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Numerical Investigation of Boiling and Thermal Characteristics of Micro-Fin Tubes for Alternative Refrigerants with Varying Fin Height and Fin Material

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Abstract— Micro-fin (micro-groove) tubes enhance in-tube evaporation and condensation by suppressing boundary-layer growth and enlarging the wetted surface area, and they now dominate residential HVAC and automotive heat-exchanger practice. This paper reports a numerical study of the boiling and thermal characteristics of a horizontal micro-fin tube of 100 mm inner and 115 mm outer diameter, carrying four refrigerants — R22, R134a, R407C and R410A — at two fin heights, 5 mm and 3 mm. Three-dimensional models were generated in a parametric CAD environment and analysed in ANSYS 15; a pressure-based CFD solution in FLUENT gave the pressure field, velocity field, convective heat transfer coefficient and total heat transfer rate, while a steady-state thermal solution using the peak convective coefficient from the CFD stage gave the surface heat flux for aluminium, copper and titanium alloy fins. Reducing the fin height from 5 mm to 3 mm raised the heat transfer coefficient by 34.89 % for R22, 139.11 % for R134a, 58.41 % for R407C and 43.62 % for R410A, and raised the total heat transfer rate by 57.78 %, 39.47 %, 7.95 % and 9.72 % respectively. The associated pressure penalty was modest and, for three of the four fluids, negative: the pressure rose by 6.41 % for R22 but fell by 0.92 %, 6.42 % and 5.22 % for R134a, R407C and R410A. Surface heat flux for the 3 mm geometry was higher by 52.01 %, 27.49 % and 30.40 % for aluminium, copper and titanium alloys respectively, with copper alloy giving the highest absolute flux. R410A at 3 mm fin height with copper alloy fins is therefore identified as the best-performing combination examined.
Keywords— *Micro-fin tube; boiling heat transfer; CFD; refrigerants; fin height; heat flux; ANSYS*

1. INTRODUCTION

Conventional energy resources are depleting at a rate that makes the sustainable operation of thermal plant increasingly difficult, and a large share of the effort to contain that decline has been directed at the augmentation of heat transfer surfaces. Augmentation techniques are conventionally grouped into three families. *Active* techniques require a direct input of external power — surface vibration, electrostatic fields, injection or suction. *Passive* techniques require none; they use geometrical or surface modification of the flow channel, or an insert placed within it. *Compound* techniques combine two or more of the above.

Passive methods have attracted the greater share of research attention because they are simple, they can be retrofitted into an existing smooth-tube exchanger, they preserve the mechanical strength of the parent tube, and they can be withdrawn for cleaning. Within the passive family the principal sub-classes are treated surfaces, rough surfaces, extended (finned) surfaces, displaced enhancement devices, swirl-flow devices, coiled tubes, surface-tension devices and additives. Extended surfaces are the most widely commercialised: they enlarge the effective heat transfer area and, in their modern forms, simultaneously disturb the flow field so that the convective coefficient itself is raised rather than merely the area on which it acts.

The micro-fin tube is the limiting case of this idea applied to the tube interior. Helical or axial grooves of sub-millimetre height are formed by displacing material from the internal wall, so that the tube retains its plain external profile while the internal surface is corrugated. Two mechanisms then act together. First, the repeated interruption of the developing thermal boundary layer prevents

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the layer from thickening along the tube, which holds the local convective coefficient near its entrance value over a much greater length. Second, the internal wetted area available to the refrigerant increases without any increase in tube envelope. Because the enhancement is obtained inside a tube whose outside diameter is unchanged, micro-fin tubes deliver a high heat transfer coefficient at a low pressure-drop penalty, they permit a reduction in the mass of copper required to build a given condenser or evaporator, and — because the tube bore is smaller — they reduce the refrigerant charge that the equipment must carry. The last point has become decisive as the industry has moved away from high-GWP fluids and as charge-minimisation has become a design objective in its own right.

The choice of working fluid interacts strongly with the choice of internal geometry. R22, the hydrochlorofluorocarbon that dominated air-conditioning practice for four decades, is being displaced under the Montreal Protocol by R134a, by the near-azeotropic blend R410A and by the zeotropic blend R407C, each of which has a different density, specific heat, thermal conductivity and viscosity and therefore a different response to a given fin geometry. A geometry optimised for one fluid is not necessarily optimal for another.

The present work therefore addresses two coupled design questions. First, for a fixed tube envelope, which of two candidate fin heights — 5 mm or 3 mm — gives the better balance of heat transfer enhancement against pressure penalty, and does the answer depend on the refrigerant? Second, for the better geometry, which of three candidate fin materials — aluminium alloy, copper alloy or titanium alloy — sustains the highest surface heat flux? The questions are answered by a two-stage numerical study: a CFD solution for the boiling characteristics of each refrigerant in each geometry, followed by a steady-state thermal solution for each fin material driven by the convective coefficient obtained from the first stage.

2. LITERATURE BACKGROUND

Samuel and Julien predicted air-to-refrigerant two-phase flow boiling for alternatives to HCFC-22 inside enhanced-surface tubing and established that correlations developed for smooth tubes systematically under-predict the coefficients obtained on enhanced surfaces [1]. Brown and co-workers subsequently formulated the enhancement problem in exergetic terms, treating the pressure drop as an exergy loss to be traded against the exergy gain from the improved coefficient, which is the framework adopted implicitly in the present comparison [2].

For the specific fluids studied here, Torrella et al. measured the variation of the boiling heat transfer coefficient for R407C inside the horizontal tubes of a shell-and-tube evaporator on an operating vapour-compression plant, and showed that the zeotropic glide of the blend produces a coefficient that varies markedly along the tube [3]. Mongey, Hewitt and McMullan assessed R407C as a direct replacement for R22 and reported the capacity and efficiency penalties that follow from the substitution [7], while Aprea and Greco evaluated the greenhouse consequences of the same substitution experimentally [8]. The regulatory background to these substitutions was set out by Manzer [4] and revisited by Wuebbles and Calm [6].

Work directly on micro-fin geometry begins with Kaul, Kedzierski and Didion, who measured horizontal flow boiling of alternative refrigerants inside a fluid-heated micro-fin tube and quantified the enhancement relative to a smooth tube of the same bore [5]. He et al. extended the fluid set to near-azeotropic mixtures such as R290/R32 in horizontal tubes [9]. More recently Lin et al. reviewed two-phase flow heat transfer in micro-fin tubes across the available database [11], and Vidhyarthi and co-workers produced a comprehensive assessment of two-phase flow boiling in micro-fin tubes using both pure and blended eco-friendly refrigerants [10]. The consistent finding across this literature is that enhancement ratios of two to three are achievable with pressure penalties well below those of tube inserts, but that the optimum groove geometry is fluid-dependent — which is precisely the gap the present parametric study addresses.

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3. GEOMETRY AND NUMERICAL METHOD

3.1 Tube geometry

Two three-dimensional models of the micro-fin tube were generated. Both share an inner diameter of 100 mm and an outer diameter of 115 mm; the models differ only in the depth to which material is removed to form the internal fin. Each fin has a cross-section of 2 mm × 2.20 mm and the fins are distributed around the bore at an included angle of 102.5°. The models were built by sketching the two bounding circles on a datum plane, extruding to form the annulus, sketching the fin profile and removing material by an extruded cut to a depth of 5 mm in the first model and 3 mm in the second. Both models were exported in IGES format for import into the analysis environment.

3.2 CFD model for boiling characteristics

The imported geometry was meshed in ANSYS Workbench and named selections were created for the inlet and outlet faces. The mesh was exported to FLUENT, where a pressure-based solver was used with the energy equation enabled and a viscous model active. Each refrigerant was defined as a new fluid material using the properties listed in Table 1, the inlet boundary condition was imposed, hybrid initialisation was applied and the solution was advanced for 30 iterations to convergence. Post-processing gave contour plots of static pressure, velocity magnitude, surface heat transfer coefficient and total heat transfer rate.

Table 1. Thermophysical properties of the four refrigerants used in the CFD stage.

Property	R22	R134a	R407C	R410A
Density (kg/m³)	1409.2	1376.7	1380.7	1349.7
Specific heat, cp (kJ/kg·K)	1.090	1.281	0.787	1.370
Thermal conductivity (W/m·K)	113.5	103.9	127.9	151.3
Viscosity (Pa·s)	346.0	384.2	384.6	346.4

3.3 Governing equations

The solver advances the conservation equations for mass, momentum and energy in their steady, incompressible form. Continuity is expressed as

$$\partial(\rho u_i)/\partial x_i = 0$$

the momentum balance in the i-direction as

$$\partial(\rho u_j u_i)/\partial x_j = -\partial p/\partial x_i + \partial/\partial x_j \left[\mu \left(\partial u_i/\partial x_j + \partial u_j/\partial x_i\right) \right]$$

and the energy balance as

$$\partial(\rho u_i\, c_p\, T)/\partial x_i = \partial/\partial x_i \left(k\, \partial T/\partial x_i\right)$$

where ρ is density, u the velocity component, p the static pressure, μ the dynamic viscosity, c_p the specific heat at constant pressure, k the thermal conductivity and T the static temperature. The surface heat transfer coefficient reported in Section 4 is recovered from the converged wall heat flux q and the wall-to-bulk temperature difference as $h = q / (Tw - Tb)$, where Tw is the wall temperature and Tb the bulk fluid temperature; the total heat transfer rate is the integral of q over the wetted internal area. In the second stage the steady conduction equation for the solid is solved with a convective boundary condition of the form $q = h\,(Ts - Tf)$ applied to the outer surface, Ts being the surface temperature and Tf the film reference temperature.

3.4 Thermal model for heat flux

The second stage used a steady-state thermal analysis on the same two geometries. For each fin height the largest convective heat transfer coefficient found in the CFD stage was applied as the film coefficient on the outer surface: 5.8×10^{-2} W/m²K (R410A) for the 5 mm model and 8.56×10^{-2} W/m²K (R134a) for the 3 mm model. A fixed temperature was imposed on the fin surfaces and the mesh was refined from the default to a fine setting. The analysis was repeated for three fin materials whose properties are given in Table 2, and the resulting temperature and heat-flux fields were extracted.

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Table 2. Properties of the three candidate fin materials.

Material	Density (kg/m³)	Thermal conductivity (W/m·°C)	Young's modulus (MPa)	Poisson's ratio
Aluminium alloy	2770	144	—	—
Copper alloy	8300	401	110 000	0.34
Titanium alloy	4620	21.9	96 000	0.36

4. RESULTS AND DISCUSSION

4.1 Boiling characteristics

Tables 3 and 4 collect the converged CFD results for the 5 mm and 3 mm geometries respectively. For the 5 mm tube the peak static pressure lies between 1.185×10^5 Pa and 1.199×10^5 Pa across the four fluids and the peak velocity between 7.285 m/s and 7.729 m/s, a narrow band that reflects the similar densities of the four refrigerants. The heat transfer coefficient, by contrast, varies by a factor of 1.6 across the fluid set, from 3.58×10^{-2} W/m²K for R134a to 5.80×10^{-2} W/m²K for R410A, and the total heat transfer rate varies by more than a factor of two, from 2416 W for R407C to 5104 W for R410A.

Table 3. Boiling characteristics, fin height 5 mm.

Parameter	R22	R134a	R407C	R410A
Pressure (Pa)	1.185E+05	1.199E+05	1.199E+05	1.188E+05
Velocity (m/s)	7.651E+00	7.294E+00	7.285E+00	7.729E+00
Heat transfer coefficient (W/m²K)	4.70E−02	3.58E−02	5.29E−02	5.80E−02
Total heat transfer rate (W)	2672.07	2999.99	2416.16	5103.85

Table 4. Boiling characteristics, fin height 3 mm.

Parameter	R22	R134a	R407C	R410A
Pressure (Pa)	1.261E+05	1.188E+05	1.122E+05	1.126E+05
Velocity (m/s)	1.163E+01	9.299E+00	9.500E+00	9.739E+00
Heat transfer coefficient (W/m²K)	6.34E−02	8.56E−02	8.38E−02	8.33E−02
Total heat transfer rate (W)	4215.98	4184.04	2608.20	5600.08

Reducing the fin height to 3 mm changes the picture in two ways. The velocity field intensifies substantially — the peak velocity rises by between 26 % and 52 % — because the shallower groove leaves a larger free-flow area at the tube wall and the fluid is accelerated over the fin crests rather than being trapped between them. The heat transfer coefficient rises accordingly, and it rises furthest for the fluid that performed worst at 5 mm. Table 5 sets out the percentage differences.

Table 5. Percentage change in boiling characteristics on reducing fin height from 5 mm to 3 mm.

Parameter	R22	R134a	R407C	R410A
Pressure	+6.41	−0.92	−6.42	−5.22
Velocity	+52.01	+27.49	+30.40	+26.01
Heat transfer coefficient	+34.89	+139.11	+58.41	+43.62
Total heat transfer rate	+57.78	+39.47	+7.95	+9.72

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The 139.11 % gain recorded for R134a is the single largest effect in the study. R134a has the lowest thermal conductivity of the four fluids (103.9 W/m·K) and the second-highest viscosity, so at 5 mm fin height its performance is limited by a thick, poorly conducting film held in the deep grooves; removing 2 mm of groove depth releases that limitation. R410A, which has the highest thermal conductivity (151.3 W/m·K) and the highest specific heat, is already performing near the geometric limit at 5 mm and gains proportionally less, although it retains the highest absolute heat transfer rate in both geometries at 5104 W and 5600 W.
The pressure result is the practically important one. Enhancement techniques are usually rejected on pumping-power grounds, yet here the 3 mm geometry raises the pressure by only 6.41 % for R22 and *lowers* it for the other three fluids, by up to 6.42 % for R407C. The shallower fin therefore delivers a large gain in heat transfer at essentially no hydraulic cost, which is the outcome that makes the geometry attractive for retrofit into existing condenser and evaporator circuits where fan and compressor sizing is already fixed.

4.2 Thermal characteristics

The steady-state thermal results are given in Table 6. The peak surface temperature is 121.11 °C in every case, which is expected because the imposed boundary temperature is common to all runs and the tube is thin relative to its diameter; the discriminating quantity is the heat flux.
Table 6. Surface heat flux (W/m²) and peak temperature for the three fin materials.

Quantity	Fin height 5 mm			Fin height 3 mm		
	Al alloy	Cu alloy	Ti alloy	Al alloy	Cu alloy	Ti alloy
Temperature (°C)	121.11	121.11	121.11	121.11	121.11	121.11
Heat flux (W/m²)	17.459	17.460	17.450	37.906	37.910	37.869
Increase (%)	—	—	—	+52.01	+27.49	+30.40

The heat flux for the 3 mm geometry is more than double that of the 5 mm geometry for every material, an increase of 52.01 % for aluminium alloy, 27.49 % for copper alloy and 30.40 % for titanium alloy relative to their respective 5 mm values. Comparing materials at fixed geometry, copper alloy gives the highest flux — 17.460 W/m² at 5 mm and 37.910 W/m² at 3 mm — followed within a fraction of a percent by aluminium alloy, with titanium alloy lowest. The three materials are separated by less than 0.15 %, which is a much smaller spread than their thermal conductivities (401, 144 and 21.9 W/m·°C) would suggest. This is the signature of a convection-controlled problem: with a film coefficient of order 10^{-2} W/m²K the external thermal resistance dominates so completely that the conduction resistance of the fin material is almost irrelevant to the flux. The practical reading is that copper alloy is confirmed as the preferred material on thermal grounds, but the margin over aluminium is negligible, so for this duty the selection can reasonably be made on cost, formability and corrosion grounds instead — an argument that favours aluminium for the fin stock and reserves copper for the tube itself.

4.3 Selection of the operating combination

Bringing the two stages together, the design choice can be framed as a ranking on three criteria: absolute heat duty, enhancement obtained per unit of pressure penalty, and material cost. On absolute duty, R410A at 3 mm fin height is the clear leader at 5600 W, some 34 % above the next fluid. On enhancement per unit penalty the ranking inverts: R407C and R410A both gain heat transfer while *losing* pressure, so their effective penalty is negative, whereas R22 pays 6.41 % more pressure for its 57.78 % gain in duty. R134a occupies an intermediate position, gaining the most in coefficient terms at essentially zero pressure cost, but from the lowest starting point.
For a new-build evaporator or condenser the recommendation is therefore unambiguous: R410A in a 3 mm micro-fin geometry. For a retrofit into plant originally charged with R22, the results indicate that the conversion to R407C — which is normally accepted with a capacity penalty — can be made close to capacity-neutral if the tube is changed to the 3 mm geometry at the same time,

because the 58.41 % gain in coefficient offsets much of the loss attributable to the fluid change. This is a useful result for the large installed base of R22 equipment now being converted.

5. CONCLUSIONS

- Reducing the internal fin height of a 100/115 mm micro-fin tube from 5 mm to 3 mm raises the boiling heat transfer coefficient for every refrigerant examined: by 34.89 % for R22, 139.11 % for R134a, 58.41 % for R407C and 43.62 % for R410A.
- The total heat transfer rate rises correspondingly by 57.78 %, 39.47 %, 7.95 % and 9.72 % for R22, R134a, R407C and R410A respectively, with R410A delivering the highest absolute rate of 5600 W at 3 mm fin height.
- The hydraulic penalty is negligible. Pressure rises by only 6.41 % for R22 and falls by 0.92 %, 6.42 % and 5.22 % for R134a, R407C and R410A, so the enhancement is obtained essentially free of additional pumping power.
- Surface heat flux at 3 mm fin height exceeds that at 5 mm by 52.01 %, 27.49 % and 30.40 % for aluminium, copper and titanium alloys. Copper alloy gives the highest flux in both geometries, but the three materials differ by less than 0.15 % because the duty is convection-controlled.
- The combination of 3 mm fin height, R410A and copper alloy fins is recommended for the geometry studied. Where cost dominates, aluminium alloy may be substituted with a thermal penalty of well under one per cent.

Micro-groove technology continues to move from residential and commercial air conditioning into large industrial refrigeration plant, supported by the manufacturing developments in tube-to-fin assembly that avoid crushing the internal fins during expansion. Extension of the present parametric approach to helical rather than axial grooves, and to the low-GWP fluids now entering service, is the natural continuation of this work.

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