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DESIGN OF AN FINNED TUBE EVAPORATOR WITH CROSS-FLOW FOR A REFRIGERATION MACHINE OPERATING WITH CO_2 IN SUBCRYTIC CYCLE

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Abstract. *The emission of some gases used in the refrigeration has caused a worrisome environmental impact, as they cause the disintegration of the ozone layer. CO_2 - Carbon Dioxide (R744) has been the subject of research in recent years because it is a refrigerant fluid that is not aggressive to the environment and presents good thermodynamic properties. In the design of an air-water refrigeration machine using components that operate in a subcritical cycle, the specific design of one of the heat exchangers was abalyzed, the crossed-flow finned tube type evaporator. The mathematical modeling of the project was programmed in the MATLAB® software (MATrix LABoratory), aided by the EES® software (Engineering Equation Solver). A complete approach to the evaporator design was performed by analyzing the convective coefficients (monophasic and biphasic) of heat exchange of the refrigerant and secondary fluid (air), through correlations in the literature. The cooling capacity of the evaporator was 3.47 kW, working at a pressure of 19.7 bar and the evaporation temperature of - 20 °C, its dimensions was obtained in which the total length of the tube was 14 m. The idealized machine has a cycle with a COP (Performance Coefficient) of 3.47, being designed for a research laboratory and will allow the production of cold air at 8.1 °C that can be used to aid a room air conditioning system of laboratories and can also be applied in laboratory tests of domestic refrigeration machines.*

Keywords: *Refrigeration machine, CO_2 (R744), finned tube evaporator.*

1. INTRODUCTION

The emission of some gases used in refrigeration has caused a disturbing environmental impact because it provides the disintegration of the ozone layer that protects the planet from ultraviolet solar radiation. This situation characterizes a recurrent need to use less aggressive gases, and which have the minimum efficiency necessary to guarantee their insertion in the market (Faria, 2013).

CO_2 - Carbon Dioxide (R744) is a natural fluid that exists in abundance in nature. It has been the subject of research in the last years because it is a refrigerant fluid that not only is not aggressive to the environment, but also presents high enthalpy and specific mass variation when compared to the most commonly used fluids (Oliveira, 2013).

In the market there are in abundance equipment that works on the basis of conventional refrigerants, such as CFCs (chlorofluorocarbons) and HCFCs (hydrofluorocarbons). However, it is challenging to develop refrigeration machines that work with CO_2 and can contribute to current research, as well as advocating a very promising and innovative idea in the refrigeration industry (Lorentzen, 1994).

There is little information in Brazil about manufacturers of refrigeration systems components and heat pumps operating with CO_2. This is an important alternative because it is a natural fluid, not aggressive to the ozone layer and has some thermal properties, such as the latent heat of vaporization, which makes its application viable in refrigeration (Silva, 2016).

The high working pressure of CO_2 requires that the design of the installation and safety measures is made with special criteria, requiring a greater specialization of all factors and components involved in the system, from design, implementation, installation, operation and maintenance (Pereira and Primo, 2012).

To Cleto (2008), one of the major advantages of using CO_2 in refrigeration is due to the reduction in the load of fluids that present the greatest restrictions of use (ammonia and hydrocarbons), however, it is essential that the CO_2 system presents an energy efficiency level equal to or better than a conventional system for the same application.

This work aims to analyze the specific design of one of the heat exchangers of the refrigeration machine in question, the evaporator. This equipment is of the cross-flow finned type, where the primary fluid (CO_2) flows inside the finned tubes which are arranged in serpentine form and the secondary fluid (air) flows in cross flow between the finned tubes aided by ventilation forced by an exhaust fan.

2. METHODOLOGY

In order to analyze CO_2 as a cooling fluid, it was proposed the design of an air-water refrigeration machine using components that operate in a subcritical cycle, allowing the development of research that may contribute scientifically to the rise of R744. This will also allow the study of thermodynamic cycles, better choices of evaporation and condensation temperatures to optimize COP (Coefficient of Performance), analysis of heat exchange coefficients in heat exchangers and influence of hot and cold source conditions on the cycle operation.

The idealized machine will be installed in a research laboratory and will allow the production of cold air (through the evaporator) that can be used to aid an air conditioning system of laboratory rooms and can also be applied in laboratory tests of domestic refrigeration machines.

In this section it is presented the cycle of operation and the components of the idealized refrigeration machine, besides the mathematical modeling of the finned tube evaporator.

2.1 Subcritical cycle of CO_2 and main components of the system

A refrigeration cycle is a diagram that demonstrates in a schematic way the complete system, besides representing the behavior of the refrigerant throughout this system.

The subcritical cycle is the most commercially widespread when it comes to the use of CO_2 as a refrigerant fluid. According to Zhang et al. (2011), practicable values for condensing temperature and evaporation in a CO_2-based cooling system would be, respectively, 20 °C and -20 °C. The compression process is assumed to be adiabatic because no heat exchange occurs with the external environment. With respect to the condensation and evaporation pressures, these are, generally, about 40 bar and 25 bar, respectively (Silva, 2008).

In general, the machine under analysis was designed to use a reciprocating semi-hermetic compressor of 1.08 kW power based on pre-set condensation and evaporation temperatures of respectively 10 °C and -20 °C. The compressor will be coaxial flow type countercurrent tubes with a heat exchange rate of 4.82 kW. The expansion device will be an electronic valve. The schematic of the idealized machine and the components required for its operation are shown in Figure 1.

2.2 Design of the evaporator

The evaporator is the refrigerating machine component in which the refrigerant fluid undergoes a change of state from the liquid phase to the gas phase. Being the most important component of a refrigeration system, since it is responsible for the removal of heat from the environment, the efficiency of the system depends directly on the design and proper operation of the same.

According to Zhang et al. (2011) overheating temperatures between 0 °C and 20 °C are common in CO_2 refrigeration machines operating in subcritical cycles and are important in preserving the compressor life. There is a variation of the COP with the rise of the overheating temperature. Faced with such a situation, in this project an overheating temperature of 10 °C will be considered. There will be a reduction in the COP, however this measure is fundamental to avoid the entry of liquid in the compressor.

Among the possible air evaporators, the serpentine model with finned tubes and crossflow was chosen for this project in order to improve the heat exchange between the fluids. The type chosen can be seen in Figure 2, where heat exchange occurs with the fins delimiting the air flow and the tubes, together with the fins, allowing the thermal exchange between them.

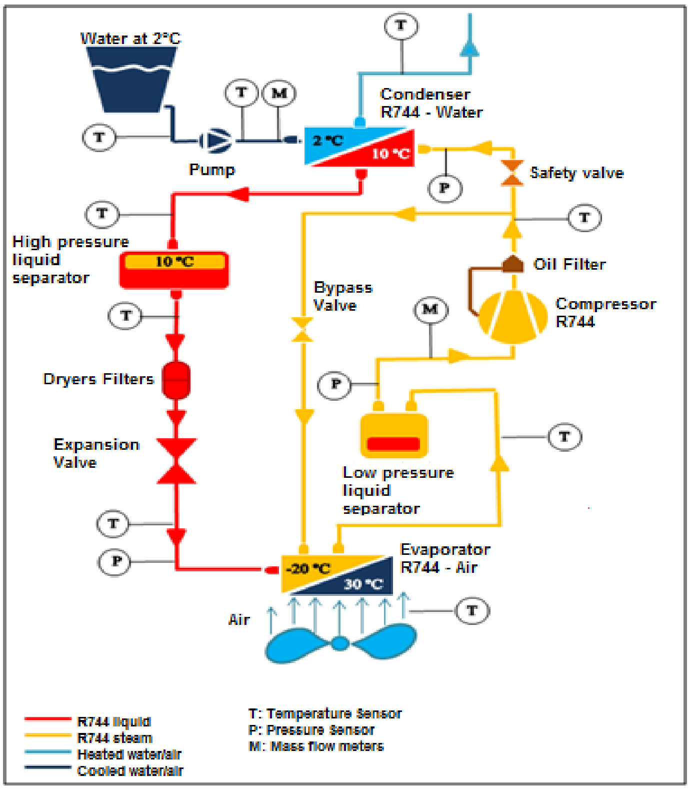

Figure 1: Idealized refrigeration machine.

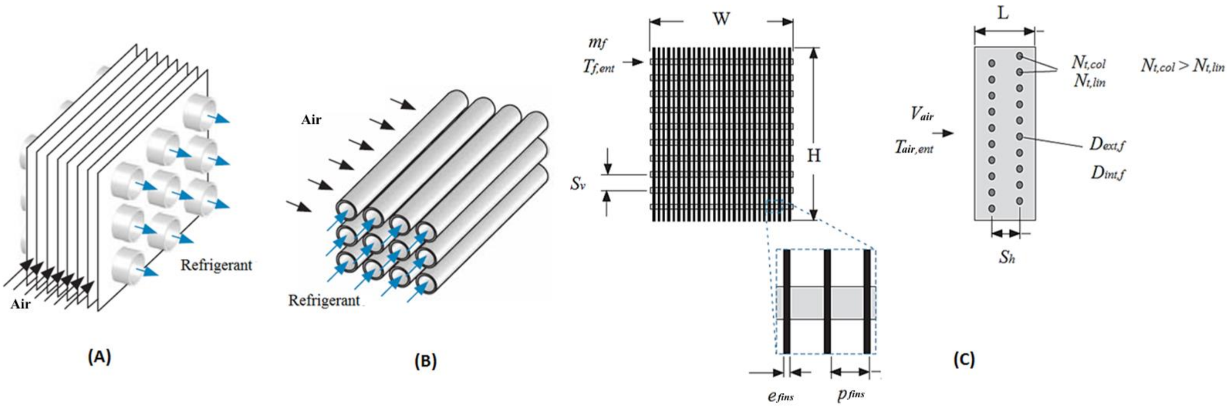

Figure 2. Heat exchanger, (a) with finned cross-flow, (b) without fins and (c) schematic to aid in the design.
Source: Nellis and Klein (2009)

According to Stoecker and Jones (1985), when the temperature of an evaporator, which cools air, drops below 0 °C, surface ice will occur. Ice is damaging because it behaves as an insulator when it forms thicker layers, preventing thermal exchange, and reducing airflow in forced convection coils.

To solve this impasse, Althouse et al. (2003) proposes the implementation of a bypass system using an electronic operated solenoid valve. When the valve is open, the refrigerant causes defrost in the evaporator due to the heat exchange with the hot gas leaving the compressor. For the operation of the valve, a timer is used to release the electric current that causes it to open, thereby allowing the hot gas to pass through. At the end of the count, the current is cut off and the valve closes, ending the defrost.

For the forced convection it will be necessary to use a fan of the axial type. Since the ability to draw heat from the environment by the cross flow evaporator depends directly on the speed of air produced in the turbomachine, handling that velocity is of utmost importance. For this, it becomes necessary to use an electronic device, the frequency inverter, which is the support of induction motor driven systems that require variable rotation.

2.3 Modeling procedures

In this section a complete approach of the evaporator design is made, stratifying the flow of the refrigerant fluid in two regions: evaporation and overheating. The determination of the convective coefficients of heat exchange of the refrigerant (single phase and biphasic) and secondary fluid (air) was performed using the correlations present in the literature, and the responses were compared for the biphasic region (CO_2) and defined the most appropriate to the project. In the region of single-phase CO_2 flow, the correlations of Dittus-Boelter, Sieder and Tate, and Gnielinski were used. The correlations developed by Kim, Youg and Webb were used for the air flow. In the biphasic flow region (CO_2), the Kandlikar and Jung correlations were used.

2.3.1 Single-phase convective CO_2 coefficient in the evaporator

In the evaporator, coolant flows into the tube. Eq. (1) and Eq. (2) are presented by Incropera (2008) first considering a laminar flow of the fluid.

$$Nu = \frac{H_f D_i}{k_f} = 4.364 \tag{1}$$

$$Re_D = \frac{\rho u D_i}{\mu} \tag{2}$$

Where Nu is the Nusselt number, Re_D é is the Reynolds number ($Re \leq 2300$) for the fluid, u is the flow velocity and μ is the dynamic viscosity. H_f is the convective coefficient and k_f is the thermal conductivity of the fluid. Finally, D_i is the inner diameter of the tube through which the fluid flows.

Also according to Incropera (2008), if the flow is turbulent and fully developed, the following three correlations are normally used. Empirical correlations are used because of the complexity in analyzing the conditions of this type of flow.

The first one is that of Dittus-Boelter, given by Eq. (3).

$$Nu_D = 0.023 Re_D^{\frac{4}{5}} \mathrm{Pr}^n \tag{3}$$

In wich $0.7 \leq Pr \leq 160$, $Re_D \geq 10000$, $\frac{L}{D} \geq 10$, $n = 0.3$ for heating ($T_s > T_m$), as in the condenser and $n = 0.4$ for cooling ($T_s < T_m$), as in the evaporator. Being L the length and D is the diameter of the tube, T_s is the temperature of the fluid at the outlet and T_m is the mean temperature of the fluid. Finally, Pr is the Prandtl number, given by Eq. (4).

$$Pr = \frac{\mu C_p}{k} \tag{4}$$

Where C_p is the specific heat and k is the thermal conductivity of the fluid.

The second correlation is that of Sieder and Tate, given by Eq. (5).

$$Nu_D = 0.027 Re_D^{\frac{4}{5}} \left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{5}$$

In wich $0.7 \leq Pr \leq 16700$, $Re_D \geq 10000$ and $\frac{L}{D} \geq 10$.

The third correlation is that of Gnielinski, given by Eq. (6).

$$Nu_D = \frac{\left(\frac{f}{8}\right)(Re_D - 1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{6}$$

Where $0.5 \leq Pr \leq 2000$, $3000 \leq Re_D \leq 5x10^6$ and $\frac{L}{D} \geq 10$. The friction factor f can be calculated by Eq. (7), developed by Petkhov for plain tubes and can be found in Machado (1996). The subscript "s" refers to output. g is acceleration of gravity.

$$f = (0.790 \ln(Re_D) - 1.64)^{-2} \tag{7}$$

As seen above in the Reynolds number ranges for applying the equations, care must be taken with the transition region ($2300 \leq Re_D \leq 10^4$). When $Re_D > 10^4$ the flow is fully turbulent, applying the Dittus-Boelter equation. In situations where the flow is in the transition region, the Gnielinski correlation will be applied to ensure greater accuracy and to avoid that the convective coefficient is overestimated.

2.3.2 Correlations for the convective coefficient of the air without evaporator

In Nellis and Klein (2009) it is found the generic equations to obtain the convective coefficient of air as a function of the number of Stanton S_t. The complexity lies in the definition of the Colburn Factor, J, which depends directly on the geometry of the exchanger. This factor can be obtained by means of empirical correlations for a specific geometry, according to Eq. (8) and Eq. (9).

$$H_{ar} = GC_{p,air}S_t \tag{8}$$

$$J = S_t Pr^{2/3} \tag{9}$$

Where G is the mass flow, given by Eq. (10). In turn, $\dot{m}$ is the mass flow of the fluid.

$$G = \frac{4\dot{m}}{\pi D_i^2} \tag{10}$$

Kim, Youg and Webb in 1999, as reported by Lauar (2011), developed studies and experiments to determine the Colburn factor in an evaporator with geometry similar to that which will be projected in this work. The calculations are presented in Eq. (11) to Eq. (17).

$$H_{ar} = \frac{JG_cC_{p,air}}{Pr_{air}^{\frac{3}{2}}} \tag{11}$$

$$Pr_{Lar} = \frac{\mu_{air}C_{p,air}}{k_{air}} \tag{12}$$

$$J_{kim} = J_3 1.043\left[Re_{Dc}^{-0.14}\left[\frac{S_v}{S_h}\right]^{-0.564}\left[\frac{p_{fins}}{D_h}\right]^{-0.123}\left[\frac{S_v}{D_h}\right]^{1.17}\right]^{(3-N_{t,lin})} \tag{13}$$

$$J_3 = 0.163Re_{Dc}^{-0.369}\left[\frac{S_v}{S_h}\right]^{0.106}\left[\frac{p_{fins}}{D_h}\right]^{0.0138}\left[\frac{S_v}{D_h}\right]^{0.13} \tag{14}$$

$$Re_{Dc} = \frac{G_cD_c}{\mu_{air}} \tag{15}$$

$$G_c = \frac{\dot{m}_{air}}{A_{disp,air}} \tag{16}$$

$$D_c = D_{ext} + 2e_{fins} \tag{17}$$

In wich J is the Colbum factor, G_c is the mass flow in the collar (section available for air passage), $C_{p,air}$ is the specific heat of the air, Pr_{air} is the Prandtl number of the air, S_h is the distance between consecutive tubes horizontally, S_v is the distance between consecutive tubes vertically and e is the thickness. With regard to the subscripts, "disp" means available, "ext" means external, "c" refers to the channel, "h" means horizontal, "D" refers to the diameter and "lin" means line.

2.3.3 CO_2 biphasic convective coefficient in the evaporator

Especially in condensers and evaporators, the fluid flows in two-phase, liquid and vapor, in which makes the calculations to need empirical correlations, often for specific conditions, rather than more general equations. In this case, the mass flow and Reynolds and Prandtl numbers are some of the factors that determine the viability of certain equations.

Patiño et al. (2014) in his paper analyzes several empirical correlations for the calculation of the convective coefficient of refrigerant in evaporators. The methodology is adopted in this work, which consists of the analysis of the flow through finite volumes. Experimental work reported by Patiño et al. (2014) measured the heat transfer coefficients and concluded that the Jung and Kandlikar correlations have good predictions for CO_2. Therefore, both are used in this project to estimate the biphasic convective coefficient of the refrigerant in the evaporator.

The Kandlikar correlation is specific for cryogenic fluids, in other words, they reach very low evaporation temperatures. Ghiaasiaan (2008) provides this calculation procedure based on Eq. (18) to Eq. (34).

$$H_f = \max(H_{NBD}, H_{CBD}) \quad (18)$$

$$H_{NBD} = \{0.6683Co^{-0.2}(1-x)^{0.8}f_2(Fr_{f0}) + 1058Bo^{0.7}(1-x)^{0.8}F_{fl}\}H_{f0} \quad (19)$$

Where H_f is the heat transfer coefficient of the refrigerant by convection, F_{fl} is the fluid surface parameter in Kandlikar correlations, Co is the convection number and Bo is the boiling number.

For $0.5 \le Pr_L \le 2000$ and $2300 \le Re_{L0} < 10^4$, use Eq. (20).

$$Nu_{L0}^{*} = \frac{(Re_{L0}-1000)\left(\frac{f}{2}\right)Pr_L}{\left[1+12.7\left(Pr_L^{\frac{2}{3}}-1\right)\left(\frac{f}{2}\right)^{0.5}\right]} \quad (20)$$

For $0.5 \le Pr_L \le 2000$ and $10^4 \le Re_{L0} < 5x10^6$, use Eq. (21).

$$Nu_{L0}^{*} = \frac{(Re_{L0})\left(\frac{f}{2}\right)Pr_L}{\left[1.07+12.7\left(Pr_L^{\frac{2}{3}}-1\right)\left(\frac{f}{2}\right)^{0.5}\right]} \quad (21)$$

$$Nu_{L0} = Nu_{L0}^{*}\left(\frac{\mu_L}{\mu_w}\right) \quad (22)$$

$$Nu_{L0} = \frac{H_{L0}D}{k_L} \quad (23)$$

$$f = [1.58\ln(Re_{L0}) - 3.28]^{-2} \quad (24)$$

For R744, $F_{fl} = 1.75$

$$Re_{L0} = \frac{GD}{\mu_L} \quad (25)$$

$$Pr_L = \frac{\mu_L Cp_L}{k_L} \quad (26)$$

$$Co = \left(\frac{\rho_g}{\rho_f}\right)^{0.5}\left[\frac{1-x}{x}\right]^{0.8} \quad (27)$$

$$Bo = \frac{q''_w}{Gh_{fg}} \quad (28)$$

$$q''_w = \left[1058\left(Gh_{fg}\right)^{-0.7}F_{fl}H_{L0}(T_w - T_{sat})\right]^{3.33} \quad (29)$$

$$Fr_{f0} = \frac{G^2}{\rho_f^2 gD} \quad (30)$$

For $Fr_{f0} \ge 0.4$, use Eq. (31).

$$f_2(Fr_{f0}) = 1 \quad (31)$$

For $Fr_{f0} < 0.4$, use Eq. (32).

$$f_2(Fr_{f0}) = (25Fr_{f0})^{0.3} \quad (32)$$

$$H_{L0} = H_{f0} \quad (33)$$

$$H_{avg,Kand} = \left(\frac{1}{L}\right)\int_0^L H_{Kand}(z)dz \tag{34}$$

In what q''_w is the heat flux through the surface, f is the ventilation friction factor, g is the local gravity acceleration, $H_{avg,Kand}$ is the average Kandlikar convective coefficient, Fr^* is the dimensionless Froude number, ρ is the specific mass, and h is specific enthalpy. In regard to subscripts, "f" is fluid, "NBD" refers to nucleated boiling, "max" é maximum, "CBD" refers to boiling convection, "f0" is a liquid-vapor mixture assumed to be a saturated liquid, "L" or "l" is liquid, "L0" is the mixture assumed to be liquid, "w" is parade, "g" é gas, "fg" is liquid-vapor, "sat" is saturated and "avg" is average.

Jung's correlation was made based on Chen's correlation and can be found in Patiño *et al.* (2014). Eq. (35) a Eq.(41) were proposed to obtain the CO_2 convective coefficient.

$$H_{TP} = NH_{SA} + F_P H_L \tag{35}$$

$$H_{SA} = 207\left(\frac{k_L}{b_d}\right)\left(\frac{q"b_d}{k_L T_{sat}}\right)^{0.745}\left(\frac{\rho_v}{\rho_l}\right)^{0.581} Pr_l^{0.533} \tag{36}$$

$$b_d = 0.0146\beta\left[\frac{2\sigma}{g(\rho_l-\rho_v)}\right]^{0.5} \tag{37}$$

$$H_L = 0.023Re_l^{0.8}Pr_l^{\,0.4}\left(\frac{k_L}{Di_f}\right) \tag{38}$$

$$F_p = 2.37\left(0.29+\frac{1}{X_{tt}}\right)^{0.85} \tag{39}$$

$$N = 4048X_{tt}^{1.22}Bo^{1.13} \text{ for } X_{tt} < 1 \tag{40}$$

$$N = 2 - 0.1X_{tt}^{-0.28}Bo^{-0.33} \text{ for } 1 < X_{tt} < 5 \tag{41}$$

In which H_{SA} is the convective coeficiente of Stephan e Abdelsalam`s nucleated boiling, H_{tp} is the biphasic convective coefficient, H_L is the convective coefficient of the liquid, b_d is the bond number, N is a function of the Martinelli Parameter (X_{tt}) and Boiling number (Bo), F_p is the modified Chen factor, β is the contact angle (35° = 0.61 rad) and H_{TP} is the biphasic convective coefficient. In relation to subscripts, "v" is steam.

2.3.4 Basic dimensioning equations

The calculation methodology in the evaporator is based on Nellis and Klein (2009). There are many possible geometric configurations for heat exchangers, however for refrigerant-air heat exchange the cross-flow finned type is widely used. The calculation procedures are based on Eq. (42) to Eq. (62).

$$R_{tot} = \frac{1}{(UA)} = R_{int} + R_{cond} + R_{a,ext} + R_{ext} \tag{42}$$

$$R_{int} = \frac{1}{H_f \pi D_{int,f} L_{tube}} \tag{43}$$

$$L_{tubo} = N_{t,col}N_{t,lin}W \tag{44}$$

Where R_{int} is the convective resistance between the coolant and the inner surface of the tube, R_{cond} is the conductive resistance across the wall of the tube, $R_{a,ext}$ is the resistance of the fouling of condensed water on the outer wall of the tube, R_{ext} is the resistance between the air and the surfaces of the fins and the tube, H_f is the convective coefficient of the refrigerant fluid, $D_{int,f}$ is the inner diameter of the tube, L_{tube} is the total length of the tubes, $N_{t,col}$ is the total of tubes per column, $N_{t,lin}$ is the total of tubes per line and W is the width of the evaporator.

$$R_{a,ext} = \frac{R''_{a,ext}}{A_{total,ext}} \tag{45}$$

In which $A_{total,ext}$ is the total area of heat transfer, $R''_{a,ext}$ is the water fouling factor, obtained in the EES`s library, using: $R''_{a,ext} = foulingfactor('Closed - loop\ treated\ water')$.

$$R_{cond} = \frac{\ln\left(\frac{D_{ext,f}}{D_{int,f}}\right)}{2k_m \pi L_{tube}} \tag{46}$$

$$R_{ext} = \frac{1}{\eta_0 H_{air} A_{total,ext}} \tag{47}$$

Where $D_{ext,f}$ is the outer diameter of the tube, k_m is the thermal conductivity of the material, η_0 is the overall surface efficiency and H_{air} is the convective coefficient of air.

$$\eta_0 = 1 - \left(\frac{A_{totat,fins}}{A_{total,ext}}\right)(1 - \eta_{fins}) \tag{48}$$

In which $A_{totat,fins}$ is the total area of the fins, η_{fins} is the efficiency of the fins, obtained in the EES`s library, using: $\eta_{fins} = eta\_fin\_annular\_rect(e_{fins}, De_f/2, r_{fins,ef}, H_{air}, k\_copper)$.

$$\eta_{fins} = \frac{2r[K_1(mr_o r)I_1(mr_o) - I_1(mr_o r)K_1(mr_o)]}{K_0(mr_o r)I_1(mr_o) + I_0(mr_o r)K_1(mr_o)}\left\{\frac{1}{[mr_o(1-r^2)]}\right\} \tag{49}$$

$$m = \sqrt{\frac{2H_{air}}{k_m e_{fins}}} \tag{50}$$

$$r = \frac{r_{ext}}{r_o} \tag{51}$$

Where $r_o = r_{fins,ef}$ is the effective radius of the fins, r_{ext} is the external radius of the refrigerant tube, k_m is the thermal conductivity of the fins material, I_0, I_1 are modified Bessel functions of the first species, K_0, K_1 are modified Bessel functions of second species.

$$A_{totat,fins} = 2\left(\frac{W}{p_{fins}}\right)\left(HL - N_{t,col}N_{t,lin}\left(\frac{\pi D_{ext}^2}{4}\right)\right) \tag{52}$$

Where p_{fins} is the distance between the fins, H is the height of the exchanger and L is the length of the exchanger.

$$A_{tube,free} = \pi D_{ext} L_{tube}\left(1 - \frac{e_{fins}}{p_{fins}}\right) \tag{53}$$

$$A_{total,ext} = A_{total,fins} + A_{tube,free} \tag{54}$$

Where $A_{tube,free}$ is the total area of the tube in direct contact with air and e_{fins} is the thickness of the fins.

$$u_m = \frac{\dot{V}_v}{A_{disp,air}} \tag{55}$$

In which u_m is the mean air velocity in the passage section, $\dot{V}_v$ is the volumetric flow rate of air produced by the fan and $A_{disp,ar}$ is the area of the section available for air passage.

$$A_{disp,air} = (H - N_{t,lin}D_{ext,f})W\left(1 - \frac{e_{fins}}{p_{fins}}\right) \tag{56}$$

$$W_{c,ef} = \frac{W - N_{fins}e_{fins}}{N_{fins}} \tag{57}$$

Where $W_{c,ef}$ is the effective width for the rectangular channel model and N_{fins} is the number of fins.

$$N_{fins} = \frac{W}{p_{fins}} - 1 \tag{58}$$

$$H_{c,ef} = H \tag{59}$$

$$D_h = \frac{W_{c,ef} H_{c,ef}}{2(W_{c,ef} + H_{c,ef})} \tag{60}$$

In which $H_{c,ef}$ is the effective height of rectangular channels and D_h is the effective hydraulic diameter in the rectangular channel.

$$\dot{m}_{c,air} = \frac{\dot{V}_v \rho_{air}}{N_{fins}} \tag{61}$$

$$A_{total,fins} = 2\left(\frac{L_{tube}}{p_{fins}}\right) \pi \left[r_{fins,ef}^2 - \left(\frac{D_{ext,f}}{2}\right)^2\right] \tag{62}$$

Where $\dot{m}_{c,air}$ is the mass flow of air in the channel and ρ_{air} is the density of the air.

With the aforementioned equations it is possible to calculate the total resistance R_{tot} and consequently the conductance (UA), in other words, this is the first step for the design of this type of exchanger. The next step is the calculation of the heat rate $\dot{Q}_{evap}$, with the equations in sequence. As the evaporator will be a cross-flow heat exchanger, it is preferable to use an alternative procedure, known by the NUT-effectiveness method, or ε-NUT method, presented in sequence (Incropera, 2008) with Eq. (63) to Eq. (77).

$$F_{evap} = \frac{L_{evap}}{L_{tube}} \tag{63}$$

$$R_{ext,evap} = \frac{R_{ext}}{F_{evap}} \tag{64}$$

$$R_{a,ext,evap} = \frac{R_{a,ext}}{F_{evap}} \tag{65}$$

$$R_{int,evap} = \frac{R_{int}}{F_{evap}} \tag{66}$$

$$R_{cond,evap} = \frac{R_{cond}}{F_{evap}} \tag{67}$$

In which L_{evap} is the length of the stretch at which evaporation occurs, F_{evap} is the ratio of the evaporation length in relation to the total, $R_{ext,evap}$ is the air-surface external resistance in the evaporation stretch, $R_{a,ext,evap}$ is the resistance of the encrustation of water in the evaporation section, $R_{int,evap}$ is the internal resistance in the evaporation section and $R_{cond,evap}$ is the conductive resistance in the evaporation section.

$$\dot{C}_{f,evap} = \dot{m}_f c_{p,f,evap} \tag{68}$$

$$\dot{C}_{air,evap} = \dot{m}_{ar} c_{p,air,evap} F_{evap} \tag{69}$$

$$\dot{m}_{air} = u_m H W \rho_{air} \tag{70}$$

$$\dot{C}_{min,evap} = MIN(\dot{C}_{f,evap}, \dot{C}_{air,evap}) \tag{71}$$

$$\dot{C}_{max,evap} = MAX(\dot{C}_{f,evap}, \dot{C}_{air,evap}) \tag{72}$$

$$\dot{Q}_{max,evap} = \dot{C}_{min,evap}(T_{air,ent} - T_{f,ent}) \tag{73}$$

Where $\dot{C}_{f,evap}$ is the thermal capacity of the refrigerant fluid, $\dot{m}_f$ is the mass flow of refrigerant, $c_{p,f,evap}$ is the specific heat of the refrigerant in the evaporation, $\dot{C}_{air,evap}$ is the thermal capacity of the air, $\dot{m}_{air}$ is the mass air flow and $c_{p,air,evap}$ is the specific heat of the air in the evaporation stretch. In addition, "MIN" means minimum and "MAX" means maximum.

$$NUT_{evap} = \frac{(UA)_{evap}}{\dot{C}_{min,evap}} \tag{74}$$

$$C_r = \frac{\dot{C}_{min,evap}}{\dot{C}_{max,evap}} \tag{75}$$

$$\varepsilon_{fc,evap} = 1 - \exp\left[\left(\frac{1}{C_r}\right)\left(NUT_{evap}\right)^{0.22}\left\{exp\left[-C_r\left(NUT_{evap}\right)^{0.78}\right] - 1\right\}\right] \tag{76}$$

$$\dot{Q}_{evap} = \varepsilon_{fc,evap}\dot{Q}_{max,evap} \tag{77}$$

In which U is the global heat transfer coefficient, A is the thermal exchange area, UA is the thermal conductance, C_r is the ratio of the heat capacity rates, $\varepsilon_{fc,evap}$ is NUT cross flow effectiveness in the evaporation stretch and $\dot{Q}_{evap}$ is the actual heat transfer rate in the evaporation stretch.

The NUT effectiveness method was also used to calculate the overheating stretch. In addition to the modification of the refrigerant flow, which is no longer biphasic, an important change in the definition of the proportion of the length was performed. The entire step-by-step follows the same as above, where F_{super} is used instead of F_{evap}, given by Eq.(78).

$$F_{super} = \frac{L_{super}}{L_{total}} \tag{78}$$

Where L_{super} is the length of the section where overheating occurs and F_{super} is the ratio of the overheating length to the total.

3. RESULTS AND DISCUSSION

The mathematical modeling of the project was programmed in the MATLAB® software (MATrix LABoratory) and the creation of an input data sheet for the calculations was developed in the MICROSOFT OFFICE EXCEL® software. Properties for R744 were taken from the CoolProp package for MATLAB® containing the thermodynamic properties of various fluids. A graphical interface for the results of the evaporator design has been developed in MATLAB®, providing a pleasant interface with easy manipulation. The library of functions and properties of the EES® (Engineering Equation Solver) was also used to perform some calculations.

The evaporator calculations were made in MATLAB® and a graphical interface was created for this component. There it is shown the convective coefficient obtained for the air, the Kandlikar and Jung average convective coefficients after 10000 title discretization points, the cooling capacity required by the cycle, the refrigerant capacity of the defined evaporator, the air outlet temperature and the total the geometry chosen for the characterization of the exchanger. The heat rate required in the evaporator was calculated from an enthalpy analysis in order to find the coolant input title in the evaporator, in other words, the enthalpy of the refrigerant in the evaporation temperature line equal to the enthalpy of the subcooler provided a title of 0.239.

The calculations are made with the objective of finding the cooling capacity of the evaporator and the temperature of the air after the thermal exchange. All properties are obtained considering the amount of water vapor contained in the air, and the relative humidity of the environment to be cooled is an input variable. To begin with, an EXCEL® table was created for easy control of input variables (Table 1).

Table 1 – General data for the evaporator inlet.

GENERAL INLET DATA	VALUE	UNIT
Refrigerant fluid (primary)	R744	
Secundary fluid	Air	
Refrigerant inlet title on the evaporator	0.239	
T_evap: Evaporation Temperature	-20	[ºC]
T_ae: Air Inlet Temperature	30	[ºC]
T_SA: Superheating Degrees	10	[ºC]
Vm_f: Massic flow of the refrigerante fluid	58	[kg/h]
Relative air humidity considered	0,7	
Vel_air: Speed of air exiting the fan	2	[m/s]
Di_f: Inner Diameter of the Refrigerant Fluid Tube	6.35	[mm]
De_f: Outside diameter of the refrigerant tube	7.93	[mm]
n: Number of convective coeficientes to calculate the average value	10000	
T_fs_SR: Temperature that the refrigerante exits the subcooler	7	[ºC]
L_evap: Length of the evaporation stretch	12	[m]
L_super: Length of the overheat stretch	2	[m]
T_Cond: Condensing temperature	10	[ºC]

In all equations that depended on the title, the average title of each control volume was applied. Figure 3 shows the window with the output variables and graphs of Kandlikar and Jung convective coefficients. The red message alerts if the heat rate taken by the proposed heat exchanger is greater than the required heat rate.

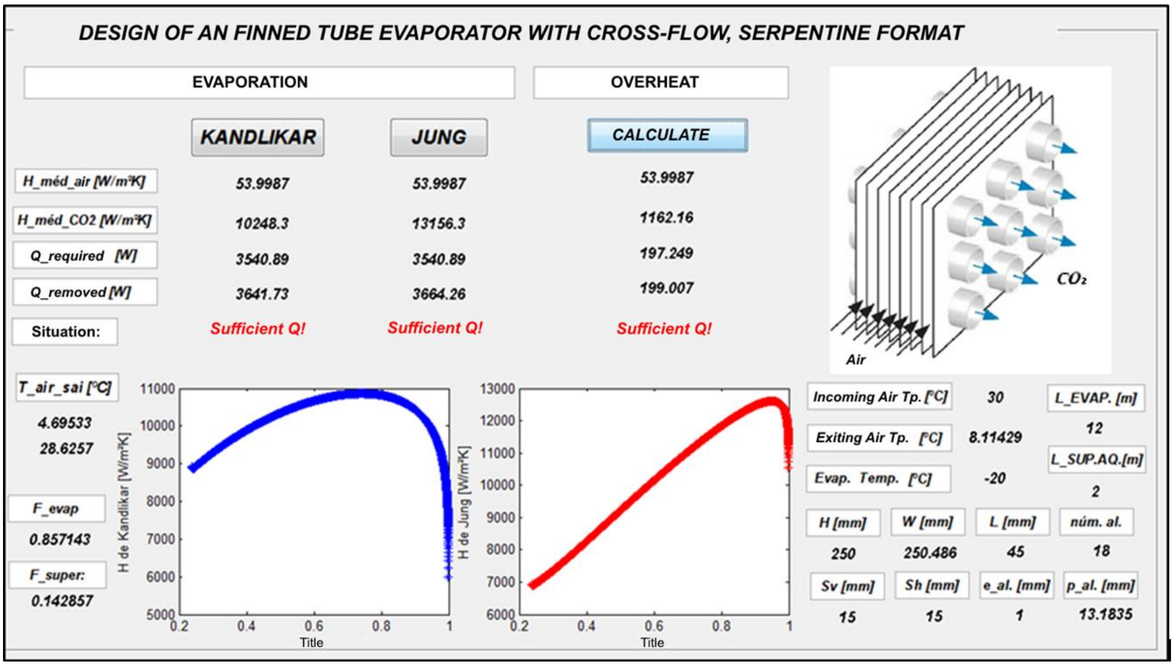

Figure 3. Results of the evaporator design.

In the calculations of the evaporator the control of the parameters of entrance was given by means of an Excel spreadsheet due to the facility in making changes and comparisons. Numerous advantages are observed in this type of treatment. Analyze the influence of each of the parameters, both external and the geometry of the equipment itself, on the refrigeration capacity can result in important studies on such type of exchangers. As an example, it can be observed that the increase of air velocity provides greater thermal changes, however the air cools less; the increase in the number of fins provides greater thermal changes and their impact gradually reduces; increasing the distances between the tubes results in an increase in the convective air coefficient; among other findings. The geometry chosen for the evaporator is shown in Table 2.

Table 2. Output data from the MATLAB® program for the evaporator design.

HEAT EXCHANGE DATA	VALUE	UNITY
S_v: Vertical distance between the tubes	15	[mm]
S_h: Horizontal distance between the tubes	15	[mm]
H: Exchange height	250	[mm]
L: Exchange depth	45	[mm]
n_tub_V: Number of tubes in VERTICAL way	17	
n_tub_H: Number of tubes in HORIZONTAL way	3	
n_tub_TOTAL: Total number of tubes	51	
n_curves_V: Number of diameter curves S_v	50	
n_curves_H: Number of diameter curves S_h	2	
W: Exchanger Width	250.49	[mm]
n_fins: Number of fins	18	
p_fins: Distance between the fins	13.183	[mm]
e_fins: Fin thickness	1	[mm]
P_air: Air pressure (atmospheric)	101300	[Pa]
g: On-site gravity	9.78	[m/s²]
FE_a: Factor of encrustation of the water that condenses on the tube	0.0002	[m²K/W]
k_tubes: Conductivity of the piping material (copper)	385	[W/m.K]
k_fins: Conductivity of the fin material (copper)	385	[W/m.K]

The cooling capacity of the evaporator was 3.74 kW, operating at a pressure of 19.7 bar and the evaporation temperature of - 20 °C. Defined the geometry, as results can be highlighted the air temperature at the evaporator outlet, 8.1 ºC, and the total length of the refrigerant tube, 14 meters. he chosen exchanger is quite compact, with a height and width of approximately 0.25 meters, depth of 0.045 meters and also has only 18 fins to facilitate the passage of air.

4. CONCLUSIONS

The program made for the evaporator calculates the refrigerating capacity according to the conditions of geometry, air intake, evaporation and overheating length, refrigerant flow and pipe diameter. The total length of the refrigerant tube was 14 meters, with a height and width of approximately 0.25 meters, depth of 0.045 meters and also has only 18 fins.

The COP of the defined cycle takes into account the input and output enthalpies of the compressor as well as the input of the evaporator. The theoretical COP of the cycle was 3.47. The air outlet temperature of the evaporator was 8.1 °C, and it can be piped and distributed to a room air conditioning system and made available to the research laboratory of domestic refrigeration machines.

It is intended after the assembly of the machine to contribute with the current research in refrigeration, regarding the need to eliminate the gases that have chlorine in its composition, through experiments. From the outset it was highlighted the fact that CO_2 presents itself as an important alternative because it is a natural fluid and not aggressive to the ozone layer, besides having thermal properties that makes its application feasible in refrigeration.

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6. RESPONSIBILITY NOTICE

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