

5 **HEAT TRANSFER TUBES, INCLUDING METHODS
 OF FABRICATION AND USE THEREOF**

Related Applications

 This application claims the benefit of U.S. Provisional
10 Application Serial No. 60/374171 filed April 19, 2002.

Field of Invention

 The present invention relates generally to heat transfer tubes,
their method of formation and use. More particularly, the present invention
relates to an improved boiling tube, a method of manufacture and use of that
15 tube in an improved refrigerant evaporator or chiller.

Background of the Invention

 A component device of industrial air conditioning and
refrigeration systems is a refrigerant evaporator or chiller. In simple terms,
20 chillers remove heat from a cooling medium that enters the unit, and deliver
refreshed cooling medium to the air conditioning or refrigeration system to
effect cooling of a structure, device or given area. Refrigerant evaporators on
chillers use a liquid refrigerant or other working fluid to accomplish this task.
Refrigerant evaporators on chillers lower the temperature of a cooling
25 medium, such as water (or some other fluid), below that which could be
obtained from ambient conditions for use by the air conditioning or
refrigeration system.

 One type of a chiller is a flooded chiller. In flooded chiller
applications, a plurality of heat transfer tubes are fully submerged in a pool of
30 a two-phase boiling refrigerant. The refrigerant is often a chlorinated-
fluorinated hydrocarbon (*i.e.*, "Freon") having a specified boiling temperature.
A cooling medium, often water, is processed by the chiller. The cooling

medium enters the evaporator and is delivered to the plurality of tubes, which are submerged in a boiling liquid refrigerant. As a result, such tubes are commonly known as "boiling tubes." The cooling medium passing through the plurality of tubes is chilled as it gives up its heat to the boiling refrigerant.

5 The vapor from the boiling refrigerant is delivered to a compressor which compresses the vapor to a higher pressure and temperature. The high pressure and temperature vapor is then routed to a condenser where it is condensed for eventual return through an expansion device to the evaporator to lower the pressure and temperature. Those of ordinary skill in the art will appreciate
10 that the foregoing occurs in keeping with the well-known refrigeration cycle.

It is known that heat transfer performance of a boiling tube submerged in a refrigerant can be enhanced by forming fins on the outside surface of the tube. It is also known to enhance the heat transfer ability of a boiling tube by modifying the inner tube surface that contacts the cooling
15 medium. One example of such a modification to the inner tube surface is shown in U.S. Patent No. 3,847,212, to Wither, Jr., et al., which teaches forming ridges on a tube's inner surface.

It is further known that the fins can be modified to further enhance heat transferability. For example, some boiling tubes have come to be
20 referred to as nucleate boiling tubes. The outer surface of nucleate boiling tubes are formed to produce multiple cavities or pores (often referred to as boiling or nucleation sites) that provide openings which permit small refrigerant vapor bubbles to be formed therein. The vapor bubbles tend to form at the base or root of the nucleation site and grow in size until they break
25 away from the outer tube surface. Upon breaking away, additional liquid refrigerant takes the vacated space and the process is repeated to form other vapor bubbles. In this manner, the liquid refrigerant is boiled off or vaporized at a plurality of nucleate boiling sites provided on the outer surface of the metallic tubes.

30 U.S. Pat. No. 4,660,630 to Cunningham et al. shows nucleate boiling cavities or pores formed by notching or grooving fins on the outer surface of the tube. The notches are formed in a direction essentially

perpendicular to the plane of the fins. The inner tube surface includes helical ridges. This patent also discloses a cross-grooving operation that deforms the fin tips such that nucleate boiling cavities (or channels) are formed having a greater width than the surface openings. This construction permits the vapor
5 bubbles to travel outwardly through the cavity, to and through the narrower surface openings, which further enhances heat transferability. Various tubes produced in accordance with the Cunningham et al. patent have been marketed by Wolverine Tube, Inc. under the trademark TURBO-B®. In another nucleate boiling tube, marketed under the trademark TURBO-BII®,
10 the notches are formed at an acute angle to the plane of the fins.

In some heat transfer tubes, the fins are rolled over and/or flattened after they are formed so as to produce narrow gaps which overlie the larger cavities or channels defined by the roots of the fins and the sides of adjacent pairs of fins. Examples include the tubes of the following United
15 States patents: Cunningham et al U.S. Pat. No. 4,660,630; Zohler U.S. Pat. No. 4,765,058; Zohler U.S. Pat. No. 5,054,548; Nishizawa et al U.S. Pat. No. 5,186,252; Chiang et al U.S. Pat. No. 5,333,682.

Controlling the density and size of nuclear boiling pores has been recognized in the prior art. Moreover, the interrelationship between pore
20 size and refrigerant type has also been recognized in the prior art. For example, U.S. Patent No. 5,146,979 to Bohler purports to increase performance using higher pressure refrigerants by employing tubes having nucleate boiling pores ranging in size from 0.000220 square inches to 0.000440 square inches (the total area of the pods being from 14% to 28% of
25 the total outer surface area). In another example, U.S. Patent No. 5,697,430 to Thors et al. also discloses a heat transfer tube having a plurality of radially outwardly extending helical fins. The tube inner surface has a plurality of helical ridges. The fins of the outer surface are notched to provide nucleate boiling sites having pores. The fins and notches are spaced to provide pores
30 having an average area less than 0.00009 square inches and a pore density of at least 2000 per square inch of the tube's outer surface. The helical ridges on the inner surface have a predetermined ridge height and pitch, and are

positioned at a predetermined helix angle. Tubes made in accordance with the inventions of that patent have been offered and sold under the trademark TURBO BIII®.

The industry continues to explore new and improved designs by which to enhance heat transfer and chiller performance. For example, U.S. Patent No. 5,333,682 discloses a heat transfer tube having an external surface configured to provide both an increased area of the tube's external surface and to provide re-entrant cavities as nucleation sites to promote nucleate boiling. Similarly, U.S. Patent No. 6,167,950 discloses a heat transfer tube for use in a condenser with notched and finned surfaces configured to promote drainage of refrigerant from the fin. As shown by such developments in the art, it remains a goal to increase the heat transfer performance of nucleate boiling tubes while maintaining manufacturing cost and refrigeration system operation costs at minimum levels. These goals include the design of more efficient tubes and chillers, and methods of manufacturing such tubes. Consistent with such goals, the present invention is directed to improving the performance of heat exchange tubes generally and, in particular, the performance of heat exchange tubes used in flooded chillers or falling film applications.

Summary of the Invention

The present invention improves upon prior heat exchange tubes and refrigerant evaporators by forming and providing enhanced nucleate boiling cavities to increase the heat exchange capability of the tube and, as a result, performance of a chiller including one or more of such tubes. It is to be understood that a preferred embodiment of the present invention comprises or includes a tube having at least one dual cavity boiling cavity or pore. While the tubes disclosed herein are especially effective in use in boiling applications using high pressure refrigerants, they may be used with low pressure refrigerants as well.

The present invention comprises an improved heat transfer tube. The improved heat transfer tube of the present invention is suitable for boiling or falling film evaporation applications where the tube's outer surface

contacts a boiling liquid refrigerant. In a preferred embodiment, a plurality of radially outwardly extending helical fins are formed on the outer surface of the tube. The fins are notched and the tips bent over to form nucleate boiling cavities. The roots of the fins may be notched to increase the volume or size
5 of the nucleate boiling cavities. The top surface of the fins are bent over and rolled to form a second pore cavity. The resultant configuration defines dual cavity pores or channels for enhanced production of vaporization bubbles. The internal surface of the tube may also be enhanced, such as by providing helical ridges along the internal surface, to further facilitate heat transfer
10 between the cooling medium flowing through the tube and the refrigerant in which the tube may be submerged. Of course, the present invention is not limited by any particular internal surface enhancement.

The present invention further comprises a method of forming an improved heat transfer tube. A preferred embodiment of the invented method
15 includes the steps of forming a plurality of radially outwardly extending fins on the outer surface of the tube, and bending the fins on the outer surface of the tube, notching and bending the left over (remaining between notches) material to form dual cavity nucleate boiling sites which enhance heat transfer between the cooling medium flowing through the tube and the boiling
20 refrigerant in which the tube may be submerged.

The present invention further comprises an improved refrigerant evaporator. The improved evaporator, or chiller, includes at least one tube made in accordance with the present invention that is suitable for boiling or falling film evaporation applications. In a preferred embodiment, the exterior
25 of the tube includes a plurality of radially outwardly extending fins. The fins are notched. The fins are bent to increase the available surface areas on which heat transfer may occur and to form nucleate dual cavity boiling sites, thus enhancing heat transfer performance.

It is an object of the present invention to provide an improved
30 heat transfer tube.

It is another object of the present invention to provide an improved heat transfer tube that is suitable for both flooded and falling film evaporator applications.

5 It is another object of the present invention to provide an improved heat transfer tube that defines least one dual cavity nucleate boiling site.

It is another object of the present invention to provide a method of manufacturing a heat transfer tube for boiling and falling film applications, wherein at least one dual cavity nucleate boiling site is located on the outer
10 tube surface to enhance the heat transfer capability of the tube.

It is another object of the present invention to provide an improved nucleate boiling tube for applications wherein fins formed on the outer tube surface have been bent to provide additional surface area for convective vaporization to thereby enhance the heat transfer capability of the
15 tube.

It is still another object of the present invention to provide a heat transfer tube which includes surface enhancements to the outer tube surface that can be produced in a single pass by finning equipment.

It is still another object of the present invention to provide a
20 heat transfer tube which includes surface enhancements to the inner tube surface which facilitate flow of liquid inside the tube, increase the internal surface area, and facilitate contact between the liquid and internal surface area so as to further enhance the heat transfer capability of the tube.

It is still another object of the present invention to provide a
25 method to make an improved heat transfer tube that defines at least one dual cavity nucleate boiling site.

It is still another object of the present invention to provide an improved refrigerant evaporator.

It is yet another object of the present invention to provide an
30 improved refrigerant evaporator having at least one heat transfer tube having at least one dual cavity nucleate boiling site.

It is yet another object of the present invention to provide an improved refrigerant evaporator having a plurality of heat transfer tubes wherein each of such tubes defines a plurality of dual cavity nucleate boiling sites.

5 It is yet another object of the present invention to provide an improved refrigerant evaporator having at least one heat transfer tube that is provided with dual-cavity nucleate boiling sites.

 It is yet another object of the present invention to provide a method of forming a heat transfer tube by bending the fins to define multiple
10 cavity nucleate boiling sites.

 These and other features and advantages of the present invention will be demonstrated and understood by reading the present specification including the appended drawings.

15 **Brief Description of the Drawings**

 Fig. 1 is an illustration of a refrigerant evaporator made in accordance with the present invention.

 Fig. 2 is an enlarged, partially broken away axial cross-sectional view of a heat transfer tube made in accordance with the present invention.

20 Fig. 3 is an enlarged, partially broken away axial cross-sectional illustration of a preferred embodiment of a heat transfer tube made in accordance with the present invention.

 Fig. 4 is a photomicrograph of the outer surface of the tube of Fig. 2 subsequent to fin-bending.

25 Fig. 5 is a cross-section taken along line 3-3 in Fig. 4.

 Fig. 6 is a cross-section taken along line 4-4 in Fig. 4.

 Fig. 7 is a photomicrograph of an outer surface of a heat transfer tube made in accordance with the present invention subsequent to root and fin notching but prior to fin-bending.

30 Fig. 8 is a schematic depiction of the outer surface of the tube of Fig. 3.

Fig. 9 is a graph comparing an efficiency index for the tube of the present invention and a heat exchange tube made in accordance with the inventions disclosed in U.S. Patent No. 5,697,430.

Fig. 10 is a graph comparing the inside heat transfer performance of the tube of the present invention and a heat exchange tube made in accordance with the inventions disclosed in U.S. Patent No. 5,697,430.

Fig. 11 is a graph comparing the pressure drop of the tube of the present invention and a heat exchange tube made in accordance with the inventions disclosed in U.S. Patent No. 5,697,430.

Fig. 12 is a graph comparing the overall heat transfer coefficient U_o in refrigerant HFC-134a at varying heat fluxes, Q/A_o .

Detailed Description of the Preferred Embodiments

Referring now in detail to the drawings, in which like numerals indicate like parts throughout, Fig. 1 shows a plurality of heat transfer tubes made in accordance with the present invention generally at 10. The tubes 10 are contained within a refrigerant evaporator 14. Individual tubes 10a, 10b and 10c are representative, as those of ordinary skill will appreciate, of the potentially hundreds of tubes 10 that are commonly contained in the evaporator 14 of a chiller. The tubes 10 may be secured in any suitable fashion to accomplish the inventions as described herein. The evaporator 14 contains a boiling refrigerant 15. The refrigerant 15 is delivered to the evaporator 14 from a condenser into a shell 18 by means of an opening 20. The boiling refrigerant 15 in the shell 18 is in two phases, liquid and vapor. Refrigerant vapor escapes the evaporator shell 18 through a vapor outlet 21. Those of ordinary skill will appreciate that the refrigerant vapor is delivered to a compressor where it is compressed into a higher temperature and pressure vapor, for use in keeping with the known refrigeration cycle.

A plurality of heat transfer tubes 10a-c, which are described in greater detail herein, are placed and suspended within the shell 18 in any suitable manner. For example, the tubes 10a-c may be supported by baffles

and the like. Such construction of a refrigerant evaporator is known in the art. A cooling medium, oftentimes water, enters the evaporator 14 through an inlet 21 and into an inlet reservoir 24. The cooling medium, which enters the evaporator 14 in a relatively heated state, is delivered from the reservoir 24 into the plurality of heat exchange tubes 10a-c, wherein the cooling medium gives up its heat to the boiling refrigerant 15. The chilled cooling medium passes through the tubes 10a-c and exits the tubes into an outlet reservoir 27. The refreshed cooling medium exits the evaporator 14 through an outlet 28. Those of ordinary skill will appreciate that the example flooded evaporator 14 is but one example of a refrigerant evaporator. Several different types of evaporators are known and utilized in the field, including the evaporator on absorption chillers, and those employing falling film applications. It will be further appreciated by those of ordinary skill that the present invention is applicable to chillers and evaporators generally, and that the present invention is not limited to brand or type.

Fig. 2 is an enlarged, broken away, plan view of a representative tube 10. Fig. 3, which is an enlarged cross-sectional view of a preferred tube 16, is readily considered in tandem with Fig. 2. Referring first to Fig. 2, the tube 10 defines an outer surface generally at 30, and an inner surface generally at 35. The inner surface is preferably provided with a plurality of ridges 38. Those of ordinary skill in the art will appreciate that the inner tube surface may be smooth, or may have ridges and grooves, or may be otherwise enhanced. Thus, it is to be understood that the presently disclosed embodiment, while showing a plurality of ridges, is not limiting of the invention.

Turning to the exemplary embodiment, ridges 38 on the inner tube surface 35 have a pitch "p," a width "b," and a height "e," each determined as shown in Fig. 3. The pitch "p" defines the distance between ridges 38. The height "e" defines the distance between a ceiling 39 of a ridge 38 and the innermost portion of the ridge 38. The width "b" is measured at the uppermost, outside edges of the ridge 38 where contact is made with the ceiling 39. A helix angle "θ" is measured from the axis of the tube, as also

indicated in Fig. 3. Thus, it is to be understood that the inner surface 35 of tube 10 (of the exemplary embodiment) is provided with helical ridges 38, and that these ridges have a predetermined ridge height and pitch and are aligned at a predetermined helix angle. Such predetermined measurements may be varied as desired, depending on a particular application. For example, U.S. Patent No. 3,847,212 to Withers, Jr. taught a relatively low number of ridges, at a relatively large pitch (0.333 inch) and a relatively large helix angle (51°). These parameters are preferably selected to enhance the heat transfer performance of the tube. The formation of such interior surface enhancements is well known to those of ordinary skill in the art and need not be disclosed in further detail other than as disclosed herein. It is to be recognized, for example, that U.S. Patent No. 3,847,212 to Wither, Jr. et al. discloses a method of formation, and formation, of interior surface enhancements.

The outer surface 30 of the tubes 10 is typically, initially smooth. Thus, it will be understood that the outer surface 30 is thereafter deformed or enhanced to provide a plurality of fins 50 that in turn provide, as described in detail herein, multiple dual-cavity nucleate boiling sites 55. While the present invention is described in detail regarding dual cavity nucleate pores, it is to be understood that the present invention includes heat transfer tubes 10 having nucleate boiling sites 55 made with more than two cavities. These sites 55, which are typically referred to as cavities or pores, include openings 56 provided on the structure of the tube 10, generally on or under the outer surface 30 of the tube. The openings 56 function as small circulating systems which direct liquid refrigerant into a loop or channel, thereby allowing contact of the refrigerant with a nucleation site. Openings of this type are typically made by finning the tube, forming generally longitudinal grooves or notches in the tips of the fins and then deforming the outer surface to produce flattened areas on the tube surface but have channels in the fin root areas.

Turning in greater detail to Figs. 2 and 3, outer surface 30 of tube 10 is formed to have a plurality of fins 50 provided thereon. Fins 50 may be formed using a conventional finning machine in a manner understood with

reference to U.S. Pat. No. 4,729, 155 to Cunningham et al., for example. The number of arbors utilized depends on such manufacturing factors as tube size, throughput speed, etc. The arbors are mounted at appropriate degree increments around the tube, and each is preferably mounted at an angle
5 relative to the tube axis.

Described in even greater detail, and focusing on Figs. 7 and 8, the finning disks push or deform metal on the outer surface 30 of the tube to form fins 50, and relatively deep grooves or channels 52. As shown, the channels 52 are formed between the fins 50, and both are generally
10 circumferential about the tube 10. As shown in Fig. 3, the fins 50 have a height, which may be measured from the innermost portion 57 of a channel 52 (or a groove) and the outermost surface 58 of a fin. Moreover, the number of fins 50 may vary depending upon the application. While not limiting, a preferred range of fin height is between .015 and .060 inches, and a preferred
15 count of fins per inch is between 40 to 70. It is then to be understood that the finning operation produces a plurality of first channels 52, as shown in Figs. 7 and 8.

After fin formation, the outer surface 57 of each fin 50 is notched to provide a plurality of second channels 62. Such notching may be
20 performed using a notching disk (see reference in U.S. Patent No. 4,729,155 to Cunningham, for example). The second channels 62, which are positioned at an angle relative to the first channels 20, interconnect therewith as shown in Figs. 7 and 8. The notching operation described in U.S. Patent No. 5,697,430, is one appropriate method for performing this notching operation so as to
25 define the second channels 62, and to form a plurality of notches 64.

After notching, the outer surface 57 of the fins 50 are flattened or bent over by means of a compression disk (see reference in U.S. Patent No. 4,729,155 to Cunningham , for example). This step flattens or bends over the top or heads of each fin, to create an appearance as shown, for example as in
30 Figs. 7 and 8. It is to be understood that the foregoing operations create a plurality of pores 70 at the intersection of channels 52 and 62. These pores 70 define nucleate boiling sites and each defined by a pore size. More

particularly, referring in detail to Fig. 3, this first flattening or bending operation forms the primary nucleate boiling cavity 72.

After flattening, the fins 50 are rolled or bent once again by a rolling tool. The rolling operation exerts a force across and over the flattened fin heads 58. The fins 50 are bent or rolled by a tool so as to at least partially cover the fin notches 64 and thereby form secondary boiling cavities 74 between the bent fins 50 and the fin notches 64. The secondary cavities 74 provide extra fin area above the primary cavities 30 to promote more convective and nucleation boiling. Thus, pores 70 are formed at the intersection of channels 52 and 62. Each pore 70 has a pore opening, which is the size of the opening from the boiling or nucleation site from which vapor escapes. The preferred embodiment of the present invention defines two cavities, primary cavity 72 and secondary cavity 74, which enhances performance of the tube.

The tube 10 is preferably notched in the first channels 52 between the fins ("fin root area") to thereby form root notches in the root surface. The notching is accomplished using a root notching disk. While root notches of a variety of shapes and sizes may be notched in the fin root area, formation of root notches having a generally trapezoidal shape are preferable. While any number of root notches may be formed around a circumference of each groove 20, at least 20 to 100, preferably forty-seven (47), root notches per circumference are recommended. Moreover, root notches 26 preferably have a root notch depth of between 0.0005 inches to 0.005 inches and more preferably .0028 inches.

Enhancements to both the inner surface 35 and the outer surface 30 of tube 10 increase the overall efficiency of the tube by increasing both the outside (h_o) and inside (h_i) heat transfer coefficients and thereby the overall heat transfer coefficient (U_o), as well as reducing the total resistance to transferring heat from one side to another side of the tube (R_T). The parameters of the inner surface 35 of tube 10 enhance the inside heat transfer coefficient (h_i) by providing increased surface area against which the fluid may contact and also permitting the fluid inside tube 10 to swirl as it traverses

the length of tube 10. The swirling flow tends to keep the fluid in good heat transfer contact with the inner surface 14 but avoids excessive turbulence which could provide an undesirable increase in pressure drop.

Moreover root notching the outer surface 30 of the tube and bending (as opposed to the traditional flattening) of the fins 50 facilitate heat transfer on the exterior of the tube and thereby increase the outside heat transfer coefficient (h_o). The root notches increase the size and surface area of the nucleate boiling cavities and the number of boiling sites and help keep the surface wetted as a result of surface tension forces which helps promote more thin film boiling where it is needed. Fin bending results in formation of an additional cavities (such as secondary cavity 74) positioned over each primary cavity 50 which can serve to transfer additional heat to the refrigerant and through the liquid vapor inter-phase of a rising vapor bubble escaping from the secondary cavity 60 by means of convection and/or nucleate boiling depending on heat flux and liquid/vapor movement over the outside surface of the tube. As one skilled in the art will appreciate, the outside boiling coefficient is a function of both a nucleate boiling term and a convective component. While the nucleate boiling term is usually contributing the most to the heat transfer, the convective term is also important and can become substantial in flooded refrigerant chillers.

Tube 10 of the present invention in respects outperforms the tube disclosed in U.S. Patent No. 5,697,430 (designated as "T-BIII® Tube" in the subsequently-described tables and graphs), which is currently regarded as the leading performer in evaporation performance among widely commercialized tubes. In order to allow a comparison of the improved tube 10 of the present invention (designated as "New Tube" in the subsequently-described tables and graphs) to the T-BIII® Tube, Table 1 is provided to describe dimensional characteristics of the New Tube and T-BIII® Tube:

TABLE 1
DIMENSIONAL CHARACTERISTICS OF COPPER TUBES
HAVING MULTIPLE-START INTERNAL RIDGING

TUBE DESIGNATION	T-BIII [®] Tube	New Tube
PRODUCT NAME	Turbo-BIII [®]	Turbo-EDE [®]
FPI = fins per inch (fpi)	60	48
Posture of Fins	Mangled	Mangled
FH = Fin Height (inches)	.0215	.021
A _o = True Outside Area (ft ² /ft)	Unknown	Unknown
d _i = Inside Diameter (inches)	.645	.652
e = Ridge Height (inches)	.016	.014
p = Axial Pitch of Ridge (inches)	.0516	.0457
N _{RS} = Number of Ridge Starts	34	44
l = Lead (inches)	1.76	2.01
θ = Lead Angle of Ridge from Axis (°)	49	45
b = Ridge Width Along Axis (inches)	.0265	.0184

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Table 2 compares the inside performance of the New Tube and T-BIII Tube. Both tubes are compared at constant tube side water flow rate of 5 GPM and constant average water temperature of 50°F. Comparisons in Table 2 are based on nominal 3/4 inch outside diameter tubes.

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TABLE 2

**TUBE SIDE PERFORMANCE CHARACTERISTICS
OF EXPERIMENTAL COPPER TUBES HAVING
MULTIPLE-START INTERNAL RIDGING**

5

	T-BIII Tube	New Tube
u = Intube Water Velocity (ft/s)	4.89	4.78
C_i = Inside Heat Transfer Coefficient Constant (From Test Results)	.075	0.077
f_D = Friction Factor (Darcy)	0.0624	0.0623
$\Delta p_e / ft$ = Pressure Drop per Foot	0.187	0.177
St_e / St_s = Stanton Number Ratio (enhanced/Smooth)	2.52	2.59
$\Delta p_e / \Delta p_s$ = Pressure Drop Ratio (Enhanced / Smooth)	3.34	3.16
$\eta = (St_e / St_s) / (\Delta p_e / \Delta p_s)$ = Efficiency index	0.78	0.82

The data illustrates the reduction in pressure drop and increase in heat transfer efficiency achieved with the New Tube. As can be seen in Table 2 and graphically illustrated in Fig. 11, the pressure drop ratio ($\Delta p_e / \Delta p_s$) relative to a smooth bore tube, at 5 GPM constant flow rate, for the New Tube is approximately 5% less than for the T-BIII Tube. Also from Table 2 and graphically illustrated in Fig. 10, one can see that the Stanton Number ratio (St_e / St_s) of the New Tube is approximately 2% higher than for the T-BIII® Tube. The pressure drop and Stanton Number ratios can be combined into a total ratio of heat transfer to pressure drop and is defined as the “efficiency index” (η), which is a total measure of heat transfer to pressure drop compared to a smooth bore tube. At 5 GPM, the efficiency index η for the New Tube is .82 and for the T-BIII® Tube is .78, resulting in an approximately 5% improvement with the New Tube, as graphically illustrated in Fig. 9, at this GPM. At 7 GPM (usual operating condition), higher percentage improvement would be obtained.

Table 3 compares the outside performances of the New Tube and the T-BIII® Tube. The tubes are eight feet long and each is separately suspended in a pool of refrigerant temperature of 58.3 degrees Fahrenheit. The water flow rate is held constant at 5.3 ft/s and the inlet water temperature is such that the average heat flux for all tubes is held at 7000 Btu/hr ft² which is constant. The tubes are made of copper material, have a nominal 3/4 inch outer diameter, and have the same wall thickness. All tests are performed without any oil present in the refrigerant.

TABLE 3
OUTSIDE AND OVERALL PERFORMANCE CHARACTERISTICS OF
EXPERIMENTAL COPPER TUBES HAVING MULTIPLE-START
INTERNAL RIDGING

	T-BIII Tube	New Tube
h_o = Average Boiling Coefficient based on Nominal Outside Area HFC-134A Refrigerant (Btu/hr ft ² F)	10,000	13,000
U_o = Overall Heat Transfer Coefficient, Based on Nominal Outside Area in HFC-134a Refrigerant (Btu/hr ft ² F)	1,960	2,250

Fig. 11 is a graph comparing the overall heat transfer coefficient U_o in HFC-134a refrigerant at varying heat fluxes, Q/A_o , for the New Tube and T-BIII® Tube. At a 7,000 (Btu/hr ft²) heat flux, the enhancement of the New Tube over the T-BIII® Tube is 15% at a water flow rate of 5 GPM (also shown in Table 3).

The foregoing is provided for the purpose of illustrating, explaining and describing embodiments of the present invention. Further modifications and adaptations to these embodiments will be apparent to those skilled in the art and may be made without departing from the spirit of the invention or the scope of the following claims. Moreover, the person of ordinary skill in the art will appreciate that the present invention provides a fin

having a unique profile that creates nucleate boiling sites having multiple cavities, such as a dual cavity. The present invention provides such a unique profile without shaving off any metal to create the pore, and then provides an improved manufacturing method of forming an improved heat transfer tube.

- 5 Yet further, use of one or more of such tubes in a flooded chiller results in improved performance of the chiller in terms of heat transfer. Thus, the foregoing explanation and description of the preferred embodiments in exemplary, and the invention is set forth in the appended claims.

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CLAIMS

What is claimed is:

1. A heat transfer tube suitable for use in a refrigerant
5 evaporator comprising:
 - an outer surface, said outer surface comprising a plurality of radially outwardly extending helical fins with channels extending between adjacent fins, said fins being grooved to define notches;
 - at least one nucleate boiling pore is formed at the
10 intersection of notch and a channel;
 - the fins being notched and bent such that adjacent fins form a channel extending between neighboring nucleate boiling pores, said pores thus defining a primary nucleate boiling cavity; and
 - the fins further being bent over so as to define a
15 secondary nucleate boiling cavity.
2. A method of fabricating a heat transferring tube for contacting a refrigerant and an inner surface for contacting a cool medium to be refreshed, the method comprising:
 - 20 (a) forming a plurality of helical ridges on the inner side of the tube;
 - (b) forming a plurality of radially outwardly extending fins on the outer surface of the tube;
 - (c) notching said fins to provide a primary nucleate
25 boiling cavity by forming a plurality of first notches in a first direction and by forming a plurality of second notches in a second direction; and
 - (d) bending over said fins to provide a second primary nucleate boiling cavity in communication with said primary nucleate boiling cavity,
 - 30 wherein the process defines a nucleate boiling pre at the intersection of said notches, said pore having a primary cavity and a secondary cavity.

3. An improved refrigerant evaporator, comprising:
a shell;
a refrigerant contained within said shell; and
5 at least one heat transfer tube contained with said shell
and submerged in said refrigerant, said heat transfer tube including at least
one nucleate boiling pore defining a primary cavity and a secondary cavity.

10