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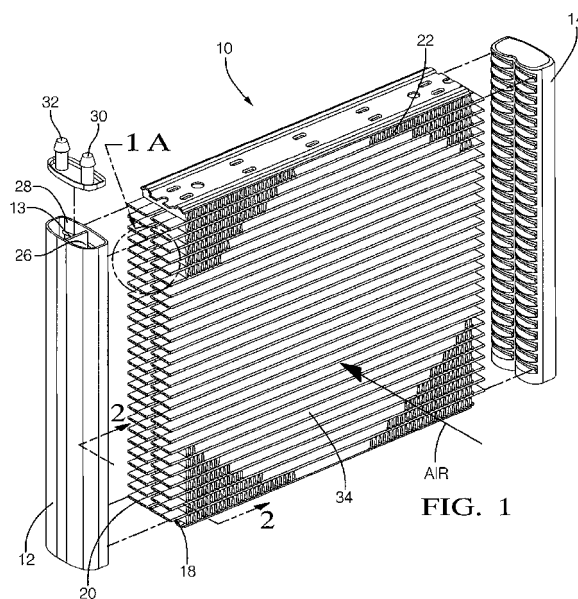
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(54) **Folded tube for an evaporator and evaporator assembly therewith**

(57) An evaporator assembly having a first header, a second header, at least two banks of evaporator tubes extending therebetween and in hydraulic communication with the first and second headers. At least one of the evaporator tube may be folded from a unitary strip clad aluminum folded having a thickness (t). The evaporator tube includes a height (h) which is measured from the bottom exterior surface to the top exterior surface, and a corner radius (r_c) defined by the transition radius from the flange segments to the channel walls. The bottom wall includes a width (2w), the corrugated portion includes alternating flange segments abutting the interior surface and channel walls connecting the alternating flange segments, at least one of the alternating adjacent flange segments includes a length (a) cooperating with adjacent the channel walls to define a channel having a width (b). The evaporator tube also includes a number of ports per millimeter width (PPMW) in a range of 0.40 to 1.0 as defined by the equation $PPMW = 2/(a+b+t)$; a Port Shape (PS) ratio of 0.05 to 0.6 as defined by the equation $PS \text{ ratio} = a/b$; a non-dimensional gauge (NDG) ratio of 0.11 to 0.21 as defined by the equation $NDG \text{ ratio} = t/h$; and a non-dimensional corner radius (NDCR) ratio of 0.10 to 0.5 as defined by the equation $NDCR \text{ ratio} = r_c/2t$.



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Description

TECHNICAL FIELD OF INVENTION

5 **[0001]** The present invention relates generally to a tube for a heat exchanger; specifically, to a fabricated tube for an evaporator; and more specifically, to a folded evaporator tube.

BACKGROUND OF INVENTION

10 **[0002]** A heat exchanger assembly such as a radiator, condenser, or evaporator for use in a motor vehicle typically includes an inlet header, an outlet header, a plurality of tubes hydraulically connecting the headers for fluid flow therebetween, and external fins interconnecting the tubes. The headers, tubes, and fins are typically assembled into a unitary structure and brazed to form the heat exchanger assembly.

15 **[0003]** A first heat transfer fluid, such as a liquid coolant, flows from the inlet header to the outlet header through the plurality of tubes. The first heat transfer fluid is in contact with the interior surfaces of the tubes while a second heat transfer fluid, such as ambient air, is in contact with the exterior surfaces of the tubes. Where a temperature difference exists between the first and second fluids, heat is transferred from the higher temperature fluid to the lower temperature fluid through the walls of the tubes. It is known to provide internal fins within the passageways of the tubes to increase the surface area available for heat transfer, as well as to increase the structural integrity of the tubes. The internal fins extend substantially the length of the tubes and define a plurality of channels or ports for the flow of a heat transfer fluid between the headers.

20 **[0004]** Heat exchanger tubes having a plurality of channels are also known as multi-port tubes. A known method of manufacturing multi-port tubes is by extruding a billet of deformable heat conductive material through a die. The extrusion process allows for the formation of the internal fins to have intricate geometric features to improve heat transfer efficiency that other known manufacturing process could not readily provide. However, the extrusion process is known to be expensive because of the need to frequently replace the extrusion die in order to maintain the desired dimensions of the intricate geometric features. Extruded tubes are also prone to corrosion attacks from road salt and acidic rain and require extensive corrosion inhibition coatings for motor vehicle applications, which add to the complexity of manufacturing and cost.

25 **[0005]** Another known method of forming multi-port tubes is by folding a sheet of pliable heat conductive material. Typically, a flat elongated sheet of metallic material is folded to form a tube having multiple ports defined by internal corrugated folds. The internal corrugated folds form the internal fins that define the shape and size of the ports. Folded tubes provide numerous advantages over extruded tubes in terms of lower cost and ease of manufacturing for the tube itself as well as for the final assembly of the heat exchanger. One advantage is that a folded tube can be formed from a sheet of clad aluminum that offers superior corrosion protection without the need for applying additional coatings. Another advantage is that due to the presence of cladding on the tube, other components of the heat exchanger, such as the headers and air fins, need not be clad, thereby simplifying the material system for corrosion protection. A further advantage is that since the headers do not need to be clad, the headers can be formed with extrusion technology to reduce the cost of manufacturing.

30 **[0006]** However, a shortcoming of a fold tube is that the thickness, or gage, of the sheet of heat conductive material limits the geometry and number of ports that the folding process can provide. The geometry and number of ports are important factors for applications in evaporator type heat exchangers to meet heat transfer requirement within a given core package.

35 **[0007]** For applications in evaporator type heat exchangers, there is a long felt need for folded tubes to have a geometry and features that can provide equivalent, if not better, heat transfer efficiency as that of the extruded tubes.

SUMMARY OF INVENTION

40 **[0008]** One aspect of the invention is an evaporator tube folded from a unitary strip of heat conductive material having a thickness (t). The folded tube includes a cross sectional shape having a bottom wall with two opposing tube edges transitioning into a pair of top walls spaced from and substantially parallel to the bottom wall, a pair of abutted central walls bent substantially perpendicularly out of the top walls and extends toward the bottom wall, a corrugated portion extending substantially perpendicularly out of each of the central walls toward the corresponding tube edge.

45 **[0009]** The bottom wall includes a width (2w), the corrugated portion includes alternating flange segments abutting the interior surface of the tube and channel walls connecting the alternating flange segments; at least one of the alternating adjacent flange segments includes a length (a) cooperating with adjacent channel walls to define a channel having a width (b). The evaporator tube includes a height (h) which is measured from the bottom exterior surface to the top exterior surface of the tube and a corner radius (r_c) defined by the transition radius from a flange segment to the channel wall.

[0010] The evaporator tube also includes a number of ports per millimeter width (PPMW) in a range of 0.40 to 0.80 as defined by the equation $PPMW = 2/(a+b+t)$; a Port Shape (PS) ratio of 0.05 to 0.5 as defined by the equation $PS \text{ ratio} = a/b$; a non-dimensional gauge (NDG) ratio of 0.11 to 0.21 as defined by the equation $NDG \text{ ratio} = t/h$; and a non-dimensional corner radius (NDCR) ratio of 0.10 to 0.5 as defined by the equation $NDCR \text{ ratio} = r_c/2t$.

[0011] Another aspect of the invention is an evaporator assembly having a first header, a second header, at least two banks of evaporator tubes extending between and in hydraulic communication with the first and second headers. At least one of the evaporator tube includes a unitary strip clad aluminum having a thickness (t) folded into a cross sectional shape having a bottom wall with two opposing tube edges transitioning into a pair of top walls spaced from and substantially parallel to the bottom wall, a pair of abutted central walls bent substantially perpendicularly out of the top walls and extend toward the bottom wall, and a corrugated portion extending substantially perpendicularly out of each of the central walls toward the corresponding tube edge.

[0012] The bottom wall includes a width (2w) and the corrugated portion includes alternating flange segments abutting the interior surface and channel walls connecting the alternating flange segments. At least one of the alternating adjacent flange segments includes a length (a) cooperating with adjacent channel walls to define a channel having a width (b). The evaporator tube includes a height (h) measured from the bottom exterior surface to the top exterior surface and a corner radius (r_c) defined by the transition radius from the flange segment to the channel wall.

[0013] The evaporator tube also includes a number of ports per millimeter width (PPMW) includes a range of 0.40 to 0.80 as defined by the equation $PPMW = 2/(a+b+t)$; a Port Shape (PS) ratio having a range of 0.05 to 0.5 as defined by the equation $PS \text{ ratio} = a/b$; a non-dimensional gauge (NDG) ratio range of 0.11 to 0.21 as defined by the equation $NDG \text{ ratio} = t/h$; and a non-dimensional corner radius (NDCR) ratio range from 0.10 to 0.5 as defined by the equation $NDCR \text{ ratio} = r_c/2t$.

[0014] Folded evaporator tube having the above critical parameters provide evaporators with improved heat transfer performance, reduced refrigerant pressure drop, increased burst strength, increased robustness of brazing process, and reduced heat exchanger mass per unit volume.

BRIEF DESCRIPTION OF DRAWINGS

[0015] This invention will be further described with reference to the accompanying drawings in which:

Fig. 1 is a perspective view of an evaporator having two banks of folded evaporator tubes for a motor vehicle.

Fig. 1A is a detail view of the evaporator of Fig. 1.

Figs. 2A-E show the intermediate stages in the formation of the folded evaporator tube for the evaporator of Fig. 1.

Fig. 3 is a cross-section of a folded evaporator tube having geometric features of the present invention.

Figs. 4A-D are graphs showing the effects of the ranges of the critical parameters of the folded evaporator tube on the operating characteristics of the evaporator shown in Fig. 1.

DETAILED DESCRIPTION OF INVENTION

[0016] Referring to the Figs. 1 through 3, wherein like numerals indicate corresponding parts throughout the views, is an embodiment of an evaporator 10 having evaporator tubes 16 with features that improve the operating characteristics of the evaporator 10. Shown in Fig. 1 is a perspective view of an evaporator 10 having dual banks 18, 20 of evaporator tubes 16 for use in a motor vehicle. The evaporator 10 is typically housed in a HVAC module of a motor vehicle and includes a plurality of evaporator tubes 16 hydraulically connecting two spaced apart headers 12, 14 for a two-phase refrigerant flow therebetween. For an exemplary two-pass evaporator 10, the first header 12 is typically an inlet/outlet header defining a cavity that includes a substantially center partition 13 extending the length of the first header 12 and separates the cavity into an inlet chamber 26 and an outlet chamber 28. The inlet chamber 26 is in hydraulic communication with an inlet port 30 and the outlet chamber 28 is in hydraulic communication with an outlet port 32. The first bank 18 of substantially parallel evaporator tubes 16 hydraulically connects the inlet chamber 26 to the return header 14 and a second bank 20 of evaporator tubes 16 connects the return header 14 to the outlet chamber 28. External fins 22 are disposed between and interconnect the evaporator tubes 16 to increase the surface area available for heat transfer. The evaporator tubes 16 and fins 22 together define the core 34 of the evaporator 10 through which ambient air flows.

[0017] During normal operating conditions, a partially expanded two-phase refrigerant flows into the inlet chamber 26 of the first header (inlet/outlet header) by way of the inlet port 30 and continues through the first bank 18 of evaporator tubes 16 to the second header (return header) 14. From the second header 14, the two-phase refrigerant flows through the second bank 20 of evaporator tubes 16 to the outlet chamber 28 of the first header 12 and exits the outlet port 32. As the two-phase refrigerant flows through the evaporator tubes 16, the two-phase refrigerant continues to expand into a vapor phase by absorbing heat from the ambient air. To further increase the heat transfer efficiency, the evaporator tubes 16 include internal geometric features having specific critical parameters that provide for improved performance

of the evaporator 10.

[0018] Shown in Fig. 1A is a view of the evaporator 10 of Fig. 1 at detail section 1A. Fig. 1A shows dual banks of folded B-type evaporator tubes 16. The B-type evaporator tube 16 shown is typically formed by folding a sheet of heat conductive material to define a series of internal channels 36 for refrigerant flow. The channel walls 72 formed from the folding of the sheet of heat conductive material act as internal fins to increase the area available for heat transfer. Figs. 2A-E show the typical stages in the formation of a folded B-type evaporator tube.

[0019] Shown in Fig. 2A is a partial sheet of heat conductive material strip 50, preferably a continuous clad aluminum strip 50, having a first surface 52 and a second surface 54 extending along a longitudinal A-axis. The heat conductive material strip 50 is longitudinally fed into a multi-station roll forming apparatus having pairs of rollers arranged to symmetrically plastically deform the heat conductive material strip 50 to form a corrugated portion 56 on both sides of the A-axis. Each of the corrugated portions 56 includes a series of alternating crests 57 and joining segments 63. The alternating crests 57 may be that of sharp corners to substantially flat surfaces that corresponds to the first and second surfaces 52, 54 of the heat conductive material strip 50, the significant of which will be discussed below.

[0020] Intermediate stations in the roll forming apparatus successively further deform the heat conductive material strip 50 to the intermediate configuration shown in Fig. 2B. The corrugated portion 56 is folded inward toward the second surface 54 such that the crests 57 on the same side as the second surface 54 are oriented toward and are in contact with the second surface 54. The fold of the corrugated portion 56 defines an abutting surface 58. The folded corrugated portion 56 is folded again toward the second surface 54 such that a portion of the first surface 52 defines an exterior tube edge 60 as shown in Figs. 2C and 2D. The abutting surfaces 58 of the corrugated portion 56 on either side of the A-axis are abutted upon each other and brazed forming the B type evaporator tube 16 having a center seam that runs the length of the tube as shown in Fig. 2E. On leaving the roll forming apparatus, the continuous evaporator tube 16 is cut to the desired length.

[0021] Shown in Fig. 3 is a cross sectional view of the folded B-type evaporator tube 16 of Figs. 2A-E having a central wall 62, two opposing tube edges 60, a bottom wall 64, and a pair of top walls 66. The folded evaporator tube 16 also has internal horizontal flange segments 68, defined by the crests 57 of the folded material strip 50, abutting the interior surface 70 of the folded tube and transitioning to channel walls 72 defined by the joining segments 63. As disclosed herein, the center walls 62, tube edges 60, bottom wall 64, top walls 66, and corrugated portions 56 are formed by folding a continuous strip 50 of clad aluminum. The terms "bottom", "upper", and "horizontal" are arbitrary, as the evaporator tube could be in any orientation. Likewise, the central walls 62 need not be exactly in the center of the width of the cross section, but typically will be.

[0022] The B-type evaporator tube is preferably folded from a clad aluminum strip 50 having a stock thickness of (t) and includes a width (2w) that is measured from external tube edge 60 to external tube edge 60. A typical B-type evaporator tube for use in automotive applications has a width (2w) in the range from 10mm to 30 mm for an evaporator 10 having a dual bank of evaporator tubes 16. The height (h) of the evaporator tube 16 is measured from the exterior surface of the bottom wall 64 to the exterior surface of the top wall 66. The length of the flange segments 68 abutting the interior surface 70 of the tube surface is shown as (a). The distance of the channel 36 defined between the adjacent intersections of the channel walls 72 and interior surface 70 of the tube is shown as (b). The angle between the channel wall 72 and the interior surface 70 is shown as (θ). The corner radius of the transition from the flange to the channel wall 72 is shown as (r_c).

[0023] The hydraulic parameter and the wetted parameter of the folded evaporator tube 16 is defined as:

$$\text{Hydraulic Diameter, } D_h = \frac{2(a+b)(h-3t)}{(a+b + \frac{2(h-3t)}{\sin \theta})},$$

where

$$\theta = \tan^{-1} \left(\frac{2h-6t}{b-a} \right)$$

Wetted Perimeter, $P_w = (\text{no. of ports}) \times (\text{perimeter per port})$

$$= \frac{2w}{(a+b+t)} \times \left(a + b + \frac{2(h-3t)}{\sin \theta} \right) \text{ (approx.)}$$

[0024] It was surprisingly found that evaporator tubes 16 having features with certain dimensional ranges, defined in terms of critical parameters, offer improvements in heat transfer performance, reduced refrigerant pressure drop, increased burst strength, increased robustness of brazing process, and reduced heat exchanger mass per unit volume for evaporators. The critical parameters are identified as follows: number of ports per unit millimeter width (PPMW); port shape ratio (PS ratio); Non-dimensional gauge (NDG ratio); and Non-dimensional Corner Radius (NDC ratio). The formulas for the critical parameters for applications in evaporators 10 are provided in the Table 1 below:

Table 1

	Critical Parameters	Formula
1	No. of ports per millimeter width (PPMW)	$2/(a+b+t)$
2	Port Shape Ratio (PS ratio)	a/b
3	Non-dimensional gage (NDG ratio)	t/h
4	Non-dimensional corner radius (NDCR ratio)	$r_c/2t$

[0025] Figs. 4A-D are graphs showing the effects of the ranges of the critical parameters of the B-type evaporator tube 16 on the performance of the evaporator 10. Each of the graphs presents performance characteristics of evaporator 10 as illustrated by five curves labeled (1) through (5), corresponding to: (1) heat transfer performance, (2) refrigerant pressure drop, (3) burst strength of the tube, (4) the amount of braze contact across tube width for robustness of brazing process, and (5) heat exchanger mass per unit volume. Each of the respective critical parameters is denoted on the X-axis and the relative change in performance of the evaporator 10 is denoted on the Y-axis. The graphs show the relative changes in performance of the evaporator 10 with the corresponding changes in respective critical parameters.

[0026] Referring to Fig. 4A, for an evaporator tube 16 having a fixed width ($2w$), the theoretical minimum number of ports is 2 such that the evaporator tube 16 is only supported by the center wall 62 where $a = b = w$, $t \approx 0$. The theoretical maximum number of ports can only be limited by the thickness (t) of the channel wall 72 or material gage. If the number of ports is too large, the cross-section of the tube is filled with a large number of vertical channel walls 72 which decreases the flow cross-section significantly. For understanding the impact of variation of number of ports, different operating conditions were evaluated where the parameters including θ are kept constant. So effectively this means changing a and b by the same amounts. As the PPMW is increased from $1/w$ to $2/t$, the performance characteristic curves (1), (2), (3), and (5) increased while performance characteristic curve (4) decreased.

[0027] Referring to Fig. 4B, the PS ratio can be adjusted by changing both a and b , such that $(a+b)$ must be kept constant to keep constant number of ports. The port shape also changes with h . However, changing h affects other critical parameters. Changing port shape ratio relates to angle θ and the hydraulic diameter D_h , which will affect some of the performance characteristics listed above. As the PS ratio is increased from 0.0 to 1.0, the shape of the port changes from a triangular cross section to a rectangular cross-section. As shown in Fig. 4B, the performance characteristic curves (2) through (5) increased while performance characteristic curve (1) decreased.

[0028] Referring to Fig. 4C, the effect of gage is simpler to understand than that for other parameters. The requirement for higher gage comes typically from the corrosion and mechanical strength issue. However, the heat transfer is also affected by gage to some extent. At very low gage, the webs defining the channels 36 do not conduct heat well and as a result has low fin efficiency. It also reduces burst strength. As the NDG ratio is increased from 0.0 to 0.333, the performance characteristic curves (2), (3), and (5) increased, while performance characteristic curve (4) remain substantially unchanged and performance characteristic curve (1) initially increased and then dropped significantly.

[0029] Referring to Fig. 4D, the impact of corner radius (r_c) is more on the manufacturing aspect of the tube rather than on the product characteristics. Nevertheless, this feature will have secondary impact on the performance characteristics as depicted below. With larger corner radius, the definition of θ becomes less clear. The largest corner radius is when the web as formed by a channel wall 72 and immediate adjacent flange segments 68 takes the shape of "S". As the NDC ratio is increased from 0.0 to 0.5, the performance characteristic curve (1) slight dropped and then increased, performance characteristic curves (2), and (5) increased, while performance characteristic curve (4) remains unchanged,

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and performance characteristic curve (3) initially increased and then dropped.

[0030] Based on the performance shown on the graphs, it is desirable to for an evaporator 10 having dual banks of evaporator tubes 16 to have geometric features that corresponds to the critical parameters presented in Table 2 below:

Table 2

	Parameters Name	Working Range	Desired Range	Current Tube
1	No. of ports per millimeter width (PPMW)	1/w - 2/t	0.4 - 1.0	0.57
2	Port Shape Ratio (PS ratio)	0.0 - 1.0	0.05 - 0.6	0.39
3	Non-dimensional gage (NDG ratio)	0.05 - 0.333	0.11 - 0.21	0.186
4	Non-dimensional corner radius (NDCR)	0 - 1.0	0.1 - 0.5	0.288

[0031] For a typical evaporator tube 16 having a width of 17mm to 18 mm for use in a an evaporator 10 having dual banks of evaporator tubes 16, the following tube dimensions listed in Table 3 based on the above described critical parameters were found to offer significantly improved performance.

Table 3

Dimension	Value (mm)
w	8.8
h	1.4
t	0.26
a	0.9
b	2.3
# of ports	10
r_c	0.15
θ	41.5°
D_h	0.738

[0032] An evaporator 10 having dual banks of B-type evaporator tubes 16 with critical parameters as provided herein has the advantage of having improved heat transfer performance, reduced refrigerant pressure drop, increased burst strength of the tube, increased robustness of brazing process, and reduced heat exchanger mass per unit volume for evaporators. Evaporator tube 16 folded from a sheet of clad aluminum offers superior corrosion protection over that of extruded tubes. For evaporators having tubes folded of clad aluminum, the primary brazing alloy comes from the cladding. This cladding on the tube alone is sufficient to braze and adhere the primary components such as the evaporator tubes 16, external fins 22 and headers 12, 14 together. Some remaining components such as end-plugs of the headers and the crossover, which are much smaller in size, need to be clad as well, but the amount of clad required from these is minimal. Thus, it is a benefit to have unclad headers 12, 14 and cores 34 which offers both material cost saving and less complication in maintaining material quality and consistency. Also, this enables headers to be made by less costly extrusion process.

[0033] While this invention has been described in terms of an evaporator having dual banks of evaporator tubes and two- passes, it is not intended to be so limited, but rather only to the extent set forth in the claims that follow. The evaporator tube 16 as disclosed may be used for evaporators 10 having greater than two banks of evaporator tubes 16 and a plurality of refrigerant passes.

Claims

1. A folded evaporator tube comprising,
a unitary strip of heat conductive material having a thickness (t) folded into a cross sectional shape having:

a bottom wall with two opposing tube edges transitioning into a pair of top walls spaced from and substantially

parallel to said bottom wall defining an interior surface;
a pair of abutted central walls bent substantially perpendicularly out of said top walls and extends toward said bottom wall;
a corrugated portion extending substantially perpendicularly out of each of said central walls toward said corresponding tube edge;
wherein said bottom wall includes a width (2w),
wherein said corrugated portion includes alternating flange segments abutting said interior surface and channel walls connecting said alternating flange segments,
wherein at least one of said alternating adjacent flange segments includes a length (a) cooperating with adjacent said channel walls to define a channel having a width (b); and
a number of ports per millimeter width (PPMW) includes a range of 1/w to 2/t as defined by the equation:

$$\text{PPMW} = 2/(a+b+t).$$

2. The folded evaporator tube of claim 1, wherein the PPMW includes a range of 0.40 to 1.0.
3. The folded evaporator tube of claim 1, wherein the PPMW is 0.57.
4. The folded evaporator tube of claim 1, further comprising a Port Shape (PS) ratio having a range of 0.0 to 1.0 as defined by the equation:

$$\text{PS ratio} = a/b.$$

5. The folded evaporator tube of claim 4, wherein said PS ratio includes a range of 0.05 to 0.60.
6. The folded evaporator tube of claim 4, wherein said PS ratio is 0.39.
7. The folded evaporator tube of claim 1, wherein evaporator tube further comprising a height (h) measured from the bottom exterior surface to the top exterior surface of said evaporator tube, and a non-dimensional gauge (NDG) ratio range of 0.05 to 0.333 as defined by the equation:

$$\text{NDG ratio} = t/h$$

8. The folded evaporator tube of claim 7, wherein said NDG ratio includes a range of 0.11 to 0.21.
9. The folded evaporator tube of claim 7, wherein NDG ratio is 0.186.
10. The folded evaporator tube of claim 1, further comprising:

a corner radius (r_c) defined by the transition radius from said flange segments to said channel walls, and
a non-dimensional corner radius (NDCR) ratio range of 0.0 to 1.0 as defined by the equation:

$$\text{NDCR ratio} = r_c/2t.$$

11. The folded evaporator tube of claim 10, wherein the NDR ratio includes a range of 0.10 to 0.50, preferably the NDCR ratio being 0.288.
12. An evaporator assembly comprising,
a first header;
a second header;

at least two banks of evaporator tubes extending between and in hydraulic communication with said first and second headers;
wherein at least one of said evaporator tube is made according to anyone of claims 1 to 11.

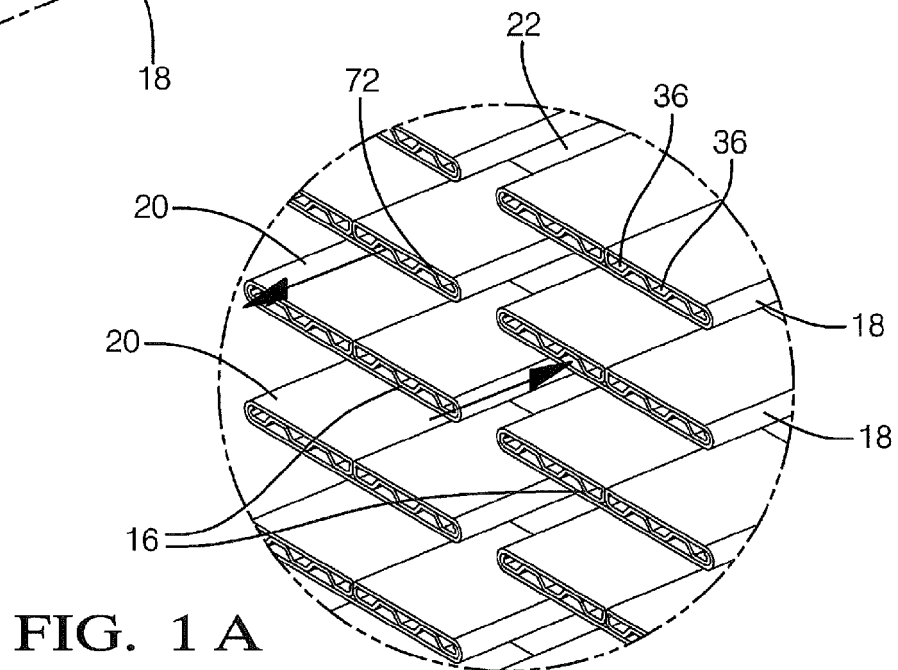
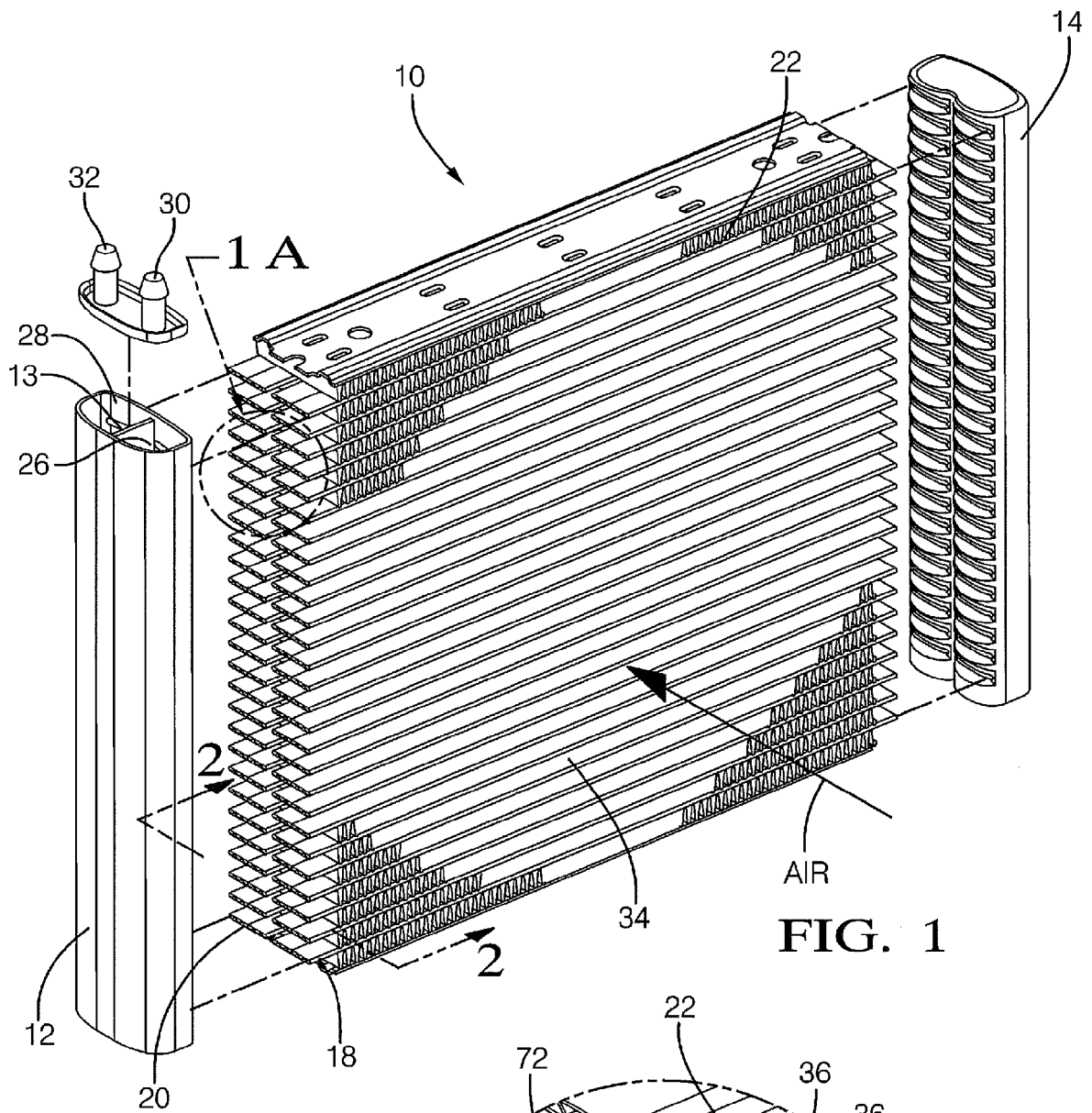
5 **13.** The evaporator assembly of claim 12, wherein the intersection of said channel wall with said interior surface defines an angle (θ) of 41.5°;
w = 8.8 mm;
h = 1.4 mm; and
t = 0.26 mm.

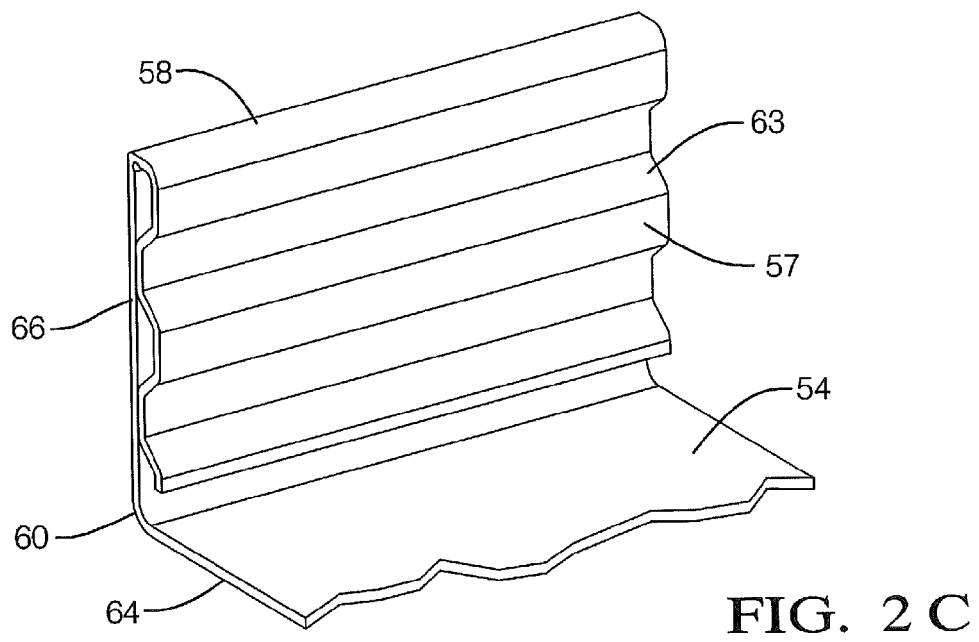
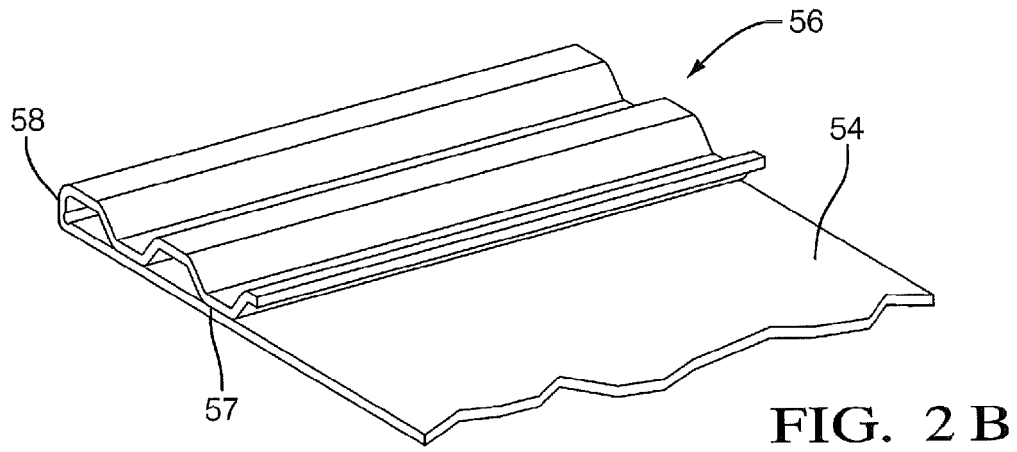
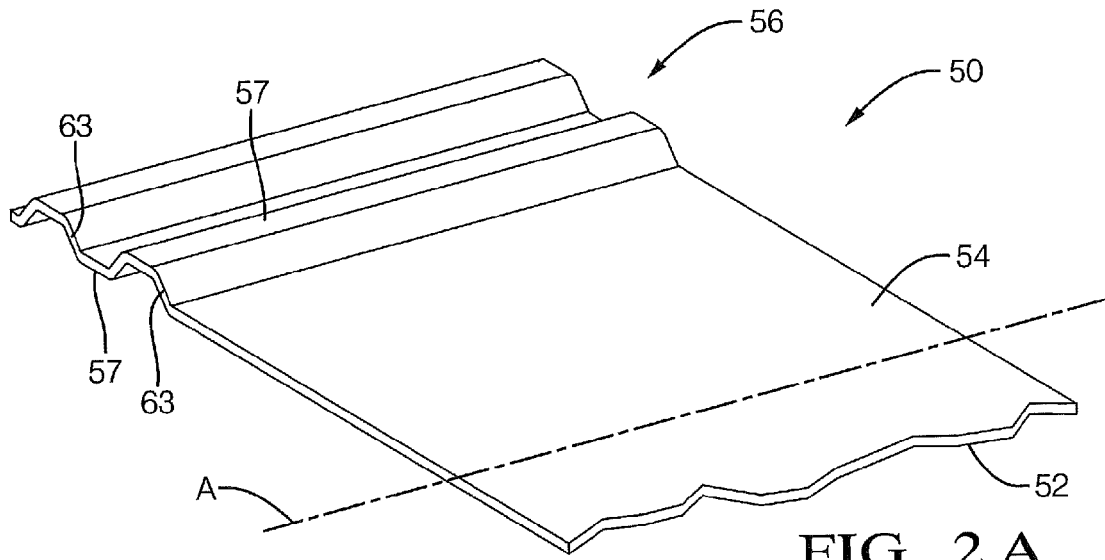
10 **14.** The evaporator assembly of claim 13, wherein:

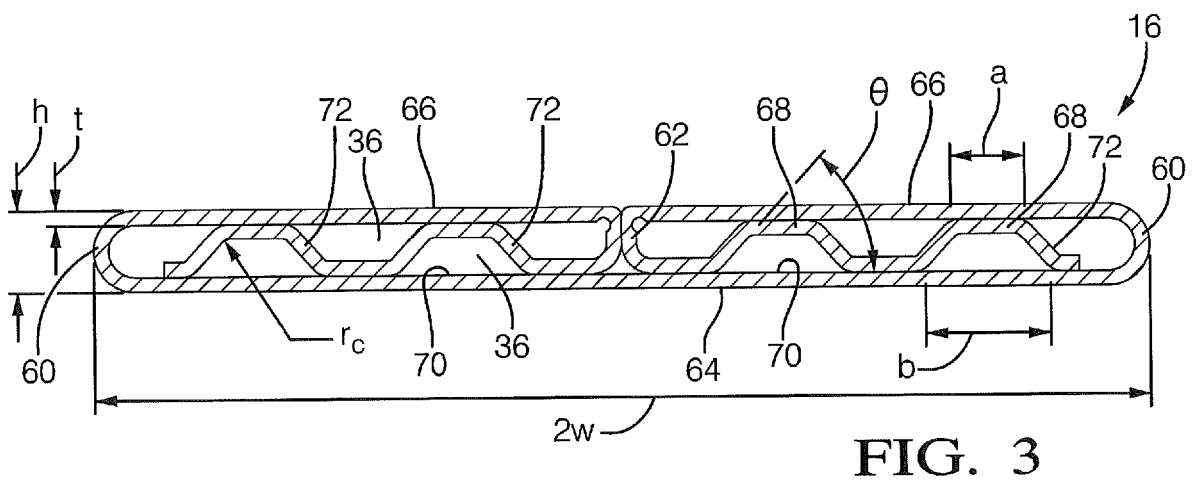
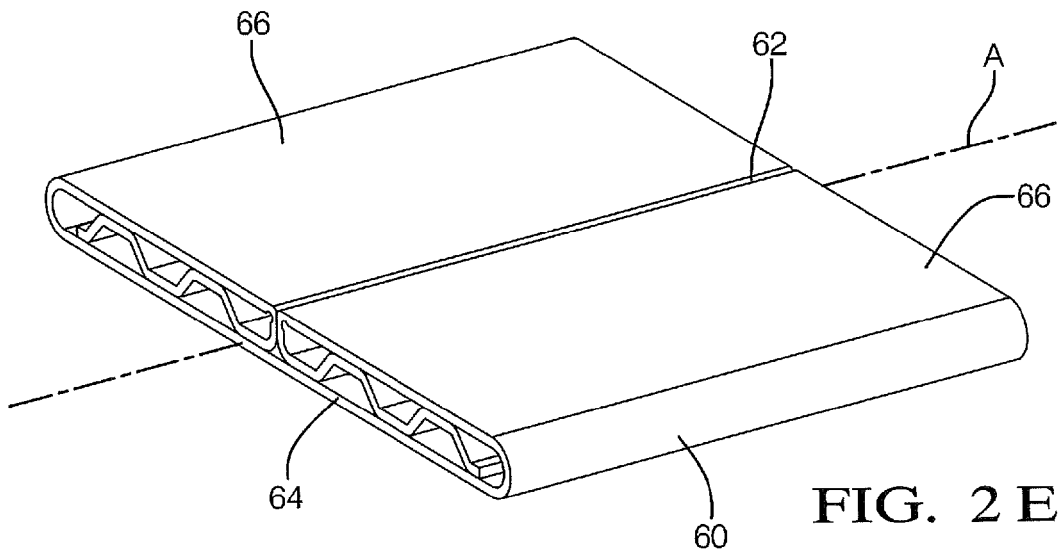
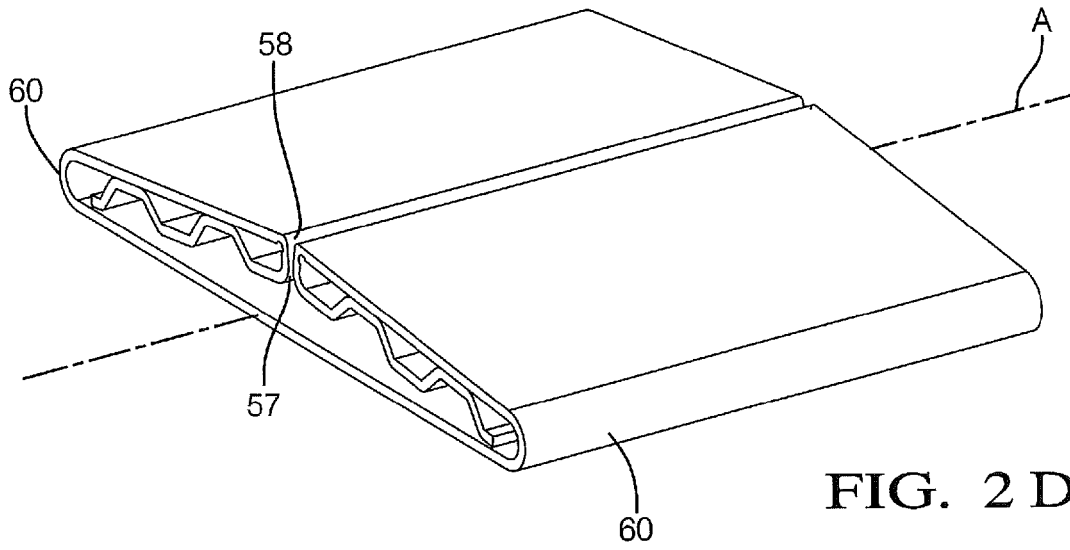
a = 0.9 mm;
b = 2.3 mm; and
15 $r_c = 0.15$.

15. The evaporator assembly of claim 14, further including a Hydraulic Diameter (D_h) of 0.738 as defined by the formula:

$$D_h = \frac{2(a+b)(h-3t)}{(a+b + \frac{2(h-3t)}{\sin \theta})}, \text{ where } \theta = \tan^{-1}\left(\frac{2h-6t}{b-a}\right)$$







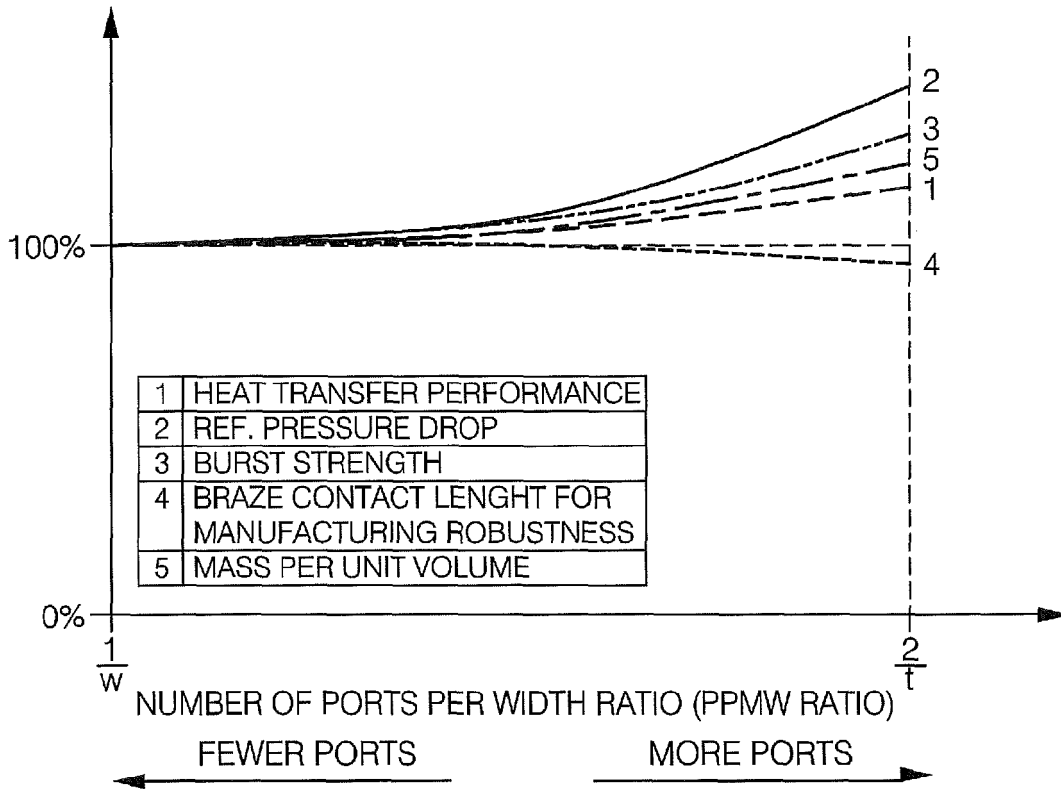


FIG. 4 A

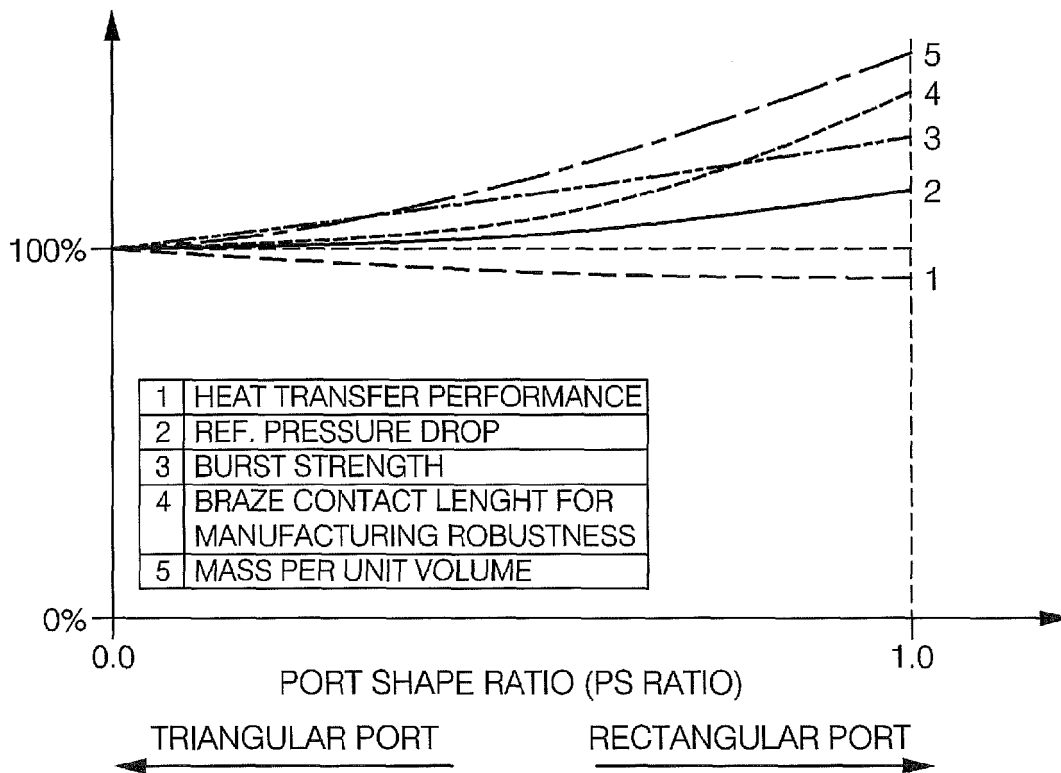


FIG. 4 B

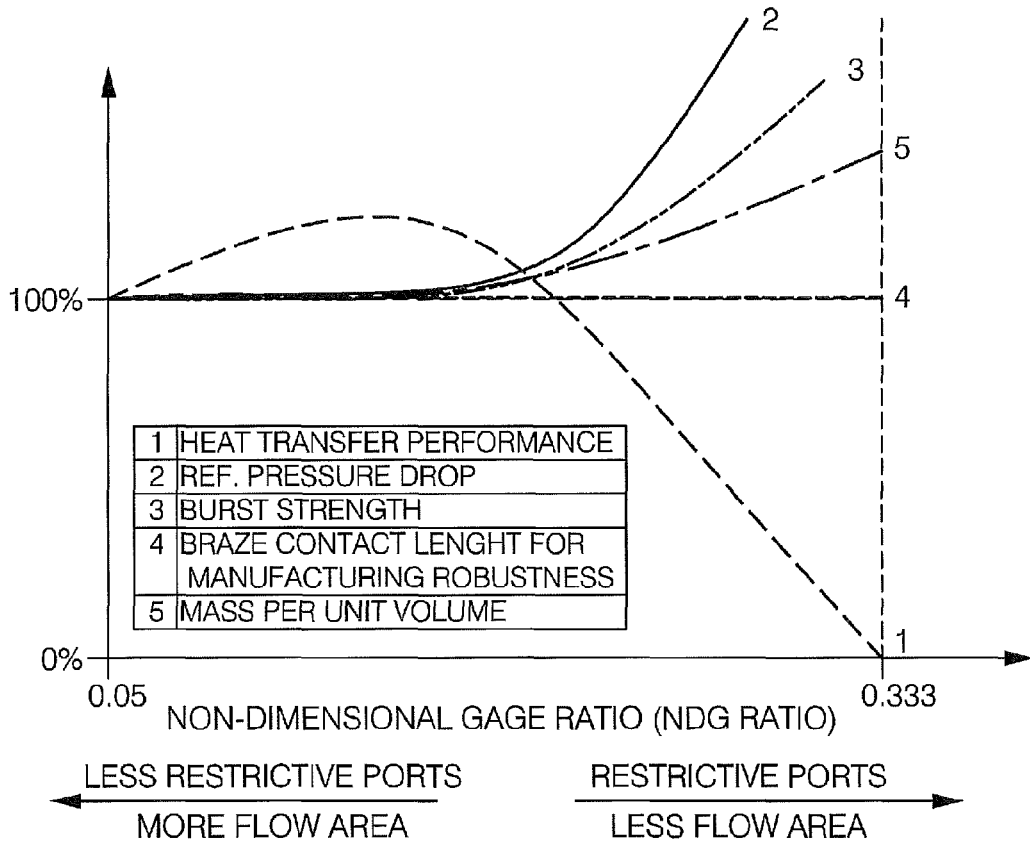


FIG. 4 C

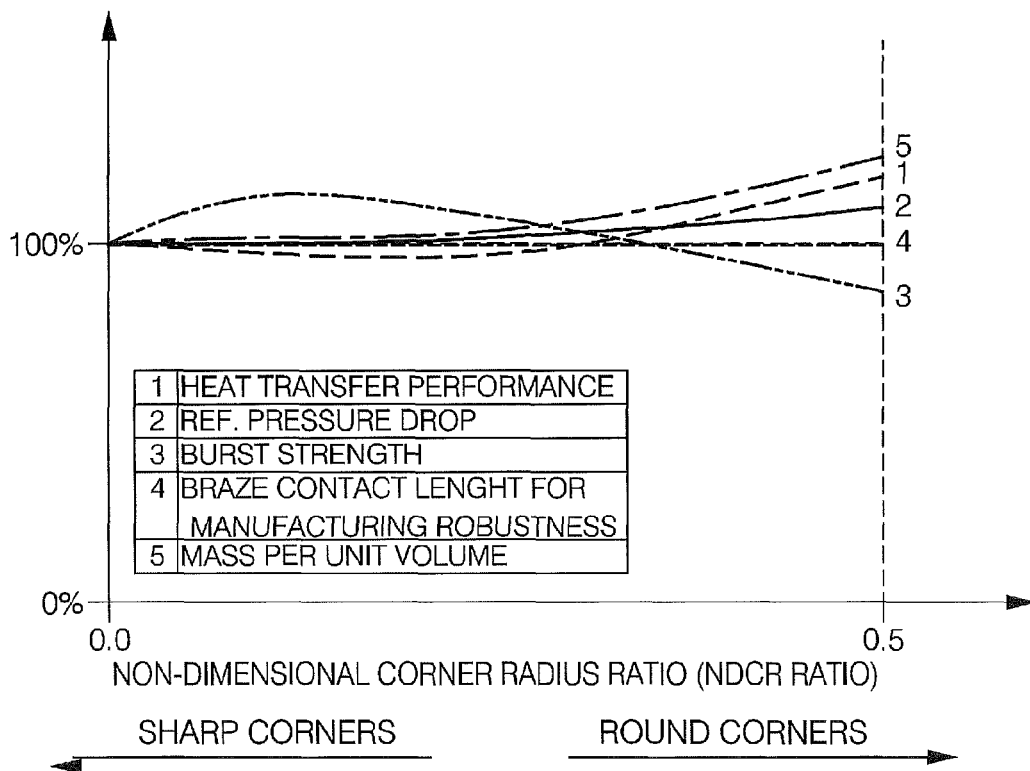


FIG. 4 D