Advances in Air-Cooled Heat Exchanger Technology for Residential Air-Conditioning

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Key words: Air-cooled heat exchanger, Residential, Vortex generator, Micro-fin tube, Flat tube, Condenser, Evaporator, Heat transfer mechanism, Predictive model

ABSTRACT: This paper describes the recent work on advanced technology concepts applied to air cooled heat exchangers for residential air-conditioning. The concepts include vortex generators for the air-side, micro-fin or flat tubes for the refrigerant-side. Advances in understanding of heat transfer mechanisms, predictive models are discussed.

Nomenclature

A	: surface area [m^2]
A_f	: frontal area [m^2]
A_o	: air-side surface area [m^2]
A_p	: plain tube surface area [m^2]
A_{tot}	: total surface area [m^2]
D	: diameter [m]
D_h	: hydraulic diameter [m]
D_i	: inside diameter [m]
D_o	: outside diameter [m]
dP/dz	: pressure gradient [Pa/m]
e	: rib height [m]
H	: heat exchange width, channel height [m]
h	: heat transfer coefficient [W/m^{2}K]
L	: flow length [m]
N	: number of microfins, dimensionless
p_f	: groove pitch normal to microfins [m]
p_n	: groove pitch normal to tube axis [m]
Re	: Reynolds number, defined where used, dimensionless
t_b	: fin base thickness [m]

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Greek symbols

α	: helix angle [degrees]
β	: fin apex angle [degrees]
η	: surface efficiency

1. Introduction

This paper addresses technology advances applicable to air-cooled heat exchangers used in residential air-conditioning. As will be shown, significant changes are underway in the design of the various heat exchanger types.

Residential air-conditioning evaporators and condensers typically use finned tube heat exchangers made with 7.0 or 9.5 mm diameter tubes expanded onto enhanced, aluminum fin geometries. Candidate designs for advanced technology include: (1) Smaller diameter round tubes and parallel flow circuiting, (2) Use of oval tubes, (3) Brazed aluminum heat exchangers with parallel flow circuiting, and a flat tube all copper condenser, which competes with the brazed aluminum design. Considerable work has been done on vortex generators for fin-tube heat exchangers, which is reviewed.

Air-cooled condensers and evaporators use special geometries on the refrigerant side. The second part of the paper reviews advances in the tube-side technology for condensers and evaporators.

2. Air-side fin geometries

2.1 Fin geometries for round tubes

Residential air-conditioning typically uses air-cooled evaporators and condensers. Finned tube heat exchangers have been used for many years. Over the past 25 years, there has been a trend toward use of smaller tube diameters.

Currently used tube diameters are in the range of 7.0-to-9.5 mm. A variety of enhanced fin geometries have been developed as illustrated in Fig. 1. Webb and Kim[1] surveys these fin geometries, and Kang and Webb[2] have compared the performance of the various fin geometries.

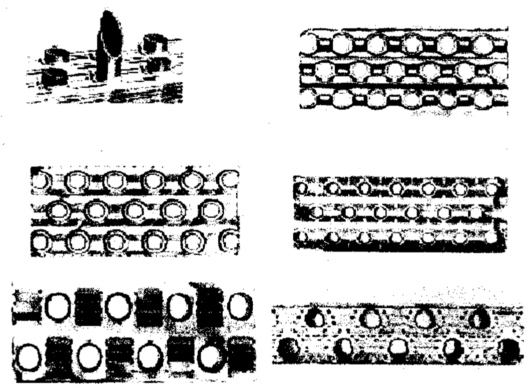


Fig. 1 Photos of enhanced fin geometries.

Kang and Webb conclude that the Fig. 1 convex louver fin provides the highest heat transfer performance (j-factor), with very good j/f ratio.

2.2 Vortex generators

There have been many recent investigations of "vortex generators" for application to plate-fin geometries. Figure 2 from Fiebig et al.[3] illustrates several types of vortex generators that have been investigated. A vortex generator is formed by cutting the metal fin and bending the metal upward to form an obstruction to the flow. These obstructions form a pair of stream-wise vortices that mix the flow in the wall region as illustrated in Fig. 2. The vortices are gradually dissipated and must be re-established as shown in Fig. 2. Vortex generators are primarily of benefit to laminar flow. The magnitude of the average enhancement over the surface is strongly dependent on the fraction of the surface that is "scoured" by the vortices. Hence, the enhancement will be increased if the area density of vortex generators is increased. Experimental data of Fiebig[4] shows that for 30° angle of attack, the "delta and rectangular winglets" give higher heat transfer and pressure drop than the "delta wings."

2.3 Vortex generators on circular finned tubes

In round tube heat exchangers, horseshoe vortices formed at the intersection of the tube

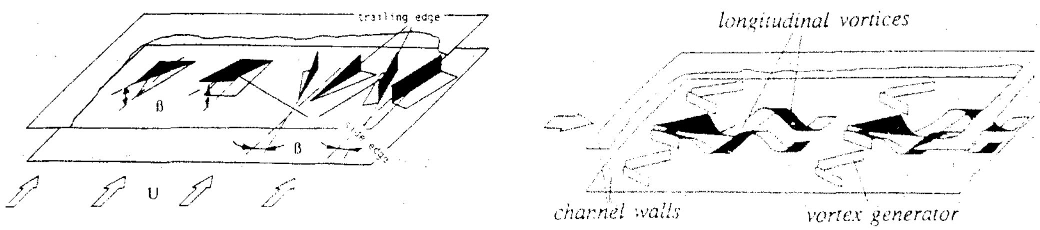
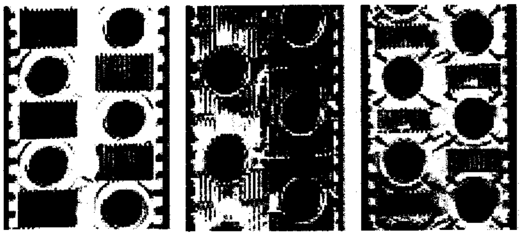


Fig. 2 Wing-type vortex generators: From left, Delta wing, Rectangular wing, Delta winglet, and Rectangular winglet pair. Right hand side figure illustrates vortices generated from vortex generators.

and the fin provide inherent enhancement on the fin. The optimum location of vortex generators on round fin-tube geometry was studied by Fiebig et al.[5] They used a winglet pair having '⌃' configuration with 45° angle of attack. An approximate optimum location was found just behind the tube with winglets one tube diameter apart. In the Reynolds number (channel spacing based) range 2,000 to 5,000, they obtained 20% heat transfer enhancement, relative to a tube with plain fins. The pressure drop was decreased as much as 10% compared with the case of no vortex generators. Fiebig et al.[3] extended the study to a 3-row heat exchanger geometry. The average heat transfer enhancement was 55% to 65% for the in-line tube arrangement, and 9% for the staggered arrangement. The corresponding pressure drop increase was 20% to 45%, and 3%, respectively.

Significant decrease of the pressure loss was achieved by Torii et al.[6] with vortex generators mounted in a '⌄' configuration. The vortex generators were installed behind the first row only for the three-row configuration. For the staggered geometry, the heat transfer enhancement was 10% to 30%. However, the pressure loss was reduced 34% to 55%. For the in-line geometry, the heat transfer enhancement was 10% to 20%, and the pressure loss reduction was 8% to 15%. Torii et al.[6] attributed the significant performance improvement to the separation delay, reduction of form drag, and removal of the poor heat transfer zone behind the tube.

Lozza and Merlo[7] tested two-row finned-tube heat exchangers having 2.0 mm fin pitch with various enhanced geometries and vortex generators. Figure 3 shows the louver fin geometries tested. All geometries had 2.0 mm fin pitch. Figure 3(b) may be considered as a high performance conventional louver fin geometry, which has a significantly greater fraction of the surface covered by louvers. The Fig. 3(b) surface provides 14% higher j-factor than does



(a) Louver fin fin A (b) Louver fin fin B (c) Louver fin with vortex generators

Fig. 3 Fin configurations tested by Lozza and Merlo.[7]

the Fig. 3(a) geometry. Figure 3(c) shows a louver fin geometry with winglet vortex generators 1.6 mm high. The Fig. 3(c) geometry provides approximately the same j-factor as the Fig. 3(a) geometry and 12% smaller j-factor than the Fig. 3(b) geometry. Thus, it appears that the addition of vortex generators to a louver fin geometry is not as good as allocating the same area to louvers. Further, the j/f ratio of the Fig. 3(c) geometry is not as good as the Fig. 3(b) geometry.

Lozza and Merlo[7] also tested 2-row plain fin geometries, with and without winglet vortex generators. The plain fin/vortex generator surface (like Fig. 3(c) without louvers) provided 50% j-factor increase and 60% friction increase compared to the plain fin geometry. This enhancement is greater than that found by Fiebig et al.[5] who used vortex generators only in the tube wake region.

Based on the above comparison, it does not appear that vortex generators provide greater enhancement than can be obtained from conventional slit or louver fin geometries, when applied to round tubes.

2.4 Vortex generators on flat or oval finned tubes

Valencia et al.[8] investigated the effect of vortex generator location for a single flat tube

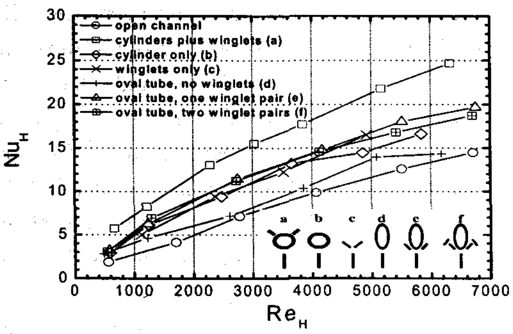


Fig. 4 Average fin surface Nusselt number for the circular and oval tubes with vortex generators (O'Brien et al.[9]).

geometry. The best performance (50% heat transfer enhancement and 36% pressure drop increase relative to a plain fin) was obtained when vortex generators are located upstream of the tube. This enhancement level is quite small compared to that of the louver fin geometry.

O'Brien et al.[9] investigated the effect of vortex generators on an oval tube heat exchanger. Their results are shown in Fig. 4. Addition of the single winglet pair to the oval tube geometry yielded 38% heat transfer enhancement with 10% pressure drop increase at Re=500. The highest heat transfer (highest pressure drop as well) was observed for the case of circular tube with winglets. Higher enhancement will be obtained from a conventional oval tube/louver fin geometry, as found by Webb and Iyengar,[10] as described in the next section.

2.5 Future advancements

Typically used finned-tube heat-exchanger designs have round tubes and serpentine refrigerant circuiting. Future advancements in finned tube design are expected to address the following possibilities and include parallel flow refrigerant circuiting:

(1) Smaller diameter round tubes (e.g., 4-to-6 mm)

(2) Oval tubes expanded on plate fins.

(3) Brazed aluminum (or copper) heat exchangers having flat tubes.

Smaller diameter tubes will result in lower air pressure drop. Oval tubes will also yield lower air pressure drop, and permit more efficient use of slit fins. Use of slit fins on round tubes results in lower fin efficiency and limits the extent of the slit area. However, use of the oval tube will allow a greater fraction of the fin surface to contain slit fins and the resulting fin efficiency will be higher than for a round tube.

Webb and Iyengar[10] compared the air-side performance of the 1,024 fins/m Fig. 5 oval tube geometry with that of a 2-row finned tube heat exchanger having 8.0 mm diameter round tubes and the Fig. 1 convex louver fins (CLF) at 807 fins/m, which was tested by Wang et al.[11] Operating at the same frontal velocity, the comparison shows that oval tube h-value is 10% higher and the ΔP is 17% lower than that of the 8.58 mm diameter round tube design. The oval tube design has approximately 20% greater surface area ratio (A/A_f) than the 807 fins/m convex louver fin design. Hence, the oval tube design offers 32% greater hA/A_{fr} (1.10×1.20) than the round tube design. The 32% greater hA/A_{fr} and 17% lower ΔP offers a significant performance improvement.

Min and Webb[12] performed CFD analysis to predict the heat transfer coefficient and pressure drop of an oval tube finned-tube heat ex-

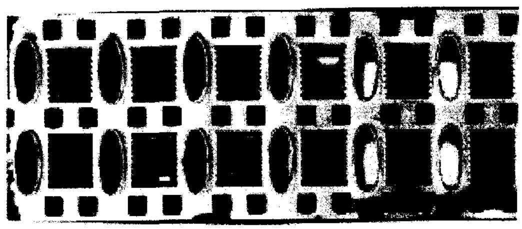


Fig. 5 Photo of fin used in the oval tube core using 8.0 mm major diameter tubes and 1.0 mm pitch louver fins.

changer having wavy fins (12 fins/in). The heat exchanger had a 3 : 1 aspect ratio oval tube and multiple rows of staggered tubes (30.5 transverse pitch and 31.8 mm row pitch). The performance was compared with the same wavy fin geometry using round tubes operating at 2.54 m/s frontal velocity. They found that the oval tube geometry gave 9.0% lower heat transfer coefficient and 49% lower pressure drop. Based on the CFD strength predictions of Webb and Iyengar[10] the oval tube geometry with 594 fins/m (15 fins/in) and 140 μm (0.0055 in) fin thickness should meet the strength requirements for condensation of R-22 and R-410A. A numerical study of Min and Webb[12] reports the effect of the aspect of the oval tube on the air-side heat transfer and friction.

3. Refrigerant-side geometries

3.1 Round micro-fin tubes

The performance of air-cooled evaporators and condensers having round tubes is improved using a special tube-side enhancement. The tube has internal micro-fins of triangular cross section that are approximately 0.25 mm high, 0.50 mm pitch at a helix angle of approximately 18 degrees. Refrigerant is either evaporated or condensed in the tube. This geometry provides 1.5-to-2.5 heat transfer coefficient increase and 1.1-to-1.3 pressure drop increase. Virtually all air cooled residential air conditioners use this tube. Air-cooled units use 7~9.5 mm diameter tubes. A 6.0 mm diameter tube is also used in window units. The micro-fin tube is made in diameters as small as 3.0 mm, which is used in heat pipes for cooling notebook computers. Current manufacturing trends are to make higher fins, which provide increased internal surface area.

Many experimental works have addressed measurements of the heat transfer coefficients for vaporization and condensation of refrigerants in tubes with different micro-fin geometries. Recent literature surveys on micro-fin tubes are provided by Thome,[13] Newell and Shah[14] for vaporization, and Cavallini et al.,[15,16] Newell and Shah[14] and Liebenberg et al.[17] for condensation. Rather than repeating a detailed literature survey, this paper reviews recent efforts on fundamental understanding and on optimization of fin geometry.

There are several variants of the microfin geometry:

(1) A single-groove direction microfin structure, which is the most common.

(2) A herringbone groove structure.

(3) A cross-grooved structure.

The basic algebraic dimensions that define the single-groove microfin geometry are shown in Fig. 6. These are:

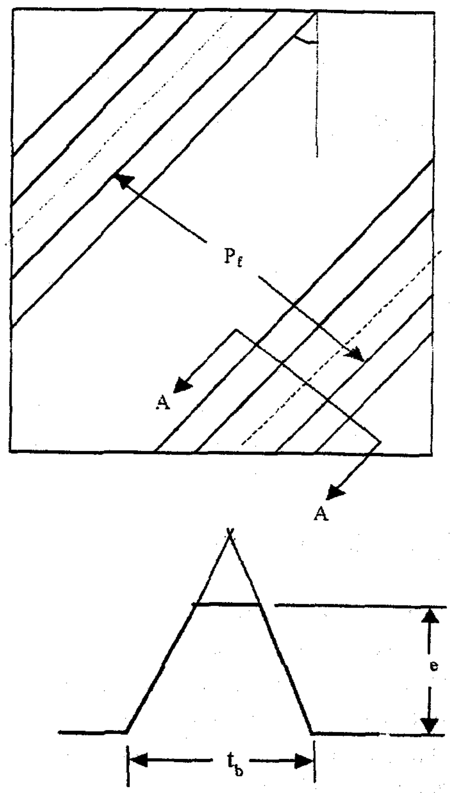


Fig. 6 Geometric parameters of a micro-fin tube.

(1) The rib layout parameters defined by the rib height (e), the rib pitch normal to the fins (p_f), and the rib helix angle (α). Alternately, one may define the rib pitch normal to the tube axis as $p_n = p_f / \cos\alpha$. The number of fins (N) is given by $N = \pi D_i / p_f = \pi D_i \cos\alpha / p_n$.

(2) The rib shape parameters are defined by the fin base thickness (t_b), and the apex angle of the fin (β). Thus, the principal dimensions are e, p_f, t_b, α, and β. Alternately, one may replace p_f by $N = \pi D_i / p_f$.

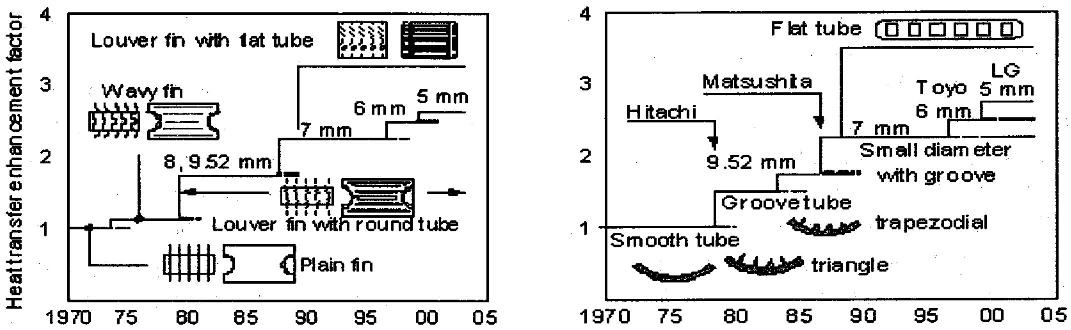
(3) The fin tip may be either a circular arc, or have a flat top. It appears that tubes made by pulling a seamless tube over a plug will have a curved fin tip. However, tubes made by embossing a flat strip and seam welding will tend to have a flat fin tip.

For micro-fins having a flat fin tip, the area, relative to that of a plain tube of the same diameter to the root of the fins (D_i) is $A/A_p = 1 + (\sec\beta/2 - \tan\beta/2)2e/p_f$. For a constant tube diameter, decrease of the fin pitch (p_f) will increase the number of fins, which provides increased A/A_p. For fixed fin pitch, the number of fins increases with increasing tube diameter.

There is little difference in the geometry of microfin tubes offered by most manufacturers. For a 9.5 mm diameter tube, typical values are $e = 0.20 \sim 0.25$ mm, $p_f = 0.46$, and $\alpha = 18°$. Tube manufacturers typically offer only one helix an-

Table 1 Optimum microfin configutaion (Dimensions in mm, G in kg/m^{2}s, q in kW/m^2)

References	Tube spec.	Test condition	Optimum configutarion (vaporization)	Optimum configutarion (condensation)
Ito and Kimura[20]	$D_o = 12.7$, $D_i = 11.2$	R-22 $40 \leq G \leq 203$ $3.5 \leq q \leq 52$	$\alpha = 7$ deg $p_n = 0.5 \sim 1.0$ $e = 0.2$	
Yasuda et al.[18]	$D_o = 9.52$, 7.94 $D_i = 8.8$, 7.22 $\beta = 40$ deg	R-22 $100 \leq G \leq 300$ $q = 10$	$\alpha = 18$ deg $p_n = 9.52$, $N = 60$ $p_n = 0.48$	$\alpha = 30$ deg $D_o = 9.52$, $N = 50$ $p_n = 0.53$
Tsuchida et al.[22]	$D_o = 9.52$ ($D_i = 8.92$) $\beta = 40$ deg	R-22 $Q = 10$ kW/m^2 $150 \leq G \leq 600$	$\alpha = 7$ deg, $e = 0.25$ $N = 70$ ($p_n = 0.40$)	$\alpha = 25$ deg, $e = 0.25$ $N = 80$ ($p_n = 0.32$)
	$D_o = 7$ ($D_i = 6.4$) $\beta = 40$ deg		$\alpha = 18$ deg, $e = 0.2$ $N = 60$ ($p_n = 0.32$)	$\alpha = 25$ deg, $e = 0.18$ $N = 60$ ($p_n = 0.30$)
	$D_o = 5$ ($D_i = 4.2$) $\beta = 40$ deg		$\alpha = 6$ deg, $e = 0.15$ $N = 45$ ($p_n = 0.29$)	$\alpha = 10$ deg, $e = 0.15$ $N = 45$ ($p_n = 0.29$)
	$D_o = 4$ ($D_i = 3.3$) $\beta = 40$ deg		$\alpha = 6$ deg, $e = 0.13$ $N = 40$ ($p_n = 0.26$)	$\alpha = 10$ deg, $e = 0.13$ $N = 40$ ($p_n = 0.26$)
Chamra et al.[19]	$D_o = 15.88$, $e = 0.35$ $p = 0.58$, $\beta = 30$ deg	R-22 $72 \leq G \leq 289$	$\alpha = 20$ deg	$\alpha = 27$ deg
Houfuku et al.[21]	$D_o = 7$ ($D_i = 6.4$ mm)	R-410A $G = 250$	$\alpha = 16$ deg, $\beta = 22$ deg $N = 54$ ($p_n = 0.37$) $e = 0.22$	$\alpha = 16$ deg, $\beta = 22$ deg $N = 54$ ($p_n = 0.37$mm) $e = 0.22$
Ishikawa et al.[23]	$D_o = 7$ ($D_i = 6.4$ mm) $\alpha = 16$ deg, $\beta = 10$ deg $e = 0.24$	R-22 $160 \leq G \leq 320$	$N = 80$ ($p_n = 0.25$)	$N = 80$($p_n = 0.25$ mm)



(a) Heat transfer enhancement by high performance fin shape

(b) Tube side heat transfer enhancement

Fig. 7 Advances in air-side and refrigerant side technology for residential condensers.

gle (approximately 18°) for both condensation and evaporation.

Although the evaporation coefficient attains a maximum at approximately $\alpha=18°$, Yasuda et al.[18] and Chamra et al.[19] have shown that the optimum helix angle for condensation is approximately 30°. Considerable recent work has been done to reduce the material content of the microfins. This work has basically involved reducing the fin base thickness (t_b) and the apex angle (β). Note that A/A_p is not influenced by the fin base thickness.

Work to optimize the micro-fin geometry is summarized in Table 1. Research at Hitachi Cable includes publications of Ito and Kimura,[20] and Yasuda et al.[18] for 9.5 mm tube diameter, Houfuku et al.[21] for 7.0 mm tube diameter and Tsuchida et al.[22] for 4.0 and 5.0 mm tube diameters. Early 9.5 mm diameter microfin tubes, as described by Yasuda et al.[18] had $e=0.2$ mm and $\beta=90°$. By 1989, Hitachi had increased their fin height to 0.25 mm and β decreased to 40° as reported by Yasuda et al.[18] Typically 60 fins were used in a 9.52 mm O.D. tube, as reported by Yasuda et al.,[18] for which $p_f=0.47$ mm. Recent work to optimize the microfin geometry for a 7.0 mm O.D. tube reported by Houfuku et al.[21] use $e=0.22$ mm, $p_f=0.37$ mm ($n=54$), $\beta=22°$, and $\alpha=16°$. Tsuchida et al.[22] found that smaller fin height is preferred for small diameter tubes — 0.13 mm for 4.0 diameter, and 0.15 mm for 5.0 mm diameter. Included in all of this work were efforts to reduce the material content by reducing the fin base thickness and the fin apex angle. Ishikawa et al.[23] investigated the number of fins for 7.0 mm tube diameter with $\beta=10°$ with $e=0.21\sim0.24$ mm. They found that 80 fins ($p_f=0.26$ mm) was the optimum value. Note that this optimum is different from that reported by Houfuku et al.,[21] who found $e=0.22$ mm, $p_f=0.37$ mm ($N=54$), $\beta=22°$, and $\alpha=16°$.

Much data have been published reporting evaporation and condensation coefficients, which are typically referred to the nominal tube area. These data have been used to develop correlations to predict the condensation and evaporation coefficients, and the frictional pressure drop. However, little work has been done to relate the tube performance to the tube internal geometry parameters.

Figure 7 provides a summary of the performance improvements that have been made in the technology of residental air-cooled condensers. Figure 7(a) shows air-side improvements, and Fig. 7(b) shows tubes side improvements. The figure clearly shows how reduced tube diameter has contributed to the improvements. Future adoption of "flat-tube" technology promises further improvements.

3.2 Flat tubes used in brazed aluminum condensers

Mechanically expanded finned tube condensers having round tubes and serpentine refrigerant circuiting are no longer used in the automotive industry. These have been replaced by brazed aluminum designs having parallel flow refrigerant circuiting. The brazed aluminum heat exchangers are seriously considered as condensers of residential air-conditioners. The basic design concept is illustrated in Fig. 8 and is described in Chapter 14 of Webb and Kim.[1] These condensers use a flat, extruded aluminum tube having a minor diameter as small as 1.35 mm, and hydraulic diameters in the range of 1.0 mm. A flat tube presents less projected frontal area to the air stream, and hence will reduce the air-side pressure drop. Further, use of "parallel-flow" refrigerant circuiting significantly improves refrigerant-side performance.

Figure 9 shows several tube geometries used in these condensers. Two variants of the tubeused in this condenser are shown in Fig. 9. The tubes contain membrane webs between the flat surfaces for pressure containment.

The Fig. 9(a) tube contains a smooth inner surface, whereas the Fig. 9(b) tube contains 0.20 mm high "micro-fins." The Fig. 9(b) tube is an advancement over the plain inner surface of Fig. 9(a). The Fig. 9(c) advanced technology tube is 0.44 mm hydraulic diameter and has 0.3 mm wall thickness. Surface tension force enhances the condensation process by pulling the condensate from the fin tips of the Fig. 9(b) tube. The mechanism is discussed by Yang and Webb,[24] who also provide a model to predict the condensation coefficient.

Small tube minor diameter is preferred for the brazed aluminum condensers, because this will provide more air-side surface area for the same tube pitch, and lower pressure drop, because the small minor diameter provides less profile drag on the tube. Further, the condensing coefficient increases as the hydraulic diameter decreases. As the hydraulic diameter is decreased, the dP_f/dz increase is greater than the hA_i/L for the same mass velocity. Refrigerant pressure drop will act to reduce the driving temperature difference between the air and the refrigerant for fixed condenser exit saturation temperature.

Typically, one constrains the refrigerant pressure drop to a fixed value. The condenser frictional pressure drop (P_f) is the sum of the

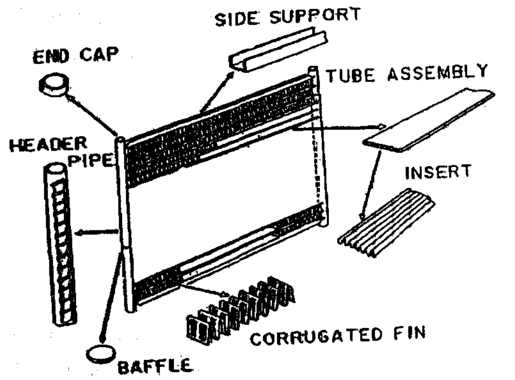
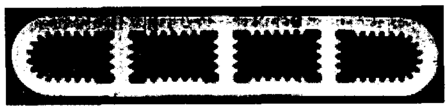


Fig. 8 Illustration of brazed aluminum, parallel flow condenser.

(a) 3×16 mm tube cross-section (0.5 mm wall) with a smooth inner surface



(b) 3×16 mm tube cross-section with 0.20 mm high "micro-fins"

(c) 1.35×20 mm advanced extrusion technology tube having 0.3 mm wall

Fig. 9 Tubes used in parallel flow refrigerant condensers.

$(dP_f/dz)_{pass}HN_p$, where $(dP_f/dz)_{pass}$ is the frictional pressure gradient in each pass and N_p is the number of passes. Because the cross-section flow area decreases with decreasing D_h, the mass flow rate in a small D_h tube will be smaller than for a tube having larger A_c (for fixed mass velocity). Therefore, a reduced circuit length (or number of passes) will be required to condense a flow at the same mass velocity. Therefore, one may use a larger dP_f/dz in the small hydraulic diameter tubes to obtain the same total circuit refrigerant pressure drop. It is probable that the reduced D_h tubes will operate at a moderately lower mass velocity than a higher D_h tube to achieve the same condenser pressure drop. Approximate relations for the tubes tested are $dP_f/dz \propto G^{2.4-3.7}$, while $h \propto G^{0.24-0.41}$.

So, a small reduction of G will significantly reduce the pressure gradient while having a smaller effect on the h-value. One may design for the desired G-value by selecting the number of tubes in parallel for each pass. A tube having smaller minor diameter will be preferred over a tube having larger minor diameter, even if the small D_h tube has no higher hA_i/L than the larger tube. This is because of the benefits derived on the air-side.

To obtain a detailed evaluation of the preferred tube geometry, it is necessary to design the condenser. This design involves selecting the required number of passes for the specific tube, and further selecting the optimum number of tubes/pass in each pass. The number of tubes used in each pass will decrease with each successive pass. The average vapor quality decreases in each successive pass, so one may decrease the number of tubes in each pass to achieve a constant mass velocity. Actually, one would probably allow the mass velocity to increase in each pass, because the lower average vapor quality will cause the dP_f/dz to decrease.

Yang and Webb[25] measured R-12 condensation heat transfer in flat extruded aluminum tubes shown in Figs. 9(a) and 9(b). Webb and Yang[26] extended the work to include R-134a. They found that the Shah[27] correlation, which is widely used for the prediction of the refrigerant condensation coefficient in larger diameter tubes, significantly overpredicted the data. However, the Akers et al.[28] correlation agreed well with the plain tube data. The condensation coefficient for the micro-fin tube is greater than that for the plain tube for vapor qualities greater than 0.5. Yang and Webb[25] proposed that the surface tension force is effective in enhancing the condensation coefficient for vapor qualities larger than 0.5. The surface tension induced enhancement was particularly strong at lower mass velocities. Webb and Ermis[29] extended the study to hydraulic diameters down to 0.44 mm. The tube geometry is shown in Fig. 9(c). Kim et al.[30,31] obtained R-22 and R-410A condensation data for similar tube geometries. The study of Webb and Ermis,[29] and Kim et al.[31,32] generally confirmed the findings by Yang and Webb.[25] Koyama et al.[32] obtained R-134a condensation data in 1.11 mm and 0.80 mm hydraulic diameter smooth aluminum extruded tubes. They found that the Haraguchi et al. correlation[33] developed for large diameter tubes highly overpredicted their data.

4. Conclusion

Brazed aluminum heat exchangers offers high promise but its introduction will require significant investment. The new all copper flat tube condenser will provide an alternative to brazed aluminum. Parallel refrigerant flow circuiting used in brazed aluminum condensers offers significant improvements over serpentine refrigerant circuiting.

Vortex generators have been studied to improve the performance of finned tube heat exchanges. However, it does not appear that vortex generators provide greater enhancement than can be obtained from conventional slit or louver fins.

Current manufacturing trend of the micro-fin tubes are to reduce material content by reducing the fin base thickness and the apex angle. Although many empirical correlations are available to predict the condensation and evaporation coefficient in micro-fin tubes, little work have been done to relate the tube performance to the internal parameters.

For the flat tube used in brazed aluminum condensers, small tube minor diameter is preferred, because this will provide more air-side surface area and lower pressure drop. Further, the condensing coefficient increases as the hydraulic diameters decreases. Moser et al.[34] model with Zhang and Webb[35] friction correlation is recommended to predict condensation coefficient in flat-tubes.

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