HEAT TRANSFER CHARACTERISTICS OF AN EVAPORATOR EQUIPPING AN AIR HANDLING UNITS FOR CEREAL SEED STORAGE FACILITY
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CARACTERISTICILE DE TRANSFER DE CĂLDURA ALE UNUI VAPORIZATOR CARE ECHIPEAZA O CENTRALA DE TRATARE A AERULUI DESTINATA UNUI DEPOZIT DE SEMINȚE DE CEREALE

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ABSTRACT
In this paper the evaporator from the air handling units (AHU) is being studied, this being connected to an air conditioning installation with mechanical compression of vapors. In a previous analysis, the heat exchanger (HEX) was checked to see if it complied with the cooling capacity relative to the heat exchange surface by finding the specific criteria equations. A follow-up to the previous article is to calculate based on the thermodynamic cycle, what happens if the refrigerant is replaced with other new refrigerants in the field of air conditioning in order to improve the performance of the air conditioning system (ACS), finding the refrigerant with a low Global warming potential (GWP) and with an appropriate flammability class. The moisture content of seeds is one of the most important storage parameters affecting their quality. In order to obtain the desired effect, the temperature and humidity of the air must be maintained in the warehouse with an air treatment unit. In this study, it is desired to replace the refrigerant and preserve the refrigerating power from the evaporator, by changing the degree of sub-cooling from the condenser. Due to the fact that the temperature in the ambient environment changes substantially during a calendar year and the condenser of the air conditioning installation has the same configuration, thus the decrease in the ambient temperature will result in the appearance of the sub-cooling process and an increase in the performance of the installation.

REZUMAT
In aceasta lucrare se studiază vaporizatorul de la o unitate de tratare a aerului (UTA), ce este racordată la o instalație frigorifică cu comprimare mecanică de vapori. Într-o analiză anterioară, schimbătorul de căldură (HEX) a fost verificat pentru a vedea dacă a respectat capacitatea de răcire în raport cu suprafața de schimb de căldură prin găsirea ecuațiilor criteriale specifice. O continuare a articolului anterior este de a calcula pe baza ciclului termodinamic, ce se întâmplă daca agentul frigorific este înlocuit cu alți agenți frigorifici noi in domeniul aerului condiționat, in scopul îmbunătățirii performantelor sistemului de aer condiționat (ACS), pentru a identifica agentul frigorific cu un potențial de încălzire globală (GWP) scăzut și cu o clasă de inflamabilitate adecvată. Conținutul de umiditate al semințelor este unul dintre cei mai importanți parametri de depozitare care afectează calitatea acestora. Pentru a obține efectul dorit trebuie menținute in depozit o temperatură şi o umiditate a aerului, utilizând o centrala de tratare aer. In acest studiu se dorește înlocuirea agentului frigorific şi păstrarea puterii frigorifice de la vaporizator, prin modificarea gradului de subrăcire de la condensator. Datorită faptului că temperatura din mediul ambiant se modifică substanțial pe parcursul unui an calendaristic şi condensatorul instalaţiei de climatizare aer are aceeași configurație, prin scăderea temperaturii ambiante va rezulta apariția procesului de subrăcire şi o creștere a performanței instalaţiei.

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INTRODUCTION

Cereals are one of the most important sources of nutrition for most of the world's population. Maintaining the quality of cereal grains is essential between harvest and use. An important preservation method is the control of moisture content. Excess moisture content can encourage the growth of bacteria and fungi and the development of insects and mites in the seeds.

Grain drying involves both heat and mass transfer processes. Mathematical modeling of grain drying can provide a better understanding of the physical and thermal processes during drying *(Ebrahimifakhar et al., 2020).*

Storage conditions of temperature and relative humidity are essential to preserve the quality of the seeds and to ensure storage stability. Seed moisture content, storage temperature, relative humidity, air O_2 and CO_2 concentrations, and storage duration are the most critical parameters for seed postharvest life. The rate of degenerative or seed damage processes increases as the moisture content of the seeds increases. If the moisture content is high enough, the biological activity in the seed will produce enough heat to spoil the seeds if they are not well aerated. Seeds with high moisture are more prone to heat damage than seeds with lower moisture levels. Seeds with moisture content above 18% are subject to excessive damage during storage and processing. Thus, the storage time is reduced, and quality losses occur in seeds with high moisture (>18%). However, seeds with moisture content below 15% are increasingly resistant to impact damage. The respiration rate of seeds with lower moisture content decreases by over 18% *(Kibar et al., 2019).*

The refrigerant is a fluid substance or mixture, usually used in a thermodynamic cycle with a reversible phase transition from a liquid to a gas and back. Refrigerators, air conditioners, heat pumps, water coolers and many other devices use refrigerants as an intermediate fluid to transfer heat between two sources.
A brief history of the past, present, and likely future of natural refrigeration agents is presented in figure 1 *(Abas et al., 2018).*

In a previous article, the evaporator in the AHU was checked in terms of cooling capacity relative to the heat exchange surface by finding in the literature the criterion equation of the indoor convection coefficient that verifies the dimensions of the evaporator in the equipment *(Uta, I. et al., 2022).*

The objective of this study is to analyze, based on the thermodynamic cycle, which parameters or which components change at the air conditioning system, by replacing the refrigerant with other agents in the field of air conditioning, keeping the rest of the data unchanged.

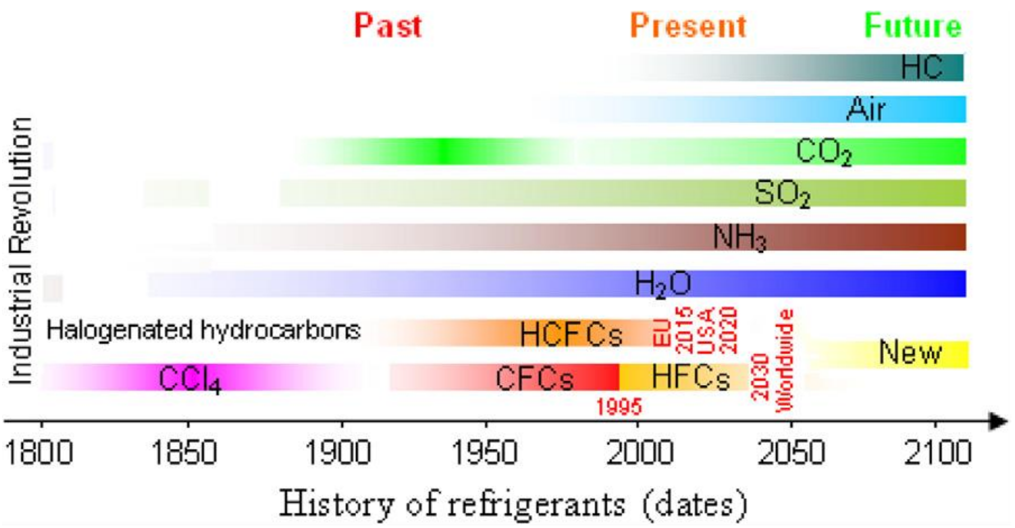


Fig. 1 – Diagram with the history of synthetic and natural refrigerants
(Abas et al., 2018; Uta, I. et al., 2022)

MATERIALS AND METHODS

The technical characteristics of the air conditioning installation

Within the Thermodynamics department of the POLITEHNICA University of Bucharest is an air conditioning installation fully equipped with an air treatment plant and refrigeration unit, Figure 2, where operating parameters can be determined *(Uta I. et al., 2022).*

Fig. 2 - General views of the experimental system

Figure 3 exemplifies the hydraulic diagram of the ACS, which shows the main elements of the installation such as the compressor, the evaporator, the condenser and the rolling valve. Figure 4 shows the thermodynamic cycle of ACS with characteristic state points *(Uta, I. et al., 2022).*

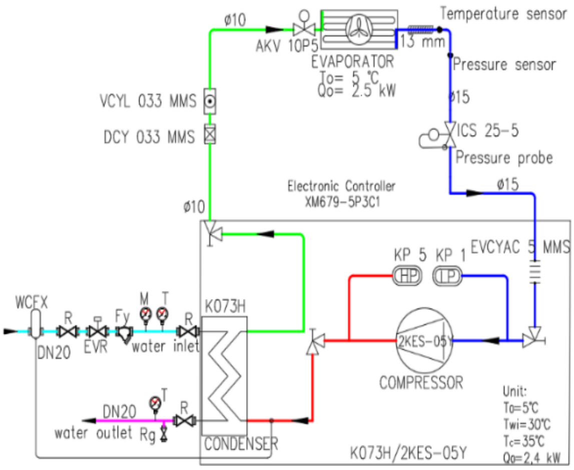


Fig. 3 - Hydraulic diagram of the refrigeration system

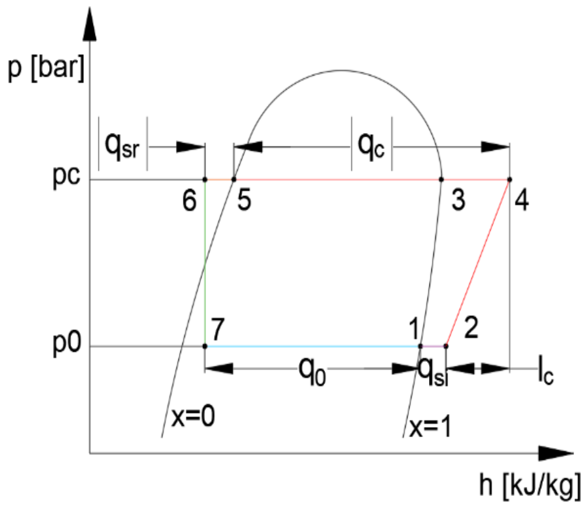


Fig. 4 - Thermodynamic cycle of the VCRS

Heat transfer takes place on the surface of the evaporator by cooling the humid air by the following ways: forced convection on the outside of the HEX, mass exchange by condensing the relative humidity of the air on the surface of pipes and sheets, conduction through the metal wall of pipes and of sheets, convection to boiling.

The evaporator is a surface heat exchanger that achieves the refrigeration effect by forced convection with a fan. The process takes place with phase change, in which the refrigerant takes heat from the cooled environment or from the air in our case, and the cooled air reaches the desired room.

The technical data of the AHU evaporator are known, they have been measured and it consists of: copper pipes, fins or aluminum sheets, with galvanized sheet metal housing, with a single row of pipes *(IDewa B, 2017*; *Uta I. et al., 2022).*

Table 1

Characteristics of the AHU evaporator *(www.komfovent.com)*

Evaporator coil	**Length [mm]: 180**	**Pressure drop [Pa]: 7**	
Coil code	DX-G10-01R-0755-0300-130/-10-1×05C-24F-M1-C40-IS2-RC-1×½/1×22		Int. headers
Air volume [m³/s]	0.28	Capacity [kW]	2.50
Sensible capacity [kW]	1.62	Latent capacity [kW]	0.88
Safety on capacity [%]	56	Air off dry bulb [°C]	25.2
Air on dry bulb [°C]	30.0	Air off wet bulb /RH [°C / %]	19.7 / 61
Air on wet bulb/RH [°C / %]	21.9 / 50	Face velocity [m/s]	1.27
Air pressure drop coil (Wet) [Pa]	7	Fluid flow rate [kg/hr]	58.35
Air pressure drop coil (Dry) [Pa]	5	Subcooling temperature [K]	5.0
Rfrigerant R134A		Superheat temperature [K]	5.0
Condensing temperature [°C]	35.0	Condensation [kg/hr]	1.22
Evaporating temperature [°C]	5.0	Inlet connection	1x½
Refrigerant Pressure Drop [kPa]	0	Outlet connection	1x22
Fin spacing [mm]	2.4	Connection type	Filet
Fin material	Al	Internal volume [dm³]	0.72
Max high pressure [bar]	42	Drain pan type	Flat
Max high temperature [°C]	80		

The evaporator for cooling the air, works with forced convection, the transmission of heat between the hot air and the refrigerant when boiling inside the pipes takes place by forced convection on the outside, mass exchange, conduction through the metal wall and boiling convection. *(Stamatescu, C., 1979)*

The compressor of the air conditioning system is a component that circulates the refrigerant throughout the installation and in which the theoretical compression process takes place, it allows the mechanical increase of the refrigerant pressure in order to suck the vapors from the evaporator and force them to enter the condenser. The compressor is of the semi-hermetic type, with the dimensions shown in figure 5 and with the technical data from table 2, where it was selected with Software Bitzer.

Table 2

Compressor data sheet

Refrigeration compressor Bitzer, type 2KES-05Y-40S							
	Speed compressors	**Bore**	**Stroke**	**No of cylinder**	**Dead space volume**	**Max temperature**	**Max pressure**
Unit of measurement	[rot/min]	[mm]	[mm]	[cylinder]	[%]	[°C]	[bar]
Rotation	nr	D	S	i	ε_0	t	p
Value	1550	30	33	2	5	70	32
Type: Semi-hermetic Reciprocating Compressors							

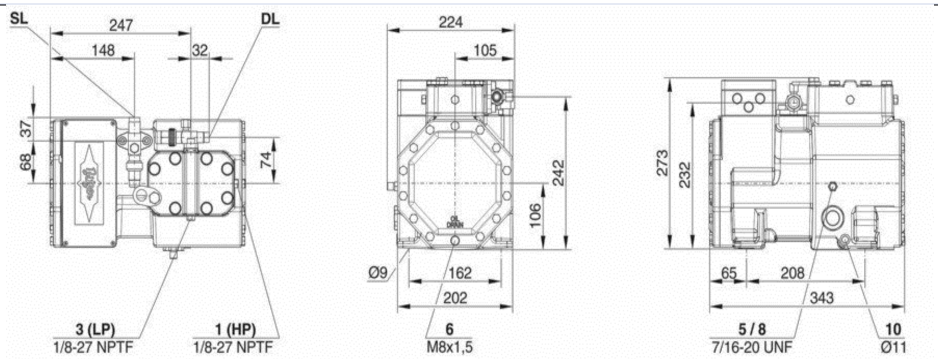


Fig 5 - Compressor dimensions and connections
(www.bitzer.de)

The condenser is a surface heat exchanger that has the role of dissipating the heat accumulated in the refrigerant into the water. The amount of heat is equal to the sum of the heat absorbed in the evaporator and the heat produced by the compression of the agent vapors in the compressor.

The heat transfer medium is water at a temperature lower than that corresponding to the condensing pressure. The condenser process can be compared to the evaporator process except that it is of the opposite "sign", is the conditional change is from vapor to liquid.

The air conditioning system condenser is of multitube type and has the dimensions according to figure 6, and the technical and operational data can be found in table 3.

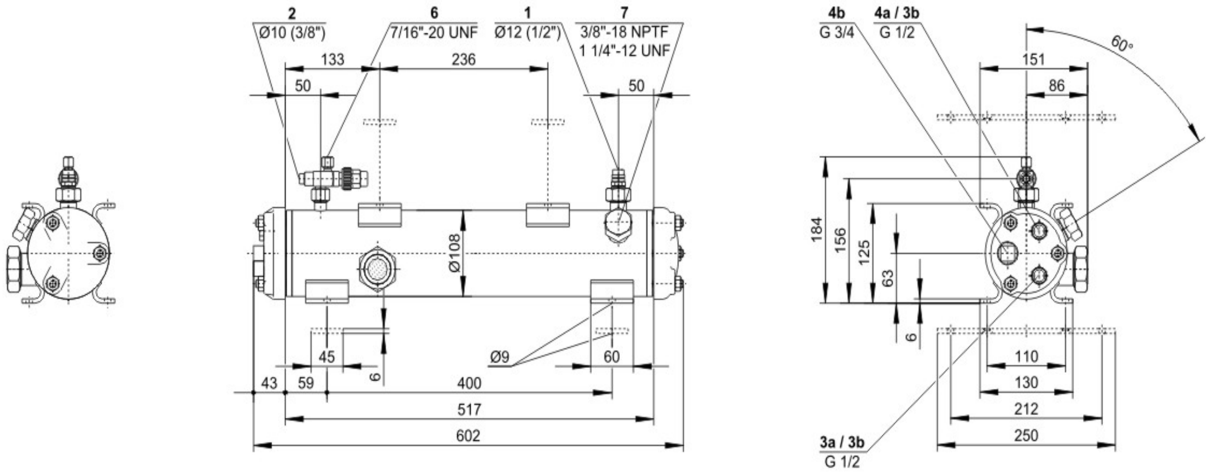


Fig. 6 - Condenser dimensions and connections *(www.bitzer.de)*

Table 3

Condenser data sheet

Condenser Bitzer, type K073H		Water-cooled multitube exchanger	
Weight [kg]	11	Height [mm]	184
With [mm]	602	Jacket tube diameter [mm]	108
Refrigerant inlet [mm]	12 / ½”	Refrigerant outlet [mm / ”]	10 / 3/8”
Coolant inlet (2 passes) [”]	2 x1/2”	Coolant outlet (2 passes) [”]	¾”
Coolant inlet (4 passes) [”]	1/2”	Coolant outlet (4 passes) [”]	1/2”
Receiver volume refrigerant [dm^3]	3,4	Max pressure refrigerant [bar]	33
Max. Operating Temperature [°C]	120	Max pressure coolant side [bar]	10

Mathematical analysis of the evaporator from the air conditioning system

This article is a continuation of the mathematical analysis of the evaporator performed in a previous work, where the chosen criterion equation was the one that verifies the dimensional data of the heat exchanger, such as: pipe diameters, pipe length, number of fins, fin thickness, but at the same time we also calculate technical parameters: cooling capacity, indoor convection coefficient, coefficient global heat exchange *(Uta, I. et al., 2022).*

By changing the order of the equations, the previous article *(Uta, I. et al., 2022)*, one can determine the resulting cooling capacity or what other data will be modified by replacing the refrigerant R134a with other refrigerants in the field of air conditioning, while maintaining the rest of the parameters unchanged.

The notations used in the formulas below can be identified in the Nomenclature (Table 4) at the end of the paper work. With these data, the air flow surfaces through the exchanger can be determined and the mass flow of humid air can we calculated:

$$\Delta H = H_1 - H_2 \left[\frac{kJ}{kg}\right] \tag{1}$$

$$\dot{m}_{aum} = \frac{\dot{Q}_o}{\left(1 + x_1\right) \cdot \Delta H}\left[\frac{kg}{s}\right] \tag{2}$$

Calculation of the humid air volume flow and the mass vapor flow:

$$\dot{V} = \frac{\dot{m}_{aum}}{\rho_m} \cdot 3600 \left[\frac{m^3}{h}\right] \tag{3}$$

$$\dot{m}_v = \dot{m}_{aum} \cdot \left(x_1 - x_2\right) \left[\frac{kg}{s}\right] \tag{4}$$

Equivalent diameter calculation *(Neacsu E, 1977):*

$$D_{equ} = 2 \cdot H \left[m\right] \tag{5}$$

The length of all sheets and the length of the measured pipe:

$$L_{fin} = n_{fin} \cdot s \left[ml\right] \tag{6}$$

$$L_{pipe} = L - L_{pipe} \left[m\right] \tag{7}$$

Flow rate and air rate of flow through HEX *(Neacsu E, 1977)*:

$$A_{exchanger} = \left(L_{pipe} \cdot H_{exchanger}\right) - \left(n_{pipes} \cdot L_{pipe} \cdot d_e\right) \left[m^2\right] \tag{8}$$

$$w_{air} = \frac{\dot{m}_{aum}}{\rho_m \cdot A_{exchanger}} \left[\frac{m}{s}\right] \tag{9}$$

The surface of the pipe without fins reported at 0.8 meters of pipe:

$$A_B = \pi \cdot d_e \cdot \left(L - n_{fin} \cdot \frac{s}{L}\right) \left[m^2\right] \tag{10}$$

The surface of the fins is 0.8 meters of pipe:

$$A_N = \left[\left(R \cdot H - \frac{\pi \cdot d_e^{\,2}}{4}\right) \cdot 2 + \left(2 \cdot R + 2 \cdot H\right) \cdot s\right] \cdot \frac{n_{fin}}{L} \left[m^2\right] \tag{11}$$

Total area reported 0.8 meters of pipe *(Stamatescu C., 1979):*

$$A_T = A_B + A_N \,[m^2] \tag{12}$$

In order to establish the outdoor convection coefficient, the heat transfer characteristics of the air are determined, the Reynolds criterion and the Nusselt criterion have been calculated.

To determine the exterior convection coefficient, the following data are required: the kinematic viscosity of the air at medium temperature $\nu_{air} \left[\frac{m^2}{s}\right]$, average air density at medium temperature $\rho_{air} \left[\frac{m^3}{kg}\right]$, thermal conductivity of air at medium temperature $\lambda_{air} \left[\frac{W}{m \cdot K}\right]$, dynamic viscosity of air at medium temperature $\eta_{air} \left[\frac{N \cdot s}{m^2}\right]$, thermal conductivity of aluminum $\lambda_{Al} \left[\frac{W}{m \cdot K}\right]$, its values are established φ from *(Stamatescu C., 1979).*

$$\eta_{air} = \nu_{air} \cdot \rho_{air} \left[\frac{N \cdot s}{m^2}\right] \tag{13}$$

$$\mathrm{Re} = \frac{w_{air} \cdot D_{equ}}{\nu_{air}} \left[-\right] \tag{14}$$

In order to determine the Nusselt criterion, the values of the constants *C* and *n* are needed. They are chosen from *Stamatescu C., (1979)*: $C = 0.096; n = 0.72$

$$Nu = C \cdot \mathrm{Re}^n \cdot \left(\frac{d_e}{a}\right)^{-0,54} \cdot \left(\frac{H}{a}\right)^{-0,14} [-] \tag{15}$$

Determination of the exterior convection coefficient *(Stamatescu,C., 1979):*

$$\alpha_n = \frac{Nu \cdot \lambda_{air}}{a} \left[\frac{W}{m^2 \cdot K}\right] \tag{16}$$

It is considered that the real distribution of temperatures in the rib is replaced by a constant average temperature, where it was shown that by solving the differential equation of conductivity under the conditions of uniqueness from the variation of temperatures in a solid depend on a parameter *(Stamatescu C., 1979):*

$$m = \sqrt{\frac{2 \cdot \alpha_n}{\lambda_{Al} \cdot s}} [-] \tag{17}$$

Feature and efficiency fin *(Stamatescu C., 1979):*

$$X = \frac{\varphi \cdot d_e}{2} \cdot m [-] \tag{18}$$

$$\eta_n = \frac{\tanh \cdot (X)}{(X)} [-] \tag{19}$$

The external convection coefficient was obtained by analytical calculation, referring to the entire ribbed surface of HEX.

Convection coefficient in the dry air area relative to the total area *(Stamatescu C., 1979):*

$$\alpha_c = \frac{\alpha_n}{A_T} \cdot (\eta_n \cdot A_N + A_B) \left[\frac{W}{m^2 \cdot K}\right] \tag{20}$$

Exterior convection coefficient relative to the total area *(Stamatescu C., 1979):*

$$\sigma = 4 \cdot \alpha_C \left[\frac{W}{m^2 \cdot K}\right] \tag{21}$$

$$\alpha_{ext} = \frac{\sigma \cdot (h_m - h_p)}{t_m - t_p} \left[\frac{W}{m^2 \cdot K}\right] \tag{22}$$

The refrigerant for ACS is R134a, thus determining its thermophysical properties at the inlet to the evaporator and at the evaporating temperature of 5°C, such as the density of R134a , $\rho_{R134a} \left[\frac{kg}{m^3}\right]$ kinematic viscosity R134a – liquid, $\nu_{R134a,l} \left[\frac{m^2}{s}\right]$ thermal conductivity R134a – liquid , $\lambda_{R134a,l} \left[\frac{W}{m \cdot K}\right]$, specific heat R134a – liquid , $c_{p_{R134a,l}} \left[\frac{J}{kg \cdot K}\right]$, dynamic viscosity R134a – liquid , $\eta_{R134a,l} \left[\frac{N \cdot s}{m^2}\right]$ *(ASHRAE, 2009)*

$$\eta_{R134a,l} = \rho_{R134a,l} \cdot \upsilon_{R134a,l} \left[\frac{N \cdot s}{m^2}\right] \tag{23}$$

The diameter of the inner pipe can be deduced from the existing evaporator measurements and thus determine the flow rate of the agent inside the existing pipe and the inner surface of the pipe on a pipe 0.8 meters long.

$$A_{pipe} = \frac{\pi \cdot d_i^2}{4} \left[m^2\right] \tag{24}$$

R134a agent flow rate through the pipe *(ASHRAE, 2009):*

$$w_{R134a} = \frac{\dot{m}_o}{A_{teava} \cdot \rho_{R134a,l} \cdot 5} \left[\frac{m}{s}\right] \tag{25}$$

In the following, we apply Shah criterion equations which check the existing HEX to determine the interior convection coefficient, the pipe length, and the overall heat exchange coefficient. *(Shah M, 1982)*

In the calculation of the criteria equations for the indoor convection coefficient, the following data are required: the thermal conductivity of copper *(Neacsu E, 1977*): $\lambda_{Cu} \left[\frac{W}{m \cdot K}\right]$.

The evaporator pipes are made of copper and have a thickness of: $\delta_{pipe}[m]$ gravitational acceleration *(ASHRAE, 2009*): $g \left[\frac{m}{s^2}\right]$, latent heat of evaporating for R134a to $t_0 = 5°C]$: $\alpha_{fg} \left[\frac{kJ}{kg}\right]$ *(Elsayed A, 2011)*

Determination of the Reynolds criterion *(Neacsu E, 1977; Mostafa G., 2008)*

$$\mathrm{Re}_{R134a} = \frac{\rho_{R134a,l} \cdot w_{R134a} \cdot di}{\eta_{R134a,l}} \left[-\right] \tag{26}$$

Determination of the Reynolds criterion at the evaporator inlet *(Elsayed A., 2011; Shah M., 1982)*

$$\mathrm{Re}_{R134a,l} = \mathrm{Re}_{R134a} \cdot \left(1 - x_4\right) \left[-\right] \tag{27}$$

Determining the Prandtl criterion *(Collier J., 1996)* and the coefficient of mass velocity of the fluid *(Mostafa G., 2008):*

$$\mathrm{Pr}_{R134a,l} = \frac{c_{pR134a,l} \cdot 10^3 \cdot \eta_{R134a,l}}{\lambda_{R134a}} \left[-\right] \tag{28}$$

$$q'' = \frac{\dot{Q}_o}{A_{pipe} \cdot 5} \left[\frac{W}{m^2}\right] \tag{29}$$

$$G = \frac{\dot{m}_o}{A_{pipe} \cdot 5} \left[\frac{kg}{m^2 \cdot s}\right] \tag{30}$$

The coefficient Bo and the coefficient α_f are calculated *(ASHRAE, 2009; Shah, M, 1982)*

$$Bo_{R134a} = \frac{q''}{G \cdot \alpha_{fg}} \left[-\right] \tag{31}$$

$$\alpha_f = 0{,}023 \cdot \mathrm{Re}_{R134a,l}^{\;0{,}8} \cdot \mathrm{Pr}_{R134a}^{\;0{,}4} \cdot \frac{\lambda_{R134a,i}}{d_i} \left[\frac{W}{m^2 \cdot K}\right] \tag{32}$$

This determines the convection coefficient inside the pipes *(ASHRAE, 2009; Shah, M, 1982)*

$$Co = \left(\frac{1 - x_4}{x_4}\right)^{0{,}8} \cdot \left(\frac{\rho_{R134a,\upsilon}}{\rho_{R134a,l}}\right)^{0{,}5} \left[-\right] \tag{33}$$

$$Frl = \frac{G^2}{\rho_{R134a,l}^{\;2} \cdot g \cdot d_i} \left[-\right] \tag{34}$$

$$\alpha_{R134a} = F \cdot \left(Bo_{R134a,l}^{0{,}5}\right) \cdot \exp\left(2{,}47\left(Co\left(0{,}38 \cdot Frl^{-0{,}3}\right)^n\right)^{-0{,}15}\right) \cdot \alpha_f \left[\frac{W}{m^2 \cdot K}\right] \tag{35}$$

Global heat exchange coefficient *(Collier, J., 1996)*

$$k = \frac{1}{\left(\frac{1}{\alpha_{ext}}\right) + \left(\frac{A_T}{A_{pipe}}\right) \cdot \left(\frac{1}{\alpha_{R134a}} + \frac{\delta_{pipe}}{\lambda_{Cu}}\right)} \left[\frac{W}{m^2 \cdot K}\right] \tag{36}$$

The average temperature difference is calculated by neglecting the steam overheating heat, so the total length of the HEX pipe can be determined.

Maximum and minimum temperature difference at the evaporator *(Mostafa, G., 2008; Collier, J., 1996):*

$$\Delta t_{\max} = t_1 - t_0 \left[{}^oC\right] \tag{37}$$

$$\Delta t_{\min} = t_2 - t_0 \left[{}^oC\right] \tag{38}$$

Average logarithmic temperature difference at evaporator *(Mostafa, G., 2008; Collier, J., 1996):*

$$\Delta t_{m\log} = \frac{\Delta t_{\max} - \Delta t_{\min}}{\ln\left(\frac{\Delta t_{\max}}{\Delta t_{\min}}\right)} \left[{}^oC\right] \tag{39}$$

The cooling capacity of the evaporator is determined by the equation below:

$$\dot{Q}_o = \left(L_{total} \cdot k \cdot \Delta t_{m\log} \cdot A_T\right) \cdot 0.8 \left[W\right] \tag{40}$$

RESULTS

The calculations have been done using Engineering Equations Solver and the following operational conditions of the VCRS: evaporating temperature: $t_0 = 5°C$; condensing temperature: $t_c = 35°C$; cooling capacity: $\dot{Q}_o = 2.5\ kW$; subcooling degree: $\Delta_{sr} = 5°C$; superheating degree: $\Delta_{si} = 5°C$; refrigerant R134a inside evaporator pipes; moist air outside evaporator pipes and the following parameters: inlet air temperature in the evaporator: $t_{ia} = 30°C$; relative humidity of the air at the inlet: $\varphi_{ia} = 50\%$; air temperature at the outlet: $t_{oa} = 25.2°C$; relative humidity of the air at the outlet: $\varphi_{oa} = 61\%$; air flow by volume: $\dot{V} = 1000\frac{m^3}{h}$;

Below, a graphical comparison will be made between different refrigerants in the field of air conditioning, to see what the differences are if the R134a refrigerant is replaced.

In figure 8, the degree of subcooling at the outlet of the condenser was analyzed in relation to the convection coefficient inside the pipe for different refrigerants in the field of air conditioning. notice how the value of the function decreases in relation to the variable.

This graph can be divided into two areas, agents with the lowest values around the refrigerant R134a, and in the upper part of the graph refrigerants around R410a, with the highest values of the indoor convection coefficient, at the degree of subcooling with the lowest values, as the value of the degree of subcooling increases, the values of α_{int} decrease for the studied refrigerants.

In order to have a high-performance air-conditioning system, the variation of the degree of subcooling in relation to the cooling capacity for the studied refrigerants as substitutes for R134a was analyzed. It is observed in figure, that there are quasi-constant values of the cooling capacity for all refrigerants in relation to the value of the degree of subcooling.

In the lower part of the graph, there are two refrigerants R1234yf and R515A, the latter has as a base agent R1234ze(E) being an azeotropic agent, in the middle part of the graph are the newly studied refrigerants such as R454B, R452B and R466A, having better performances compared to R134a, and in the upper area of the graph are represented refrigerants, R410A, R32 and R407C, with the best performances of the refrigerating power, with an average increase of 40% compared to R134a.

This analysis resulted in a quasi-constant refrigerating power for all agents studied as substitutes for R134a, thus the total refrigerating power increases due to the fact that we have an increase in density due to vaporization pressure and due to the reduction of the compression ratio, a higher mass flow result.

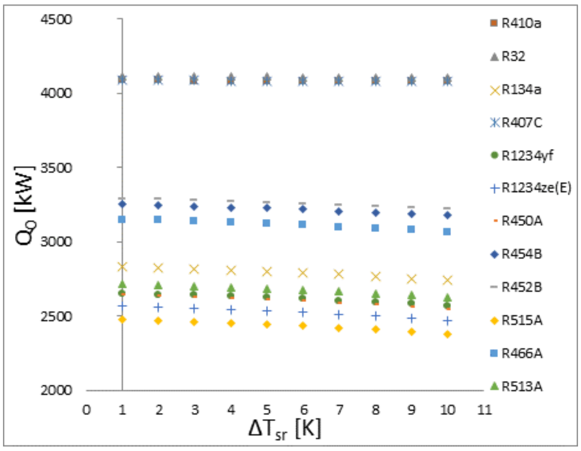


Fig. 7 - Influence of cooling degree on cooling capacity for R134a substituents

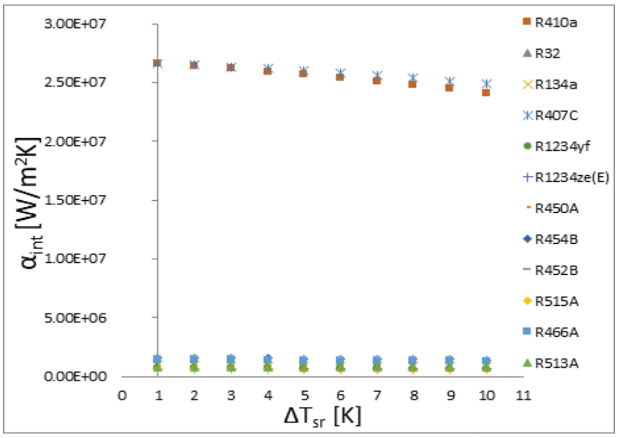


Fig. 8 - Influence of the degree of subcooling on the internal convection coefficient for R134a substituents

The influence of the degree of subcooling at the exit from the condenser was studied, taking into account the heat exchange coefficient for different refrigerants in the field of air conditioning, where a slight decrease in the overall heat exchange coefficient is observed as the value of the degree of subcooling increases for refrigerants in the lower part of the graph, and in the upper part refrigerants have quasi-constant values as the value of the degree of subcooling increases.

The graph in figure 9 can be divided into three groups, respectively in the lower part you can find the refrigerants R515A and R1234yf, with the lowest values, in the middle part you can find the newly studied agents R454B, R452B and R466A, and in the lower part at the top of the graph, there are 3 refrigerants with the best performances at the lowest values of the degree of subcooling, they have an average increase of 40% compared to R134a. The best value of the global heat exchange coefficient is with refrigerants R410A, R32 and R407C.

In figure 10, the influence of the degree of subcooling over the quality at the inlet to the evaporator was examined, where it has a tendency to decrease linearly in relation to the variation of the degree of subcooling; the more liquid enters the evaporator, the higher the efficiency of the heat exchanger.

The graph in figure 10 can be divided into three groups, respectively at the bottom is the cooling agent R32 with the lowest values, most agents occupy the idle of the graph, where the refrigerant R513a is highlighted with an increase of 10% compared to R134a, and in the upper part of the chart there are 2 refrigerants with the highest values, respectively R407C, R1234yf, they have an average increase of 20% compared to R134a.

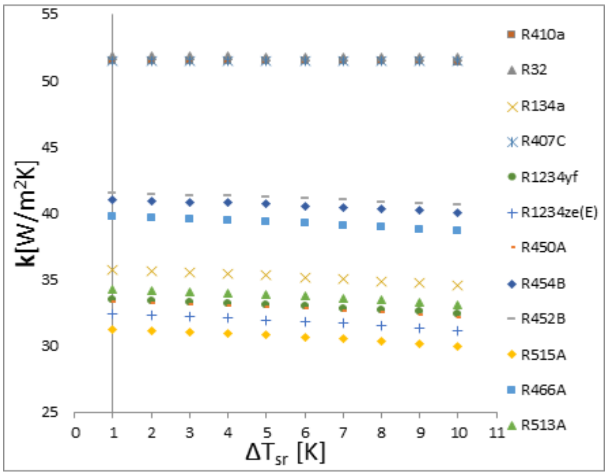


Fig. 9 - Influence of the degree of subcooling on the overall heat exchange coefficient for R134a substituents

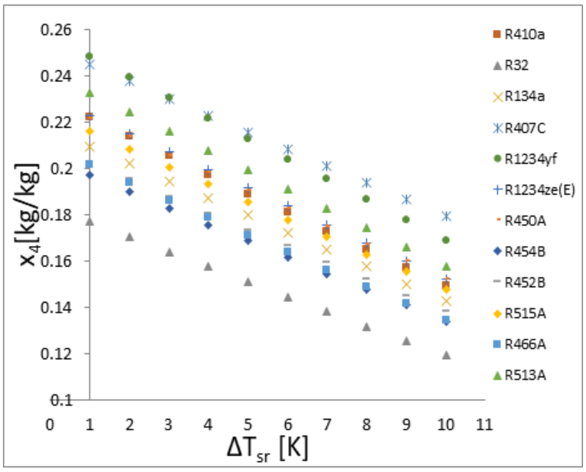


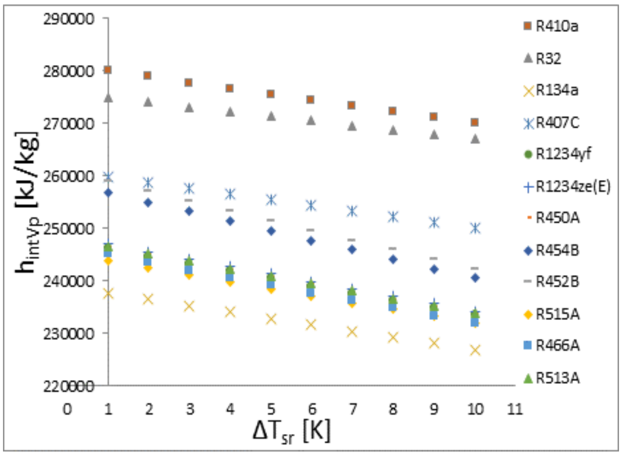
Fig. 10- Influence of the degree of subcooling on the vapor inlet quality for R134a substituents

The influence of the degree of subcooling is represented in figure 11, taking into account the enthalpy of the refrigerant at the entrance to the evaporator, the general orientation for all refrigerants is of linear decrease compared to the increase of the degree of subcooling.

As in the previous graphs and in this graph, it can be divided into 3 areas, in the lower area there are agents around R134a, in the middle part there are new studied agents R454B, R452B and R466A, but also R407C, and in the upper part they remained only agents R32 and R410a with the best performances.

In the graph from figure 12, the degree of subcooling is studied according to the enthalpy of the refrigerant at the condenser exit, so the general direction of refrigerants is increasing in relation to the degree of subcooling. At the bottom of the figures, agent R466A is represented, with the lowest values, and agent R452B is represented with the highest values.

It can be observed that at low values of the degree of subcooling, all agents have similar results, and as the degree of subcooling increases, several antifreeze agents with the highest values R452B and R454B are differentiated.



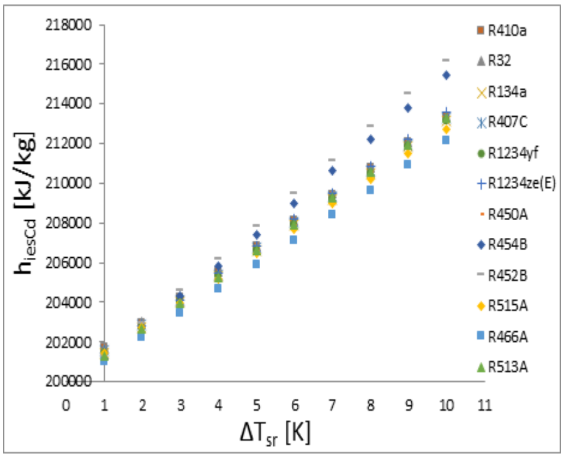


Fig. 11 - Influence of the degree of subcooling on the enthalpy at the input to the evaporator for R134a substituents

Fig. 12- Influence of the degree of subcooling on the enthalpy at the outlet of the condenser for R134a substituents

CONCLUSIONS

The values of vapor quality at the inlet of the evaporator directly influence the convection coefficient inside the pipes, so we analyzed the variation of the degree of subcooling in relation to the convection coefficient inside the pipes, where we noticed that as the degree of subcooling increases, t the cooling capacity of the heat exchanger decreases, because not enough liquid enters it to realize the heat transfer; the degree of subcooling is influenced by the condensation temperature.

The refrigerant R1234ze(E) is considered a resident agent of R134a, having similar or weaker characteristics in terms of energy and refrigeration performance, with a GWP of 1, but with a flammability class of A2L, this agent can be replaced without making too many changes in the air conditioning system.

A perspective refrigerant with the best performance is the agent R452B, it had the highest values in the graphs among the new refrigerants studied, with a GWP of 676, with a flammability class of A2L, but to replace R134a, the whole installation must be checked, so that it can withstand the higher working pressure, the refrigerant oil must be replaced and the rolling valve must be adjusted to achieve the appropriate liquid injection.

The air conditioning system maintains its cooling capacity when the refrigerant is replaced, and, by adjusting the sub-cooling process, the performance of the air conditioning system increases.

In the next stage, an experimental study will be performed on the air conditioning system with air treatment plant to analyze the installation parameters and model the results, comparing them with the technical data from the equipment manufacturer, and then the refrigerant will be replaced and a comparative study between agents will be performed.

Table 4

NOMENCLATURE

Symbols			
A_{pipe}	Pipe area [m^2]	$Pr_{R134,l}$	Prandlt's liquid criterion [-]
d_i	Inside diameter [m]	Q_0	Cooling capacity [kW]
H_{sc}	Height exchanger [m]	$q^{''}$	The transferred heat flow [W/m^2]
H_1	Enthalpy at point 1 [kJ/kg]	Re	Reynolds criterion [-]
H_m	Average enthalpy [kJ/kg]	Re_{R134a}	Reynolds criterion for R134a [-]
h_p	Enthalpy at the wall [kJ/kg]	$Re_{R134a,l}$	Reynolds criterion liquid R134a [-]
k	The overall heat exchange coefficient [W/m^2 K]	R	The length of the rectangular fin [m]
L_{pipe}	Measured pipe length [m]	s	The thickness of a fin [m]
m_o	Mass flow [kg/s]	t_m	Medium temperature [°C]
m_{aum}	Mass flow rate of moist air [kg/s]	t_p	Wall temperature [°C]
n_{fin}	The number of fins	V	Volumetric flow of moist air [m^3/h]
n_{pipe}	The number of pipes	w_{air}	Air flow speed [m/s]
Nu	Nusselt criterion [-]	X	Characteristic of the fin [-]
w_{R134a}	Flow rate of R134a [m/s]	x_1	The moisture content of the air [kg/kg]
Greek Symbols			
α_n	Convection coefficient for outside [W/m^2K]	$\lambda_{R134a,l}$	Thermal conductivity of R134a [W/mK]
$\lambda_{R134a,l}$	Thermal conductivity of R134a [W/mK]	ν_{air}	Kinematic viscosity of air [m^2/s]
ν_{air}	Kinematic viscosity of air [m^2/s]	$\nu_{R134a,l}$	Kinematic viscosity of R134a [m^2/s]
$\nu_{R134a,l}$	Kinematic viscosity of R134a [m^2/s]	η_n	Yield fin[-]
η_n	Yield fin[-]	η_{air}	Dynamic air viscosity [N s/m^2]
η_{air}	Dynamic air viscosity [N s/m^2]	$\eta_{R134a,l}$	Dynamic viscosity of R134a [N s/m^2]
$\eta_{R134a,l}$	Dynamic viscosity of R134a [N s/m^2]	ρ_{air}	Air density [kg/m^3]
ρ_{air}	Air density [kg/m^3]	$\rho_{R134a,l}$	Density of liquid R134a [kg/m^3]
$\rho_{R134a,l}$	Density of liquid R134a [kg/m^3]	$\rho_{R134a,v}$	Vapor density of R134a [kg/m^3]
$\rho_{R134a,v}$	Vapor density of R134a [kg/m^3]	ρ_m	Medium density [kg/m^3]
ρ_m	Medium density [kg/m^3]	φ_m	Average relative humidity of the air [%]
φ_m	Average relative humidity of the air [%]	$\lambda_{R134a,l}$	Thermal conductivity of R134a [W/mK]

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