Exergetic analyses of a domestic heat pump for drying

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ABSTRACT
The use of heat pumps for heating and cooling a building or a single room is a technology that has been widely developed in recent times, the main advantage is that they provide higher efficiency compared to other heating methods such as fossil fuel combustion. But in many regions of the world, it is still a little or nothing known technology, since environmental and economic conditions have not allowed its development and use. This paper analyzes from the exergetic point of view the use of a domestic heat pump with the dual purpose of taking advantage of both the higher temperature side in an herbal dryer and the low temperature to cool a room. COP of the heat pump was 6.5.
Keywords: domestic heat pump, conditioned air, herbs dryer, COP.

1 INTRODUCTION

To heat the air required in a dryer of vegetables, grains and other foods, electric heaters have been used, although there are also gas, oil, solar collectors, and heat pumps this depends on the level of temperature required in each product. The heat pump is an efficient and environmentally friendly technology when both sides, hot and cold, are used, because the energy consumption is slightly lower, [1].

Heat pump systems offer economical heat recovery alternatives from various sources for use in various industrial, commercial, and residential applications. As the cost of energy continues to rise, it becomes imperative to save energy and improve overall energy efficiency,[2] In this sense, the heat pump becomes a key component in an energy recovery system with great energy saving potential. Improving heat pump performance, reliability and its environmental impact has been a constant concern,[3]

The results obtained from the application of heat pumps for the drying of agricultural products have shown that they are a reliable system that guarantees the quality of the final product, mainly using as a source of energy the solar, some hot air current, the energy of the subsoil, and others, to achieve this it is necessary to control the drying temperature, relative humidity, air velocity inside the dryer, drying time, etc. To improve the efficiency of the heat pump dryer, some factors must be known that allow it, such as: include the installation cost, drying performance, such as temperature, speed and relative humidity of the air, the performance of the heat pump dryer, the power required to run the system and the payback period, [4].

Among the main advantages of the heat pump dryer are: product quality can be significantly improved by drying at low temperatures where the air drying potential can be maintained by reducing air humidity; the temperature range can be maintained between 20 to 100 C (with auxiliary heating) and a relative humidity of 15-80% by adding a humidification system; and excellent environmental control for some products by reducing energy consumption, [2].

In this work a heat pump system is used with dual purpose, on the one hand, to take advantage of the chilly air current to provide air conditioning to a room and on the other to use the hot current for an herbal dryer in conditions of forced convection.

2 EQUIPMENT DESCRIPTION

The heat pump system used in this work has a capacity of 4,103 kW (14,000 BTU/h) of energy, approximately 1.17 TR (TR stands for tons of refrigeration). The refrigerant used R410A, is the most environmentally friendly one. It has a control that allows to adjust the evaporator fan at three speeds, its
Airflow in standard conditions ranges between a maximum of $450 \ m^3/h$ ($0.125 \ m^3/s$), and minimum of $350 \ m^3/h$ ($1.1 \ m^3/s$); its hot air fan has similar characteristics, the drive motor power of both fans is 250 W, a rotary compressor of 1.13 kW input power, it has a R410A displacement of 11.4 cc/rev. We present the scheme of the heat pump and the drying tunnel used in this study in figure 1.

Air in the drying tunnel is conducted by means of an axial-flow fan, the use of this is optional, the velocity of air can be varied between 1.0 and 4.0 m/s; There is a panel of two electrical resistances of 1750 W each one, they are managed independently, which allows the selection of the energy to be provided, there are optional too; The test chamber is 20 cm width, 20 cm length and 40 cm height, it has an air outlet designed as a vertical vent. We present the scheme of the drying tunnel used in this study in figure 2.
In order to conduct the experimental evaluation, 6 K-type thermocouples, (Nickel-Aluminum), were adapted and properly calibrated prior to installation, four of which were placed on the pipe through which the refrigerant circulates, ensuring that physical contact was as good as possible; two of them were located at the entrance and exit of the compressor, another one at the outlet of the condenser and a fourth thermocouple, at the entrance of the evaporator. The other two ones were placed at the outlet of the chilly air to the room and at the exit of the warm air. The energy of this last warm air flow is discarded into the environment, but further action should follow with it. We used A 407113 CFM model, Extech brand metal vane anemometer to measure the magnitude of both mass flows: with a measuring range of 0.5 to 126 m/s and an accuracy of 2 %. We recorded these values using an Arduino circuit with their immediate registration.

3 EXPERIMENTAL METHODOLOGY

The heat pump was installed in a 150 m$^3$ volume room, following the manufacturer's recommendations. The orientation of this room is south-north, with the main access door facing east, the unit's warm air discharge pipe goes through a west-facing window. We installed two thermocouples more inside to know the average indoor temperature. Environmental conditions were also recorded during the equipment's testing period. The evaluation months were March and April, of this year during the first and the third week of each month, scheduled from 10:00 to 18:00 hours, the five working days. All of them occurred at full load of the compressor, i.e., its operation was not limited and the temperature inside the room was adjusted to 22 °C. The average ambient temperatures for the selected months (and relative humidity) were 28.5 °C (40 %), and 28.6 °C (60 %), respectively. We obtained the thermodynamic properties of the R410A refrigerant using the EES software and compared to the thermodynamic diagram in which the thermodynamic cycle was drawn, as depicted in figure 3.

4 THEORETICAL ANALYSIS

We present the energy and exergy analysis for the diagram of the thermodynamic cycle shown in figure 1. To simplify the analysis, we made following assumptions: the entire thermodynamic cycle was considered under permanent flow conditions, and pressure loss in the condenser and evaporator, as well as in pipes, were neglected. The reference environmental state for the system is water at an environment temperature of 25 °C and P = 101 kPa; we considered the evaporator capacity constant; the heat transfer of the system to the surroundings was zero, except for the evaporator and condenser.

To perform the calculation of the energies involved and thus obtain the COP (coefficient of performance) and the exergetic efficiency of all the components of the cycle, we applied the equations of conservation of mass, energy, and exergy [5, 6]. The mass flow conservation equation is:
\[ \Sigma \dot{m}_{in} = \Sigma \dot{m}_{out} \]  

(1)

Where:

\( \dot{m} \) is the mass flow of input and output, respectively.

The energy conservation equation is:

\[ \dot{Q}_{in} + \dot{W}_{in} + \Sigma \dot{m}_{in} (h + \frac{v^2}{2} + gz) = \dot{Q}_{out} + \dot{W}_{out} + \Sigma \dot{m}_{out} (h + \frac{v^2}{2} + gz) \]  

(2)

In which \( \dot{Q} \) is the heat flow transfer; \( \dot{W} \) is the supplied power; \( h \) is the specific enthalpy; \( V \) is the velocity; \( gz \) is both input and output potential energy. The overall equation of the exergy balance is:

\[ \Sigma (1 - \frac{T_0}{T_k}) * \dot{Q}_k - \dot{W} + \Sigma \dot{m}_{in} \Psi_{in} - \Sigma \dot{m}_{out} \Psi_{out} = \dot{E}_x_d \]  

(3)

Where:

\( T_0 \) is the temperature of the environment; \( T_k \) is the temperature at which heat transfer takes place; \( \Psi \) is the specific exergy flow, and \( \dot{E}_x_d \) is the destroyed exergy flow.

The specific discharge of the refrigerant or air is:

\[ \Psi_{r,a} = (h - h_0) - T_0(s - s_0) \]  

(4)

In which \( s \) is the entropy of the refrigerant or the air. Using the notation in figure 1, with the conditions indicated above,

\[ q_c + w_{in} = h_2 - h_1 \]
\[ q_{con} = h_3 - h_2 \]
\[ h_3 = h_4 \]
\[ q_{eva} = h_1 - h_4 \]

The COP is determined as:
\[ \text{COP} = \frac{q_r}{w} \]  

The loss of availability of each component is calculated according to the following expression:

\[ W = \sum_{\text{out}} m b - \sum_{\text{in}} m b - \sum Q_j \left(1 - \frac{T_0}{T_j}\right) + l_t \]  

In which \( b = \left(h + \frac{v^2}{2} + gz - T_0 s\right) \), \( T_j \) is the temperature of each element, and \( l_t \) is total irreversibility. So, for each component:

- For the compressor: \( \psi_1 + w_{in} = \psi_2 - \sum q_i \left(1 - \frac{T_0}{T_i}\right) + i_c \)
- For the condenser: \( \psi_2 = \psi_3 - \sum q_i \left(1 - \frac{T_0}{T_i}\right) + i_{con} \)
- For the capillary tube: \( \psi_3 = \psi_4 + i_{ct} \)
- For the evaporator: \( \psi_4 = \psi_1 - \sum q_i \left(1 - \frac{T_0}{T_i}\right) + i_{eva} \)

The total energy of the air conditioning cycle is the sum of destruction of exergy of each component,

\[ \Delta \psi_t = \Delta \psi_c + \Delta \psi_{con} + \Delta \psi_{ct} + \Delta \psi_{eva} \]  

The total loss of exergy is:

\[ i_l = i_c + i_{con} + i_{ct} + i_{eva} \]  

We calculated the exergetic efficiency as:

\[ \eta_{Ex} = \frac{i_l}{w} \]  

5 RESULTS AND DISCUSSION

We present experimental values in table 1. Temperatures correspond to the average recorded values in the evaluation interval (12:00 to 18:00 h), and it was observed that the value remained constant in the reported months with minimal variation. The electrical power supplied to the compressor and fans was 1.42 kW with variations of less than 3%.

The operation time of the system (per day) was, on average, 4 hours, and 6 hours, for the months of March and April, respectively. The corresponding total electricity consumption was 5.60 kW-h, and
8.40 kW-h. The low-temperature airflow to the room (8.5 °C and 0.122 kg/s) came from the evaporator output and presented almost no variation.

The mass flow of hot air at the condenser outlet was 0.175 kg/s with an average temperature of 48.5 °C, this heat flow is the one discharged into the environment and can be used for some type of application, for example, drying medicinal and aromatic herbs, fruits; in such case the device would serve as an air conditioning unit and as a heat pump. The read values were constant. In order to calculate other parameters, we used the average temperature in the two months, the estimated error was less than 1.15 %. As observed, the longest working time of the device occurred in April (6.0 h).

Using the temperature values, we plotted the thermodynamic diagram for R410A coolant, considering that the work of the compressor was isentropic, and that the behavior of the capillary tube was isoenthalpic, as shown in figure 3. Using the EES software, we obtained enthalpy and entropy values in the four states of the thermodynamic cycle (table 2). We also obtained the values of enthalpy and entropy for refrigerant at environmental conditions of T = 29.8 °C and P = 101 kPa

<table>
<thead>
<tr>
<th>Table 1. Thermodynamic cycle temperatures</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature Entrance (°C)</td>
</tr>
<tr>
<td>Compressor</td>
</tr>
<tr>
<td>Condenser</td>
</tr>
<tr>
<td>Capillary tube</td>
</tr>
<tr>
<td>Evaporator</td>
</tr>
<tr>
<td>Power (kW)</td>
</tr>
<tr>
<td>Operation time (h)</td>
</tr>
<tr>
<td>Electric consumption (kWh)</td>
</tr>
</tbody>
</table>

Source: Own creation

Figure 3. Thermodynamic cycle diagram
The values obtained from the thermodynamic diagram of the coolant as compared to those provided by the EES software are the same (< 0.03% difference).

Table 2. Enthalpy and entropy of the cycle

<table>
<thead>
<tr>
<th>Cycle point</th>
<th>Enthalpy (kJ/kg)</th>
<th>Entropy (kJ/kg K)</th>
<th>Exergy (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>421.7</td>
<td>1.8121</td>
<td>74.256</td>
</tr>
<tr>
<td>2</td>
<td>438.2</td>
<td>1.7910</td>
<td>92.826</td>
</tr>
<tr>
<td>3</td>
<td>247.0</td>
<td>1.1660</td>
<td>92.915</td>
</tr>
<tr>
<td>4</td>
<td>247.0</td>
<td>1.1637</td>
<td>91.019</td>
</tr>
</tbody>
</table>

Source: Own creation

We calculated the operating coefficient of the air conditioning system with the expressions, which represent the difference of enthalpies in the evaporator multiplied by the mass flow and divided by the power supplied to the compressor, its value is: COP = 6.5.

At $T_0$ ambient temperature, we obtained enthalpy and entropy values of 483.8 kJ/kg and 2.183 kJ/kg K. With the compressor operating velocity and the displaced volume, we obtained the mass flow of the coolant, which was 0.048 kg/s. The exergetic efficiency of the compressor was 1.269, that of the condenser was equal to 1.269, the capillary tube turned out to be 0.814, and that of the evaporator, 0.818. The average exergetic efficiency considering the four elements, was 4.3.

The energy from this equipment in the form of heat flow that can be taken advantage of is equal to $\dot{Q}_H = 3.26$ kW, which represents a significant amount that is currently wasted and can be applied to in some beneficial application, such as the drying of vegetables and medicinal herbs, within the same household premises that currently use this type of device.

6 CONCLUSIONS

We evaluated a domestic heat pump in order to take advantage of the warm air coming from the condenser and the cold side to keep a room at a temperature of 22 °C. The evaluation months were March and April of 2023, from the experimental analysis conducted we found that the monthly electrical power consumption was 5.60 kW-h, and 8.40 kW-h, respectively. The air flow supplied to the room was 0.122 kg / s at a temperature of 8.5 °C and on the hot side the air flow was 0.175 kg / s at a temperature of 48.5 °C, providing an approximate heat flow that will be used in the dryer to dehydrate some aromatic herb. The COP of the heat pump was 6.5 and the exergetic efficiency was 4.3. $\dot{Q}_H = 3.26$
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