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MODELLING FIN-AND-TUBE GAS-COOLER FOR TRANSCRITICAL CARBON DIOXIDE CYCLES

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ABSTRACT

In CO_2 transcritical refrigeration cycles fin-and-tube coils are still interesting as possible gas cooling devices, due to their lower costs when compared with recent aluminium minichannels heat exchangers. In spite of the very high working pressures, a commercial coil with four ranks of 3/8" copper tube and louvered fins has been studied as a gas-cooler in a test rig built at University of Padova for testing CO_2 equipment operating with air as a secondary fluid. The test rig consists of two closed loop air circuits acting as heat sink and heat source for gas-cooler and evaporator respectively. The tested refrigerating circuit consists of two tube-and-fin heat exchangers as the gas-cooler and the evaporator, a back-pressure valve as the throttling device, a double-stage compound compressor equipped with an oil separator and an intercooler. A full set of thermocouples, pressure transducers and flow-meters allows measurement and recording of all the main parameters of the CO_2 cycle, enabling to perform heat balance both air-side and refrigerant-side.
Tests were focused on two different gas-coolers, with continuous and cut fins, and on two different circuit arrangements. Test on each heat exchanger were run at three different inlet conditions, both for CO_2 and air. A simulation model was developed for this type of heat exchanger and the most outstanding models proposed for the carbon dioxide supercritical cooling heat transfer coefficients were implemented and compared in the code.
The model results are compared with the experimental data for the finned coil: emphasis is given to the effect of heat conduction through fins between adjacent tubes ranks on the system efficiency.

1. INTRODUCTION

According to the necessity of decreasing greenhouse effect, new fluids need to be investigated as refrigerants. Carbon dioxide seems to be promising, because of its environmental friendliness, and its excellent thermodynamic and transport properties, like high specific heat, high thermal conductivity and low viscosity. The "traditional" finned coil heat exchangers can be still considered as an opportunity in CO_2 transcritical cycles. The gas cooling process needs to be investigated taking into account two points of view: the great variation of thermophysical properties and the great decrease of temperature occurring along the heat exchanger. Very poor evidence is given in the open literature to the study of finned coils gas-cooler with "macro" tubes (i.e. internal diameter of the pipe larger than 3 mm). Recently, Hwang et al (2005) proposed an experimental and numerical study of a three rows finned coils gas-cooler with 7.9 mm outside diameter. In the present paper a set of tests was conducted and experimental results were compared with numerical predictions obtained trough a finite volume simulation software.
Although simulation results gained by our software has been found to be in agreement with experimental results for a large number of refrigerants and test conditions, a systematic deviation was seen using CO_2. Therefore, some correlations to predict CO_2 heat transfer coefficient was tested, but the heat conduction along the heat exchanger, which has not been considered by the software, was deemed to be the most important cause of the results disagreement. This kind of phenomenon is not considered by the simulation software at the moment, and it has never been pointed out as a significant problem by any paper published in literature. Furthermore, it seems to be relevant just to finned coil heat exchangers, since it was shown to be not important in microchannel heat exchangers (Asinari et al, 2004).

2. TEST RIG DESCRIPTION

The CO_2 circuit (Fig. 1) carries out a double compression with gas intercooling between the two compression stages and single throttling, and it is equipped with an internal heat exchanger. The compressor is a two-stage semi-hermetic reciprocating unit running at 1450 rpm (50 Hz). The nominal volumetric flow rate of the low pressure stage (one cylinder) is 3.0 m^3/h, while of the second stage is 1.74 m^3/h (vol. ratio 1.7). The lubricant is a PAG oil 46 ISO grade.

The compressor is instrumented with a thermocouple inserted in the high pressure suction chamber. A pressure tap is drilled in the same chamber. Comparison of the compressor performance before and after inserting the thermocouple and the pressure tap was made without finding any measurable difference. The compressor power input is recorded with an electronic transducer (with an accuracy ±0.5 % of the reading value).

The intercooler (IC) heat flow is rejected to a water-loop. The cooling water inlet temperature and flow rate are controlled through an auxiliary circuit (temperature stability ±0.05 °C). The IC is a copper tube-in-tube heat exchanger with the CO_2 flowing inside three pipes (ID 4 mm, OD 6 mm) fed in parallel and inserted into a 20 mm ID (22 mm OD) copper tube. The water flows inside the outer tube in counter-current to the CO_2. The IC was designed for 1 °C temperature approach between the two fluids. The water volumetric flow rate is measured with an electromagnetic flow (accuracy ±0.2 % of the reading value).

The internal heat exchanger used is a copper-steel 10 m length tube-in-tube heat exchanger with the high pressure fluid flowing inside three pipes (ID 6 mm, OD 8 mm) fed in parallel and inserted into a 21 mm ID (26.7 mm OD) steel tube. The low pressure CO_2 flows inside the outer tube in counter-current to the high pressure CO_2. The internal heat exchanger was designed for 2 °C temperature approach between the two fluids. The throttling device used in the tests is a back-pressure valve: this allows the operator to set and keep constant the gas-cooler (GC) outlet pressure.

The CO_2 circuit is equipped with an oil separator. A special accumulator with sight glasses is inserted for visual inspection of the oil returning to the compressor crankcase. A metering valve is also installed to control the lubricant level in the compressor and to avoid any hot gas by-pass through the oil drainage.

Air is the external fluid for both the gas-cooler and the evaporator. For the tests here reported finned coils with round copper tubes (8.22 mm ID and 9.52 mm OD) are employed. The aluminium fins are louvered with 2.1 mm fin spacing. The face area for both exchangers is 500x500 mm. Air temperature at the inlet of each heat exchanger is controlled by way of two separated closed-loop wind tunnels. The air ducts lay-out was designed through CFD simulation targeted at getting very uniform velocity and temperature distribution over the entire heat exchanger face area. The centrifugal fans are equipped with inverter to adjust the volumetric flow rate that is measured with ISA 1932, AISI 316L nozzles with pressure taps integrated in the nozzle body. The nozzles were installed according to EN-ISO Standard 5167. Air pressure drop through the nozzle is measured by a strain gauge pressure transducer, with an accuracy of ±1 Pa while air temperature downstream the nozzles is recorded with a thermocouple. In this way, according to EN-ISO Standard 5167, the estimated accuracy in volumetric flow rate measurement is ± 0.8 % of reading.

The air in the closed loop serving the GC is re-conditioned by a cooling finned coil fed with water kept at an inlet constant temperature in an auxiliary circuit equipped with a commercial chiller and control system. The air inlet temperature to the GC is then kept at the desired value by electrical heaters, PID controlled. Another PID feedback loop is used to control the air dry bulb temperature at the evaporator inlet by compensating the CO_2 system cooling power with electrical heating. The air temperature stability thus achieved in steady-state conditions was ±0.05 °C. Nine thermocouples are placed, evenly distributed, just before the GC inlet. Since the high temperature change of the CO_2 through the gas cooling process leads to a scattered distribution of the air temperature over the outlet section (depending on the CO_2 pipe arrangement in the finned coil), an air mixer is placed after the GC and the air temperature is measured again by nine thermocouples placed after the air mixer. The same thermocouple arrangement was adopted for the evaporator.

CO_2 temperatures are measured with thermocouples placed inside mixing chambers at the inlet and outlet of each heat exchanger. A thermocouple is also placed at the IC water inlet, whereas the water temperature change across the IC is measured with a four junctions type T thermopile.

All the thermocouples for air and CO_2 are T type: the complete measuring chain, including multimeter and the ±0.01 °C reference ice point was calibrated against a ±0.02 °C accuracy Pt100 thermometer. Thus a ±0.05 °C accuracy is estimated for all the temperature measurements.

R744 pressures are recorded with strain-gauge transducers at the outlet of each heat exchanger and inside the compressor 2nd-stage (high pressure) suction chamber. The accuracy is ±1 kPa for evaporator and compressor chamber pressures and ±2 kPa for gas-cooler pressures, according to the calibration report from the manufacturer. Differential pressure transducers were used for pressure drop recording across GC, evaporator and IC (accuracy ±400 Pa).

CO_2 mass flow rate is measured by a Coriolis mass flow meter placed upstream of the throttling valve. The claimed accuracy is ±0.1 % of reading. IC water volumetric flow rate was measured by an electromagnetic meter (accuracy ±0.2 % of reading).

All the measurements are real time acquired and elaborated. The particular lay-out of the air tunnels and the accuracy of the instruments led to a heat balance error for each component and for the complete system lower than ±1 %.

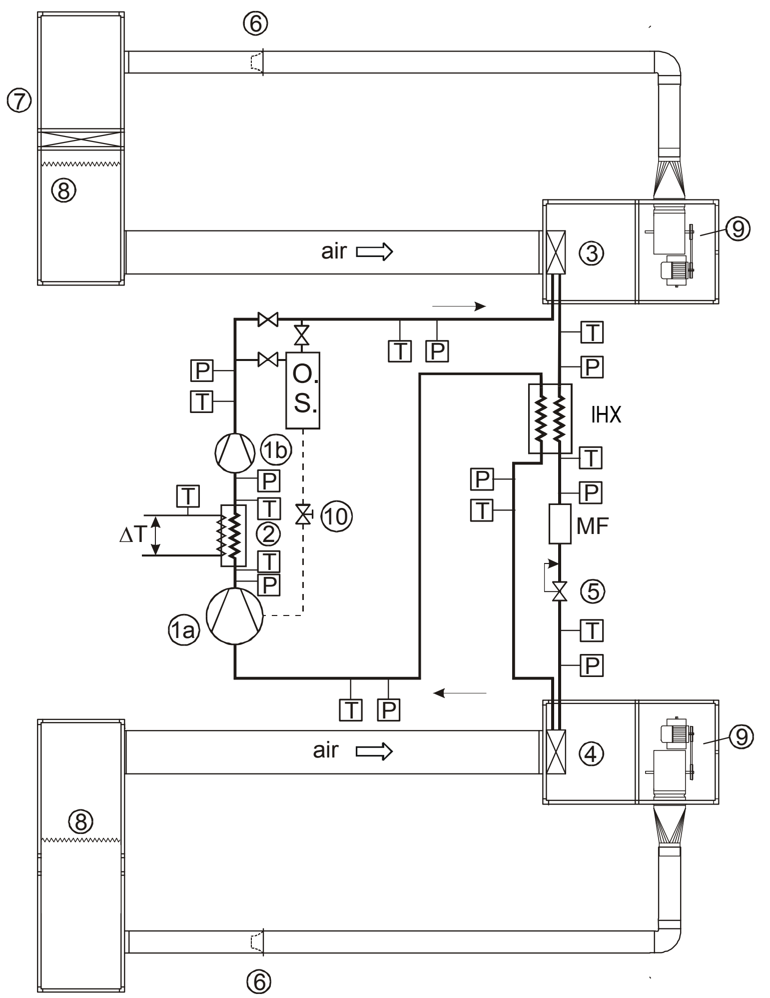

Figure 1: Test rig lay-out. 1a,b two-stage compressor, 2 intercooler, 3 gas-cooler, 4 evaporator, 5 back-pressure valve, 6 nozzle, 7 auxiliary finned coil, 8 electrical heater, 9 centrifugal fan, MF Coriolis mass flow meter, OS oil separator, IHX internal heat exchanger, P pressure transducer, T thermocouple, ΔT thermopile

3. GAS-COOLERS DESCRIPTION

Three different devices were tested , dubbed as “A”, “B” and “C” throughout the paper. The main characteristics of the three finned coils are listed in Table 1. “A” and “B”, as depicted in Fig. 2, are identical but for the fins. Referring to Fig. 4, in coil “B”, the fins insisting on adjacent tube rows are separated along line DE. This expedient contributes in reducing heat conduction between adjacent tube rows. Coil “C” presents the same fin arrangement as for coil “B”, with different tubes lay-out, as in Fig. 3. A simple scheme of the fin type used is given in Fig. 4.

Table 1: Specifications of the heat exchangers

Heat exchanger		Gas-cooler	Row pitch	[mm]	19
Type		Finned coil	Geometry		staggered tubes
Dimension (W x H)	[mm]	500 x 500	Fin pitch	[mm]	2.1
Number of circuits		2	Fin thickness	[mm]	0.1
Number of tubes per row		20	Fin type		louver aluminium (Fig. 4)
Number of rows		4			
Refrigerant path		vertical, counter-current	Tube outside diameter	[mm]	9.52
			Tube thickness	[mm]	0.65
Tube pitch	[mm]	25	Tube type		Smooth

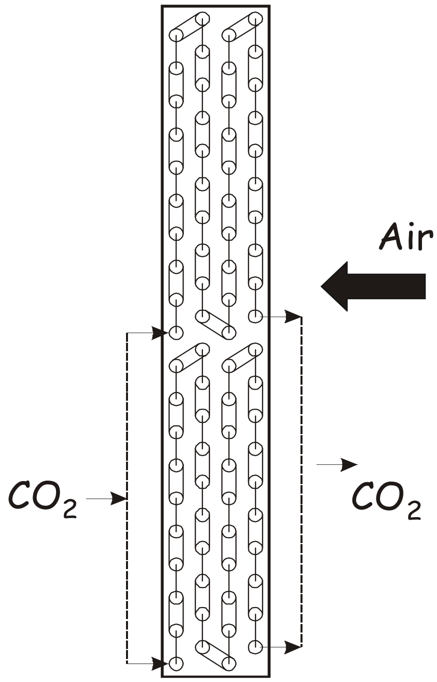

Figure 2: coils A and B

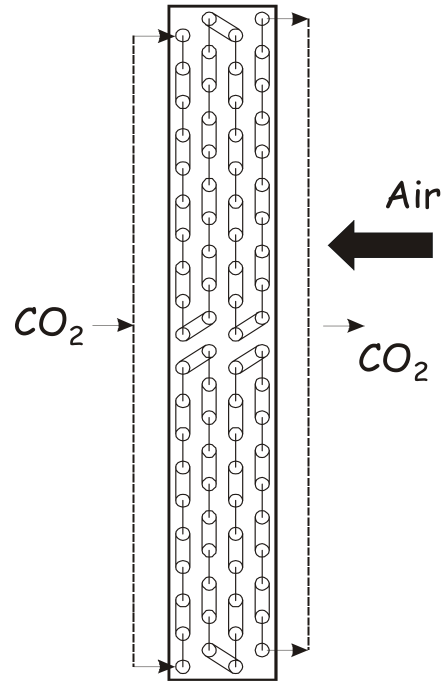

Figure 3: coil C

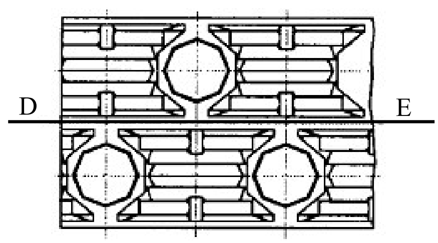

Figure 4: Simple scheme of the investigated fin type

4. THE SIMULATION MODEL FOR THE FINNED COIL

The heat transfer analysis has been applied to the case of a finned coil gas-cooler, by developing a simulation model. Since the flow configuration of a finned coil does not conform to the elementary and well known parallel or counter flow patterns, the heat transfer area is subdivided into a three-dimensional array of cells that conforms to the true flow pattern of the air and of the refrigerant streams. Each tube is divided into cells and the total volume, in the form of a parallelepiped, is subdivided into individual nodes, each including a small stretch of tube and the related fins.

The numerical approach to the definition of the circuits is accomplished by means of two arrays, one for the refrigerant [PR(N)] and one for the air [PA(N)]. Each element of the PR vector (preceding refrigerant) indicates the index of node preceding the N-node, along the refrigerant flow.

As described for the refrigerant, the PA(N) value (preceding air) represents the air-side node preceding the N-node according to the air flow direction.

This simple approach allows a finned coil to be solved regardless of how complicated is the lay-out of the tube circuits. For the thermal performance determination, the refrigerant flow rate in each circuit is calculated by an iterative procedure, being known the air flow rate, its inlet temperature as well as the refrigerant pressure at the inlet of the gas-cooler. The well known method of the secant is employed.

The refrigerant and air-side heat transfer coefficients are also input parameters in the program. The model can deal with carbon dioxide in transcritical conditions, in this case the heat transfer coefficient is calculated from equations described in literature, such as Gnielinski's equation (1976), Pitla et al. equation (2002) and also Dang and Hihara's equation (2004).

The friction factor is calculated by the Colebrook-White's equation, while pressure drops in the curves are locally accounted for by adding an equivalent tube length equal to 50 times the tube diameter.
The air side heat transfer coefficient, including surface efficiency was evaluated experimentally, by feeding the coils with "hot" water (inlet temperature 28.3 °C). An overall heat transfer coefficient U was evaluated from Equation (1), being Q the measured heat flux, A_0 the external heat transfer area and ΔT_{lm} the logarithmic mean temperature difference. U is defined by Equation (2):

$$Q=UA_0\Delta T_{lm} \qquad (1)$$

$$\frac{1}{U}=\frac{A_0}{A_i}\frac{1}{h_i}+\frac{A_0}{A_i}\frac{r_i}{k}\ln(r_0/r_i)+\frac{1}{\eta_0}\frac{1}{h_0} \qquad (2)$$

(r radius, h heat transfer coefficient, η surface efficiency, k tube thermal conductivity, i: internal, o: external). h_i was evaluated with the Gnielinski equation, so it was possible to obtain $\eta_0 h_0$ from equation (2).
It's worth noting that, the investigated finned coil promotes four passes R744 side, with air flowing in cross flow. According to Kays and London (1998) the efficiency of this configuration is almost identical to pure counterflow configuration.
The thermodynamic and thermophysical properties of the refrigerants considered in the calculations are obtained from the database REFPROP 7.0 by NIST.
A more detailed description of the simulation model is provided in (Casson et al., 2002).

5. EXPERIMENTAL RESULTS – COMPARISON BETWEEN NUMERICAL AND EXPERIMENTAL RESULTS

More than 100 tests were run with coil "A", with gas cooling pressures from 7.8 MPa to 12.0 MPa. The gas-cooler "A" performance was compared with "B" and "C" in several operating conditions. In this paper three different working pressure are analysed from 7.9 MPa up to 9.1 MPa. The mentioned pressure range were considered particularly meaningful because in this working conditions R744 Prandtl number is recognized to undergo the most "dramatic" variation. The test conditions to evaluate the gas-cooler performance are listed in Table 2.

Table 2: Test conditions

Test	CO_2 inlet pressure [MPa]	CO_2 inlet Temperature [°C]	CO_2 mass flow rate [kg/h]	Air inlet temperature [°C]	Air inlet velocity [m/s]
1	7.9	87.0	169.0	20.3	1.6
2	8.6	97.6	167.1	21.5	1.6
3	9.1	107.8	166.4	23.0	1.6

Three sets of tests were conducted at the mentioned conditions, and the experimental results gained are summarized in the next table.

Table 3: Experimental results

Finned coil	Test	CO_2 outlet temperature [°C]	Air outlet temperature [°C]	Approach [°C]	Heat flux (CO_2 side) [kW]
A	1	33.0	39.5	12.6	9.0
	2	33.0	42.7	11.5	10.2
	3	33.1	45.1	10.0	10.8
B	1	32.1	40.6	11.8	9.5
	2	31.0	43.7	9.6	10.6
	3	30.9	46.0	7.9	11.2
C	1	32.2	40.8	11.8	9.5
	2	31.0	43.8	9.5	10.6
	3	30.9	46.2	7.9	11.2

In order to analyse the fluid temperature distribution in the heat exchangers, a set of thermocouples was arranged along one circuit of the finned coil, as it is illustrated on Fig. 5. The thermocouples were attached to the outside surface of fifteen U-bends, using a silicone based heat transfer compound to improve the thermal contact to the surface. Then every single thermocouple and its corresponding U-bend were suitably insulated towards the outside. Referring to the test number 3, for the gas-cooler named as A, the temperature profiles are shown in Fig. 5.

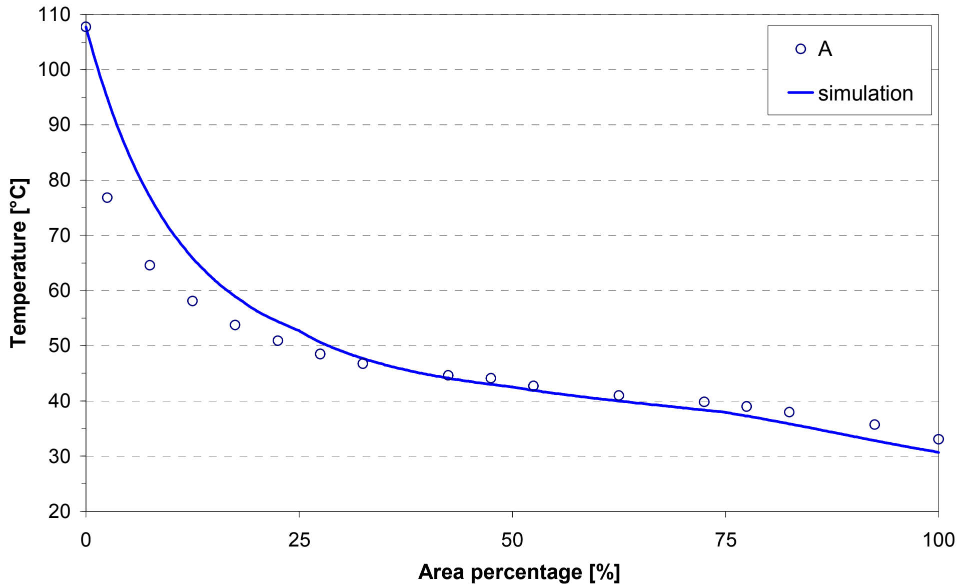

Figure 5: Temperature profiles at 9.1 MPa (test conditions 3, finned coil "A")

The "simulation" line shown in the figure relates to the temperature profile obtained by the numerical simulation software. In each test a significant difference between the predicted temperatures and measured values with coil A was observed. Although the approach and the temperature values are different between tests at different refrigerant and air inlet conditions, in the first row (0 % - 25 %) of the heat exchanger the measured temperature values are always inferior to the calculated ones, while they are superior in the last three rows (25 % - 100 %).
Several CO_2 heat transfer coefficient correlations were tested, but without any significant change in prediction of values. All CO_2 specific correlations usually refer to Gnielinski's correlation that was chosen as the reference to be used in the simulation software to evaluate the temperature profile and the heat flux. The other correlations were selected considering the different approach to the heat transfer coefficient evaluation. Pitlas's correlation makes use of an average between Nusselt numbers referred to bulk and wall temperatures, while Dang and Hihara's correlation uses a film temperature to evaluate CO_2 properties and therefore the heat transfer coefficient.
Different correlations don't give rise to different results because the heat transfer resistance is chiefly concentrated on the air side in finned coils, therefore the CO_2 side heat transfer coefficient doesn't seem to be as important as the air side coefficient. In the light of what it was seen, the reason of the difference between predicted and measured values must lie in a phenomenon characterizing gas cooling process and not modelled by the numerical code.

Table 4: Simulation results by using three different correlations

Correlation	**Gnielinski**		**Pitla et al.**		**Dang and Hihara**	
Test	**CO_2 outlet temperature [°C]**	**Heat flux [kW]**	**CO_2 outlet temperature [°C]**	**Heat flux [kW]**	**CO_2 outlet temperature [°C]**	**Heat flux [kW]**
1	32.2	9.5	32.2	9.5	32.3	9.5
2	31.0	10.6	31.0	10.6	31.2	10.6
3	30.7	11.3	30.7	11.3	30.9	11.2

Unlike condensation or evaporation processes, in which the fluid temperature is almost uniform for a large part of the heat exchanger, the gas cooling process involves a significant temperature glide along the heat exchanger. In particular, the CO_2 temperature gradient is higher in the first 20 % (following R744 flow direction) of the heat transfer area. Therefore it seems to be reasonable to consider the thermal conduction from first high temperature tubes to low temperature tubes throughout continuous fins as the most important factor of performance penalization and temperature profile disagreement for finned coils. As a consequence of the conduction heat transfer from hotter tubes to colder ones, the efficiency of the heat exchanger deviates from the overall counterflow behaviour.
To confirm this hypothesis, the continuous fins were cut to eliminate the thermal conduction from a row to the next one. In this way a separated row finned coil was tested (named as “B” in Table 3). A 3.7 % to 5.6 % heat flux improvement was gained, as it can be seen in Table 3, while Fig. 6 shows a better agreement of “B” temperature profile compared with “A” profile. In the first row of the coil (0 % - 25 %) the temperature values increased while in the second part of the heat exchanger the temperatures decreased approaching the simulation profile. That is consistent with a decrease in thermal conduction.

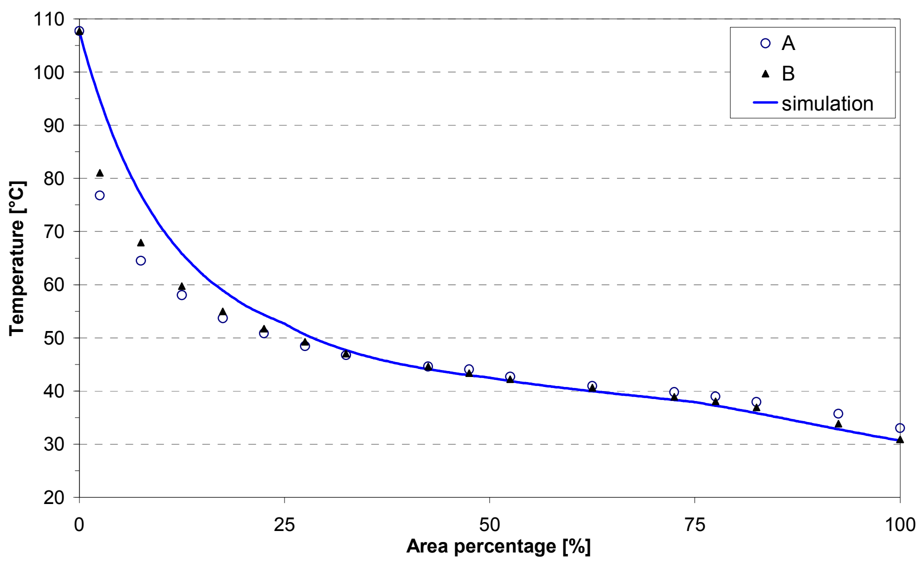

Figure 6: Improvement of the temperature profile obtained by using the finned coil B (test conditions 3)

This improvement can be clearly translated in terms of COP, since a low value of the CO_2 temperature at its outlet increases the cooling capacity. Using the “cut-configuration”, a 5.7 % to 6.6 % increase of coefficient of performance can be obtained, referring to a conventional cycle with the same operating conditions. The considered reference cycle is the simplest kind of a transcritical CO_2 system using single stage compression, without the presence of a Suction Line Heat Exchanger. The compressor is simulated with constant isentropic efficiency equal to 0.6. Isobaric processes except for compression and throttling were considered, with -10 °C evaporation temperature and 5 °C vapour superheating. Furthermore, a higher COP improvement can be reached by an optimization of gas-cooler pressure.
The configuration represented in Fig. 3 was chosen in a second set of tests to limit the thermal conduction between the two circuits along the vertical direction. Basically, from a geometric point of view, the two circuits were symmetrically fed, to reduce the thermal conduction in the horizontal middle section of the heat exchanger.
Referring to the comparison between “C” configuration and “B” one, Fig. 7 shows a slightly better approach just in the first tube row (high temperature zone), while in the other three tube rows measured temperatures are almost the same, and the same is also for the overall thermal performance.
This experimental results are an evidence that the thermal bridge made by fins contributes to penalize heat transfer because of thermal conduction between different tube rows, not between the tubes in the same row.

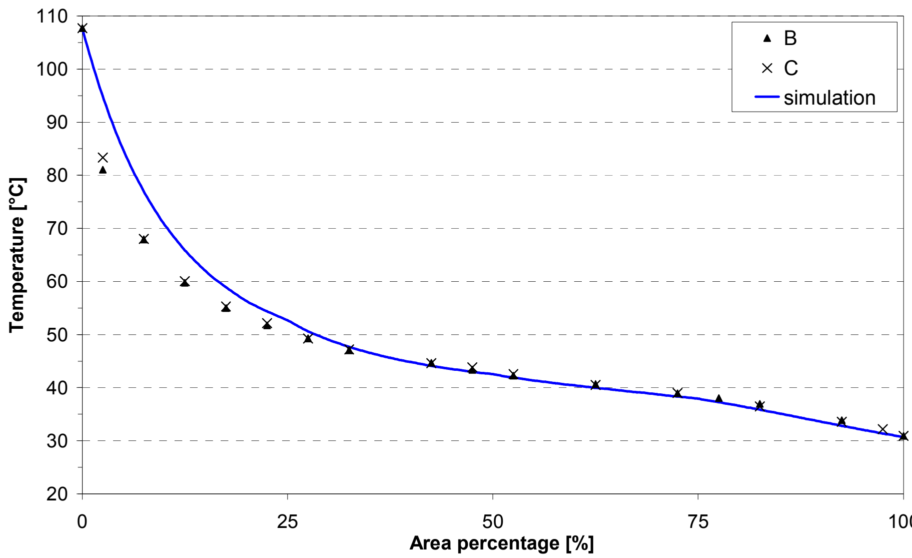

Figure 7: Comparison of the temperature profiles of the two different circuit arrangements in the same finned coil

CONCLUSIONS

Experimental tests and numerical analysis on two identical finned coils, one with continuous fins, the other with separated fins per each tube row, indicates that "internal" heat conduction through the fins is an important factor in finned coils CO_2 gas-coolers. This aspect is strictly linked to the high temperature variation of carbon dioxide during the gas cooling process in a transcritical refrigeration cycle. The icrease in temperature approach between CO_2 outlet temperature and air inlet temperature was found to improve the efficiency of the refrigerating cycle. This aspect offers a quite promising technological opportunity, since it is easy to build a coil with fins separated between the rows.

Future work will be aimed at developing a new more detailed code that keeps into account also the heat conduction through the fins: this will be a fundamental tool for the optimized design of finned coil gas-coolers.

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