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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, temperature control on industrial environments and laboratories, ambient comfort, among others. In these applications, the refrigeration system can assume varying degrees of criticality. In cases of frozen and refrigerated food storage, any refrigeration failure directly impacts on its preservation.

The Frigorífico Novo Meriti Ltda. is one of the most important cold store in the State of Rio de Janeiro due to its size and location. Some improvement opportunities necessary for a better maintenance and operation planning of its cold storage units were noted, considering its typical high energy consumption, frequent maintenance operations, the importance and high costs of its control equipment and difficulties to access them.

This study aims to develop a tool that allows a better knowledge of the operational conditions of refrigeration equipment and its transport lines, based on the graphics and database produced. It can be utilized as a prediction resource of normal operating conditions of each component from the refrigeration system to provide the identification of abnormal events to signalize corrective, preventive and predictive maintenance needs.

A computational model for the refrigeration system from the new refrigeration chamber was developed based on the First and Second Laws of Thermodynamics. Data available from previous measurement reports encompass suction and discharge pressure and temperature, voltage and current feeding the compressor, superheating and sub-cooling temperatures. This modelling will enable to evaluate the behavior and approximate theoretical performance of each component from the refrigeration cycle.

2. LITERATURE REVIEW

Santos and Costa Filho (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.

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

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.

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.

3. MATERIALS AND METHODS

3.1 The Study Case

The refrigeration system under study is intended to meet the needs of the new cold storage room from Frigorífico Novo Meriti. This chamber is in a shed, built in 2014 to increase the storage and distribution capacity of the frigorific, which since 2007 has already owned a chamber of cooled and another of frozen products, both refrigerated by an ammonia system. The design of the new shed, as shown in Figure 1, includes a freezing chamber at -20ºC, two carcass chambers at 10 ºC, a chill chamber up to 0 ºC and a large antechamber at12°C, the last for decreasing the temperature gradient created during the opening of the chamber doors and to minimize the effects caused by the high humidity contained in the external 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 from the ground level. Further, the walls and roof are insulated with 50.6 millimeters of PUR layer.

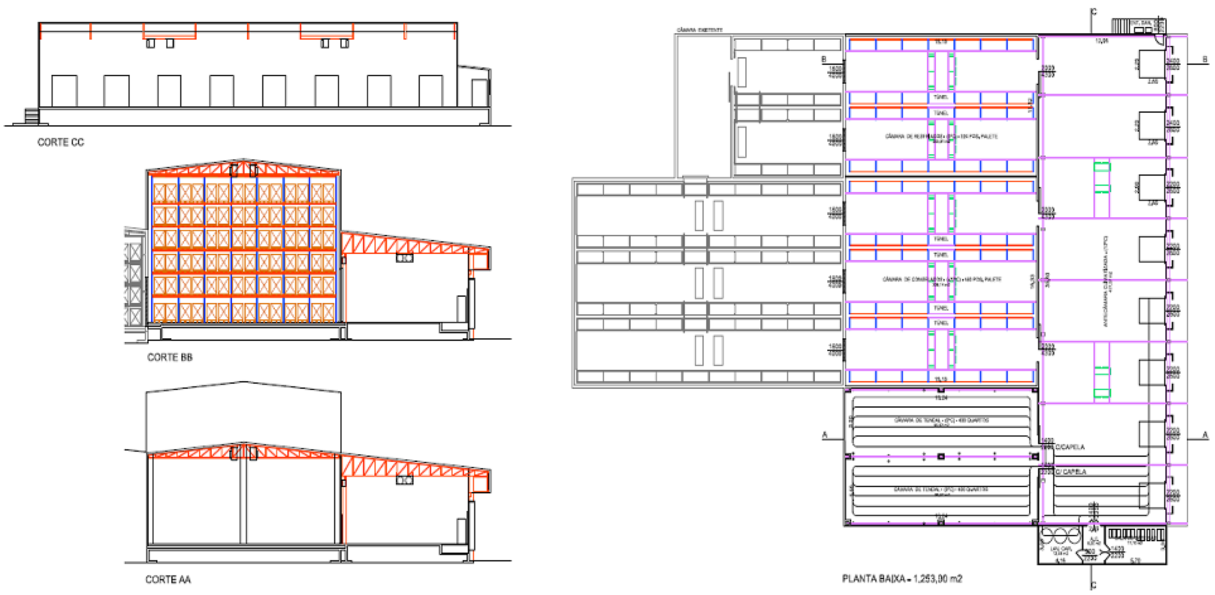

Figure 1. Design of the cold store shed.

The chamber is refrigerated by four evaporating units manufactured by Heatcraft, model BHL400. Each one has its own thermostatic expansion valve. Two condensing units also manufactured by Heatcraft, model BZT100H6, independently operate in parallel since each one is connected to two evaporating units, as shown in Figure 2. Each condensing unit is equipped with a Copeland scroll compressor, model ZB95KSE, with nominal capacity of 14HP. Both condensing units are in the machine room.

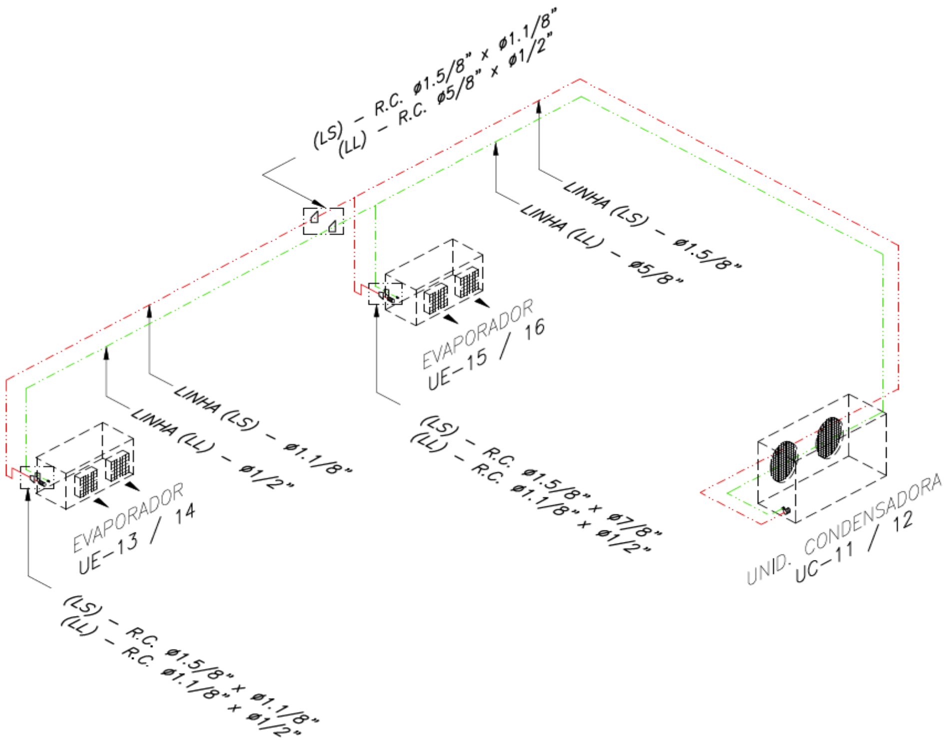

Figure 2. Isometric drawing of the refrigerant lines.

This refrigeration 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. Its setpoint is around 0° C, regardless of the present weather conditions. Surrounding temperatures up to 40° C, concomitant with large demands for product entrances and exits 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 be neither frozen nor exposed to negative temperatures.

As illustrated in Figure 2, the condensing and evaporating units are interconnected by pipelines called Liquid (LL) and Suction (SL) Lines. In the first, saturated liquid exits the condenser and continues to lose heat, due to the non-insulated piping, until it reaches the expansion valve at the subcooled liquid state. In the other, superheated steam flows from the end of the evaporator coil to the compressor suction, 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. Table 1 presents the characteristics of both refrigerant lines.

The Table 2 shows data from the evaporator and condenser coils.

The working fluid 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
T Connections	1	1
Equivalent length	65,918 m	65,918 m
Insulation thickness	40 - 50 mm	-
Kind of insulation	Polyurethane (PUR)	-

Table 2a. Evaporator Coil Data.

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. Condenser Coil Data.

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 of the Refrigeration System

The computational modeling had utilized data generated in the monthly maintenance days on March, April, June, July, August, September and October of 2016, producing 14 scenarios. The input data were the two compressors suction and discharge pressures and temperatures, their voltages and supply currents, and the superheating and sub-cooling temperatures. The software AutoCAD and Microsoft Excel, both of general use in Engineering, and the free software CoolPack were utilized as tools. The routines of the computational modeling were developed in Excel through Applied Visual Basic.

The code was prepared by taking the working fluid thermodynamic properties and considering the localized and distributed pressure drops in the feeding lines of the Condensing and Evaporating Units and also the single-phase and two-phase pressure drops within the heat exchanger coils during the latent heat absorption or rejection process for phase change.

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

For searching the fluid properties at the desired points, it was necessary to build a search algorithm and calculate the pressure drops. It was decided to avoid the use of state equations for the R404a, due to the errors accumulated in these empirical equations. It was preferred to create an algorithm for the direct search or interpolation of these properties in the thermodynamic property tables, since the manufacturers raise such data experimentally for building them. These data were processed together with the specific volume and viscosity equations provided by SOLVAY, 2017. The thermodynamic property table from DUPONT, 2016, were exported to excel from a pdf format through a file in txt format. From this, it was generated another one in the format csv that generated the worksheets of saturated and non-saturated liquids and vapors, through an organization process executed by a VBA routine. Tests were performed through comparisons among the values contained in the spreadsheets with the correspondents generated by the IPU CoolPack software.

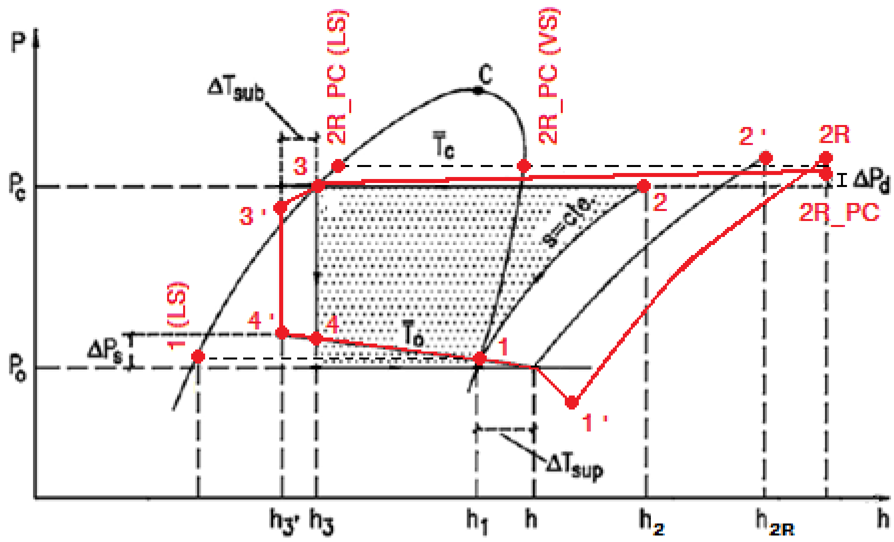

Figure 3. Refrigeration cycle indicating the points used in the modeling.

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

For this purpose, the mass flow rate 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 from the compression power at the time of the voltage (V) and current (I) measurements of the compressor, as expressed by 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 through 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 ° and a filter. With this pressure loss, the properties at the point 2R_PC and subsequently at the points 2R_PC (VS) and 2R_PC (LS) 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 in the system, the single-phase pressure drop was calculated to the suction line. The properties at the point 1’_PC were sought with this pressure variation.

Since inside the coils there are single and two-phase flows, it was necessary to determine the lengths where each one occurs. The equivalent lengths of the coils were calculated through the lengths of the fins, from the equivalent lengths for the halogenated fluids to 180 ° curves and their respective amounts. The total length was related to the thermal load required in the chamber design from the measurements from each specific month. This ratio was then used to determine the superheated vapor lengths in the evaporator and condenser units, through the enthalpy variations 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 in the evaporators. Through these lengths, the single-phase pressure drops in the condenser and evaporator were calculated. The remaining length in each heat exchanger was attributed to 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, according to WOLVERINE TUBE, INC, 2016.

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 repeating the closest values in the first and last discretization points. 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 losses were calculated.

An equivalent process was done in the evaporator. However, before calculating the pressure drop, it was necessary to find the properties at point 3, calculate the monophasic pressure drop in the line of liquids to find the properties at the point 3’ and the initial title of the evaporator unit by a lever rule using the enthalpies at 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 that 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 ones at point 4’. For a better understanding, one can use the term "load gain calculation" to explain this process, although this has no physical meaningful. At each value returned in the algorithm, the pressures were multiplied by 2, since there are two evaporator units, whose input and output properties are identical.

After all pressure drops were calculated, the search algorithm returns the properties at points 4 and 4’ that are now possible to be collected. With this last stage the search for the properties at 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 on March of 2016 for the refrigeration system from 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 & COSTA FILHO, 2008, were applied to the main components of the refrigeration circuits.

4. RESULTS

4.1 Thermodynamics Properties

The algorithm returns the thermodynamic properties in the format shown in Table 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

The Figure 4 presents the heat flows in and out, where the heat output flows out are greater than the heat input flow to every month.

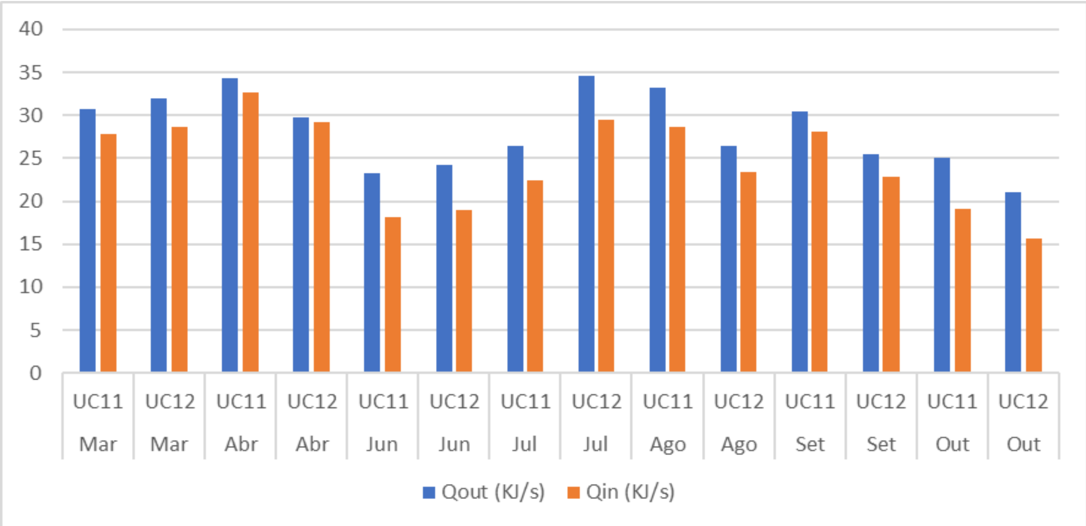

Figure 4. Heat flows in and out.

4.3 Energy Analysis

The Figure 6 presents the refrigeration cycles' COP's, the correspondent Carnot's COP's and the % Carnot.

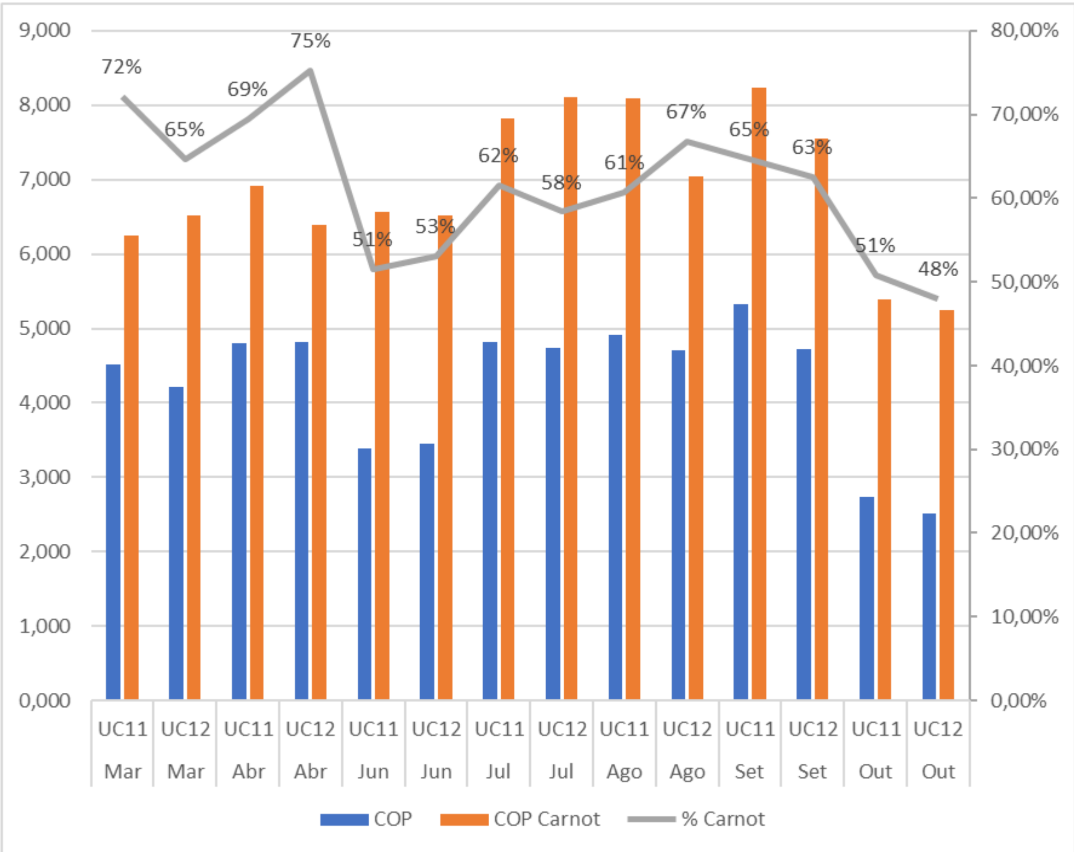

Figure 5. COP of the Cycle, COP Carnot, % Carnot.

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 the working fluid R404a, and taking into account the localized and distributed pressure drops in the feeding lines of the Condensing (Suction Lines) and Evaporating Units (Liquid Lines) and the single-phase and two-phase pressure drops 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 Property 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 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.

6. ACKNOWLEDGEMENTS

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