DIMENSIONING TUBE DIAMETER OF AIR COOLING COIL FOR CARBON DIOXIDE

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ABSTRACT

To study the influence of the tube diameter in a cooling coil working carbon dioxide as the refrigerant, calculations were carried out a specified finned plate and tube coil as an example. The main dimensions and the tube/fin pitching were held constant, only the tube diameter and the number of the parallel tube circuits were varied. The refrigerant side heat transfer was calculated using Thome and El Hajal -correlation in the two-phase region and Gnielinski -correlation in the superheating region. The pressure drop was evaluated applying Müller-Steinhagen-Heck and Darcy-Weinbach correlations.

Important criteria for the optimization of the mass flow velocity are the capacity of the coil and the saturation temperature at the end of the evaporator. When the mass flow velocity increases, the increasing heat transfer coefficient also rises the level of the evaporation temperature. On the other hand, the increase of the pressure loss lowers the pressure at the end of the tube circuit. Other variables affected are the air velocity, the air-side surface area and the fin efficiency. In principle, as the sum of these partly opposite effects an optimum tube dimension could be expected. However, the example shows that in the air cooling coil because of the dominance of the air-side heat transfer resistance and the superheating region, this kind of clear optimum could not be found in the range studied.

1. INTRODUCTION

The intstallations using carbon dioxide as the refrigerant increase rapidly in numbers and areas of applications. The properties of carbon dioxide differ in many respects from the traditional halocarbon refrigerants. As to the design and dimensioning of heat exchangers, the most important thermal features of carbon dioxide are good heat transfer and low pressure loss because of good thermodynamic properties. Often, when an air cooling evaporator or an air cooled condenser for carbon dioxide are requested, a manufacturer dimensions his coil using a traditional pipe size selected originally on the basis of the properties of halocarbon refrigerants. Therefore, a question of the optimum configuration of heat exchangers arises. Particularly, the influence of the tube diameter or the mass velocity is interesting. An important question is if a better performance could be achieved when the coil would be optimized for the particular properties of carbon dioxide.

For dimensioning of coils, a number of computer models, more or less detailed, have been set up. Some of them have been applied to sudy the influence of the circuit design and optimization of the coil, see e.g. Domanski (2004). Also earlier investigations on optimization of the tubes have been made but mainly for halocarbon refrigerants. Higher heat transfer and lower pressure drop of carbon dioxide have a clear influence. The purpose of this study was to find out how strongly the pipe size effects on the performance of an air cooling coil when using carbon dioxide as the refrigerant. When the mass velocity of carbon dioxide is increased, the heat transfer coefficient also increases rising the level of the evaporation temperature. On the other hand, the increase of the pressure loss lowers the pressure at the end of the tube circuit. With the tube diameter also air velocity decreases as well as air-side heat transfer and the fin efficiency. At least in principle, as the sum of all these partly opposite effects, an optimum tube dimension could be possible.

Because of high heat transfer rate of carbon dioxide, for the purposes of this sudy also a relatively simple modeling gives the required accurracy. The following simplifying assumptions were made:

1. a dry coil (no frosting);
2. the properties of air and CO_2 were determined at the average temperatures of air and refrigerant;
3. a smooth velocity profile of air;
4. the contact resistance between tube and fin was ignored
5. the cross flow in each section were used in modeling the coil.
6. the influence of the tube bends was omitted

2. HEAT TRANSFER COEFFCIENT OF CARBON DIOXIDE

The average heat transfer coefficient of carbon dioxide in each section of the coil was calculated by integrating the local values. However, using general models of evaporation heat transfer did not yield satisfactory results and therefore the main focus was on particular models for CO_2. Kattan, Thome and Favrat (1998) proposed a flow boiling model based on the local flow pattern that covers the annular flow, the annular flow with partial dryout, the intermittent flows, the stratified-wavy flow and fully stratified flow. Because typical mass flow velocities are well below 500 kg/sm^2 one can see from a flow pattern map for CO_2 (Figure 1) that no other flow patterns will encounter. However, it must be noted that actual flow patterns are difficult to predict and the map was created for a 6 mm tube and for a heat flux of 15 kW/m^2, which might give wrong predictions for other very different configurations. Since several independent researchers reported that the Kattan-Thome-Favrat model did not work satisfactorily when predicting the heat transfer coefficient of CO_2 evaporation, Thome and El Hajal (2004) modifed it to fit for the specific properties of CO_2.

For the stratified regime, the local heat transfer coefficient is calculated as a weighed average from dry and wet parts of the perimeter:

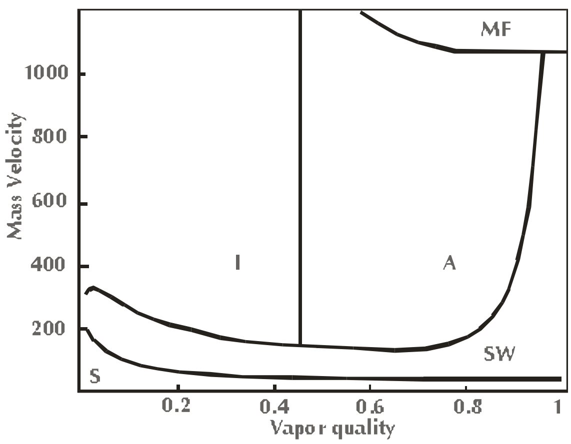


Figure 1. Principle of Thome - El Hajan flow map. S = stratified, SW = stratified wavy, I = intermittent, A = annular, MF = misty.

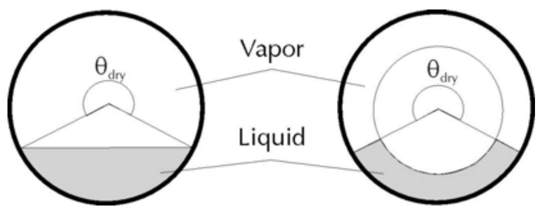


Figure 2. Transformation of stratified flow to “effective” annular flow of Thome and El Hajal model.

$$h_r = \frac{\theta_{dry} h_v + (2\pi - \theta_{dry}) h_{wet}}{2\pi} \tag{3}$$

θ_{dry} is the angle which is not wetted by the liquid phase (Fig. 2). θ_{dry} was calclated on the basis of the void fraction ε and the limits of the flow ranges applying the procedure given by Kattan, Thome and Favrat (1998). ε was calculated using the Rouhani and Axelsson (1970) drift flux model:

$$\varepsilon = \frac{x}{\rho_v} \left\{ (1 + 0.12(1-x)) \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right) + \frac{1.18}{\dot{m}} \left[\frac{g\sigma(\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4} (1-x) \right\}^{-1} \tag{4}$$

The heat transfer coeffcient h_{wet} between the tube wall and the liquid phase was calculated as a weighted mean from the values for convective and nucleate boiling regimes:

$$h_{wet} = (Sh_{cb}^3 + h_{nb}^3)^{1/3} \tag{5}$$

where

$$Nu_{nb} = \frac{h_{nb}\delta}{k_l} = 0.0133 \left[\frac{4G(1-x)\delta}{(1-\varepsilon)\mu_l} \right]^{0.69} Pr_l^{0.4} \tag{6}$$

The first term in the brackets is the liquid film Reynolds number. The thickness δ of the liquid film is defined as a virtual thickness liquid flowing in the segment of anlgle 2π - θ_{dry} filling the same cross sectional area (Fig. 2). The nuclear heat transfer coefficient is given by the formula which is a modification of the original formula to fit better the experimental values for CO_2:

$$h_{nb} = 3970 + 38.9 p_r^{0.12} (-\log p_r)^{-0.55} M^{-0.5} q^{0.67} \tag{7}$$

p_r is reduced pressure and M molar mass.

Another special property of CO_2 is the fact that the convective boiling contribution is larger than the nucleate one. Therefore a ceofficient S as a suppression factor was introduced:

$$S = \frac{(1-x)^{1/2}}{0.121 Re_l^{0.255}} \tag{8}$$

Heat transport at the dry part of the perimeter (h_v) was calculated using the simple well-known Dittus-Boelter-correlation.

3. PRESSURE DROP OF CARBON DIOXIDE

For frictonal pressure drop, on the basis of comparison of Didi *et al*. (2002) the correlation of Müller-Steinhagen and Heck (1986) was selected. The method gives the pressure gradient as a combination of the gradients of pure liquid (a) and vapour (b) flows:

$$\left(\frac{dp}{dL} \right)_f = Y(1-x)^{1/3} + bx \tag{9}$$

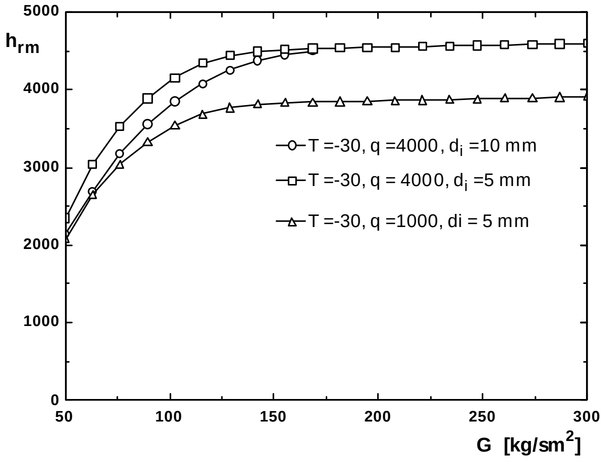


Figure 3. Example of variation of the average evaporation heat transfer coefficient. x = 0 – 0.95.

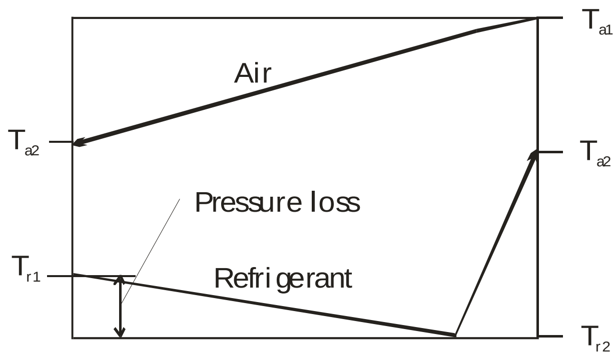


Figure 4. Schematic temperature profiles (pure counter flow).

$$Y = a + 2(b - a)x$$

The gradients a and b are calculated using the friction factors ξ_l and ξ_v:

$$a = \xi_l\, G^2 / 2\rho_l\, D \qquad (10)$$
$$b = \xi_v\, G^2 / 2\rho_v\, D$$

$$\xi_l = 0.184 Re_l^{-0.2} \qquad (11)$$
$$\xi_v = 0.184 Re_v^{-0.2}$$

Reynolds numbers R_{el} and R_{ev} are calculated the total flow as liquid and vapour, correspondingly.

The Equations (9-11) give also local values and the mean value for the whole length was calculated by integrating between x_{in} and 1. The pressure loss in the superheating part of the tube can be calculated using ξ_v from Equation (11).

4. MODEL OF THE COIL

The total capacity has to be calculated by integrating the local values because the variation of the tube-side heat transfer coefficient with the quality depends on the flow regime. For an element of the coil heat flow is

$$d\dot{Q} = U(x(L))(T_a - T_r)\frac{dA_a}{dL}dL = GA_c \Delta h_{lv} dx \qquad (12)$$

This equation cannot bee integrated analytically because of an implicit and non-linear variation of x with L. Also the superheating part of the tube has to bee included in the total heat balance of the coil.

Air side heat transfer was calculated using correlations of Kim, Yun and Webb (2003). The fin efficiency was evaluated using the model of Schmidt modified by Perrotin and Clodic (2003). The total heat transfer resistance from inside to air is:

$$R_{tot} = \frac{1}{A_i}\left(\frac{1}{\overline{h}_r} + \frac{1}{h_{as}\ A_a / A_i} + A_i R_{tube}\right) \tag{13}$$

$\overline{h}_r$ is the average heat transfer coefficient of carbon dioxide (inside), h_{as} effective outside (average) heat transfer coefficient including the influence of the fin efficiency. Formula (14) was applied for each section. Air temperatures leaving the sections were caculated using heat balance of the air side:

$$d\dot{Q} = d\dot{m}_a c_{pa}(T_{a,in} - T_{a,out}) \tag{14}$$

The capacity of the sections was calculated assuming a cross-flow configuration (both flows unmixed) and for the effectiviness the formula (Mills 1999)

$$\varepsilon = \frac{T_{a1} - T_{a2}}{T_{a1} - T_{r1}} = 1 - \exp\left\{\frac{N_{tu}^{0.22}}{R_C}\left[\exp(-R_C N_{tu}^{0.78}) - 1\right]\right\} \tag{15}$$

N_{tu} is the number of transfer units (= $UA/\dot{m}_a c_{pa}$), R_C the capacity ratio. T_{a1} and T_{a2} are the inlet and outlet air temperatures, T_{r1} the inlet refrigerant temperature. The coil was divided into sections and for each section the balance equations were set up.

The area required for the superheating section A_{sh} is obtained from the energy balance

$$\dot{m}_r \Delta h_{sh} = A_{sh} \overline{\Delta T}_{sh} / R_{sh} \tag{16}$$

The equations of heat transfer and pressure drop have to be solved iteratively because of interdependence of the variables. The tool applied was EES (Engineering Equation Solver).

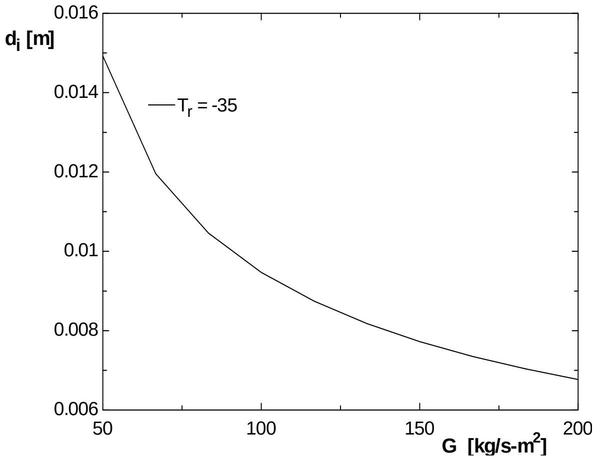
5. MAIN RESULTS

The influence of two adverse effects of heat transfer coefficient and pressure drop can be combined by defining the apparent over-all heat transfer coefficient (U-value):

$$U_{app} = \frac{\dot{Q}_{tot}}{A_a(T_{a1} - T_{r2})} \tag{17}$$

For comparisons, some parametrs have to be fixed. When dimensioning a coil natural preset parametrs are the temperature approach $T_{a2} - T_{r1}$ and the superheating $T_{a1} - T_v$. Figure 4 shows schematically these temperatures assuming the pure counter flow profiles for the simplicity. The other fixed parameters are the air frontal velocity, the inlet temperature of air, the inlet quality of the refrigerant and the geometric dimensions of the coil (see appendix). Two temperature levels were studied: the air inlet –30 and –10 ^oC. In the former case, carbon dioxide enters the evaporator tubes at the quality of 0,1 and in the last case at 0,4 (a direct expansion system).

When the external dimensions and the tube pithces are contstant, the following changes with increasing mass velocity G can be noticed: decreasing tube diameter (an example in Figure 5), increasing air-side heat transfer area, decreasing air velocity, decreasing surface efficiency,



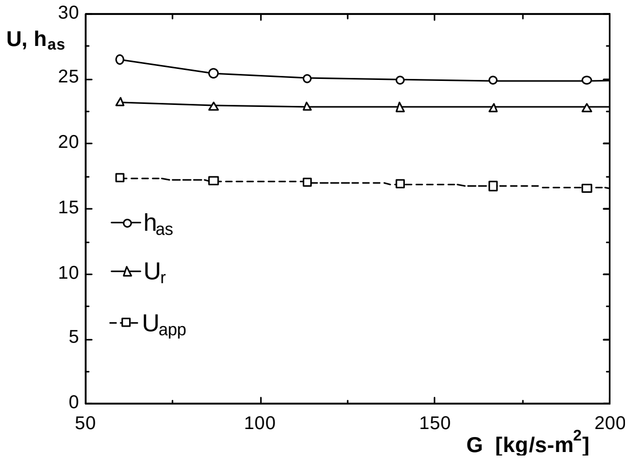
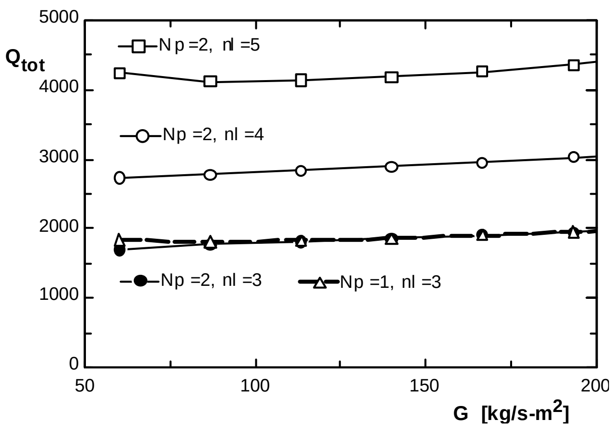


Figure 5. Tube diameter versus mass velocity for saturation temperature -35 ^{o}C.

Figure 6. Effective air-side heat transfer coefficient, local mean U-value and apparent U-value. T_{a1} = -30 ^{o}C, n_l = 4, N_p = 2.



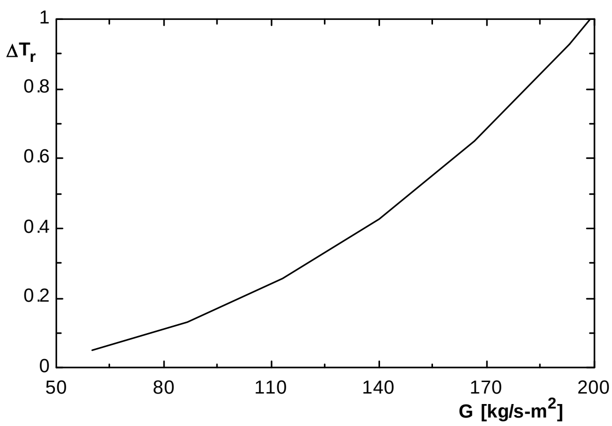
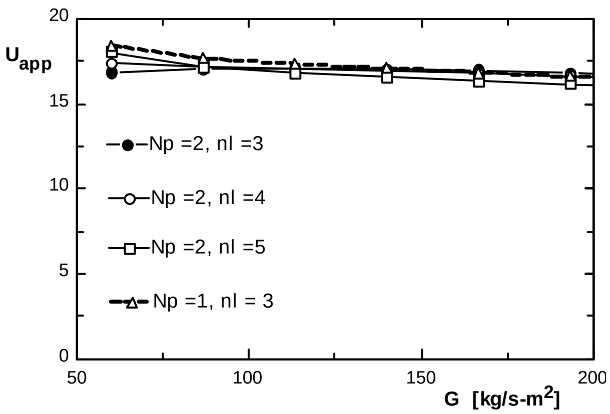


Figure 7. Total capacity of the coil. T_{a1} = -30 ^{o}C.

Figure 8. Change of saturation temperature. T_{a1} = -30 ^{o}C. n_l = 4.



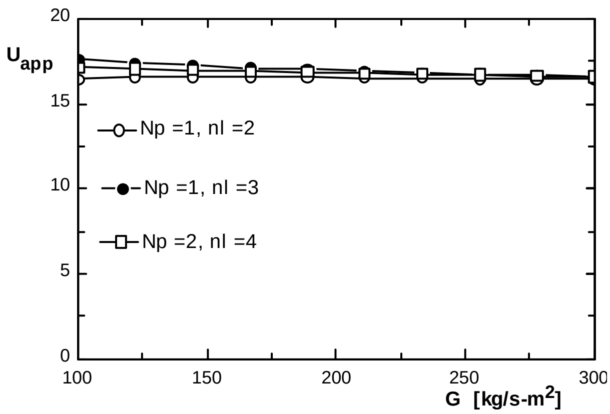


Figure 9. Apparent U-value for the case T_{a1} = -30 ^{o}C. n_l is number of tube rows, N_p number of circuits.

Figure 10. Apparent U-value for the case T_{a1} = -10 ^{o}C.

decreasing effective air-side heat transfer coefficient (Figure 6), increasing refrigerant-side heat transfer coefficient (Figure 3), decreasing local U-value (Figure 6). As the sum of these partly opposite effects, a weak minimum of the total capacity of the coil can occur, Figure 7. The apparent U-value decreases monotonously with increasing *G* as shown in Figure 9 and Figure 10.

6. DISCUSSION AND CONCLUSIONS

The sample results show that a decrease of mass velocity gives monotonously lower apparent U-value. A weak minimum of the total capacity of the coil can be found with a suitable combination of the parameters. The variations are within 10% in range studied. At higher values of *G*, the evaporation pressure is lower which means a lower COP of the compressor. However, this change of the saturation temperature is smaller than 1 K which corresponds less than 40 kPa in the pressure. The evaporation temperature can be rised if a smaller temperature approach $T_{a2} - T_{r1}$ is used but simultaneously, the capacity of the coil decreases rapidly. By far, in direct expansion coils the superheating determines often the possible upper level of evaporation temperature. Thus for an effective utilization of the coil, a small superheating is important. As known, this is not easy to achieve in the practice when injection of a mixture of gas and liquid into several tube circuits is required. However, carbon dioxide as the refrigerant the number of branches can be much smaller and thus the problems of flow distribution are easier to manage.

To find an optimum economic configuration of the coil, the price of the coil and the price of electricity should be considered. When examining the coil en block, also the power consumption of the fan due to the pressure loss should be taken into the count.

NOMENCLATURE

D	diameter [m]
g	gravitational acceleration [m^2/s]
G	mass velocity [kg/sm^2]
h	heat transfer coefficient [W/Km^2]
k	thermal conductivity [W/Km]
L	length, m
$\dot{m}$	mass flow rate [kg/s]
M	molar mass [kg/kmol]
p_r	reduced pressure
q	heat flux [W/m^2]
R	thermal resistance [K/W]
	flow capacity ratio
x	vapor fraction
σ	surface tension [N/m]
μ	dynamic viscosity [kg/sm]
ξ	friction factor
σ	surface tension [N/m]
μ	dynamic viscosity [kg/sm]
ξ	friction factor
θ	perimeter angle [rad]

Subscripts

a	air
as	air side, effective
c	flow area
dry	vapor part
i	refrigerant side
l	liquid
Lv	vaporization
o	air side
s	superheating
v	vapor
wet	liquid part

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APPENDIX. The parameters of the coil studied

Air pressure	100 kPa	Fin thickness	0,3 mm
Height of the coil	0,48 m	Superheating	5 K/0 K
Longitudinal tube pitch	35 mm	Temperature approach	5 K
Transverse tube pitch	40 mm	Frontal air velocity	2,5 m/s
Fin pitch	6 mm	Frost thickness	0