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A Numerical Study of an Evaporator Coil for a Refrigeration Secondary Loop with CO_2

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Abstract

Environmental concerns have dictated the identification and the development of substitutes for the current synthetic refrigerants replacement. Carbon dioxide is a natural alternative suitable for medium and low temperatures. Heat transfer between the refrigerant and the refrigerated medium occurs in the evaporator, whose design must be adapted to take advantage of the favorable properties of carbon dioxide. This paper presents a mathematical model to study in detail a counter-current type air/CO_2 coil with corrugated fins. The model applies the conservation equations for mass, momentum and enthalpy on small volume elements. The solution procedure of these equations is based on the Forward Marching Technique. The computed results allow for detailed information on the air and refrigerant states in the coil. Local results can be integrated to obtain global values for engineering purposes. It is possible to qualify tube rows by determining their individual capacities in terms of their location in the coil. Experiments have been performed on an experimental facility in CTEC-V Laboratories. Results from this installation as well as those collected from the open literature have been used to validate the developed model. Agreement between the predictions and experimental values is satisfactory. Calculations performed on a typical 3.5 kW capacity refrigerated cabinet coil at -24 ˚C, highlight some advantages of CO_2:

- small CO_2 side pressure drop and small corresponding temperature variation
- nearly uniform temperature and humidity distributions on the air side.
- possibility of using long circuits for simple coil geometry and impacting positively on refrigerant distribution and the cost of the coils.

INTRODUCTION

As a natural fluid offering environmental advantages, carbon dioxide has thermo physical and transport properties that places it among the preferred alternatives for synthetic refrigerants such as HCFCs and HFCs. In order to fully exploit the advantages related to the use of carbon dioxide, several researchers over the last few years have devoted time and effort to clarify the issue. Among the pioneering works were those of Lorentzen (1993 and 1995). Since then research was performed on systems, equipment and heat transfer processes involving this refrigerant. In the case of supermarket applications, particular interest is directed towards the use of secondary loops refrigeration. Such an approach based on CO_2 with phase change is a potentially interesting alternative to dry expansion or sensible heat transfer, since pressures are within the range of traditional refrigeration applications at moderate temperatures. In such cases, the combined CO_2 latent heat due to phase change with its excellent thermophysical properties, significantly reduce the total charge and refrigerant flow rates, as it is clearly outlined by Pearson (1993) and Inlow et al. (1996) respectively, just to mention these two papers. However in order to be competitive with existing direct evaporation systems, secondary loops must be designed such that inefficiencies related to extra heat exchangers and temperature gradients are minimized. Other developpements continue to be proposed. For example Sawalha et al. (2005) have studied the case of a refrigeration system that simulated real operating conditions in a supermarket using CO_2 as refrigerant. Dispenza et al. (2005) performed a feasibility study on a hypermarket in Sicily (south of Italy), in which they recommended two different configurations: the first configuration uses carbon dioxide in a three-stage supercritical cycle; the second configuration combines the use of propane and CO_2 in cascade. Both options appear to be suitable for supermarket refrigeration, based on the TEWI criterion which is comparable for both cases. However the technology is considered to be still at an R-D stage. Modeling work on evaporator coils such as is presented here has also received a great deal of attention from researchers because coils are the main heat transfer component in a refrigeration set up at the cold end. Moreover most of this work is also applicable for refrigerants other than CO_2. In this respect, Liang et al. (1999) have used refrigerant R134A in their evaporator coil. However the computation approach is also applicable to CO_2. Model validation was based on global performance parameters such as refrigerating capacity. More recently, H. Jiang et al. (2006) have modeled their heat exchanger making provision for the flexibility in the treatment of several refrigerant circuits. These models use a tube by tube approach in their computations. The mathematical model proposed in this paper was developed to study in detail a counter-current type air/CO_2 coil with corrugated fins. The model applies the conservation equations for mass, momentum and enthalpy to a coil divided according to an incremental approach into small volume elements. In the process appropriate CO_2 correlations for heat transfer and pressure drop are used. The solution procedure of these equations is based on the Forward Marching Technique. The results obtained by means of these computations allow for detailed information on the fluids states in the coil. On the air side, temperature, relative humidity and external pressure drop are among the important parameters that can be obtained. On the refrigerant side flowing in the tube, saturation temperature, and internal pressure drop and vapor quality along the coil can be tracked. Local results can be integrated to obtain global values for engineering purposes. It is possible to qualify tube rows by determining their individual capacities as a function of their location in the coil. As an example, calculations have been performed on an evaporator coil typically used in refrigerated cabinets with three doors. The coil capacity is approximately 3.5 kW and the air inlet temperature is -24 ˚C. The results obtained on the basis of this case have been used to highlight the main advantages of CO_2 as a refrigerant.

THEORETICAL MODEL

Important parameters including refrigerant properties, geometry, configuration layouts and other operating conditions significantly influence operation and performance. It is the purpose of this contribution to deal with the development of a mathematical model for design and analysis that will be used to harness these parameters for a rational design of air-CO_2 wavy fin evaporator coils which are standard in refrigeration. Refrigerant CO_2 flows inside horizontal tubes while air flows across the coil and over the fins, on the outside of the tubes. Flow of refrigerant through its progression in the coil can be distributed in sub cooled, two-phase and superheated regions along the tube and tube lengths are allocated according to the desired performance and operating conditions, as detailed in Aidoun et al. (2004). The resulting computer programs are designed to tackle staggered tube situations, accounting for co-current or counter-current flows and updating for pressure drop related temperature and refrigerant properties variations along the flow. In the present work, this tool is first used to perform simulations of the effect of a staggered tube coil arrangement in counter flow with phase change, on performance. It is then applied to examine the parametric distributions under different operating conditions. Performance is expressed in terms of coil capacity, internal and external pressure drops, temperature distributions (air, wall and refrigerant).

The model relies on the resolution of the conservation equations for mass, momentum and enthalpy, with the main assumptions stated below:

- steady state flow conditions;
- one-dimensional flow for CO_2 inside tubes and air across the coil;
- negligible thermal losses to the environment;
- uniform temperature and air flow before each pass;
- no condensation and no frost accumulation.

Conservation equations are applied on an element of control volume as represented in Fig.1-b.

$$Q_i = \dot{m}_a \left(Ha_i - Ha_{i+1}\right) = \dot{m}_{co2}\left(Hco_{i+1} - Hco_i\right) \quad (1)$$

$$Q_i = Ug_i . \pi . D_{ou} . \Delta L_i . \Delta T_{lmtd} \quad (2)$$

$$Ug_i = \left[\frac{A_{ou}}{hin_i A_{in}} + \frac{1}{\eta_f \ \ hou_i} + \frac{A_{ou} Ln\left(\frac{D_{ou}}{D_{in}}\right)}{2\pi\lambda_T . \Delta L_i} \right]^{-1} \quad (3)$$

ΔT_{lmtd} is the logarithmic temperature difference and Ug_i is the overall heat transfer coefficient.

Air side pressure losses and the heat transfer coefficient have been respectively calculated by using the Colburn and friction coefficients available from Wang et al. (2002). The Colburn equations for N rows are:

$$J = 0.0646\, Re_{Dc}^{\ J_1} . GF_1 . (\tan\theta)^{-0.692} Nrang^{-0.737} \quad (4)$$

Re_{Dc} is the Reynolds number based on the collar diameter.

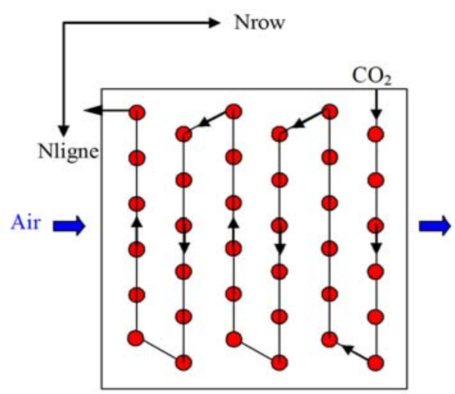

Fig.1-a: Coil configuration

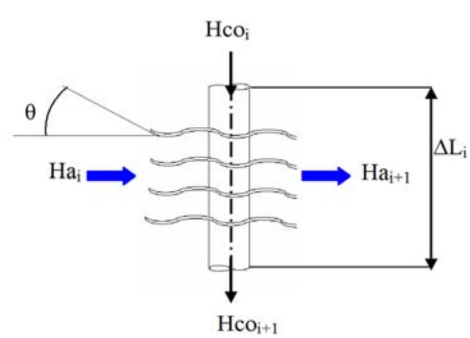

Fig. 1-b: Volume element

$$GF1 = \left(\frac{D_c}{D_h}\right)^{J_2} . \left(\frac{F_s}{P_t}\right)^{-1.03} \left(\frac{P_l}{D_c}\right)^{0.432} \quad (5)$$

$$J_1 = -0.0545 - 0.0538 \tan\theta - 0.302 Nrang^{-0.24} . GF_2 . \tan\theta^{-0.256} \quad (6)$$

$$GF_2 = \left(\frac{F_s}{P_l}\right)^{-1.3} \left(\frac{P_l}{P_t}\right)^{0.379} \left(\frac{P_l}{D_h}\right)^{-1.35} \quad (7)$$

$$J_2 = -1.29 Nrang^{-0.166-1.08\tan\theta} . GF_3 \quad (8)$$

$$GF_3 = \left(\frac{P_l}{P_t}\right)^{1.77-9.43\tan\theta} \left(\frac{D_c}{D_h}\right)^{0.229-1.43\tan\theta} \left(\frac{F_s}{P_t}\right)^{-0.174\ln(0.5Nrang)} \quad (9)$$

θ is the deviation angle of the fin corrugation relative to the horizontal, D_c and D_h are respectively the external and hydraulic diameters. F_s is the fin spacing. P_l and P_t are respectively the longitudinal and transverse spacings of the tubes. Re_{Dc} is the Reynolds number relative to the external diameter. The friction factor is defined as:

$$f = 0.228 \, Re_{Dc}^{f_1} . (\tan\theta)^{f2} \left(\frac{F_s}{P_l}\right)^{f3} \left(\frac{P_l}{D_c}\right)^{f4} \left(\frac{D_c}{D_h}\right)^{0.383} \left(\frac{P_l}{P_t}\right)^{-0.247} \quad (10)$$

Where:

$$f_1 = -0.141 \, .GF_4 . (\tan\theta)^{-0.472} \, N_{rang}{}^{-0.049+0.237\tan\theta} \quad (11)$$

And the corresponding factors are:

$$GF_4 = \left(\frac{F_s}{P_l}\right)^{0.0512} \left(\frac{P_l}{P_t}\right)^{0.35} \left(\frac{P_t}{D_h}\right)^{0.449\tan\theta} \quad (12)$$

$$f_2 = -0.562 (\ln(Re_{Dc}))^{-0.0923} N_{rang}{}^{0.013} \quad (13)$$

$$f_3 = 0.302 \, Re_{Dc}^{0.03} . \left(\frac{P_l}{P_t}\right)^{0.026} \quad (14)$$

$$f_4 = -0.306 + 3.63 \tan\theta \quad (15)$$

The heat transfer coefficient calculation for CO_2 with phase change has been calculated by using the correlations developed

by Bennet-Chen and modified by Hwang et al. (1997). These are based on the superposition principle which consists in assuming that the heat exchange coefficient is equal to the sum of two exchange coefficients respectively representing nucleate boiling heat transfer and convective heat transfer.

$$h = h_{nb} + h_{bc} \quad (16)$$

Correlations based on the homogeneous model have been selected for the calculation of linear pressure losses, Rohsenow et al. (1998).

$$\Delta P_{\Delta L i} = \left[\frac{f}{2D_{in}} . \Delta L_i . v_{tp(i+1)} + (v_{tp(i+1)} - v_{tp(i)})\right] G^2 \quad (17)$$

Pressure losses in bends have been determined by using the correlation due to Geary (1975).

$$\Delta P_s = f_s \frac{L_{ben}}{D_{in}} \frac{G^2 . X^2}{2.\rho_g} \quad (18)$$

$$f_s = \frac{803.52 \times 10^{-6} . Re_g^{0.5}}{\exp(0.215 \, C_d / D_{in}) X^{1.25}} \quad (19)$$

The properties of air above the dew point temperature have been determined from the conventional psychometric relations, summarized in the ASHRAE Fundamentals Handbook, ASHRAE (1993).

$$Cp_m = Cp_a + \omega Cp_v \quad (20)$$

In the vicinity of the wall, the following correlation is used:

$$C_{pw} = \frac{H_w - H_{ev}}{T_w - T_{ev}}$$

(21)

H_w and H_{ev} are air enthalpies based on the wall and on the CO_2 evaporation temperatures respectively.
When the wall temperature is below the dew point and assuming that the air is saturated at the wall temperature condition, use of the following equations is made:

$$Cp_m = Cp_a + \omega_m . Cp_v \quad (22)$$

and

$$\omega_m = \frac{\omega + \omega_{sat}}{2} \tag{23}$$

SOLUTION PROCEDURE AND VALIDATION

The modular structure of this model is such that it can handle different coil arrangements. The tubes are tagged in terms of the coordinate parameters Nligne (vertical position) and Nrow, (horizontal position). Once the configuration, the tube length and other geometric characteristics are selected, the program sets the overall pattern with the number of passes over which computations will be performed. The solution procedure adopted is based on an incremental approach which consists in resolving the equations applied on each volume element in the inverse direction of the CO_2 flow path. Iterations are performed up to the point where inlet conditions are attained. During the computations the return bends are automatically sensed and the cumulative count of the total pressure losses is performed. Thermodynamic properties of CO_2 and air are respectively determined by using NIST-REFPROP subroutines, NIST (1998) and conventional psychrometric relations, ASHRAE (1993). The experimental set up of the CTEC-Varennes has been used for validation. It is represented in Fig.2. Evaporating carbon dioxide is the working fluid in the loop of interest (L1), which includes a CO_2-air coil with aluminium wavy fins and copper tubes. The loop is well instrumented for the purposes of heat and mass transfer balances and fluid flow. It is located in a closed, well insulated room with two compartments corresponding to the inlet and the outlet of the coil: air flows from one compartment to the other through a duct enclosing the coil. Air circulation is maintained by a blower. The configuration employed in this particular instance is similar to that represented on Fig.1-a, arranged in one circuit. Other characteristic details of the installation, measurements uncertainties and operational procedure can be found in Ouzzane et al. (to appear in 2008). The computed results have then been compared with those obtained by experiments performed on the above-mentioned test bench.

Table 1 Comparison of results from measurements and computations

Case		Capacity (W)		X%	ΔP *(kPa)*	Te *(°C)*	
		Air	CO_2	CO_2	CO_2	Air	CO_2
1	Model	4438.3	4530.6	58.0	195.2	-23.0	-29.9
	Expt.	4083.3	-	51.4	171.6	-22.4	-29.6
2	Model	4615.9	4673.3	97.0	122.1	-23.8	-29.7
	Expt.	4556.7	-	90.0	111.4	-23.7	-29.2
3	Model	1557.2	1550.0	36.5	38.6	-22.8	-24.8
	Expt.	1495.9	-	35.5	44.4	-22.7	-24.9
4	Model	1871.1	1798.4	20.0	107.7	-24.9	-28.0
	Expt.	1640.0	-	18.3	97.0	-24.5	-27.8

These are shown in Table 1, in terms of capacity, quality, internal pressure drop and temperatures on the air side and the refrigerant side respectively. The evaporator coil experimented upon has the following characteristics :

- frontal section of 0.61m×0.32m for air flow and a depth of 0.22m.
- the tubes are finned with corrugated aluminium fins in the number of 157 fins per meter.
- the coil is 8 rows deep and 10 rows high.

The operating conditions are:

- air inlet temperatures between -21˚C and -10˚C.

For CO_2 temperatures vary between -27˚C and -23˚C.

- air velocities in the range of 1m/s to 3 m/s
- CO_2 mass flow rates in the range of 0.002 kg/s to 0.100 kg/s.

The heat balance that links the air side and the refrigerant side is very good. The agreement between the model predictions and the experimental data is satisfactory. In the worst case the predictions are within 12 % of the experimental values. The temperatures on the air side and the refrigerant side are particularly well predicted. In any case the predictions are all within the uncertainty range of the used correlations as stated by their authors.

RESULTS

The prediction results obtained by using the developed calculation tool refer to a practical case selected for this study and are based on an existing evaporator coil commonly used in three-door freezing cabinets. It is of a staggered tube configuration and counter-current flow, as depicted in Fig. 1-a. Refrigerant CO_2 in phase change vaporizes while flowing inside tubes of 8.7 mm diameter, 1.88 m length per pass and 90 m circuit total length. On the air side, corrugated aluminium fins of rectangular shape were set at 197 fins per meter.

Air enters at the evaporator front at -24˚C; refrigerant CO_2 enters at a saturation temperature of -30˚C and a quality of X=0 from the rear of the coil. When CO_2 reaches the end of the circuit (at the coil front) the quality becomes X=1 (complete evaporation). Average air temperature distributions for each tube pass are represented in Fig.3. Temperature along each tube row is uniform and at the tube exit, temperature is constant and equal to -27.4 ˚C. It is interesting to note in this case that besides this uniformity of temperature behind each tube row, the air temperature drop across tube rows, starting with a relatively small average value of 0.5 ^oC for the first two rows, decreases gradually down to approximately 0.2 ^oC towards the last rows. Due to the thermodynamic properties of carbon dioxide, particularly its high saturation pressures for the temperatures levels used in refrigeration, the temperature decrease accompanying pressure losses inside the tube is modest. Overall, this fact coupled with the controlling effect of heat transfer by the air side due to its high thermal resistance, results in a balanced air temperature distribution across the coil. This gradual decrease with a small temperature gradient favours a good refrigeration load repartition over the

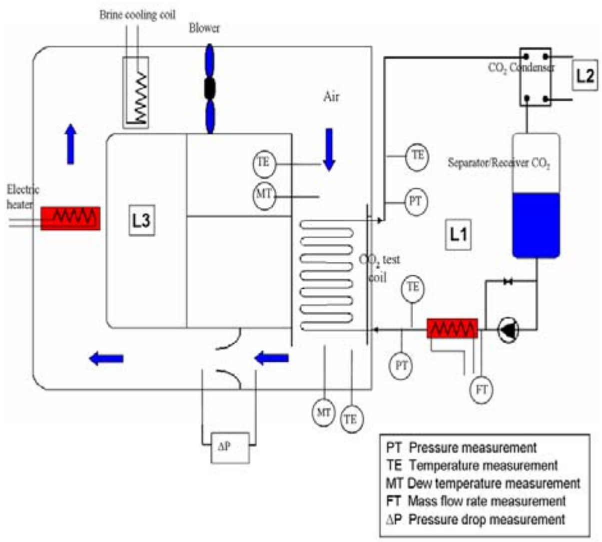

Fig.2: Experimental set-up schematics

entire heat transfer surface, for given refrigeration conditions. In case of frost deposition it is expected to be more uniform and to occur over a longer period of time than for an ordinary synthetic refrigerant. The distribution of the cumulative pressure losses as a function of the tube length is presented in Fig.4. It is worth noting that 80% of the total losses occur in the first half of the circuit, corresponding to increasing qualities, vapour velocities and better heat transfer. Over the total length of 90 meters, including 80 return bends (180^o bends), the total pressure drop losses for CO_2 amounts only to 80 kPa. This fact highlights one the many advantages resulting from the favourable thermo physical and transport properties of CO_2. This pressure drop results in the diminution of the saturation temperature by 1.8 ˚C as can be seen in Fig. 5.

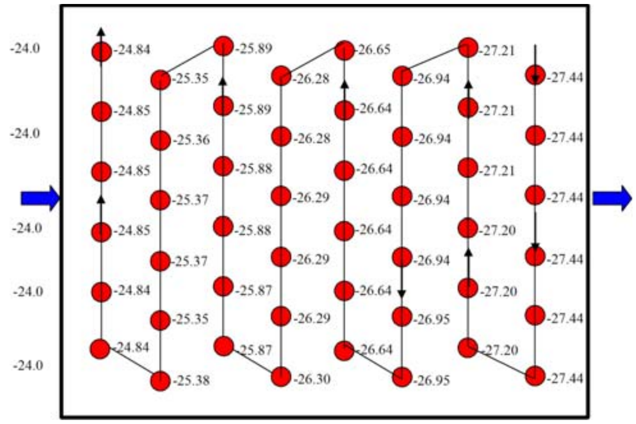

Fig. 3 : Distribution de la température moyenne de l’air

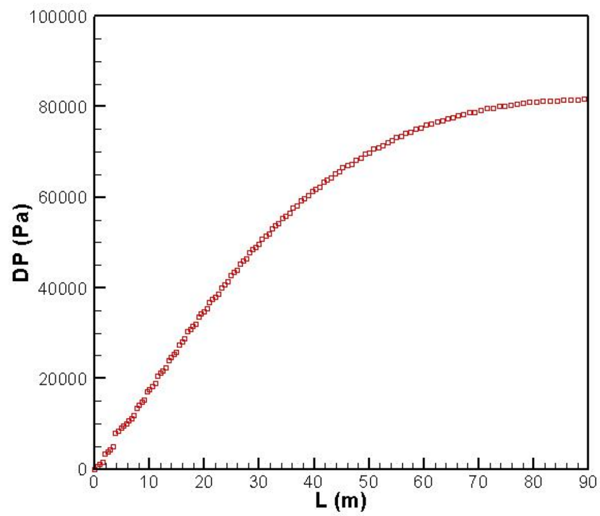

Fig.4: Cumulative pressure losses of CO_2

For the sake of comparison with the synthetic refrigerants ordinarily used in refrigeration systems, this temperature reduction value is rapidly attained in relatively short tube lengths. In order to maintain a reasonably constant temperature across an evaporator this temperature drop must be small (less than 1^{o}C say). This condition needs several small length circuits with synthetic refrigerants, while far fewer circuits may be required with carbon dioxide. The distribution of the air temperature shown in Fig.5 and represented by blank squares has eight flat plateaus, corresponding to the number of rows deep. Since these plateaus are essentially horizontal, the assumption of an average temperature before each pass is justified. In order to outline the advantages offered by the thermo physical properties of CO_2, a comparative study with

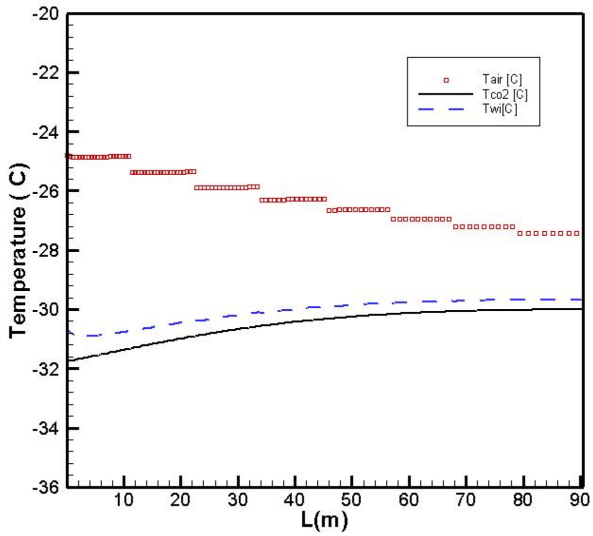

Fig.5: Temperature distribution of CO_2, air and wall

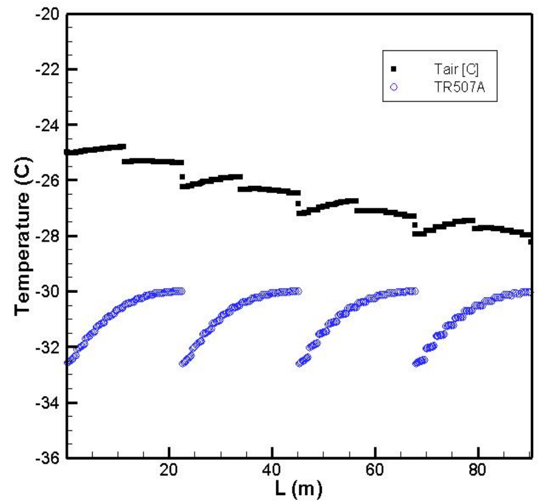

Fig. 7: Temperature distribution of air and R507A

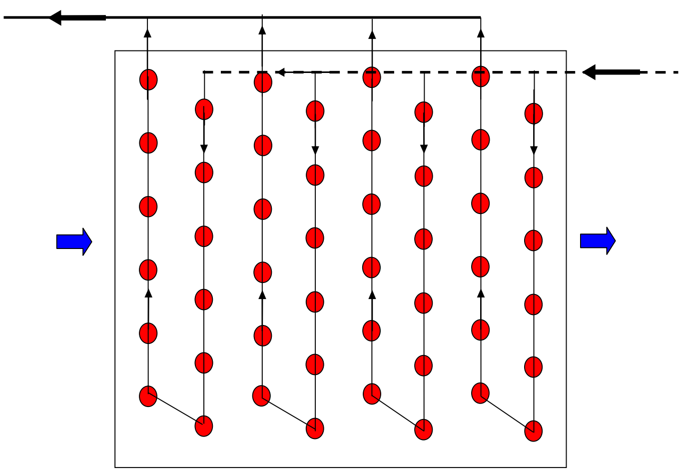

Fig. 6: Four circuits' configuration for the study of R507A

Refrigerant R507 has been carried out. Several iterative tests have been performed in order to obtain a reasonable temperature drop. It was found out to be impossible to use less than four circuits. Therefore the configuration selected was that of four circuits as shown in Fig.6. In such a case the circuits are well balanced and the temperature drop in the saturation temperature is of the order of 2.4 ^{o}C in each circuit. Air temperature distribution also presents eight plateaus. In contrast with CO_2 however, these plateaus are not horizontal as Fig. 7 shows, a fact that explains the lack of uniformity in the air temperature before each pass.

CONCLUSION

A theoretical model allowing simulation of an air/CO_2 evaporator coil with corrugated aluminium fins has been developed and validated. The structure of the resulting tool is such that details on the temperature distribution, vapour quality, CO_2 pressure and other parameters along the circuit can be obtained. The results from these computations allow for detailed information on the fluids states in the coil. On the air side, temperature, relative humidity and external pressure drop are among the important parameters that can be obtained but only representative results have been presented in this paper for the sake of conciseness. On the refrigerant side, saturation temperature, internal pressure drop and vapor quality along the coil can be tracked. Local results can be integrated to obtain global values for engineering purposes. It is possible to qualify tube rows by determining their individual capacities as a function of their location in the coil. Experiments have been performed on an experimental facility of the CTEC-V Research Centre. Results from this installation as well as those collected from the open literature have been used to validate the developed model. Agreement between the model predictions and the experimental values is satisfactory.

As an example, calculations have been performed on an evaporator coil typically used in refrigerated cabinets with three doors. The coil capacity is approximately 3.5 kW and the air inlet temperature is -24 ˚C. Comparison with refrigerant R507A shows that in general several circuits are necessary, while for CO_2 a single circuit configuration was sufficient to obtain the same or even smaller temperature drop, which was acceptable in refrigeration practice. More specifically, over a tube length of 90 meters, the temperature drop was ΔT_{ev}= 1.8 C. By means of this model some advantages offered by the thermo physical properties of this refrigerant have been demonstrated, among others:

- Air temperature after every tube pass is nearly constant.
- Pressure drop for CO_2 is very small in comparison to other refrigerants ordinarily used in current systems and results in correspondingly small drop in saturation temperature. This has a positive impact on the temperature distribution within the freezing cabinet.
- It is possible to use longer circuits with CO_2 (in comparison to other refrigerants), therefore reducing their number for a given capacity. This greatly simplifies the coil geometry, impacting positively on refrigerant distribution and the cost of the coils.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A	area, m^2
Cd	centre to center distance of bend
Cp	specific heat, J/kg ^oC
D	diameter, m
Fs	fin spacing, m
f	friction factor
G	mass flux, kg/m^2 ^oC
GF	geometry factor
h	convective heat transfer coefficient, kW/m^2 ^oC
H	enthalpy, kJ/kg
J	colburn heat transfer factor
L	length, m
$\dot{m}$	mass flow rate, kg/s
N_f	number of fins
N_{ligne}	height coordinate
N_{row}	row number
P_l	longitudinal tube pitch
P_t	transversal tube pitch
P	pressure, Pa
Q	capacity, kW
Re	Reynolds number
T	temperature, ^oC
U_g	overall heat transfer coefficient, W/m^2 ^oC
v	specific volume, m^3/kg
x	quality
ΔL	element of length, m
ΔP	pressure drop, Pa
ΔT	temperature drop, ^oC
λ	thermal conductivity of tube, kW/m ^oC
η	efficiency
ω	humidity ratio, kg vapor/kg total moist air
ρ	density, kg/m^3
θ	corrugated angle, degrees

Subscripts

a	air
b	bare
bc	convective boiling
ben	bend
c	collar
co	CO_2
Dc	collar diameter
ev	evaporation
f	fins
g	gas
h	hydraulic
in	inner
l	liquid
m	mean
nb	nucleate boiling
ou	outer
o	overall
s	singular
sat	saturation
T	tube
t	transversal
tp	two phase
v	vapor
w	wall
wi	internal wall

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