ENVIRONMENTAL FRIENDLY HEAT EXCHANGERS

F.ZOGGIA[(a)], S.FILIPPINI[(a)], C.PERFETTI[(a)], G.LOZZA[(b)]
[(a)] LU-VE Contardo, Viale della Liberazione, 21040 Uboldo (Va), Italy
[(b)] Dept.of Energy, Politecnico di Milano, Milano, Italy

ABSTRACT

Alternative solutions to traditional HFC plants, like indirect systems (making use of brine coolers) and natural refrigerants, are playing an increasing role in the refrigeration industry.
Brine coolers are a well-known technology, but a more widespread diffusion can be anticipated if better heat transfer capabilities can be achieved, reducing coolers size and cost, as well as pumping energy consumption. On another side, the novel technology of CO_2 cycles is characterized by excellent environmental performance, having negligible direct GWP (Global Warming Impact). However large attention must be paid to the indirect GWP, to obtain with CO_2 comparable annual COP of HFC or ammonia plants or, if possible, better. Hence, the availability of efficient, reliable and affordable heat exchangers is a relevant issue for CO_2 applications, as well as for brine coolers.
To discuss the above subjects, this paper includes two separate sections, regarding (i) the utilization of microfin tubes in brine coolers and (ii) air cooled heat exchangers for CO_2 direct cycles.

1. THE UTILIZATION OF MICROFIN TUBES IN BRINE COOLERS

1.1. General remarks

The utilization of low freezing solutions (usually made of ethylene glycol and water) is becoming a widely diffused solution in refrigeration plants. In this plant arrangement, the low freezing solution is cooled down in the refrigeration machine room, enabling the utilization of hazardous refrigerants like ammonia or hydrocarbons with zero GWP and reducing the refrigerant charge. The cooling brine is distributed in the liquid phase to the various cold users (cold rooms, cabinets, etc.), where the air coolers can be operated at a very low ΔT, while for evaporators a minimum ΔT of 6-8K is required to drive the expansion device.There are of course some drawbacks: (i) more hardware (i.e. costs) is needed, (ii) some additional energy is required, due to the circulation pump and to the double heat transfer from the refrigeration cycle to the cold room, (iii) the thermophysical properties of glycol solutions are not favourable: therefore, larger coolers are needed, negatively impacting on the investment cost. This paper discusses the adoption of microfin tubes in air coolers, as a measure to improve the heat transfer and to remove, as far as possible, the outlined drawbacks.

1.2 Background and experience

In the recent past, the authors' company had successfully introduced advanced heat transfer surfaces in the typically conservative market of heat transfer equipments for refrigeration. Starting from 1986, pioneer studies were undertaken to develop high performance fin geometries, characterized by louvered or wavy surfaces and by small absolute dimensions[1], and to make use of the technology of micro-fin tubes, at that times utilized just for some small air-conditioning application. Micro-fin

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[1] This represents a fundamental design philosophy, tending to apply small diameter tubes and very compact fin geometries even for large capacities: for instance the authors' company makes use of 3/8" (9.52 mm) tubes for condensers and dry-coolers exceeding 1200 kW (ΔT_1=15K), with coil lengths up to 12800 mm. The rationale behind this philosophy is to increase the heat transfer coefficients (proportional to $D^{-0.55\text{-}0.65}$ in a typical fin pack, by using a proportionally lower spacing between the tubes), and to reduce the tubes weight and costs (even if more tubes and more parallel feeding are used).

tubes with helical grooves provide significant enhancements of the heat transfer coefficient during evaporation and condensation, with moderate improvements of the pressure losses: a large number of papers can be found in the literature about this subject. However, when applied to single-phase fluids (i.e. liquids) the situation is much more uncertain: the larger surface made available by micro-fins improves the heat transfer per meter of tube, but usually also improves the pressure loss at a faster rate. The relative improvement of both heat transfer and pressure loss depends on the Reynolds number *Re*. Different results may occur, depending on the flow regime: since most refrigeration applications for low freezing solutions actually operate in laminar or transition regimes (i.e. Re<3000-3500), one may move from laminar to turbolent achieving abrupt changes in the heat transfer characteristics. Therefore the utilization of micro-fin tubes is not straightforward as it is for two-phase flows. Particular geometries are to be developed to account for the heat transfer / pressure drop behaviour at low Reynolds numbers.
The selected test method was to compare the performance of two air coolers having the very same characteristics, reported in tab.1, one using the new proposed tubes, one using the conventional smooth tubes. Compared to a more direct type of investigation (for instance, by using an electrically heated test rig, providing a known heat flux directly to the tube outer surface), in this way the results (even if affected by somewhat larger uncertainties) will surely take into account all the effects encountered in the industrial application (like entrance effects after bends and headers, deformation of the internal grooves due to mechanical expansion and so on).

Tab.1: Characteristics of the aircoolers.

Length of the fin pack	1620 mm
Number of tubes / row	18
Number of rows	4
Tubes diameter	12.7 mm
Tubes spacing	42x36 mm
Fin spacing	7 mm
No. / diameter of fans	2x500mm

1.3. The experimental apparatus

The capacity of the two air coolers of tab.1 was measured by means of a calibrated cold room in the LuVe laboratories (fig.1, leftmost side). It consists of a double insulated room, the external one kept at the same temperature of the internal one (the test room) to minimize the thermal losses. The refrigeration capacity of the coolant is balanced by the electric power provided to the fans and by the warm water feeding the ‘balancing coils’. Mass flows and temperatures (inlet, outlet) of the coolant and of the warm water are measured, as well as the electric power introduced, so that the air cooler capacity is double checked, after a pretty long period of stabilization (typically 12 hours), also useful to remove all the moisture from the air in the room (“dry” test). The difference between the two capacity measurements (direct on the aircooler and indirect on the heat introduced in the cold room) is typically within 2%.
The in-tube heat transfer coefficient is derived from the measured capacity by the following procedure: (i) the inlet air temperature (room temperature) is measured, by averaging the indications of 8 thermocouples distributed on the coil front area, (ii) the refrigerant inlet/outlet temperatures are also measured, (iii) the air flow is measured by a test conducted in a wind tunnel, (iv) from capacity, airflow and inlet temperature, the air outlet temperature is calculated, by assuming dry operation (no latent heat), (v) the log-mean temperature difference can be evaluated, as well as the overall heat transfer coefficient, for a known capacity and inner surface (i.e. the internal surface of the smooth tube, taken as the reference surface also for the micro-fin tube), (vi) the in-tube heat transfer coefficient is derived from the overall one, by using a tube-side fouling factor of 0.1 m^2K/kW and a fin-side heat transfer coefficient derived from wind-tunnel tests (fig.1, rightmost side) of the fin geometry used for the actual heat exchanger (Lozza and Merlo, 2001).
The procedure is rather indirect, therefore cumulative measurement errors may lead to a rather large uncertainty on the final value of the heat transfer coefficient (about 10%), but, to a large extent, they do not affect the comparative results between smooth and micro-fin tubes.
Pressure losses are also measured. However we measure a differential pressure at the headers inlet and outlet, including the losses from: (i) the straight part of the tubes, (ii) the bends, (iii) the two

headers. The tube type only affects the first source of loss: to derive indications useful for general predictions, it was necessary to "separate" the tubes losses, by an empirical prediction of the bends and headers losses. It introduces again some uncertainties in the obtained values, but it doesn't affect the comparison between smooth and microfin tubes, using the same bends and headers.

Figure 1: LU-VE Contardo Laboratory.

1.4. Results obtained

Results are expressed by means of non-dimensional numbers (*Nu*, *Re*, *Pr*, friction factor *f*) by using the thermo-physical properties of the 34% ethylene glycol. In our tests, conducted at room temperature of about 0°C and glycol inlet temperature of about –10°C, the Prandtl number was about 60. We investigated Reynolds numbers in the range of 1000-5000, by varying the solution mass flow. Non-dimensional numbers are used for the following reasons: (i) raw data obtained should be adjusted for slight variations of the test conditions (i.e. air-fluid temperature difference, glycol average temperature, bringing about different thermophysical properties), (ii) non-dimensional correlations are introduced in coil rating calculations, to extend the validity of the results to other single-phase fluids or to other fluid conditions.

Fig.2 shows the experimental results relative to a number of test runs on two coolers, exactly identical except for the type of tube used. It can be seen that, for micro-fin tubes, advantages are found in terms of heat transfer capabilities, with respect to the smooth ones, at Reynolds exceeding 2500-3000, while in laminar flow no (or negligible) improvements are encountered. At fully developed turbulent flows, *Re*>5000, a 40÷50% heat transfer augmentation was estimated. As far as the pressure losses are concerned, we can basically say that no significant differences were found between the two tubes. A somewhat larger pressure loss occurs at the same flow rate, due to a smaller cross area (thickness of the groves). However it must be said that in laminar flow (low velocities) pressure loss are very small and measurements become less accurate; into addition the empirical estimation of the headers pressure loss may be affected by large approximations.

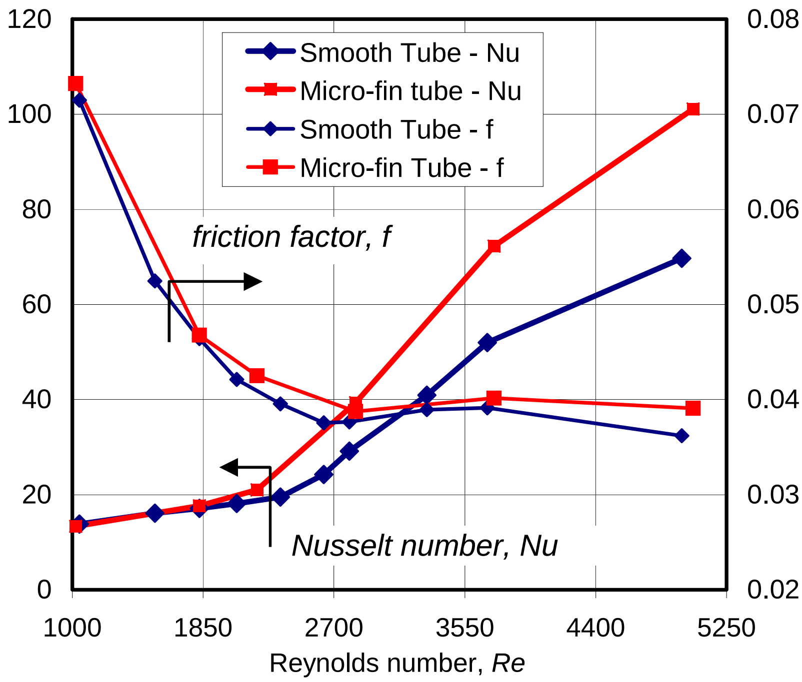

Figure 2: Nusselt number and friction factor vs. Re.

1.5. Applications

We estimated the variation of the capacity of typical aircoolers, as a consequence of the adoption of microfin tubes. We used our rating prediction code, whereby empirical correlations for heat transfer and pressure loss augmentation of microfin tubes were introduced, as from fig.2. The results are reported in tab.2, for three Reynolds numbers possibly encountered in real applications, for SC11 conditions (room temperature 0°C, ethylene glycol 34% solution, entrance temperature –10°C). The capacity improvement is very small at low *Re*, then increases significantly up to 7%, but for turbulent flow no further gains are obtained. In fact, even if the heat transfer enhancement is larger at elevated *Re* (fig.2), the tube-side heat transfer coefficient is large enough to have a modest influence on the overall performance, because most of the heat resistance is now concentrated on the air-side (fin-side heat transfer coefficient, referred to the tube surface, inclusive of the fin efficiency and of the contact resistance – Lozza and Merlo, 2001 – is about 900 W/m^2-K, for the 55x27.5 louvered geometry here considered, with a fin pitch of 6 mm, at an air velocity of 3.7 m/s).
As a conclusion, it can be said that significant performance improvements can be obtained in most operating conditions, without detrimental effects, apart from a moderate rise of the tubes cost.

Table 2: Relative capacity, tube-side heat transfer and pressure loss of a typical aircooler with smooth or microfin tubes, operating with ethylene glycol 34% solution at three Reynolds numbers, obtained by varying the solution flow rate.

	Capacity, relative to smooth case at Re=1800			Tube side heat transfer coefficient, W/m^2K			Liquid pressure loss, kPa		
Re	microfin	smooth	var %	microfin	smooth	var %	microfin	smooth	var %
1800	102.4	100.0	2.4%	551	517	6.6%	35.2	33.6	4.8%
2500	155.7	146.4	6.3%	1594	1238	28.8%	65.6	62.7	4.6%
3500	179.4	168.2	6.7%	2567	1815	41.4%	109	104	4.8%

2. AIR COOLED HEAT EXCHANGERS FOR CO_2 CYCLES

2.1. General remarks

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. 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 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. However, it is well known that the indirect effect must be considered: if the CO_2 refrigeration cycles were less efficient than traditional ones (lower COP), larger electricity consumption would bring about larger emissions of CO_2 and of other pollutants from power stations. The appropriate choice of the heat exchanger technology is a fundamental condition for obtaining COP values from CO_2 cycles allowing for a real reduction of the greenhouse effect. CO_2 is significantly different from all other refrigeratants, posing peculiar problems to heat exchanger designers: their discussion is the subject of this section.

2.2. CO_2 heat exchangers

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

- Evaporators, which are included in all possible plant configurations: 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 the ones for halogenated fluids.

- Gas-coolers, which are included in direct cycles 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. With a critical pressure of 73.8 bar, operating pressures much larger than those of conventional cycles will be adopted.

Supercritical cycles performances 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 entrance of the expansion device. This is very important to obtain an 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 performances are concerned.

2.3. 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. Rather than oversizing the evaporator and the refrigerant lines, it is preferable to adopt expedients to limit the design pressure to 60 bar or even less (safety valves, pump-down to remove liquid from the evaporator). Such pressure values are not dramatically higher than those normally used in refrigeration (e.g. 30-35 bar) and do not impose any special design, even if larger thickness of tubes and headers are usually adopted (e.g. from 0.35 to 0.5 mm for 3/8” copper tubes).

It is interesting to determine if an evaporator designed for conventional refrigerants can operate correctly for CO_2, with no or limited modifications. 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 lower pressure loss 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. Therefore a reduced number of parallel feedings must be adopted (about one half). Table 3 shows that a 10-15% larger capacity may be obtained by the same evaporator when running on CO_2 rather than on R404A, with optimized number of feedings. Into addition, the utilization of microfin tubes is rather questionable with CO_2, having elevated heat transfer capabilities.

Table 3: Comparative performance of an unit coolers with R404A and CO_2. SC2 implies a room temperature of 0°C and –8°C evaporation for SC4 figures are –25°C and –31°C.

fluid	R404A	CO_2	CO_2
type of tube	microfin	microfin	smooth
SC2 rel. capacity	100.0	110.6	108.2
SC4 rel. capacity	100.0	117.7	112.0

The specific cost (€/kW) of CO_2 evaporators is not really different from the one of conventional coolers. Lu-Ve and associated companies have 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.

2.4. 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 3 shows this situation very clearly: 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 power. This provides notable advantages in terms of reduced front area, of electricity required for ventilation and of the initial cost of the fans and of their regulators.

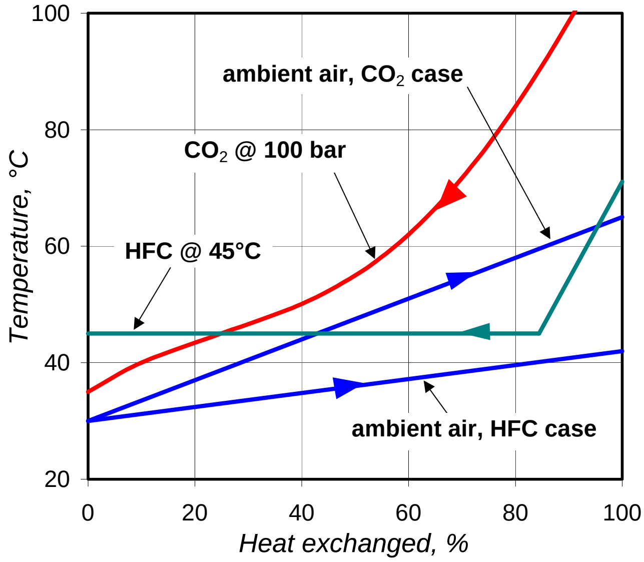

Figure 3: Heat transfer for a CO_2 gas cooler and for condensation of a conventional refrigerant.

To quantify these statements, a calculation method was developed capable of accounting for the particular distribution of the ΔTs between CO_2 and air (fig.3), provided that flows are arranged counter-current. The heat exchanger is divided 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 correlation for single phase flows from Gnielinski (1976). Figure 4 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 4 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) equal power and different final temperatures, (ii) at a final ΔT of 3 K, varying the power. The solution considered uses a fin geometry of 25 x 21.65 mm, with fin spacing of 2.1mm and 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.4). The number of parallel feedings is optimized in all cases. The following solutions are proposed in table 4**:**

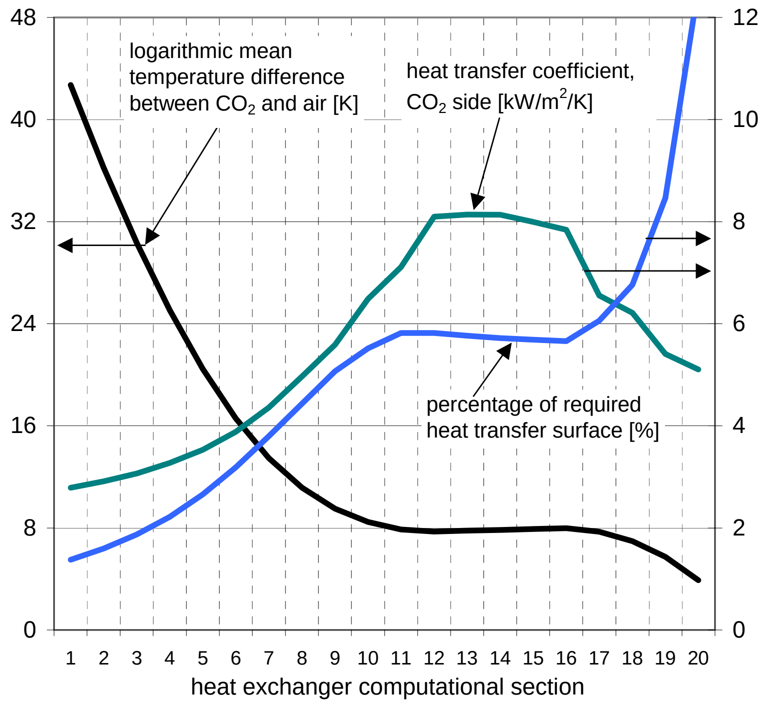

Figure 4: Variations of some parameters in the computational sections of a CO_2 gas cooler

- The first solution is the R404A reference (in normal production).
- The second solution presents the same fin pack 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). This is caused by the very large ΔT between CO_2 and air, at equal airflow.

- The third solution thoroughly exploits the above-mentioned (fig.3) possibility of reducing the airflow, using only one fan instead of three. The heat 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).

Table 4: Comparative performances of air cooled condensers with R404A and CO_2 at a pressure of 100 bar.

fluid	R404A	CO_2	CO_2
number of fans (8 poles)	3	3	1
coil front 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 (saturated)	25.3°C (ΔT=0.3)	28.8°C (ΔT=3.8)
or: (relative) thermal rating, at cooler outlet temp. 28°C	100 (ΔT=15)	158 (ΔT=3)	96.0 (ΔT=3)

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, 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 (end 2005: their number is rapidly increasing…), including large units (fig.5). Some particular manufacturing solutions were defined: (i) the fin pack is properly interrupted to allow for different thermal expansion and to avoid thermal conduction along the fin thickness: in fact, a large ΔT occurs in gas coolers (es: 120°->20°C), much higher than in condensers; (ii) the pressure test is carried out in four steps: (1) with air at 30 bar in a water pool to detect major leaks, (2) with water at 170 bar, (3) again with air at 30 bar to detect leaks caused by the previous pressurization; (4) the coil is de-hydrated by vacuum pumping to a pressure of 2 mbar.

Fig.5: A large CO_2 gas cooler with water-spray ready for shipping at LU-VE workshop.

The CO_2 gas cooler product can therefore be considered “proven technology”. This achievement was made possible because of the company’s design strategy, consisting of the utilization of high performance heat transfer surfaces and of miniaturized geometries (small diameter tubes) even for large heat exchangers. This is now precious for CO_2 applications, without the need of resorting to ‘exotic’ (and unproven!) technologies, such as aluminium heat exchanger with micro-channels sometimes proposed for automotive air-conditiong, hardly applicable to the refrigeration field, which requires much larger units and does not allow for large scale production.

2.5. Water spray

Water spray is another feature developed by the authors’company 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 reduce the condensation temperature, or, in the CO_2 case, the gas cooler outlet temperature, which dramatically affects the cycle performance. The water injection system is clearly visible in fig.5. 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. Two systems were developed: 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.

3. 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). The fin-and-tube geometries used for conventional fluids are perfectly adequate to CO_2 application, provided that small diameter tubes are used 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).

As far as air coolers operating with low freezing solutions at low Reynolds numbers, the utilization of microfin tubes brings about moderate advantages, i.e a 5-7% capacity improvements in the flow regimes more frequently used in applications.

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