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DESIGN OF A COUNTER-CURRENT COAXIAL CONDENSER FOR A REFRIGERATING MACHINE OPERATING WITH CO_2 IN A SUBCRITICAL CYCLE

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Abstract. *The emissions of some gases used in refrigerating systems have caused a worrying environmental impact, because they cause the disintegration of the ozone layer. The CO_2 - Carbon Dioxide (R744) has been researched in recent years because it can be used as a refrigerant that is not so aggressive to the environment and presents good thermodynamic properties. Using an idealized air-water refrigerating machine working in a sub-critical cycle, a counter current coaxial condenser was projected and analyzed. The mathematical modeling of the project was programmed in MATLAB® software (MATrix LABoratory). A complete approach of the condenser design project was performed by analyzing the refrigerant heat transfer convective coefficients (single phase and biphasic) and the secondary fluid coefficients (single phase). These coefficients were calculated using the correlations present in the literature. The pressure drop was also calculated. The heat exchange rate between the fluids in the condenser was 4.82 kW operating at a pressure of 45.02 bar and a condensation temperature of 10 °C. A value of 26.5 meters was obtained for the condenser total length. The idealized refrigerating machine has a COP (Coefficient of Performance) of 3.47 being designed for a research laboratory and that allow the production of cold air at 8.1 °C which can be used to aid a room air conditioning system or be applied in laboratory domestic refrigerating machine tests.*

Keywords: *Refrigerating machine, CO_2, R744, counter-current coaxial condenser*

1. INTRODUCTION

The emissions of gases used in refrigeration systems are causing a worrying environmental impact once it induces the disintegration of the ozone layer that protects the planet from ultraviolet solar radiation. This situation features the recurrent demand of using less aggressive gases that have the minimum efficiency necessary to guarantee their insertion in the market (Faria, 2013).

The CO_2 - Carbon Dioxide (R744) is a natural fluid that exists in abundance in nature. It has been the object of research in the last years because it is a refrigerant fluid, which is not so aggressive to the environment and presents a high enthalpy and specific mass variation when compared to other currently used refrigerant fluids (Oliveira, 2013).

There is an abundance of conventional refrigerants, such as CFCs (chlorofluorocarbons) and HCFCs (hydro fluorocarbons) based equipment in the market. However, it is challenging to develop refrigerating machines that work with CO_2 and that can contribute to current research defending very promising and innovative ideas in the refrigeration business (Lorentzen, 1994).

There are little technical information about manufacturers of refrigeration system components and heat pumps operating with CO_2 in Brazil. Although the CO_2 can be an important alternative because it is a natural fluid, non-aggressive to the ozone layer and has some thermal properties, such as a vaporization latent heat, which makes its application viable in refrigeration (Silva, 2016).

The high working pressure of CO_2 demands special criteria for the installation design and safety requirements, requiring a greater specialization of all elements and components involved in the system, including design, implementation, installation, operation and maintenance (Pereira and Primo, 2012).

The purpose of this work is to analyze the project design of one heat exchanger in the refrigerating machine: the condenser. The condenser is a concentric tube condenser, where the primary fluid (CO_2) flows into the central tube and the secondary fluid (water) flows counter-current into the annular space (between the two tubes).

2. METHODOLOGY

In order to analyze CO_2 as a coolant, it was proposed the design of an air-water refrigerating machine using components that operate in a subcritical cycle, allowing the development of research that can scientifically contribute 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 operation cycle.

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 be applied in laboratory tests of domestic refrigerating machines.

In this section the cycle of operation and the components of the idealized refrigerating machine, in addition to the coaxial condenser design mathematical model will be presented.

2.1 Subcritical CO_2 cycle and main components of the system

The refrigeration cycle diagram demonstrates in a schematic way the complete system and shows 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 and can be seen in Figure 1. According to Zhang et al. (2011), common values for condensing temperature and evaporation in CO_2-based refrigeration systems would be, respectively, 20 ° C and -20 ° C. The compression process is considered adiabatic because the heat exchange with the surroundings is very small. The condensation and evaporation pressures are generally about 40 bar and 25 bar, respectively (Silva, 2008).

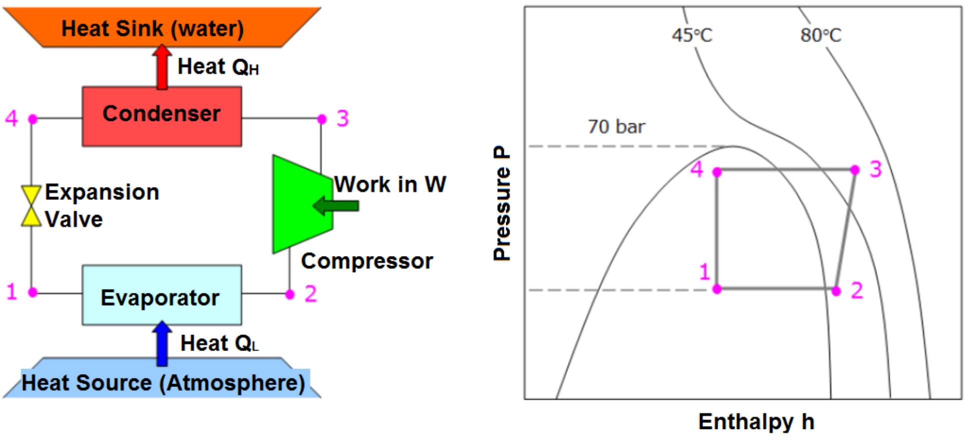

Figure 1. Subcritical CO_2 cycle. Source: Oliveira (2013).

The machine under analysis was designed to use a semi hermetic reciprocating compressor of 1.08 kW power based on pre-set condensation and evaporation temperatures of respectively 10° C and -20° C. The evaporator used is a cross-flow finned tubes evaporator with a cooling capacity of 3.74 kW. The expansion device is an electronic valve.

The diagram of the idealized refrigerating machine and the components required for its operation are shown in Figure 2.

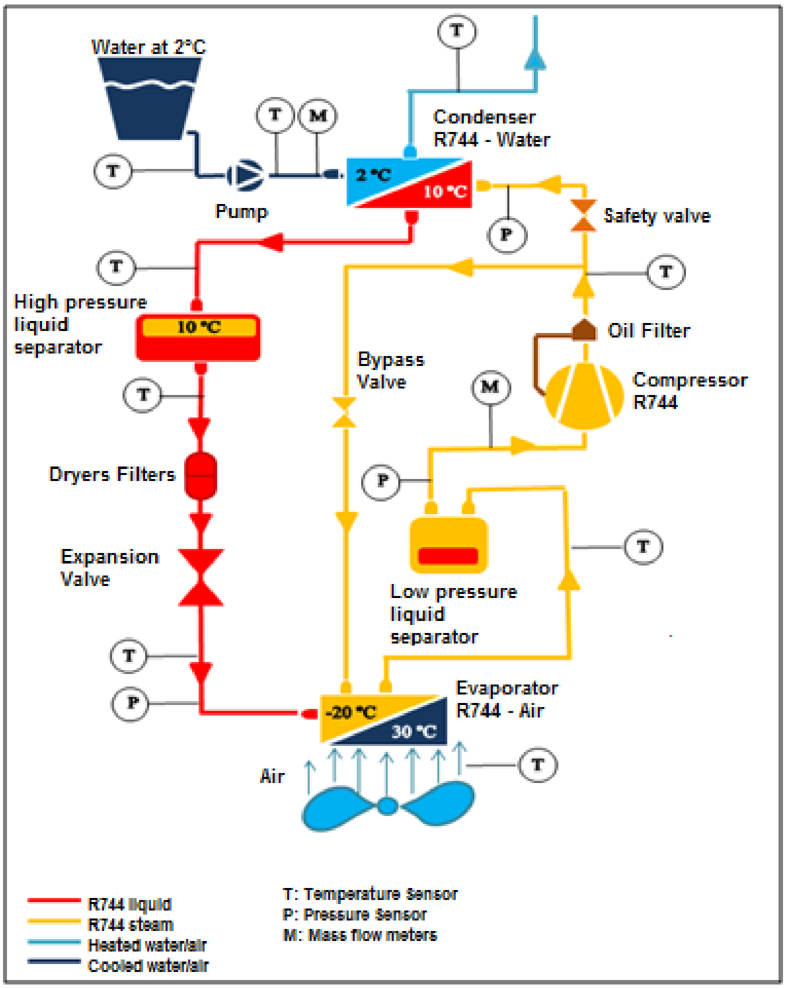

Figure 2: Idealized refrigerating machine and components.

2.2 Condenser design project

The condenser used was designed according to the model proposed by Yamagushi et al. (2011). The water flows upwardly between the tubes and the refrigerant flows downwardly in the countercurrent inner tube. Yamaguchi et al. (2011) developed a simulation model for CO_2 heat pump. During the research, it was necessary to project a coaxial heat exchanger to act as a cooler (in the case of CO_2 the term condenser gives way to cooler, once this fluid does not change phase, remaining as a steam during the heat exchange process). Figure 3 shows the aforementioned coaxial heat exchanger scheme. Its constructive aspects are identical to that designed heat exchanger.

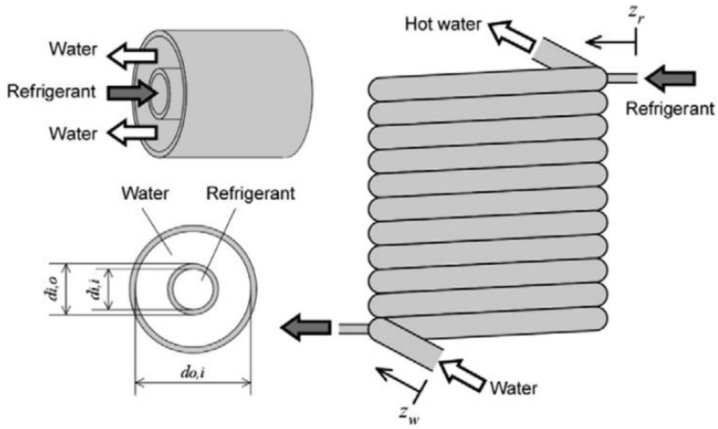

Figure 3. Idealized coaxial condenser. Source: Yamaguchi et al. (2011).

2.3 Modeling procedures

A complete approach of the condenser project was performed stratifying the flow of refrigerant fluid in the condenser in three regions: desuperheating, condensation, and sub-cooling. The determination of the refrigerant heat transfer convective coefficients (single phase and biphasic) and the secondary fluid coefficients (single phase) were performed using the correlations present in the literature. The results for the coefficients were compared for the biphasic region (CO_2) and the most appropriate coefficient was defined for the project. In the single-phase flow region, the correlations of Dittus-Boelter, Sieder and Tate, and Gnielinski were used. In the region of biphasic flow (CO_2), the correlations of Akers, Shah and Chato were used. The CO_2 pressure drop was also determined in all regions. In the single-phase region, the pressure drop was determined by Yoon equation and in the biphasic region by the Muller equation.

2.3.1 Single-phase convective CO_2 coefficient and water coefficient at the condenser

In the coaxial condenser, the secondary fluid is counter current flowing water. The refrigerant flows inside the internal round tube. The below equations, Eq. (1) and Eq. (2) are presented by Incropera (2008) considering it as a laminar flow.

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

$$Re = \frac{\rho u L}{\mu} \tag{2}$$

Nu is the Nusselt number, Re is Reynolds number ($Re \leq 2300$) for laminar, u is the flow velocity and μ is the dynamic viscosity. H_f is the convective coefficient, k_f is the thermal conductivity of the fluid. D_i is the internal tube diameter.

Inside the heat exchanger, for a constant heating rate, the convective coefficient of water H_a in fully developed laminar flow can be obtained with Eq. (3) and Eq. (4). In these equations the internal diameter is not used, instead the hydraulic diameter D_h is used.

$$Nu_D = \frac{H_a D_h}{k_a} = 4.364 \tag{3}$$

$$D_h = D_i - d_e \tag{4}$$

Where k_a is the water thermal conductivity and $d_e$is the inner tube external diameter. The subscript "D' refers to the diameter.

The water flows in the annular region between the concentric tubes, which requires careful calculation of the Reynolds number. The calculation of this dimensionless number is done using Eq. (5) to Eq. (7).

$$Re_D = \frac{G_a D_h}{\mu_a} \tag{5}$$

$$G_a = \frac{4\dot{m}_a}{[\pi(D_i^2 - d_e^2)]} \tag{6}$$

$$Re_D = \frac{4\dot{m}_a}{\pi(D_i + d_e)\mu_a} \tag{7}$$

Where μ_a is the water dynamic viscosity, G_a is the mass flow per area and $\dot{m}_a$ is the water mass flow.

According to Incropera (2008), if the flow is turbulent and fully developed, the following three correlations can be used. Empirical correlations are used because of the complexity behind this type of flow conditions.

The first correlation is the correlation of Dittus-Boelter, given by Eq. (8).

$$Nu_D = 0.23 Re_D^{\frac{4}{5}} \mathrm{Pr}^n \tag{8}$$

Where $0.7 \leq Pr \leq 160$, $Re_D \geq 10000$, $\frac{L}{D} \geq 10$, $n = 0.3$ for heating ($T_s > T_m$), in the condenser, and $n = 0.4$ for cooling ($T_s < T_m$), in the evaporator. L is the length and D the tube diameter, T_s is the outlet fluid temperature and T_m is the average temperature. Pr is the Prandtl number, given by Eq. (9).

$$Pr = \frac{\mu C_p}{k} \tag{9}$$

Where C_p is the specific heat and k the fluid thermal conductivity.
The second correlation is from Sieder and Tate, given by Eq. (10).

$$Nu_D = 0.027 Re_D^{\frac{4}{5}} \left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{10}$$

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

The third correlation is from Gnielinski, given by Eq. (11).

$$Nu_D = \frac{\left(\frac{f}{8}\right)(Re_D - 1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{11}$$

Where $0.5 \leq Pr \leq 2000$, $3000 \leq Re_D \leq 5x10^6$ and $\frac{L}{D} \geq 10$. The friction factor f is calculated using Eq. (12) that was developed by Petkhov for smooth tubes and can be found in Machado (1996). The subscript "s" refers to output. g is the gravity acceleration.

$$f = (0{,}790 \ln(Re_D) - 1.64)^{-2} \tag{12}$$

Special care must be taken when applying the equations in for a Reynolds number close to the transition region $(2300 \leq Re_D \leq 10^4)$. When $Re_D > 10^4$ the flow is fully turbulent, and we can apply the Dittus-Boelter equation. In situations where the flow is in the transition region, the Gnielinski correlation can be used to ensure greater precision and to avoid that the convective coefficient of water is overestimated.

2.3.2 Biphasic CO_2 convective coefficient in the condenser

Mainly in condensers and evaporators, the fluid flows as a two-phase mixture, liquid and steam, which causes the need of empirical correlations for the calculation of variables. These empirical correlations are often for specific conditions, rather than general situations. In this case, the mass flow and Reynolds and Prandtl numbers are some of the factors that define the viability of certain equations.

According to Ghiaasiaan (2008), three studies presented correlations to calculate the convective coefficient (H) in the condensation section of horizontal tubes, as a function of the quality. These studies are: Akers, dated 1959, Shah, dated 1979 and Chato, dated 1998. The equations for each correlation are presented below.

Akers correlation is recommended for horizontal tubes. According to Collier and Thome (1994) it is still valid for several refrigerants at any quality value. Eq. (13) to Eq. (15) show these correlations.

$$\frac{H_F D}{k_L} = C Re_{TP}^n Pr_L^{\frac{1}{3}} \tag{13}$$

$$Re_{TP} = G\left[(1-x) + x\left(\frac{\rho_L}{\rho_g}\right)^{\frac{1}{2}}\right]\left(\frac{D}{\mu_L}\right) \tag{14}$$

$$Pr_L = \frac{\mu_L C_p}{k_L} \tag{15}$$

Where $C = 0.0265\ and\ n = 0.8$ for $Re_{TP} > 50000$ or $C = 5.03\ and\ n = \frac{1}{3}$ for $Re_{TP} < 50000$. Besides, $0 \leq x \leq 1\ and\ G < 200\ kg/m^2 s$.

H_F is the heat exchange coefficient, G is the mass flow per area, D is the diameter, x is the quality, k_L is the fluid thermal conductivity, ρ is the density, μ is the dynamic viscosity, C and n are constants, Re_{TP} is the Reynolds biphasic number and Pr_L is the liquid part Prandtl number. C_p is the specific heat. The subscripts "L" and "g" represent the liquid and gaseous state respectively.

Shah's correction is one of the most widely used, particularly for interpreting the forced convection that occurs during condensation. Eq. (16) to Eq. (18) show the correlation to obtain the convective coefficient.

$$\frac{H_F}{H_{L0}} = (1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{\left(\frac{P}{P_{cr}}\right)^{0.38}} \tag{16}$$

$$\frac{H_{L0}D}{k_L} = 0.023\left(\frac{GD}{\mu_L}\right)^{0.8} Pr_L^{0.4} \tag{17}$$

$$Re_{L0} = \frac{G(1-x)D}{\mu_L} \tag{18}$$

Where H_{L0} is the heat transfer coefficient (assumed as liquid), P_{cr} is the critical pressure and P is the condensing pressure. The subscript "L0" refers to the mixture assumed to be liquid.

In Collier and Thome (1994), this correlation was presented without restrictions of application. However, in Ghiaasiaan (2008), the following recommendations were presented: $10.8 < G < 1599\ kg/m^2s$, $Re_L > 350$ for tubes, $Re_L > 3000$ for rings, $0.002 < \left(\frac{P}{P_{cr}}\right) < 0.44$, $Pr_L > 0.5$ and $0 < x < 1$.

Chato's correlation is based on experiments performed with various refrigerant fluids and mixtures, in tubes with internal diameter ranging between 3.14 and 7.04 mm and quality between 0.1 and 0.9. According to Ghiaasiaan (2008), the transition between flow regimes around $G = 500kg/m^2s$ is not smooth and can lead to unreal jumps and discontinuities when mass flow approaches this value.

For $G \geq 500\frac{kg}{m^2s}$ or $G < 500\frac{kg}{m^2s}$ with $Fr^* > 20$, Eq. (19) to Eq. (22) are valid.

$$Nu_F = \frac{H_F D}{k_L} = 0.023(Re_L)^{0.8} Pr_L^{0.4}\left[1 + \frac{2.22}{X_{tt}^{0.89}}\right] \tag{19}$$

$$Ga = \left[\frac{\rho_L \Delta\rho g D^3}{\mu_L^2}\right] \tag{20}$$

$$Fr^* = 0.025 Re_L^{1.59}\left[\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right]^{1.5}\left(\frac{1}{\sqrt{Ga}}\right)\ for\ Re_L \leq 1250 \tag{21}$$

$$Fr^* = 1.26 Re_L^{1.04}\left[\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right]^{1.5}\left(\frac{1}{\sqrt{Ga}}\right)\ for\ Re_L \geq 1250 \tag{22}$$

For $G < 500\frac{kg}{m^2s}$ and $Fr^* < 20$, Eq. (23) to Eq. (28) are valid.

$$Nu_F = \frac{0.23Re_{g0}^{0.12}}{1+1.1X_{tt}^{0.58}}\left[\frac{GaPr_L}{Ja_L}\right]^{0.25} + \left(1 - \frac{\theta_L}{\pi}\right)\left[0.0195Re_L^{0.8}Pr_L^{0.4}\sqrt{1.376 + \frac{c_1}{X_{tt}^{c_2}}}\right] \tag{23}$$

$$X_{tt} = \left(\frac{\rho_g}{\rho_f}\right)^{0.5}\left(\frac{\mu_f}{\mu_g}\right)^{0.1}\left(\frac{1-x}{x}\right)^{0.9} \tag{24}$$

$$Ja_L = \frac{c_{P,L}(T_{sat} - T_W)}{h_{fg}} \tag{25}$$

$$\theta_L \approx \pi - cos^{-1}(2\alpha - 1) \tag{26}$$

$$\alpha = \frac{1}{1+\frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)} \tag{27}$$

$$Fr_{L0} = \frac{G^2}{\rho_L^2 g D} \tag{28}$$

For $Fr_{L0} \leq 0.7$, Eq. (29) and Eq. (30) can be used, and for $Fr_{L0} > 0.7$, $c_2 = 1.655$ and $c_1 = 7.242$.

$$c_1 = 4.172 + 5.48Fr_{L0} - 1.564Fr_{L0}^2 \tag{29}$$

$$c_2 = 1.773 - 0.169Fr_{L0} \tag{30}$$

Where Nu_F is the Nusselt number of the liquid or steam film, X_{tt} is the Martinelli parameter, Ga is the Galileo number, Fr^* is the dimensionless Froude number, Ja_L is the Jakob number of the liquid, T_{sat} is the saturation temperature, T_w is the wall temperature, α is the empty fraction, c_1 and c_2 are constants and $h_{fg}$is the latent heat of vaporization.

2.3.3 Pressure drop

According to Oliveira (2013), the Yoon equation for the pressure drop ΔP, Eq. (31), together with the Blasius equation for the friction factor f, Eq. (32) and Eq. (33), can be used for the calculation of the pressure loss of monophasic CO_2.

$$\Delta P = \frac{fG^2L}{2\rho d_i} \tag{31}$$

$$f = 0.316Re^{-\frac{1}{4}}, if\ Re \leq 20000 \tag{32}$$

$$f = 0.184Re^{-\frac{1}{5}}, if\ Re \geq 20000 \tag{33}$$

According to Faria (2013), the pressure drop in a two-phase flow can be obtained by means of the fluid momentum equations. The calculation is done by equating the momentum variation to the forces applied in the control volume. Owing to this momentum balance, the equation of Muller, Eq. (34) to Eq. (42), found in Machado (1996) can be used. Its first term refers to the fluid loss of velocity and the increase of the specific weight to maintain the flow, once the fluid is condensing. The second term calculates the pressure drop due to friction. The third term describes the pressure gain resulting from the loss of potential energy from the downward flow.

$$\frac{\partial P}{\partial z} = -G^2\frac{d}{dz}\left(\frac{x^2v_v}{\alpha} + \frac{(1-x^2)v_l}{1-\alpha}\right) + \left(\frac{dP}{dz}F\right) + gsen(\theta)(\alpha\rho_v + (1-\alpha)\rho_l) \tag{34}$$

$$\alpha = \frac{1}{1+\frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)} \tag{35}$$

$$\left(\frac{dP}{dz}F\right) = \frac{fv_lG^2(1-x)^2}{2d}\phi_{l0}^2 \tag{36}$$

$$\phi_{l0}^2 = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2} \tag{37}$$

$$f = \frac{64}{Re} \text{ for } Re \leq 2300 \tag{38}$$

$$f = \frac{0.316}{Re^{0.25}} \text{ for } 2300 \leq Re \leq 20000 \tag{39}$$

$$f = [1.82\log(Re) - 1.64]^{-2} \text{ for } Re \geq 10000 \tag{40}$$

$$Re_l = \frac{G(1-x)D}{\mu_l} \tag{41}$$

$$Re_v = \frac{GxD}{\mu_v} \tag{42}$$

Where $g\ (+)$ is the downward flow and $g\ (-)$ is the upward flow. To find the constant C, it is necessary to analyze each phase flow, according to Table 1. The subscript "v" refers to the steam or vapor state. z is ordered in the tube length direction and v is the kinematic viscosity.

Table 1: Criteria for choosing the constant C

LÍQUIDO	VAPOR	C
Turbulent	Turbulent	20
Laminar	Turbulent	12
Turbulent	Laminar	10
Laminar	Laminar	5

2.3.4 Fundamental equations for dimensioning

For condenser design, some basic heat transfer equations should be highlighted. The calculation procedures are based on Eq. (43) to Eq. (53).

$$q_f = \dot{m}_f(h_{fs} - h_{fe}) \tag{43}$$

$$q_a = \dot{m}_a Cp_a(T_{as} - T_{ae}) \tag{44}$$

$$q_f = q_a = q \tag{45}$$

$$T_{as} = \frac{\dot{m}_f(h_{fs} - h_{fe})}{\dot{m}_a Cp_a} + T_{ae} \tag{46}$$

$$q = U\pi d_e L \Delta T_{ml} \tag{47}$$

$$\frac{1}{U} = \frac{d_e}{H_f d_i} + \frac{d_e \ln\left(\frac{d_e}{d_i}\right)}{2k} + \frac{1}{H_a} \tag{48}$$

$$\Delta T_{ml} = \frac{(T_{fe} - T_{as}) - (T_{fs} - T_{ae})}{\ln\left(\frac{T_{fe} - T_{as}}{T_{fs} - T_{ae}}\right)} \tag{49}$$

$$\Delta L = \frac{\Delta q}{U\pi d_e \Delta T_{ml}} \tag{50}$$

$$q = H_a \pi d_e L(T_{w,a} - T_a) \tag{51}$$

$$q = \frac{2\pi L k(T_{w,f} - T_{w,a})}{\ln\left(\frac{De_f}{Di_f}\right)} \tag{52}$$

$$q = H_f \pi d_i L(T_f - T_{w,f}) \tag{53}$$

Where q is the heat exchange rate, $\dot{m}$ is the mass flow, h is the enthalpy, U is the global heat transfer coefficient and H is the convective coefficient. The subscripts "f" and "a" are used for the refrigerant and water, respectively. The subscript "s" is used to represent the output and the subscript "e" to represent the input. The subscript "w" is used to represent the wall of the tube.

3. RESULTS AND DISCUSSION

The mathematical modeling of the project was programmed in the MATLAB® software (MATrix LABoratory) and an input data sheet was created for the calculations using EXCEL®. The properties of the refrigerating fluid R744 were taken from the CoolProp package for MATLAB® which contains thermodynamic properties of various fluids. A graphical interface for the results of the condenser design was developed using MATLAB®, providing a pleasant interface and easy manipulation.

An algorithm was developed in MATLAB® that allows the calculation of the tube length in the subcooling, condensation and desuperheating sections, as well as the refrigerant and water pressure drops at these sections. The following EXCEL® table was created for controlling the input variables (Table 2).

With a didactic purpose, a graphical interface was created using MATLAB® containing buttons for calculations of the tube length according to the desired correlation, pressure drop of both water and refrigerant fluid and the water output temperature in each section. The condenser modelling provided the results shown in Figure 4 and in Table 3.

In subcooling, as well as in desuperheating, the calculations are simple because the flow is not biphasic. The calculations of heat transfer in biphasic regime are more complex. In many cases, it is hard to determine the convective coefficient, since the flow is in constant variation. In subcooling, the tube length was found using the CO_2 inlet and outlet flow rates and temperatures and the water inlet temperature. After finding the water's inlet temperature at the condenser, the volume was divided in 10,000 volumes. Each volume was attached to the fluid title that varies along the condensation. This way, the variations of enthalpy were obtained.

Table 2. MATLAB input data for the condenser's design project.

INPUT DATALIST	VALUE	UNIT
Primary fluid - Refrigerant	R744	nap
Secondary fluid	Water	nap
T_cond: Condensation Temperature	10	[ºC]
T_ae: Secondary fluid Inlet Temperature	2	[ºC]
Vm_f: Primary Refrigerant mass flow	58	[kg/h]
Vm_a: Secondary fluid mass flow	400	[kg/h]
Di_f: Inner diameter of the refrigerant fluid tube (1/4")	6.35	[mm]
De_f: External diameter of the refrigerant fluid tube	7.93	[mm]
Di_a: Internal diameter of the secondary fluid tube (9/16")	14.28	[mm]
De_a: External diameter of the secondary fluid tube	15.28	[mm]
T_sup: Sub-cooling step degrees	3	[ºC]
L_SR_FREE: Estimated piping length from subcooling outlet to expansion valve inlet.	1	[m]
téta_graus: Serpentine inclination angle	5	º
T_cr: Critical temperature of the refrigerant	31.06	[ºC]
g: Local gravity	9.78	[m/s²]
P_cr: Critical pressure of the primary fluid	7384000	[Pa]
n: Number of finite volumes for calculations	10000	
Te_f_D: Inlet temperature of the refrigerant in the desuperheating (shortly after the compressor outlet)	74.3	[ºC]
L_D_FREE: Estimated piping length from the desuperheating exit to the compressor inlet	1	[m]

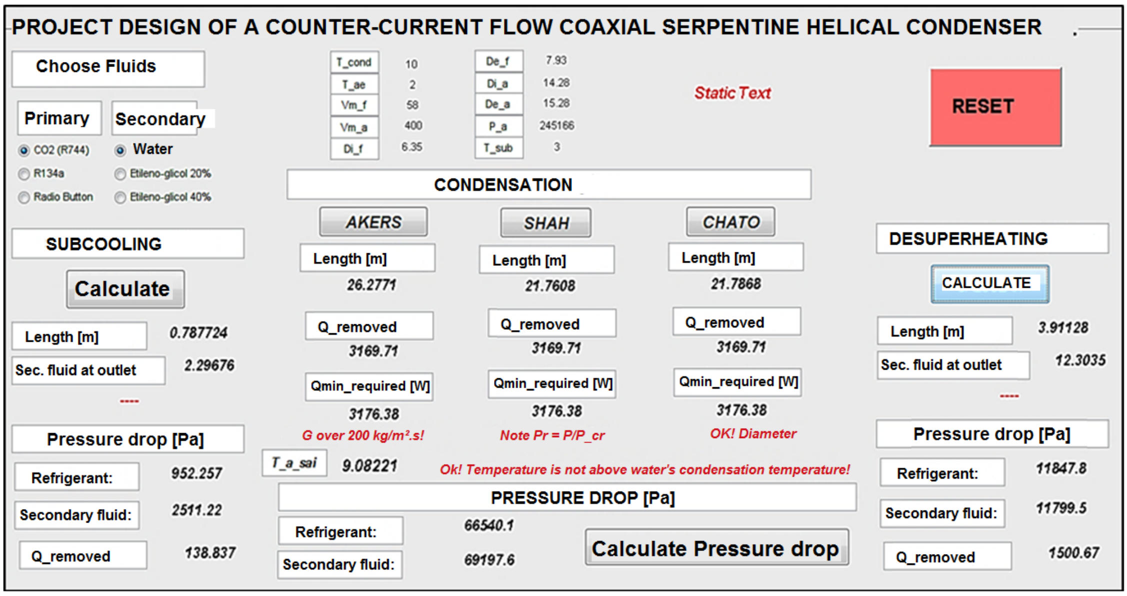

Figure 4. Condenser project design results.

A final result of 26.5 meters was obtained for the coaxial condenser tube length using Chato's correlation. The convective coefficient's and condensation length's calculation results using Shah and Chato correlations were very close. On the other hand, Akers correlation is recommended for a refrigerant mass flow, G, lower than 200 kg/m²K but in the project conditions the fluid mass flow is higher than this value.

The refrigerant pressure drop was calculated in order to analyze how much the real situation differs from the ideal situation. In the condensation section, the total pressure drop was approximately 58 kPa. This pressure variation could not be detected by the manometer sensitivity. The secondary fluid's pressure drop in the subcooling, condensation and desuperheating was calculated in order to size the water pump. It was considered that the water flows 1m before and 1m after the thermal exchange with the refrigerant. For the secondary fluid was found the approximated value of 74 kPa of pressure drop.

Table 3. Condenser project design MATLAB® program output data.

OUTPUT DATALIST	VALUE	UNIT
L_SR: Calculated tube length – Subcooling	0.7877	[m]
T_as: Secondary fluid temperature at Subcooling outlet	2.29676	[ºC]
P_f_SR: Refrigerant pressure drop in the Subcooling	952.26	[Pa]
P_a_SR: Secondary fluid pressure drop in the Subcooling	2511.22	[Pa]
L_C: Calculated tube length - CONDENSATION	21.79	[m]
P_f_C: Refrigerant pressure drop in the Condensation	58073.5	[Pa]
P_a_C: Secondary fluid pressure drop in the Condensation	60312.6	[Pa]
Te_a_D: Secondary fluid temperature after condensation (desuperheating inlet)	9.08	[ºC]
L_D: Pipe length where the desuperheating occurs	3.91	[m]
P_f_D: Refrigerant pressure drop from the desuperheating until compressor inlet	11847.8	[Pa]
P_a_D: Secondary fluid pressure drop in the desuperheating	11799.5	[Pa]
P_a_TOTAL: Secondary fluid total pressure drop Subcooling + Condensation + Desuperheating	74.62332	[kPa]
L_total: Condenser total length	26.49	[m]

The refrigerant's heat rejection rate in the condenser was 4.82 kW, operating at a pressure of 45.02 bar and a condensation temperature of 10 °C.

4. CONCLUSIONS

The program for sizing the condenser was able to calculate the tube length required to achieve the desired heat exchange rate. The water temperature at the heat exchanger outlet and the pressure losses of the fluids were also calculated. The resulting length was 26.5 meters.

The thermodynamic cycle COP takes into account the input and output enthalpies of the compressor as well as the input of the evaporator. It was obtained a theoretical cycle COP of 3.47. The air temperature at the evaporator outlet was 8.1 ° C. The cold air can be piped and distributed to a room air conditioning system or to a research laboratory of domestic refrigeration machines.

The proposal of designing and constructing this machine is to contribute with the current research in refrigeration, regarding the need to eliminate the gases that have chlorine in its composition. The CO_2 presents itself as an important alternative because it is an abundant natural fluid and not aggressive to the ozone layer. Besides this gas has thermal properties that makes possible its application in refrigeration.

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6. RESPONSIBILITY NOTICE

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