ID: 295

AIR COOLED HEAT EXCHANGERS FOR CO_2 REFRIGERATION CYCLES

FILIPPINI S.(*), MERLO U.(**)
(*) LU-VE Group, Via Caduti della Liberazione 53, Uboldo, 21040, Italy
stefano.filippini@luve.it
(**) LU-VE Group, Via Caduti della Liberazione 53, Uboldo, 21040, Italy
umberto.merlo@luve.it

ABSTRACT

Alternative solutions to traditional HFC plants are becoming more and more of an issue, with the objective of lowering the carbon foot-print. This is why the use of natural refrigerants in the refrigeration industry has shown a significant increase in recent years. CO2 seems to be an excellent and environmental friendly solution as it is non toxic and non flammable which are interesting advantages in comparison with other natural refrigerants such as hydrocarbons and ammonia. The real challenge is to design plant with an efficiency level equal to or higher than current HFC plants.

The paper describes the main parameters affecting CO2 cycle performance, focusing on the contribution that can be made by the efficient design of ventilated air heat exchangers..
CO2 properties are quite different from current HFC properties and pose important problems for designers because of the high pressure operating conditions. On the other hand, its high heat exchange characteristics, good thermo-physical properties and low pressure drop open up interesting possibilities in the definition of high performance heat exchangers.
The article illustrates first the key points of air cooler unit design, the differences compared to HFC products, and underlines the necessity of having low internal volume. DX air coolers and pump air coolers are analysed.
However, the CO2 gas cooler is the really challenging product. This unit cannot be considered a rearrangement of existing HFC condensers and a complete re-design is required.
The working pressure is almost 4 time higher (120 bar) and the temperature double (150°C). These characteristics require a different design which is able to match the severe design conditions and at the same time obtain an advantage from the completely different fluid properties. In comparison to an HFC condenser, a gas cooler can ensure a much higher level of air heating through the coil and low air flow is required with a consequent lower fan power input.
The paper illustrates the main steps in good gas cooler design (material selection, mechanical resistance, coil arrangement for high efficiency) and the development of a highly precise calculation method of such heat exchangers; several test results confirm the great accuracy of the software prediction. Design activity was focused on having a coil with small diameter tubes, low refrigerant charge and circuiting which can lower the approach temperature. Finally, for an efficient refrigeration plant it is necessary to ensure low gas cooler outlet temperature also in summer conditions. This is in fact a key parameter to reduce compressor power input and the solution of spraying demineralized water on coil surface appears to be very attractive. A detailed explanation of this technology is included, showing significant effects on energy saving, explaining the correct way to spray water on finned surface and indicating the physical properties of spray water.

1. INTRODUCTION

In the refrigeration industry, the utilization of “natural” fluids, including CO_2, is often proposed as a radical solution to eliminate the greenhouse effect caused by halogenated hydrocarbons belonging to the HFC category. CO_2 is a greenhouse gas, indeed the most important and the most notorious, but the quantities involved, even if used by the refrigeration industry on a massive scale, would be very small compared to those produced by combustion processes. Its GWP (Global Warming Potential) is in any case very low compared to the HFCs (one against several thousands). Furthermore, CO_2 does not exhibit any problem of flammability nor of impact on the ozone layer. Yet there is a serious risk that the use of CO_2 may not be entirely good, regarding greenhouse effect mitigation. Even though the direct contribution is practically zero, the indirect effect would be increased if the CO_2 refrigeration cycles were less efficient than traditional ones, due to larger electricity consumption bringing about larger emissions of CO_2 and of other pollutants from power stations, consuming more fossil fuels. For this reason, it is always worth bearing in mind that the technical solutions used to improve the environmental aspects cannot disregard the achievement of elevated thermodynamic efficiency. The appropriate choice of heat exchanger technology is a fundamental condition for obtaining COP values from CO_2 cycles allowing for a real reduction of the greenhouse effect. However, CO_2 is significantly different from all the other halogenated and non-halogenated fluids and it poses peculiar problems to heat exchanger designers: their discussion is the subject of this paper [6].

2. CO_2 CYCLES FOR REFRIGERATION

Before looking at the specific issues regarding heat exchangers, the possible methods of using CO_2 in refrigeration equipment and the aspects relative to the thermodynamic cycle must be focused upon, albeit only in summary. Three solutions are possible (fig.1):

- The simplest solution – with a normal refrigeration cycle operating with CO_2 (on the left in fig.1) – is the most difficult from the point of view of application: the low critical temperature of the CO_2 acts in such a way that, in order to be able to transfer heat to the ambient air, supercritical cycles would have to be used, therefore using pressures much higher than those used in normal use in the refrigeration industry. The heat exchanger used to cool down the high pressure CO_2 is named “gas cooler”: it will be analysed in detail later.
- In the diagram in the centre of fig.1 liquid CO_2 is produced by the evaporation of another fluid in a refrigeration cycle; it supplies the evaporators representing the cooling load; circulation being maintained by a simple pump.
- The third solution involves a cascade refrigeration cycle: there is a high temperature cycle acting as condensing media for the low temperature cycle operating on C_{O2}: in this case the C_{O2} never works at high pressures because it *de facto* condenses at a temperature close to the evaporation of the superior cycle (for example 0/-10°C).

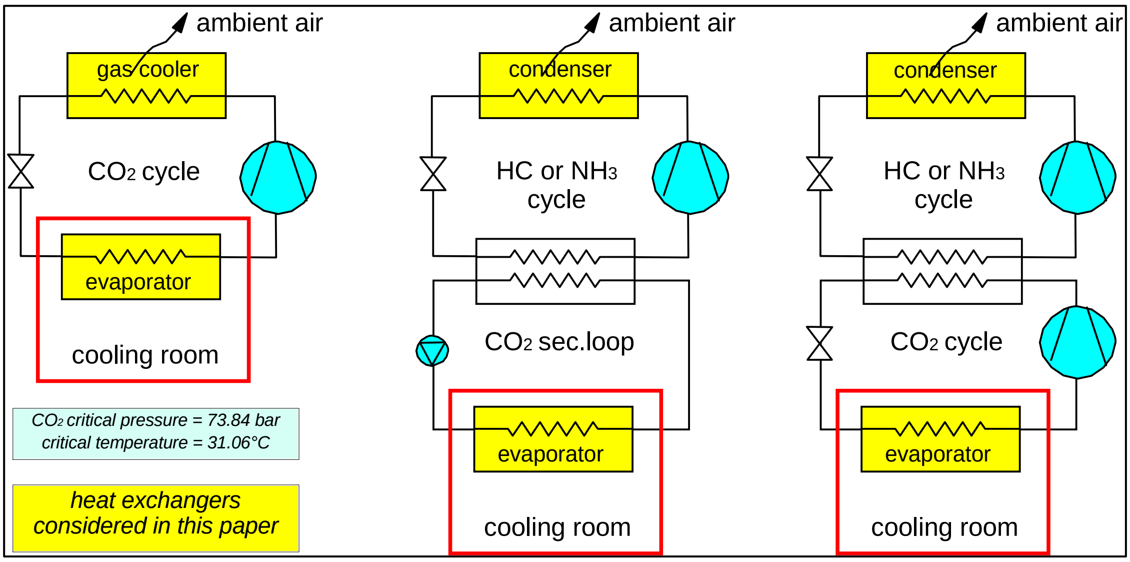

Fig.1: Possible uses of CO_2 in refrigeration equipment

All cases require a CO_2 evaporator operating at moderate pressure (from 14.3 to 38.6 bar passing from -30 to +4°C), discussed in section 3. The CO_2 condensers of solutions 2 and 3 are not of interest as they do not operate with air. In solution 1, the gas cooler represents an innovative device. To understand its operating conditions, reference must be made to the thermodynamic cycle of a CO_2 refrigerator as in the first scheme of fig.1. The form of the cycle, shown in figure 2, moves significantly away from that of a conventional cycle with condensing at constant temperature. In order to transfer heat to the ambient at a sufficiently high temperature, the maximum pressure is above the critical (73.84 bar) [5] and particularly high compressor outlet temperatures are shown.

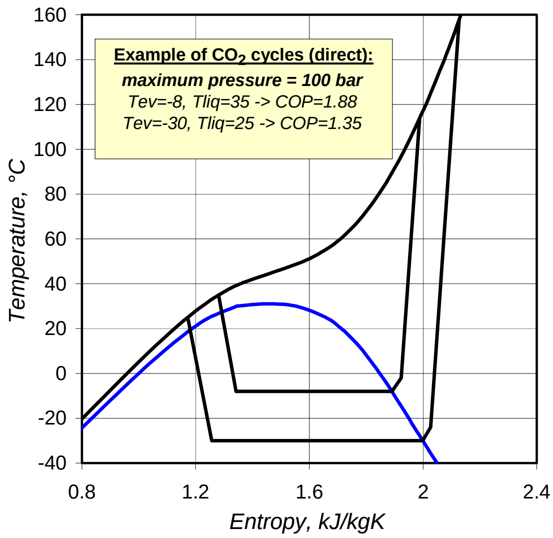

Fig.2: CO_2 refrigeration cycle

Given evaporation temperature by the application, the performance of a refrigeration cycle is determined by:

- maximum cycle pressure, at compressor outlet;
- liquid temperature at the gas cooler outlet;
- compressor efficiency (always set at 70% in this work[1]);
- effectiveness of any gas/liquid (GLHX) exchanger (set at 0.6 when present)
- superheating of the gas at the intake of the compressor (set at 6 K).

CO_2 thermodynamic properties used in this study are from Refprop. The influence of the first two parameters on COP is shown in figure 3, for evaporation temperature of -8°C, with or without GLHX. The gas cooler outlet temperature is the fundamental parameter influencing both COP and optimum pressure.

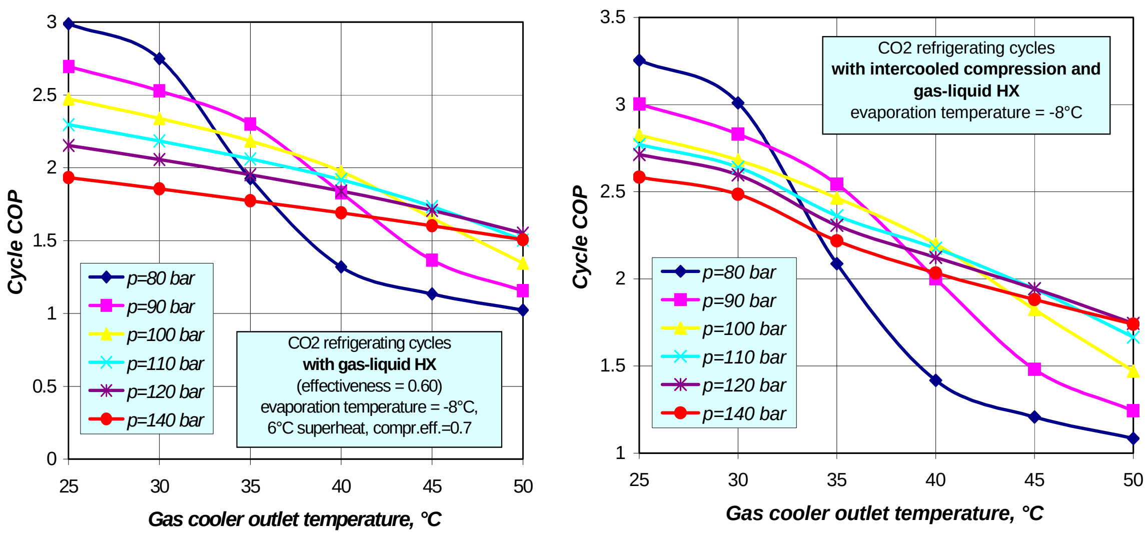

Fig.3: Performance of CO_2, cycles with and without GLHX, with evaporation at -8°C

[1] This assumption is relevant for predicting the actual value of the COP and may influence comparisons between CO_2 and conventional fluids. As this is not the scope of this study, this assumption is only for providing realistic COP values and does not influence the comparison among cycles using the same fluid.

It can clearly be seen in figure 3 that, given such a gas cooler outlet temperature, there is a pressure which maximises the COP: this tendency does not occur in cycles with conventional refrigerants (HC, HFC, NH_3) in which the lower the condensation pressure, the higher the COP. It can also be seen that the presence of the GLHX certainly improves the COP. Similar results are found for different evaporation temperatures. Thus, the gas cooler outlet temperature is a fundamental parameter for the cycle and constitutes the most important specification for the design of the exchanger.

3. EVAPORATORS

A CO_2 evaporator for refrigeration applications does not have to undergo especially high working pressures. However it is necessary to prevent overpressures caused by prolonged standstill of the equipment or by defrosting, when the temperature can rise well over that of the cooling room. Normally, max. working pressures of 45-60 bar are required. Such values do not impose a special design, but only an adjustment of tube and header wall thickness.
It is important to determine if an air cooler designed for HFC refrigerants can operate correctly for CO_2 and to estimate the capacity variations. It should be stated in advance that CO_2 thermo physical properties are favourable to obtaining elevated heat transfer performance. Table 1 shows that, compared to R404A, CO_2 has higher specific heat, higher thermal conductivity and lower viscosity.

Table1: Some thermo physical properties of CO_2 and R404A (source: NIST Refprop)

Temperature		-8 °C		-30 °C	
Fluid		CO_2	R404A	CO_2	R404A
Saturated liquid density	[kg/m^3]	972.1	1182.8	1073.5	1258.3
Saturated vapour density	[kg/m^3]	76.30	23.76	37.10	10.65
Saturated liquid specific heat	[J/(kg · K)]	2239.2	1347.1	1990.8	1273.8
Sat'd liquid thermal conductivity l.s.	[W/(m · K)]	124.2	81.5	155.1	91.3
Saturated liquid viscosity	[μPa · s]	123.7	198.0	181.6	264.1
Evaporation heat	[kJ/kg]	253.6	172.8	302.8	189.6

This last fact, along with the greater vapour density, allows fewer pressure drops (in terms of temperature variation due to the pressure variation) at the same mass velocity. Considering that (at equal capacity) the larger heat of evaporation brings about a lower through-flow, pressure drop reductions at the same power turn out to be very significant indeed. Table 2 shows the results of a theoretical prediction of a LU-VE unit cooler (Type F35HC 69 E 7) running on CO_2 (in terms relative to R404A) at two different evaporation temperatures, in the following hypotheses;

- Unchanged specifications: a slight increase in power at -8°C, becoming more consistent at low temperatures (from 3.5 to 11%); fluid velocity and pressure drops are very low.
- Reducing the number of feeds: in-tube velocity return to optimal values and 6-7% capacity improvement is shown compared to the previous case
- Reducing the number of inlets and using smooth instead of micro fin tubes: micro fin tubes are useful with poor refrigerant heat transfer coefficient: their usefulness is reduced at high evaporation temperature, but remains significant at low temperature with a low density fluid.

Table 2: Comparative performance of F35HC 69 E 7 unit coolers for R404A and CO_2

	fluid		R404A	CO_2		
	type of tube		micro fin			smooth
	no. of parallel inlets		N	N	N/2	N/3
Tev= –8°C, ΔT_1= 8K	rating (rel. to R404A)	[%]	100.0	103.5	110.6	108.2
	Mass flow rate	[kg/(m^2·s)]	76.8	71.6	149.8	231.0
	Pressure drop	[K]	0.32	0.025	0.25	0.66
Tev= –30°C, ΔT_1= 6K	rating (rel. to R404A)	[%]	100.0	111.1	117.7	112.0
	Mass flow rate	[kg/(m^2·s)]	53.4	43.4	90.8	137.0
	Pressure drop	[K]	0.66	0.03	0.20	0.52

4. GAS COOLERS

The gas cooler design is notably more complex, also due to the higher operating pressure (up to 150 bar), and poses some relevant peculiarities.

4.1 Thermodynamic aspects

The fundamental aspect for the thermodynamic design is that, as a consequence of the high average temperature along the upper isobar (responsible for the modest COP values), with CO_2 it is possible to bring the cooling air to much higher temperatures than those occurring with a refrigerant having a condensation phase at a constant temperature. Figure 6 shows this situation very clearly: it is evident that with CO_2 an air ΔT 2-3 times greater can be obtained. Consequently, it is possible to use an airflow reduced by the same proportion at equal thermal capacity. The large reduction in the airflow gives notable advantages in terms of reduced front area of the fin pack (i.e. unit foot print) and of electric power required for ventilation.

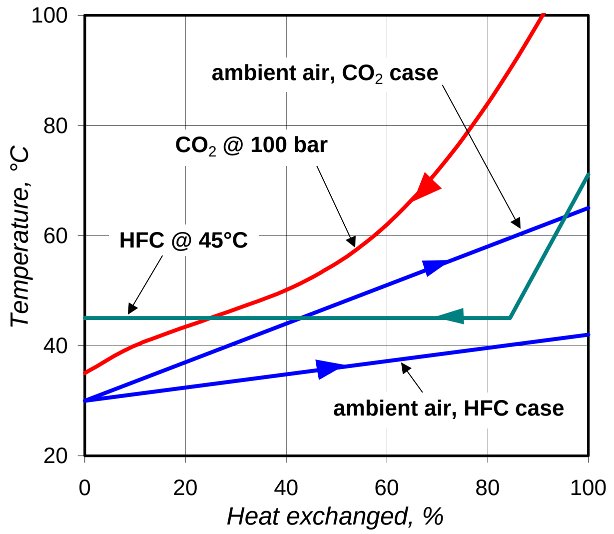

Fig.6: Heat transfer diagram for a CO_2 gas cooler and for an HFC condenser

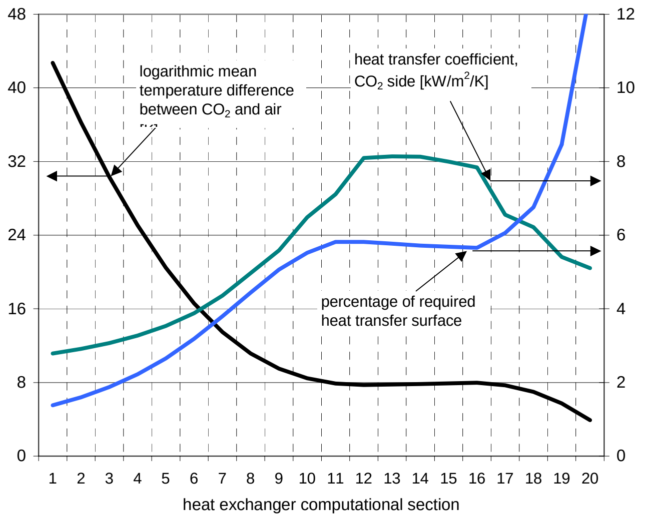

Fig.7:Variations of some parameters in the computational sections of a CO_2. gas cooler

To quantify these statements, a calculation method was developed capable of accounting for the particular distribution of the ΔTs between CO_2 and air (as in figure 6), provided that flows are arranged to run counter current[2]. The exchanger is subdivided into 20 computational sections: for each one an independent evaluation is done of the average logarithmic ΔT and of the in-tube heat transfer coefficient, with the Gnielinski correlation for single phase flows. Figure 7 shows an example of how some important parameters vary in the computational sections. It can be noticed that: (i) the heat transfer coefficient presents a maximum close to the critical point , (ii) the required surface area increases significantly in the cold end, due to the reduced ΔT between the two fluids and to the low liquid velocity.

Table 3 shows a comparison between a R404A condenser of standard production type SAV8T 3131 (168 kW capacity at ΔT=15K) and CO_2 gas coolers of the same capacity range. Since the CO_2 outlet temperature plays a preponderant role, the comparison was carried out in two ways: (i) at equal capacity, varying the final temperature, and (ii) at a final ΔT of 3 K, varying the capacity. In detail the solutions proposed in table 3**:**

- The first solution is the R404A reference.
- The second solution presents the same fin dimensions (frontal area and rows) and the same ventilation.

The rating is exuberant (last line) or, as an alternative, a very reduced ΔT can be obtained (the 0,2K value is only in theory, however, and with perfect counter flow) all of which is caused by the very large ΔT between CO_2 and air (at equal air flow). The above mentioned possibility of reducing the airflow was not exploited in this solution

---

[2] In plate-fin coils with 3-4 rows (or more) it is usually possible to arrange the circuiting in order to obtain a fluid path very close to counter flow, with negligible influence on the predicted performance.

- The third solution thoroughly exploits this possibility, using only one fan instead of three. The exchanger surface is redistributed to best adapt to a reduced airflow: the number of rows is doubled and the front section was halved, with a heat transfer surface practically the same as the original. The thermal rating at final ΔT of 3K is slightly less than the reference (-4%) in the presence of major reductions in the dimensions (50%), in the ventilation power (66%) and in the noise level (4.8dB).

Table 3: Comparative performances of air cooled condensers with R404A and CO_2 under the following conditions: air temperature 25°C, condensation R404A 40°C, CO_2 pressure 100 bar

		R404A	**CO_2**	**CO_2**
number of fans (8 poles)		3	3	1
Air quantity	[m^3/h]	37.800	40.000	15.000
Air outlet temperature (at equal capacity)	[°C]	38,6	46,1	62,7
front coil area	[m^2]	5,28	5,28	2,56
number of rows		3	3	6
number of feeds		66 (std)	22	21
tube specifications		3/8"micro fin	5/16" smooth	5/16" smooth
Fluid inlet temperature	[°C]	65,0	115,0	115,0
Fluid outlet temperature, at equal capacity	[°C]	40,0 (condensation)	25,2°C	28,8°C
Relative thermal capacity with CO2 gas cooler outlet temperature 28°C		100 (ΔT=15)	158 (ΔT=3)	96.0 (ΔT=3)
Fluid pressure drop (at equal capacity)	[kPa]	12,7 (=0,3K)	67,2	88,1

Even if this is a theoretical calculation, it gives some clear indication concerning the possibility of using the CO_2 peculiarity to design an efficient heat exchanger. In fact if a gas cooler, because of high pressure, poses serious difficulty to designers it may also open new doors leading to the exploitation of innovative construction characteristics, which are very different from traditional HFC condensers. In general one can conclude that the use of CO_2 could bring about significant reductions in the size of the equipment (in relation to the reduced ventilation) compared to equipment with similar ratings for conventional refrigerants, even with small final ΔT values (for example, 3K as in tab. 3).

4.2. Coil geometry definition

An important aspect to be analyzed is the coil arrangement. As described in section 3, the refrigerant flow involved in the plant is reduced compared to HFC as is pressure drop (considered in K). These aspects combined with high pressure (forcing the reduction of refrigerant charge) suggest designing a coil with small diameter tubes. A profound technical analysis made first with CFD simulation [3], [4] and thereafter verified in the laboratory by using a sophisticates wind tunnel, offers as best compromise the use of a fin geometry of 25 x 21.65mm, with spacing of 2.1mm and louvered turbolators [2]. The tube diameter is 5/16". The fin shape configuration is the result of long research activity aimed at achieving very high efficiency with a coil configuration with lower air flow and higher coil depth than a normal HFC condenser. In Fig 9 is shown the deviation between CFD simulation and test result at wind tunnel at 1 and 3 m/s air face velocity. The max deviation is 7,5% for air side heat transfer coefficient, similar value is for air pressure drop.

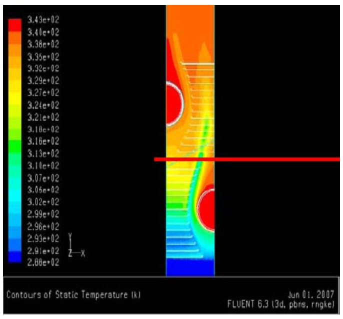

Fig 8 - Fin CFD simulation

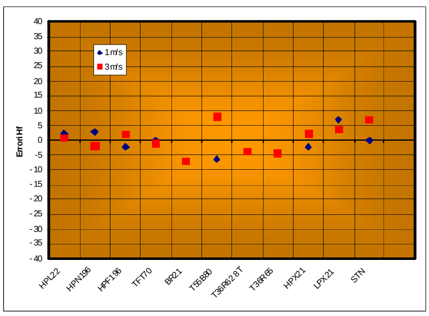

Fig 9 - deviation CFD vs test results

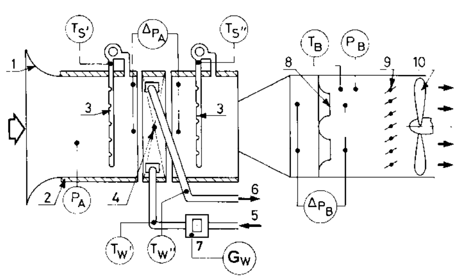

Fig 10 - Wind tunnel LU-VE laboratory

The fin pack is deliberately interrupted to allow for different thermal expansion and to avoid thermal conduction along the fin thickness [1]: it should be remembered that a large ΔT occurs in gas coolers (es: 120°->20°C), much higher than in condensers.

As a result of cooperation with an important tube manufacturer, a special copper alloy was introduced, tested and finally approved by TUV München. Its denomination is K65 and it has an yield point twice that of normal copper, allowing the use of such material and ensuring a maximum operating pressure of 120 bar.

Tab.4: Chemical composition of Copper alloy K65

Materials composition	**Cu %**	**Fe %**	**Pb %**	**Zn %**	**P %**	**Others**
Tube K65	rest	2,1 ÷ 2,6	0,03 (max)	0,05 ÷ 0,2	0,015 ÷ 0,15	0,20

4.3 Spray system

As anticipated on page 3, the gas cooler outlet temperature is a key point in ensuring good COP also at max. ambient temperature. When in fact the outdoor temperature is 35°C or higher, the plant efficiency drops off and power consumption rises. Referring to the left-hand graph in Fig 3, for an ambient temperature of 35°C a CO_2 gas cooler outlet temperature of 37°C with COP = 1,7 (pressure 90 bar) can be assumed. Indeed, by using a spray system the CO_2 gas cooler outlet temperature can be 30°C with COP = 2,3 (pressure 90 bar). The increase is 35%. Of course this is valid at high ambient temperatures that occur for a limited period of the year (depending on geographic location); however the spray system can be an interesting solution.
Water spray is a feature developed by LU-VE for conventional condensers and dry-coolers which proves to be of particular interest for CO_2 applications [8]. The idea behind water spray is rather simple. In most applications, extreme summer conditions which occur for a few hours per year impose an over-sizing of the heat dissipation devices and/or severe penalties on the cooling capacity and the COP. It is therefore convenient to spray some water, just for those periods on the coil surface to dramatically reduce the condensation temperature, or, in the CO_2 case, the gas cooler outlet temperature
Serious attention has been paid to the definition of the water quality to be sprayed. Sophisticated tests were done in the laboratories, in particular concerning the resistance to corrosion and limescale deposits on aluminium fins with special protective coatings with conditions of different water qualities (see Fig 11). To work properly, the spray water needs to have the following characteristics:

- be in accordance with European Directive 98/83/EC
- PH in the range 6 to 8
- Conductibility < 1500 μS/cm
- Chloride < 200 mg/l (200 ppm)

This water, before being nebulized, has to undergo a softening process to reduce its hardness which has to be between 2 and 4 °F (or 1,1 - 2,2 °H). If indeed normal water is sprayed without any softening process, rapid limescale deposition may occur as is well shown in Fig 12. The result can be seen there of the test we carried out using normal water (with rather low hardness of only 18°F) and with soft water (3°F).

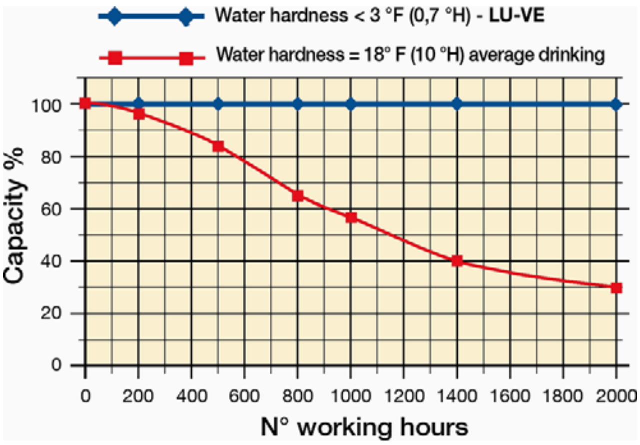

Fig. 12 – performance test with different water hardness

Fig 13 – CO2 gas cooler with spray (summer capacity =500kW)

After only 500 working hours the limescale deposition reduced the performance by nearly 20%. We suggest using the spray system for a period of 200 ÷ 500 hours/year depending on the location of the plant.

No problems concerning the hygienic aspect can occur (i.e. legionella) because most water is evaporated and any that remains is evacuated and not recycled as is the case in cooling towers.

5. CONCLUSIONS

The applications of CO_2 in the refrigeration industry could shortly become an important reality. From the heat exchanger point of view, the utilization of CO_2 poses some problems (greater operating pressures) but also offers notable opportunities, especially in the most difficult design case of the gas coolers. We have seen that reductions of the airflow and of the coil front area can be achieved, at equal capacity and with very small final ΔT values (this last being an essential parameter for obtaining a good COP of the cycle). It brings about lower fan consumption, smaller size and some production cost savings, counterbalanced by the increased use of copper resulting from the thicker tube walls and headers. The fin-and-tube geometries used for conventional fluids are perfectly adequate to CO_2 application in the case of LU-VE production, which has for many years concentrated on small diameter tubes even for large units. The use of the water spray system helps the plant to significantly improve general efficiency at high ambient temperatures.

6. REFERENCES

[1]. Wang CC, Recent progress on the air-side performance of Fin-tube Heat Exchangers, International Journal of Heat Exchanger1524-5608/vol1 (2000), pp 49-76.
[2]. Lozza G., Merlo U. An experimental investigation of heat transfer and friction losses of interrupted and wavy fins for fin-and-tube heat exchangers. International Journal of Refrigeration 24 (2001) pp. 409-416
[3]. Patankar S.V. Numerical heat transfer and fluid flow, Mc Graw-Hill, New York, 1980.
[4]. Sunden B., Brebbia C.A., Advanced computational methods in heat transfer VII, Proceeding of the Seventh International conference on advanced computational methods in heat transfer, Halkidiki, Greece, April 22-24, 2002.
[5]. Ashrae Handbook (2009), Fundamentals
[6]. Ashrae Handbook (2010), Refrigeration.
[7]. Colombo E., Macchi E., Merlo U., Strategy for innovation in heat exchanger design: computational approach combined with experimental tests leads to performance improvement. Summer Heat Transfer Conference – Westin St. Francis, San Francisco, CA, USA (2005)
[8]. G. Lozza, S. Filippini, F. Zoggia, Using "water-spray" techniques for CO2 gas coolers. XII European Conference on "Technological Innovations in Air Conditioning and Refrigeration Industry", June, 2007, Milan, Italy