Experimental Assessment of the Thermal Performance of a Heat Pump Dryer System Based on the Variations in Compressor Discharge Pressure on Oregano Drying

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Received: 15 October 2020; Accepted: 11 November 2020; Published: 1 December 2020

Abstract: The current study shows an empirical analysis to establish the effects of the variations in compressor discharge pressure on the drying performance of aromatic herbs, in terms of the coefficient of performance (CoP), moisture content (MC), specific moisture extraction rate (SMER), drying temperature, drying time and energy consumption. In conducting the research, a heat pump drying system was utilized as a mechanism for dehydrating herbs, seeds, and fruits. It was used thanks to its benefits like higher efficiency and its low power consumption. Three levels of discharge pressure were considered, 1380 kPa, 1100 kPa, and 827 kPa, using 1,1,1,2-tetrafluoroethane (R134a) as a refrigerant and oregano leaves as the main product. The findings show that, concerning the same oregano moisture sample, the lower the compressor discharge pressure, the lower drying temperature, also, the higher drying time was obtained. Despite the fact that the CoP decreased with the compressor discharge pressure, in comparison with the baseline case, it remained essentially the same for the other two cases.

Keywords: heat pump dryer; compressor discharge pressure; energy efficient; coefficient of performance; moisture content; oregano

1. Introduction

Herbs and spices are high significant nutrition products; as components in various dishes, beverages or health and beauty care goods, these are fundamental. The herbs and spice global market presented a significant growth in 2012 of USD $12 billion; it is estimated to reach a value of USD $16.6 billion by 2019 [1]. Aiming at supplying the demand for safe and high-quality products, the food industry requires to provide fresh products; products offering the least (or none) industrial processing. Products ready to consume or which require minimum cooking procedures [2]. Fresh farming crops like fruits, vegetables, and leaves are mainly composed of a high percentage of both, bound and unbound water. The water contents in foods constitutes the environment for the flourishing of bacteria during the products’ storage [3].

Drying is a procedure that is used to steady products by reducing their water activity, moisture contents, and quality loss [4,5]. The drying of farming goods constitutes a valuable and convenient preservation method for reducing damage after harvest; it also expands those products’ lifecycle and helps reducing transport costs [6]. Heat or power generation is the key issue concerning dehydration procedures. Electrical, oil, gas, solar heaters, and heat pump dryers (HPD) are used to take the drying air to the required temperature according to the produces’ drying necessities [7,8]. In the developed
countries, the power consumption ratio of drying procedures is between 10 and 25% of the total energy consumption. Drying is one of the most significant energy-intensive processes with low thermal efficacies between the range of 25–50% [9].

A HPD is a machine which transforms medium-low-temperature heat energy into medium-high temperature heat energy. This conversion is generated by the heat pump using a small amount of high-grade energy. This makes heat pumps reduce power consumption. HPs are ecofriendly devices since they do not use fossil fuels, instead, they use a refrigerant in a closed process [10]. HPDs have been extensively utilized in dehydrating vegetables and fruits [4], grains and farming products thanks to the fact that these can operate at low drying temperatures. HPDs are very attractive for scholars since they have the advantage of controlling drying temperatures and humidity levels; in fact, these pumps offer multiple options to perform drying [11].

SMER is the parameter used to analyze the thermic features of an HPD’s system performance. This rate defines the efficacy of a heat pump and it is defined as, the ratio between the amount of moisture evaporated and the energy consumed by the compressor and the blower [12]. SMER numbers directly measures the energy efficiency in removing moisture from the dried material [13].

Origanum vulgare, commonly known as oregano, is a seasoning aromatic plant globally used to spice various dishes, and processed meat: salads, pizza, and sausages among others [14]. Oregano, as well as other aromatic herbs are dehydrated using temperatures which range from 40 to 60 °C. Drying temperatures between 50 and 60 °C have been selected for their usefulness when drying diverse medicinal plants. It has been found that a minimal quality loss occurs at temperatures around 50 °C when drying herbs and spices [3].

Jew’s mallow, spearmint, and parsley were dried using an HPDs at a drying air temperature of 55 °C and a velocity of 2.7 m/s. Herbs in different sizes were dried with and without the stem. Small herbs without their stems require low energy consumption and short drying time [15]. Some researchers have also used HPDs in order to dry products at temperatures between 35 and 55 °C. Chapchaimoh et al. [16] used a closed HPD dehydrating ginger at 50 °C. For this case, SMER equaled 0.06 kg water/MJ, while the specific energy consumption for air drying was 16.67 MJ/water kg. Taşeri et al. [17] used an HPD for drying Pomace. This case had an initial moisture content of 2.57 (dry basis) to final a moisture content of 0.1 g water/g dry matter (dry basis) at a drying air temperature of 45 °C for three different air drying velocities, 1.5, 2 and 2.5 m/s; 1040, 840 and 720 min were the time lapses required to achieve the desired final moisture content for each velocity. In 2017, Aktaş et al. [6] dried mint leaves from 9 g water/g dry matter to 0.1 g water/g dry matter at 35 °C and a velocity of 3 m/s; the procedure was carried out using an HPD. When the velocity in the air was 2.5 m/s the CoP for the whole system was 2.64, while the CoP the HPD was 4.188. Also, the system’s total energy consumed was 2.969 kWh. Aktaş et al. [18] dried bay leaves using HPDs, temperature, humidity and air speed were controlled by a programmable logic controller (PLC) in order to run the dryer efficiently. Through an artificial neural network, the best drying conditions were adjusted (based on empirical tests); these showed that the suitable drying temperature was 45 °C. Harchegani et al. [19] found a CoP of 0.76, a SMER of 0.1 kg/kWh when an HPD was used for drying rice until it reached 11–12% of moisture content on wet basis (w.b) at 35 °C (drying temperature).

Authors like Shen et al. [20] empirically analyzed a heat pump for dual mode drying: single-stage and cascade cycle. In both modes, the provided air temperature reached 70 °C, at an ambient temperature of 0 °C for the cascade cycle mode and 20 °C for the single stage mode. A higher CoP was reported when the HPD runs under a single-stage cycle for an ambient temperature of 10 °C. Song et al. [4] evaluated an HPD combined with far-infrared radiation in Chinese Yam chips at a drying temperature 50 °C. The results showed that the simultaneous use of the HPD and the far-infrared radiation increases the drying rate by reducing the drying time. The specific power consumption reduced by about 44.08% when the drying process was conducted in the dual mode. Mohammadi et al. [21] studied the consequences of air recirculation when a heat pump was used for dehydrating kiwifruit slices.
It was noticed by the authors of this paper that HPDs have been used in dehydration processes of farming produces. Nevertheless, there is no evidence of studies on the effect of compressor discharge pressure variations on the thermal performance in heat pumps when herb drying. When a heat pump based on a refrigeration vapor compression cycle is used for drying herbs, the pressure at compressor discharge determines the drying temperature. Low pressure at compressor discharge causes a long time for dehydrating, and high energy consumptions, while extremely high pressure generates high values of drying temperature, eliminating more water from herbs than necessary, this reduces the quality of products. Other consequences are related to high temperatures in the motor compressor, bringing out damage to the winding and changes in the lubricating oil viscosity. This paper studies the effects of compressor discharge pressure variations on the thermal performance of a heat pump dryer system used for oregano drying. Through this study, it was possible to determine the optimum drying condition for oregano when refrigerant R134a is used as a working fluid in a heat pump.

2. Materials and Methods

2.1. Experimental Setup

The experimental test bench, where measurements were carried out, is detailed in Figure 1a (schematic diagram) and Figure 1b (experimental arrangement), which mainly consists of a heat pump system to be operated with R134a, a drying room, airflow ducts, and a psychometric room, a piston compressor, a finned cross-flow evaporator, a finned cross-flow condenser, a centrifugal blower and some auxiliary elements like measurement and control equipment. List part showed at Figure 1a is described at Table 1.

![Figure 1. Experimental test bench for Oregano drying: (a) Schematic diagram of HPDS test bench used for the experimental analysis; (b) Experimental arrangement for Oregano drying.](image_url)

Table 1. Components and specifications of the HPD system.

| Component | Type | Technical Specifications |
|-----------|------|--------------------------|
| 1 Compressor | Alternative hermetic piston, Embraco FFU 100HAK model | 248.56 W, 115–127 V, 5 A, 60 Hz |
| 2 Evaporator | Finned cross-flow heat exchanger | 8 fin/in |
| 3 Compressor Air | SunFlow centrifugal fan, 172*150*50 mm HBL model | 0.37 kW, 2.1 m³/h |
| 4 Blower | Temperature display and registration unit | Electronic interface made with Arduino 5 V, 1.5 A |
| 5 High and low pressure sensor | Bourdon manometer | 0–3447.3 kPa; 0–827 kPa |
| 6 Pipeline for air recirculation | Thermal isolated pipe | 4” = 0.10 m, aluminum, fiber glass as isolating system |
| 7 Tray 1 | Grill | 34 cm × 34.5 cm |
| 8 Tray 2 | Grill | 34 cm × 34.5 cm |
| 9 Baffle | Aluminum deflector | degree tilt 45° |
| 10 Condenser | Finned cross-flow heat exchanger | 8 fin/in |
An existing refrigeration system was used and modified in order to adapt it with a drying chamber. A compressor, capillary tube, evaporator, and a condenser heat exchanger were used without modification because the purpose of our research was to study the effect of compressor pressure discharge on drying efficiency, and factors such as the thermal capacities of the evaporator and the condenser are subject to the cooling capacity of the compressor, which, according to the “global company in refrigeration solutions - Embraco” technical manual, is 151 watts when the evaporation temperature is −35 °C and the condensing temperature is 45 °C. According to the above operating conditions, the diameter of the capillary tube was selected using Danfoss capillary tube selector software, which suggested a diameter of 1.4 mm and a length of 1.5 m. Table 1 presents the components and specifications of the HPD system used in this study.

2.2. Experimental Design

Tests were performed according to an experimental design with just one factor (in this case moisture ratio was selected as the main drying characteristic) and three levels of compressor discharge pressure (1380 kPa as the baseline, 1100 kPa, 827 kPa). Table 2 lists the ranges and conditions of the experimental runs covered in the present study. Considering that recommends temperature for drying herbs are between 35 to 55 °C, three different pressure at discharge compressor were selected taking account of the saturation temperature for the refrigerant R134a. According to saturation tables for R134a, at 827, 1100, and 1380 kPa corresponds to a saturation temperature equivalent to 36.27, 46.46, and 54.69 °C, respectively, drying temperatures values suitable for dehydrating herbs. Compressor velocity remained constant for the 3 selected discharge pressures. This velocity was 3600 RPM, nominal compressor velocity. The discharge compressor pressure was modified charging the system with more or less refrigerant mass. The constant parameters during the experiments are illustrated in Table 3.

Table 2. Technical specifications of sensor and measurement devices.

| Instruments                        | Properties                          | Range             | Accuracy  | Uncertainty |
|------------------------------------|-------------------------------------|-------------------|-----------|-------------|
| Temperature and humidity sensor    | Power supply 3–5.5 VDC              | −40–80 °C        | ±0.1 °C   | ±0.1 °C     |
| Low pressure                       | Mechanical manifold gauge           | 0–11884.49 kPa   | ±1%       | N/A         |
| High pressure                      | Mechanical manifold gauge           | 0–23768.3 kPa    | ±1%       | N/A         |
| Velocity                           | Multifunction vane thermo-anemometer| 0.6–32 m/s       | ±5%       | 0.1 m/s     |
| Digital balance                    | BOECO-3000 g                        | 0–3000 g         | ±0.01 g   | ±0.2 g      |
| Thermographic Camera              | FLIR E4                             | −20–250 °C       | ±2%       | 0.15 °C     |
| Superheating and subcooling calculator | 24 preprogrammed refrigerants and a Type K thermocouple | −40–200 °C | ±0.1 °C | ±0.2 °C |
| Voltage and current measurement device | Digital clamp multimeter, UT207A | 0–600 VAC | ±1.5% | N/A |
| Thermogravimetric Analyzer        | Thermogravimetric analyzer Q6000 SDT | 0.02% | 0.0001 g |
| Data acquisition system           | Arduino-based data Acquisition system | 10 points for temperature measurement | N/A | N/A |

Table 3. Constant parameter during the experiments.

| Parameter                      | Operating Range |
|--------------------------------|-----------------|
| Air velocity                   | 1.2 m/s         |
| Refrigerant mass flow          | 0.0031 kg/s     |
| Surrounding air temperature    | 25 °C (average) |
| Surrounding relative humidity  | 68% (average)   |
| Volumetric air flow            | 195 m³/h        |
| Atmospheric pressure           | 85.2 kPa        |
Taking in account that this work aims to study the effect of the discharge pressure of HPD in the process of drying herbs, it is necessary to determinate if some of the discharge pressures levels have a statistical effect on the drying performance. In this case, analysis of variance (ANOVA) evaluates the importance of one or several factors, comparing the means of the variable studied. If detailed information about the differences between specific means is needed, a method for multiple comparison should be used, as the so called Fisher least significant difference (LSD) method. Statistical analysis was done by one-way ANOVA, followed by LSD (least significant difference) test with P lower than 0.05 as a limit of significance. An experimental run, plus three replicas, was performed as follows: Initially, the moisture content of Oregano samples around 70%, measured by means of thermal gravimetric analysis (TGA). In order to carry out the experiments, 100 g of Oregano were used for each of the two trays in all tests. The drying process started by introducing the trays inside the heat pump, which had a temperature of near to the drying temperature, for each pressure considered in the work. Then, every 10 min, trays were extracted from the HPDS for monitoring of moisture removal, thermographic scan; before of relocation of the trays inside HPDS, they were shifted in their position and were rotated 180° clockwise, to ensure uniform velocity and heat flux pattern in each tray. By means of a damper, fresh air incoming to HPDS was used to retain a high pressure to ensure a constant drying temperature, every time that a high value of compressor discharge pressure was noticed, the damper was completely opened until the high pressure reached its nominal value. The variation of compressor discharge pressure was established varying the refrigerant charge into the cycle, in order to obtain the corresponding pressures of 1380 kPa, 1100 kPa and 827 kPa. During the drying procedure, the electric current of the compressor and the drying blower, the manometric pressure (high and low side), the subcooling and superheating, the temperature of the compressor housing, and the temperatures and humidity in selected parts of the heat pump were continuously registered. Figure 2 shows the drying process of the material (fresh oregano) and the thermography for the final product at 1380 kPa.

![Drying process](image)

**Figure 2.** Drying process of fresh Oregano: (a) Oregano leaves before (left) and after (right) of drying process, (b) thermography for final product at 1380 kPa.

### 2.3. Model Development

The energetic evaluation was made based on performance metrics as heating capacity ($\dot{Q}_h$), coefficient of performance of the heat pump ($\text{CoP}_{\text{system}}$), and the specific moisture extraction rate (SMER).

- **Heating capacity**

  The condenser heating capacity was determinate by the following equation:

  $$\dot{Q}_h = m_r(h_{c2} - h_{co})$$

  Here, $h_{c2}$ and $h_{co}$ in [kJ/kg] are the enthalpy of the refrigerant before and after leaving the condenser, respectively.

- **Refrigerant mass flow rate**
The mass flow rate of the refrigerant through the condenser was calculated from an energy balance between the drying air and the refrigerant, and was determined by the following equation:

\[
\dot{m}_r = \frac{\dot{m}_a c_p (T_{co} - T_{ci})}{(h_{ci} - h_{co})}
\]  

(2)

Here, \(\dot{m}_a\) is the mass flow rate of air passing through the condenser coil [kg/s]. \(T_{co}\) and \(T_{ci}\) are the air temperatures at the outlet and inlet condenser respectively.

The \(\text{CoP}_{\text{system}}\) of the HPDs was calculated by the following equation:

\[
\text{CoP}_{\text{system}} = \frac{Q_h}{W_{\text{comp}} + W_{\text{fan}}}
\]

(3)

where \(W_{\text{comp}}\) and \(W_{\text{fan}}\) are the power consumed by the compressor and drying fan respectively and can be expressed as Equations (4) and (5).

\[
W_{\text{comp}} = \dot{m}_r (h_2 - h_1)
\]

(4)

\[
W_{\text{fan}} = V \times I_{\text{fan}}
\]

(5)

where \(h_2, h_1\) are the refrigerant enthalpy’s at the outlet and inlet at the compressor respectively, \(V\) is the input voltage to the fan motor and \(I_{\text{fan}}\) is the current consumed by the fan motor.

In order to evaluate the effect of the compressor pressure discharge over the heating of the compressor the superheating of the refrigerant at the suction compressor was evaluated using the following equation:

\[
\Delta T_{sh} = T_1 - T_{sat}
\]

(6)

where \(T_1\) is the refrigerant measured temperature at the compressor inlet and \(T_{sat}\) is the saturation temperature read at the inlet compressor. The superheating was measured at 30 cm from the entrance of the compressor using a compact subcool/superheat calculator.

- **Drying and energy parameter**

  The variation for the moisture ratio (MR) during the experiment was calculated using the following equation:

  \[
  \text{MR} = \frac{M_t - M_e}{M_o - M_e}
  \]

(7)

where \(M_t\) and \(M_e\) were the moisture contents at the time \(t\) and initial time on a dry basis, respectively, kg; \(M_o\) was the equilibrium moisture content.

The specific moisture extraction rate (SMER) is the ratio of the moisture evaporated from the wet product to the energy input to the HPDs and it was calculated as follows:

\[
\text{SMER} = \frac{\dot{m}_{\text{w}}}{E_i}
\]

(8)

Where \(\dot{m}_{\text{w}}\) is the evaporated mass water and \(E_i\) is the electrical energy input to the HPDs and was calculated as Equation (9).

\[
E_i = W_{\text{ec}} + W_{\text{fan}}
\]

(9)

\(W_{\text{ec}}\) is the compressor power consumption and was calculated as follows:

\[
W_{\text{ec}} = V \times I_{\text{comp}}
\]

(10)

\(V\) is the input voltage to the compressor motor and \(I_{\text{comp}}\) is the current consumed by the compressor.
In order to validate the result of the ANOVA, Fisher’s LSD test, in which all pair of means were compared with the null hypotheses using the \( t \)-statistic, was performed:

\[
t_0 = \frac{|y_i - y_j|}{\sqrt{\frac{2 \times MSE}{n}}}
\]  

(11)

where \( t_0 \) is the \( t \)-statistic, \( y_i \) and \( y_j \) are the means that are being compared, MSE is the error mean square and \( n \) is the number of observations.

Assuming a two-sided alternative hypothesis, the pair of means \( i \) and \( j \) would be declared significantly different if:

\[
|y_i - y_j| > LSD
\]

(12)

where \( LSD \), the least significant difference, is calculated thus, for the case of samples with the same size and it was calculated as follows:

\[
LSD = t_{\frac{\alpha}{2}, a(n-1)} \sqrt{\frac{2 \times MSE}{n}}
\]

(13)

where \( t_{\frac{\alpha}{2}, a(n-1)} \) is the \( t \)-statistic evaluated with a significance level \( a = 0.025 \), \( a \) is the number of means and \( n \) is the number of observations

3. Results and Discussion

3.1. Subsection Impact of Variation in Compressor Pressure Discharge on Oregano Drying Characteristics

Colombian regulations have set the allowed moisture content for various aromatic herbs and other goods to “around 14%.” Figure 3 shows the moisture contents obtained for each considered compressor discharge pressure. As expected, the drying time increased as the compressor discharge pressure decreased. For the baseline case, the shortest drying time was almost 150 min, followed by the cases in which the compressor discharge pressures were 1100 and 827 kPa, with almost 270 and 450 min of drying time, respectively, which were about twice and three times the HPD’s nominal capacity drying time. This is because the lower gradient in the dehumidification rate and the controlled air velocity makes the moisture removal rate strongly dependent on temperature. This tendency can be explained by considering the drying temperature profiles.

![Figure 3](image-url)  
Figure 3. Moisture content variation as a function of compressor discharge pressure.
Figure 4 shows the air humidity ratio at the entrance of the drying chamber for each pressure evaluated. When the discharge pressure compressor was 827 kPa a 0.012 kg H$_2$O/kg dry air was observed, while for 1380 kPa the air humidity ratio was 0.009 kg H$_2$O/kg dry air. This lower value of air humidity ratio causes a major capacity of drying air to absorb water vapor from Oregano leaves, increasing the SMER of the heat pump dryer system due to the reduction of drying time.

Figure 5 indicates that the baseline case reached an average of 46 °C. The other cases showed, average drying temperatures of 7 °C and 13 °C, lower than in the baseline case. Several studies presented considerable increases in the drying time, with decreasing temperature levels [22,23].

3.2. Efficiency of Oregano Drying Process in HPDs

As for indicators of the performance of the heat pump, CoP and SMER were chosen as the most representatives for determining the efficiency of such systems. In general, CoP obtained was in the range of 0 to 4 for Single-stage motor-driven HPD configuration. Figure 6 plots the coefficient of
performance for each case considered in this work, resulted in a reduction near to 20% when compressor discharge pressure decreases from 1380 kPa to 1100 kPa and 827 kPa.

The COPs values obtained in each experiment are in accordance with those reported by the manufacturer in the compressor data sheets, which can present a maximum COP of 2.13 when the evaporation temperature is −5 °C and the temperature of condensation is 55 °C. According to the manufacturer, a COP value of 1.14 is obtained when the evaporation temperature is 35 °C and the condensation temperature is 45 °C. The results obtained in the present study are not significantly different from those reported by the manufacturer.

Figure 7 shows the SMER, as calculated using Equation (8). The CoP value for the HPD system was calculated using Equation (3). The observed trend established that as the compressor discharge pressure decreased, so did the CoP and the SMER, with the latter being more noticeable than the CoP. A 23% reduction of the CoP was observed with respect to the baseline case, while the SMER decreased by 44%. This can be explained by the isentropic height corresponding to the work of the compressor and the heat dissipated in the condenser, where an appreciable increase in the isentropic height was observed when passing from the base pressure to 1100 and 827 kPa, which also represented a proportional increase in compressor work that resulted in a lower CoP and a lower SMER.

Figure 7. Specific moisture extraction rate.
3.3. Analysis of Variance for the HPDS Drying Process

The experimentation involved 4 runs for each level of compressor discharge pressure, in which every 30 min was measured the air temperature on the drying chamber. The average air temperatures obtained for each tested pressure, are summarized in Table 4.

Table 4. Average air temperatures (°C) for each compressor discharge pressure.

| Pressure    | 1380 kPa | 1100 kPa | 827 kPa |
|-------------|----------|----------|---------|
| Temperature | 46.3     | 39.2     | 32.9    |
|             | 46.7     | 39.9     | 34.0    |
|             | 46.1     | 39.5     | 34.8    |
|             | 46.3     | 39.5     | 33.9    |

The null hypothesis stated that there would be no difference between the means for each compressor discharge pressure, and alternative hypothesis stated that there would be at least a small difference between a pair of means.

\[
H_0: \mu_{1380\, \text{kPa}} = \mu_{1100\, \text{kPa}} = \mu_{827\, \text{kPa}}
\]

\[
H_A: \mu_{1380\, \text{kPa}} \neq \mu_{1100\, \text{kPa}} \neq \mu_{827\, \text{kPa}}
\]

The results of the analysis of variance, performed with a significance level of \( \alpha = 0.005 \), can be seen in Table 5. Each run consisted of nine experimental measured points for the case of 1380 kPa and 12 and 16 experimental measured points for the cases of 1100 and 827 kPa, respectively. In statistical terms, this amount of data was enough for a statistical analysis. Considering that the probability was lower than the \( p \)-value, the null hypothesis can be rejected; this indicates that, with 95% of confidence, it can be stated that at least a pair of compressor discharge pressures had different drying temperatures due to the effect of the variation on the discharge pressure of the compressor. In the same way, it can be said that the total variation in all three experimental points was 236.06. Of this amount, 233.76 was due to differences between compressor pressure discharge and 2.3 was due to the difference between pressures of the same magnitude. Correlating with the corresponding degrees of freedom, the mean squares that reflected the real magnitude of each source of variation were obtained. Therefore, the differences due to the compressor pressure discharge was found to be 116.88 with an error of 0.38; this means that the first was pressure discharge was 305.17 times larger than the second, which indicated that the differences observed between the pressure types are significant and were not due to small sample variations or experimental errors.

Table 5. Analysis of variance for the drying temperature.

| Source of Variations | Sum of Squares | Degree of Freedom | Mean Squares | \( F \)-Value | \( p \)-Value | \( F \) critical Value |
|----------------------|---------------|------------------|--------------|--------------|--------------|----------------------|
| Between groups       | 233.76        | 2                | 116.88       | 305.17       | 9.2258 \times 10^{-7} | 5.14325285           |
| Inside groups        | 2.30          | 6                | 0.38         |              |              |                      |
| Total                | 236.06        | 8                |              |              |              |                      |

Regarding that the null hypothesis was rejected in the ANOVA, it is known that some of the treatment or factor level means are different. As the ANOVA doesn’t identify which means are different, Fisher’s least significant difference (LSD = 1.2364) test was performed to validate the previous statement, using the t-statistic (2.4469). Results are summarized in Table 6.
Table 6. Results of the LSD means comparison.

| Comparison               | \(|\bar{y}_i - \bar{y}_f|\) | Result  |
|--------------------------|-------------------------------|---------|
| 1380 kPa vs 1100 kPa     | 6.8                           | >1.2364 |
| 1380 kPa vs 827 kPa      | 12.5                          | >1.2364 |
| 1100 kPa vs 827 kPa      | 5.6                           | >1.2364 |

From this analysis, it can be seen that there are significant differences between all pairs of means, or in other words, there are significant differences between all pairs of compressor discharge pressure means. This implies that pressure produces different drying temperatures.

3.4. Behavior of Superheating with the Variation of the Compressor Discharge Pressure

In Figure 8, the superheating of the refrigerant in the compressor suction can be observed, for each of the pressures considered in the analysis. The highest overheating temperature was obtained in the case of 827 kPa, with a temperature around 64 °C, followed by the case of 1100 kPa with an average value of 54 °C. Both are significantly higher than the base case of 1380 kPa, which is due to the low amount of refrigerant in the system, which causes an external obstruction in the evaporation coil due to the formation of frost on the outer surface of the evaporator, reducing the mass flow of air towards the condenser, affecting the ability to remove moisture from the product in the drying chamber, which can be evidenced by the drop in the coefficient of performance, and the SMER for the pressures of 827 and 1100 kPa. It can be observed that the subcooling of refrigerant at the compressor outlet does not have a significant effect on the operation of the system.

![Figure 8. Superheating profile for all operation discharge pressure.](image)

By varying the refrigerant charge in the system, we found that the pressure on the low side decreased as the compressor discharge pressure decreased. To prevent this, it would be necessary to modify the characteristics of the capillary tube, an activity that was not carried out in each experiment, since the object of the study was to evaluate the effect of operating a heat pump operating under different compressor discharge pressures without modifying its components. Is suggested the use of an expansion valve would allow maintaining a constant evaporation pressure while modifying the discharge pressure.

Low evaporation temperatures require larger capillary diameters. As the diameter of the capillary was preserved, it is to be expected to have a lower refrigerant mass flow than required and therefore high values for superheating.
As can be seen in the Figure 9, the effect of lowering the discharge pressure of the refrigerant in the compressor causes a drop in the suction pressure, and in turn a very low evaporation temperature. When the discharge pressure was 827 kPa, the evaporation temperature decreased to $-36^\circ$C, causing a partial obstruction of the evaporator due to the formation of frost on its surface and therefore a reduction of the airflow to the condenser and the drying chamber, which affected the convective coefficient of the air and increased its absolute humidity, as shown in Figure 2, reducing the ability of the air to absorb moisture from the product in the drying chamber.

![Figure 9. P-h diagram for the drying heat pump.](image)

4. Conclusions

An empirical study was conducted to determine the influence of decreasing compressor discharge pressures on the drying process of an aromatic herb: oregano. The results indicate that the most affected drying parameter was the drying temperature, which led to longer drying times for the same final moisture content of the product.

Based on superheating results in the compressor suction, it was established that operating the heat pump at a pressure lower than or equal to 1100 kPa can seriously affect the compressor’s performance due to overheating. This caused a decrease in the viscosity of lubricating oil, shortening the compressor’s life. On the other hand, very low pressures in the compressor’s discharge imply a greater thermal load for the condenser, so that much of its area would be used to lower the temperature of the refrigerant to reach the saturation point; it compromises the condensation and results in a decrease in the drying capacity, and therefore, longer drying times.

Both, the CoP and SMER decreased along with the compressor discharge pressure. This is mainly due to the increase in the energy demanded by the compressor. This demand is caused by an increase in the compressor’s isentropic height, as the discharge pressure decreased. When the discharge pressure was 1380 kPa, a compression ratio of 6.34 was obtained, while 1100 kPa caused an increased 9.61 ratio; this means a 51.6% higher than the base case, causing a 73% increase in the power absorbed by the compressor.

Heat pump drying is a viable method for oregano drying. By calibrating the pressure at the compressor’s inlet and outlet, high values for CoP, SMER can be reached; additionally, this can lower operational costs. Currently, HPDs are not massively used in Colombia; solar drying is commonly used instead. This practice makes it difficult to control the temperature and time in the drying process. From these results, it can be concluded that, by using HPDs and adjusting the compressor’s discharge pressure, different air-drying temperatures could be achieved. This can be used for setting the air-drying temperature for other types of herbs and products without a high impact on the environment (since common fossil fuel are not being used).
In further works, it is highly recommended to use a variable speed compressor controlled by a variable frequency drive (VFD) in order to get different working pressures for the compressor; in that way, it could be possible to reach different drying temperatures without manipulating the refrigerant charge on the HPD. In turn, this would reduce the risk of refrigerant leaks (polluting the environment). It is suggested to pay attention to factors such as leaks and obstructions in the refrigeration pipes which imply drops in the compressor discharge pressure up to 15% of the nominal operating value.

This study evaluated the effect of the variation of the pressure in the compressor discharge on the performance of a heat pump for dried herbs, these evaluated pressure values were reached by varying the refrigerant load in the system, while the speed was maintained constant. This obviously keeps the temperature of the motor coil constant, as does the heat generated by friction between the moving parts. Using a variable speed drive could bring benefits over the experimentation carried out since it could reduce the level of superheating in the compressor suction, which is very expensive due to the low refrigerant load in the system. The reduction of start-up peaks could bring energy savings, which would achieve higher COP and SMER values.

**Author Contributions:** Conceptualization, A.S.-H., A.D.-M. and A.F.R.-M.; methodology, A.S.-H., A.D.-M., A.F.R.-M. and E.D.-G.; software, A.S.-H.; validation, A.S.-H., A.D.-M. and A.F.R.-M.; formal analysis, A.S.-H., A.D.-M. and A.F.R.-M.; investigation, A.S.-H., A.D.-M. and A.F.R.-M.; data curation, A.S.-H., A.D.-M. and A.F.R.-M.; writing—original draft preparation, A.S.-H., A.D.-M. and A.F.R.-M.; visualization, A.S.-H., A.D.-M., A.F.R.-M. and E.D.-G.; writing—review and editing, A.S.-H., A.D.-M., A.F.R.-M. and E.D.-G.; supervision, A.S.-H., A.D.-M., A.F.R.-M. and E.D.-G. All authors have read and agreed to the published version of the manuscript.

**Funding:** This research was funded by Institución Universitaria Pascual Bravo, grant number IN201506.

**Acknowledgments:** We would like to thank the Direction of Technology and Innovation of the Institución Universitaria Pascual Bravo for the financial support to carry out the research project “Implementing didactic tools for the teaching of thermal sciences” at the Institución Universitaria Pascual Bravo. The project identified with the code SM201801. We also wish to acknowledge the help provided by the students María Camila Moncada, Diego Cárdenas for their support in the execution of the experimental tests.

**Conflicts of Interest:** The authors declare no conflict of interest. The funders had no role in the design of the study; in the collection, analyses, or interpretation of data; in the writing of the manuscript, or in the decision to publish the results.

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