Energy and exergy analysis of a portable air conditioner system.

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Abstract. In this analysis the main objective is to evaluate the thermodynamic behavior of a portable air conditioning system. The power is 4,103 kW. To achieve this, the device was implemented with pre-calibrated thermocouples, taking care that there was always a proper contact with the pipes in order to minimize the differences between the measurement and the actual value. The ESS software and the coolant thermodynamic diagram were used to determine the value of the properties at the considered points. The refrigerant used by this device is the R410A. Although the first law of thermodynamics can be used to assess the energy consumption, it makes no distinction between quantity and quality of energy. Energy analysis provides only the amount of supplied energy. Exergy analysis provides the maximum useful work that can be obtained by a system that is in balance with the environment. The COP (Coefficient of Performance), turned out to be 7.35, which is acceptable for this type of device. The exergetic efficiency was 4.17. Finally, the hot air flow coming from the condenser was 3.15 kW, this value is significant and will be used in some drying application.

1. Introduction
The Energy Information System, SENER, 2018, reports that in Mexico City there are, in small offices, (less than 60 m²-surface) and room houses, 7 million window type air conditioning units installed, with an average electricity consumption of up to 4 kW-h, considering that they operate a minimum of 8 hours a day for at least 260 days per year, [1]. For larger spaces, other types of air conditioning systems are used, for example: mini Split, multi Split and so-called "Chillers", which have the same thermodynamic cycle, but their operation is different. It is known that this type of devices discards an amount of energy, in the flow of air passing through the condenser and its final destination is precisely the environment, [2-3]. The first law of thermodynamics is used to assess energy consumption, it makes no distinction between the quantity and quality of it. Energy analysis provides only the amount supplied. The analysis of the exergy provides the maximum useful work that can be obtained a system that is in balance with the environment.

Some projects have been carried out in which commercial-type air conditioning systems similar to the one observed in this work, were evaluated for example: in 2002, an air conditioning system that uses the R 134A as a working fluid and the compressor is of the reciprocating type, in which the output pressure of the R 134A was reduced from a value of 2.0 Bar to 1.0 Bar with decreases of 0.2 Bar, was evaluated. Thermodynamic efficiency resulted around 52.00 %. This low efficiency is largely due to the compressor, since, reciprocating agents have the highest number of losses, the maximum COP is almost 4, [4]. In 2017 a study was performed on a unit with a reciprocating compressor, using different refrigerants, R22, R134A, R290, R410A, R407C. The maximum COP value was 5 and corresponded to R22. The exergetic efficiency at an evaporator temperature of 10 °C was 32.00% and also corresponded to R22, [5]. In 2009, an air conditioning system which uses a reciprocating compressor was evaluated with different coolant fluids, namely R125, R134A, R22 and R717, the working pressure was 1 Bar, the best COP value was 4.5 for the R314A, [6]. The objective of the present work was to evaluate a portable air conditioning unit, properly instrumented, to obtain COP and exergetic efficiency. The aim was to
calculate the amount of energy that is discarded into the environment and could be applied to beneficial uses, such as drying medicinal herbs.

2. Equipment description
The portable air conditioner system used in this work is of a widely-recognized trademark in the area of refrigeration and air conditioning. It 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 is the most environmentally friendly one, R410A. The device physical dimensions are 0.51 m front, 0.41 m wide and a height of 0.87 m. 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 consumed power of each fan is 800 W. The external view of the used unit is presented in Figure 1.

![Portable air conditioner system, (a) front, (b) back.](image)

In Figure 2, the front and back cover of the unit have been removed, and each component of the system can be observed. Located at the bottom, stands a rotary compressor of 1.13 kW input power, it has a R410A displacement of 11.4 cc/rev. Behind it the condenser coil is located and exactly and further behind stands the fan used to drive the air that will take the energy required for the coolant to condense. The evaporator with its respective fan is located at the right top and at the uppermost point, the cold air outlet. The drive motor power of both fans is 250 W.

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 cold air to the room and at the exit of the hot air. The energy of this last hot air flow is discarded into the environment, but further action should follow with it. A 407113 CFM model, Extech brand metal vane anemometer was used to measure the magnitude of both mass flows; with a measuring range of 0.5 to 126 m/s and an accuracy of 2 %. The PQ2071 model Extech brand meter, with an accuracy of 0.01% was used for the measurement of electrical power, voltage and amperage. The record of these values was done using an Arduino circuit with their immediate registration.

3. Experimental methodology
The portable air conditioner unit was installed in a 6 m wide -8 m long and 3 m high room, following the manufacturer's recommendations. The orientation of this room is south-north, with the main access door facing east, the unit's hot air discharge pipe goes through a west-facing window. Three
thermocouples were installed inside to know the average indoor temperature. Environmental conditions were also recorded during the equipment's testing period. The evaluation months were April, June and August, during the third week of each month, scheduled from 13:00 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 29.8 °C (41%), 28.3 °C (60%), and 27.1 °C (63 %), respectively.

The thermodynamic properties of the R410A refrigerant were obtained using the EES software and compared to the thermodynamic diagram in which the thermodynamic cycle was drawn, as depicted in Figure 4.

4. Theoretical analysis

The energy and exergy analysis is presented for the diagram of the thermodynamic cycle shown in Figure 3. The thermodynamic cycle of the air conditioning unit considers the four components: compressor, condenser, capillary tube and evaporator.

In order to simplify the analysis following assumptions were made:

- 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

![Figure 2.- Parts of the portable air conditioning system.](image)

![Figure 3. Flow diagram of the equipment for the portable air conditioner.](image)
the system is water at an environment temperature of 25 °C and P = 101 kPa; the evaporator capacity was considered 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, the equations of conservation of mass, energy and exergy [1,7] were applied. The mass flow conservation equation is:

\[ \sum m_i = \sum m_{out} \]  

where \( m \) is the mass flow of input and output, respectively. The energy conservation equation is:

\[ \dot{Q}_{in} + \dot{W}_{in} + \sum m_i \left( h + \frac{V^2}{2} + gz \right) = \dot{Q}_{out} + \dot{W}_{out} + \sum m_{out} \left( h + \frac{V^2}{2} + gz \right) \]  

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:

\[ \sum \left( 1 - \frac{T_0}{T_k} \right) * Q_k \ - \ \dot{W} + \sum m_i \Psi_{in} - \sum m_{out} \Psi_{out} = \dot{E}_d \]  

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 \( \dot{E}_d \) is the destroyed exergy flow. The specific discharge of the refrigerant or air is:

\[ \Psi_{fr} = (h - h_0) - T_0(s - s_0) \]  

in which \( s \) is the entropy of the refrigerant or the air. Using the notation in Figure 3, with the conditions indicated above, for the compressor,

\[ q + W_{in} = h_2 - h_1 \]  

for the condenser:

\[ q_{co} = h_3 - h_2 \]  

For the capillary tube:

\[ h_3 = h_4 \]  

And, for the evaporator

\[ q_{ev} = h_1 - h_4 \]  

Finally, the COP is determined as:

\[ COP = \frac{q_r}{\dot{W}} \]  

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

\[ \dot{W} = \sum m_{out} \dot{m} b - \sum m_{in} \dot{m} b - \sum Q_j \left( 1 - \frac{T_0}{T_j} \right) + I_t \]
In which \( b = \left( h + \frac{v^2}{2} + gz - T_0 S \right) \), \( T_j \) is the temperature of each element, and \( I_t \) is total irreversibility. So, for each component:

For the compressor,

\[
\psi_1 + w_{in} = \psi_2 - \sum q_j \left( 1 - \frac{T_0}{T_i} \right) + i_c
\]  

(11)

for the condenser:

\[
\psi_2 = \psi_3 - \sum q_j \left( 1 - \frac{T_0}{T_i} \right) + i_{con}
\]  

(12)

For the capillary tube,

\[
\psi_3 = \psi_4 + i_{ct}
\]  

(13)

And, for the evaporator

\[
\psi_4 = \psi_1 - \sum q_j \left( 1 - \frac{T_0}{T_i} \right) + i_{eva}
\]  

(14)

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

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

(15)

The total loss of exergy is:

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

(16)

The exergetic efficiency is calculated as:

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

(17)

5. Results and discussion.

Experimental values are presented in Table 1. Temperatures correspond to the average recorded values in the evaluation interval (13:00 to 18:00 h), and it was observed that the value remained nearly 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 10 min, 3 hours and 40 min, and 3 hours and 25 min, for the months of April, June and August, respectively. The corresponding total electricity consumption was 5.95 kW•h, 5.25 kW•h, and 4.9 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 47.8 °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 practically constant. In order to calculate other parameters, the average temperature in the three months was used, the estimated error was less than 0.1 2%. As observed, the longest working time of the device occurred in in April (5.65 h).

Using the temperature values, the thermodynamic diagram was plotted 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 4. Using the EES software, enthalpy and entropy values were obtained in the four states of the thermodynamic cycle (Table 2). The values of enthalpy and entropy for refrigerant were obtained at environmental conditions of \( T = 29.8 \) °C and \( P = 101 \) kPa
Table 1. Thermodynamic cycle temperatures

| Temperature (°C) Entrance: | April | June | August |
|---------------------------|-------|------|--------|
| Compressor                | 4.48  | 4.42 | 4.45   |
| Condenser                 | 39.71 | 39.80| 39.7   |
| Capillary tube            | 28.57 | 28.61| 28.57  |
| Evaporator                | 4.64  | 4.62 | 4.65   |
| Power (kW)                | 1.42  | 1.421| 1.419  |
| Operation time (h)        | 4.16  | 3.66 | 3.41   |
| Electric consumption (kWh)| 5.95  | 5.25 | 4.9    |

The values obtained from the thermodynamic diagram of the coolant as compared to those provided by the EES software are almost the same (< 0.01% difference).

Table 2. Enthalpy and entropy of the cycle.

| Cycle point | Enthalpy (kJ/kg) | Entropy (kJ/kg K) | Exergy (kJ/kg) |
|-------------|------------------|-------------------|---------------|
| 1           | 422.7            | 1.8021            | 74.236        |
| 2           | 441.2            | 1.8021            | 92.736        |
| 3           | 245.8            | 1.1560            | 92.975        |
| 4           | 245.8            | 1.1617            | 91.249        |

The operating coefficient of the air conditioning system was calculated with the aforementioned 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 = 7.35.

At $T_0$ ambient temperature, enthalpy and entropy values of 483.8 kJ/kg and 2,183 kJ/kg K are obtained. With the compressor operating velocity and the displaced volume, the mass flow of the coolant is obtained, which was 0.048 kg/s. The exergetic efficiency of the compressor was 1.269, that of the capacitor 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 4.17.

The energy from this equipment in the form of heat flow that can be taken advantage of is equal to $Q_H=3.15 \text{ kW}$, 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
A portable air conditioning unit has been fully implemented in order to thermodynamically evaluate it. To achieve this aim, a series of thermocouples were installed, duly calibrated and in total contact to the points of interest. Based on the thermodynamic diagram of the refrigerant and the EES software, the values of the required variables were obtained. The entire thermodynamic cycle was considered under permanent flow conditions; pressure losses in the condenser and evaporator were neglected, as well as in pipes.

The COP of the air conditioning unit, 7.35, is quite acceptable, although it was less than that reported by the manufacturer. The exergetic efficiency of the cycle yielded a value of 4.17. The high heat flow, 3.15 kW, is within the acceptable range to be used in the drying of vegetables and medicinal herbs at household scale.

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