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COMPUTATIONAL MODELING AND ENERGY EVALUATION OF A REAL COOLING SYSTEM OF A REFRIGERATION CHAMBER

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Abstract. *This study was developed to theoretically scale out, through computational tools, the efficiency of a real refrigeration system of simple compression that works with refrigerant fluid R404a. The object of study is composed by a refrigeration chamber of the Novo Meriti Frigorífico, located at Estrada São João Caxias, 1752, São João de Meriti, RJ. Two identical refrigeration systems consisting of one condensing unit and two evaporator units, which are connected to each other by two lines of refrigeration pipes, are responsible, in parallel, for cooling the chamber. The system was designed and selected to operate at a maximum surrounding temperature of 40° C and provide an internal temperature around 0° C on cold room for 24 hours a day, 7 days a week. The data utilized for modeling were collected by measuring equipment at monthly maintenance routines. With this, the computational modeling was designed to offer, through thermodynamic properties and pressure drops, an operational image for the main components of the system for 7 months. Consequently, it was possible to perform an energy analysis under the operating conditions throughout the covered months. This work was possible through collaboration between 3 companies and the authors, they are Frigorífico Novo Meriti Ltda., RGT Engenharia de Sistemas Ltda. and Heatcraft Brasil Ltda. These companies participate by releasing information and data on the operation, maintenance and design, respectively, in addition to support and authorization to remain in the installations of the refrigerator units.*

Keywords: *Modelling; Pressure drop; biphasic and monophasic fluid flow; energy efficiency; thermodynamics; heat transfer; simple compression refrigeration system.*

1. INTRODUCTION

Refrigerators have wide range of application, such as preservation of food and medicines, refrigeration of equipment, control of temperature on industrial environments and laboratories, comfort, among others. In these applications, the refrigeration system can assume varying degrees of criticality. In cases of storage of frozen and refrigerated food in refrigerators, the inexistence or failure of the refrigeration directly compromises the preservation of food.

The Frigorífico Novo Meriti Ltda. is in the state of Rio de Janeiro and it is one of the most important refrigerators of the state because of its size and location. Some opportunities of improvement necessary for a better planning of the maintenance and operation of its cold storage units were perceived in this refrigerator, due to the importance and high costs of its control equipment, to constant maintenance, to difficulties to access them, and to the constant energy consumption.

The objective of this study is to develop a tool that allows a better knowledge of the operational conditions of refrigeration equipment and transport lines, based on the graphics and database produced. These can be utilized as a standardization of normal operating conditions of each component of the refrigeration system to provide, on a punctual manner, the identification of abnormal events to realize the preventive and predictive maintenances.

For such, a computational model of the refrigeration system on the new refrigeration chamber of the Novo Meriti Frigorifico, was developed based on published information of previous measurements. The data utilized refers to pressure and temperature, the upstream and downstream of the compressor, overheating and undercooling temperatures in addition to the voltage and current lines supply

This modelling will enable to evaluate the behavior and approximate theoretical performance through the application of the First and Second Laws of Thermodynamics and the Energy Analysis in each component that acts within refrigeration cycle.

2. LITERATURE REVIEW

Bueno (2014) presented a numerical model to simulate the flow of refrigerant fluids on an evaporator of finned tube, where it makes a detailed analysis of the flow regimes assumed by the refrigerant inside the tubes.

Wolverine (2006) provides a simplified model for calculating the pressure drop of biphasic flows in finned tubes.

Santos (2008) lists the application of the 1st and 2nd Laws of Thermodynamics applied to the compressor, the condenser, the expansion valve, and the evaporator. This was done during the comparison of the performance of the same air-conditioning system in the cities of Rio de Janeiro, Brazil and Istanbul, Turkey by utilizing three different refrigerant fluids.

Lauar (2011) modeled the simulation of a finned tube condenser, where the behavior of the refrigerant fluid flow during its passage through the heat exchanger is detailed.

3. MATERIALS AND METHODS

3.1 The Refrigerator – Study Object System

The system under study is intended for the refrigeration of the new cold storage room of Novo Meriti Frigorífico. The chamber in question is in a shed, built in 2014 to increase the storage and distribution capacity of the frigorific, which since 2007 already has a chamber of cooled and another of frozen products, both refrigerated by an ammonia system. The design of the new shed, as shown in Fig. 1, has a freezing chamber (temperature starting at -20ºC), two carcasses chambers (temperature starting at 10ºC) a chill chamber (temperature up to 0ºC) and a large antechamber, (temperature starting at 12 ° C) to decrease the temperature gradient created during the opening of the chambers doors and to minimize the effects caused by high humidity contained in ambient air. The shed has a layered insulation on the floor, which is made, from the ground to up, of an asphalt blanket film, polyurethane (PUR) sheets, concrete and the floor finish itself thus totaling an elevation of approximately 1 meter of the ground level. The walls and roof are insulated with 50.6 millimeters of PUR layer.

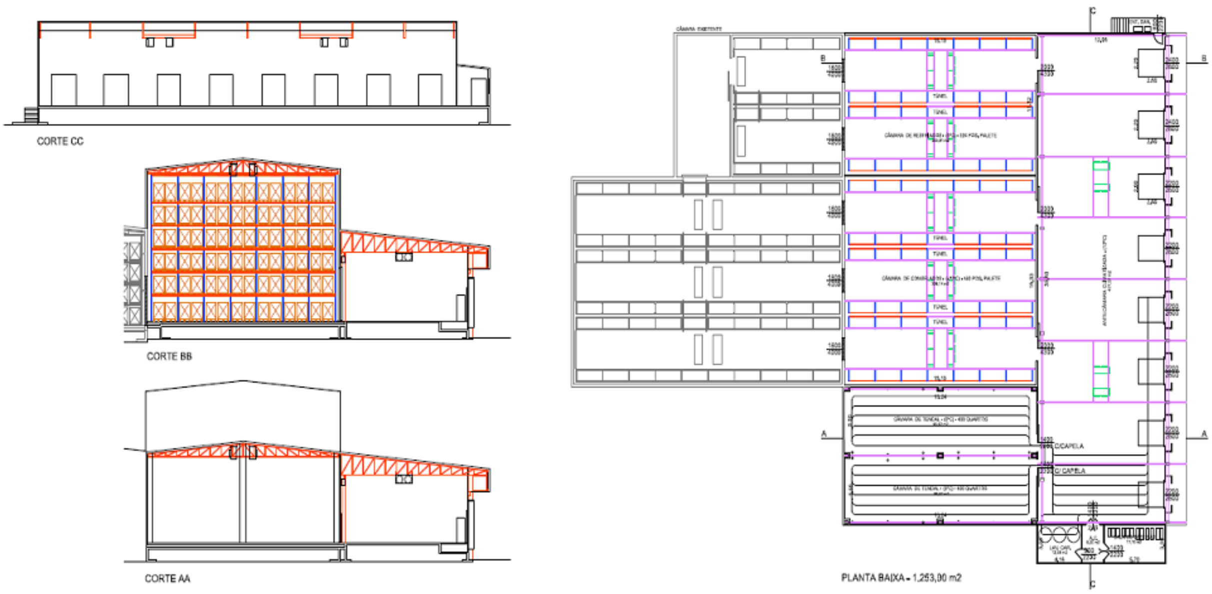

Figure 1. Design of the cold store shed.

The chamber is cooled by four evaporator units manufactured by Heatcraft, model BHL400, and two condensing units also manufactured by Heatcraft, model BZT100H6, located in the machine room. Each of the condensing units is equipped with a Copeland scrool compressor, model ZB95KSE, with nominal capacity of 14HP. These two compressors work in parallel and divide the work of compressing the superheated steam from the evaporators. The condensing units operate in parallel and are connected each one to two evaporator units, as shown in Fig. 2. Each evaporator unit has a thermostatic expansion valve. Thus, each set of one condensing unit plus two evaporator units has parallel and independent operation of one another.

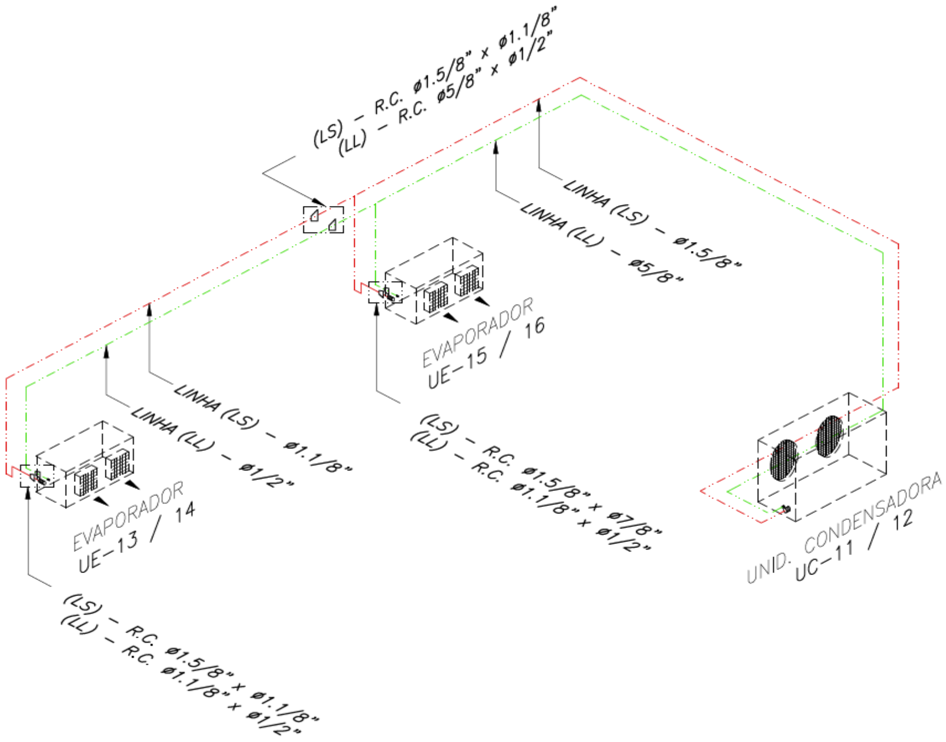

Figure 2. Illustration of the disposition of equipments to each system line.

The system was designed to serve a cold room with 18.20 m of length; 11.30 m of width and 13.00 m of height resulting in a floor area of 205.66 m^2 and a volume of 2673.6 m^3 excluding the prism below the roof; all insulated by a wall of 150 mm of PUR and a door, also insulated with 8.6 m^2. The setpoint designed for the internal temperature of the studied chamber was around 0° C, regardless of the present weather conditions. Surrounding temperatures up of 40° C, concomitant with large demands for output and input of products are challenges to be faced on the hottest months of the year. Its temperature control is of the utmost importance, since processed products such as yogurts and cheeses are stored in the chamber which can't be frozen or exposed to negative temperatures.

The condensing and evaporating units are interconnected by pipelines called Liquid Line (LL), where the saturated liquid flows out of the condenser and continues to lose heat, due to the non-insulated piping, until it reaches the expansion valve in the state of under cooling liquid, and of Suction Line (SL), through which the overheated steam flows, now through a pipe insulated in its entire length by a foam layer of rigid polyurethane (PUR), with a thickness varying between 40 and 50 mm, surrounded by aluminum plates of 0.5 and 0.7mm of thickness, from the end of the evaporator coil to the suction of the compressor in the condenser unit. The characteristics of the flow lines can be observed in Tab. 1.

Each condenser and evaporator unit has a coil for heat exchange and the realization of condensation and evaporation, respectively. Table 2a and 2b shows data on coils of the evaporator and the condensers, respectively.

The working fluid in the system is R404a refrigerant which is a near azeotropic blend of 3 halogens refrigerants (HFC-125, HFC-143a and HFC-134a) in different mass proportions [ASHRAE designation: 404A (44/52/4)].

Table 1. Physical Characteristics of Liquid and Suction Lines

	SL	LL
Material	Copper	Copper
pipeline diameter	1.5/8"	5/8"
90º Curves	10	10
45º Curves	-	-
T Connections	1	1
Equivalent length	65,918 m	65,918 m
Insulation thickness	40 - 50 mm	-
Kind of insulation	Polyurethane (PUR)	-

Table 2a. Data of the Evaporator Coils.

Evaporator Unit (Heat Transfer) 1 Unit	10175 Kcal/h
Evaporator Unit (Heat Transfer) 4 Units	40700 Kcal/h
	47,33 KW
	47334,1 J/s
EU - Length (L)	1,752m
EU - Height (H)	0,889 m
EU - Qnty. of Pipelines	84 Pipelines
EU - Qnty. of Curves (180º)	42 Curves
EU - Linear Length	147,168 m
EU - Equivalent Length (*0,75m)	31,5 m
EU - Total Length	179,668 m

Table 2b. Data of the Condenser Coils.

Condensing Unit (Heat Transfer) 1 Unit	22463 Kcal/h
Condensing Unit (Heat Transfer) 2 Units	44926 Kcal/h
	52,25 KW
	52248,94 J/s
CU - Length (L)	1,37 m
CU - Height (H)	0,875 m
CU - Qnty. of Pipelines	112 Pipelines
CU - Qnty. of Curves (180º)	56 Curves
CU - Linear Length	153,44 m
CU - Equivalent Length (*0,75m)	42,0 m
CU - Total Length	195,44 m

3.2 Computational Modeling

The computational modeling of the refrigeration system was done for the monthly maintenance days, carried out in March, April, June, July, August, September and October of 2016, which totals 14 scenarios. The compressor suction and discharge pressures and temperatures of the two condensing units mentioned above were taken as input data and the values of total (not only useful) overheating and undercooling were considered. The voltages and supply currents were also collected for each refrigeration system. The software AutoCAD and Microsoft Excel (of general use in Engineering) and CoolPack (although specific, is free) were utilized as tools. The routines of computational modeling were based on Excel through Applied Visual Basic.

The code was planned by taking the thermodynamic properties of said fluid and taking into account the localized and distributed pressure drops on the feed lines of the Condensing Units (Suction Lines) and Evaporator Units (Liquid Lines) plus single-phase and two-phase pressure drops within the heat exchanger coils during the latent heat absorption process for phase change.

After the survey of the physical data of the lines, the understanding of the operation of the refrigeration system and the recognition of the proportions and dimensions of the project, it was evident that the pressure drops could not be disregarded in the computational modeling. Figure 3 shows in red a representation of the refrigeration cycle with marking of the points used in the modeling compared to an ideal cycle.

For searching the properties on the points, it was necessary to program a search algorithm and calculations of pressure drops. In the search algorithm of properties, it was decided to avoid the use of state equations for the R404a fluid, due to the errors accumulated in the empirical equations. As the algorithm would use formulas or routines to identify the properties at points of interest, it was preferred to create an algorithm for the direct search or interpolation of these properties, since the manufacturers raise such data experimentally for the development of the tables of thermodynamic properties, together with the specific volume and viscosity equations provided by Solkane in the SOLVAY FLUOR Solkane - Information Service - Solkane 404A Thermodynamics. The thermodynamic properties table of DuPont, DuPont Suva 404a Thermodynamic Properties (SI), were exported to excel from a pdf format through a file in txt format. Later, from this was generated a another one in the format csv that generated the worksheets of saturated and non-saturated liquids and vapors, through an organization process done by a VBA routine. Tests were performed to verify the values contained in the spreadsheets with IPU CoolPack software.

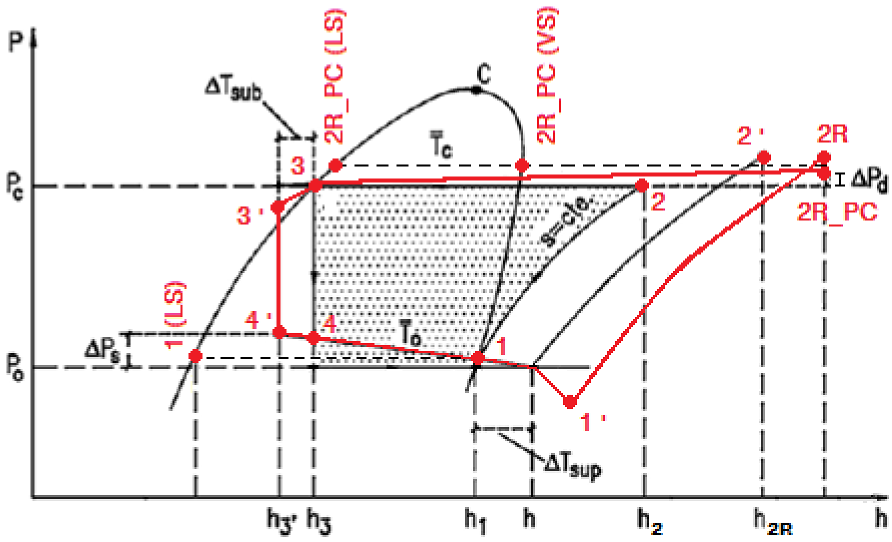

Figure 3. Cooling cycle with the points used on the modeling.

After the execution of the algorithm to search for the properties of points 1 ', 1, 1 (LS) and 2R, whose input data are already known, the knowledge of the pressure drops is necessary for the search of properties of the other points. The point 1 (LS) corresponds to the Saturated Liquid at the same pressure as the Saturated Vapor at point 1.

For this purpose, the mass flow was calculated during the operation of the compressors, using the First Law of Thermodynamics, taking the actual compression work applied to the refrigerant. This work was calculated by means of the compression power at the time of the voltage (V) and current (I) measurements of the compressor, as explained in Eq. (1).

$$P = \frac{\mathrm{V} \times \mathrm{I} \times \sqrt{3} \times \eta \times \cos\varphi}{1000} \tag{1}$$

Where η is the efficiency of the compressor and cos φ is the power factor of the installation.

The friction factors were calculated by the Colebrook Equations to determine the single-phase pressure drops. The single-phase pressure drop between the compressor output and the condenser input were calculated after the termination of the algorithm. Such loss has been dimensioned to an equivalent length of 6 m since there are 1 m of tube, 6 curves of 90 ° plus a filter. With this pressure gradient, the properties of the 2R_PC point and subsequently the 2R_PC (VS) and 2R_PC (LS) points were found, which corresponds to the Saturated Vapor and Liquid at the same pressure as the point 2R_PC. However, at point 1’, in the opposite direction of the flow of the fluid in the system, the single-phase pressure drop was made to the suction line. The properties at points 1’_PC were sought with this pressure variation.

Since inside the coils there are two types of flow, single-phase and two-phase, it was necessary to dimension the lengths where each one occurs. The equivalent length of the coils was calculated through the lengths of the fins, from the equivalent lengths in the halogenated fluids to 180 ° curves and their respective amounts. The total length was related to the thermal load required in the chamber design for the measurements of that specific month. This ratio was then used for the design of the superheat length in the evaporator and superheat units in the condenser units, through the variations of enthalpies already dimensioned with the respective superheated and saturated vapors at points 2R_PC and

2R_PC (VS), in the condensers, and at points 1'_PC and 1, on the evaporators. Over these lengths, the single-phase pressure drop in the condenser and evaporator were calculated through an existing algorithm. The remaining length in each exchanger was destined for the biphasic flow and its pressure drops.

In the heat exchangers, the flow regime during the phase change was adopted as Homogeneous in Separated Pipes. According to Wolverine Tube, Engineering Thermal Innovation the homogeneous model is the simplest technique for the analysis of biphasic flows. In this model the slip between the phases is disregarded and the biphasic mixture is treated as a pseudo fluid that obeys the usual equations of the single-phase flows, with properties obtained by a weighted average of the properties of the individual phases. In this way the phases are considered in equilibrium, with the same pressures, velocities and temperatures, and the standard analysis method of fluid mechanics and heat transfer can be applied (Wolverine Tube, Engineering Thermal Innovation - Engineering Data Book III, 2006).

In the condenser, the biphasic flow was treated through a discretization of the tube by varying the titer from 0.999 to 0.001 with steps of 0.001. The 1 and 0 titers were avoided due to the mathematical discontinuities in the Void Fraction calculation equation, present in the Pressure Drop Momentum, and then compensated by the doubling of the pressure drop values in the first and last discretization. The next challenge was to unite the existing discretization with that of Pressure Drop Friction that does not depend on the title. It was then necessary to create a routine in which the known loss of load multiplied by 1.1 was used for the dimensioning of the lengths of each discretization. After this calculation the friction loss was calculated.

An equivalent process was done in the evaporator. However, before calculating the pressure drop, it was necessary to find the properties of point 3, calculate the monophasic pressure drop in the line of liquids to find the properties of the point 3’ and the initial title of the evaporator unit by a lever rule using the enthalpies of points 1 and 1 (LS) through the enthalpy of this last point, all previously scaled by the search algorithm. Once the resulting pressures in the evaporator discretization are always greater that the incoming pressures when calculating the pressure drop, it is important to emphasize how it occurred. In the evaporator, the pressure drop was dimensioned through the code rhat was already developed for the condenser by utilizing an inverse discretization that started from the final properties at point 1 for the initial and unknown at point 4’. For a better understanding, one can use the term "load gain calculation" to explain this process, although this is not correct in engineering. At each value returned in the algorithm, the pressures were multiplied by 2, since they are two evaporator units, whose input and output properties are identical.

After all pressure drops were calculated, the search algorithm returns the properties of points 4 and 4’ that are now possible to be collected. With this last stage the search for the properties of all the points presented as pertinent in the refrigeration cycle, as shown in Figure 4, which is the modeling of the thermodynamic properties of the system during the data collection in March of 2016 for the cooling system of the Condensing Unit 11 (CU11). With these data the Energy and Exergy Balance equations from the 1st and 2nd Laws of Thermodynamics, respectively, according to SANTOS, 2008, were applied in the main components of the line thus being the two Compressors in parallel, the two capacitors in parallel, the four evaporators in parallel two by two and the expansion valves also in parallel. Then, the computational modeling and the energy evaluation are finished.

4. RESULTS OBTAINED

4.1 Thermodynamics Properties

The algorithm returns the thermodynamic properties in the format shown at Tab. 3.

Table 3. Modeling of the thermodynamic properties of the system during the data collection on March of 2016.

#	YEAR	MONTH	UNIT	POINT	Temperatura (ºC)	Temperature (K)	Pressure (K.Pa)	PD_1P (T) (K.Pa)	PD_1P (S) (K.Pa)	PD_2P (K.Pa)	Enthalpy (KJ/Kg)	Entropy (KJ/K.Kg)	Volume (m3/Kg)	Surface T. (N.m)	Viscosity (µPa.s)
1	2016	3	UC11	1	-11,3	261,85	413,11	0,443796	104,7114893	---	361,82	1,62072	0,04841	---	10,79721
2	2016	3	UC11	1 (LS)	-11,3	261,85	421,25	---	---	---	184,31	0,94207	0,000839099	---	---
3	2016	3	UC11	1'_PC	4,7	277,85	307,9547143	---	---	---	377,16	1,70050	0,072034547	---	11,41001
4	2016	3	UC11	1'	3	276,15	310,2642	---	---	---	375,67	1,69452	0,070897649	---	11,3449
5	2016	3	UC11	2	---	---	---	---	---	---	---	1,62072	---	---	---
6	2016	3	UC11	2'	---	---	---	---	---	---	---	1,69452	---	---	---
7	2016	3	UC11	2R	61	334,15	2033,9542	0,008803781	---	---	405,56	1,65725	0,010142321	---	13,5663
8	2016	3	UC11	2R_PC	61	334,15	2033,945396	---	59,78940792	300,8758511	405,56	1,65725	0,010142382	0,000624246	13,5663
9	2016	3	UC11	2R_PC (VS)	44,79042939	317,9404294	2033,945	---	---	---	385,78	1,59659	0,008662871	---	12,94547345
10	2016	3	UC11	2R_PC (LS)	44,4872715	317,6372715	2033,945396	---	---	---	272,23	1,23907	0,0011	---	99,11561263
11	2016	3	UC11	3	36,3206846	309,4706846	1673,280137	3,611959187	---	---	257,08	1,19220	0,001013262	---	111,4010444
12	2016	3	UC11	3'_PC	27,63280238	300,7828024	1669,668178	---	---	---	256,92	1,19170	0,000967108	---	123,4860972
13	2016	3	UC11	3'	27,7206846	300,8706846	1347,507756	---	---	---	242,35	1,14499	0,000967522	---	---
14	2016	3	UC11	4	-11,3	261,85	---	---	---	---	257,0772323	---	---	---	---
15	2016	3	UC11	4'	3,863601844	277,0136018	689,0492785	---	---	275,9392785	242,3530954	1,14980	0,0009	---	---
16	2016	3	UC12	1	-11,8	261,35	405,96	0,437430222	106,6622998	---	361,52000	1,62092	0,04926	---	10,77806
17	2016	3	UC12	1 (LS)	-11,8	261,35	414	---	---	---	183,66	0,93952	0,000837902	---	---
18	2016	3	UC12	1'_PC	3,7	276,85	298,86027	---	---	---	376,40	1,70025	0,074065727	---	11,37171
19	2016	3	UC12	1'	2	275,15	303,36944	---	---	---	374,89	1,69356	0,072324162	---	11,3066
20	2016	3	UC12	2	---	---	---	---	---	---	---	1,62092	---	---	---
21	2016	3	UC12	2'	---	---	---	---	---	---	---	1,69356	---	---	---
22	2016	3	UC12	2R	62	335,15	1999,4804	0,009301111	---	---	407,3286254	1,66352	0,010464157	---	13,6046
23	2016	3	UC12	2R_PC	62	335,15	1999,471099	---	70,01148097	313,569341	407,33	1,66352	0,010464231	0,000532862	13,6046
24	2016	3	UC12	2R_PC (VS)	44,06465471	317,2146547	1999,471	---	---	---	385,71	1,59724	0,008880604	---	12,91767628
25	2016	3	UC12	2R_PC (LS)	43,75689053	316,9068905	1999,471099	---	---	---	270,84	1,23477	0,0011	---	100,1978123
26	2016	3	UC12	3	34,9040372	308,0540372	1615,890277	3,482849115	---	---	254,63	1,18427	0,001004928	---	113,5901043
27	2016	3	UC12	3'_PC	26,016308	299,166308	1612,407428	---	---	---	254,4693544	1,183789694	0,000959641	---	126,343181
28	2016	3	UC12	3'	26,1040372	299,2540372	1291,947669	---	---	---	239,6768632	1,136261801	0,000960038	---	---
29	2016	3	UC12	4	-11,8	261,35	---	---	---	---	254,627267	---	---	---	---
30	2016	3	UC12	4'	7,602804301	280,7528043	771,3042185	---	---	365,3442185	239,6768632	1,138533126	0,0009	---	---

4.2 First Law of Thermodynamics

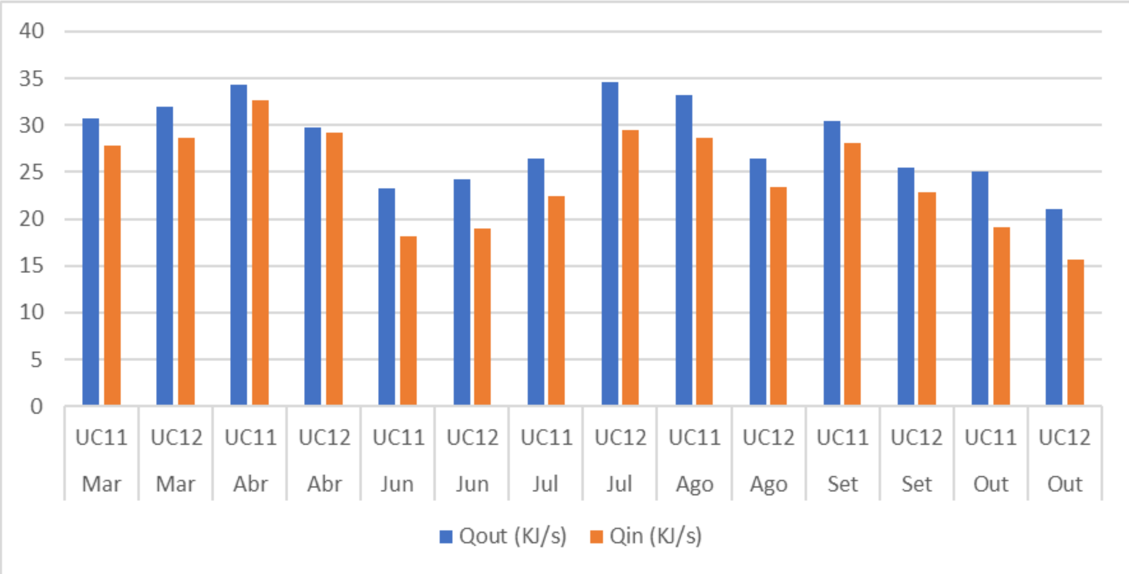

Figure 4. Heat flows in and out.

4.3 Second Law of Thermodynamics

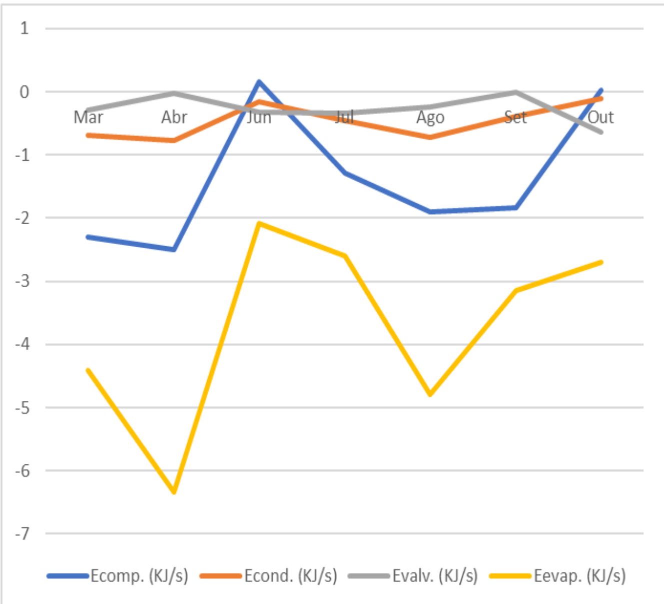

Figure 5a. Rate of Exergy Destruction (UC11)

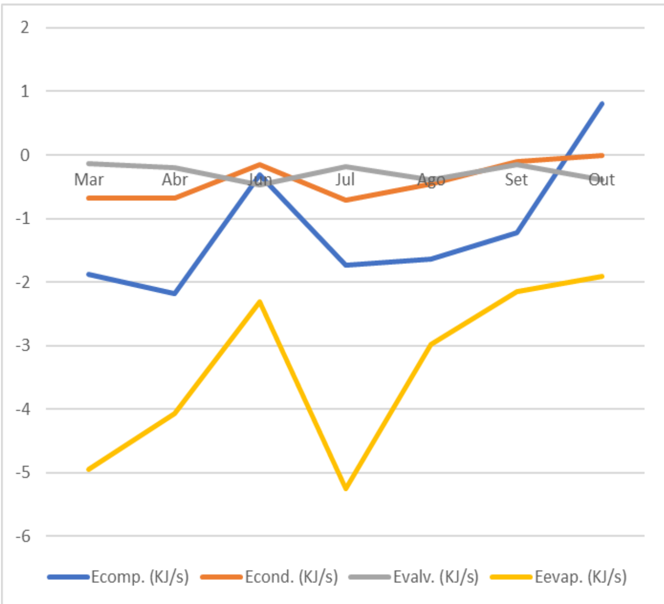

Figure 5b. Rate of Exergy Destruction (UC12)

4.4 Energy Analysis

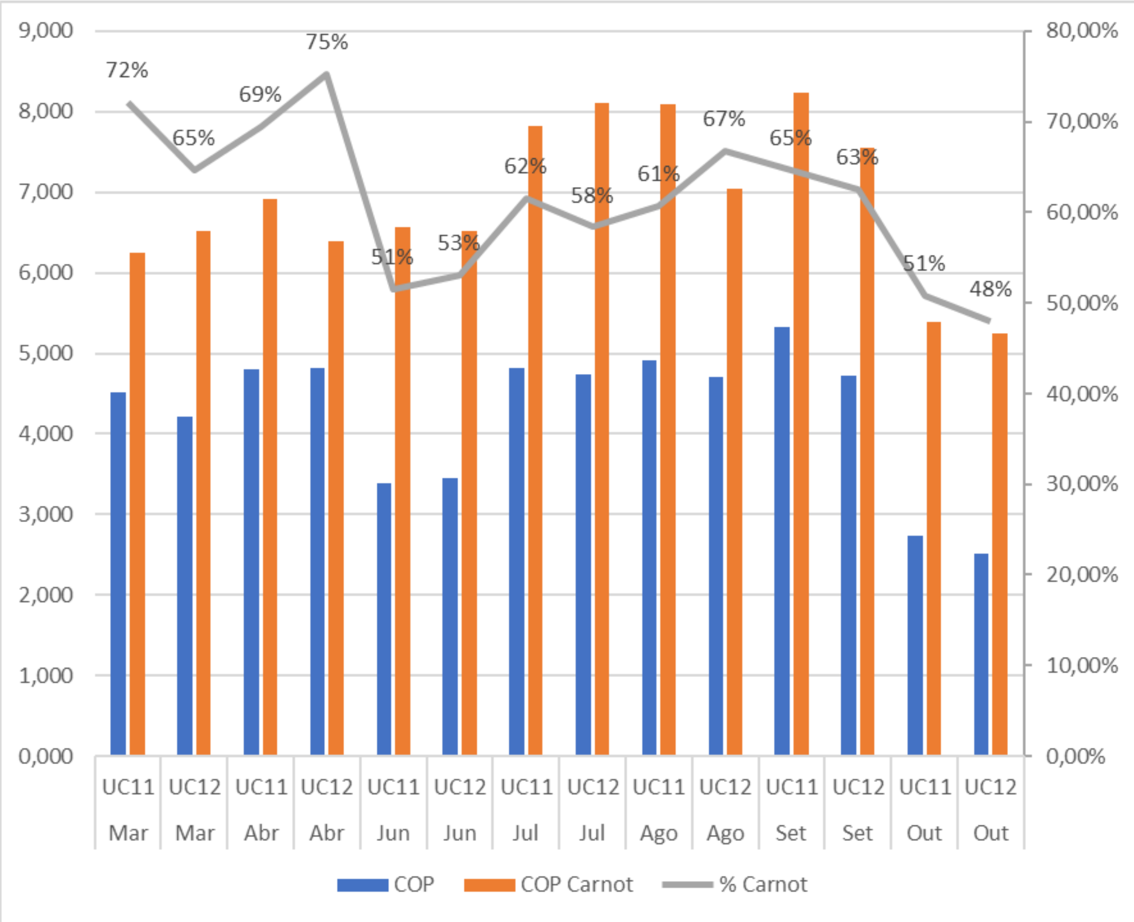

Figure 6. COP of the Cycle, COP Carnot, % Carnot.

5. CONCLUSIONS

This modeling aimed to predict the performance of the refrigeration system by applying the First and Second Laws of Thermodynamics in each component of the refrigeration cycle. This was planned by utilizing the thermodynamic properties of said fluid (R404a), and taking into account the localized and distributed pressure drops in the feed lines of the Condensing Units (Suction Lines) and Evaporator Units (Liquid Lines) and the single-phase and two-phase pressure drop within the heat exchanger coils. Altogether, the properties of the usual points of a real refrigeration system were calculated.

The properties were collected based on tables of thermodynamic properties supplied by Dupont, the refrigerant manufacturer of the R404a, plus some equations of the specific volume and viscosity supplied by the manufacturer Solkane. The Dupont's Properties Table was plotted on a spreadsheet of the Microsoft Office Excel 2016 software, that made possible to search directly or interpolate properties through a programmed algorithm in Visual Basic. The calculated properties with the Solkane formulas were then calculated in order to be inserted on the already collected or interpolated temperatures.

At the end of the modeling, it was verified that the modeled data were approximately matching what was verified by the measurements taken. This analysis can be done on the table 3 through the properties of the point 3', measured in the system, versus the properties of the point 3'_PC which were determined by the modeling.

It concluded, therefore, that even using two-phase empirical pressure drop equations and the approximations considered, the criteria adopted together with measurements / readings were well used to build a suitable modeling that can provide a better cost and logistics scaling plus a warning system if any component is damaged.

On the energy evaluation, it is noticeable that on months in which the exergy variation rate of the compressors was positive were also the months with the lowest efficiencies in the respective refrigeration lines. Therefore, we can consider as an indication that the compressors require predictive maintenance.

Another very important observation is that the second law on heat exchangers indicates the proper working of condensers and evaporators because the heat flow coming out of the system is always greater than the one coming in.

That is, there is no leak or any other fault in the coils. Through the modeled data, the rate of exergy destruction of the compressors takes the second place in this type of refrigeration system, the biggest one stay with the evaporators.

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