The utilization of microfin tubes in air coolers with low freezing solutions

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Introduction

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 and it is distributed in the liquid phase to the various cold users (cold rooms, cabinets, etc.). With respect to the traditional solution, consisting of the distribution of halogenated refrigerants from the machine room to the cold users, it offers some advantages and some drawbacks. The advantages are:

- The refrigeration cycle and its operating fluids are strictly contained within the machine room. The possibility of leakages is much lower and they are easily detected. Much lower refrigerant charges are necessary, or, even better, it is possible to safely utilize hazardous refrigerants like ammonia or hydrocarbons, as an alternative to greenhouse gases like the halogenated refrigerants HFC.
- The distribution of the cold energy to the various users is very simple, not requiring expansion devices located on the evaporators to control the outlet quality of the gas, but simply distributing a liquid circulated by a pump.
- The air cooler can be operated at a very low ΔT, while for evaporators a minimum ΔT of 6-8 K is required to superheat the vapour, to drive the expansion device.

The first reason is probably the most important in recent realizations, overwhelming the ‘hystorical’ drawbacks of the low freezing solutions:

- More hardware is needed, i.e. a refrigerant / solution heat exchanger (evaporator) in the machine room and a pump to the circulate a consistent solution flow rate.
- Some additional energy requirement, due to the above mentioned pump and to the double heat transfer from the refrigeration cycle to the cool air (via the solution), requiring a double ΔT. This can be minimized according to the third point depicted above.
- The thermophysical properties of glycol solutions are not favourable to the achievement of elevated heat transfer coefficient in the air cooler: therefore, larger coolers are needed, negatively impacting on the economics (investment cost), or larger ΔT occur, negatively impacting the thermodynamics and the economics again (energy costs).

The purpose of this paper is to discuss the adoption of microfin tubes in air coolers, as a measure to improve the heat transfer in such heat exchangers and to remove, as far as possible, the drawback outlined by the latter point. Reference is made to fin-and-tube air coolers of medium-elevated capacity produced by LuVe Contardo (a major european manufacturer of heat exchangers for refrigeration applications), now utilizing a particular type of micro-fin tube developed in cooperation with a well-known copper tube supplier (Trefimetaux).

Background and experience

In the recent past, LuVe had successfully introduced advanced heat transfer surfaces in a typically conservative market, like the one of heat transfer equipments for refrigeration. Starting from 1986, pioneer studies were undertaken to develop high performance fin geometries, characterized by louvered or wave 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. Those efforts passed to the industrial production in 1988-1992, firstly applied to evaporators (unit coolers) and then to air-cooled condensers. Therefore LuVe has an unbeaten experience in using advanced heat transfer surfaces, compared to other manufacturers which adopted micro-fin tubes (now a generalized solution) in very recent times.

Micro-fin 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 same (qualitative) results may be simply obtained by a larger liquid velocity within the tubes: therefore, micro-fin tubes (relatively expensive) should offer a better result than adopting higher velocity (inexpensive!).

The situation is pretty complicated, because the relative improvement of both heat transfer and pressure loss depends on the Reynolds number *Re*. Just to give some example, in a fully developed turbulent flow (Re>5000) in a straight pipe, the heat transfer coefficient (h) depends on $Re^{0.8}$, while pressure loss on $Re^{1.8}$ (approximately): to obtain, for instance, a 50% improvement in h, velocity should increase by $1.5^{1/0.8}$=1.66, bringing about $1.66^{1.8}$=2.49 times larger pressure loss (Δp). Therefore a micro-fin tube offering 50% heat transfer improvement can be considered only if its pressure loss per meter is less than 2.49 the one of a straight tube, a relatively simple situation to be achieved. However, when we go to laminar or transition flow (Re<3000÷3500) the situation becomes less clear, because if we stay within the laminar flow (Re<2000) h depends on $Re^{1/3}$ and Δp on *Re* (therefore a large Δp occurs for a modest improvement of h), but, adopting a larger velocity, one may move from laminar to turbolent achieving a much larger improvement of the heat transfer. Since most refrigeration applications for low freezing solutions actually operate in laminar or transition regimes, 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 tube geometry selected by Trefimetaux presents the geometric characteristics shown in tab.1. Its behaviour with single-phase fluids was determined by means of the experimental investigation described below. The selected test method was to compare the performance of two air coolers having the very same characteristics, reported in tab.2, 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), therefore providing a reliable indication of the real advantages obtainable.

Tab.1: Geometry of the investigated tube [mm].

Outer diameter	12.7
Minimum thickness	xx
Groove height	xx
Number of grooves	xx
Helical angle	25°
Inner/outer surface	

[1] This represents a fundamental design philosophy, tending to apply small diameter tubes and very compact fin geometries even for large capacities: LuVe production makes use of ½” (12.7 mm) tubes for large evaporators up to 200 kW (ΔT_1=8K) and, mostly, of 3/8” (9.52 mm) tubes for condensers and dry-coolers exceeding 1000 kW (ΔT_1=15K), with coil lengths up to 12800 mm. The rationale behind this philosophy is to increase the convective heat transfer coefficients (proportional to $D^{-0.55÷0.65}$ 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).

The experimental apparatus

Tab.2: Characteristics of the aircoolers.

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

The capacity of the two air coolers of tab.2 was measured by means of a calibrated cold room in the LuVe laboratories in Uboldo (Va, Italy), schematically shown in fig.1. 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[2] of 0.1 m^2K/kW and a fin-side heat transfer coefficient derived from wind-tunnel tests[3] of the fin geometry used for the actual heat exchanger. 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, therefore 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 pressure loss: to derive indications useful for general pressure loss prediction, 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.

Results are therefore expressed by means of non-dimensional numbers (*Nu, Re, Pr,* friction factor *f*) by using the thermo-physical properties of the 34% ethylene glycol – water solution, as reported by [xx]. 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.

figure possibili> apparecchio (fig.1), cella (fig.2), figura schema della cella (fig.3 Merlo?)

[2] In our experience, a better matching of the experimental results with well-know single-phase correlations (i.e. Gnielinski [xx]) is obtained if a fouling factor is introduced even for closed circuit applications, probably due to some dirt always present in common water circuits with steel tubes.

[3] The experimental apparatus and the procedure for testing fin geometries is described in [xx].

Results obtained

Fig.4 shows the experimental heat transfer results, expressed in terms of Colburn factor j, defined as $j = Nu \cdot Re^{-1} \cdot Pr^{-1/3}$, as a function of the Reynolds number Re, together with the prediction formulated at Pr =60 with the Sieder-Tate correlation [ashrae] in laminar regime (Re<2000) and the Gnielinski correlation [xx] for turbulent flow (Re>2300), as used in our computational tools for smooth tubes. It can be seen that:

- For smooth tubes, the agreement between correlations and experimental data is pretty good for laminar flow, tends to be good again at Re>5000, but in the transition zone the Gnielinski correlation, for the Pr considered, seems to overestimate the Colburn factor.
- For micro-fin tubes, advantages are found, with respect to the smooth ones, at Reynolds exceeding 3000, while in laminar flow no (or negligible) improvements are encountered. At *Re* >5000, a 40÷50% heat transfer augmentation was estimated.

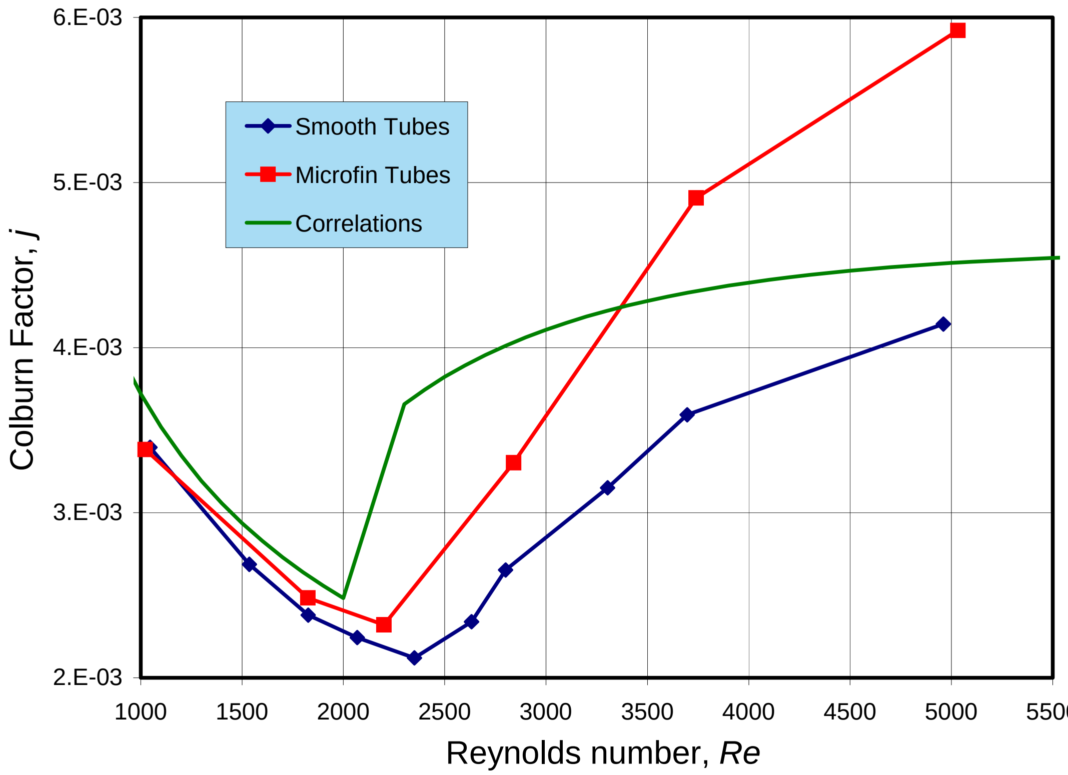

Fig.4: Colburn factor as a function of Re: experimental results vs. literature correlations.

For the pressure losses, the situation is shown in fig.5 in terms of friction factor f. 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) and therefore higher velocity, but it is practically the same at the same velocity. The comparison with simple pressure loss correlations (f=64/Re in laminar flow, $f=0.24/Re^{0.22}$ in turbulent flow) shows a good agreement in turbulent flow, while experimental values are significantly higher in laminar flow. 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.

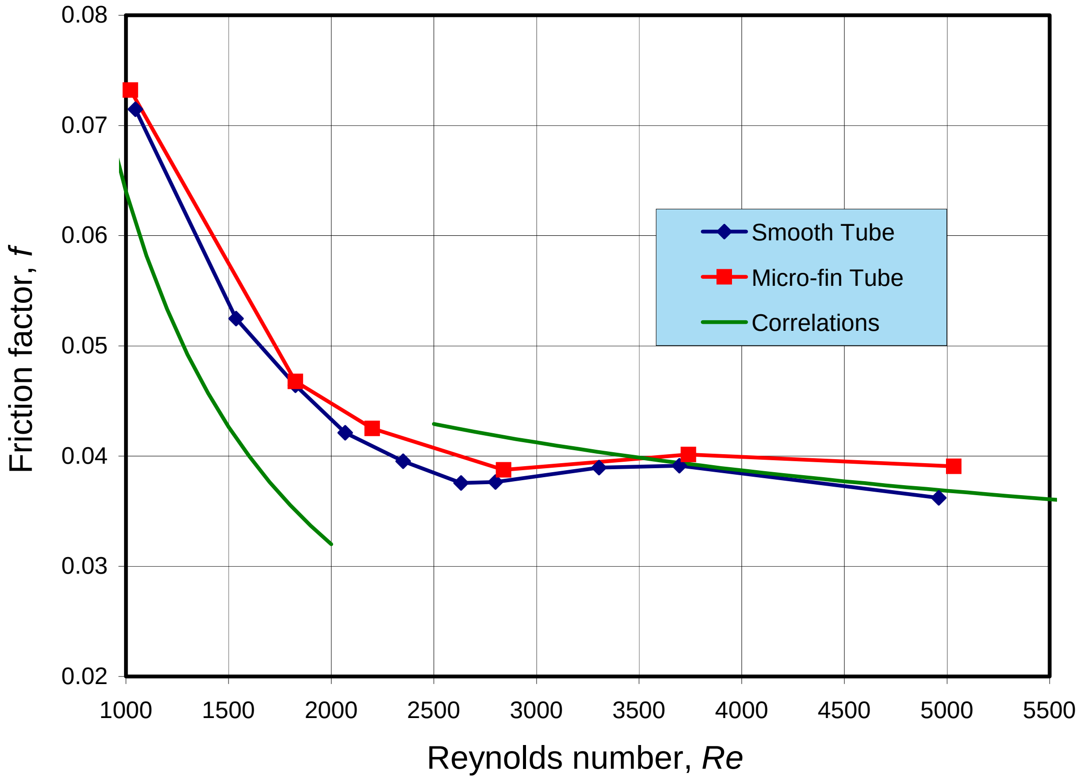

Fig.5: Friction factor as a function of Re: experimental results vs. literature correlations.

CFD studies

The experimental results are, under certain points of view, a little surprising. For instance, one may ask: (i) why pressure losses are unaffected, in spite of a much larger wetted inner surface, (ii) why heat transfer is improved only for turbulent flow? To try a better understanding of the physical phenomena occurring within the grooved tubes, an analysis was carried out by Computational Fluid Dynamics methods (Fluent™).

Applications

Conclusions

[1] Gnielinski V., New equations for heat and mass transfer in turbolent pipe and channel flows, International Chemical Engineering 16 (1976) 359-368