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MATHEMATICAL MODELING OF AN AIR CONDITIONER TO SIMULATE THE COOLING OF SPACES

Flávio Júnior Santiago Silva
Diego de Sousa Lima
Iago Gomes Costa
Department of Mechanical Engineering, Nucleus of Research and Development in Self-sustaining Energy (NPDEAS), UFPR - Federal University of Paraná, CP 19011, 81531-980 Curitiba, PR, Brazil.
e-mails: flaviojrsantiago@gmail.com, diegofine@gmail.com and iago_gomes_costa@hotmail.com.

Wellington Balmant
José Coelho Viriato Vargas
Department of Mechanical Engineering, Nucleus of Research and Development in Self-sustaining Energy (NPDEAS), UFPR - Federal University of Paraná, CP 19011, 81531-980 Curitiba, PR, Brazil.
e-mails: wbalmant@gmail.com, vargasjvcv2@gmail.com.

André Bellin Mariano
Department of Electric Engineering, Nucleus of Research and Development in Self-sustaining Energy (NPDEAS), UFPR - Federal University of Paraná, CP 19011, 81531-980 Curitiba, PR, Brazil.
e-mail: andrebmariano@gmail.com

Abstract. *Absorption refrigeration technology was introduced to address some serious issues such as the energy crisis, increased fuel prices, and environmental problems associated with conventional compression refrigeration systems. This work presents mathematical modeling to simulate the air conditioning of two spaces. For this, the development of the model was based on the application of the principles of conservation of mass and energy in a transient regime for the simple control volumes. After the equation was developed, a computational code was developed in Fortran®90 software to obtain numerical answers according to the system parameters. Liked result, this article showed in the input profiles of the temperature of the thermal fluids of each control volume and the Coefficient of Performance (COP) for the first cooled space was 0.54 (COP = 0.54) and for the second space, the COP was 0.25 (COP = 0.25). In this way, it is possible to conclude that the proposed mathematical model can be used as a useful tool for simulation in other conditions of operation, control and, thus, avoid future problems in the installation of the system in environments, in addition to performing analyzes for optimization of the main components of an absorption refrigeration system.*

Keywords: *mathematical modeling, computer simulation, absorption refrigeration.*

1. INTRODUCTION

With the development of the economy and industry, the demand for energy has increased significantly, which has led to a drop in fossil energy reserves and consequently rising prices, and serious problems of environmental pollution (Tietenberg, 2016). In addition, it is expected that in the next ten years, there will be a continuity of the trend of acceleration of the growth of the Brazilian population according to the Secretary of Energy Planning and Development, which predicts an average growth rate of 0.6% between 2021 and 2031, going from about 214.1 million in 2021 to a level of 226.3 million inhabitants in 2031, and thus a significant increase in electricity consumption. And to meet this increase in electricity consumption in Brazil, the Ministry of Mines and Energy developed a Ten Year Expansion Plan (EP, 2021) for energy generation 2021 - 2031, which estimates an increase from 261 to 333 million tons equivalent to petroleum (toe) for energy generation in the country. In addition to this expansion forecast, the expansion plan is also intended for the generation of renewable energy through initiatives in solar and wind energy. However, even so, it will be far from reaching in 2030 the levels observed in developed countries necessary to meet the country's demand and mainly because it still has investments aimed at the exploitation of non-renewable energy (EPE, 2021).

Currently, this use of electricity is led by three large sectors, industrial, commercial and residential, because according to the last energy balance released by the Ministry of Mines and Energy, the industry sector showed an

increase of 3 million TEP (Tons Equivalent to Petroleum) or 36.6% in the final consumption of electricity, the commercial sector consumed 15.7%, and the residential sector 27.6%, thus becoming the 3 sectors that consume the most electricity. As a result, Brazil generated about 398.3 million tons of carbon dioxide released into the atmosphere. And Among these sectors, the subsectors stand out among the sectors mentioned above, according to their level of consumption.

In the residential sector, consumption of refrigerators was 7.7% and HVAC (Heating, ventilating and air conditioning) was 23.9%, while in the commercial sector, consumption of refrigerators was 14% and HVAC with 24.7%. This shows the significant impact that such systems have on the country's energy matrix. Regarding the industrial sector, electricity consumption by the HVAC and refrigeration subsectors varies according to the type of industry, but it is estimated that about 10% of the consumption is the product of these sectors. Thus, analyzing all the percentages mentioned, the HVAC and Refrigeration subsectors, are equivalent to about 31.60% in the residential sector and 38.70% in the commercial sector of total electricity consumption, respectively (DOE, 2021). With this, many studies and researchers are looking for a way to reduce the world's electrical energy consumption, as there has been substantial growth in the demand for refrigeration and air conditioning devices to meet various engineering and comfort requirements. According to the International Institute of Refrigeration (IIR), conventional vapor compression refrigeration (VCRS) systems consume about a fifth of all electricity generated worldwide (Nikbakhti et al 2020).

Currently, the best-known refrigeration systems are mechanical vapor compression, absorption refrigeration, air refrigeration, thermoelectric refrigeration, and thermomagnetic refrigeration (Hermes, 2006 & Martinho, 2013). Two refrigeration systems stand out for being the best known, the vapor compression refrigeration system that basically uses electrical energy for its operation. On the other hand, absorption refrigerators have the advantage of using a non-electrical energy source, ensuring flexibility for the designer/owner/operator (Herold; Radermacher and Klein, 2016). In other words, this type of refrigerator stands out precisely because of the development of new technologies that use other energy sources.
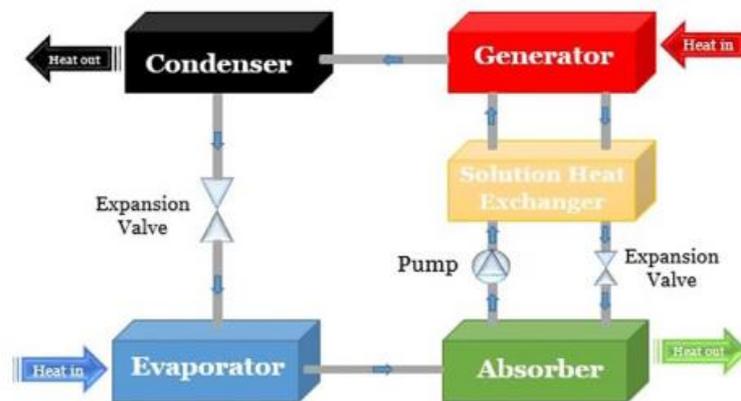

Figure 1: Basic cycle of operation of the absorption refrigeration system

Briefly, the absorption refrigeration system (Figure 1) operates according to a heat-fed refrigeration cycle, where a secondary or absorbent fluid in the liquid phase is responsible for absorbing the primary fluid or refrigerant, in the form of vapor. Heat-powered refrigeration cycles are so defined because the energy responsible for operating the cycle is mostly thermal. This type of refrigerator is formed by: a condenser, an evaporator, two expansion valves, an absorber, a mixing pump, a regenerating heat exchanger, a generator, and a rectifier. The consumption of electrical energy in an absorption cycle is minimal when compared to the cycle by mechanical vapor compression since only the pump uses this type of energy to raise the pressure of the refrigerant solution, with low specific volume, formed in the absorber. (Pereira, 2006).

The refrigeration cycle analysis can be calculated by the Coefficient of Performance - COP, which is the rate of cooling capacity, from the heat that is taken from the environment to be cooled, the rate of heat supplied to the system, and the energy or work to cool the environment (Martinez, 2018). In the case of the absorption refrigeration cycle, the energy used by the system is given by the sum of the amount of heat exchanged in the generator and the work done by the pump. Thus, the COP can be determined by:

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{ger} + \dot{W}_{pump}} \tag{1}$$

Where:

$\dot{Q}_{evap}$ represents the evaporator transfer rate;

$\dot{Q}_{ger}$ represents the transfer rate of the generator;

$\dot{W}_{pump}$ represents the pumping power.

This work aims to present a mathematical model for simulating the air conditioning of spaces through an ammonia absorption refrigeration system.

2. METHODOLOGY

With the objective of simulating the application of the refrigerator in spaces, a mathematical model was developed, in which the physical problem consists of an ammonia absorption refrigeration system with a capacity of 5 TR, a reservoir, and 2 cassette type fan coil, and two environments. Each of these components was considered as a single control volume (CV), in which the laws of mass and energy balance will be applied in order to simulate the profiles of temperatures that enter each control volume, such as the time that the thermal fluid passes through each CV and cools the spaces. The defined control volumes are described below

Control Volume 1: Refrigerator
Control Volume 2: Reservoir
Control Volume 3: Fan coil 1
Control Volume 4: Space 1
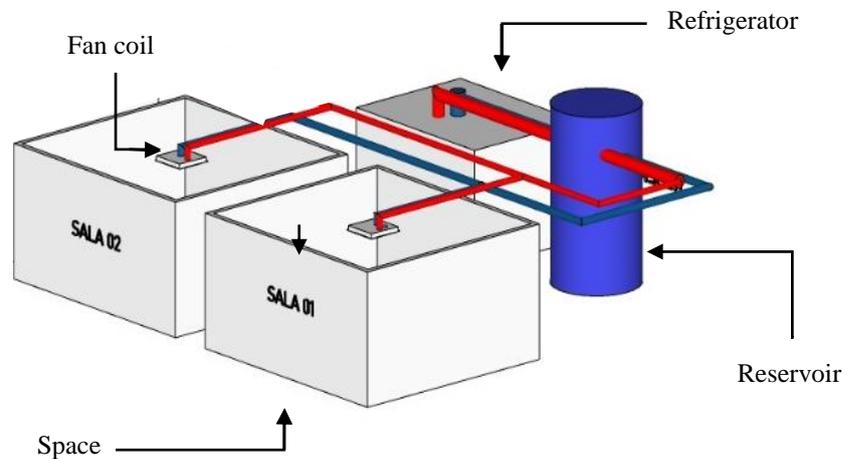Control Volume 5: Fan coil 2
Control Volume 6: Space 2

Figure 3: Scheme of operation of the refrigeration system for space cooling.

To calculate the equation of the proposed system and presented in Figure 3, the following initial considerations were made: i) The energy balance in the absorption refrigeration system will be performed only in the evaporator on the waterside, in a single volume, ii) The system will be modeled in a transient operation regime. iii) All thermal fluid that enters the fan coil and acclimatizes the room, returns to the reservoir independently.

2.1 Control Volume 1: Refrigerator

Initially, a mass and energy balance was carried out in CV_1 (Refrigerator), where the mass and energy flows that enter and leave this first volume of control, regarding the heat loss that occurs in the process, as shown in equation (2). Consequently, the other processes that occur in the refrigerator are modeled based on the same principle.

$$E_r = m_r.C_w.T_r \tag{2}$$

$$\frac{dE_r}{dt} = \dot{m}_{re}.C_w.T_{re} - \dot{m}_{re}.C_w.T_{re}.\dot{Q}_r \tag{3}$$

$$\frac{dE_r}{dt} = \frac{\dot{m}_{re}.T_{re} - \dot{m}_{re}.T_r}{m_r} - \frac{\dot{Q}_r}{m_r.C_w} \tag{4}$$

After identifying the elements that enter and leave the volume, equation (3) demonstrates the application of the mass and energy balance, where the input parameters are based on the energy produced in the refrigerator as a function of

time, represented by the mass flow rate ($\dot{m}_{re}$) of the thermal fluid entering the refrigerator, the (C_w) the specific heat, and the temperature initial that the thermal fluid enters the refrigerator (T_{re}), as well as the parameters that leave the refrigerator, being the same items that enter: the flow of water ($\dot{m}_{re}$) that the fluid leaves the refrigerator, the specific heat, as well as the temperature at which the thermal fluid leaves the refrigerator. Emphasizing the refrigerator's cooling capacity ($\dot{Q}_r$) of 17,580 kW. In equation (4), the temperature of the same parameters is modeled as a function of time.

2.1 Control Volume 2: Reservoir

For the equation of CV$_2$, this control volume is characterized by two steps, that is, for the mathematical modeling this component was considered the same factors of the previous control volume, but changing only input parameters for a water reservoir, where there is an input of the thermal fluid, the output of this fluid towards the Fan coil (CV$_3$), by the return of this fluid from CV$_3$ to CV$_2$ and the exit of the thermal fluid towards the refrigerator (CV$_1$). So we have equation (5) that guides the mathematical modeling of the control volume 2. Where, (E_t) represents the energy produced in the reservoir, the specific mass flow rate that the thermal fluid entering the reservoir ($\dot{m}_{re}$), the specific heat (C_w) and the temperature that the fluid produces (T_{re}) at CV$_2$.

$$E_{re} = m_{re}.C_w.T_{re} \tag{5}$$

$$\frac{dE_{re}}{dt} = \dot{m}_{re}.C_w.T_r - \dot{m}_{re}.C_w.T_{re} + \dot{m}_f.C_w.T_f - \dot{m}_f.C_w.T_{re} + U_{re}.A_{re}.(T_a - T_{re}) \tag{6}$$

$$\frac{dT_{re}}{dt} = \frac{\dot{m}_{re}.T_r - \dot{m}_{re}.T_{re} + \dot{m}_f.T_f - \dot{m}_f.T_{re}}{m_{re}} - \frac{U_{re}.A_{re}.(T_a - T_{re})}{m_{re}.C_w} \tag{7}$$

Equation (6) is integrated as a function of time in VC$_2$, in which the volume input parameters are first assigned, being: ($\dot{m}_{re}.C_w.T$) mass flow, specific heat and the temperature at which the thermal fluid enters the reservoir, with the same fluid coming from CV$_1$, as well as the fluid leaving the reservoir represented by the nomenclatures ($\dot{m}_{re}.C_w.T_{re}$), and consequently, the CV$_2$, receives the return fluid coming from the fan coil, where the mass flow of the fluid coming from the fan coil is now calculated ($\dot{m}_f$), the specific heat (C_w) and the temperature that thermal fluid to the reservoir (T_{re}) also, by the Global Coefficient of the Reservoir (U_{re}) identified through the literature, as the area of the reservoir (A_{re}) that directly influence the behavior of the system the temperatures, environment (T_a), and the reservoir itself (T_{re}).

Thus, equation (7) consequently assigns the temperature of the thermal fluid being integrated as a function of time. Initially, we have the same input and output equations of the system (CV$_2$), and by adding to ($U_{re}.A_{re}.(T_a - T_{re})$) equivalent to the process that occurs when the fluid leaves the reservoir at a lower temperature towards the cassette fan coil (VC$_3$), that is, the fan coil will receive this fluid, already with the lowest water temperature. This refrigerant fluid acclimates the environment and soon after with a more ambient temperature, it will return to the reservoir, taking into account the Global Reservoir Coefficient (U_{re}), the area of this reservoir (A_{re}), and the ambient temperature at which the fluid cools the room and the temperature of the water in the reservoir.

2.3 Control Volume: Fan Coil 1

For the equation of the control volume 3 (CV$_3$), firstly, some system operation information is presented. i) the CV$_3$ corresponds to the first fan coil that uses a chilled water system to cool the environment and is more compact and robust in its design. ii) The fan coil is inside the environment in which the air conditioning will be carried out.

Thus, similarly to the equation already developed, for the mathematical modeling of CV$_3$, the same factors of CV$_1$ and CV$_2$ were considered, modifying the component that produces energy, being the first fan coil, where we have the mass flow (m_{f1}) of the fluid that enters the fan coil, the specific heat (C_w) and the temperature (T_{f1}) that the thermal fluid enters the fan coil as shown in equation (8).

$$E_{f1} = m_{f1}.C_w.T_{f1} \tag{8}$$

$$\frac{dT_{f1}}{dt} = \frac{\dot{m}_{f1}.T_{re}}{m_{f1}} - \frac{\dot{m}_{f1}.T_{f1}}{m_{f1}} + \frac{U_f.A_{f1}.(T_{s1} - T_{f1})}{m_{f1}.C_w} \tag{9}$$

In equation (9), the integration of the thermal fluid temperature of fan coil 1 (dT_{f1}) as a function of time (d_t) is obtained, where the thermal fluid from the reservoir ($\dot{m}_{re}.T_{re}$) enters the fan coil ($\dot{m}_{f1}$) and consequently the cooling process takes place in the first space (T_{s1}), also considering the temperature at which the thermal fluid enters the fan coil (T_{f1}) as the area of this fan coil ($Af1$). The same fluid leaves fan coil 1, now at room temperature, and returns to the reservoir. Considering this equation the Global Coefficient (U_f) is determined for the fan coil.

2.4 Control Volume: Space 1:

To model control volume 4 (CV4), the following information is considered: the calculation of the thermal load for the first space ($\dot{Q}_{s1}$) is 28.967 BTU/h or 8.49 kW, the temperature environment ($\dot{Q}_a$), which was considered a temperature of 30ºC and how much heat ($\dot{Q}_{f1}$) the fan coil takes from the environment.

$$E_{s1} = m_{s1}.CV_{ar}.T_{s1} \tag{10}$$

$$\frac{dT_{s1}}{dt} = \frac{U_s.A_{s1}.(T_a - T_{s1})}{m_{s1}.CV_{ar}} - \frac{U_f.A_{f1}.(T_{s1} - T_{f1})}{m_{s1}.CV_{ar}} + \frac{\dot{Q}_{s1}}{m_{s1}.CV_{ar}} \tag{11}$$

Equation 10 presents the main equation for CV$_4$, defined by the energy that environment 1 generates (E_{s1}), the specific heat of air, volume, and constant (CV_{ar}), as the standard temperature defined for space 1 (T_{s1}), and thus, it guides equation (11), where the temperature at which the thermal fluid enters the environment 1 (T_{s1}) is integrated as a function of time (dT). The equation is first developed for environment 1, as a function of the global coefficient (Us) for a space 1, the area of the environment ($As1$) that directly influences the cooling process of the environment, and their respective temperatures between the room and the external environment ($T_a - T_{s1}$), divided by the flow rate that the thermal fluid enters the room and by the air constant ($m_{s1}.CV_{ar}$). Soon after, there is the equation that models the fan coil 1 within the environment, also having the global coefficient for the fan coil (U_f), the area of the fan coil 1 (A_{f1}), with the temperature difference between the environment 1 and fan coil 1 as well ($T_{s1} - T_{f1}$) and dividing also by the flow rate that the thermal fluid enters the room and by the air constant ($m_{s1}.CV_{ar}$). And the ($\dot{Q}_{s1}$) corresponds to the thermal load that the environment produces.

2.5 Mathematical Modeling of Control Volume 5 and 6

For the modeling of the other control volumes, the same ammonia absorption refrigerator, the same air conditioning capacity of 5 TR, and the same reservoir with a capacity of 500 liters were maintained, and as shown in Figures 12 and 13, each fan coil has a and return outlet of the thermal fluid individually to the reservoir. And consequently, the cooling capacity in the environment is lower, as it has already been used for air conditioning of an environment.

2.6 Mathematical model input parameters for simulation

To start the simulation of the mathematical modeling, it was necessary to define some of the input parameters of the mathematical model, which these input values are presented in Table 1 and described during the mathematical equation for each control volume. The global heat transfer coefficients were determined according to the literature (Engineering ToolBox, 2003). In this way, the influence of these parameters on the air conditioning in the environment, and the inlet temperatures of the thermal fluid of each component will be analyzed and later a parametric analysis will be carried out to verify the operational performance of the control volumes.

Table 1: Input parameters for model simulation

Parameters	Symbols	Value	Unit
Reservoir mass flow	$\dot{m}_{re}$	0.5	kg/s
Specific heat of water	Cw	4180	$J.kg^{-1}.K^{-1}$

Refrigerator Heat Transfer	$\dot{Q}_r$	17.580.3	kW
Water density	ρ_w	1000	kg/m^3
Refrigerator Volume	V_r	0.005	m^3
Mass Flow of Fancolete	$\dot{m}_f$	100	kg/s
Reservoir Volume	V_{re}	0.5	m^3
Fancolete Volume	V_f	0.005	m^3
Fancolete Global Coefficient	U_f	100	w/m²k
Fan coil Area 1	A_f	2	m^2
Fan coil Area 2	A_{f2}	2	m^2
Space Volume 1	V_{s1}	30	m^3
Space Volume 2	V_{s2}	30	m^3
Global Coefficient of Spaces	U_s	1	w/m²K
Space Area 1	A_{sl1}	48	m^2
Space Area 2	A_{sl2}	17	m^2
Heat Generation in space 1	$\dot{Q}_{sl1}$	9.376	kW
Heat Generation in space 2	$\dot{Q}_{sl2}$	2.22	kW
air density	ρ_a	1.2	kg/m^3
Specific heat of air at constant volume	CV_{ar}	717	J.kg^{-1}.K^{-1}
Environment temperature	T_a	30	°C
Global Cooler Coefficient	U_r	500	w/m²K
Refrigerator Area	A_r	10	m^2
Refrigerator Saturation Temperature	T_{st}	-18.9	°C

2.7 Numerical Method

The numerical problem to be solved consists of numerically integrating the ordinary differential equations (ODEs), which describe the behavior of the system. The mathematical model was implemented, computationally, through the language of Fortran®90 programming. The ODEs were integrated explicitly in time using adaptive steps and using the 4th/5th order Runge-Kutta method (Kincaid and CheneY, 1991). Numerically, the time step is automatically adjusted according to the local truncation error, which is kept below a specified tolerance of 10^{-6}. As stopping criteria, the computational code developed allows calculations to be performed up to a pre-established final time or steady-state condition (Martinho, 2013).

3 RESULTS AND DISCUSSIONS

3.1 Refrigerator simulation for an environment

The mathematical model was implemented in Fortran software, as the mass and energy balance was applied using the equations and initial conditions described in the methodology of this article. With the given parameters, the program calculated the values of the inlet temperature of the thermal fluid of the control volumes: CV_1, CV_2, and CV_3, as well as simulated the temperature at which the fluid cools the environment 01 (CV_4) exposed by Figure 4. The system modeled to cool only one environment obtained a coefficient of performance of 0.546 (COP = 0.54), which the value is considered acceptable as COP values presented in experimental approaches for this type of refrigerator according to the literature.
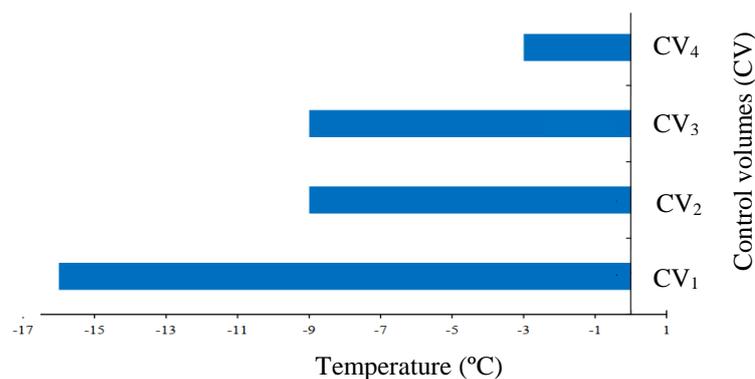

Figure 4: Thermal fluid Inlet temperature for a 1pace

Still in figure 4, in the numerical simulation of the system, the cooling temperature of each component was observed, that is, the inlet profile of the thermal fluid in which the system reaches the steady state stage. Firstly, it was simulated with a standard time of 8.000s, in order to verify if the time would be ideal to carry out the simulation, which proved to be the standard to be followed, where the CV_1 presented an inlet temperature of 30ºC, which decays according to with time, after 1.000s the system shows temperatures below -10ºC, and when it reaches 2000 seconds, the system enters a steady state with a temperature of -16ºC. CV_2, on the other hand, had a temperature of -11ºC, reaching the steady state stage. The CV_3 showed a similar result, due to the fact that the thermal fluid that leaves the reservoir towards the fan coil is at a negative temperature and with no loss. The CV_4, which refers to the air conditioning of the 1st space that starts with a temperature of 30ºC and enters a permanent regime after 2.000s. During the simulation period, it was possible to notice that before 2.000 seconds the system already presented a negative reduction in relation to temperature, which reached the steady state level.

3.2 Cooling capacity for a space

In order to simulate the refrigerator capacity of 5 TR in relation to the air conditioning of the first environment, the initial thermal load of 10.000 kW was modified, with intervals of 5000 kW, reaching up to 45.000 kW as shown in Figure 19. Firstly, it is noted that at 10.000 kW the refrigerator can cool the temperature below 0°C, when its thermal load is increased at different points, it can be seen that the graph consequently increases the room's air conditioning temperature. When the environment adopted the thermal load of 30.000 kW, it is still possible to have a thermal comfort of 20ºC, which behaves between 20 to 35 people. When this amount is exceeded, the room temperature reaches 25ºC, thus, there is no thermal comfort (Figure 5).
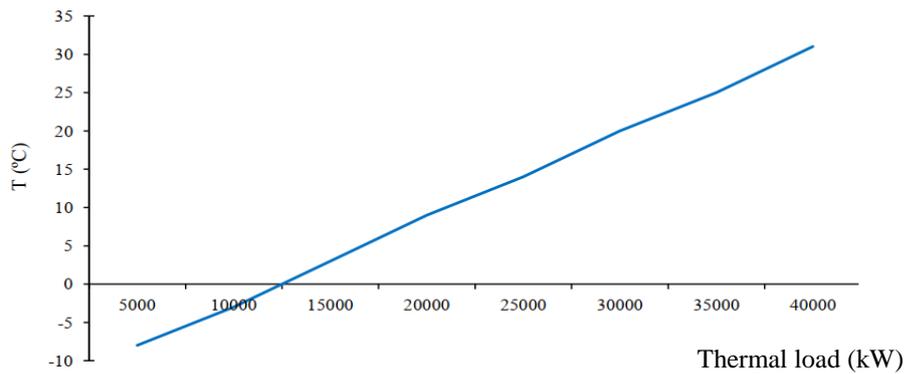

Figure 5: Refrigerator air conditioning capacity.

Figure 6 shows the result of the variation in the volume of the reservoir, as a function of the COP, the volume was changed, which was described in table 1, when the system obtained a COP of 0.53 when the volume adopted a 500 m^3, even being an acceptable COP according to the literature for the absorption refrigeration system. However, volumes were used smaller capacity as shown in the figure below. The coefficient of performance obtained better performance when the system used a volume lower than 100 m^3, that is, a container of lower capacity directly influences the cooling capacity of the environment.
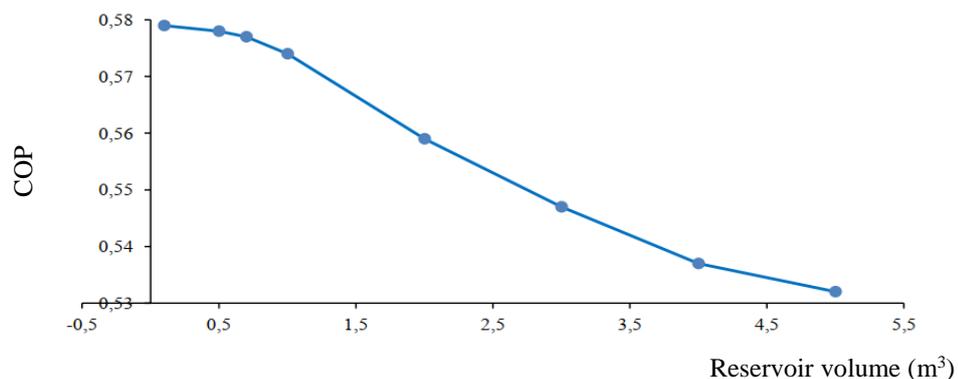

Figure 6: Reservoir volume variation.

Figure 7 shows the parameters used for the system to operate, starting with flows above 100 kg/s and thus increasing the others. Firstly, a good performance is noted when flow rates higher than those used in the methodology were adopted, so the system presented excellent COP values when mass flows of 200 kg/s were adopted, reaching an optimal flow at 300 kg/s and COP of 0.69. With this, it is possible to state that the higher the mass flow rate adopted for this fan coil, the better its efficiency, but it was also observed that each time higher flow rates were used, the system its efficiency declined again. Thus, it is understood that the ideal flow when the system uses the maximum load, the best flow is 300 kg/s.
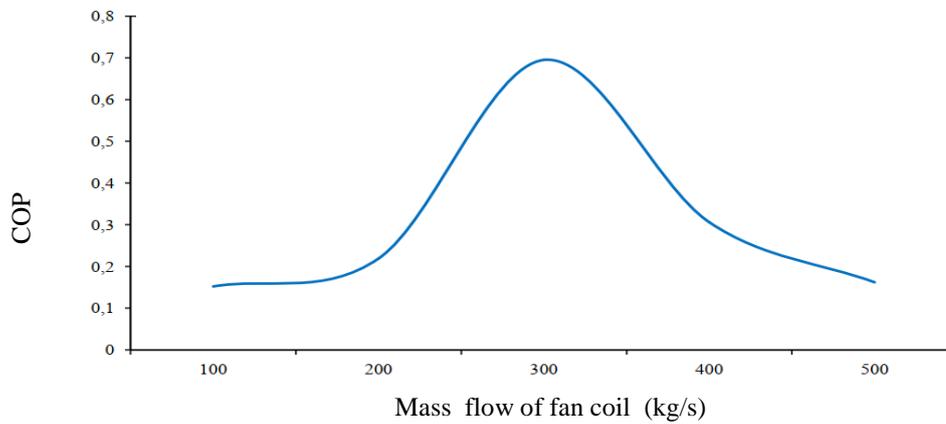
Figure 7: Analysis in function of fan coil flow

3.3 Refrigerator simulation for two spaces

To simulate the application of the refrigerator in two environments (CV_6), some of the same parameters of the first space were kept, and the thermal load calculated for environment 2 was added, which is 2.22 kW. The simulation showed a similar temperature for CV_1, when using two environments for cooling, that is, there was no change since the refrigerator is the same, and in CV_2 the temperature was lower, dropping to -7°C, and VC_5 obtained the same CV_2 temperature. In the CV_6, the temperature was -6°C, as expected because it is a smaller environment with only 15.5 m^2, little circulation of people in the environment and for having fewer factors that directly impact heat generation (Figure 6), but in relation to the refrigerated performance for the cooling of two environments, the COP dropped to 0.0258 (COP = 0.025).
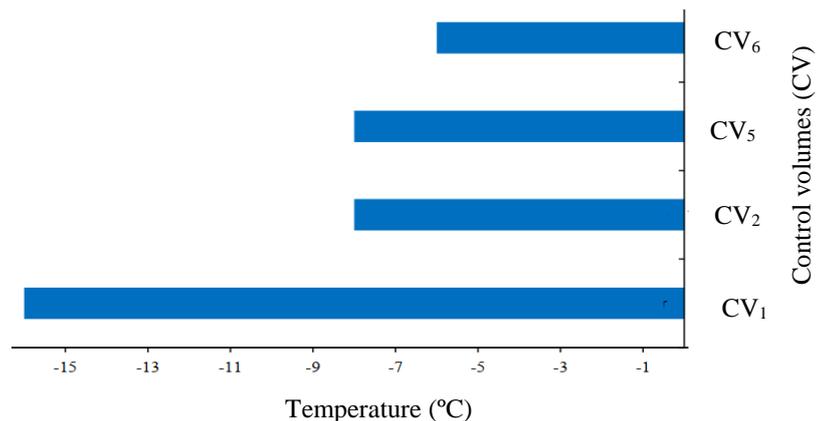

Figure 8: Temperature profiles when two-space refrigerated.

3.4 Cooling capacity for 2 spaces

In figure 8, the temperature variation when changing the thermal load of space 2 (CV_6) as a function of time was analyzed. The load of this environment has already been calculated and presented a heat generation of 3.000 kW was used to simulate the mathematical model. Thus, to carry out the analysis, the variation of loads of the two environments was carried out, where it was necessary to vary these loads and thus identify the maximum value supported by the CV_6, and also considering the maximum load found for the CV_4 that presented a maximum load of 30.000 kW to maintain thermal comfort. Thus, variations with a value range of 2.000 kW were analyzed, starting from the standard load

calculated previously. When the ambient load exceeded 18.000 kW, the system presented a temperature above the allowed ideal, which lost all comfort. Up to 18.000 kW, the model exhibited a temperature of 23ºC, which is ideal for this environment according to the ABNT NBR 16401-1. Air-Conditioning Installations – Central and Unitary Systems, 2008. Based on this, the refrigerator can maintain its cooling quality in the environments (CV_4 and CV_6) at their maximum loads and still make them pleasant, since this change will not compromise the functioning of the system of air conditioning.
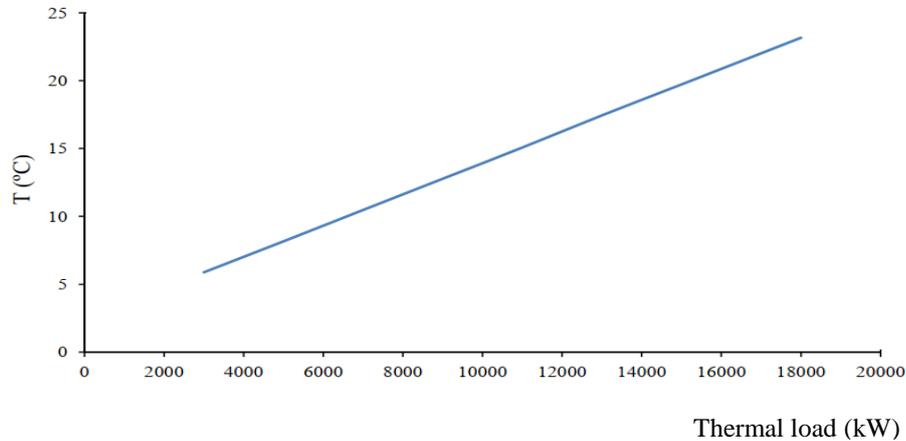
Figure 9: Refrigerator cooling capacity for 2 spaces.

Figure 40 shows the optimization of the fan coil mass flow when the system admitted the maximum load for the VC6 of 18.000 kW. Initially, when it adopted flow rates between 0.1 and 10 kg/s, the system performed below expectations for an absorption refrigeration system, with a COP of 0.207 (COP = 0.2). As a result, variations from 40 kg/s were carried out, throughout the changes it was noticed that when the system adopted a flow rate of 80 kg/s, the model obtained a better performance, thus obtaining a COP of 0.416 (COP = 0.4) and when this flow was exceeded, the system reduced its COP, thus note that the best flow rate for the system when the CV_6 is at its maximum load is 80 kg/s.
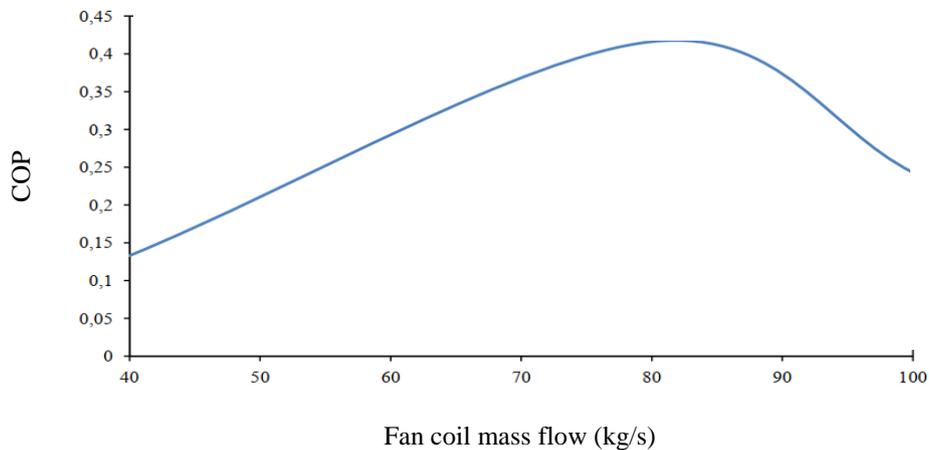
Figure 10: Optimization of fan coil mass flow for maximum load.

4. CONCLUSION

With the models developed for the two environments, numerical simulations were performed via the Fortran®90 software to simulate the entrance of the thermal fluid in the control volumes and predict the cooling capacity in the spaces until reaching the initial condition until reaching the state stationary. Where the main conclusions of this article are: i) The modeled system was able to cool an environment 1 and obtained a coefficient of performance of COP 0.546 (COP = 0.54), ii) The refrigerator can cool an environment with a thermal load of up to 30.000 kW, because until then, it is possible to have a thermal comfort of 20ºC, iii) The refrigerator was able to cool both environments, but its performance for the COP was COP of 0.258 (COP = 0.25), iv) In VC6, the temperature was –6ºC, as a consequence of being a more compact environment and the cooling time was similar to that presented for an environment, v) The

refrigerator managed to cool both environments under thermal load, the second environment supporting up to 18.000 kW, when the simulation presented a temperature of 23ºC, vi) For the maximum thermal load for two environments, the system adopted a flow rate of 80 kg/s, which presented the best performance, thus obtaining a COP of 0.416 (COP = 0.4) and vii) The coefficient of performance obtained better performance when the system used a reservoir volume of less than 100 m^3. In this way, it is expected that this mathematical modeling can be used as a possible and efficient computational tool for accurate simulations in environments, sizing, and understanding the best operation for absorption refrigeration systems.

5. ACKNOWLEDGEMENTS

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