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STUDY OF THE THERMODYNAMIC PERFORMANCE OF A REFRIGERATION SYSTEM WITH VARIABLE ROTATION COMPRESSOR AND ELETRONIC EXPANSION VALVE

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Abstract. *This present paper analyzes, experimentally, the thermodynamic behavior of a vapor compression refrigeration system. The experimental workbench consists of two units, one main and one secondary. In the primary, the components of a real refrigeration cycle, a variable rotation compressor, a condenser, an electronic expansion valve (EEV), an evaporator and a refrigerant fluid R134a are available. Already in the secondary unit, with the aid of a motor pump runs a mixture of water and ethyl alcohol, exchanging heat with the primary unit to guarantee the thermal comfort of the environment. For evaluation, parameters such as evaporation pressure, condensation pressure, mass flow, thermal load, power consumption and coefficient of performance (COP) are observed. Therefore, with the experiments, it was observed that with an increase in the rotation of the compressor, there was an increase in the thermal capacity and, consequently, an improvement in the COP of the system.*

Keywords: *Refrigeration, Variable speed compressor, Eletronic expansion valve, COP.*

1. INTRODUCTION

The use of refrigeration, in general, plays an important role in the quality of life of society. Such evolutions can be seen in daily life, such as in air conditioning, for the purpose of thermal control of the environment, in food preservation, to increase the shelf life of perishable foods at ambient temperatures, in the pharmaceutical industry, for the adequate transportation of vaccines and even in the oil industry, for its refining. However, in all this evolution there are environmental disadvantages, since refrigeration systems require large consumption of electricity to be able to operate and the main forms of energy generation used by several countries are by hydroelectric or thermoelectric.

By linking these two factors, studies for the technical development of the actual refrigeration cycles are of great importance in order to obtain better thermal capacity in detriment to low energy consumption and, consequently, a reduction of the operating costs in the operation of the system.

In order to investigate the interaction between the speed of rotation of the compressor in several compressors, Tassou and Qureshi (1994), compared performances of several compressors, among them the semi-hermetic reciprocating, rotary vane compressor and open-type reciprocating, all with refrigerating capacity nominal of 25 kw. Of all the comparisons made, only reciprocating open-type showed an increase in COP with the reduction of speed. In terms of the power consumed, the system showed a reduction of 12% to 24% in power consumption when operated with open-type compressor compared to the conventional fixed-rotation compressor.

In another comparison between energy performances between variable speed and fixed rotation compressors, Lida *et al.* (1982) found a reduction in power consumption, in the order of 20% to 26%, when the system works with a compressor that varies its speed from 750 rpm to 2250 rpm in relation to a fixed-speed compressor of the same rated cooling capacity.

Nobrega (2015) conducted experimental tests on a refrigeration system with a variable speed compressor and a thermostatic expansion valve. Their research has the objective to compare the operating system COPs with variations in the rotation of the compressor with respect to a fixed speed. It was observed that the system COPs are larger when variations are made in the speed of the compressor for the same thermal load, when compared to fixing a rotation of the compressor. He also demonstrated the importance of using variable speed compressors in systems that require variations in the thermal load, making the system more energy efficient.

Regarding the use of electronic expansion valves, Yasuda (1994) developed a dynamic modeling to simulate the control of the valve by proportional integral derivative (PID) command. The main objective of his research is to evaluate the responses a modeling compressor in relation to the experiments. To perform the tests, it used inverter-drive compressor. As a result, it obtained good agreement between the results of the simulation and the experimental data.

Aprea and Mastrulo (2002) carried out a study to optimize the use in steam compression refrigeration systems. It utilized a fixed rotary compressor operating at a partial load, controlled by a thermostat compared to a variable speed compressor operating continuously by varying its speed depending on the required thermal load. it was verified that a system of variation in the rotation of the compressor can be up to 25% more efficient than systems that operate with fixed speed, whereas systems that operate by thermostatic control can be up to 15% more economical.

Still in the use of a PID controller, Outtagarts *et al.* (1997) evaluated the mass flow change as a function of the evaporator design. For each evaporator model, it used superheat values and mass flow rates of different refrigerants to define the best parameters of each of these evaporators.

Costa (2014) proposed a comparison between expansion devices. For the experiments, it was used a capillary tube and an electronic expansion valve (EEV), analyzing parameters such as thermal load, power consumption and COP. For all operating conditions, the system proved to be more efficient operating with electronic expansion valve.

Lago (2016) conducted experiments using a variable speed compressor combined with an EEV and compared to another system using variable speed compressor and a thermostatic expansion valve (TEV). For low thermal loads, the first system presented a power consumed 7% lower than the second, already for larger thermal loads, the power had very similar results. With regard to the COP of the system, the formerly obtained values slightly higher than the second.

2. EXPERIMENTAL PROCEDURE

For the analysis of the experimental study, the bench was instrumented with temperature, pressure and flow sensors, as well as meters of power consumed and a wave generator to vary the rotation of the compressor. In order to collect this data, an acquisition module is used that displays the results graphically on the screen of a computer.

Figure 1 represents a diagram of the experimental bench, where it is possible to observe the primary and secondary units of the system. The R134a refrigerant circulates in the primary unit, and the equipment comprising this unit is a variable-speed compressor, a forced-air condenser, an electronic expansion valve and an evaporator. The other unit consists of a pump, which has the function of circulating the ethyl alcohol as secondary fluid and a fancoil, which has the function of providing a thermal comfort for a given environment.

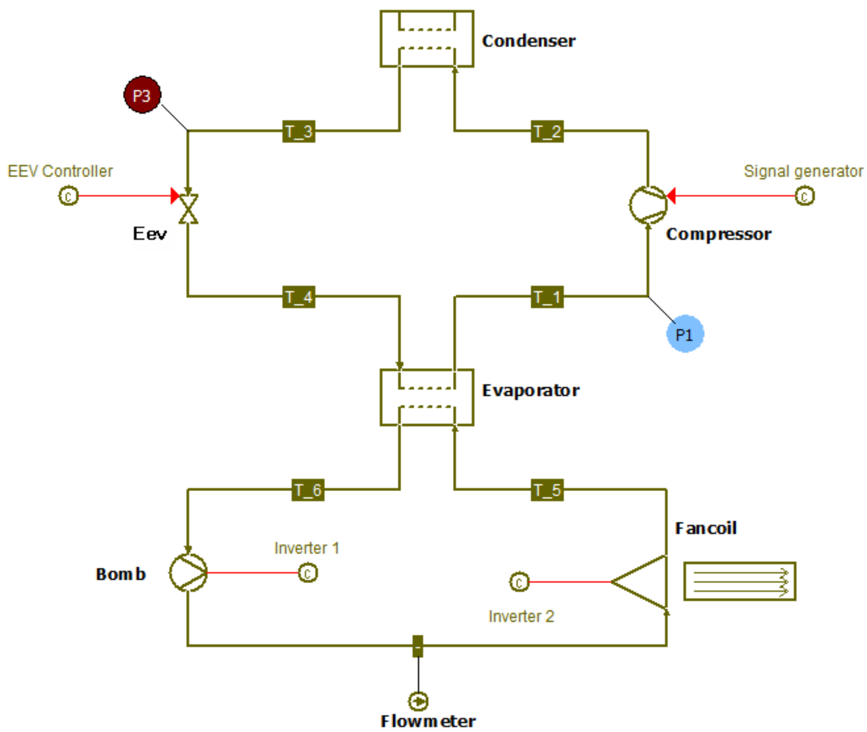

Figure 1. Experimental workbench

The primary unit provides the thermal load through the heat transfer by the counterflow flow evaporator. Once this heat transfer takes place, the secondary fluid passing through that evaporator lowers its temperature and conducts that thermal load to the fancoil, where it will cool a given environment.

Once the refrigeration cycle of the experimental stand is understood and the data collection points of the system defined, the next phase will be to define the hypotheses allowed for the control volume in each equipment in the refrigeration system. This phase is important for defining the performance calculations, which will be displayed in the next section. Table 1 shows the assumptions for each equipment in the refrigeration set. A common consideration among them is the permanent regime criterion, where the data are collected after the stabilization of the properties.

Table 1. Assumptions

Control volume	**Assumption**
Compressor	Adiabatic
Condenser	Does not do work
Expansion Valve	Adiabatic and does not do work
Evaporator	Does not do work

Based on the general energy equation (Eq. 1), hypotheses are applied for each of the equipments present in the refrigeration cycle, however adopting the permanent regime criterion for the whole system, as well as the disregard of kinetic energies and power cycle, the energy equation for control volume is reduced to Eq. (2)

$$\frac{\mathrm{d}E_{cv}}{\mathrm{d}t} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_{i}^{i} \dot{m}_i(h_i + \frac{V_i^2}{2} + gz_i) - \sum_{o}^{i} \dot{m}_o(h_o + \frac{V_o^2}{2} + gz_o) \tag{1}$$

where $\dot{Q}$ is the heat exchange rate between environments, $\dot{W}$ is the work, $\dot{m}$ is the mass flow at the input and output of the control volume, V represents the kinetic energy, z is the potential energy and g is the gravitational acceleration

$$0 = \dot{Q_{cv}} - \dot{W}_{cv} + \sum_{i}^{i} \dot{m}_i h_i - \sum_{o}^{i} \dot{m}_o h_o \tag{2}$$

Assuming, also, the same mass flow rate for the whole system, it is possible to obtain:

$$\dot{W}_{cv} = \dot{Q}_{cv} + \dot{m}(h_i - h_o) \tag{3}$$

Based on this equation and with the hypotheses allowed in the Tab. 1 the thermal load of the evaporator is defined as:

$$\dot{Q}_{evaporator} = \dot{m}_{refrigerant}(h_1 - h_4) \tag{4}$$

The thermal load of the secondary fluid passing through the evaporator can be defined as:

$$\dot{Q}_{sf} = \dot{m}_{sf}(h_o - h_i) \tag{5}$$

But,

$$h_o - h_i = C_p(T_6 - T_5) \tag{6}$$

Therefore, its final equation is defined:

$$\dot{Q}_{sf} = \dot{m}_{sf} C_p(T_6 - T_5) \tag{7}$$

The mass flow rate of the refrigerant is obtained by the equality between the amount of heat exchanged by the evaporator, as in Eq. 8.

$$\dot{Q}_{evaporator} = \dot{Q}_{refrigerant} = -\dot{Q}_{sf} \tag{8}$$

Thus, by combining the Eq. 4 and Eq. 7, it is concluded that the mass flow is a direct function of the specific heat, the temperature variation and the mass flow of the secondary and indirect fluid of the enthalpy change of the coolant passing through the evaporator, as in Eq. 9.

$$\dot{m}_{refrigerant} = \frac{\dot{m}_{sf} c_p \left(T_5 - T_6\right)}{h_1 - h_4} \tag{9}$$

Since the COP is a division between the desired effect by the energy consumed for such effect, as in Eq. 10.

$$COP = \frac{\dot{Q}_{evaporador}}{\dot{W}_{real}} \tag{10}$$

But by combining the Eq. 7, Eq. 8 with Eq. 10, the COP comes down to:

$$COP = \frac{\dot{m}_{fs} C_p(T_5 - T_6)}{\dot{W}_{real}} \tag{11}$$

3. RESULTS AND DISCUSSION

The results presented here were obtained in experiments of a variable speed compressor, manipulating its rotation in the range of 1600 rpm at 4500 rpm and an electronic expansion device, in which an overheating value of seven degrees Celsius ($7\,^{\circ}\mathrm{C}$). The pump of the secondary set was controlled so that its mass flow of alcohol was $0.087\ kg/s$. The fancoil was controlled so that the evaporation pressure was always 3 bar, as explained in figure 2.

Still in figure 2, it is possible to observe an increase in the condensation pressure of the fluid as the frequency of rotation of the compressor is also increased, this increase corresponds to a variation of 2% to 18%.

The evaporation and condensation temperatures of the system, as specified in Fig. 3, have the same performance as those of the pressure plot (Fig. 2). This is due to the uses of these pressures to acquire the corresponding temperature, by the saturation data of each pressure.

Figure 4 demonstrates the advantage in the use of variable speed compressors in steam compression refrigeration systems. An increase in the mass flow rate of the refrigerant can be observed as increments are made in the speed of the compressor. This increase is in the order of 2% for flow rates of 1800 rpm, up to 40% at 4500 rpm rotation.

Since the variable-speed compressor has the capacity to vary the mass flow rate of the refrigerant, the direct consequence of this action is the variation of the thermal load of the system, as shown in Fig. 5. It is possible to observe an increase in the thermal load that varies from 5% to 38%.

Another important parameter analyzed when making variations in the rotation of the compressor is the power consumption. In this way, Fig. 6 shows an increase in the power consumed by the compressor as the speed of the compressor is increased, in the range of 2% to 18%. This increase is explained by the increase in the system pressure ratio.

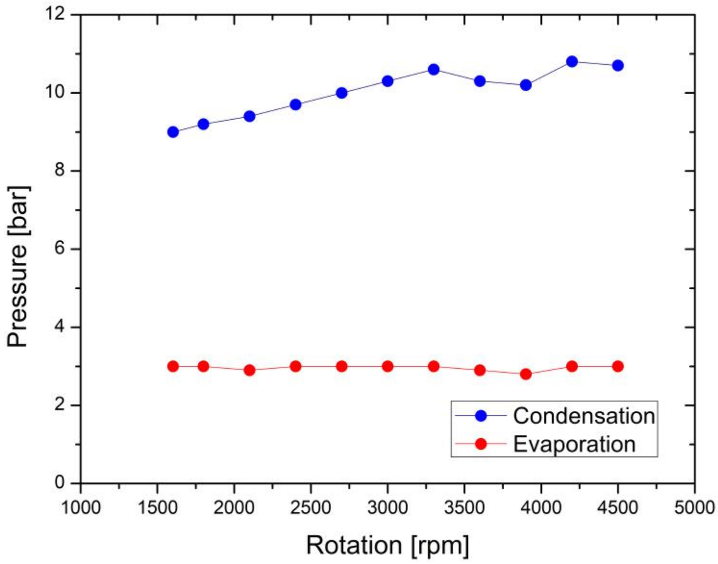

Figure 2. Evaporation and condensation pressure as a function of compressor speed

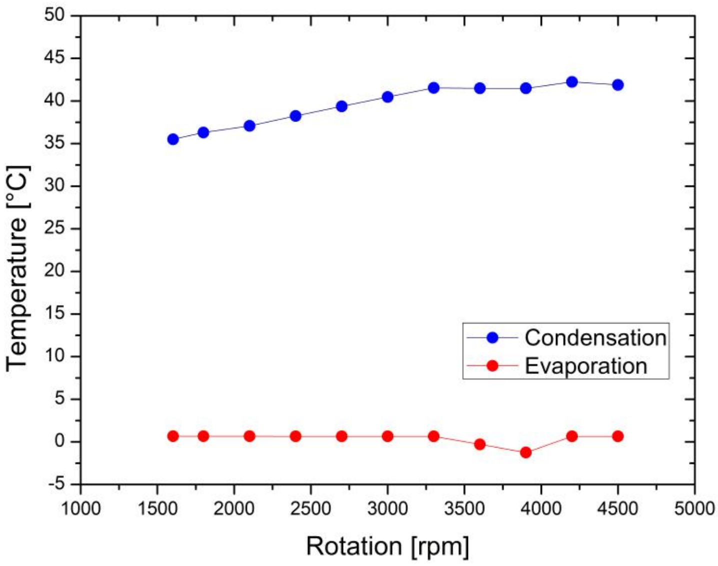

Figure 3. Evaporation and condensation temperature as a function of compressor speed

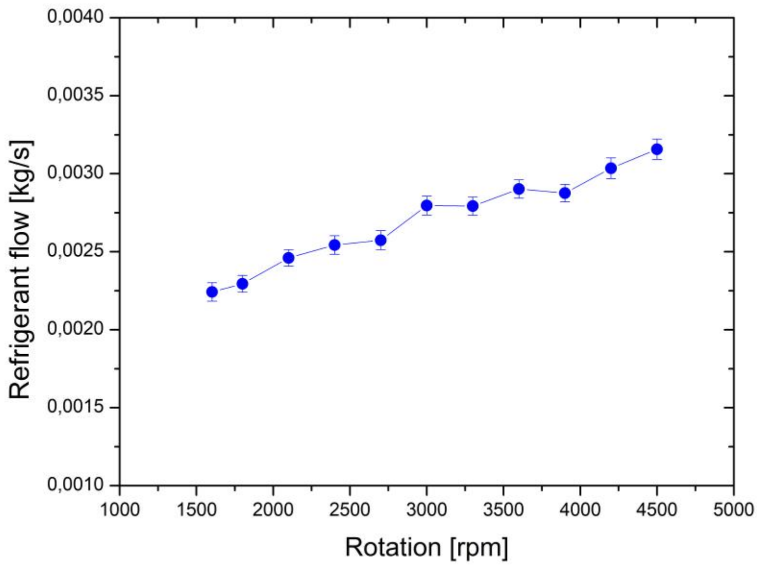

Figure 4. Refrigerant flow rate as a function of compressor speed

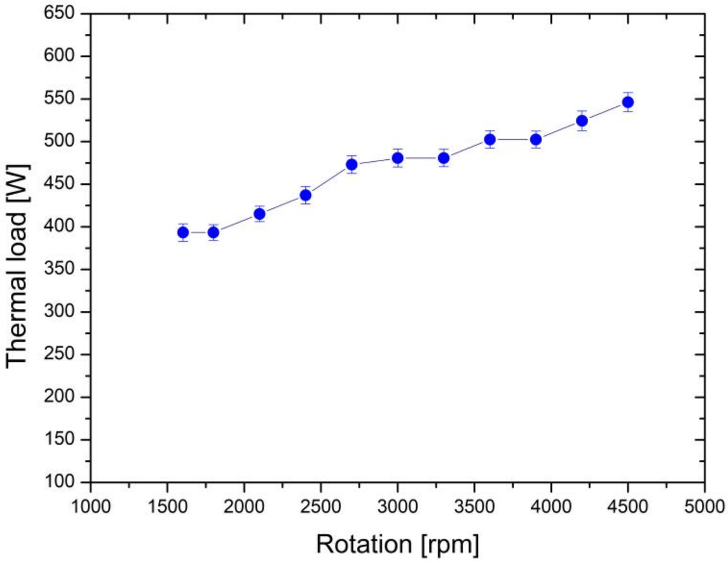

Figure 5. Thermal load as a function of compressor speed

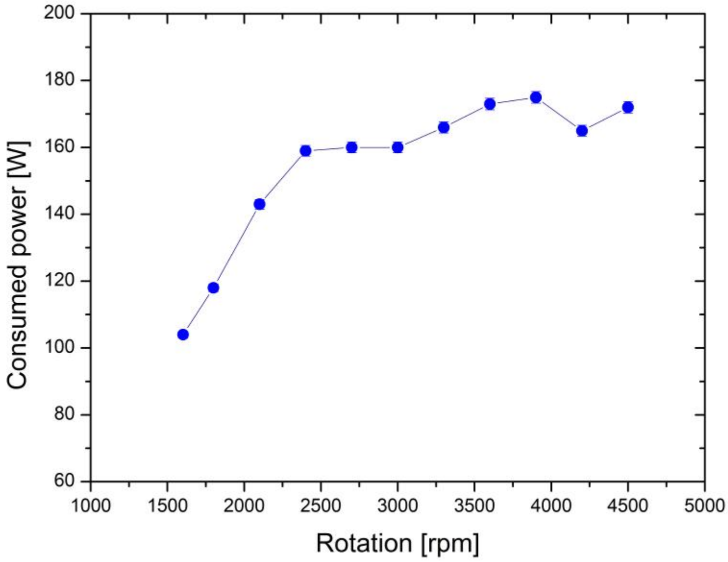

Figure 6. Consumed power as a function of compressor speed

In Figure 7, it is possible to observe that for compressor speeds of 1600 rpm and 1800 rpm, COP has higher values due to lower condensing temperature, lower power values and lower compression rates. With the increase of the rotation, from 2100 rpm to 3900 rpm, the condensation pressure and the power consumed also increase, which results in lower efficiency values. From this rotation, the COP increases due to higher values and thermal load and the stabilization of the power consumed.

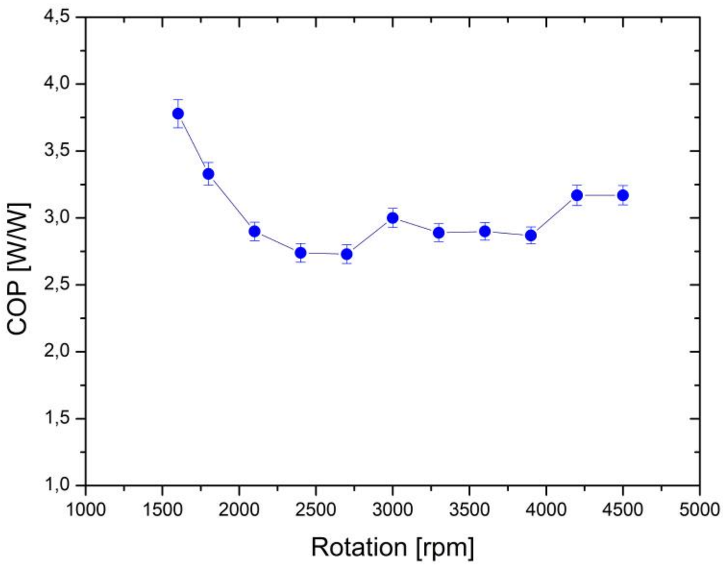

Figure 7. COP as a function of compressor speed

4. CONCLUSION

This experimental work examined the influence of the use of variable rotation compressor and EEV in refrigeration systems. To perform the experiments it was necessary to determine specific operational parameters, such as the fixing of a mass flow rate of the secondary fluid and the evaporation pressure of the system. By this, it is possible to observe that with the control in the frequency of work of the compressor, the thermal load had a variation of up to 38%. Already the power consumption, had an increase of up to 18%, as the value of the compressor frequency was added. Using the EEV, it was possible to observe constant values in the actual superheating of the system. Finally, it is possible to conclude that systems that use a variable speed compressor combined with an EEV are effective in terms of control depending on the thermal capacity.

5. ACKNOWLEDGEMENTS

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