United States Patent [19]

Kun

[11] **3,845,814**

[45] **Nov. 5, 1974**

[54] **FINNED PRIMARY SURFACE HEAT EXCHANGER**

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[73] Assignee: **Union Carbide Corporation,** New York, N.Y.

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[21] Appl. No.: **270,259**

[52] **U.S. Cl.** **165/148,** 29/157.3

[51] **Int. Cl.** **F28b 1/14**

[58] **Field of Search** 165/130, 131, 151–153, 165/166, 170, 148

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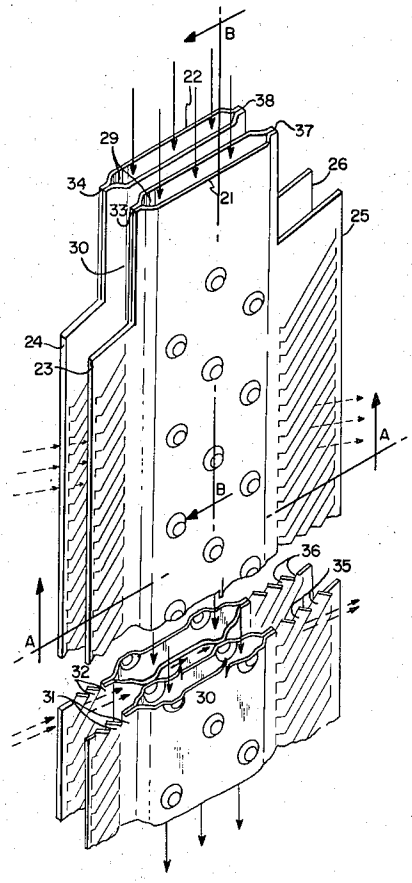
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Primary Examiner—Charles J. Myhre
Assistant Examiner—Theophil W. Streule, Jr.
Attorney, Agent, or Firm—John C. LeFever

[57] **ABSTRACT**

A heat exchanger composed of primary-surface heat exchange channel elements, each element being formed and bound by a thermally conductive material and having slotted secondary-surface fins disposed substantially along its longitudinal length. The fins, along with the slatted apertures on the fins, are oriented to provide a substantial through-flow area for an external fluid which in the operating mode will flow through passages formed between adjacent channel elements.

10 Claims, 21 Drawing Figures



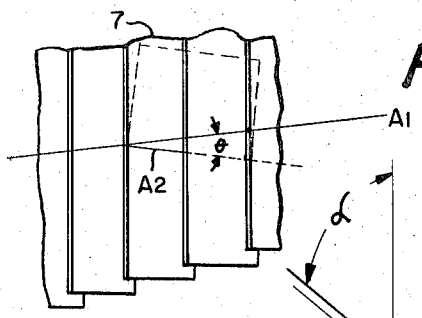


FIG. ID

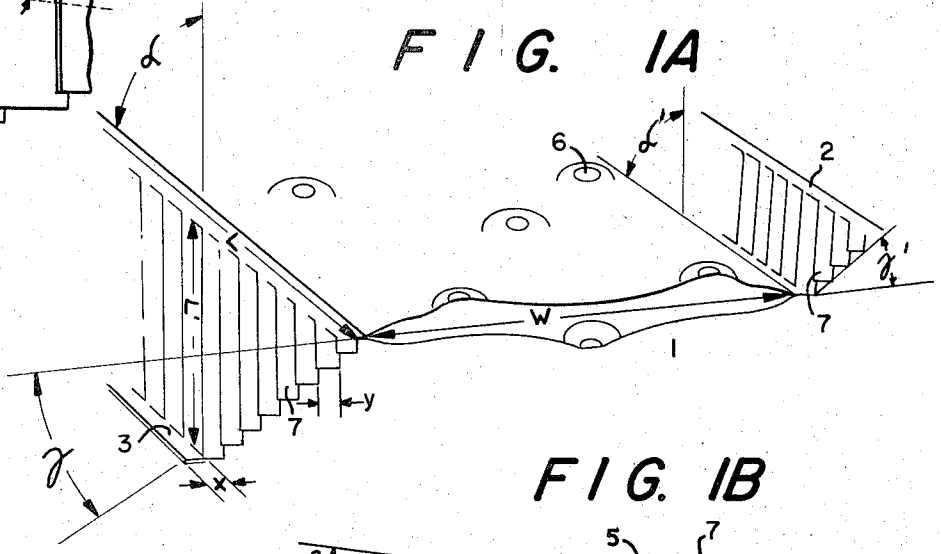


FIG. IA

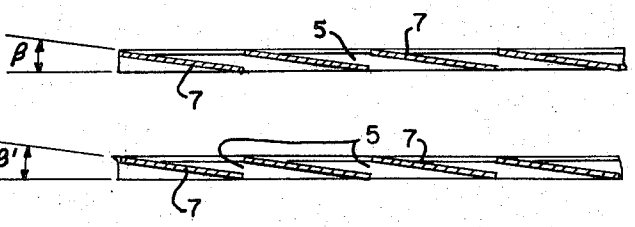


FIG. IB

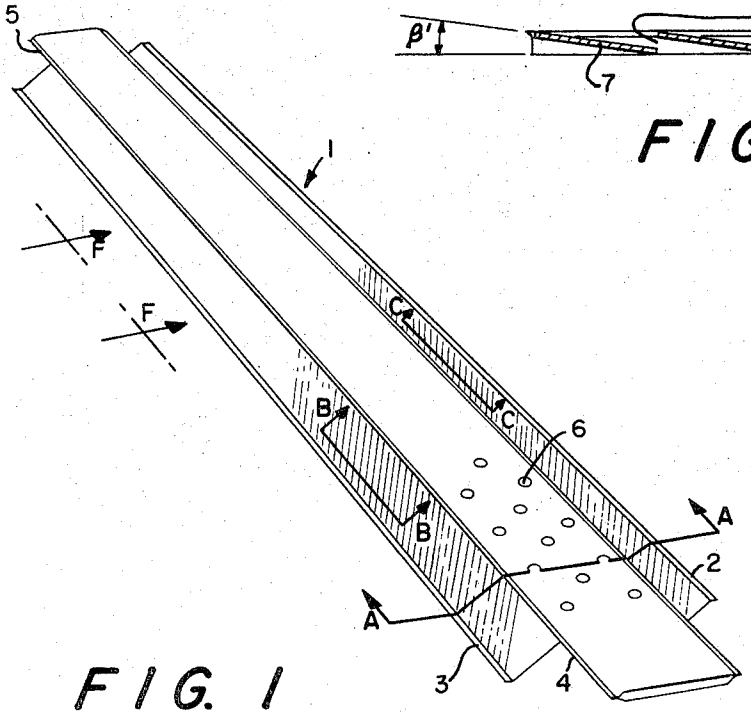


FIG. 1

FIG. IC

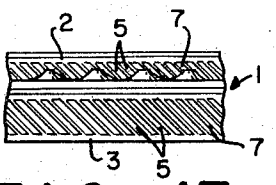


FIG. IF

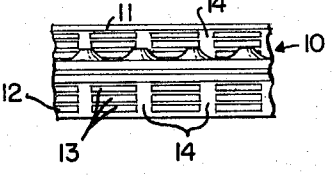


FIG. IE

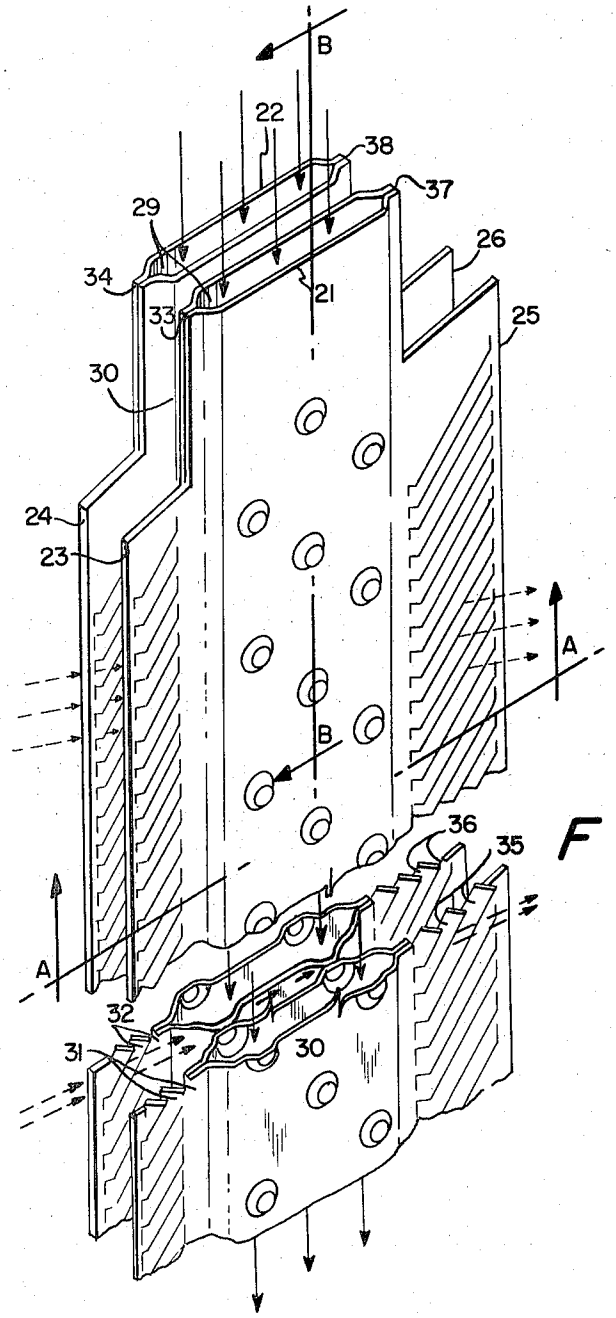


FIG. 2

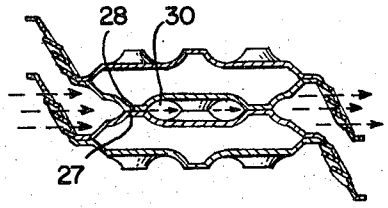


FIG. 2A

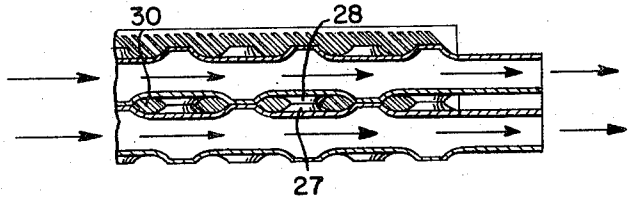


FIG. 2B

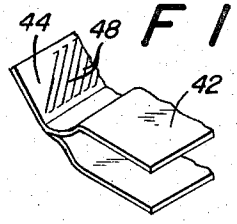


FIG. 3A

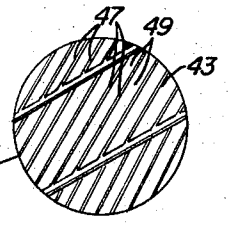


FIG. 3B

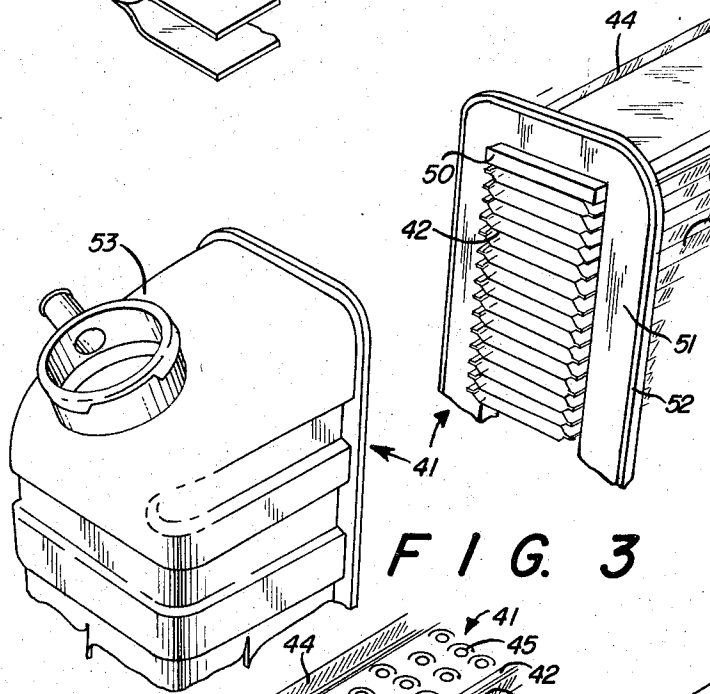


FIG. 3

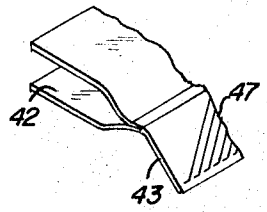
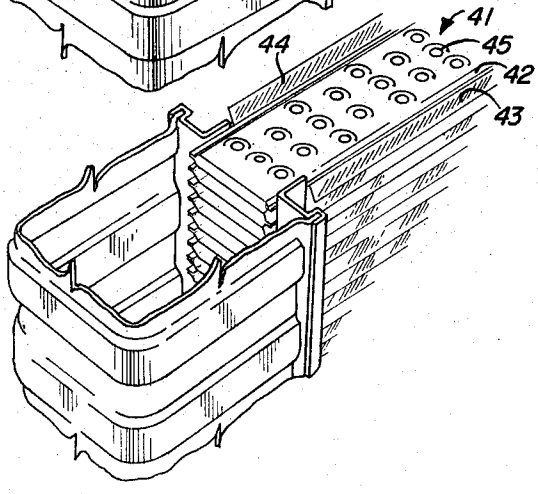


FIG. 3C



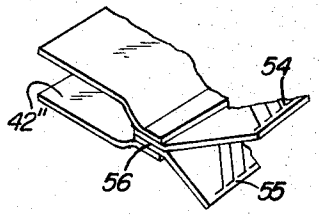


FIG. 3D

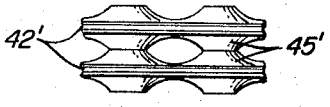


FIG. 3E

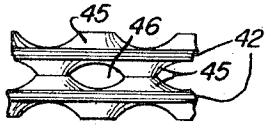


FIG. 3F

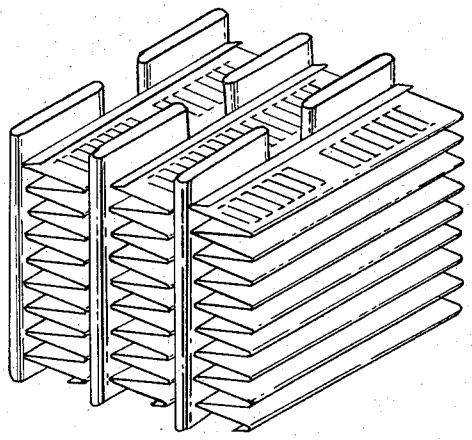


FIG. 4

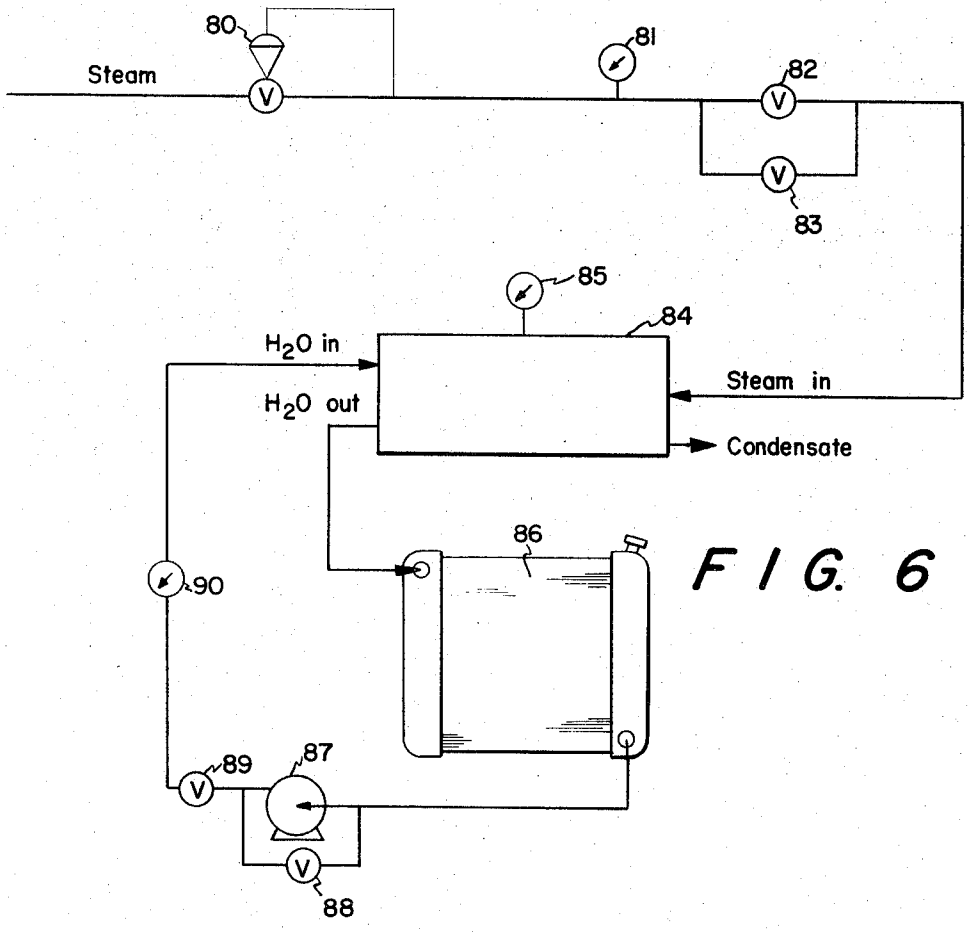
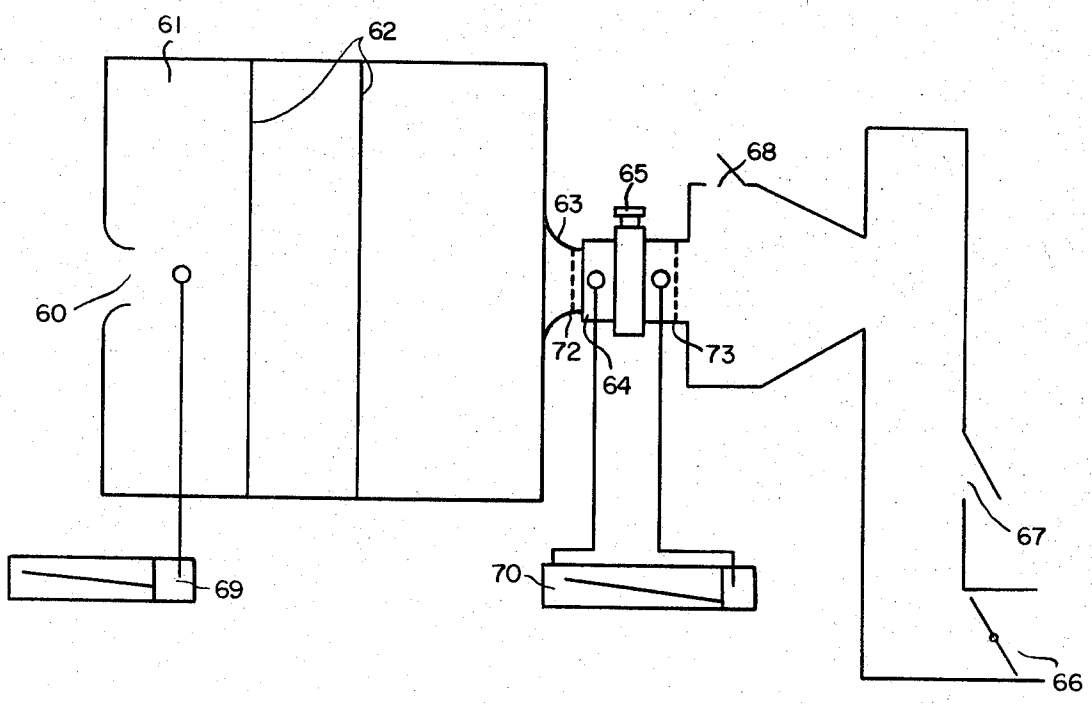

80
81
82
83
Steam
85
84
H_2O in
H_2O out
Steam in
Condensate
86
90
89
87
88

FIG. 6

FIG. 5



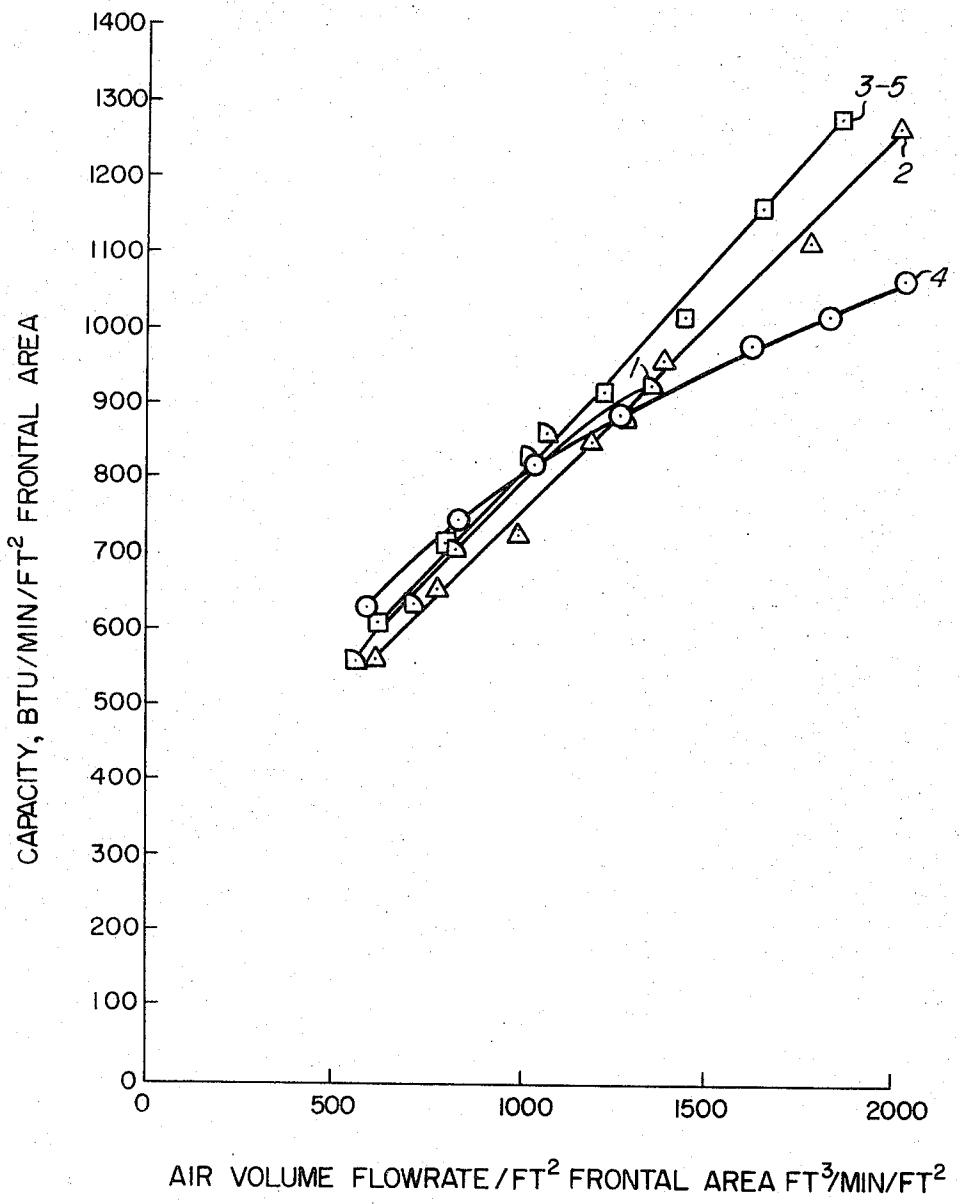

1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
100
0
0
500
1000
1500
2000
CAPACITY, BTU/MIN/FT² FRONTAL AREA
AIR VOLUME FLOWRATE/FT² FRONTAL AREA FT³/MIN/FT²
3-5
2
1
4

FIG. 7

FINNED PRIMARY SURFACE HEAT EXCHANGER

FIELD OF THE INVENTION

This invention relates to a heat exchanger composed of an array of primary-surface channelized elements each having secondary-surface slotted fins longitudinally disposed along the fore and aft sections of the channels as defined by the directional flow of a coolant medium through passage formed between adjacent channels.

BACKGROUND OF THE INVENTION

There is an increasing need in industry for compact, light weight heat exchangers capable of transferring large amounts of heat from one medium to a second medium. For example, the increased use of superconductive devices and maser applications have placed a demand for the development of heat exchangers for cryogenic environments. The need for reliable, rugged and compact heat exchangers, for operation in systems at ambient temperatures and above, has also increased. One of the major industries that is constantly searching for a compact, light weight heat exchanger for its combustion engines, is the automobile industry. Various types and styles of radiators have been designed, such as the individually finned round tubes, the hexagon-shaped air tubes with water passages between the tubes, and the flat dimpled water tubes with air flow therebetween. The pre-1942 automobile engines were designed to deliver between 50 and 125 horsepower, and radiators designed for use with them were designed to operate close to atmospheric pressure. A simple solder joined finned-copper constructed radiator was therefore sufficient to cool the low horsepower engine of the automobile without much of a threat of overheating. Various copper radiators having cup-like or frustroconical surface projections have been designed during this pre-1942 period but the finned copper radiator proved more successful and suitable for automobile applications.

The automobile industry, however, in the post-1945 era embarked upon the design of high power rated engines while simultaneously attempting to compact them as much as possible. This dual design approach coupled to the employment of improved lubricants resulted in an internal combustion engine capable of operating at high permissible temperatures. To satisfy the heat transfer requirements of such compact high power rated engines, and to avoid loss of coolant, the tube and fin copper radiators were designed to operate under pressure so as to increase the boiling temperature of the coolant. However, within the last several years, additional power operated equipment, such as air conditioners and the like, was added to the automobile thereby further increasing the demands on the internal combustion engine and consequently the duty of the heat rejection system. This has necessitated the designing of present day radiators to operate at pressures as high as 15 psig. to prevent coolant loss and overheating. The operating temperature of the automobile engine is anticipated to rise further in the near future thereby necessitating a heat transfer system operable with existing coolants under still higher pressure conditions. The conventional type finned-copper radiator will not perform satisfactorily in an increased temperature environment due to the low stress characteristics

inherent in soft solder at high temperatures, such solder being the securing medium between the tubes and fins of the radiator. In addition, the steady increase in the price of copper is causing copper to become an undesirable material for radiator applications from an economical standpoint.

In heat exchange applications requiring pressure-bearing walls as the primary heat exchange surface, new designs have recently been developed whereby thermally conductive thin material can be fabricated into channels having isocompression supports or trucated conical wall-supporting projections. The channels can be aligned in juxtaposed relationship to form a compact light weight heat exchanger having a first set of passages defined by, and bound within, the conductive walls of each channel, and a second set of passages defined by, and disposed between, the juxtaposed channels so that a first medium can be fed through one set of passages while a second cooler medium can be fed through the other set of passages thereby effecting a heat exchange between the mediums without having the mediums intermix. The specific details of these compact, economical channelized heat exchangers can be found in applicant's copending patent application Serial No. 189,659 titled "Primary Surface Heat Exchanger and Manufacture Thereof", now U.S. Pat. No. 3,757,856, granted Sept. 11, 1973 and in copending patent application Ser. No. 189,509 titled "Primary Surface Heat Exchanger," by L. C. Kun and J. B. Wulf, now U.S. Pat. No. 3,757,855 granted Sept. 11, 1973. Also in copending application Ser. No. 344,429 titled "Cross Flow Heat Exchanger" now U.S. Pat. No. 3,810,509 granted May 14, 1974.

The present invention is directed to an improvement on all types of channelized primary-surface heat exchangers, whether stayed by means of numerous support members or by wall-supporting projections, the improvement being the addition of one or more slotted fins disposed along the longitudinal edge of each channel in such a manner that it provides an efficient compact heat exchanger that is economical to construct primarily because it offers a substantial reduction in weight through optimum metal utilization.

SUMMARY OF THE INVENTION

The heat exchanger of this invention basically comprises at least one primary-surface heat exchange channel element formed and bound by at least one thermally conductive metal or plastic walled material, said element having an entrance opening and an exit opening, the improvement which comprises the addition of at least one secondary-surface heat exchange slotted fin disposed substantially along at least one longitudinal section of the channel element and wherein said slotted fin has a multitude of slatted apertures arranged in a louver configuration. The fins primarily extend outward from the longitudinal segment of the element at an angle made with a plane containing the maximum dimensional width line of, and the longitudinal dimensional length line of, said element to provide a multitude of slatted apertures in said fin when viewed in a direction parallel to the plane so defined. At least two of such channels, when aligned in juxtaposed relationship, or otherwise appropriately stayed, will form a heat exchanger having a first set of passages defined by, and bound within, the conductive walls of each channel, and a second set of passages defined by, and disposed

between adjacent channels so that a first medium can be fed through one set of passages while a second cooler medium can be fed through the other set of passages thereby effecting a heat exchange between the mediums without having the mediums intermix.

For circular channels or tubes, the maximum dimensional width line shall mean the diameter of the tube taken along a line containing the fin attachment point on the tube. When a tube has more than one fin, the maximum dimemsional width line shall mean the diameter of the tube taken along a line where the medium arithmetic mean distance of fin attachment points onto one side of the line from one half of the tube will be equal to the minimum arithmetic mean distance of the fin attachment points onto the other side of the line from the other half of the tube. For an array of two or more circular tubes, the maximum dimensional width line shall mean the diameter of the tube taken in plane perpendicular to a plane containing the center points of two adjacent tubes and the longitudinal axis of the tube.

The term primary-surface heat exchanger refers to heat exchangers wherein substantially all the material which conducts heat between two media comprises the walls separating the two media. In contrast, secondary surface heat exchangers contain a substantial amount of material in the form of fins which do not separate the media but are contacted on virtually all surfaces by a single medium. In addition, in primary surface heat heat exchanger applications wherein a pressure difference exists between the two media of the system, substantially all of the heat exchanger material is stressed pneumatically. Stated another way, primary surface heat exchanger refers to a heat exchanger consisting primarily of plates or sheets and having no separate or additional internal or external members, such as fins, so that the exchanger is constructed of plates or sheets each side of which is in contact with a different fluid, and heat transfer is substantially and directly between the plates and the fluid.

The term channel element refers to any enclosed longitudinal member having an entrance opening and an exit opening, and formed and bound by at least one thermally conductive metal or plastic wall material. Channels elements could be of the isotension, isocompression, or truncated projected type as specified in the above-mentioned patent applications, or be of the outwardly dimpled tubular types as disclosed in U.S. Pat. Nos. 1,540,913 and 1,413,163, or of the outwardly dimpled rectangular passages as disclosed in U.S. Pat. Nos. 1,641,148 and 2,354,865. The channel elements could be spaced apart by wall projected supports, by wire members, by epoxy or epoxy coated washers, by appropriately bent channels or alternately bent channels, by metal or plastic strips, or by any other conventional means.

The slotted fins can be disposed on the longitudinal aft and fore sections of the channel as defined by the direction of a medium flow through passages formed between adjacent channels, or otherwise defined as the remote sections of the channel as measured from a plane containing the center points of two adjacent channels and the longitudinal axis of the channel. The elongated slots or slatted apertures can be disposed perpendicular to the longitudinal length of the channel, but, however, when so arranged, the fins would have to be bent at a fin angle γ of about 90° with respect to a plane containing the maximum dimensional width line of, and the longitudinal dimensional length line of, the channel, if the slats in the fins are to be aligned to provide a minimum air flow path on the slats for a medium flowing through passages formed between adjacent channels, since such a design would provide maximum heat transfer. If substantial overlapping of the fins is to be avoided in an array of channels aligned parallel to the direction of flow of a medium between the channels, then the width of a 90° orientated fin will dictate the spacing between the channels. Since compactness is a desirable characteristic in a heat exchanger, the fin should preferably be disposed at an angle γ of less than 90° with respect to the plane containing the maximum dimensional width line, and longitudinal dimensional length line of the channel. On the other extreme, this fin angle γ can be 0° even though the fins would be disposed such that the slatted apertures would be substantially aligned parallel to the directional flow of a medium between adjacent channels, since in certain respects, this configuration may be desirable from an economical or mechanical viewpoint. Thus the fin angle γ, formed between the plane of the fin and the plane containing the maximum width and longitudinal lines of the channel, can be between about 0° and 90°, but preferably between about 0° and about 60°. For a given fin bending angle γ, the slot angle α, formed by the longitudinal length line of the channel and the longitudinal length line of the slot, and the slat angle β, formed by the plane of the fin and the plane of the slat member between adjacent slatted apertures, would have to be chosen to provide substantially minimum "fin frontal area" and thus substantially maximum area of the slatted aperture orientated normal to the flow of a medium through passages between adjacent channels. "Fin frontal area" is defined as the area of the projection of the entire array of fins onto a plane normal to the direction of flow of a medium through passages formed between adjacent channels. The slot angle α can vary broadly from 0° to 180° as measured clockwise from the longitudinal axis of the channel to the longitudinal length line of the slot. Preferably the slot angle α should be $90° \pm 45°$ for most applications since this would provide a sufficient uninterrupted flow path on the fin extending outward from the attached segment points on the channel. The slat angle β can vary between about 15° and about 90° for most applications although an angle between about 30° and about 60° would be preferable since it would provide sufficient slatted apertures which can be obtained through one shot of a die or the like, thus minimizing the manufacturing cost. An angle β of less than about 15° would be insufficient since it would not provide an adequate opening between adjacent slats due to finite material thickness of the fins.

The width of the slats of the louver constructed fins should be smaller than about 0.250 inch for most applications, although a slat width of less than 0.10 inch would be desirable. For the lower limit, the slat width should be at least 0.02 inch and preferably at least about 0.03 inch. The smaller the slat width, the smaller the flow path length on the slat for a medium passing through the apertures in a louver constructed fin. Usually finite fin thickness limits the minimum width dimension for the slat of a louver constructed fin.

As specified above, the orientation of the fins, slats and slatted apertures can be defined by the angles γ, α,

and β. The interrelation of these three angles can be expressed as follows:

$$\sin \theta = \cos \beta \sin \gamma - \cos \gamma \sin \beta \cos \alpha \quad (1)$$

where θ is the angle of approach defined as the angle between a first line which is parallel to the plane containing the maximum dimensional width line and the longitudinal dimensional length line of the channel, and perpendicular to the channel's longitudinal length line, and a second line formed by the intersection of a plane onto the surface of the slat where said plane is normal to the surface of the slat and contains said first line as defined above.

Thus we see that for any angle γ, α or β, the remaining two angles can vary widely for a specific angle of approach θ for a particular heat exchange operational mode. By first selecting an angle of approach θ, and then selecting a fin bending angle γ to accommodate a specific array of channels in an overall heat exchanger configuration, the slat angle β and the slot angle α can be substantially varied, and still satisfy the solution of Equation I. However, once one of these angles is selected, while maintaining γ and θ constant, the other angle becomes fixed according to the Equation I. It is thus shown that γ, α and β can vary widely but they are interrelated in accordance with Equation I which defines the angle of approach which can be selected for a particular application. An angle of approach θ of about 60° or less would be acceptable for most applications employing the fin arrangement of this invention, although an angle of approach θ of about 0° to about 45° would be preferable. As can be seen by Equation I, once you choose one of the angles γ, α or β, while keeping the angle of approach θ constant, then the other two angles can not be selected independently. Consequently, to provide a desirable fin frontal area for a particular application, the angle of approach θ, fin bending angle γ, the slat angle β and the slot angle α, all of which are related in accordance with Equation I, should be picked so that the plane of each slat will be substantially aligned such that the air flow path across the slats will be substantially minimized.

The slotted fins can be added both fore and aft of the channel as defined by the direction of a coolant medium through passages formed between adjacent channels, or otherwise defined as the remote sections of the channel as measured from a plane containing the center points of two adjacent channels and the longitudinal axis of the channel. The width of the fore and aft fins, measured normal to the longitudinal length of the channel, can advantageously be of a different size as discovered by James B. Wulf. The maximum dimensional width line of the primary-surface heat exchange channel can preferably be between about 0.5 inch and 3.0 inches for most applications and the width of the secondary-surface heat exchange fin should be less than about 0.60 inch and preferably wider than 0.10 inch. For automobile radiator applications, the width of the primary-surface heat exchange element can vary between about 0.75 inch and about 2.0 inches and the width of the secondary-surface fin can vary from about 0.2 to about 0.5 inches.

The wall material of the primary surface heat exchange channel need not be highly conductive, although preferably it should be, and can be selected from at least one of the groups consisting of metals, metal alloys, metal clads, ceramics, plastics (such as Mylar), plastic-coated metals and the like. The criteria of the material selected for the primary surface of the channel element is that it be sufficiently thermally conductive so that as a hot medium is passed through the channel, the heat of the medium will be conducted through the wall of the channel to a cooler medium external of, and adjacent to, the channel which can absorb the heat thereby successfully effecting a heat transfer between the mediums without intermixing of said mediums. In addition, the material should preferably be such that it will transmit heat laterally to the sections where the fins are attached without the need for a large temperature gradient existing through the material. Materials such as aluminum, copper, steel, zinc, brass, magnesium, titanium, beryllium and Mylar, and the alloy of such material, where applicable, are suitable for this application. The wall material of the secondary-surface heat exchange fin segment must be conductive since its efficiency is dependent upon its ability to transfer the heat localized in one area; i.e., in the primary-surface walls; to a cooler area; i.e., the area external of, and away from, the primary-surface walls; so that such heat can be absorbed by the cooler environment not in contact with the primary-surface walls. Suitable fin material would include the base metals and the alloys of aluminum, copper, carbon steel, zinc, magnesium, and beryllium.

An additional modification of the finned primary-surface heat exchanger of this invention would be an arrangement whereby two or more fins would be disposed from the same longitudinal segment on the channel so as to provide a multiple fin configuration.

The longitudinal mating edges of a channel, along with the fins, whether or not such fins are an extension of the primary-surface wall material or are composed of a separate material, can be sealed by conventional techniques, i.e., soldering, brazing, welding, or with an adhesive filled lock-joint to make the channel leak tight. An array of channels so formed can be suitably spaced apart by wall-supporting projected buttons or the like, or by conventional type stayes, to produce a compact, efficient heat exchanger. A pressurized medium, such as hot water, could then be passed through the channels while a coolant medium, such as cool air, could first pass through the slatted apertures on the louver type front fin and then pass between, and contact the outer surface of, the primary-surface walls of the channel before exiting through the slatted apertures on the rear fin thereby effecting a transfer of heat between the mediums. The fin attached channel could be appropriately bent or curved along its longitudinal length, or shaped into any curvilinear configuration, to fit the need of any particular environment.

The slats and/or the fins could also be slightly curved to provide an optimum type geometry for the heat exchanger in a particular end use application.

In a cross-flow heat exchange operational mode, the mediums can be fed through their respective passages in a heat exchanger, made according to this invention, in a perpendicular relationship or at some other angle relationship therebetween.

DESCRIPTION OF THE DRAWINGS

FIG. 1 — Isometric view of a heat exchange channel element of this invention.

FIG. 1A — An enlarged sectional view of the heat exchange element of FIG. 1 taken along line 1A—1A.

FIG. 1B — Sectional view of the heat exchange element of FIG. 1 taken along line 1B—1B.

FIG. 1C — Sectional view of the heat exchange element of FIG. 1 taken along line 1C—1C.

FIG. 1D — An enlarged view of slats of the heat exchange element of FIG. 1A.

FIG. 1E — Alternate embodiment of the heat exchange element of FIG. 1.

FIG. 1F — Sectional view of the heat exchange element of FIG. 1 taken along line 1F—1F.

FIG. 2 — Isometric view of heat exchange channel elements having fore and aft fins.

FIG. 2A — Sectional view of the heat exchange channel elements of FIG. 2 taken along line 2A—2A.

FIG. 2B — Sectional view of the heat exchange channel elements of FIG. 2 taken along line 2B—2B.

FIG. 3 — Isometric exploded view of an automobile radiator employing the heat exchange channel elements of this invention.

FIG. 3A — View taken of the longitudinal edge of a heat exchange channel element of FIG. 3.

FIG. 3B — Side view of the heat exchange elements of FIG. 3.

FIG. 3C — View taken of the longitudinal edge of a heat exchange channel element of FIG. 3.

FIG. 3D — Alternate embodiment of the heat exchange elements of FIG. 3.

FIG. 3E — Alternate embodiment of the heat exchange elements of FIG. 3.

FIG. 3F — Side view of heat exchange elements of FIG. 3 shown without the fins.

FIG. 4 — Isometric view of conventional automobile radiator.

FIG. 5 — Schematic of test apparatus for checking automobile radiators.

FIG. 6 — Schematic of test apparatus for supplying a heated medium through an automobile radiator.

FIG. 7 — Plot of Capacity vs. Volume flowrate for several radiators.

The performance of any heat exchange channelized element can be improved by adding slotted fins along the longitudinal length of the element according to this invention. FIG. 1 shows a thin wall isocompression channel 1 having a front or fore fin 2 and a rear or aft fin 3. The details of the isocompression channel, including the wall supporting projections 6, are disclosed in my copending patent application Ser. No. 189,659 now U.S. Pat. No. 3,757,856 granted Sept. 11, 1973. Fins 2 and 3 extend substantially along the longitudinal edges of channel 1 leaving finless segments 4 and 5 at opposite ends of channel 1. The finless segments 4 and 5 are designed so that when an array of channels are superimposed in touching relationship, a vertical support or retainer member 51 can be positioned in touching relationship with the edges of said segments 4 and 5 as shown in FIG. 3. An enlarged sectional view of channel 1, FIG. 1A, and FIG. 1B, show the various angles γ, α, and β at which fin 3, along with the slats 7 of the louver type arrangement in fin 3, should be orientated so as to provide a multitude of slatted apertures 5 arranged so that the substantially maximum areas of said apertures 5 are aligned perpendicular to the longitudinal axis of channel 1. For a given set of values of γ and α, maximum aperture area, as viewed from the flow path of an external medium, will correspond to minimum flow path lengths across slats 7. Fin bending angle γ is the angle formed between the plane of the fin 3, and the plane of the channel continuing both the maximum dimensional width line W and the longitudinal dimensional line L of channel 1. To minimize momemtum loss in the air flow, the unslotted tip section X (which is provided for structural strength) should preferably be parallel with the plane of the channel defined above, or to the directional flow of an external medium. The slot angle α is defined as the angle formed between the longitudinal dimensional line L′ of the slot, and the longitudinal dimensional line L of channel 1. The slat angle β can best be illustrated from FIG. 1B where it is shown as the angle formed between the plane of the slat 7 and the plane of the overall fin 3. As stated above, the sangle γ, α, and β can vary widely depending upon the angle of approach θ desired. As stated above, and shown in FIG. 1D, the angle of approach θ is defined as the angle formed between a line A_1 which is parallel to the plane containing lines L and W and perpendicular to the line L, and a line A_2 formed by the intersection of a plane on the surface of the slat, said plane being normal to the plane of the slat and containing line A_1. Aperture area, as viewed from the flow path of an external medium, approaches a maximum only when the angle of approach θ has low values approaching 0°. Since the heat transfer for a slotted fin is both a function of the width "y" of the slats 7, and the length of the slatted apertures L′, it would be advisable to first select the slot angle α to optimize the heat transfer characteristics of the fin. The slat angle β could next be selected based on the material thickness of the fin and on the manufacturing feasibility of the fin as based on mechanical and economical considerations. Fin bending angle γ could then easilty be determined for an angle of approach θ suitable for a particular heat rejection system using Equation 1. As stated previously, fin bending angle γ, slot angle α, and slat angle β, can be varied as long as they satisfy Equation 1 for a selected angle of approach θ. By determining any two of the angles γ, α and β, and having a desired angle of approach θ, the third angle can be found using Equation 1.

FIGS. 1A and 1C show fin bending angle γ', slot angle α' and slat angle β' for front or fore fin 2. Note that the fin bending angle γ' is shown as the negative of fin bending angle γ of fin 3. This is not necessary and both angles may be positive or negative depending on the direction of α and α', and β and β'. Likewise, slot angle α' and slat angle β' could be the same or different than angles α and β for fin 3. Any artisan can determine when and if it would be desirable to have different angles of γ', α' and β' for the front fin 2 as opposed to angles γ, α and β for rear fin 3. Preferably, the angles of the front and rear fins including the angle of approach for each fin, should be the same. Economy in the fabrication of the fins alone could dictate the use of the same angles for both front and rear fins.

FIG. 1F shows an enlarged side view of the finned heat exchange channel 1 of FIG. 1. Fins 2 and 3 are shown with slatted apertures 5, each formed between slats 7. FIG. 1E shows an alternate embodiment of a heat exchange element 10 having fins 11 and 12 with slots 13 parallel to the longitudinal length of element 10. The slots 13 can not extend substantially along the length of fins 11 and 12 since such a structure would not provide sufficient conductive material extending uninterruptedly outward from the primary-surface material to the tip of each fin. Therefore, slots orientated parallel with the longitudinal length of element 10,

must be regulated in length so as to provide sufficient segments **14** extending uniterruptedly outward from the primary-surface material to effectively and efficiently transfer heat from one medium in the enclosed channel to a medium external of the channel. Depending on the end use application of a heat exchanger employing the finned element of this invention, any artisan can easily determine the length of the slots necessary to provide sufficient conductive paths normal to the longitudinal axis of the heat exchange element. Thus the slots can be disposed anywhere between 0° and 180° with respect to the longitudinal axis of the heat exchange element provided it yields an angle of approach suitable for a particular end use heat exchange application.

FIGS. **2**, 2A and 2B show two isocompression heat exchange channels **21**, **22**, having front fins **23** and **24**, respectively, and rear fins **25** and **26**, respectively, juxtaposed in touching relationship with projected buttons **27** and **28** in contact. This two channel assembly is sufficient to show the dual set of passages that conventional heat exchangers have. Passages **29** in channel elements **21** and **22** define one set of confined passages independent of, and separate from, a second set of passages **30** formed between adjacent elements **21** and **22**. One fluid, shown as solid line arrows, can be fed through passages **29** while simultaneously, a second cooler fluid, shown as broken line arrows, can be fed through passages **30** to effectively cause a transfer of heat from the hotter fluid to the cooler fluid without having them intermixed. The fluid passing through the passages **30** will first pass through slatted apertures **31** and **32** in front fins **23** and **24**, respectively, thereby effecting a transfer of the heat which was conducted along the fins **23** and **24**, from the primary-surface area **33** and **34**, respectively. Likewise, when the fluid medium leaves passages **30** through apertures **35** and **36** in rear fins **25** and **26**, respectively, it again effects a transfer of heat which was conducted along the fins **25** and **26** from the primary-surface area **37** and **38**, respectively.

To illustrate an embodiment of the heat exchange elements of this invention, FIG. **3** shows a partial exploded isometric view of an automobile radiator **41** employing such heat exchange elements **42**. An array of heat exchange channel elements **42**, having front fins **43** and rear fins **44**, and aligned so that their isocompression projected buttons **45** are arranged such that they extend outward from each wall of a pair and cooperate with similar projected buttons on the wall of a second pair so as to space apart the walls of a pair to form passages between adjacent channel elements. For clarity of illustration, FIG. 3F shows the side view of two elements **42** with the fins **43** and **44** removed thereby displaying the touching relationship of isocompression projections **45**, and passages **46**. Note that the isocompression projections **45** are not aligned on opposite walls of the pair that comprises element **42**, but are offset, or non-symmetrically disposed, on such opposite walls of the pair. FIG. 3E shows an embodiment where the projections **45'** are symmetrically disposed on opposite walls of a pair of each element '. A side view of an end of element **42** and front fin **43** is shown in FIG. 3C while FIG. 3A shows the side view of the opposite end of element **42** and rear fin **44**. Fin **43** is shown with slats **47** at about a 45° angle (slot angle α) with respect to the longitudinal edge of element **42**. As shown in FIG. 3B, apertures **49** are disposed between slats **47** of front fin **43** and are aligned so that a medium flowing normal to the longitudinal length of the assembled elements **42** would flow through such apertures **49** with substantially mimimum resistance. Stated another way, the medium flow path across the slats **47** will be substantially minimized by the alignment of the slats in an orientation substantially parallel to the directional flow of such medium. Although not shown, rear fins **44** have the same type apertures between slats **48** as shown for fins **43**. Member **50** is placed on top of the array of elements **42** and then support bracket **51** is placed over the assembly along the finless channel edge segments **4** shown in FIG. **1**. This arrangement provides stability for the array of elements **42** in addition to securing proper alignment for elements **42**. Support bracket **51** must have an outer plate segment **52** adaptable for securing header **53** thereon, and in addition, it must be capable of providing a leak-tight seal to header **53** and to the channel elements **42** so that in the operational mode, a fluid fed through the elements **42** via header **53** will not leak into the space between adjacent elements **42**. A similar type header, not shown, is positioned at the opposite end of elements **42** to provide a compact efficient automobile radiator.

An alternate embodiment for the elements of this invention would be to have two fins, **54** and **55**, disposed from the same longitudinal edge **56** of element **42''** as shown in FIG. 3D. The fins could be an extension of the walls of the primary-surface material or they could be added separately as shown. It is to be understood that more than two fins from the same edge could be provided to yield a multiple fin constructed channel. A finned heat exchange element made by extending the primary-surface walls fore and aft from the enclosed channel segment, or by separately adding the fin between the mating wall edges of the enclosed channel segments, can have the mating wall edges bonded into a seam joint as for example by ultrasonic welding.

Although the fins of this invention are intended as an improvement on all channel type heat exchangers and primary-surface heat exchangers, the following example will be directed to the use of such fin in an automobile radiator which will illustrate the basic efficiency and other advantages derived from the use of such fins.

EXAMPLE

Three one-square-foot, heat exchange frontal area isostress radiators, two finless and one finned, constructed as shown in FIG. **3**, along with three conventional type copper radiators, typified by the radiator shown in FIG. **4** and modified to one-square-foot, heat exchange frontal area size, were tested in a wind tunnel shown schematically in FIG. **5**. The prime function of the test was to measure and compare the heat transfer capacity of the different type heat exchangers. This was accomplished by pumping heated coolant through the test radiators as a fixed rate as shown in FIG. **6** while varying the air flow rate as shown in FIG. **5**. As shown in FIG. **6**, steam was fed from a supply source (not shown) through a feedback control valve **80** through a 0-to-30 psi. gauge **81**. The steam was controlled by valve **82**, parallel coupled to a finer adjustment valve **83**, before being fed into a water heater tank **84**. The pressure in the tank **84** was measured by a 0–60 psi. gauge **85**. The water output from the tank **84** was fed

into a test radiator **86** and upon exiting from the radiator **86,** the water was pumped by motor **87** via parallel coupled control valve **88** and series control valve **89** back into the heater tank **84.** A water meter **90** was connected downstream of the pump **87** so as to measure the water flow through the radiator **86.** Steam condensate in the tank **84** was discharged into a recepticle, not shown. The feed rate of the heated coolant through the radiator was thus controlled at all times, and although not shown, temperature measuring device were used to record the temperature of the water into and out of the radiator.

The test schematic of FIG. **5** shows a wind tunnel used to regulate the rate and temperature of an air flow through the passages formed between adjacent heat exchange elements in the radiators being tested. Air entering the tunnel was first passed through a calibrated orifice **60,** which measured the air volume flow into a four cubic foot plenum chamber **61,** and then fed through flow straightening screens **62** into a tapered adapter section **63.** The adapter section **63** was provided to effect a smooth transition in the air flow between the plenum chamber **62** and a 1 square foot wind tunnel duct **64** coupled to a test radiator **65.** A blower (not shown) was positioned downstream of the radiator for controlling the air flow through the radiator. The heated air from radiator **65** was either exhausted via camper **66** or recirculated into the test room via damper **67** so as to provide a degree of temperature control within the room. To reduce the air flow rate through the radiator, control damper **68** was coupled downstream of the exiting air flow from radiator **65.** The tapered adapter section **63** and the flow straightening screens **62** were successful in keeping velocity variation through the test radiator **65** at a minimum. Velocity profiles and pitot tube readings made over the frontal area of different test radiator **65** indicated that air velocity variation for all the radiators tested was within ± 5 percent. One 0-to-2 inch manometer **69,** and two 0-to-4 inch manometers **70** were positioned as shown in the test circuit and used to measure the air pressure drop. Two grids, **72** and **73,** containing four thermocouples, each of which was placed in the center of one quarter of the flow passage area of the radiator **65,** measured the average inlet and exit air temperature through test radiator **65.** A Brown multipoint chart recorder (not shown) and a Rubicon potentiometer (not shown) were coupled to the grids and recorded the termocouple readings. Thus, an accurate test circuit was provided for measuring the heat transfer capacity of the test radiators.

Isocompression test radiator number **1,** similar to that shown in FIG. **2,** except it was finless, was fabricated by pressing the "right" and "left" channel halves of aluminum between male and female epoxy isocompression dies prepared as described in my above-identified patent application. The isocompression impressed channel halves were then trimmed and folded longitudinally on the edge, whereupon they were degreased, acid cleaned and Alodine treated, followed by a water rinse. After drying, an adhesive known as Resin Type EA-914, manufactured by Hysol Division of Dexter Corporation, Calif., was applied to the 13 inch long folded edge on each channel half, and mating channel halves were then joined to form a 1.625 wide channel. The channel was then placed in a fixture and rolled seam tight. After removing the excess adhesive from the channel, the adhesive in the seam was oven cured. Thereafter spacers were inserted into the tube and an adhesive was applied to each of the projected buttons and the tube walls. Inserts were inserted at the tube ends and an array of ten channels were clamped in touching relationship in a fixture and then the adhesive was oven cured. Thereafter the array of channels were again cleaned as specified above and then assembled into a header as shown in FIG. **3.** Once again adhesive was used to seal the channels to the header thus providing a test radiator.

Isocompression test radiator number **2** was prepared similar to the above except that the projected buttons were not adhesively secured together. The width of the channels measured 1.9 inches, and the length was 13 inches.

Isocompression test radiator number **3** was prepared similar to test radiator number **2** except that the seams were welded, and slotted fins were disposed on the fore and aft longitudinal edges of the channel, as shown in FIG. **3.** The overall width of the channel measured 1.7 inches and the unbent fins measured 0.2 inch and 0.5 inch at the fore and aft longitudinal edges. The length of the channel measured 13 inches. The slot angles α and α' measured 45°, the slat angle β and β' measured 40° and the bending angles γ and γ' varied for six fin configurations as follows:

TEST RADIATOR 3

Type 1	Short and long fin unbent, short fin forward
Type 2	Short fin bent 30°, long fin unbent, short fin forward
Type 3	Short fin bent 30°, long fin unbent, long fin forward
Type 4	Short fin bent 30°, long fin bent 15°, short fin forward
Type 5	Short fin bent 30°, long fin bent 30°, short fin forward
Type 6	Short fin bent 30°, long fin bent 30°, long fin forward

Three similar type conventional automobile radiators, test numbers **4, 5** and **6,** typical copper core consisting of brass tubes soldered to thin copper fins, except for varying fin and tankage configurations, were modified to one square foot heat exchange frontal area by cutting the core and reinstalling a shortened header and tank. Thus all test radiators were designed to yield one square foot heat exchange frontal area. The finned isostress test radiator cost substantially less to make than the conventional copperbrass test radiators because in the former, the cost of the base metal material was less per pound than the base metal material of the latter.

The nominal test conditions for the various radiators are shown in Table 1. Of the conventional radiators tested, test radiator **4** was found to be superior on a heat transfer per unit of air horsepower basis. Thus radiator **4** was compared to test radiators **1, 2** and **3.**

TABLE 1

TEST NO.	NUMBER OF RUNS	TEMPERATURE INLET AIR °F.	TEMPERATURE INLET WATER °F.	FLOW RATE AIR CFM	FLOW RATE WATER CFM
4	8	90	215	600–2700	15
	8	90	240	450–1650	17
	8	90	180	600–1700	17

TABLE 1-Continued

TEST NO.	NUMBER OF RUNS	TEMPERATURE INLET AIR °F.	TEMPERATURE INLET WATER °F.	FLOW RATE AIR CFM	FLOW RATE WATER CFM
	8	90	190	600–2000	17
5	8	90	200	600–1900	14
	8	90	210	600–2100	17
	8	75	185	600–2000	14
6	8	90	210	650–2500	17
1	16	90	210	400–1250	17
2	16	90	210	400–2000	17
3					
Type 1	8	75	185	600–1800	14
Type 2	8	75	185	600–1800	14
Type 3	8	75	185	600–2000	14
Type 4	8	75	185	600–2000	14
Type 5	8	75	185	600–2000	14
Type 6	8	80	185	600–2050	14

FIG. 7 shows a plot of capacity vs. air volume flow rate for the test radiators 1 through 4. The conditions for the plot was 100°F. average coolant fluid to inlet air temperature difference, using a 50–50 Prestone water coolant at a flow rate of between 14–17 gallons per minute. As illustrated in FIG. 7, radiator 3 Type 5 had a higher capacity for the same air volume of flow rate than the best conventional type radiator, radiator 4.

The result of this test proved that the finned aluminum radiator of this invention will perform comparably to the best conventional automobile radiator and cost substantially less to make because of optimum metal utilization. Thus this invention provides economical, compact, light-weight, finned heat exchange channel elements that can be assembled into an efficient heat exchanger for various applications.

Iterative calculations based on the experimental data of the test for radiator 3 in its various configurations revealed the following:

Approach Speed)	Heat Transfer Coefficient, BTU/Ft²°F Hr. Low Speed (600 FPM Approach Velocity) Primary Surface	Low Speed (600 FPM Approach Velocity) Secondary Surface	High Speed (1200 FPM Approach Velocity) Primary Surface	High Speed (1200 FPM Approach Velocity) Secondary Surface
Type 1	20.2	13.7	30.6	20.6
Type 2	20.3	23.0	30.5	39.3
Type 3	20.2	18.6	30.7	34.0
Type 4	20.2	27.0	30.5	47.0
Type 5	20.4	31.5	30.3	53.0
Type 6	20.4	30.5	30.6	52.0

Thus a radiator having a front fin with the same bent angle γ of about 30° as the rear fin, provided the best embodiment for an automobile radiator using the specific type heat exchange channel elements tested.

A radiator similar to that shown in FIG. 3 and constructed as specified for test Radiator 3 Type 5 was installed in an automobile and when subjected to local driving conditions, the radiator performed admirably.

As used herein, Mylar is a trademark of E. I. Du Pont Company, Alodine is a trademark of Amchem Products, Inc., and Prestone is a trademark of Union Carbide Corporation for ethylene glycol antifreeze.

I claim:

1. A heat exchanger comprising at least one flat primary-surface heat exchange channel element formed and bound by at least one thermally conductive walled material with a maximum dimension width line and a longitudinal length line, said element having an entrance opening and an exit opening, the improvement which comprises the addition of at least one secondary-surface heat exchange slotted fin of width greater than 0.1 inch and less than 0.6 inch disposed substantially along at least one longitudinal section of the channel element, said fin having a multitude of slatted apertures arranged in a louver configuration with adjacent slats separated by slot-shaped apertures, having the fin angle γ between about 0° and about 60° wherein γ is the angle formed between the plane of the fin and a plane containing the maximum dimension width line and longitudinal length line of said channel element, slat width of at least 0.02 inch and less than 0.10 inch, the slat angle β is between about 15° and about 90° wherein β is the angle formed between the plane of the fin and the plane of the slats and the slot angle α is between about 0° and about 180° wherein α is the angle formed between the longitudinal length line of the channel and the longitudinal length line of the slots.

2. The heat exchanger of claim **1** wherein at least two of such channel elements are aligned and spaced apart to provide a dual set of passages, the first set of passages defined by, and bound within, the conductive walls of each channel element, and a second set of passages defined by, and disposed between, adjacent channels, said fins being disposed at the fore and aft sections of the channel elements, and said section defined as the remote sections of the channel element as measured from a plane containing the center points of two adjacent channel elements and the longitudinal axis of the channel.

3. The heat exchanger in claim **2** wherein the slotted fin is fabricated with an angle of approach θ of between 0° and 70°, said angle of approach θ being defined as the angle between a first line which is parallel to a plane containing the maximum dimensional width line and the longitudinal dimensional length line of the channel element and perpendicular to the channel's longitudinal length line, and a second line formed by the intersection of a plane onto the surface of a slat on the slotted fin, said plane being normal to the surface of the slat and containing the first line as defined above.

4. The heat exchanger of claim **3** wherein said angle of approach is between 0° and 45°.

5. The heat exchanger of claim **2** wherein more than one slotted fin is disposed substantially along a longitudinal section of the channel element.

6. The heat exchanger of claim **4** for use in conjunction with internal combustion engines wherein said channel elements are flat aluminum isocompression channels, wherein the channel elements are aligned such that the planes of each element which contains the maximum dimensional width line and longitudinal length line of such elements are parallel; and wherein the angle of approach is between about 0° and about 45°.

7. The heat exchanger of claim **6** wherein the maximum dimensional width line of each primary-surface channel element is between about 0.75 inch and 2.0 inches.

8. The heat exchanger of claim **6** wherein angle β is between about 30° and about 60°; angle α is between about 45° and about 135°; and angle γ is between about 0° and about 60°.

9. The heat exchanger of claim **6** wherein the fins are an extension of the walls of the primary-surface channel elements.

10. The heat exchanger of claim **6** wherein the fins are separately added between the longitudinal edges of the primary-surface channel elements.

* * * * *

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,845,814 Dated November 5, 1974

Inventor(s) Leslie C. Kun

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 8, line 1, delete "continuing" and substitute --containing--.

Column 9, line 62, after "element", insert --42--.

Signed and sealed this 7th day of January 1975.

(SEAL)
Attest:

McCOY M. GIBSON JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents