AIR COOLED HEAT EXCHANGERS FOR CO_2 REFRIGERATION CYCLES

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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 (such as R134a, R404A, R407C, R507, R410A, etc). 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 (1 against several thousands). Furthermore, CO_2 does not exhibit any problem of toxicity and flammability nor of impact on the ozone layer. Yet there are serious risks that the use of CO_2 may not be an entirely good idea, 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 (lower COP), due to larger electricity consumptions bringing about larger emissions of CO_2 and of other pollutants from power stations, consuming more fossil fuels[1]. The appropriate choice of heat exchanger technology is a fundamental condition for obtaining COP values form 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.

2. CO_2 heat exchangers

In refrigeration plants using CO_2 as the working fluid, two types of heat exchangers are used:

- <u>Evaporators</u>, which are included in every proposed plant configuration: in direct CO_2 cycles, in binary cycles (using a low temperature CO_2 cycle and an higher temperature cycle, operated by another fluid and rejecting heat towards the ambient) and in other systems using CO_2 as the cold energy carrier, condensed by a refrigerating machine and evaporated by the users device. Evaporators, working at low temperature, do not require elevated operating pressure and therefore are not substantially different from models for halogenated fluids.
- <u>Gas-coolers</u>, which are included in direct cycles only to reject heat towards the ambient. They perform the same duty of conventional fluid condensers, but rather than condensation (implying a two-phase equilibrium) a simple transition from the expanded gas phase to the liquid state takes place. As a matter of facts, having CO_2 a very low critical temperature of 31°C, a supercritical operating pressure is necessary to maintain a temperature higher than the one of ambient receiving heat from the cycle. Being the critical pressure of 73.8 bar, operating pressures much larger than those of conventional cycles will be adopted.

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[1] Specific emissions of CO_2 from power stations vary from 350 to 800 g/kWh. The lower values are for natural gas combined cycles, the higher for conventional coal-fired stations.

The typical shape of supercritical cycles is shown in fig.1. Compared to conventional cycle rejecting most of their thermal capacity at a constant temperature, supercritical cycles performance are not only influenced by the minimum and maximum pressure, but their COP is strongly affected by the gas cooler outlet temperature, i.e. the temperature of the liquid at the expansion device entrance[2]. This is very important to obtain acceptable COP: as a matter of facts CO_2 cycles perform very brilliantly with low coolant temperatures (e.g. water-heating heat pumps, low ambient temperatures in cool regions). For a given ambient temperature, the gas cooler exit temperature is imposed by the design characteristics of the gas cooler, therefore assuming a fundamental role as far as the cycle performance are concerned.

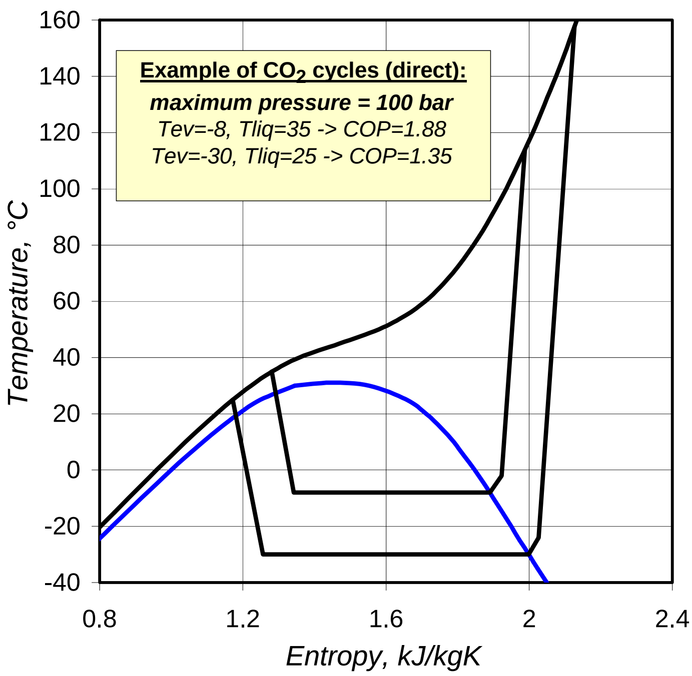

Fig.1: Examples of inverse CO_2 supercritical cycles.

3. Evaporators

A CO_2 evaporator for refrigeration applications does not have to undergo especially high working pressures (table 1). 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. Rather than oversizing the evaporator and the refrigerant lines, it is preferable to adopt expedients which can limit the project pressure to 60 bar or even less (safety valves, pump-down to remove liquid from the evaporator). Such pressure values are just a little above those normally used in refrigeration (all Lu-Ve evaporators are tested at 40 bar) and do not impose any special design, even if larger thickness of coil tubes and headers are usually adopted (e.g. from 0.35 to 0.5 mm for 3/8" copper tubes).

T	-40	-35	-30	-25	-20	-15	-10	-5	0	5	10	15	20	25	30
p	10.04	12.02	14.26	16.81	19.67	22.88	26.45	30.42	34.81	39.65	44.97	50.81	57.22	64.25	72.05

Tab.1: Relationship between temperature [°C] and pressure [bar] for CO_2.

On the other hand, it is interesting to determine if an aeroevaporator designed for conventional refrigerants can operate correctly for CO_2, with no or limited modifications, and, if so, to estimate the variations of thermal power. It should be stated in advance that the thermophysical properties of CO_2 are favourable to obtaining elevated heat transfer performance. Compared to R404A, CO_2 has higher specific heat, higher thermal conductivity and lower viscosity. This last fact, along with the greater vapour density, allows fewer pressure drops at the same mass velocity. Considering that (at equal capacity) the larger heat of evaporation brings about a lower throughflow, pressure drop reductions at the same power turn out to be very significant indeed. Table 2 shows the results of a theoretical predition of a Lu-Ve unit cooler 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.

[2] Liquid temperature also affects the performance of conventional cycles, provided that a specific heat transfer section is devoted to sub-cooling, but at a much lower extent.

- Reducing the number of feedings: in-tube velocity return to optimal values and 6-7% capacity improvement is shown compared to the previous case; reducing the number of feedings reduces the cost of the gas header and distributor.
- Reducing the number of inlets and using smooth tubes instead of microfin tubes (helically grooved microfins such as those normally used in LU-VE unit coolers): microfin tubes are particularly useful with poor refrigerant heat transfer coefficient: their convenience is very reduced at high evaporation temperature, but remains significant at low temperature with a low density fluid (-30°C).

fluid	R404A	CO_2		
type of tube	microfin			smooth
no. of parallel inlets	N	N	N/2	N/3
rating (rel. to R404A), Tev= –8°C, ΔT_1= 8K	100.0	103.5	110.6	108.2
rating (rel. to R404A), Tev= –30°C, ΔT_1= 6K	100.0	111.1	117.7	112.0

Tab.2: Comparative performance of unit coolers for R404A and CO_2. The ratios are valid for some representative models but are not applicable in general.

The last two solutions permit a modest improvement of the specific cost (€/kW) of the equipment, as long as the pressures of the design do not exceed 40-60 bar. Lu-Ve has already supplied various clients with CO_2 unit coolers (about 200 units sold – spring 2005 – for cooling rooms or refrigerated cases); up to now no visible indications have arisen of the slightest power deficit nor of any operating problems.

4. Gas coolers

The gas cooler design is notably more complex, also due to the larger operating pressure (up to 150 bar), and poses some relevant peculiarities. 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 constant temperature. Figure 2 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, of electric power required for ventilation and of the initial cost of the fans and their regulators.

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 2), provided that flows are arranged to run countercurrent[3]. The exchanger

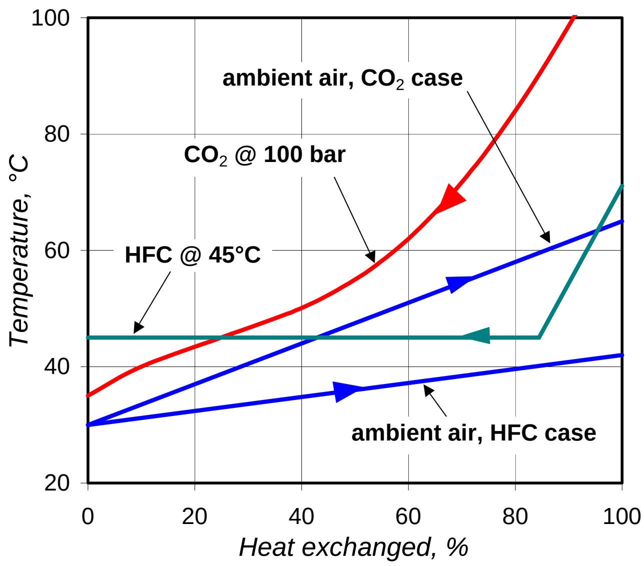

Fig.2: Heat transfer diagram for a CO_2 gas cooler and for a condenser using a conventional refrigerant.

[3] 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 counterflow, with negligible influence on the predicted performance.

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 3 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.

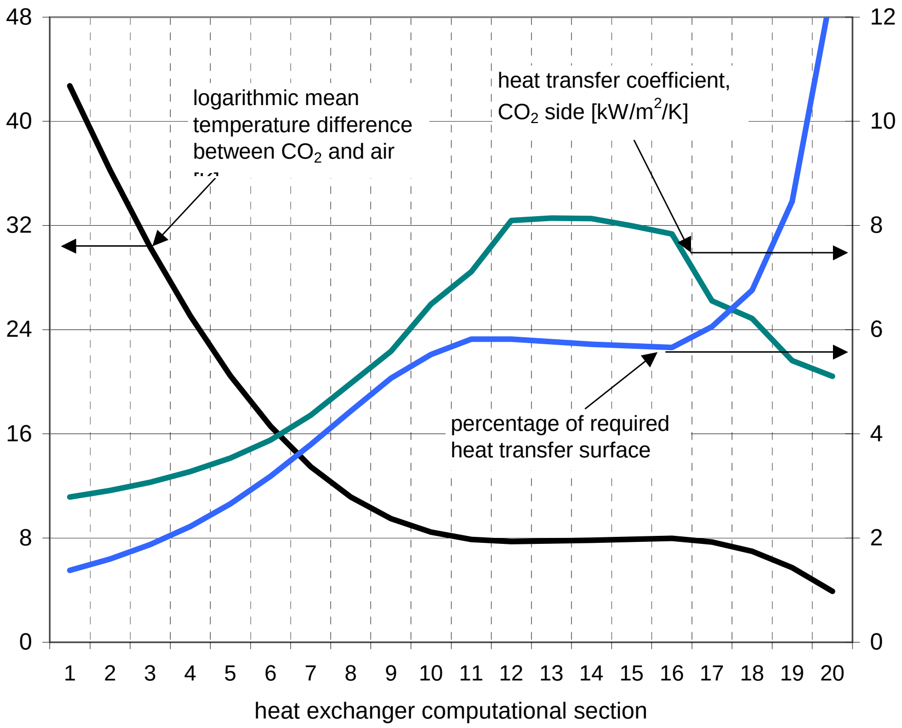

Fig.3:Variations of some parameters in the computational sections of a CO_2. gas cooler

Table 3 shows a comparison between a R404A condenser (capacity of about 170kW with initial ΔT of 15K) and CO_2 gas coolers of the same power range. Since the CO_2 outlet temperature plays a preponderant role, the comparison was carried out in two ways: (i) at equal power, varying the final temperature, and (ii) at a final ΔT of 5 K, varying the power. The solution considered uses a fin geometry of 25 x 21.65mm, with spacing of 2.1mm which is produced by LU-VE with louvered turbulators. A standard 3/8" microfin tube was used for R404A, while for CO_2 it is preferable to use a smaller diameter tube (5/16") with a thicker wall to withstand the working pressures required by gas coolers. It must be noted that 5/16" (8 mm) copper tubes with 1 mm thickness can withstand an operating pressure of 190 bar (ASTM rules), collapsing at 750 bar; the same figures for 3/8" tubes are 150 and 600 bar. These tubes are not available in microfin versions which would in any case be of little use given the elevated heat transfer coefficient (fig.3). The number of parallel feedings is optimised in all cases. The following solutions are proposed in table 4**:**

- The first solution is the R404A reference (in normal production).
- 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 0.3 value is, however, only valid for perfect counterflow) 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
- 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 an 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). It should be pointed out that in these cases the outlet air temperature is in the range of 60°C: it is therefore convenient to place the fans at the coil inlet (forced draft) to avoid thermal stress to the motor and to increases the mass air flow, compared to the usual solution of induced draft (fan at coil outlet).

fluid	**R404A**	**CO_2**	**CO_2**
number of fans (8 pole)	3	3	1
front coil area, m^2	5.28	5.28	2.56
number of rows	3	3	6
number of inlets	66 (std)	22	21
tube specifications.	3/8"x 0.35 microfin	5/16"x 1.0 smooth	5/16" x 1.0 smooth
fan positioning	induced-draft	induced-draft	forced-draft
cooler outlet temperature, at equal power (air at 25°C)	40°C (condensation)	25.3°C (ΔT=0.3)	28.8°C (ΔT=3.8)
or: (relative) thermal rating cooler outlet temp. =30°C	100 (ΔT=15)	158 (ΔT=3)	96.0 (ΔT=3)

Tab.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.

In general, the optimum solutions may vary depending on the design survey and on the requirements imposed by the compatibility with existing models, for industrial reasons. However, 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. 4).

LU-VE has gained a good experience with CO_2 gas coolers with about 20 units in operation (mid 2006: their number is rapidly increasing…), including some large equipment (see for instance fig.4). Some particular manufacturing solutions were defined:

- the fin pack is properly interrupted to allow for different thermal expansion and to avoid thermal conduction along the fin thickness: it should be remembered that a large ΔT occurs in gas coolers (es: 120°->20°C), much higher than in condensers;
- the pressure test is carried out in three steps: (i) with air at 30 bar in a water pool to detect major leaks, (ii) with water at 170 bar, (iii) again with air at 30 bar to detect leaks caused by the previous pressurization; (iv) the coil is de-hydrated by vacuum pumping to a pressure of about 2 mbar.

The CO_2 gas cooler product can therefore be considered "proven technology" in the refrigeration field. This achievement was made possible because of the design strategy adopted by LU-VE, consisting of the utilization of high performance heat transfer surfaces and of miniaturized geometries (small diameter tubes) even for large heat exchangers. This strategy and the following manufacturing experience are now precious for CO_2 applications, without the need of resorting to 'exotic' (and unproven!) technologies, such as aluminium heat exchanger with microchannel sometimes proposed for automotive air-conditiong, hardly applicable to the refrigeration field (requiring much larger units and not allowing for large scale production).

Fig.4: A large CO_2. gas cooler ready for shipping at LU-VE workshop.

5. Water spray

Water spray is another feature developed by LU-VE for conventional condensers and dry-coolers which resulted of particular interest for CO_2 applications. The idea behind water spray is rather simple. In most applications, extreme summer conditions, occurring for few hours per year, impose an over-sizing of the heat dissipation devices and/or severe penalties of the cooling capacity and of the COP. It is therefore convenient to spray some water, just for that periods, on the coil surface to dramatically reduce the condensation temperature, or, in the CO_2 case, the gas cooler outlet temperature, which strongly affects the cycle performance as we discussed in chapter 2 (see for instance fig.3). Therefore water spray is a precious feature for gas coolers: the water injection system is clearly visible in fig.8. Water consumption is very limited on an yearly basis, provided that a proper control system is adopted, because it is used for few hours/year (i.e. 200-500). No hygienic problems may occur (i.e. legionella) because most water is evaporated and the remainder is evacuated (not recycled as for cooling towers).

However water spray poses an important issue, given by the deposition of solids on the fin surface, depending on the water characteristics. LU-VE offer two systems: the standard one is rather inexpensive, including a sweetener, and it is suggested for short yearly periods of water injection; a second one is much more sophisticated, including a reverse osmosis plant to guarantee an unlimited coil life even if used for thousands of hours/year. Both systems can be optimized for CO_2 application and are readily available for applications.

6. 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 presures) 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. At present, for the refrigeration sector (wide capacity range, small production volumes) it would not seem necessary nor convenient to adopt particular geometries with excessively miniaturised specifications, which could however be opportune for smaller application with a large production volumes (for instance, automotive air conditioning).