MATHEMATICAL MODELING OF AN EVAPORATOR BY USING DIFFERENT CRITERIAL EQUATIONS
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MODELAREA MATEMATICĂ A UNUI VAPORIZATOR CU AJUTORUL ECUATIILOR CRITERIALE

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ABSTRACT

The paper aims to compare and choose the most suitable criteria equation for the computation of the convective heat transfer coefficient on the refrigerant side of the evaporator. The comparison is carried out by considering a given air handling unit equipped with a specific evaporator operating with refrigerant R134a. A mathematical model is given. The results obtained for criteria equations available in the work of Shah (1982), Gungor and Winterton (1987) and Kandlikar (1999) have been compared. The third criterion equation of Shah (1982) is selected because it leads to a value for the evaporator pipe length close to the one measured on the air handling unit and also it can be applied for a wider range of refrigerants used in the field of air conditioning.

REZUMAT

Lucrarea își propune să compare și să aleagă cea mai potrivită ecuație criterială pentru calculul coeficientului de transfer de căldură de convecție pe partea de agent frigorific a unui vaporizator. Comparația se realizează luând în considerare o anumită unitate de tratare a aerului echipată cu un vaporizator specific care funcționează cu agent frigorific R134a. Se realizează un model matematic. Au fost comparate rezultatele obținute pentru ecuațiile criteriale disponibile în lucrările lui Shah (1982), Gungor și Winterton (1987) și Kandlikar (1999). A treia ecuație criterială a lui Shah (1982) este selectată deoarece conduce la o valoare pentru lungimea conductei vaporizatorului apropiată de cea măsurată pe unitatea de tratare a aerului și de asemenea, poate fi aplicată pentru o gamă mai largă de agenți frigorifici utilizați în domeniul aerului condiționat.

INTRODUCTION

In zootechnical farms, the climatization and ventilation systems are decisive for animal welfare. The ventilation systems from the zootechnical spaces control luminosity, air temperature and humidity, as well as air composition, pressure and movement.

Current research in the field of air conditioning focuses mainly on improving the overall performance and efficiency of the system. The evaporator is a component that significantly influences the overall performance of the entire air conditioning system also known as air handling unit (AHU). However, the effect of design changes to evaporators is not always intuitive, and tests are needed to verify and quantify their effects (*Mason and Kincheloe, 2021*).

In order to properly design an evaporator for a given AHU application, one must choose the appropriate criteria equations that describe the heat transfer on the refrigerant side and on the air side. In literature, there are several criteria equations available that describe the phase change heat transfer inside the evaporator and also the heat transfer on the air side (*Jin and Wen, 2020*). In the particular case of evaporators, the cooled air is actually moist air and not dry air.

The purpose of this paper is to compare and choose the most suitable criteria equation for the phase change heat transfer on the refrigerant side of the evaporator. The comparison is carried out by considering a given AHU equipped with a specific evaporator operating with refrigerant R134a. The geometry of the evaporator is determined by using an identical evaporator as the one mounted on the AHU.

The thermodynamic properties of the refrigerant and of the moist air are determined for given operational conditions of the AHU. The most suitable criteria equation will be the one that will lead to a value of the total pipe length closest to the pipe length of the evaporator mounted on the AHU.

This work can be useful for specialist in the field, by pointing out the criteria equations needed to design an evaporator in the field of air conditioning. The results of this work should be regarded as recommendations and no to be generalized.

MATERIALS AND METHODS

1.1. Evaporator characteristics

The evaporator is a surface heat exchanger that achieves the air-cooling effect by forced convection using a fan. The cooling process in achieved by refrigerant phase change inside the evaporator pipes. The heat needed for the refrigerant phase change is absorbed from the moist air that is circulated on the outside of the evaporator. The cooled moist air is finally inserted in the conditioned enclosure *(IDewa et al., 2016).*

Fig.1 shows the hydraulic diagram of the vapour compression refrigeration system (VCRS) which is part of the AHU and is responsible for the direct air cooling (*Collier and Thorne, 1996*). The main components of the VCRS are evaporator, compressor, condenser and expansion valve. The other visible components are different valves, regulators, filters, pressure switches, temperature sensors and other devices which are not in focus in this work. Fig. 2 shows the corresponding thermodynamic cycle of the VCRS *(Uta, 2020).*

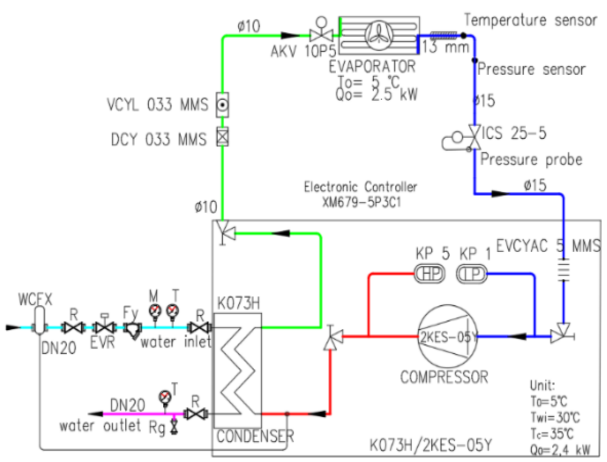

Fig. 1 - Hydraulic diagram of the refrigeration system
(*Collier and Thorne, 1996*)

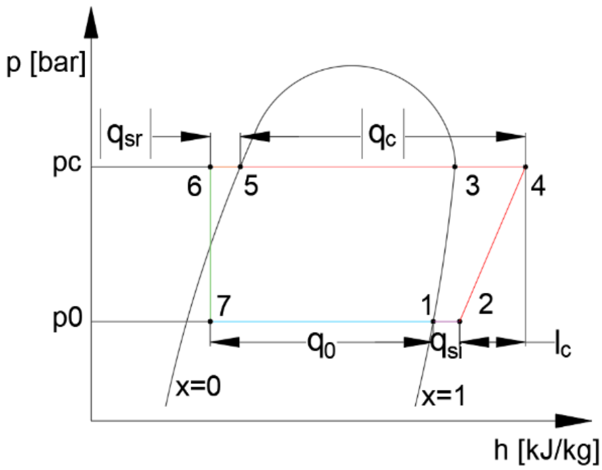

Fig. 2 - Thermodynamic cycle of the VCRS

The evaporator of the VCRS considered in the present work consists of copper pipes (coil) and aluminium fins, arranged on a single row with a galvanized sheet metal housing *(IDewa et al., 2016).* Its main characteristics and operation conditions are shown in Table 1.

Table 1

Characteristics of the AHU evaporator

(*Collier and Thorne, 1996*, *Uta, 2020, www.komfovent.com)*

Evaporator coil	**Length [mm]: 180**	**Pressure drop [Pa]: 7**	
Coil code	DX-G10-01R-0800-0300-130/-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
Refrigerant R134A		Superheat temperature [K]	5.0
Condensing temperature [°C]	35.0	Condensation [kg/hr]	1.22
Evaporating temperature [°C]	5.0	Inlet connection	1x½

Evaporator coil	**Length [mm]: 180**	**Pressure drop [Pa]: 7**	
Coil code	DX-G10-01R-0800-0300-130/-10-1×05C-24F-M1-C40-IS2-RC-1×½/1×22		Int. headers
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		

Fig. 3 and Fig. 4 show overview images of the evaporator mounted on the AHU.

Fig. 3 - Evaporator of the AHU frontal view

Fig. 4 - Evaporator of the AHU frontal view

The evaporator mounted in AHU was measured and the following data were obtained from Table 2.

Table 2

Measured values of the evaporator mounted in AHU

Evaporator coil			
Length [mm]	800	Width [mm]	30
Height [mm]	300	Number of fins [pcs]	253
The thickness of a fin [mm]	0,2	Inlet connection [mm]	16
Output connection [mm]	22	Number of liquid pipes [pcs]	5
Number of gas pipes [pcs]	5	Diameter of the liquid pipes	10
Diameter gas pipes [mm]	10	Length of a pipe [mm]	800
The total length of the pipes [mm]	8000	Total number of pipes [pcs]	10
Inside diameter of the pipe [mm]	8	Material of the pipes	Cu

The air-cooling process as it flows through the evaporator can be described in corelation with Fig. 5 and Fig. 6 (*Stamatescu et al.,1979*). At the entrance in the AHU, the fresh moist air is characterized by temperature t_1, humidity content x_1, enthalpy H_1 and relative humidity φ_1. The temperature of the evaporator pipe on the outside is t_p which is higher than the evaporating temperature t_0 (see Fig. 5).

Before entering the evaporator, the fresh moist air is mixed with warmer air coming from the conditioned enclosure and thus the thermodynamic state M will result. State M is characterized by temperature t_M, humidity content x_M, enthalpy h_M and relative humidity φ_M. As the moist air, having the thermodynamic state M, flows over the evaporator pipes, it will cool down towards t_p (process M–P in Fig.6).

Temperature t_2 is known, because it is the desired temperature of the air entering the conditioned enclosure after passing through the AHU. State 2 will result at the intersection between the t_2 isotherm and the line connecting the states M and P. State 2 is characterized by humidity content x_2, enthalpy H_2 and relative humidity φ_2. From Fig. 6 one can notice that the air-cooling process leads to a decrease in the humidity content from x_1 to x_2. Also, the air cools down from t_1 to t_2, leading to a temperature difference Δt between the inlet and the outlet of the AHU (see Fig. 5). At the same time, between the temperature of the moist air at the outlet of evaporator t_2 and the evaporating temperature t_0 there is a temperature difference Δt_{vp} needed to ensure a proper heat transfer process.

Inside the evaporator pipe, the refrigerant will evaporate and then will undergo a superheating process characterized by the superheating degree Δt_{si}, due to the heat absorbed from the air in the cooling process M-2, visible in Fig. 6.

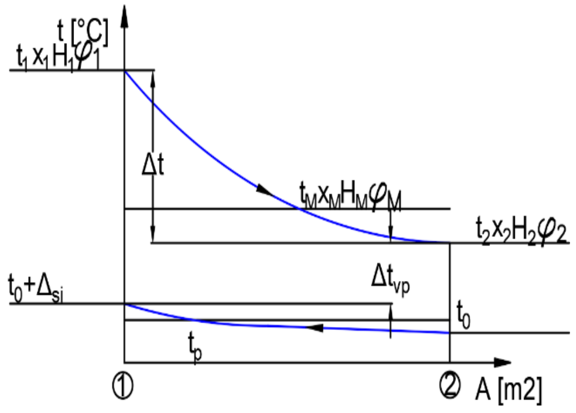

Fig. 5 - Variation of temperatures and humid air (vapor gas) along the contact surface with the wall of a pipe (*Stamatescu et al.,1979*)

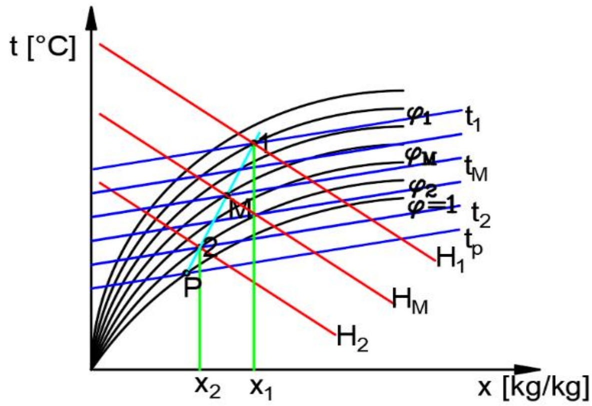

Fig. 6 - Specific thermodynamic states at the inlet and outlet of the air from the evaporator (*Stamatescu et al.,1979*)

In the next section, a mathematical model that describes the geometry of the evaporator and the heat transfer mechanism is presented.

Mathematical model

In a first stage the mass flow rate, volume flow rate and heat transfer area on the moist air side are calculated starting from the dimensions obtained from measuring the evaporator mounted on the AHU.

The moist air mass flow rate can be computed as:

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

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

The humid air volume flow rate and the mass vapor flow rate are:

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

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

The evaporator characteristics are: height H_{sc} $[m]$, the length of a pipe L $[m]$, the width of the rectangular aluminium fin H $[m]$, the length of the rectangular aluminium fin R $[m]$, the distance between two aluminium fin *a* [m], the thickness of an aluminium fin, *s* $[m]$ and pipe outer diameter d_e $[m]$.

The equivalent diameter can be calculated as:

$$D_{ech} = 2 \cdot H \; [m] \tag{5}$$

The length of all fins and measured pipe length is determined:

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

$$L_{pipe} = L - L_{fin} \; [m] \tag{7}$$

Flow area and moist air flow speed through the evaporator can be determined *(Neacsu and Naghi, 1977):*

$$A_{evaporator} = (L_{pipe} \cdot H_{sc}) - (n_{pipe} \cdot L_{pipe} \cdot d_e) \;\; [m^2] \tag{8}$$

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

The surface of the pipe without aluminium fins for 0.8 meters of pipe length is:

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

The surface of the aluminium fins corresponding to 0.8 meters of pipe is:

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

Total heat transfer area for 0.8 meters of pipe length is (*Stamatescu et al.,1979*):

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

It was calculated with 0.8 meters as the length of a pipe, because the existing evaporator was measured and it resulted that the length of a pipe has the value of 0.8 meters, and these data are in detail in Table 2.

In a second stage, the convective heat transfer coefficient on the moist air side is determined. In order to do so, the following data are required: kinematic viscosity of air at mean temperature ν_{air} $\left[\frac{m^2}{s}\right]$, mean density of air at mean temperature ρ_{air} $\left[\frac{m^3}{kg}\right]$, thermal conductivity of air at the mean temperature λ_{air} $\left[\frac{W}{m\cdot K}\right]$, the dynamic viscosity of the air at the mean temperature η_{air} $\left[\frac{N\cdot s}{m^2}\right]$, the thermal conductivity of aluminum, λ_{Al} $\left[\frac{W}{m\cdot K}\right]$ (*Stamatescu et al.,1979*).

With the data presented before, the Reynolds number can be computed:

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

$$Re = \frac{w_{air}\cdot D_{ech}}{\nu_{air}} [-] \tag{14}$$

In order to determine the Nusselt criterion, the values of the constants C and n are needed, and they are chosen from (*Stamatescu et al.,1979*). $C = 0{,}096$; $n = 0{,}72$

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

The convection coefficient on the outside of the aluminium fin is (*Stamatescu et al.,1979*):

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

For the calculation of the aluminium fin efficiency, a fin is exemplified in Fig. 7 in the most general case, as well as the notations used below: α_N - is the surface of the fin, α_B - is the base surface of the rib, α_i - is the total heat transfer surface of the fin (*Stamatescu et al.,1979*).

It is considered that the real distribution of temperatures in the fin is replaced by a constant average temperature t_N, where it was shown that by solving the differential conductivity equation under the conditions of uniqueness in Fig. 7, the variation of temperatures in solids depends on a parameter (*Stamatescu et al.,1979*).

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

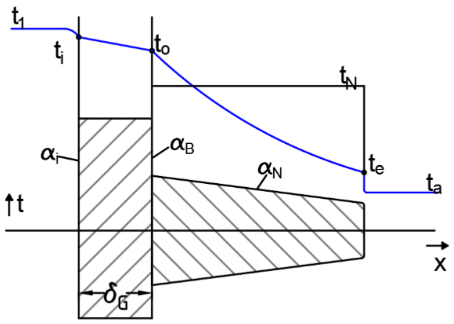

Fig. 7 - Temperature variation in a finned solid (*Stamatescu et al.,1979*)

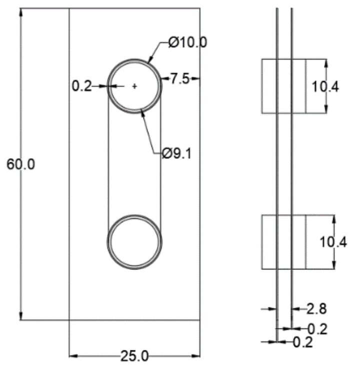

Fig. 8 - Representation of rectangular fins with actual measured data see in Table 2 (*Stamatescu et al.,1979*)

In Fig. 8 and Fig. 9 there are two sketches with the geometric details of the fins, where the thickness of the fins, the width of the rectangular fins, the outer radius of the pipe, the length of the rectangular fins can be found (*Stamatescu et al.,1979*).

The aluminium fins efficiency can be determined (*Stamatescu et al.,1979*):

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

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

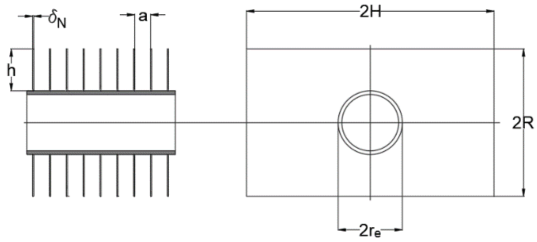

Fig. 9 - Detail with the dimensions of the fins
(*Stamatescu et al.,1979*)

The convective heat transfer coefficient on the moist air side was obtained by analytical calculation, referring to the entire finned surface of the evaporator.

Convection coefficient for dry air relative (*Stamatescu et al.,1979*):

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

The convective heat transfer coefficient relative (*Stamatescu et al.,1979*):

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

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

In formula (22), t_m represents the average value between the air temperature at the inlet to the evaporator and the air temperature at the outlet from the evaporator, and H_m represents the average value between the enthalpy of the air at the inlet and the enthalpy of the air at the outlet from the evaporator, *Hp* and *tp* are the values of the enthalpy and the air temperatures at the wall.

In a third stage, the convective heat transfer coefficient on the refrigerant side is determined as presented next.

The refrigerant of the VCRS mounted on the AHU is R134a. Its thermophysical properties are determined at an evaporating temperature of 5°C. The thermophysical properties needed to compute the convective heat transfer coefficient on the refrigerant side are: density ρ_{R134a} $\left[\frac{kg}{m^3}\right]$, kinematic viscosity in liquid state $\nu_{R134a,l}$ $\left[\frac{m^2}{s}\right]$, thermal conductivity in liquid state $\lambda_{R134a,l}$ $\left[\frac{W}{m \cdot K}\right]$, specific heat of R134a in liquid state $c_{p_{R134a,l}}$ $\left[\frac{J}{kg \cdot K}\right]$ and dynamic viscosity in liquid state $\eta_{R134a,l}$ $\left[\frac{N \cdot s}{m^2}\right]$ (*ASHRAE: Fundamentals, 2009).*

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

The measurements of the existing evaporator show the diameter of the inner pipe d_i. With this value, the inner surface of the pipe for a length of 0.8 meters is:

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

The refrigerant velocity is (*ASHRAE: Fundamentals, 2009):*

$$w_{R134a} = \frac{\dot{m}_0}{A_{pipe} \cdot \rho_{R134a,l} \cdot 5} \left[\frac{m}{s}\right] \tag{25}$$

In relation (26), in order to calculate the velocity of the refrigerant in the existing evaporator, we need the mass flow rate ($\dot{m}_0$), the area of the pipe (A_{pipe}), the density of the liquid fluid ($\rho_{R134a,l}$) and the number of liquid pipes that enter the evaporator, where in our application it has the value of 5, this is in Table 2.

Next, the equations proposed by Shah in 1982, are used as an example how to determine the inner convective heat transfer coefficient, the overall heat transfer coefficient, and the pipe length (*Shah, 1982*).

In calculating the criterion equations for the inner convective heat transfer coefficient, the following data are required: thermal conductivity of copper *(Neacsu and Naghi, 1977):* λ_{Cu} $\left[\frac{W}{m \cdot K}\right]$, the copper pipe thickness: δ_{pipe} $[\mathrm{m}]$ gravitational acceleration *(ASHRAE: Fundamentals, 2009)*: g $\left[\frac{m}{s^2}\right]$, latent heat of evaporating for R134a at $t_0 = 5°C$ *(Elsayed, 2011)*: α_{fg} $\left[\frac{J}{kg}\right]$.

The Reynolds number can be determined *(Neacsu and Naghi, 1977), (Ghiaasiaan, 2008):*

$$Re_{R134a} = \frac{\rho_{R134a,l} \cdot w_{R134a} \cdot d_i}{\eta_{R134a,l}} \, [-] \tag{26}$$

The Reynolds number at evaporator inlet (*Shah, 1982*) is:

$$Re_{R134a,l} = Re_{R134a} \cdot (1 - x_4) [-] \tag{27}$$

The Prandtl number *(Collier and Thorne, 1996)* and the coefficient of mass velocity of the fluid *(Ghiaasiaan, 2008)* are:

$$Pr_{R134a} = \frac{c_{p_{R134a,l}} \cdot 10^3 \cdot \eta_{R134a,l}}{\lambda_{R134a,l}} \; [-] \tag{28}$$

$$q'' = \frac{\dot{Q}_0}{A_{pipe} \cdot 5} \left[\frac{W}{m^2}\right] \tag{29}$$

$$G = \frac{\dot{m}_0}{A_{pipe} \cdot 5} \left[\frac{kg}{m^2 \cdot s}\right] \tag{30}$$

The coefficient *Bo* and the coefficient α_f (*ASHRAE: Fundamentals, 2009, Shah, 1982*). are:

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

$$\alpha_f = 0{,}023 \cdot Re_{R134a,l}{}^{0,8} \cdot Pr_{R134a}{}^{0,4} \cdot \frac{\lambda_{R134a,l}}{d_i} \left[\frac{W}{m^2 \cdot K}\right] \tag{32}$$

The convective heat transfer coefficient inside the pipe (*ASHRAE: Fundamentals, 2009, Shah, 1982*) is n=0 and F=14,7:

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

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

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

The overall heat transfer coefficient *(Collier and Thorne, 1996)* is:

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

The average temperature difference is calculated by neglecting the superheating process.

Maximum and minimum temperature differences for the evaporator (*Ghiaasiaan, 2008*),

$$\Delta t_{max} = t_1 - t_0 \; [°C] \tag{37}$$

$$\Delta t_{min} = t_2 - t_0 \; [°C] \tag{38}$$

Mean logarithmic temperature difference at the evaporator *(Ghiaasiaan, 2008, Collier and Thorne, 1996):*

$$\Delta t_{mlog} = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)} \; [°C] \tag{39}$$

The total length of the pipe required to carry out the heat transfer process is:

$$\mathrm{L}_{total} = \left(\frac{\dot{Q}_0 \cdot 10^3}{k \cdot \Delta t_{mlog} \cdot A_T}\right) / 0{,}8 \; [ml] \tag{40}$$

The table also presents other criteria equations, available in the literature that can be used instead of relationship (33). In the next section an analysis is carried out in order to see which equation verifies the existing heat exchanger in the laboratory.

Table 3

Criteria equations for the inner convective heat transfer coefficient

Author	Criterion Equations
Shah (1982)	$Re_{R134a} = \frac{\rho_{R134a_l} \cdot w_{R134a} \cdot d_i}{\eta_{R134a}}$ $Re_{R134a_l} = Re_{R134a} \cdot (1 - x_4)$ $Bo_{R134a} = \frac{q''}{G \cdot \alpha_{fg}}$ $\alpha_f = 0{,}023 \cdot Re_{R134a_l}{}^{0,8} \cdot \; Pr_{R134a}{}^{0,4} \cdot \left(\frac{\lambda_{R134a}}{d_i}\right)$ $Frl = \frac{G^2}{\rho_{R134a_l}{}^2 \cdot g \cdot d_i}$
	$Co = \left(\frac{1 - x_4}{x_4}\right)^{0,8} \cdot \left(\frac{\rho_{R134a_v}}{\rho_{R134a_l}}\right)^{0,5}$ $\alpha_{R134a1} = 230 \cdot Bo_{R134a}{}^{0,5} \cdot \alpha_f$ $\alpha_{R134a2} = 1{,}8 \cdot (\, Co(0{,}38 Frl^{-0,3})^n)^{-0,8} \cdot \alpha_f$ $\alpha_{R134a3} = F \cdot Bo^{0,5} exp(2{,}47 \cdot (\, Co \cdot (0{,}38 \cdot Frl^{-0,3})^n)^{-0,15}) \cdot \alpha_f$ $\alpha_{R134a4} = F \cdot Bo^{0,5} \cdot exp(2{,}74 \cdot (\, Co \cdot (0{,}38 \cdot Frl^{-0,3})^n)^{-0,1}) \cdot \alpha_f$ $\mathrm{L}_{total} = \frac{\dot{Q}_0}{k \cdot \Delta t_{mlog} \cdot A_T}$

Author	Criterion Equations
Gungor and Winterton (1987)	$Re_{R134a} = \frac{\rho_{R134a_l} \cdot w_{R134a} \cdot d_i}{\eta_{R134a}}$ $Re_{R134a_l} = Re_{R134a} \cdot (1 - x_4)$ $Bo_{R134a} = \frac{q''}{G \cdot \alpha_{fg}}$ $Frl = \frac{G^2}{\rho_{R134a_l}{}^2 \cdot g \cdot d_i}$ $\alpha_f = 0{,}023 \cdot Re_{R134a_l}{}^{0,8} \cdot Pr_{R134a}{}^{0,4} \cdot \left(\frac{\lambda_{R134a}}{d_i}\right)$
	$E = 1 + 3000 \cdot Bo^{0,86}$ $S = 1{,}12 \cdot \left[\frac{x_4}{1 - x_4}\right]^{0,75} \cdot \left(\frac{\rho_{R134a_l}}{\rho_{R134a_v}}\right)$ $E_2 = Frl^{(0,1-2Frl)}$ $S_2 = Frl^{(1/2)}$ $\alpha_{R134a5} = E \cdot E_2 \cdot \alpha_f + S \cdot S_2 \cdot \alpha_f$ $\mathrm{L}_{total} = \frac{\dot{Q}_0}{k \cdot \Delta t_{mlog} \cdot A_T}$
Author	Criterion Equations
Kandlikar (1999)	$Re_{R134a} = \frac{\rho_{R134a_l} \cdot w_{R134a} \cdot d_i}{\eta_{R134a}}$ $Re_{R134a_l} = Re_{R134a} \cdot (1 - x_4)$ $Bo_{R134a} = \frac{q''}{G \cdot \alpha_{fg}}$ $Frl = \frac{G^2}{\rho_{R134a_l}{}^2 \cdot g \cdot d_i}$ $\alpha_f = \frac{Re_{R134a_l} \cdot Pr_{R134a} \cdot \left(\frac{f}{2}\right) \cdot \left(\frac{\lambda_{R134a}}{d_i}\right)}{1{,}07 + 12{,}7 + \left(Pr_{R134a}^{2/3} - 1\right) \cdot \left(\frac{f}{2}\right)^{0,5}}$
	$f = \left[1{,}58 \cdot \ln(Re_{R134a_l}) - 3{,}28\right]^{-2}$ $f_2 = 1$ $\alpha_{R134a6} = \alpha_f \cdot (0{,}6683 \cdot \left(\frac{\rho_{R134a_l}}{\rho_{R134a_v}}\right)^{0,1} \cdot x^{0,16} \cdot (1-x)^{0,64} \cdot f_2 + 1058 \cdot Bo^{0,7} \cdot (1-x)^{0,8} \cdot Fr_{R134a})$ $\alpha_{R134a7} = \alpha_f \cdot (1{,}136 \cdot \left(\frac{\rho_{R134a_l}}{\rho_{R134a_v}}\right)^{0,45} \cdot x^{0,72} \cdot (1-x)^{0,08} \cdot f_2 + 667{,}2 \cdot Bo^{0,7} \cdot (1-x)^{0,8} \cdot Fr_{R134a})$ $\mathrm{L}_{total} = \frac{\dot{Q}_0}{k \cdot \Delta t_{mlog} \cdot A_T}$

RESULTS

In this section, a comparison between different criterion equations used for the calculation of the inner convection heat transfer coefficient that will lead to a pipe length value closest to the actual one corresponding to the evaporator mounted on the AHU, is carried on. In order to determine the inner convection heat transfer coefficient, it was necessary to study different well-known authors in the field of refrigeration, heat exchangers and mass transfer. The authors studied are: Shah (1982), Gungor and Winterton (1987) and Kandlikar (1999), as presented in Table 2.

The calculations have been done using Engineering Equations Solver (*Klein, S. A.: Engineering Equation Solver*) and the following operational conditions of the VCRS: evaporating temperature: $t_0 = 5℃$; condensing temperature: $t_c = 35℃$; cooling capacity: $\dot{Q}_o = 2500\,W$; subcooling degree: $\Delta_{sr} = 5℃$; superheating degree: $\Delta_{si} = 5℃$; refrigerant R134a inside evaporator pipes; moist air outside evaporator pipes and the following parameters: inlet air temperature in the evaporator: $t_{\mathrm{ia}} = 30℃$; relative humidity of the air at the inlet: $\varphi_{\mathrm{ia}} = 50\%$; air temperature at the outlet: $t_{\mathrm{oa}} = 25{,}2℃$; relative humidity of the air at the outlet: $\varphi_{\mathrm{oa}} = 61\%$; air flow by volume: $\dot{V} = 1000\frac{m^3}{h}$;

Fig.10 presents the results obtained for the inner convective heat transfer coefficient using different equations available in literature. One can notice that the equations of Kandlikar (1999) lead to the highest values *as well as Gungor and Winterton*. The second criterion equation of Shah (1982) has a small value, so that this has not been taken into consideration; thus, we can say that, from Fig. 10, we do not take into account the extreme values, both the high and the low ones, and we choose those that apply to several refrigerants *(Shah, 1982)*.

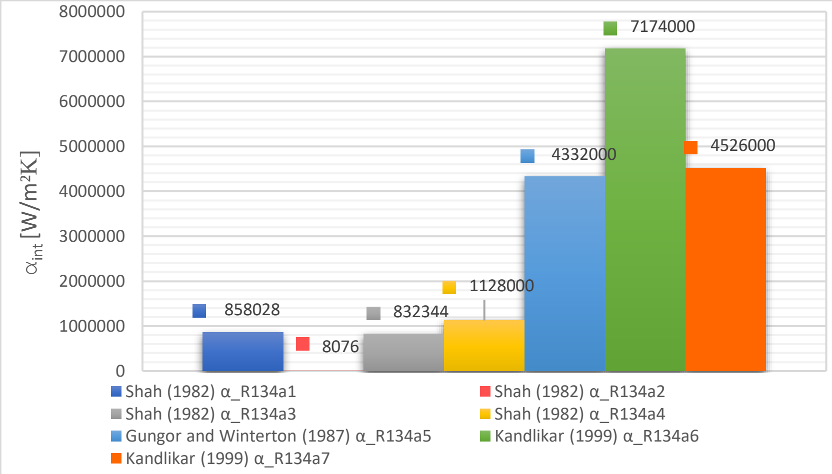

Fig. 10 - Graphic representation of α_{int} with different criteria equations
(Klein, S. A.: Engineering Equation Solver)

Fig.11 shows the overall heat transfer coefficient obtained using different equations available in literature. The same configuration of values can be observed as in Fig.10. The equations of Kandlikar (1999) *(Satish and Kandlikar,1999)* display higher values and the second equation of Shah (1982) *(Shah,1982)* leads to a small value for the overall heat transfer coefficient; thus we can say that from Fig.11, we do not take into account the extreme values, both the high and the low ones, and we choose those that apply to several refrigerants.

The rest of the values can be considered acceptable and the selected value for the overall heat transfer coefficient corresponds to the Shah's third equation (1982) applicable for a wide range of refrigerants, not only for R134a *(Shah, 1982)*. Thus, the overall heat transfer coefficient is directly influenced by the inner convective heat transfer coefficient, having a noticeable proportionality in the resulting values.

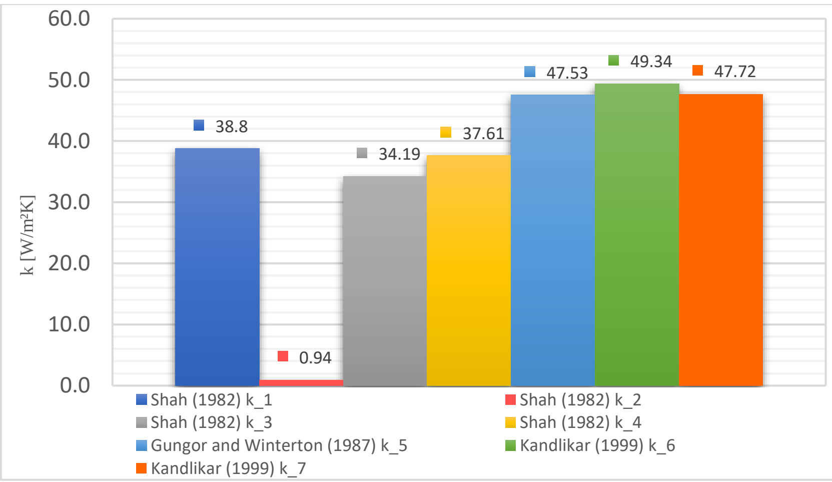

Fig. 11- Graphic representation of the overall heat transfer coefficient with different criteria equations
(Klein, S. A.: Engineering Equation Solver)

Fig. 12, the total pipe length of the evaporator obtained using different equations available in literature, is shown.

One can notice that the second equation of Shah (1982) has different values and does not correspond to the rest of the results.

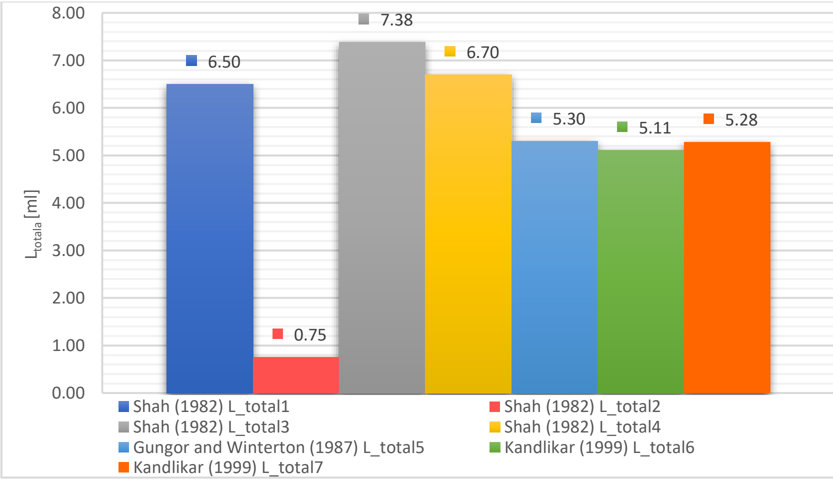

Fig. 12 - Graphic representation of total pipe length with different criteria equations
(Klein, S. A.: Engineering Equation Solver)

The criteria equation that leads to a pipe length closest to the one corresponding to the evaporator mounted on the AHU, seen in Table 2, is the third equation of Shah (1987) *(Shah,1985).* This equation is applicable for a large range of refrigerants.

As a general conclusion, from the results presented in Fig.11 to Fig.12, the third criterion equation of Shah (1982) is selected because it leads to a value for the evaporator pipe length close to the one measured on the AHU, seen in Table 2 and also because it can be applied for a wider range of refrigerants used in the field of air conditioning *(Shah, 1982).*

Other results of interest are shown in Table 3.

Table 2

Results of the criteria equations

Author	The results of the criterion equations	
Shah (1982)	$Re_{R134a} = 24962$ $Re_{R134a_l} = 20480$ $Bo_{R134a} = 0{,}8204$ $\alpha_f = 6049\ \frac{W}{m^2 \cdot K}$ $Frl = 2004$ $Co = 1{,}453$ $F = 14{,}7\ ;\ \ n = 0$ $\alpha_{R134a1} = 858028\ \frac{W}{m^2 \cdot K}$	$\alpha_{R134a2} = 8076\ \frac{W}{m^2 \cdot K}$ $\alpha_{R134a3} = 832344\ \frac{W}{m^2 \cdot K}$ $\alpha_{R134a4} = 1128000\ \frac{W}{m^2 \cdot K}$ $L_{total1} = 6{,}507\ ml$ $L_{total2} = 0{,}7494\ ml$ $L_{total3} = 7{,}378\ ml$ $L_{total4} = 6{,}7\ ml$
Author	The results of the criterion equations	
Gungor and Winterton (1987)	$Re_{R134a} = 24962$ $Re_{R134a_l} = 20480$ $Bo_{R134a} = 0{,}8204$ $\alpha_f = 6049\frac{W}{m^2 \cdot K}$ $Frl = 2004$ $E_2 = 0$	$E = 2531$ $S_2 = 1002$ $S = 0{,}7148$ $\alpha_{R134a} = 4332000\ \frac{W}{m^2 \cdot K}$ $L_{total5} = 5{,}307\ ml$
Author	The results of the criterion equations	
Kandlikar (1999)	$Re_{R134a} = 24962$ $Re_{R134a_l} = 20480$ $Bo_{R134a} = 0{,}8204$ $Frl = 2004$ $f = 0{,}006498$ $f_2 = 1$	$\alpha_f = 5596$ $\alpha_{R134a1} = 858028$ $\alpha_{R134a2} = 7174000$ $L_{total6} = 5{,}11$ $L_{total7} = 5{,}28$

CONCLUSIONS

The present work focuses on the comparison of several criteria equations available in literature used to compute the convective heat transfer coefficient inside the evaporator pipe, namely on the refrigerant side. The comparison is carried out by considering a given AHU equipped with a specific evaporator operating with refrigerant R134a. The geometry of the evaporator is determined by using an identical evaporator as the one mounted on the AHU. The most suitable criteria equation will be the one that will lead to a value of the total pipe length closest to the pipe length of the evaporator mounted on the AHU.

A description of the evaporator and its operation is given. Then, a mathematical model that describes the geometry of the evaporator, the heat transfer process outside and inside the pipes is given. The mathematical model is exemplified for the equations given by Shah (1982) and it was implemented in Engineering Equation Solver Software.

Based on the mathematical model, the results obtained for criteria equations available in the work of Shah (1982), Gungor and Winterton (1987) and Kandlikar (1999) have been compared. The values for the inner convective heat transfer coefficient, the overall heat transfer coefficient and the total pipe length of the evaporator have been compared. As a general conclusion, the third criterion equation of Shah (1982) is selected because it leads to a value for the evaporator pipe length close to the one measured on the AHU and also because it can be applied for a wider range of refrigerants used in the field of air conditioning.

Future work will involve a study of the AHU evaporator by considering its given dimensions and different refrigerants. This study will simulate the evaporator operation when the refrigerant is changed with an environmentally friendly one.

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