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EXPERIMENTAL STUDY OF A VAPOR COMPRESSION CYCLE PERFORMANCE BEHAVIOUR

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Abstract. *Vapor compression is the main way to obtain refrigerant effect, and the most accurate knowledge is needed to design a machine that has optimized energetic efficiency. In this work, a study about the thermal performance of a water-water refrigeration machine operating with R134a was performed. Three tests were done, where the hot source temperature and the cold source temperature were varied, and the corresponding variations in fluid mass flow, coefficient of performance (COP) and heat exchanges in the evaporator were evaluated. When raising the temperature of the cold source, there was an increase in COP, refrigeration capacity and refrigerant mass flow. When the hot source temperature was increased, it was observed the reduction of COP, refrigeration capacity and refrigerant mass flow.*

Keywords: *Vapor compression, refrigeration machine, energetic efficiency.*

1. INTRODUCTION

The operation of refrigeration machines involves the analysis of different physical and thermodynamic variables, such as compressor discharge temperature and pressure, refrigerant mass flow rate and hot and cold source temperatures. These variables directly influence technical parameters that are of great importance to the engineer, such as compressor consumption, refrigeration rate and coefficient of performance.

The experimental study used a refrigeration machine with R134a as the working fluid and water as the secondary fluid, present in the surroundings of the evaporator and the condenser. In this way, it is a water-water machine. The secondary fluid is divided in two stages of temperature in each of the heat exchangers. There are two sources of water, hot water and cold water, which are controlled by valves that allow manipulating the temperature of each of the heat exchangers.

For the design of a refrigeration system it is important to know the influence of the variation of each variable over the others. The objective of the present work is to analyze the effect of the condensation temperature variation on the compressor discharge temperature and the effect of the evaporation temperature variation on the mass flow. For this, three tests were carried out on the machine mentioned, generating three cooling cycles that represented the system operation.

2. TEST BENCH

The bench is a refrigerator equipped with temperature controllers, which operates according to the vapor compression cycle with the refrigerant R134a. Figure 1 shows an image of the machine.

Figure 1. View of the refrigeration machine (Duarte, 2014)

The bench has maximum cooling capacity of 3 kW and operates with water as secondary fluid in both the high pressure circuit and the low pressure circuit. Figure 2 shows the circuit of R134a where vapor compression is performed by the machine, with its main components.
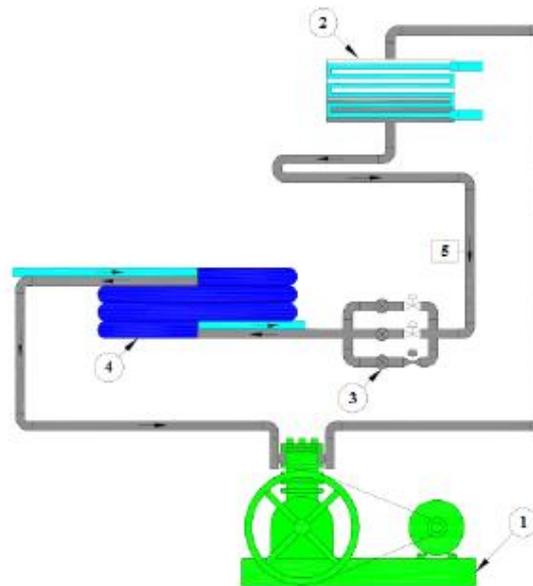

Figure 2. Vapor compression circuit and main components. 1 - Compressor, 2 - Condenser, 3 - Expansion valves, 4 - Evaporator, 5 – Flowmeter (Maia, 2005)

The water condenser is of the shell and tubes type (refrigerant on the side of the shell and water inside the pipes) and can dissipate up to 6 kW of heat. The condenser water circuit, shown in Fig. 3, is composed of two water reservoirs, one with hot water and the other with cold water. The temperature of the water pumped into the condenser inlet is adjusted by mixing the waters of the two tanks through the manual opening of the valves at each reservoir outlet.
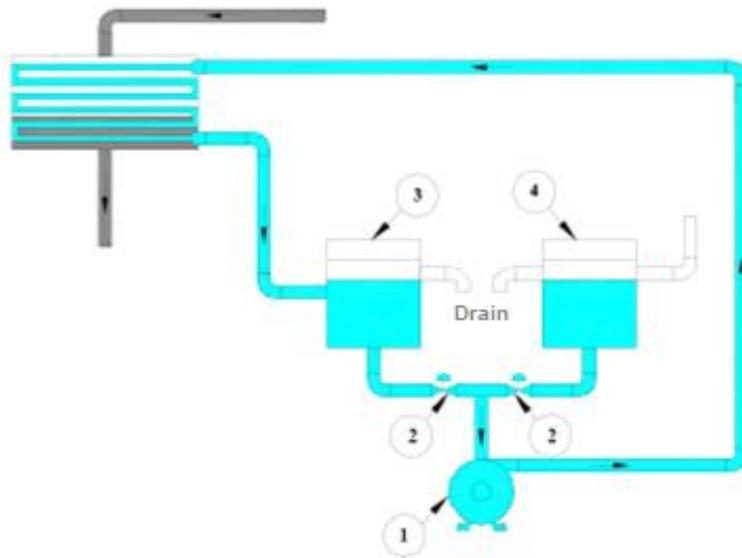
Figure 3. Condenser water circuit. 1 - Pump, 2 - Valves, 3 - Hot water reservoir, 4 - Cold water reservoir (Maia, 2005)

The system is endowed with three expansion devices, which are put in operation using manual registers mounted downstream of them. One is a needle-type manual expansion valve, the other is an electronic valve and the latter is a thermostatic valve. The evaporator consists of a tube with three inner copper tubes. The refrigerant flows through the inner tubes and the water circulates in the countercurrent outer space. As the secondary fluid, more precisely a mixture composed of 20% by volume of ethylene glycol and 80% of water was used. The cooling water circuit, shown in Fig. 4, is provided with a heating resistor of 6 kW. The water temperature is controlled by a PID controller (Proportional Integral and Derivative), which acts on the resistor, controlling the firing of a set of thyristors. The temperature of the water is measured by a thermoresistor.
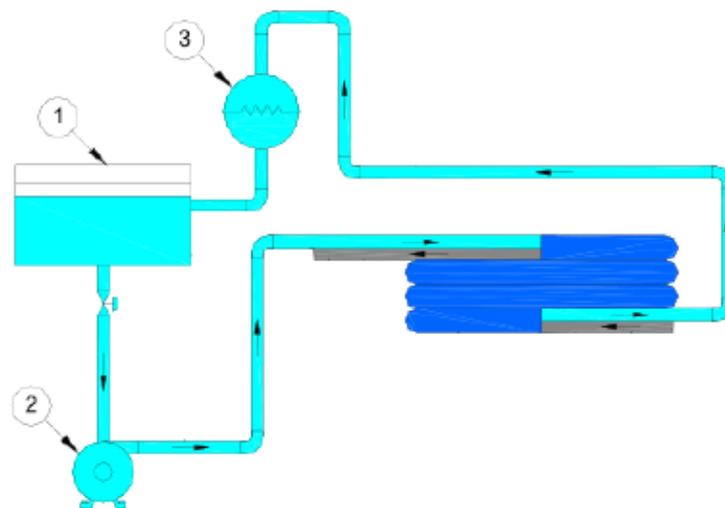

Figure 4. Evaporator water circuit. 1- Water reservoir, 2 – Pump, 3 - Electric resistor (Maia, 2005)

The bench has a reciprocating compressor equipped with two compression chambers. The diameter of the pistons is 50 mm and the stroke is 40 mm, corresponding to a displacement of 157 cm³. The maximum cooling capacity is 7.56 kW. The compressor is driven through a system of belts and pulleys by an asynchronous electric motor, three-phase 1720 rpm and 3 hp. The motor is driven by a frequency inverter that allows the rotation speed variation. A photoelectric velocity measuring system, mounted to the compressor pulley, provides, with an accuracy of 5 rpm, the compressor rotation speed. To perform the experimental tests, T-type thermocouples and Bourdon pressure gauges were installed at the compressor inlet and outlet. The machine also has a Coriolis flow meter (Duarte, 2014).

3. VAPOR COMPRESSION REFRIGERATION SYSTEM

According to Sonntag and Borgnakke (2003), the refrigeration cycle by mechanical vapor compression, used in refrigeration systems (heat pumps) is composed of four basic components: condenser, evaporator, expansion device and compressor. In this cycle, a working fluid (refrigerant) is subjected to a thermodynamic cycle consisting of condensation, expansion, evaporation and compression processes. Figure 5-a presents the scheme and 5-b the graph T vs s (Temperature vs. Entropy) of a vapor compression ideal cycle.
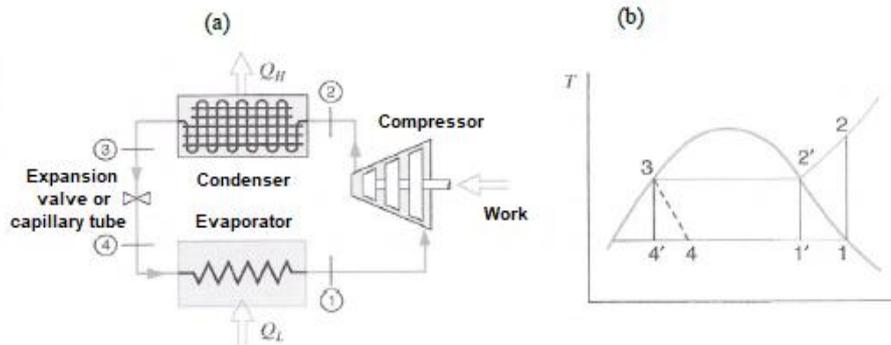

Figure 5. (a) Scheme of an ideal refrigeration system. (b) Chart of temperature versus entropy of the vapor compression refrigeration ideal cycle (Sonntag and Borgnakke, 2003)

As shown in the graph of Fig. 5-b, the cycle is subdivided into four processes, and in process 1-2 the saturated low pressure steam enters the compressor and is subjected to reversible adiabatic compression. Process 2-3, carried out in the condenser, is characterized by the heat exchange at constant pressure of which the fluid exits as saturated liquid. The process 3-4 is characterized by being an adiabatic strangulation process, which can be performed by an expansion device or by capillary tube. The purpose of this step is to reduce fluid pressure at the condenser outlet (also known as high pressure side) to the required pressure in the evaporator (also known as low pressure side). In process 4-1, carried out in the evaporator, the working fluid is boiled at constant pressure. After this step, the fluid returns to the compressor, and the cycle repeats. The chart in Fig. 5-b outlines besides the optimal cycle of refrigeration (1-2-3-4-1), the Carnot cycle (1'-2'-3-4'-1'). In the latter, the working fluid always remains within the biphasic (liquid-vapor) region. However, the actual compressors do not work in the biphasic region, that is, with mixtures of liquid and steam, they work only in the region of steam.

The above description refers to the ideal vapor compression refrigeration cycle. However, the real cycle, which effectively describes the operation of vapor compression refrigeration systems, differs from the ideal cycle as a function of the load losses of the working fluid flow and the heat transfers to the vicinity. Figure 6-a shows the schematic and 6-b the chart of Temperature vs Entropy (T vs s) of the vapor compression refrigeration real cycles. In theses cycles, the transformations are irreversible, the compression does not occur isentropically and the pressure losses in the heat exchangers (evaporator and condenser) are taken into account. In addition, in the real cycle there is overheating of the fluid at evaporator outlet and subcooling of the fluid at condenser outlet. Dashed lines 1-2 and 1-2 ', represent the compression process. In this step, there is irreversibility and heat transfer, that make the entropy vary, usually increasing.
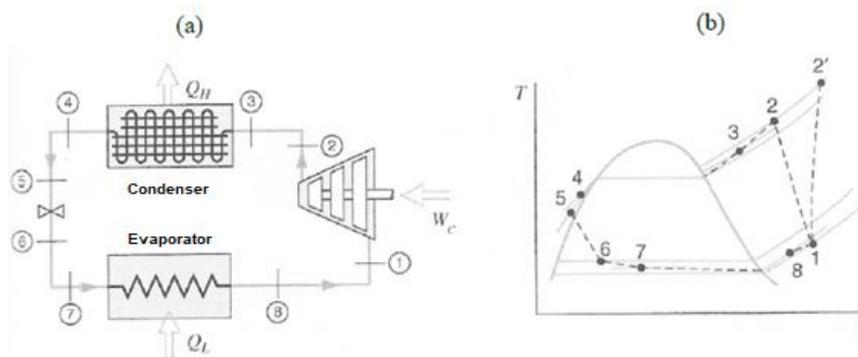

Figure 6. (a) Scheme of a real refrigeration system. (b) Chart of temperature versus entropy of the vapor compression refrigeration real cycle (Sonntag and Borgnakke, 2003)

4. METHODOLOGY

The refrigeration machine was tested for three conditions. The first was called the initial condition and served as the basis for the two subsequent tests. The second condition was obtained by heating the hot source (increasing the hot water flowing around the condenser), which raised the condensation temperature. The third and final condition was obtained by heating the cold source (increasing the hot water flowing around the evaporator), which raised the evaporation temperature. Along the tests, several parameters were monitored, and at the end, other parameters were consulted through thermodynamic properties or calculated.

A comparison of the cases is presented when the condensation temperature and the evaporation temperature are varied. Three complete measurements of parameters such as temperature (T), pressure (P), mass flow rate of the refrigerant fluid ($\dot{m}$) and ambient temperature (T_{amb}) were made to build the cooling cycles in the software EES (Engineering Equation Solver), in which the enthalpies were taken and the thermodynamic cycles were represented, what is fundamental for the analysis of this work.

The other parameters analyzed in the tests were enthalpies (h) of each cycle point, specific evaporation heat (q_{evap}), evaporation heat rate ($\dot{Q}_{evap}$), specific condensation heat (q_{cond}), condensation heat rate ($\dot{Q}_{cond}$), compressor specific work (w_{comp}), compressor work rate ($\dot{W}_{comp}$) and cooling machine performance coefficient (COP).

4.1 Modeling procedures

To perform the calculations of several parameters mentioned above it is necessary to present the equations beforehand. The specific energy exchanged in the evaporator, also identified as the specific evaporation heat q_{evap}, is calculated by Eq. (1).

$$q_{evap} = h_1 - h_5 \tag{1}$$

Where h is the fluid enthalpy. The index 1 refers to compressor inlet and evaporator outlet, while the index 5 refers to the expansion device outlet and evaporator inlet.

Once the evaporation heat has been defined, it is represented by $\dot{Q}_{evap}$ and calculated by Eq. (2).

$$\dot{Q}_{evap} = \dot{m} \cdot q_{evap} \tag{2}$$

The specific energy exchanged in the condenser, also identified as the specific condensation heat q_{cond}, is calculated by Eq. (3).

$$q_{cond} = h_2 - h_4 \tag{3}$$

Where index 2 refers to the compressor output and the condenser input, while the index 4 refers to the actual condenser output and expansion device input. The index 3, shown in the tables below, refers to the point at which the fluid is very close to the condenser outlet, in the saturated liquid curve.

Once the condensation heat has been defined, it is represented by $\dot{Q}_{cond}$ and calculated by Eq. (4).

$$\dot{Q}_{cond} = \dot{m} \cdot q_{cond} \tag{4}$$

The specific work of the compressor w_{comp} is defined according to Eq. (5).

$$w_{conp} = h_1 - h_2 \tag{5}$$

The work rate demanded by the compressor $\dot{W}_{comp}$ is defined according to Eq. (6).

$$\dot{W}_{comp} = \dot{m} \cdot w_{comp} \tag{6}$$

The COP coefficient, also known as the coefficient of performance, is defined according to Eq. (7).

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \tag{7}$$

5. RESULTS AND DISCUSSION

For the three conditions, data were measured in order to set up the refrigeration cycles. Table 1 shows the data collected in the tests.

Table 1. Quantities measured to build the thermodynamic cycles

Quantity	Unit	Case 1	Case 2	Case 3
T_1	°C	9.7	12	15
T_2	°C	87.8	84.9	85
T_3	°C	42.3	36.8	38.1
T_4	°C	41.5	36.8	37.9
T_5	°C	-9.4	-9.4	-4.2
$T_{ambient}$	°C	29.1	29.8	30.1
P_{low} (absolute)	kPa	190	180	210
P_{high} (absolute)	kPa	1650	1180	1270
$\dot{m}$	kg/h	39.1	41.0	52.2

Where T is temperature, P is pressure and s is the specific entropy. With the pressure and temperature data measured for the three cases, the thermodynamic properties were consulted using the EES software. Tables 2, 3 and 4 present the thermodynamic properties for cycles 1, 2 and 3, respectively.

Table 2. Thermodynamic properties for the cycle 1

	T (°C)	P (kPa)	h (kJ/kg)	s (kJ/kg.K)
1	9.7	190	261.6	1.004
2	87.8	1650	313.3	1.008
3	42.3	1081	111.7	0.4057
4	41.5	1081	110.5	0.4018
5	-9.4	205.5	110.5	0.4278

Table 3. Thermodynamic properties for the cycle 2

	T (°C)	P (kPa)	h (kJ/kg)	s (kJ/kg.K)
1	12	180	263.7	1.016
2	84.9	1180	316.9	1.041
3	36.8	932.6	103.5	0.3798
4	36.8	932.6	103.5	0.3798
5	-9.4	205.5	103.5	0.4013

Table 4. Thermodynamic properties for the cycle 3

	T (°C)	P (kPa)	h (kJ/kg)	s (kJ/kg.K)
1	15	210	265.7	1.011
2	85	1270	315.8	1.033
3	38.1	966.3	105.4	0.3859
4	37.9	966.3	105.1	0.385
5	-4.2	251	105.1	0.4028

Obtaining the pressure and temperature for the tests, the cycles were built on the Pressure (P) versus Enthalpy (h) graph, also using the EES software. Figure 7 shows the three cycles, partially overlapping.
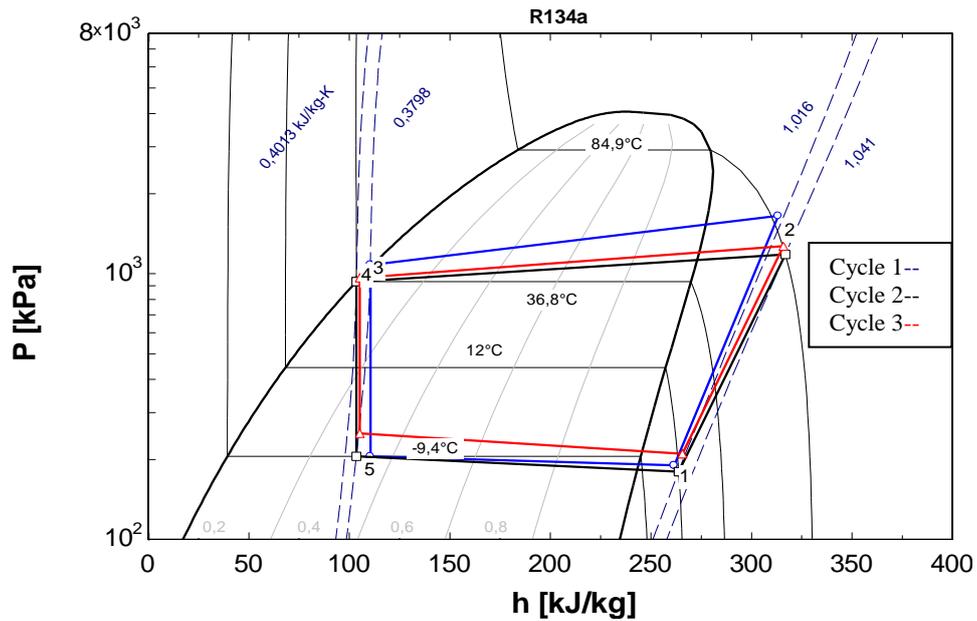
Figure 7. Graphical representation of the three cycles (cycle 1 in blue line, cycle 2 in black line and cycle 4 in red line)

Analyzing the cycles presented in Fig. 7, it is noted that the reference cycle adopted was cycle 2 (black line). Thus cycle 1 (blue line) suffered significant increase of the condensation temperature, and a slight variation of the evaporation temperature. Cycle 3 (red line) underwent significant increase of the evaporation temperature, and a slight variation of the condensation temperature. These small variations in temperature are due to the difficult control of the the refrigeration machine stabilization and the time when the tests have begun. These effects were neglected in the analysis below.

To analyze the effect of the condensation temperature variation on the compressor discharge temperature, the hot source temperature was increased. Figure 8 graphically shows the effect of this heating.
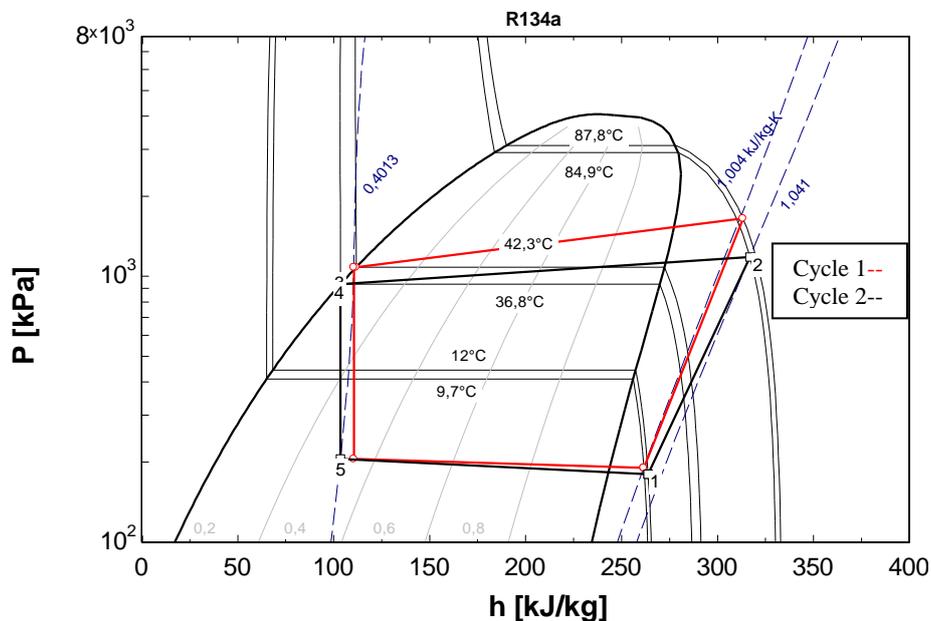

Figure 8. Graphical representation of the cycle with condensation temperature increase (cycle 1 – red line) and the reference cycle (cycle 2 – black line)

To analyze the effect of the evaporation temperature variation on the refrigerant mass flow rate, the temperature of the cold source was increased. The graphic of Fig. 9 shows the effect of this heating.
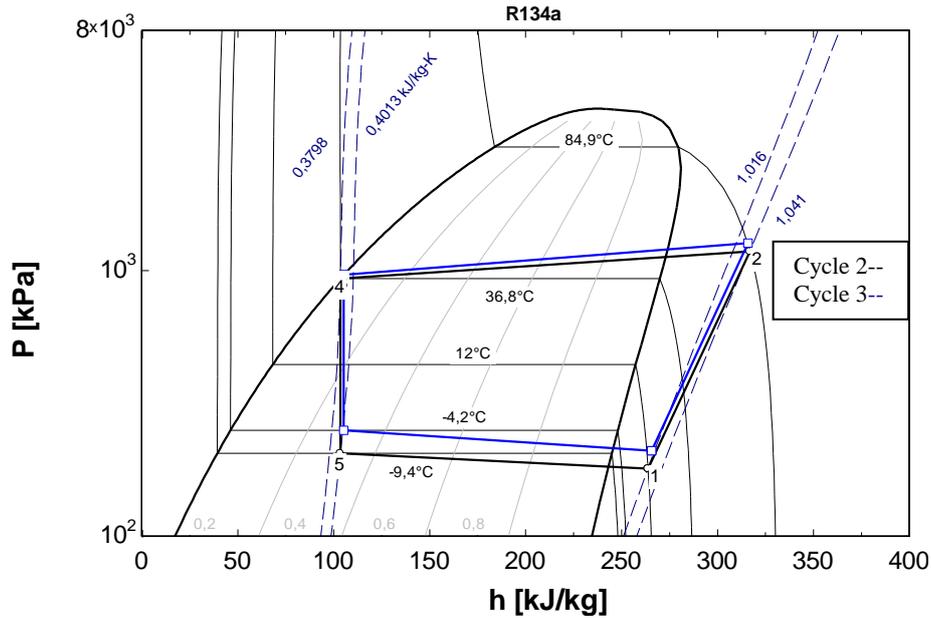
Figure 9. Graphical representation of the cycle with evaporation temperature increase (cycle 3 – blue line) and the reference cycle (cycle 2 – black line)

The mass flow rate, heat transfer rate in the evaporator and condenser, compressor consumption rate and the coefficient of performance were calculated, except the measured mass flow rate, and were organized in Tab. 5.

Table 5. Measured and calculated quantities for the three cycles.
Indexes 1, 2 and 3 for cycles 1, 2 and 3, respectively

Quantities	Units	Values
$\dot{m}_1$	kg/s	0.01086
$\dot{m}_2$	kg/s	0.01139
$\dot{m}_3$	kg/s	0.01450
$\dot{Q}_{evap-1}$	kW	1.641
$\dot{Q}_{evap-2}$	kW	1.825
$\dot{Q}_{evap-3}$	kW	2.327
$\dot{Q}_{cond-1}$	kW	2.189
$\dot{Q}_{cond-2}$	kW	2.389
$\dot{Q}_{cond-3}$	kW	3.013
$\dot{W}_{comp-1}$	kW	0.5616
$\dot{W}_{comp-2}$	kW	0.6055
$\dot{W}_{comp-3}$	kW	0.7264
COP_1	-	2.921
COP_2	-	3.014
COP_3	-	3.204

Analyzing all measured and calculated data, it was possible to identify the influence of evaporation and condensation temperatures on the machine's operation.

When increasing the condensation temperature of the machine (cycle 1), the compressor discharge temperature increased as well as the high pressure. There was also an increase in the condenser loss of charge. Regarding the heat exchange rates, the system had a reduction in the compressor demand and in the heat rates in the condenser and evaporator, due to the mass flow rate decrease. Thus, the COP of the refrigeration machine and the cooling capacity decreased.

When increasing the evaporation temperature of the machine (cycle 3), there was a significant increase in the fluid mass flow rate. The cooling capacity has increased significantly. This is a direct consequence of the fact that the cold source is at a temperature closer to the hot source, which contributes to the increase in COP. There was an increase in the compressor consumption due to the increase in mass flow rate. As a negative point, the increase of the minimum

working temperature stands out, that is, the lowest temperature that the refrigerated environment can reach becomes less.

Cycle 2 was maintained as an intermediate cycle in terms of performance, cooling capacity and mass flow rate. Cycle 1 had the worst results and cycle 3 had the best results. The graph of Fig. 10 shows the comparison between the mass flow rate and the cooling capacity of the machine for the three tests.
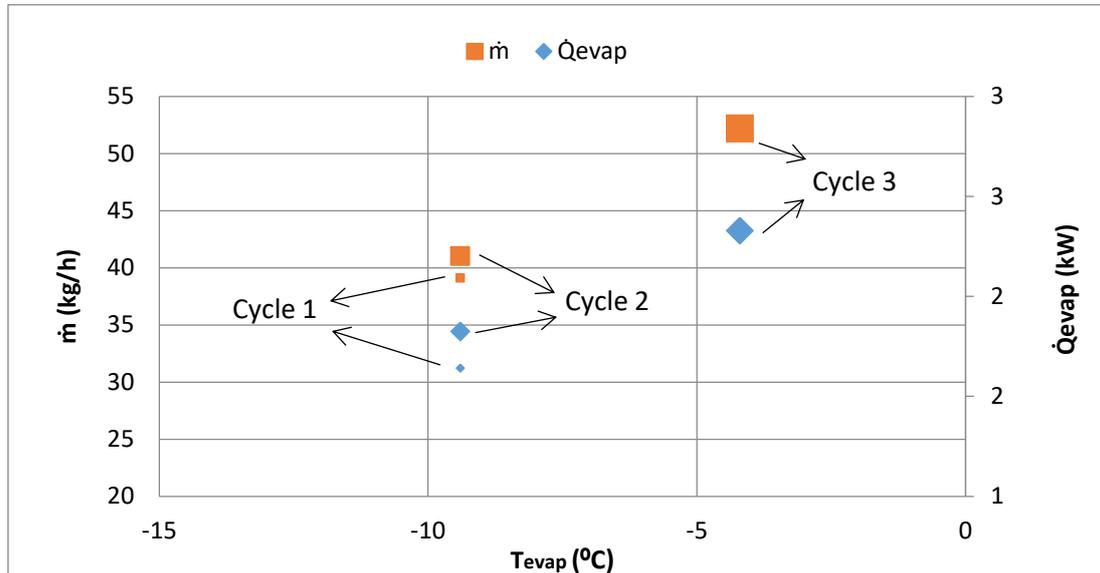

Figure 10. Comparison between mass flow and cooling capacity for the three cycles

6. CONCLUSION

Knowing the influence of each physical and thermodynamic variable on the technical parameters in the operation of a thermal equipment is a task for an engineer, and when this task is accomplished properly, it allows designing or selecting an appropriate equipment to optimize the work for which it was intended.

With the completion of the tests, it was possible to identify with cycle 1 that the increase in the condensing temperature of the refrigeration machine resulted in the reduction of the mass flow rate, increase of the compressor discharge temperature and pressure and increase of the condenser loss of charge which, consequently, resulted in reduced cooling capacity and machine performance.

On the other hand, the increase in evaporation temperature (cycle 3) resulted in increased mass flow rate, cooling capacity and machine performance. The compressor discharge temperature and pressure and the condenser pressure drop were not significantly influenced, as were the evaporator pressure and pressure drop. As a negative point of the latter case, there was an increase in the minimum ambient cooling temperature. Depending on the technical specifications required for the application destination, it may be irrelevant compared to the energetic efficiency of the machine operating under these conditions.

7. REFERENCES

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