves were evenly distributed among 10 trays, with each tray containing between 0.35 kg and 0.55 kg for each experiment. These modifications enabled a comparative comparison of the two systems under identical operating conditions to determine their effectiveness. The next experiment using a dual condenser setup involves drying 50 kg of fresh kesum leaves. Kesum leaves were uniformly spread across ten trays, each containing roughly 5 kg. Before beginning the drying process, thermocouple sensors and a hygrometer were used to determine the initial temperature and relative humidity around the leaves, providing the baseline environmental conditions for the experiment. Use Equation 3 to calculate the material's dry base moisture (Qiu et al. 2016):

$$M = (m - md)/md \tag{3}$$

where m and md stand for the material's mass (kg) and the corresponding dry material's mass (kg), respectively. The material's specific heat was determined using Eq. 4 (Huang et al. 2013).

$$cp = 837 + 3348 * (\times/((\times + 1))) \tag{4}$$

Where × is the moisture content of the dry base at any given time (kg water $kgDM^{-1}$), and Cp is the specific heat (J kg^{-1} K^{-1}).

Herb leaves were dried in a chamber using hot air at 45-55 °C. Temperature data were collected using sensors to monitor the drying conditions over a period of three to four hours. Furthermore, a Fluke TIR 32 thermal imager was used to capture the heat profile and temperature distribution, allowing for a thorough investigation of the drying system's performance. The refrigeration system and air conditioning data were routinely watched and recorded at 15-minute intervals, providing information about drying time, material quality at specific humidity levels, and overall energy usage. The energy consumption was precisely measured with a Fluke 1736 power logger. This heating system demonstrated continuous operation by recycling heat from the condenser, which improved energy efficiency and allowed for an unbroken drying process. At the conclusion of the experiment, the herb leaves were reweighed to assess the total moisture loss during the drying process. In addition, the water extracted by the evaporator was collected and properly weighed on a Kern FCE 3K1N digital scale. This stage gave a quantitative assessment of the system's drying efficiency and moisture removal capacity.

ERROR ANALYSIS

The study used a variety of measurement instruments and software to enable perfect data gathering and analysis. Temperature sensors were calibrated to ensure accuracy. Furthermore, the uncertainty associated with these tools, as well as the software's correctness, were calculated systematically. These processes are critical for reducing errors and maintaining the trustworthiness of experimental data, ensuring that the findings are strong and scientifically sound.

ERROR FOR MEASURING TOOLS

Table 2 presents the calibration findings for the thermocouple sensors utilised in this study. The sensors were calibrated before the experiment to ensure accuracy. The reference standard was a clinic thermometer, which revealed a 0.5 Kelvin variation in water readings from thermocouple sensors. The thermocouple sensors showed a variance of about ±0.3 Kelvin. The uncertainty was within acceptable bounds, falling below the normal sensor uncertainty of ± 0.5 Kelvin. This meticulous calibration emphasises the accuracy of the experimental measurements. The manufacturer specifies accuracy values for the following instruments: hygrometer (±3% RH), pressure gauge (Refco) (±0.1%), anemometer (Amprobe TMD-56) (±0.05%), and power logger (Fluke 1736) (±0.25%). Using equation 5 (Cheung, 2018), the computed uncertainty for this experiment is 16.50%.

$$\frac{\Delta \mathbf{COP}}{\mathbf{COP}} = \frac{\Delta \dot{\mathrm{m}}}{\dot{\mathrm{m}}} + \frac{\Delta \mathbf{h}}{\mathbf{h}} + \frac{\Delta \mathbf{E}}{\mathbf{E}} \quad (5)$$

where COP is coefficient of performance, ṁ is air mass flow rate, h is enthalpy and E is electrical energy.

ERROR OF SOFTWARE

Solkane and Genetron software were used to calculate the coefficient of performance (COP) for the Heat Pump Dryer's (HPD) refrigeration system. A comparison investigation of the two software products was performed to ensure their compatibility and accuracy. The COP calculations revealed that the average difference between the two software outputs was 0.23, representing an estimated 5% variation. This shows that both software tools produce findings that are closely aligned, ensuring the application's trustworthiness.

DATA ANALYSIS

The pressure-enthalpy (P-h) diagram can be created either manually on chart paper or digitally by entering experimental data into specialised software to assess the operation of the heat pump dryer (HPD) refrigeration system. Solkane and Genetron software are computer tools for creating pressure-enthalpy charts. Experimental parameters such as evaporating temperature, condensing temperature, suction superheat, and subcooling were entered into these software tools (Kalua 2017), which then calculated and displayed key outputs, including the coefficient of performance (COP) and other refrigeration system parameters, in tabular format. When the refrigeration system's experimental results were displayed on a pressure-enthalpy (P-h) chart, the enthalpy values (h), measured in kJ/kg at points 1, 2, and 4, were calculated. These enthalpy values were then used in the supplied formulae to compute the coefficient of performance (COP). The COP measures the system's efficiency in removing moisture (drying) from the leaves, with higher values indicating better efficiency. Furthermore, the heat pump (HP) unit's rated heat capacity (Q) of 5.27 kW (equal to two horsepower, 2 HP) was identified on the specification label. A Fluke 1736 power logger was used to measure the compressor's power consumption (P), which was stated in kW/hour. As a result, applying Eqs. 6 and 7 is seen as a more appropriate method for determining the system's efficiency.

$$COP = \frac{Q}{P} \quad (6)$$

$$\mathrm{COP} = \frac{(h1 - h4)}{(h2 - h1)} \quad (7)$$

where;

Q = useful heat supplied or removed by the considered system (W), P = work (electricity) required by the considered system (kW/h), h = enthalpy (kJ/kg).

A psychrometric chart (Vadoudi & Marinhas 2018) is used to assess the energy consumption of a drying system. The redesigned air heating unit utilises the respiratory heat released by fresh materials via air circulation within the drying system, allowing the materials to reach the necessary level of dryness. According to the psychrometric chart shown in Figure 5, the ambient dry bulb (dB) and wet bulb (wB) temperatures at Point 1 were 28.5°C and 27°C, respectively. A portion of the air at Point 1 was cooled as it travelled through the evaporator, bringing it to Point 2. The cooled air was then combined at Point 3 with another fraction of air coming from Point 1. The resulting mixed air was heated progressively at points 4 and 5. At Point 5, the design settings for temperature and relative humidity (RH) were set to 50°C and 25%, respectively. The sensible heating or cooling coil, as well as the heat exchanger's thermal energy consumption, were determined using Equation 8 (*Eurovent REC, 2005*).

$$Qs = q_v.\rho.cp\ (Tout - Tin) \quad (8)$$

Where *cp* is the specific heat of air in kJ/kgK (1.00 kJ/kg·K), ρ is the air density in kg/m³ (1.2 kg/m³), Qs is the sensible energy consumption in kW, and qv is the air flow rate in m³/s. *Tin* is the temperature of air entering the evaporator or condenser coil in °C, and *Tout* is the temperature of air leaving the coil in °C. Using Equation 9, the moisture extraction rate (MER), also known as the water removal rate, was calculated.

$$MER = q_v\ \rho\ (\omega_1 - \omega_2) \quad (9)$$

where ω_1 and ω_2 denote the air's moisture content (gramme per gramme). The specific moisture extraction rate (SMER) was calculated using Eqs. 10 or 11, respectively.

$$SMER = \frac{\dot{m}(\omega out - \omega in)}{Wcomp + Wblower} \quad (10)$$

$$\text{Or} \quad SMER = \frac{mi - mf}{W} \quad (11)$$

where *W* is the system's power consumption (kWh), mi is the initial mass (kg), and mf is the final mass of the product (kg). Using Equations 12 and 13, the drying system's drying rate efficiency was calculated.

$$Efficiency = \frac{(Mass\ of\ water\ removed\ from\ leaves\ \times\ vaporization\ of\ water)}{Energy\ consumption\ by\ compressor} \quad (12)$$

If the efficiency is ~ 1, then Ƞ = 1

$$E_{\text{compressor}} = m\ (removed) \times hfg \quad (13)$$

where *hfg* is the specific enthalpy (in kJ/kg) of water evaporation (latent heat). The latent heat of water vapourisation is equal to the mass of water recovered from the leaves minus the compressor's energy consumption. To determine the entire electrical power of the HPD, use a power logger, also known as Eq. 14.

$$\text{E (compressor)} = Current\ (I) \times Voltage\ (V) \times time\ (t) \quad (14)$$

In this study, the energy cost was determined in US dollars (USD) using an exchange rate of 4.48 RM to USD. The commercial electricity rate in Malaysia for 2025 has been recorded as USD 0.094 per kWh for the first 200 kWh of use and USD 0.110 per kWh for usage above 200 kWh.

PRODUCTIVITY AND SPECIFIC ENERGY CONSUMPTION

The performance of a dryer can be quantitatively assessed based on its productivity (kg/m²h) and specific energy consumption (SEC) (kJ/kg H_2O). The definitions of productivity and SEC for heat pump dryers are established according to Eq. 15 and Eq. 16 (Fatouh et al. 2006) respectively.

$$\textbf{Productivity} = \mathbf{0.109C + 0.446V + 0.184T - 9.134} \quad (15)$$

$$\textbf{SEC} = \frac{\textbf{Pcom+Pb}}{\textbf{average moisture exteraction rate}} \quad (16)$$

ENERGY-EXERGY ANALYSIS

This study used mass and energy balance equations to assess the energy and exergy performance of a heat pump dryer (HPD) system used to dry kesum and pandan leaves. The system was considered to run under steady-state and steady-flow circumstances, with the following calculation procedure supplied in Eq. 17:

$$\sum m(in) = \sum m(out)h(out) \quad (17)$$

Eq. 18 quantitatively represents the energy balance inside the system's operational framework and can be used to reflect the overall system's energy conservation.

$$\sum Q + \sum m_{in}h_{in} = \sum W + \sum P + \sum m_{out}h_{out} \quad (18)$$

In this study, the total heat (QHPD) supplied to the drying system via heat rejection from the condenser, the heat (Qa) utilised for air heating, the useful energy (ExHPD) transferred to the air, the power consumption of the air blower (Pblower), the specific energy consumption (SEC), the specific thermal energy consumption (STEC), the exergy efficiency of the drying chamber (ηex), the improvement potential (IP), and the sustainability index (SI) of the drying system are determined using Eqs. (19)–(31), as tabulated in Table 3 (Zeng et al., 2022).

TABLE 3. Energy-exergy analysis of herbs drying system

Energy-Exergy Analysis	Equation	Eq. No.
Total heat applied from HP	$Q_{HP} = V_{ref} \times q_{ref}$	(19)
Energy for heating air	$Q_a = m_{g,a} c_{p,a}(T_1 - T_0)$	(20)
Heat loss of exhaust air outlet	$Q_{loss,air} = m_{g,a} c_{p,a}(T_{out} - T_0)$	(21)
Heat loss of the wall heat transfer	$Q_{loss,wall} = 1.3A_0/(\frac{\delta}{\gamma})\sum_{i=1}^{4}(T_{wall,i} - T_0)$	(22)
Exergy for heating air	$Ex_{HP} = m_g c_{p,a}[(T_1 - T_0) - T_0 \ln\left(\frac{T_1}{T_0}\right)]$	(23)
Exergy of air blower	$Ex_{blower} = W_{blower} = P_{blower} t$	(24)
Total exergy entering the drying system	$Ex_{dc,in} = Ex_{gas} + Ex_{blower}$	(25)
Total exergy outlet the drying system		
The specific energy consumption	$SEC = \frac{Ex_{gas} + Ex_{blower}}{DR}$	(26)
The specific thermal energy consumption	$STEC = \frac{Ex_{gas}}{DR}$	(27)
Exergy of drying chamber	$Ex_{dc} = Ex_{dc,in} - Ex_{dc,out}$	(28)
Exergy efficiency of drying chamber	$\eta_{ex} = \frac{Ex_{dc}}{Ex_{dc,in}}$	(29)
The improvement potential	$IP = (1 - \eta_{ex})Ex_{dc}$	(30)
The sustainability	$SI = 1/(1 - \eta_{ex})$	(31)

TECHNO ECONOMICS ANALYSIS

A drying system's installed cost includes charges for equipment acquisition, installation, piping, wiring, and instrumentation (Kudra & Sztabert 2006). The heat pump dryer (HPD) was economically evaluated utilising a life cycle analysis approach (Qiu et al. 2016).

LIFE-CYCLE METHOD

Eq. 32 calculates a dryer's life cycle savings, often known as its cumulative net present value.

$$Vcnp = \sum_{t=1}^{t} Vnpt \tag{32}$$

Vnpt, or the present value of a dryer over a time period (*t*) expressed in terms of a year, can be computed using Eq. 33.

$$Vnpt = \frac{(1+i)^{t-1}}{(1+n)^t} \times Sd \times D \tag{33}$$

where *D* is the number of dryer days used annually, *Sd* is the daily savings (USD/d) for a dryer in a time period (*t*) by year, and i is the rate of inflation and Sd is given by Eq. 34.

$$Sd = (Cs - Cds) \times mf \tag{34}$$

where *Cds* is the cost per kilogramme of dried herbs leaves for a dryer and *Cs* is the selling price of dried herbs leaves (USD/kg) and *Cds* is calculated by Eq. 35

$$Cds = Cfd + Cd \tag{35}$$

where Eqs. 36 and 37 are used to compute *Cfd*, or the cost of fresh herbs leaves per kilogramme of dried herbs leaves (USD/kg), and *Cd*, or the cost of drying per kilogramme of dried herbs leaves in drier (USD/kg).

$$Cfd = \frac{mf}{mi} Cf \tag{36}$$

where *Cf* is the price in USD/kg for one kilogramme of fresh herbs leaves.

$$Cd = \frac{Ca}{mf \times D} \tag{37}$$

The annual cost of a dryer, or *Ca*, can be computed using Equation 38.

$$Ca = Cac + Cm + Cr \tag{38}$$

where *Cm* is a dryer's annual maintenance cost (USD), *Cr* is a dryer's annual running cost (USD), and *Cac* is a dryer's annual capital cost (USD), which is determined by Eq. 39.

$$Cac = Ccc \times \frac{n(1+n)^j}{(1+n)^j - 1} \tag{39}$$

where *j* is the dryer's life (year) and *Ccc* is its capital cost (USD). The drying system's yearly operating costs in HPD mode are determined by Eq. 40.

$$Cr = \frac{mf - mi}{SMERhpd} \times Dhpd \times Ce \tag{40}$$

where *Ce* is the electric energy cost per kWh (USD/kWh).

PAYBACK PERIOD

The payback period (*N*) is provided by Eq. 41.

$$N = \frac{\ln\left(1 - \frac{Ccc}{V1pt}(n-1)\right)}{\ln\left(\frac{1+i}{1+n}\right)} \tag{41}$$

TABLE 2. Instrument characteristics

Sensor/Location	Reading 1	Reading 2	Error
TC1/ Suction	29.4 ºC	29.2 ºC	0.2K
TC2/ Liquid	29.2 ºC	28.9 ºC	0.3K
TC3/ Discharge	29.0 ºC	29.0 ºC	0.0K
Clinic Thermometer	29.5 ºC	29.5 ºC	0.0K

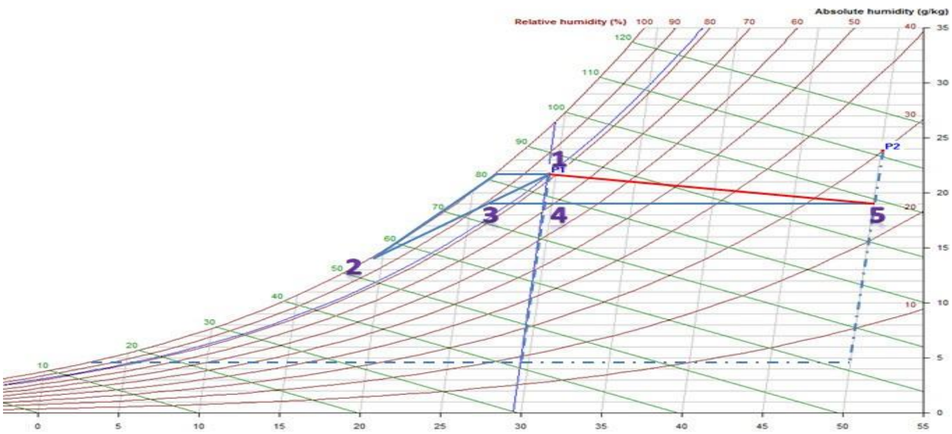

FIGURE 5. Energy Consumption for the Heat Pump Dryer on the Psychrometric Chart

RESULT AND DISCUSSION

HPD DATA AND ANALYSIS

Table 4 shows the data acquired from the heat pump dryer (HPD) system. The air temperature and relative humidity of the drying unit were meticulously recorded and grouped into tables and graphs to assist analysis in accordance with the study's objectives. Table 3 shows the recorded data from the HPD's refrigeration system. These measurements were then processed using Eqs. 42 and 43 (Rapids 2015) to calculate other parameters, such as superheat and subcooling, as shown in Table 5.

$$\begin{aligned} &Superheat = Evaporating\ temperature - \\ &suction\ line\ temperature \end{aligned} \tag{42}$$

$$\begin{aligned} &Subcooling = Condensing\ temperature - \\ &liquid\ line\ temperature \end{aligned} \tag{43}$$

Figure 6 depicts a pressure-enthalpy (P-h) curve that was used to examine the heat pump dryer's refrigeration system. The graphic was created using baseline data from the system for single and dual condenser configurations. This chart was used to determine the enthalpy values at points 1, 2, and 4. These data were then used to determine the coefficient of performance (COP) using Eqs. 4 or 5, as discussed in the previous section, or generated using dedicated software. Furthermore, air-cooled condensers are often built to operate at a temperature 2-5% higher than the maximum ambient temperature at their installation site (Nwasuka N. C et al. 2020). The heat pump dryer (HPD) had a temperature differential of at least 4.4% between the condenser air inlet and exit. As illustrated in Figure 7, the dual condenser setup produced higher amounts of superheat at the suction pipe while requiring less subcooling than the single condenser. An rise in the evaporator outlet temperature, along with a drop in the evaporation temperature, resulted in increased heat absorption by the system, boosting the coefficient of performance (COP) (SU et al. 2024).

COMPARISON OF PERFORMANCE BETWEEN SINGLE AND DUAL CONDENSER HEAT PUMP DYER TO DRY PANDAN LEAVES

Figure 8 depicts the experimental testing, which began at 17:15 and ended at 20:45 for both setups performed on different days. The data were plotted to show the relationship between temperature, relative humidity, and coefficient of performance (COP) as a function of HPD drying time for both the single (1 condenser) and dual condenser (2 condenser) configurations. The initial temperature and relative humidity surrounding the leaves were measured as 27.7°C and 85.8% RH for the single condenser setup, and 28.4°C and 95.8% RH for the twin condenser arrangement, respectively. Pandan leaves weighing between 3.5 and 5.5 kilogrammes were dried for four hours at temperatures ranging from 45°C to 55°C. The drying process used a mass flow rate of 0.135 kg/s and a volume flow rate of 0.113 m^3/s, with air velocities ranging from 2.72 to 2.85 m/s (Abusaibaa et al. 2022). After drying, the ultimate mass of the leaves was recorded as 1.12 kg and 2.11 kg, resulting in a computed moisture extraction rate (MER) or drying rate of 0.113 and 0.255 kg/h for the single and dual condenser setups, respectively. Moisture was removed as hot, dry air flowed through the evaporator, so accelerating the drying process. During testing, the coefficients of performance (COP) of the single and dual condenser setups were calculated, providing average COPs of 4.95 and 5.52. This COP 5.52 was found to be 32% higher than that of a gas engine-driven heat pump, as reported by (Pawela & Jaszczur, 2022), 57% higher than the heat pump dryer reported by (Salehi 2021) and 3.26% higher than a prior experiment by (Abdullah et al. 2025).

Based on actual testing with the present research dryer, the dual condenser arrangement improved the coefficient of performance (COP) by almost 11%, boosting it from 4.95 to 5.52 when compared to the single condenser system. The heat pump drier (HPD) used in this study consumed less electricity, consuming only 1.21 to 1.53 kWh. This system featured a high-efficiency inverter compressor and a Proportional-Integral-Derivative (PID) temperature controller. The cost of electricity to dry 3.5 and 5.5 kg of pandan leaves for four hours was calculated to be USD 0.63 and 0.65, respectively. Over the course of 20 days, with eight hours of operation per day, the system used 266.20 to 275.02 kWh of energy to generate 44.8 to 84.4 kilogrammes of dried pandan leaves. As a result, the monthly electricity cost was estimated to be between USD 5.82 and 6.04, or between USD 0.07 and 0.13 per kilogramme of dried pandan leaves.

Table 6 shows that the pandan drying time was four hours, with Specific Moisture Extraction Rates (SMER) of 0.36 and 2.26 kg/kWh in HPD mode of single and dual condensers, respectively. During the drying process, 2.38 kg and 3.39 kg of condensate water were collected from the evaporator during a 4-hour period, giving 1.12 kilogramme and 2.11 kg of dried pandan leaves, with total power consumption recorded at 6.66 and 6.88kWh,respectively.

TABLE 4. Reading from refrigeration system HPD (R-32)

Type of Parameter	Reading
Gage suction pressure (Low side)	130psig/8.97bar
Gage discharge pressure (High side)	380psig/26.21bar
Evaporating temperature	6 ºC
Condensing temperature	47 ºC
Discharge line temperature	96.7 ºC
Suction line temperature	25.7 ºC
Liquid line temperature (after condenser 1)	42.5 ºC
Liquid line temperature (after condenser 2)	41.1 ºC
Drying chamber temperature (wet/ dry bulb)	41.0/43.0 ºC
Ambient temperature	28.5 ºC
Discharge air temperature (condenser)	46.5 ºC

TABLE 5. Value of superheating and sub cooling

Calculated Result	Value
Suction Superheat = 25.7 ºC - 6 ºC	19.7K
Sub cooling 1 = 47 ºC – 42.5 ºC	4.5K
Sub cooling 2 = 47.0 ºC – 41.1 ºC	5.9K
Discharge superheat = 96.7 ºC - 47 ºC	49.7K

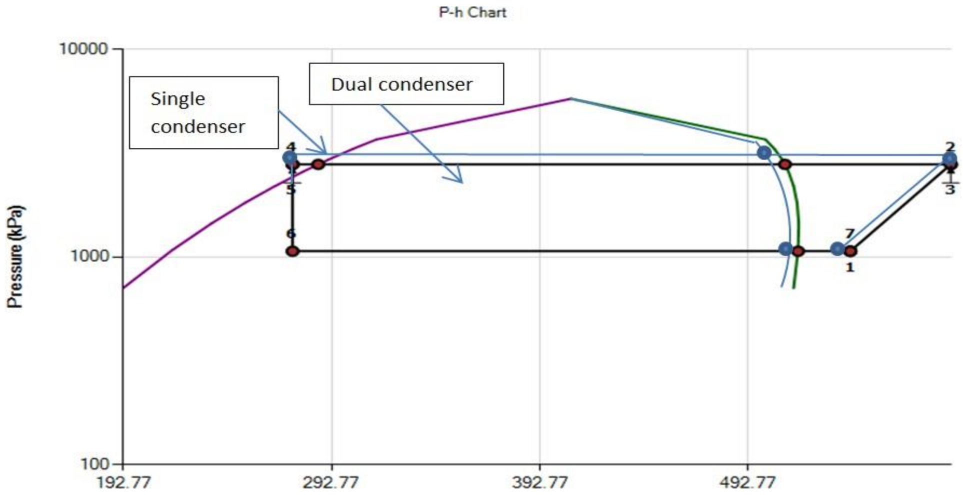

FIGURE 6. P-h chart for single and dual condenser of refrigeration system HPD

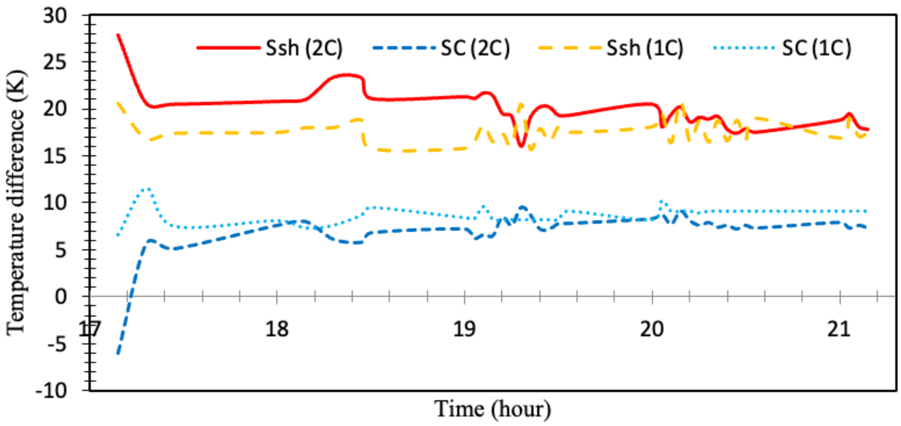

FIGURE 7. Suction super heat (Ssh) and sub cooling (SC) of a single condenser (1C) and dual condenser (2C)

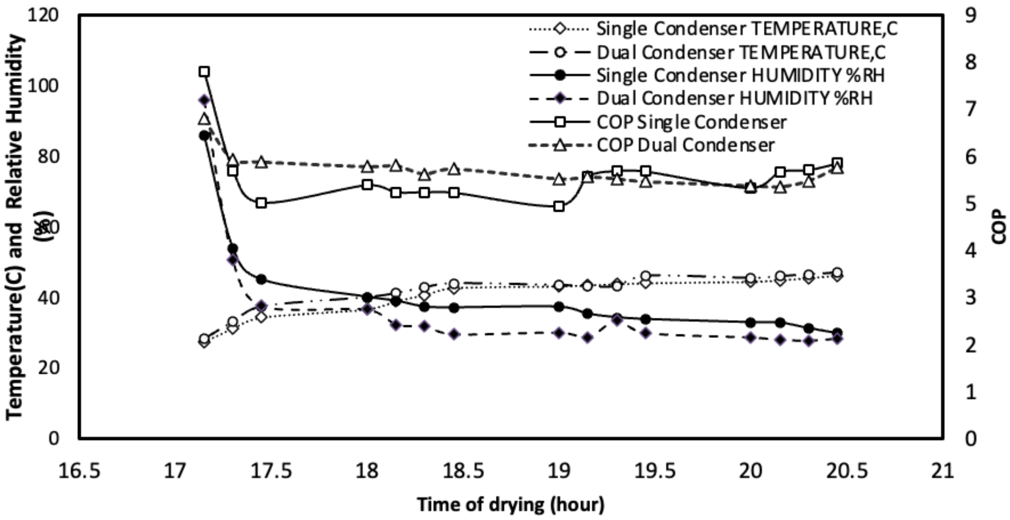

FIGURE 8. Temperature, Relative Humidity and COP Vs. Time of drying HPD

TABLE 6. Result of drying system of single condenser (HPD-1Condenser) and dual condenser (HPD-2Condenser)

Measurement	HPD-1Condenser	HPD-2Condenser
SMER (kg/kW h)	0.36	2.26
COP (average)	4.95	5.52
Time (h)	4.0	4.0
The efficiency of HPD machine (%)	80.74	78.12
The efficiency of drying rate (%)	25.65	35.37
Water removed per energy (kg/kW)	0.36	0.49
Initial and final moisture content (g water/g dry matter)	0.68,0.070	0.62,0.045

DUAL CONDENSER HEAT PUMP PERFORMANCE TO DRY PANDAN AND KESUM LEAVES

EFFECT OF SURFACE LOAD

The surface load for pandan and kesum leaves was 0.37 and 3.39 kg/m², respectively, with initial moisture levels of 61.87% and 76%. As seen in Figure 9, the drying time increased correspondingly with surface load. After 4 hours of drying, surface loads of 0.37 and 3.39 kg/m² resulted in moisture levels of 4.56% and 10.93%, respectively. Notably, both surface loads met the moisture content recommendations of 10% for kesum and 20% for pandan leaves.

DRYER PRODUCTIVITY

Surface load, air velocity, and drying temperature all have an impact on dryer productivity, which quantifies the dryer's dried mass per unit time and unit surface area. As shown in Figure 10, both kesum and pandan leaf drying experiments were productive, with moisture contents meeting the recommended levels. Dryer productivity for pandan and kesum leaves was 0.59 kg/m²h and 0.92 kg/m²h, with surface loads of 0.37 kg/m² and 3.39 kg/m², drying air velocity of 2.32 m/s, and drying temperature of 47°C, respectively.

SPECIFIC ENERGY CONSUMPTION

Specific energy consumption (SEC) is the amount of energy required to remove 1 kilogramme of water during the drying process. Figure 11 shows the SEC findings from successful drying experiments with kesum and pandan leaves under various surface load settings. Drying circumstances at 47°C, 2.32 m/s, and 3.39 kg/m² resulted in a minimum SEC of 1065.4 kJ/kg H_2O.

ENERGY-EXERGY ANALYSIS

In this investigation, the drying procedure used only electricity for energy consumption. The heat required to dry the herbs was supplied by the heat pump (HP) system as well as heat created by the herbs' breathing. During the drying process, the inverter compressor, evaporator fan, and two axial fans consumed 75% and 25% of the total energy consumption of 2 kW, respectively. As demonstrated in Table 7, the absence of exhaust air resulted in limited heat loss through the dryer walls, resulting in low overall energy usage. The wall heat loss varied between 0.15 and 0.45 kW throughout the drying process. Figure 12 depicts the similar pattern in exergy flow, which shows a rising trend in exergy input and outflow over time before stabilising near the end of the drying phase. Throughout the drying process, the exergy inflow, exergy outflow, and exergy within the drying chamber ranged from 0.5 to 1.08 kW, 0.02 to 0.47 kW, and 0.5 to 0.61 kW, respectively. Because the dryer is a closed system, the exergy output is recycled back into the drying process. Table 8 shows that as drying time rose, exergy efficiency (ηex) and sustainability index (SI) dropped, whereas improvement potential (IP) increased (Colak et al. 2008). During the drying process, exergy efficiency ranged from 56.85% to 96.33%.

ECONOMIC ANALYSIS

The current study determined that the HPD system may be used at any time and in any location, hence the yearly net present worth calculation assumes that the HPD is operational 240 days per year. Table 9 shows the cost and economic characteristics.

The price of any material has a significant impact on the economic analysis of drying systems. The products are maintained dry and serve as staple foods for many communities around the world; thus, the drying process is critical in the mass production of the product and its derivatives (United 2020). As a result, the dryer chosen for the drying process should have the lowest equipment, operating, and maintenance costs to assure a low selling price for the product and good profitability. The system requires 24000 kg and 2640 kg of dried kesum and pandan leaves every year, based on the amount used during testing. Table 10 displays the calculated yearly and cumulative net present values for the system life with kesum and pandan leaves.

Life cycle cost analysis (LCCA) for HPD equipment, energy, and maintenance. The energy cost was computed in US dollars. A home air-to-air heat pump (HP) has an expected service life of 15 years and an annual maintenance cost of 7% of the overall system cost (Kirk & Dell'Isola 1995). Drying kesum and pandan leaves had payback times of 3.24 and 6.09 months, according to the life-cycle technique. The cumulative net present values were 1,403,835.99 USD and 256,567.62 USD, respectively, while the calculated annual revenues were 147,822.48 USD and 27,046.80 USD for drying kesum and pandan leaves.

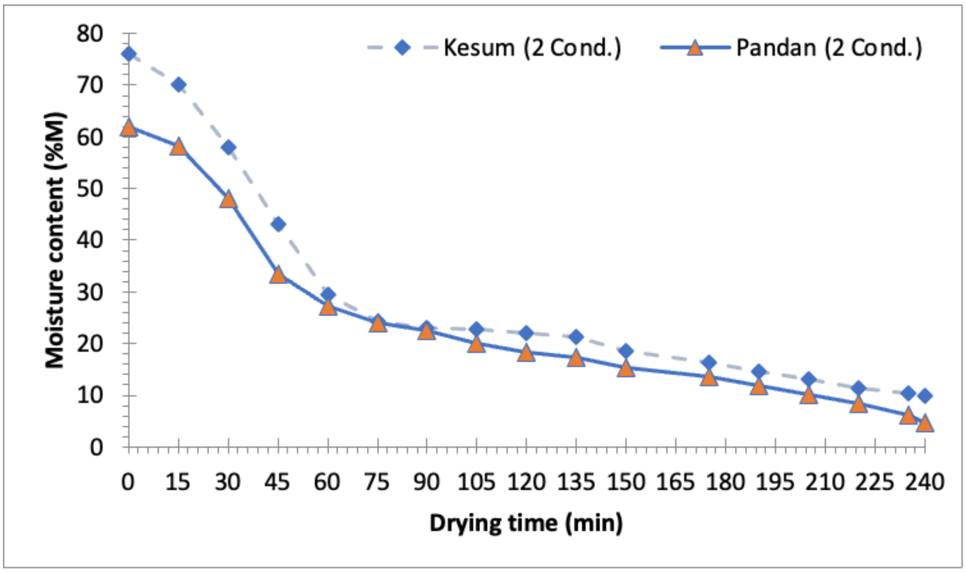

FIGURE 9. Effect of surface load of kesum, C=3.39 kg/m^2 and pandan C=0.37 kg/m^2 in the drying process

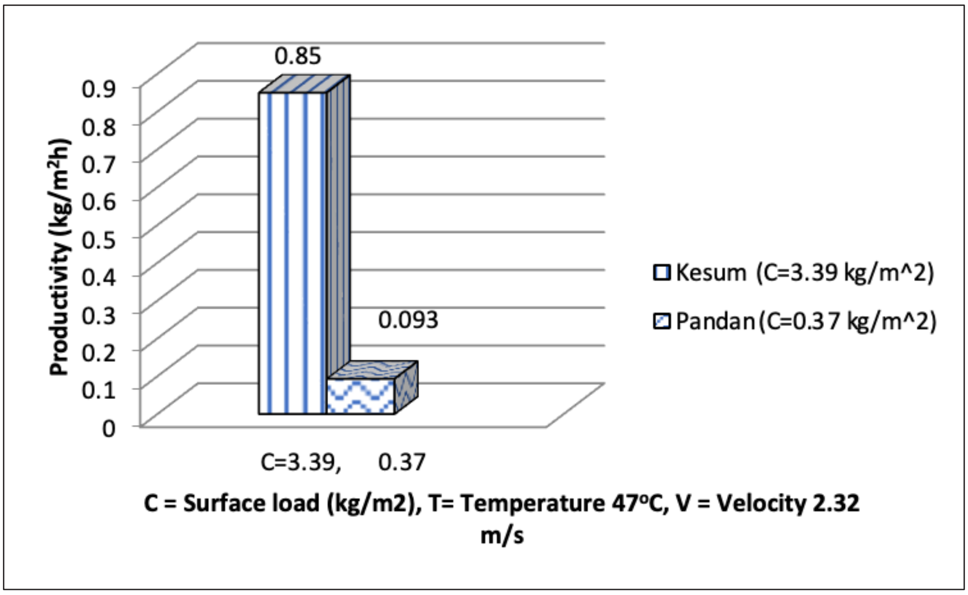

FIGURE 10. Dryer productivity for successful experiments at different surface load.

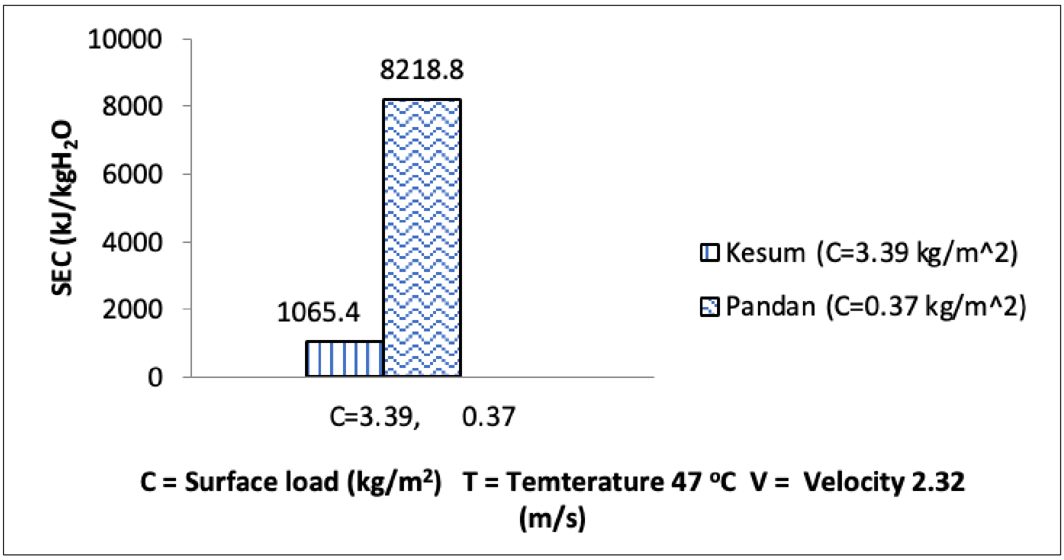

Figure 11. Effect of drying conditions on the specific energy consumption (SEC) of drying the herbs

TABLE 7. Variation of the heat loss in the drying period.

Drying time	0	15	30	45	60	75	**90**	**105**	**120**	**135**	**150**	**165**	**180**	**195**	**210**	**225**	**240**
Heat loss of Wall heat Transfer (kW).	-0.02	0.15	0.24	0.29	0.33	0.35	0.36	0.36	0.38	0.46	0.44	0.42	0.43	0.46	0.44	0.44	0.45
Percantage Of total heat Loss to total Heat (%)	-0.32	2.81	4.54	5.44	6.31	6.72	6.76	6.90	7.26	6.90	8.26	8.03	8.17	8.71	8.30	8.39	8.44

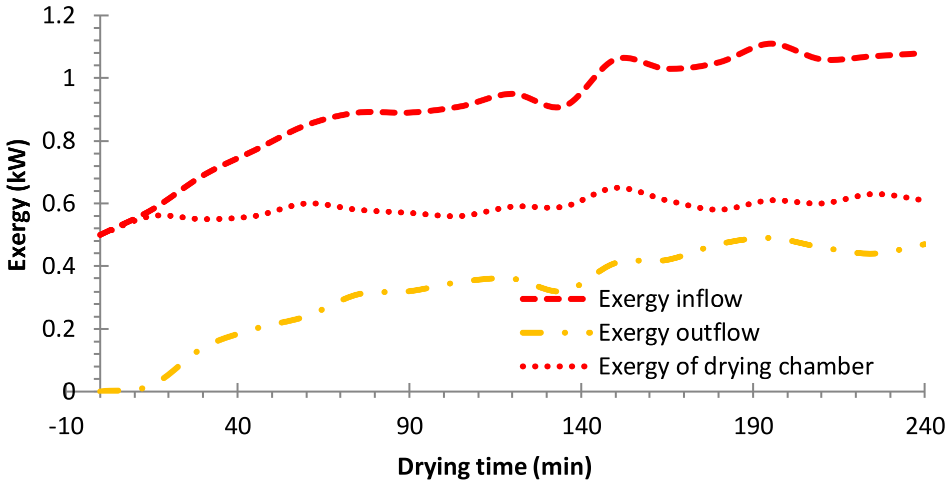

Figure 12. Effect of drying conditions on the specific energy consumption (SEC) of the herbs

TABLE 8. Variations in the evaluation indicators of the drying system.

Drying process	Drying time (min)	η_{ex}	IP (kW	SI (-)	Drying System SEC (kJ/kg)	STEC (kJ/kg)
	0	96.33	0.00	35.00	88.21	0.21
	15	96.33	0.02	27.22	99.75	13.75
	30	79.41	0.11	4.86	121.17	33.59
	45	73.5	0.15	3.77	137.30	47.91
	60	71.18	0.17	3.47	162.67	66.77
	75	65.16	0.20	2.87	175.65	76.59
	90	63.91	0.21	2.77	290.70	128.20
Drying	105	61.58	0.21	2.60	350.68	157.68
period	120	62.25	0.22	2.65	401.78	189.34
	135	64.52	0.21	2.82	430.11	193.11
	150	61.45	0.25	2.59	568.34	299.48
	165	59.05	0.25	2.44	643.42	331.75
	180	55.49	0.26	2.25	779.60	407.60
	195	55.65	0.27	2.25	1023.24	560.24
	210	56.67	0.26	2.31	1307.24	691.91
	225	58.86	0.26	2.43	1978.78	1057.78
	240	56.85	0.26	2.32	3967.54	2127.54

TABLE 9. Cost and economic parameters.

ITEM (USD)	Pandan - HPD	Kesum - HPD
Material and labor cost	8620.68	8620.68
Annually, herb cost	11,827.20	257,040.00
Annually energy cost	377.59	292.11
Annually saving	27,046.80	147,822.48
Inflation rate	2%	2%
Interest of long term investment rate	10%	10%
Life period of the dryer (years)	15	15
Price of fresh Pandan (Shopee,2024),USD/kg	4.48	10.71
Selling price of dried Pandan (Shopee,2024)	24.97	20.00
Electricity cost (USD/kWh)	0.67	0.59
Electricity cost (USD/kg)	0.37	0.02

TABLE 10. Economic of dried pandan and kesum using dual condenser HPD.

	Pandan leaves		Kesum leaves	
Year	Annual net present worth (USD)	Cumulative net present worth (USD)	Annual net present worth (USD)	Cumulative net present worth (USD)
1	24589.09	24589.09	134384.73	134384.73
2	23247.76	47836.85	127054.04	261438.77
3	21979.20	69816.05	121759.33	383198.10
4	20780.98	90597.03	113572.56	496770.67
5	19647.67	110244.70	107378.77	604149.44
6	18575.75	128820.45	101520.54	705669.98

continue...

...continue

7	17562.54	146382.99	95983.08	801653.06
8	16604.50	162987.49	90747.18	892400.24
9	15698.93	178686.42	85798.06	978198.31
10	14842.59	193529.01	81117.98	1059316.29
11	14033.04	207562.05	76693.63	1136009.92
12	13389.30	220951.35	73175.44	1209185.36
13	12543.78	233495.13	68554.49	1277739.85
14	11859.74	245354.87	64816.04	1342555.89
15	11212.75	256567.62	61280.11	1403835.99

CONCLUSION

A single-condenser heat pump dryer (HPD) demonstrated the ability to dry 3.5 kg of pandan leaves in 4 hours. This setup was primarily utilised for comparing the coefficient of performance (COP) to a dual-condenser HPD system. Meanwhile, experiments using the dual-condenser setup were carried out to dry 50 kg of kesum leaves and 5.5 kg of pandan leaves, with the goal of evaluating its performance in terms of energy efficiency, cost, drying characteristics, exergy efficiency, and techno-economic aspects of herb drying. The dual-condenser HPD successfully completed the drying process in 4 hours. The main research conclusions are as follows:

1. The experiment results show that the drier consumed little electricity, measuring 0.012 kW/kg and 0.143 kW/kg of fresh leaves for kesum and pandan, respectively. The average coefficient of performance (COPavg) of the drying system was found to be 5.70 for kesum and 5.34 for pandan. In addition, the heat pump dryer (HPD) had a specific moisture extraction rate (SMER) of 11.23 for kesum and 2.26 for pandan leaves. Notably, implementing the dual-condenser system resulted in an approximately 11% increase in COP.
2. Drying kesum leaves at 47°C, 2.32 m/s air velocity, and 3.39 kg/m² surface load resulted in a maximum dryer productivity of 0.75 kg/m²h and a minimum specific energy consumption (SEC) of 1065.4 kJ/kg H_2O.
3. The heat pump dryer (HPD)'s closed-loop design recirculates exergy outflow, reducing heat loss. The exergy efficiency in this investigation ranged from 56.85% to 96.33% during the drying process.
4. Economic research shows that the heat pump dryer (HPD) saves 147,822.48 USD for kesum leaves and 27,046.80 USD for pandan leaves annually due to low electricity prices. Over a 15-year period, the cumulative net present value of drying kesum leaves was calculated to be 1,403,835.99 USD and 256,567.62 USD for pandan leaves. According to the life cycle cost analysis (LCCA), the payback period for drying 24,000 kg of kesum leaves yearly is 3.24 months, whereas the repayment period for drying 2,640 kg of pandan leaves each year is 6.09 months.

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DECLARATION OF COMPETING INTEREST

None.

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