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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 size, through computational tools, the efficiency of a real refrigeration system of simple compression, working with refrigerant fluid R404a. The object of the study is composed of 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, the pipeline extensions of each one is their own, are responsible, in parallel, by the cooling of the chamber. The system was designed and selected to operate at a maximum ambient temperature of 40° C and provide an internal temperature around 0° C on cold room for 24 hours a day, 7 days a week. To the modeling, the data were collected by measuring equipment at monthly maintenance routines. With this, the computational modeling was designed to offer, through equations of state, thermodynamic properties and pressure drops, a working image for the main components of the system for 7 months. Consequently, it was possible to perform an energy analysis under the operating conditions in the months covered. This work was possible through the 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 of the operation, maintenance and design, respectively, plus support and authorization to stay 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 acting in diverse types of application, like food preservation, medicines, refrigeration of equipment, control of temperature and industrial environments and laboratories, comfort, among others. In these applications, the refrigeration system can assume varying degrees of criticality. In the case of the storage of frozen and refrigerated food in refrigerators, a non-existence or failure of the refrigeration committed directly in the maintenance of food.

In the state of Rio de Janeiro, stands the Frígorifico Novo Meriti Ltda., One of the most important for its size and location. The needs for improvements needed to better plan the maintenance and operation of your refrigeration units, due to the importance and high cost of your control equipment, constant maintenance and difficult access to them.

The objective of this study was 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, that act as a database of operating conditions of each one component of the refrigeration system so as to provide, in a timely manner, the identification of abnormal events.

For that, a computational model was developed of the refrigeration system of the new refrigeration chamber of the Novo Meriti Frigorifico, based on published information of measurements performed. Such data as pressures and temperatures, an upstream and downstream of the compressor, overrun and undercooling temperatures in addition to the lines' supply voltages and currents.

This model will make it possible 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 without refrigeration cycle.

2. MATERIALS AND METHODS

2.1 The Refrigerator – Study Object System

The system under study is intended for the refrigeration of the new cold storage room to the Novo Meriti Frigorifico. The chamber in question is housed in a warehouse, built in 2014 to increase the storage and distribution capacity of the refrigerator, which since 2007 already has a chilled chamber and other frozen chilled by an ammonia refrigeration system. The design of the new shed, as shown in Figure 1, has a freezing chamber, a chill chamber (temperature up to 0ºC) and a large antechamber (Temperature from -20ºC), two chambers (temperature from 10ºC) (12 ° C) temperature so as to decrease the temperature gradient created during opening the chambers doors and minimize the effects caused by high humidity contained in ambient air. The shed has a layered insulation on the floor, being formed, from the ground up, an asphalt blanket film, polyurethane (PUR) sheets, concrete and the floor finish itself, totaling an elevation of approximately 1 m of the ground level. The walls and roof are insulated with a 50.6 mm PUR layer.

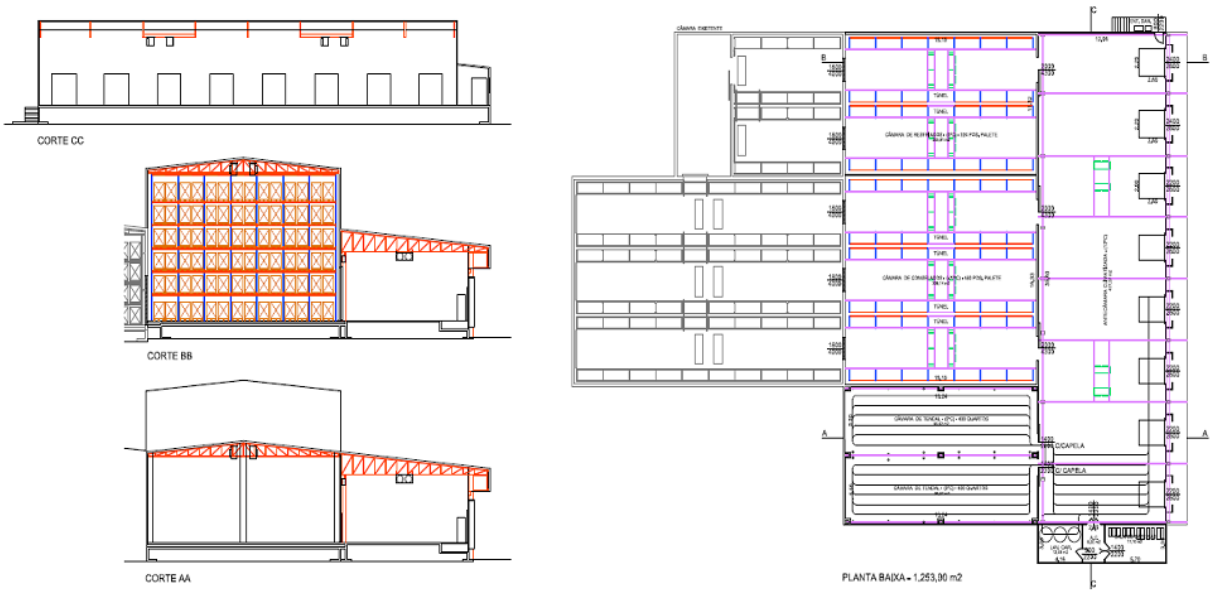


Figure 1 – Design of the cold store shed.

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 engine room. Each of the condensing units is equipped with a Copeland scrool compressor, model ZB95KSE, with nominal capacity of 14HP. These 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 to two evaporator units, as shown in Figure 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.

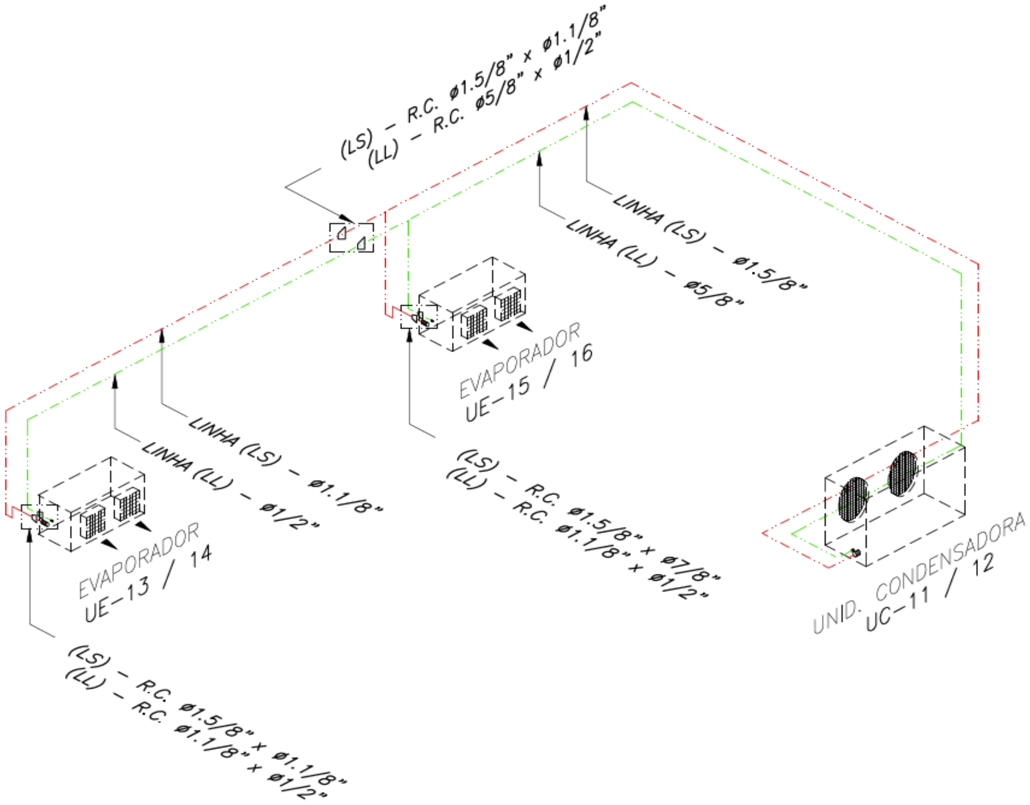


Figure 2 – Illustration of the disposition of equipments to each system line.

The system was designed to serve a cold room with 18.20 m of length; 11.30 m of wide and 13.00 m of high 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 was around 0° C, regardless of the weather conditions present. Ambient temperatures up of 40° C, concomitant with large demands for output and product entry are challenges to be faced in the hottest months of the year. Its temperature control is of the utmost importance as 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 the liquid line (LL), where the 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 (LS) through which the superheated steam flows, now through a pipe isolated in its entire length by a layer of rigid polyurethane foam (PUR), with a thickness varying between 40 and 50 mm, surrounded by aluminum plates 0.5 and 0.7mm thick, from the end of the evaporator coil to the suction of the compressor in the condenser unit. In Table 1, one can observe the characteristics of the flow lines.

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)	-

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

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

2.2 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, making 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) supercooling and undercooling were considered. The voltages and feed currents were also collected for each refrigeration system. Used as tools are the software Autocad and Microsoft Excel, of general use in Engineering and CoolPack, the latter though specific, free use. The routines of computational modeling were based on Excel through Applied Visual Basic.

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

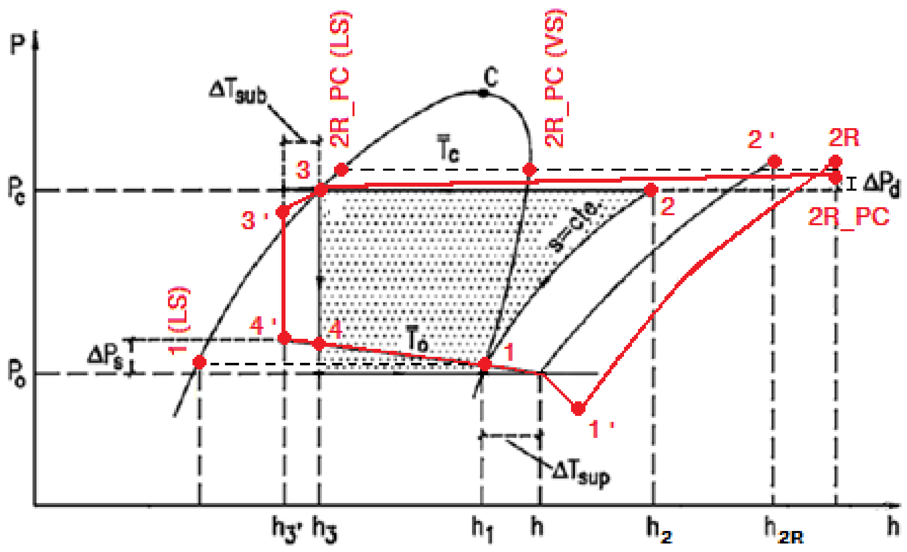
The code was planned by tapping thermodynamic properties of said fluid, and taking into account the localized and distributed load losses on the feed lines of the Condensing Units (Suction Lines) and Evaporator Units (Liquid Lines) plus losses of single-phase and two-phase loads within the heat exchanger 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 load losses 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.

To search for the properties in the points it was necessary to program the search algorithm and calculations of losses of loads.

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 file. Solkane - Information Service - Solkane 404A Thermodynamics. The thermodynamic properties table of DuPont, DuPont Suva 404a Thermodynamic Properties (SI), were exported to excel through an online file converter in pdf format to file in txt format. From this was generated a file in the format csv that generated the worksheets of 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, it is necessary to search the properties of the other points for the knowledge of the load perry. Point 1 (LS) corresponding 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 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.

To determine the losses of single-phase loads, the friction factors were calculated by the Colebrook equations. After the termination of the algorithm, the single-phase load losses between the compressor output and the capacitor input were calculated. Such loss has been dimensioned to an equivalent length of 6 m since there are 1 m of tube, 6 curves 90 ° knee 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, corresponding to the Saturated Vapor and Liquid at the same pressure as the point 2R_PC, were found. Already at point 1 ', in the direction opposite to the flow of the fluid in the system, the loss of single-phase load was made to the suction line. With this pressure variation the properties at points 1'_PC were sought.

Since inside the coils there are two types of flow, single-phase and two-phase, it was necessary to dimension the lengths where each 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 that month measurement. 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 capacitors, and at points 1'_PC and 1, on the evaporators. Over these lengths, the losses of single-phase loads 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 losses of loads.

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 pseudofluid 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 at 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 load loss 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.

In the evaporator unit an equivalent process was done. However, before calculating the loss of charge, it was necessary to find the properties of point 3, calculate the monophasic charge loss in the line of liquids to find the properties of the 3 'point and through the enthalpy of the latter the initial title of the evaporator unit by a lever rule using the enthalpies of points 1 and 1 (LS), all previously scaled by the search algorithm. As in the evaporator discretization, by calculating the pressure drop, the resulting pressures are always greater than the incoming pressures, it is important to emphasize how it occurred. In the evaporator, the load loss was dimensioned through the code already developed for the capacitor, using an inverse discretization that started from the final properties at point 1 for the initial and unknown at the 4 'point. 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 load losses are 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 the month of March of 2016 for the cooling system of the Condensing Unit 11 (UC11). 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, being therefore 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, closing the computational modeling

3. RESULTS OBTAINED

3.1 Thermodynamics Properties

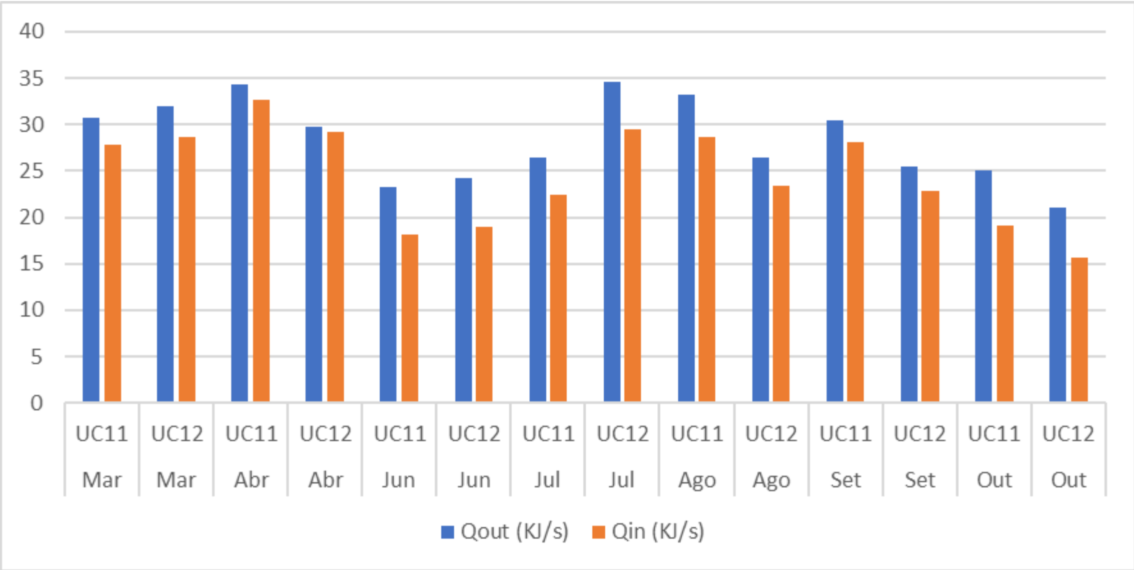
The algorithm returns the thermodynamic properties in the format shown at table 4.

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

#	ANO	MÊS	UNIDADE	PONTO	Temperatura (ºC)	Temperatura (K)	Pressão (K.Pa)	PD_1P (T) (K.Pa)	PD_1P (S) (K.Pa)	PD_2P (K.Pa)	Entalpia (KJ/Kg)	Entropia (KJ/K.Kg)	Volume (m3/Kg)	Surface T. (N.m)	Viscosidade (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	---	---

3.2 First Law of Thermodynamics

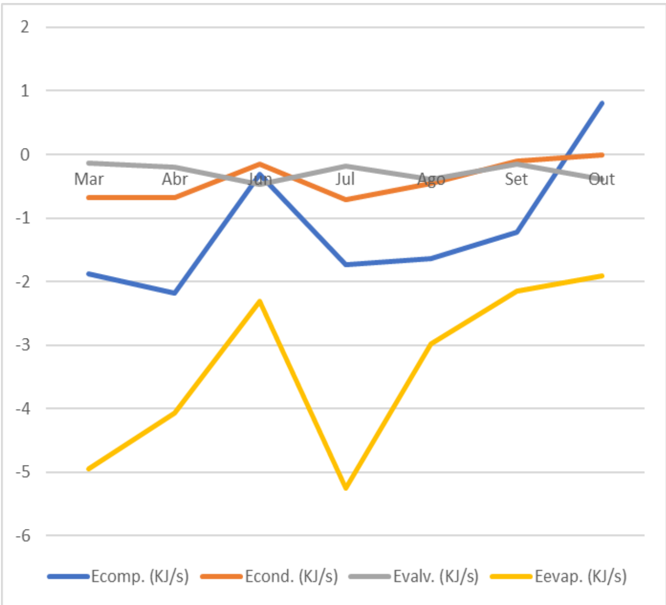
Graphic 1 – Heat flows in and out.



3.3 Second Law of Thermodynamics

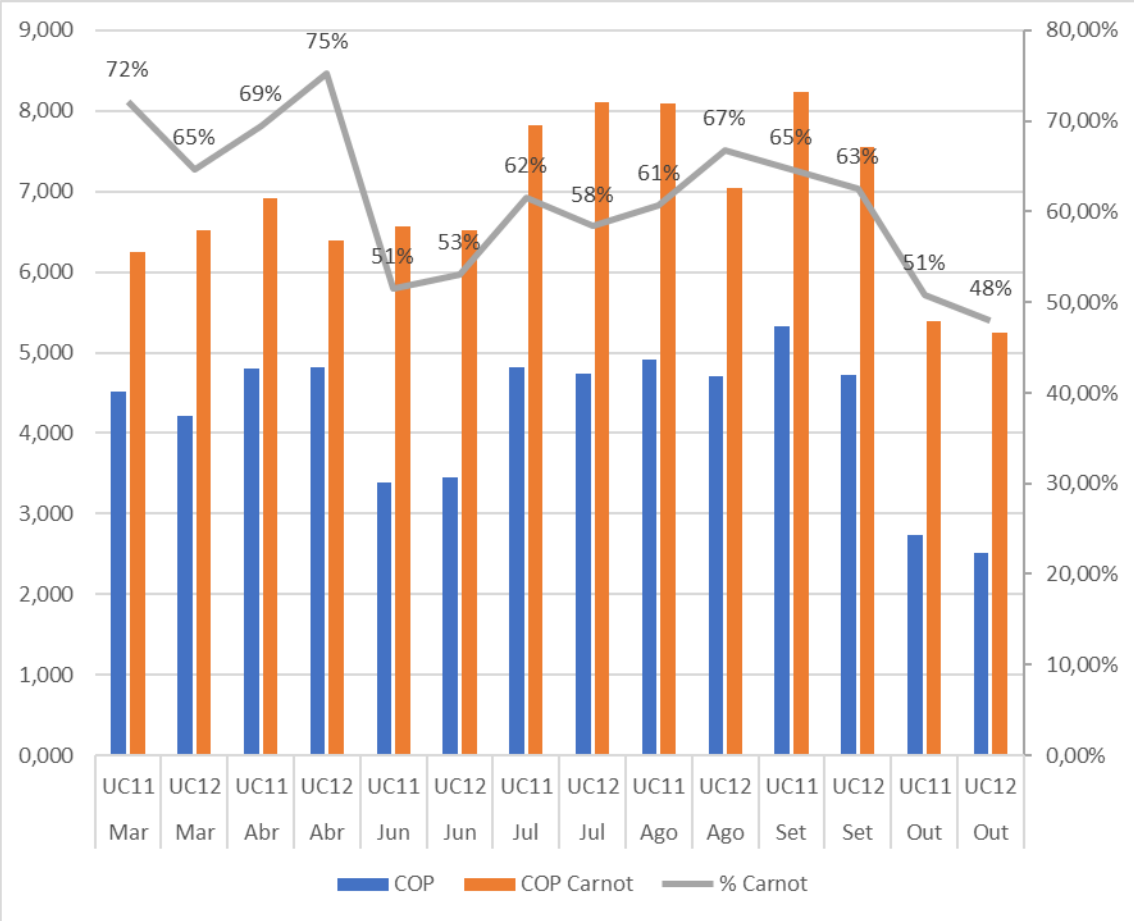
Graphic 2a – Rate of Exergy Destution at UC11.

Graphic 2b – Rate of Exergy Destution at UC12.



3.4 Energy Analysis

Graphic 3 – COP of the Cycle, COP Carnot, % Carnot.



It is perceptible 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 need predictive.

Another very important observation is that on the second law in heat exchangers indicate the proper functioning of condensers and evaporators because the heat flow coming out of the system is always greater than the coming in flow. That is, there is no leak or any other fault in the coils.

4. CONCLUSIONS

This model 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. The modeling was planned using the thermodynamic properties of said fluid, and taking into account the localized and distributed load losses in the feed lines of the Condensing Units (Suction Lines) and Evaporator Units (Liquid Lines) and the losses of single-phase and two-phase loads within the heat exchanger coils. In all, the properties of all the usual points of a real refrigeration system were calculated.

The properties were collected based on tables of thermodynamic properties supplied by the refrigerant manufacturer R404a, Dupont, plus some equations of the specific volume and viscosity supplied by the manufacturer Solkane. The Dupont property table was plotted on a Microsoft Office Excel 2016 software spreadsheet that made it possible to search directly or interpolate properties through a programmed algorithm in Visual Basic applied to Excel. The properties calculated with the Solkane formulas were then calculated in order to insert the already collected or interpolated temperatures (search and calculation algorithms in the Annex).

At the end of the modeling, it was verified that the modeled data were roughly matching what was verified by the measurements taken. This analysis is done in figure 27 through the properties of the 3 'point, measured in the system, versus the properties of the 3'_PC point which were determined in the modeling.

It is concluded that even using two-phase empirical loss-of-charge equations and the approximations considered, the criteria adopted together with measurements / readings were well used to construct a suitable model that could provide better cost and logistics a warning system if any component is damaged.

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