Air Cooled Heat Exchangers for CO_2 Refrigeration Cycles

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1. Introduction

Using "natural"refrigerants, including CO_2, is often seen as radically solving the greenhouse effect caused by halogenated hydrocarbons of the HFC category (such as R134a, R404A, R407C, R507, R410A, etc). The most important and notorious is the greenhouse gas CO_2 but the quantities involved - even if refrigeration industry uses it massively- would be very small compared with what combustion generates. Its GWP (Global Warming Potential) is very low compared with the HFCs (one in several thousands). CO_2 does not exhibit any problem of toxicity and flammability nor does it impact on the ozone layer. But serious risks from CO_2 may not be an entirely good idea, regarding the greenhouse effect. Although the direct contribution is practically zero, the indirect effect would increase if the CO_2 refrigeration cycles were less efficient than traditional ones (lower COP), as larger electricity consumptions emit larger amounts of CO_2 from power stations. CO_2 differs significantly from all the other halogenated and non-halogenated fluids, posing peculiar problems to heat exchanger designers: now discussed in this paper.

2. CO_2 Heat Exchangers

In refrigeration plants using CO_2 there are two types of heat exchangers:

• Evaporators, which are in every proposed plant configuration as direct CO_2 cycles, in binary cycles (using a low temperature CO_2 cycle and a 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 refrigeration and evaporated. Low temperature evaporators do not need elevated operating pressure and therefore do not substantially different from models for halogenated fluids.

• Gas-coolers 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) simply cross from the expanded gas phase to the liquid state. Having CO_2 - at a very low critical temperature of 31°C, a supercritical operating pressure is needed to keep a temperature higher than the ambient receiving heat from the cycle. At the critical pressure of 73.8 bar, they will operate at much larger pressures than conventional cycles.

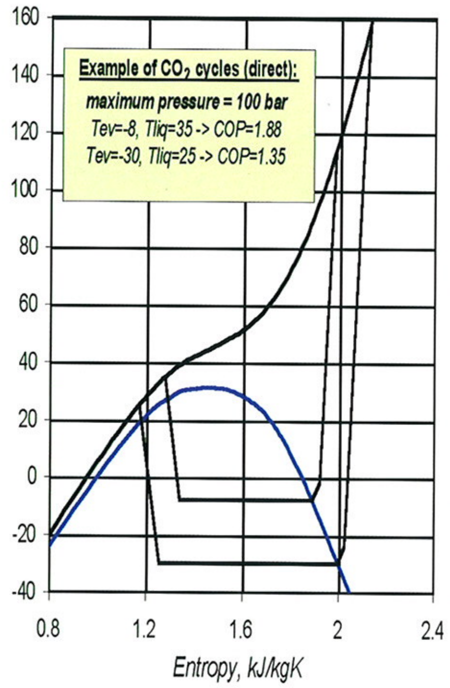

Fig.1: Examples of inverse CO_2 supercritical cycles.

Fig. 1 shows the typical shape of supercritical cycles. Compared with the conventional cycle rejecting most of their thermal capacity at a constant temperature, supercritical cycles are not only influenced by the lowest and highest 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. 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). The design characteristics for a given ambient temperature the gas cooler are imposed by the gas cooler temperature, therefore assuming a fundamental role as far as the cycle performance is concerned.

3. Evaporators

A CO_2 evaporator for refrigeration applications does not have to undergo specially high working pressures (table 1). However, to stop 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, 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 fitted (e.g. from 0.35 to 0.5 mm for 3/8" copper tubes).

Otherwise 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 thermal power variations. It should be stated in advance that the thermophysical properties of CO_2 favour to obtaining elevated heat transfer performance. Compared with R404A, CO_2 has higher specific heat, higher thermal conductivity and lower viscosity. This, 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 a lower throughflow, pressure drop reductions at the same power turn out to be very significant . Table 2 shows the theoretical predition of a LU-VE unit cooler running on CO_2 (in terms relative to R404A) at two different evaporation temperatures, in these 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 feedings: in-tube velocity return to optimal values and 6-7% capacity improvement is shown compared with the previous case; reducing the feedings reduces the cost of the gas header and distributor.

• Reducing the 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 stay significant at low temperature with a low density fluid (-30°C).

The last two improve the specific cost ($/kW) modestly of the equipment, provided the design pressures 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). So far no visible indications have shown the slightest power deficit nor any operating problems.

4. Gas Coolers

The gas cooler design is more complex, also because of the larger operating pressure (up to 150 bar), posing 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 with a refrigerant having a condensation phase at constant temperature. Figure 2 shows this 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 largely reduced airflow notably advantages in reduced front area of the fin pack, of electric power required for ventilation and of the initial cost of the fans and their regulators.

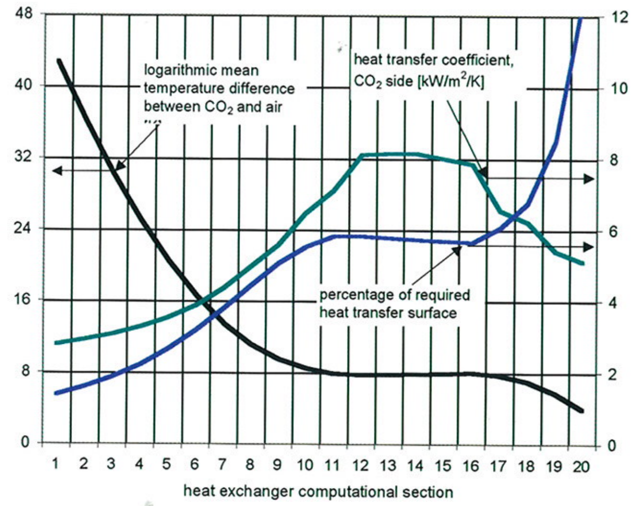

Fig.3:Variations of some parameters in the computational sections of a CO_2 gas cooler

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 2), provided that flows are arranged to run countercurrent . The exchanger is subdivided into 20 computational sections: feach one is evaluated independently with the average logarithmic ΔT and of the in-tube heat transfer coefficient, with the Gnielinski correlation for single phase flows. Figure 3 shows how some important parameters vary in the computational sections. Notice 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, because of the reduced ΔT between the two fluids and to the low liquid velocity.

Table 3 compares between a R404A condenser (capacity of about 170 kW 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, it was compared 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 21.65 mm, with spacing of 2.1mm which is produced by LU-VE with louvred turbulators. RA404A used a standard 3/8" microfin tube, while preferably for CO_2 to use a smaller diameter tube (5/16") with a thicker wall to withstand the working pressures the gas coolers need. Note 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 be of little use given the elevated heat transfer coefficient (fig.3). The number of parallel feedings is optimised in all cases. These solutions are proposed in table 4:

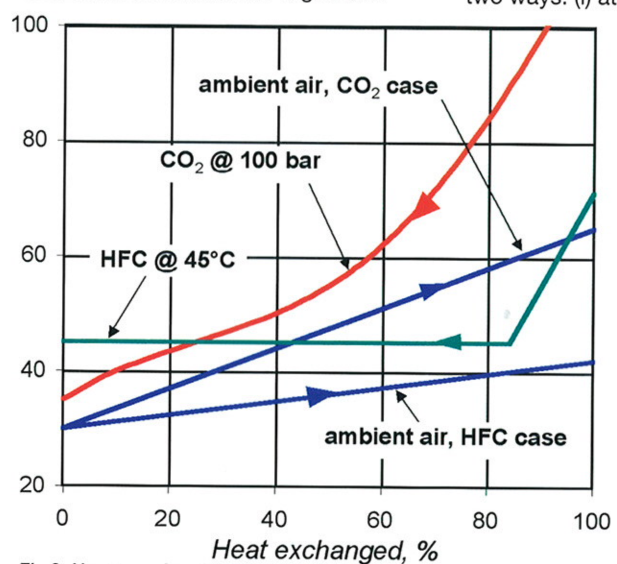

Fig.2: Heat transfer diagram for a CO_2 gas cooler and for a condenser using a conventional refrigerant.

• 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 *T can be obtained (the 0.3 value is, however, valid only for perfect counterflow). All these are caused by the very large *T 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 best to adapt a reduced airflow: the rows are 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 (4.8 dB). We point out that in these cases the outlet air temperature is in the range of 60°C: therefore it is convenient to place the fans at the coil inlet (forced draft) to avoid thermal stress to the motor and to increase the mass air flow, compared with the usual solution of induced draft (fan at coil outlet).

Generally the best solutions may vary depending on the design survey and on the needs imposed by the compatibility with existing models, for industrial reasons. However, one can conclude that the use of CO_2 could significantly reduce the size of the equipment (in relation to the reduced ventilation) compared with equipment with similar ratings for conventional refrigerants,

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

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even with small final △T values (for example, 3 K as in tab. 4).

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

- the fin pack is properly interrupted to allow different thermal expansion and to avoid thermal conduction along the fin thickness: remember that a large △T 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 pressurisation; (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 "proved technology" in the refrigeration field. This achievement was made possible because of the design strategy adopted by LU-VE, consisting of using high performance heat transfer surfaces and of miniaturised 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 micro-channel sometimes proposed for automotive air-conditiong, hardly applicable to the refrigeration field (requiring much larger units and not allowing for large scale production).

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 simple. In most applications, extreme summer conditions, occurring for few hours a year, impose an over-sizing the heat dissipation devices and/or severe penalties of the cooling capacity and of the COP. Therefore it is convenient to spray some water, just for such periods, on the coil surface to reduce the condensation temperature dramatically or, in the CO_2 case, the gas cooler outlet temperature, which strongly affects the cycle performance as 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 solids deposed on the fin surface, depending on the water characteristics. LU-VE offers 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 optimised 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 utilisation of CO_2 poses some problems (greater operating pressures) but also offers notable opportunities, specially in the most difficult design case of the gas coolers. We have seen that reducing the airflow and of the coil front area can be achieved, at equal capacity and with very small final △T values (this being an essential parameter for obtaining a good COP of the cycle). It lowers 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 geometrics 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 geometrics with excessively miniaturised specifications, which could, however, be opportune for smaller application with large production volumes (for instance, automotive air conditioning).

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