NEW GENERATION OF AIR COOLED HEAT EXCHANGERS FOR CO_2 APPLICATIONS

S. FILIPPINI, U. MERLO
LU-VE Group, Via Caduti della Liberazione 53, Uboldo, 21040, Italy
Corresponding author: stefano.filippini@luvegroup.com

ABSTRACT

In several refrigeration applications, CO2 is an excellent and environmentally-friendly solution; LU-VE has designed very efficient heat exchangers for CO2 applications and has been able to make important developments thanks to the use of its modern test ring. This plant can test the performance of CO2 finned heat exchangers, both air cooled unit coolers and gas coolers.
The new testing plant enabled the launch of a specific project for a CO2 fin-and-tube heat exchanger, with the primary aim of improving knowledge of heat exchange phenomena in evaporation, condensation and during trans-critical gas cooling. The influence of oil on internal heat exchange coefficient also enters into the scope of the research. The paper describes the testing activity, the calibration made on software that calculates product performance, and potential improvements to products.
In particular, it was possible to calibrate a specific method able to take into account the behaviour of the fluid during trans-critical cooling, properly considering all the parameters affecting real performance.

Keywords
CO2, gas coolers, heat exchangers, oil influence

1. INTRODUCTION

CO2 has been used since the middle of the 19th century as a refrigerant fluid in land-based systems and also on board ships. It offered several advantages:

- reduced cost
- good heat transfer properties
- non-toxicity in the event of leaks in machine rooms
- the condenser cooling fluid in naval applications being low-temperature seawater meant that the cycle functioned in subcritical conditions.

The introduction of synthetic refrigerants (CFC, HCFC) and the development of compressor technology in the first half of the 20th century led to the use of CO2 as a refrigerant being greatly reduced. The rediscovery of carbonic anhydride was only brought about by the environmental problems which made the use of CFCs and HCFCs in refrigeration systems less acceptable. The articles by Gustav Lorentzen in the 90s are celebrated for that reason. Numerous studies on carbonic anhydride as a secondary fluid were published subsequently, above all for commercial refrigeration systems in supermarkets.
LU-VE began to study air-cooled heat exchangers for CO2 at the beginning of the new millennium and since 2012 has been equipped with a sophisticated laboratory for the testing of gas coolers and unit coolers. This article collects the results of the experimental analyses and the calibration of the calculation model used in LU-VE for the dimensioning of gas coolers.

2. TEST PLANT

The CO2 plant was designed to be able to test condensers, gas coolers and unit coolers. The maximum operating pressure is 120 bar, while the maximum temperature is 120°C. The capacities of the plant in the standard conditions specified below are:

- Unit Coolers (SC2) : 15 kW
- Gas Coolers e Condensers (100bar, 120°C, DTmin=3°C) e (25/40/65°C) : 40 kW.

A dedicated software programme was developed in-house using LabView to monitor and acquire data. The tests were conducted making a thermal balance between the test unit and a contrast group in order to guarantee the reliability of the experimental data. This required a great deal of time for the calibration of the measurement instrumentation and in particular for the definition of dispersion inside the climatic test chamber.

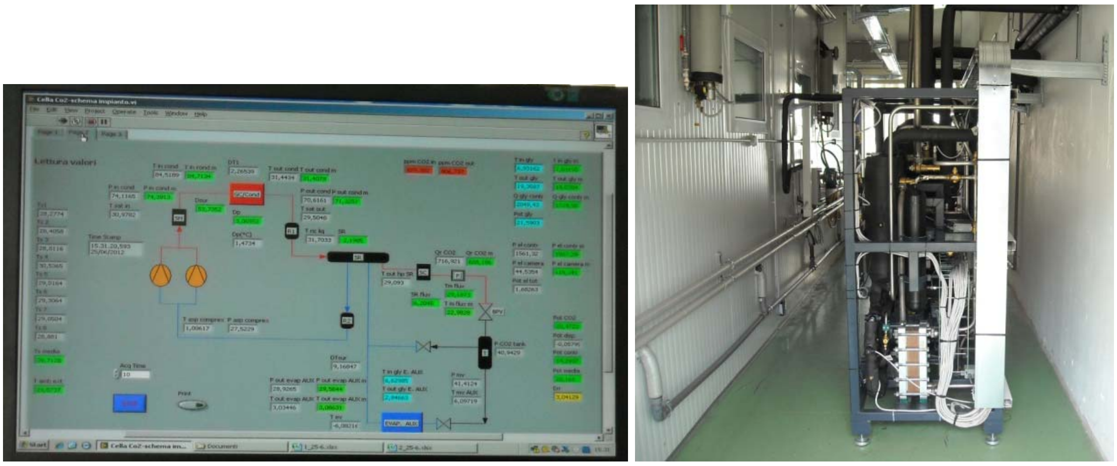

Figure 1 - Image of data acquisition codes and lateral view of the plant

2.1 Characteristics of the machines under test

Three units were tested in the laboratory, with the characteristics listed in table 1. The tube used is made of K65 alloy, containing a small quantity of iron which significantly increases mechanical resistance. The circuits were designed identically, respecting the rule of counter-current and drainage of any possible stagnant oil which, as we shall see later, is the main cause of gas cooler underperformance.
At the end of the experimental activity, a series of comparisons was made between theoretical and experimental data.

Unit	GAS COOLER 1	GAS COOLER 2	GAS COOLER 3
Fins mat.	Al	Al	Al
Tube type	Smooth	Smooth	Smooth
Tube mat.	Cu K65	Cu K65	Cu K65
Fins spacing [mm]	2,1	2,1	2,1
Fins thick. [mm]	0,12	0,12	0,12
n° tubes	40	40	40
n° rows	4	4	3
n° circuits	5	5	4
Tube diameter [mm]	7,94	7,94	7,94
Coil length [mm]	1.215	1.215	1.215
Frontal area [m2]	1,215	1,215	1,215
Fan type / Rpm	8P/465	EC/807	EC/675
Air volume rate [m3/h]	3.956	6.650	6.502

Table 1 - Characteristics of the machines under test

3. THE LU-VE CALCULATION CODE

The calculation software uses Gnielinski [1] and Colburn [2] correlations modified for heat exchange with single-phase and two-phase fluids respectively. For the calculation of pressure drop, the code uses a modified Lockhart-Martinelli [3] correlation. The initial version of the software does not consider specific corrections to these correlations. The thermodynamic and transport properties of carbon dioxide are deduced from the Refprop 8.0 programme [4]. The objective of the work is to verify the validity of these correlations for the specific case of operation with CO2.

The code calculates the heat transfer of gas coolers considering counter-current flows. However, it applies corrections for cross-flows for the de-superheating parts in condensers. The coefficients of the heat transfer correlations are obtained from experimental tests conducted in the LU-VE laboratories in standard conditions. Considering the peculiarities of the operating conditions of gas coolers compared to the most common calculation conditions, some verifications of the adaptability of the code were required in these specific circumstances.

The first analysis concerned the verification of the quality of the approximation of cross-flow. To evaluate the influence of the average logarithmic transition (and DTml in particular), the alternative “X3FLOW” code developed in LU-VE was used for the gas cooler. This allows calculations in three dimensions to be made following the real form of the circuit, dividing it into finite volumes and resolving with cross-flow the N elementary transfers using the classic equations of the ε-NTU method. Despite the most rigorous calculation of the X3FLOW code (but also computationally much more severe) the analysis comparing the overall performance obtained with the LU-VE standard and the X3FLOW code did not indicate any particular calculation deviations; as a consequence it was decided not to make any modifications to the method of calculating DTml.

The second analysis evaluated the effect of steep temperature gradients between adjacent tubes, which could lead to thermal flows by conduction along the fin. To evaluate the influence of conductive heat transfer between two rows with perceivable temperature variation of the fluid in the tubes, CFD was used to carry out some simulations.

As the reference base, a flat fin was simulated with two temperature differences between the fluid flowing in the tubes:

- 70/70°C DELTA 0°C
- 110/30°C DELTA 80°C.

The same temperature differences were maintained approximately equal (76 vs. 80°C) in the louvered fin configuration.

From the results of the four CFD simulations, it can be seen that, when the temperature differences are null, conductive heat transfer near the middle zone between the rows is likewise null (adiabatic zone), while, when we increase the temperature difference of the fluids, two different types of behaviour can be noted for the two types of fin: a) for flat fins, the conductive transfer (negative from the point of view of the overall heat transfer of the fin) amounts to 37% of the total, a significant value; b) whereas in the case of louvered fins, the cuts act as interruptions to the conductive transfer, reducing the value to approximately 4% of the total, in this case a negligible amount.

Figure 2 shows, for the four simulated configurations, the trend of the fin temperature in the centre line zone of the rows, 0.1mm before and 0.1mm after (these surfaces are used for the calculation of the conductive thermal flow). In addition, the coloured maps show the fin temperature. As can be noted, the specific configuration of the louvers permits excellent heat transfer radially to the tubes (high fin efficiency), impeding damaging transfer between the zones around the tubes in adjacent rows (thermal break).

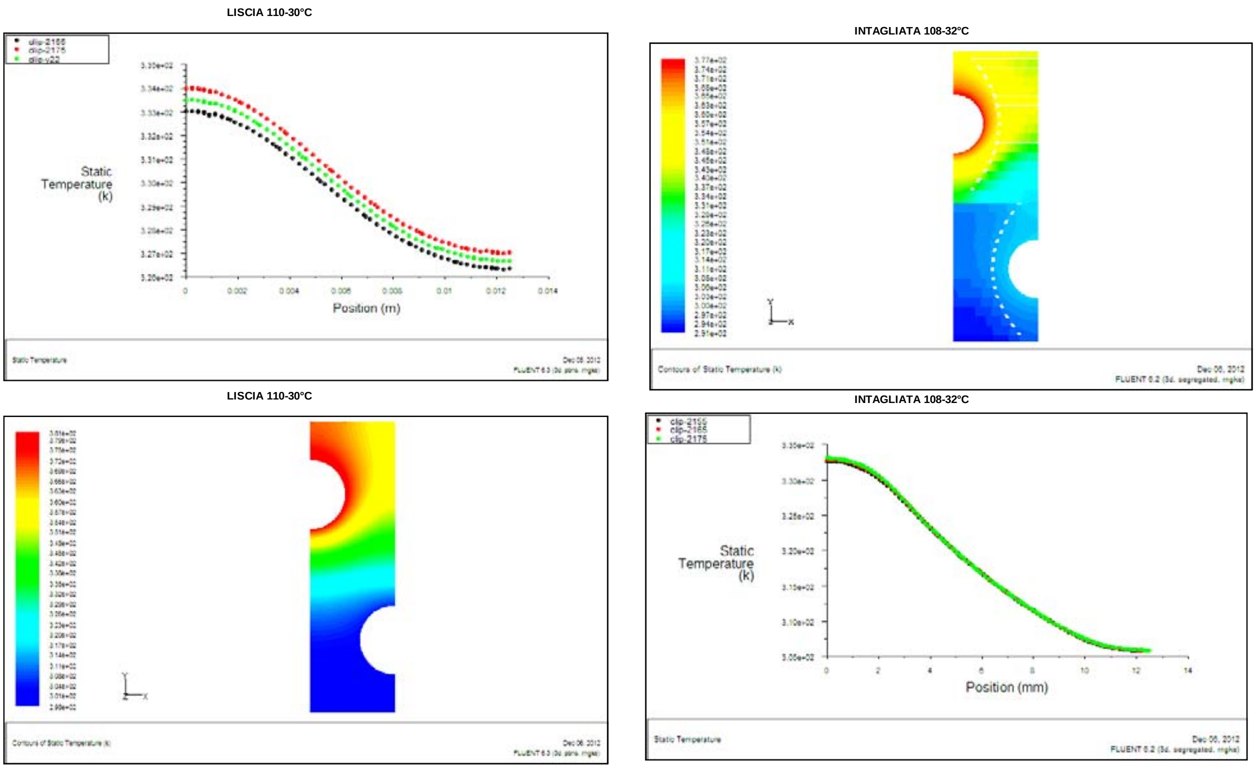

Figure 2 - CFD simulated configurations

4. EXPERIMENTAL RESULTS AND COMPARISONS WITH LU-VE CODE IN TRANS-CRITICAL CONDITIONS

The horizontal axis of Figure 3A shows the value of the error between the calculation data and the experimental data of thermal capacity, while the vertical axis shows the capacity deduced experimentally. The data are divided by type of machine and the calculation was performed at imposed capacity. An average error of capacity calculation of the order of 7% is noted.

Figure 3B shows the deviations in the calculation of pressure drop. In this case, the data are more dispersed and the biggest errors are denoted at low pressure drop. This can also be attributed to the fact that the experimental measurement of small pressure variations is much more difficult.

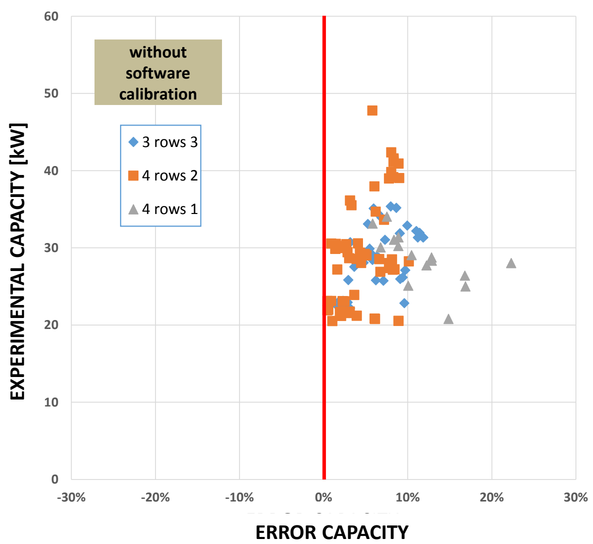

Figure 3A - Value of the error on thermal capacity

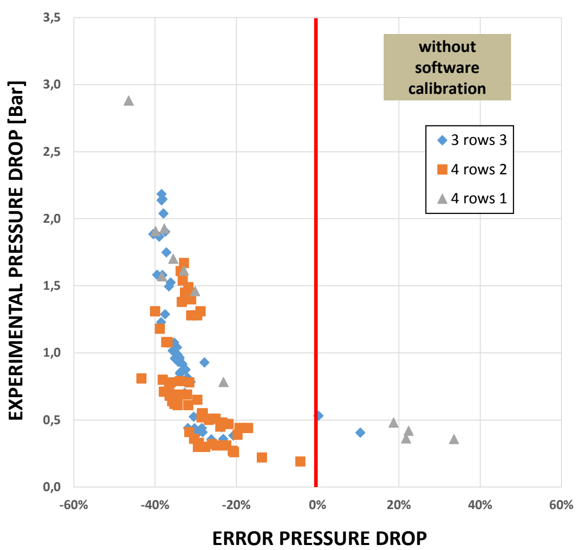

Figure 3B - Deviations in pressure drop calculation

After thorough verification of the tested units we concluded that the differences between calculations and measurements can be attributed to the presence of lubricant oil in the circuit. As has in fact been observed in the literature [5], the formation of a film of oil inside the tubes of the circuit can cause a significant reduction of the coefficient of internal exchange, even over 70%, also when there is very low oil concentration (in the region of 1%) [6]. The oil percentage was measured and a value of 1.08% was found.

An "RHTC" (multiplication factor for internal heat transfer coefficient) was therefore sought, reducing the coefficient of internal heat exchange and increasing the pressure drop, on an analysis of the experimental data. Some studies were found in the literature concerning the determination of the heat transfer coefficient in hypercritical zones, in the presence of oil in the circuit. The variables which were investigated and correlated to the penalization of the heat transfer coefficient are as follows:

RHTC =f(ID, ρv, flux, Press., xoil, TCO2, type of oil)

An analysis of the published data shows an excellent correspondence between our conclusions and those cited [7], in particular with mass concentrations of oil around 3-5%. Bibliographic research was then conducted on the available correlations, confirming as the most valid that of Gnielinski [1] (valid for 2300<Reynolds<5000000).

The corrective coefficients identified depend little on operating conditions. The graph in Figure 4A shows the trend of the theoretical/calculated capacity compared to experimental capacity after the application of the corrective coefficient. The graph displays two demarcation lines of the percentage error between the two values of ±10%. Figure 4B shows the trend of pressure drop errors as a function of their values expressed in bar. The distribution is contained within ±15%, a very substantial value.

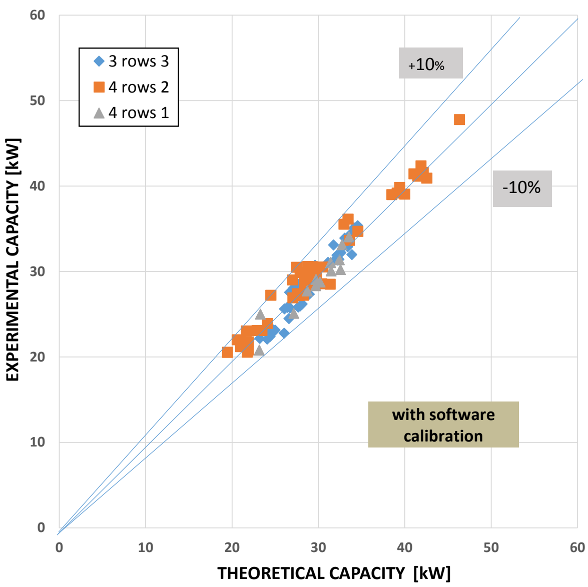

Figure 4A – Validation plot of predicted capacity

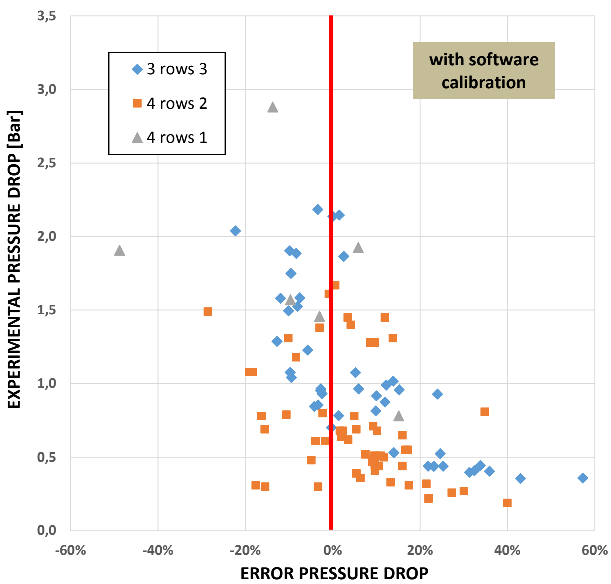

Figure 4B – Error of pressure drop

Figure 5 shows the trend of capacity error as a function of the DT between CO2 outlet and air inlet (DTmin). The errors between the theoretical and experimental capacity increase as the DTmin decreases. Therefore, considering the impact of the uncertainties of temperature measurement both air-side and CO2-side, it is not advised to go below a DTmin of 2K for the definition of exchange capacity. Furthermore, lowering the temperature difference means that the size of gas cooler has to be increased; this increase is much more than proportional.

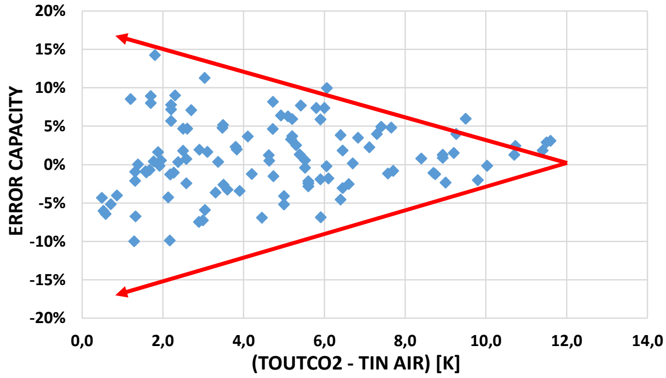

Figure 5 - Capacity deviation as function of DTmin

5. EXPERIMENTAL RESULTS AND COMPARISONS WITH LU-VE CODE IN SUBCRITICAL CONDITIONS

The same units tested in trans-critical conditions were then measured in subcritical conditions. Deviations between measured and calculated capacity were observed also in these cases. Calibration gave rise to different coefficient values for the three heat exchange zones (gas de-superheating, condensation, liquid sub-cooling). In order to make a better evaluation of the contribution of the 3 zones, different tests were carried out, increasing and reducing the de-superheating zone as well as the sub-cooling one. By doing this, the correction factors could be determined.

We observed in the condensation zone that the value of the heat exchange coefficient deduced from current condensation correlations for smooth tube matched the measured data quite well; therefore we assumed that due to the phase change and the mixing generated by it, the oil does not separate, so that a correction factor was not necessary. The performance is most penalized in the de-superheating zone, with a phenomenon similar to the one observed in transcritical operation; in addition, the correction factor is similar to the one defined for transcritical conditions.

In the liquid single-phase zone the heat exchange coefficient is less penalized than in the gas zone, for the reason that the oil is more easily mixed/dragged through. Furthermore, the liquid zone is the smallest one inside the heat exchanger and its behaviour does not greatly influence the overall performance of the test unit.

The graph in Figure 6 shows the trend of the error between the theoretical/calculated capacities compared to experimental capacity after the application of the corrective coefficients, as a function of inlet pressure to the unit. Calibration lets us align all the experimental data within a range of ±10%. The errors are independent of pressure; in particular, also in the zone near the critical point.

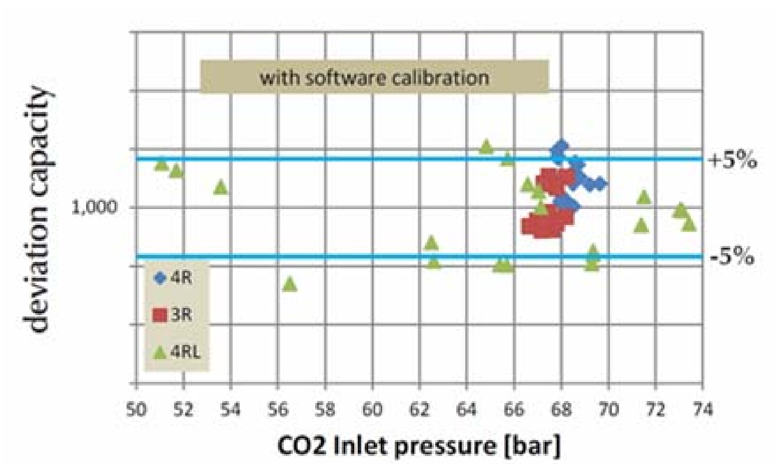

Figure 6 - Trend of capacity error as function of CO2 inlet pressure

The trend of the points for the three tested configurations are well distributed around the unit value; this means that modifying only the heat exchange coefficient and the pressure drop does not lead to a further improvement in the distribution of errors.

Figure 7 shows the trend of pressure drop errors on the internal CO2-side (calculation/experimental), as a function of the value measured, expressed in bar. The distribution is contained within ±20%, a commonly accepted value. The errors increase as the measured value decreases, due to experimental uncertainty.

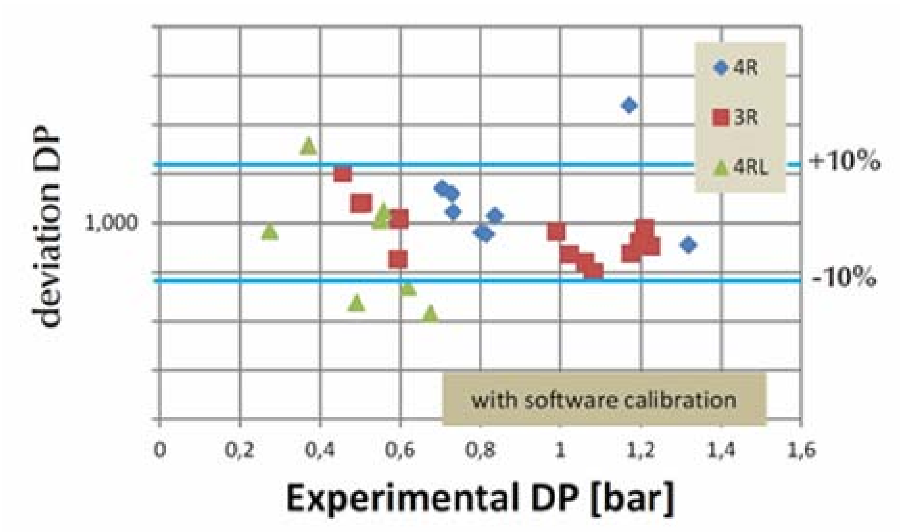

Figure 7 - Trend of pressure drop errors

One important observation could be made on the graph in Figure 8, where it can be seen that, as sub-cooling gradually decreases, the errors of the estimation of capacity become greater. This confirms that under 2K sub-cooling it is difficult to precisely define the change of state end zone; the deviation is significantly influenced by measurement uncertainties, increasing when temperature differences are quite small.

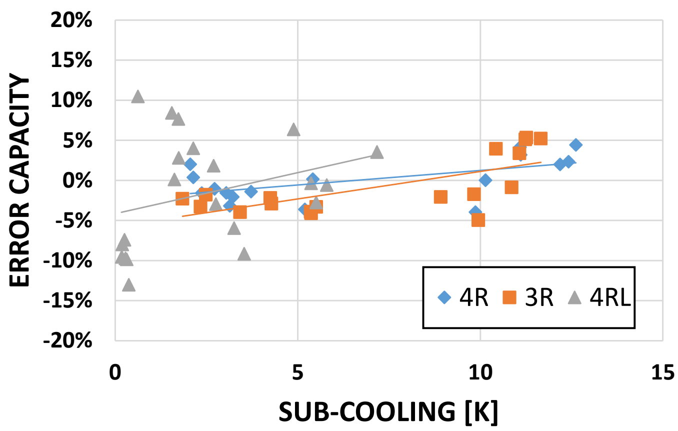

Figure 8 – Capacity error as function of sub-cooling

6. CONCLUSIONS

The battery of tests highlighted the initial deviations between the calculation software and experimental data. To correct the capacity errors, multiplicative coefficients were employed for heat transfer coefficient and pressure drop. The coefficients take into account the penalisation of internal heat exchange due to oil, which generates a film between the internal surface of the tube and the CO_2. The coefficients introduced are in line with the values reported in the literature [8]. The oil film phenomenon is evident in heat exchange with single-phase CO_2, while it is much reduced in two-phase heat exchange.

7. REFERENCES

1. *New equations for heat and mass-transfer in turbulent pipe and channel flow*, 1976, 359-368 p. (Gnielinski: Karlsruher Institut für Technologie)
2. 1933, *A method of correlating forced convection heat transfer data and a comparison with fluid friction,* 174-210 p. (Colburn: Trans Am Inst Chem Eng)
3. Chem. Eng. Prog 45.1, 1949, *Proposed correlation of data for isothermal two-phase, two-component flow in pipes*, 39-48 p. (Lockhart, Martinelli)
4. *REFPROP 8.0." NIST Standard reference database 23"*, 2007 (Lemmon, Huber, McLinden)
5. International Journal of Refrigeration, 2007, *Effect of lubricant oil on cooling heat transfer of supercritical carbon dioxide*, 724-731 p. (Dang, Iino, Fukuoka, Hihara: University of Tokyo)
6. International Refrigeration and Air Conditioning Conference, 2010, *Study on flow and heat transfer characteristics of supercritical carbon dioxide cooled with different types of lubricating oil*, Purdue, Paper 1143, (Dang, Hoshika, Hihara, Kaneko: University of Tokyo)
7. International Refrigeration and Air Conditioning Conference, 2010, *Heat Rejection from R744 near the critical point* (Kondu, Hrnjak: University of Illinois at Urbana-Champaign)
8. International Refrigeration and Air Conditioning Conference, 2012, *The effect of inner grooved tubes on the heat transfer performance of air-cooled heat exchangers for CO2 heat pump system*, Purdue, Paper 1268, (Kaji, Yoshioka, Fujino: Daikin Industries LTD.)