An all purpose primary-surface heat exchanger, comprising an array of parallel channels formed and bounded by thin heat conductive walls, at least one wall of which has on at least a portion of its surface substantially uniformly disposed unidirectional truncated conical wall-supporting projections formed from the wall. The projections are arranged so as to mate with and to abut supportingly against corresponding wall-supporting projections of a similar adjacent wall. The walls, so arranged, are sealed at the wall edges in a manner to form and isolate alternate enclosed channels from intervening open channels so that the alternate channels may contain and conduct a first fluid, and the intervening channels may contain and conduct a second fluid at a different temperature, thereby effecting heat exchange between the fluids.

6 Claims, 12 Drawing Figures
3,757,855 PRIMARY SURFACE HEAT EXCHANGER

FIELD OF THE INVENTION

This invention relates to a thin metal or plastic plate heat exchange channel element having on a portion of its surface substantially uniformly disposed unidirectional wall-supporting truncated conical projections.

BACKGROUND OF THE INVENTION

The need for lightweight, inexpensive heat exchange elements for various heat transfer applications has been in demand by industry for a long time. The automobile industry has constantly been searching for a compact, light weight radiator for use in cooling the internal combustion engine. 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 passages with air flow therebetween. The pre-1942 automobile engines were designed to deliver between 50 and 125 horsepower and required radiators operable close to atmospheric pressure. A simple solder joint of finned-copper constructed radiator was therefore sufficient to cool the low horsepower engine of the automobile without much of a threat of over-heating. Various copper radiators having cup-like or frusto-conical 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 higher 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 reduce the 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.

An alternate solution to the conventional type finned-copper radiators for heat transfer applications in the automobile, is to replace the soft soldered copper fins with aluminum fins. Although aluminum is less expensive than copper, the fusion bonding of aluminum fins to the tubes of a conventional radiator by brazing techniques is expensive. In addition, if a corrosive flux is used, the deposits left by the salt bath of the brazing process must be meticulously removed. Alternate brazing techniques and methods, i.e., vacuum brazing, are still in the experimental stage and when perfected their high cost will probably overwhelm the savings otherwise gained in the use of aluminum rather than copper for producing automobile radiators. Other proposals have been advanced, such as the use of adhesive bonding between the fins and tubes of a radiator. However, the low thermal conductivity of present day adhesives renders this approach inefficient for radiator applications.

In heat exchange applications requiring pressure-bearing walls as the primary heat exchange surface, the present invention enables such walls to be fabricated from thinner thermally conductive material than is presently required of conventional type primary heat exchangers. In order to utilize relative thin sheet materials, the walls of conventional type primary heat exchangers have to be stayed by means of numerous support members so as to reduce stress in the walls. However, stayed walls are normally not practical because of the following reasons:

a. high stress concentrations are still produced in the wall at the point of attachment of the stays;

b. a substantial amount of material is required in the stays, and in heat exchangers such stays contribute only indirectly if at all to heat transfer; and

c. the numerous stays are tedious and expensive to install, particularly in heat exchangers where the spacing between walls is very small and often inaccessible.

The present invention overcomes the above drawbacks by providing an all-purpose, primary-surface heat exchange channel element having on at least a portion of its surface substantially uniform disposed unidirectional truncated conical wall-supporting projections. The heat exchange element is economical to fabricate and when employed in stacked units, they are admirably suited as a heat exchanger for use with internal combustion engines.

SUMMARY OF THE INVENTION

The primary-surface heat exchanger of this invention basically comprises at least one channel element formed and bound by at least one thin walled, thermally conductive metal or plastic material, such channel element having an entrance opening, an exit opening and substantially uniformly disposed on a portion of its wall surface unidirectional wall-supporting truncated conical projections formed from the wall in a dimensional relationship to be discussed hereinafter. The wall-supporting truncated conical projection have load bearing segments at their extremities which are shaped and arranged so as to mate with and abut against corresponding load bearing segments or wall-supporting truncated conical projections on a second surface. At least two such channels, when aligned in juxtaposed relationship, 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, 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 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 addition, heat exchange 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 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.

A truncated conical projection from a wall of a thin sheet material is a protrusion in the wall of the material having a cone angle $\theta$ and a radius of curvature $R$ as shown in FIG. 1A. Cone angle $\theta$ equals the acute interior angle measured between the horizontal undeformed surface of the wall adjacent the protrusion and the substantially straight segment along the sloped side of the protrusion, and the radius of curvature $R$ equals the radius of the surface segments on both sides of the bounding line of intersection formed by the protrusion and the undeformed surface of the wall adjacent the protrusion. The surface segment at the extremity of the truncated conical projection is a load bearing segment and is shaped so as to mate with, and abut against, similar type surface segments at the extremities of truncated conical projections on a second wall. For stacking or abutting two or more channel elements together, the wall-supporting unidirectional truncated conical projections are disposed in a pre-aligned space relationship on the surface of each element so that when the walls are juxtaposed, the load bearing segments at the outer extremities of the wall-supporting projections hereafter referred to as buttons, will be in touching relationship. With reference to any adjacent pair of pressure withholding walls, wherein the buttons of both walls project inward into the space between the walls, the forces due to the pressure either external or internal of the pair will be substantially balanced, i.e., the secured contact between the buttons will sustain by tension or compression the entire force due to the pressure and no other structural member will be needed to absorb the load. Thus the pressure force will be counter-balanced by a restraining force developed within the pair of walls without the necessity of any external structure.

With reference to any adjacent pair of pressure withholding walls, wherein the buttons of both walls project outward from the space between the wall, the pressure either external or internal of the pair will not be balanced and a member external of the pair will be needed on each exposed face of the pair to absorb the load by supportive contact with the buttons in either tension or compression. Thus a restraining force will not be developed within the pair of walls to counterbalance the pressure force. In a series, stack or array of walls, the member external of the pair may be yet another wall with buttons matching those of the juxtaposed surface of the pair.

With reference to any series, stack or array of truncated conical projected pressure withholding walls wherein the buttons of the two outermost walls of the stack project inward toward the stack, the forces due to pressure will be substantially balanced throughout the stack and no other structural member will be needed to absorb the pressure load and to restrain the walls from deflecting outward from the stack.

With reference to any series, stack or array of truncated conical projected pressure withholding walls wherein the buttons of the two outermost walls of the stack project outward from the stack, the forces due to pressure will not be balanced within the stack and a structural member will be needed juxtaposed in supportive contact with the buttons of each outermost wall to absorb the pressure load and restrain the stack.

Since the truncated conical projected wall of the channel element of this invention is designed as a primary-surface heat exchange channel element, its wall material need not be highly conductive and thus can be selected from at least one of the groups consisting of metals, metal alloys, metal clads, plastics (such as Mylar), plastic-coated metals and the like. The criteria of the material selected for the heat exchange channel element is that it be only 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. At least one material selected from the group consisting of copper, steel, brass, titanium and Mylar is suitable for this application.

"Substantially uniformly disposed wall-supporting projections" is intended to be broad enough to include a pattern of wall-supporting projections having a progressive variation in spacing along at least one axis of the heat exchange element. In addition, as hereinafter stated, additional wall-supporting projections can be provided along the curved portion of the channel which may have a spacing relationship different from that of wall-supporting projections occupying the central portion of the heat exchanger element.

The dimensions of, and the dimensional relationship between, the wall-supporting truncated conical projections on the wall surface of the channel are somewhat restrictive depending on the end use environment of the heat exchange channel. The pattern of wall-supporting truncated conical projections can be arranged in a square, diamond, triangle or any other design configuration depending somewhat on the actual shape of the channel and the intended differential pressure to which the wall of the channel will be subjected in its intended environment. To minimize the resistance to flow and maximize the heat transfer effectiveness of any defined flow area of a heat exchange channel of this invention, the wall-supporting truncated conical projections should be designed and arranged in only such size number and pattern as will provide the restraint necessary to withstand the maximum differential pressure for which the channel wall is designed in its intended environment. This will result in maximum primary-surface area being available for heat transfer.

Once the desired size and pattern of the wall-supporting truncated conical projections are determined for efficient heat transfer in an intended end use pressurized environment, the projections can be imparted to the surface of a thin-walled thermally con-
ductive sheet of material by any conventional technique such as pressing, stamping, rolling or the like.

A thermally conductive truncated conical projected sheet, so prepared, can be longitudinally folded upon itself with the projections facing either inwardly or outwardly, and the folded sheet segments spaced sufficiently apart so as to define a passage therebetween. When the buttons project inwardly of the passage, they should match and contact with buttons extending inwardly across the passage from the opposite wall. The width of the passage so formed is thereby defined by the projected heights of the wall-supporting buttons from the undeformed surface of the walls. Since stress concentration may occur at the bending area of the sheet in its intended operational environment, additional wall-supporting truncated conical projections may be disposed within the vicinity of such area so as to equalize the stresses throughout the channel structure. The longitudinally mating edges of the sheet can then be suitably sealed by conventional techniques, i.e., soldering, brazing, welding or with an adhesive filled lock-seam joint, to make it leak-tight. This unidirectional wall-supporting truncated conical projected channel is then ready for use as a heat exchange element. When a channel is formed with buttons projecting inwardly and when intended for internal pressurization, then the button contacting surfaces within the passages should be bonded together by conventional means as soldering, brazing or with an adhesive. An array of channels so formed with the wall-supporting projected buttons in touching relationship, can then be appropriately assembled to produce a compact, efficient, primary-surface heat exchanger. When the wallsupporting truncated conical projections are disposed outwardly, then the channels can be superimposed in button touching relationship wherein the heights of the projected buttons from the undeformed surface of the walls will define the size of the passage between adjacent channels. When the wall-supporting truncated conical projections are disposed inwardly, then the channels will have to be spaced apart by some external structure so as to define a passage between adjacent channels. A pressurized medium, such as hot water, could then be passed through the channels while a coolant medium, such as cool air, could be passed between, and contact the outer surface of, the channels thereby effecting a transfer of heat between the mediums. The wall-supporting truncated conical projected sheet could also be fabricated into a circular or spiral channel, or any multiple sided channel by appropriate bending and/or folding techniques. The heat exchange channelized elements so formed can also be shaped into any curvilinear configuration and then superimposed one on the other leaving defined passages therebetween to form a simple or complex geometry heat exchanger having multiple confined channelized passages and multiple separate passages defined by, and between, the outer surfaces of adjacent heat exchange channelized elements. By passing a medium through the channelized passages defined while directing a second coolant medium through the passages defined by, and between, the outer surfaces of adjacent elements, an effective, large, primary-surface heat exchanger is obtained. In a cross-flow heat exchange operational mode, the heat exchanger of this invention will provide a low frontal area and a low external fluid pressure drop. Frontal area is the area of the projection of the entire array of heat exchange channels onto a plane normal to the direction of fluid flow through the channelized passages. Low external fluid pressure drop is the static pressure drop across the length of the flow path of the external coolant medium.

The mediums can be fed through their respective passages in a mutually parallel relationship, a perpendicular relationship or at any angle relationship therebetween.

DESCRIPTION OF THE DRAWINGS

FIG. 1 — A graph of applied pressure vs. surface deflection for 30° and 45° truncated conical projections in an aluminum wall.

FIG. 1A — Truncated cone surface.

FIG. 2 — Isometric view of an automobile radiator employing the heat exchange elements of this invention.

FIG. 2A — View taken of the longitudinal edges of a heat exchange element of FIG. 2.

FIG. 2B — Side view of elements 1 of FIG. 2.

FIG. 2C — Alternate embodiment of elements 1 of FIG. 2.

FIG. 2D — Alternate embodiment of the longitudinal edges of elements 1 of FIG. 2.

FIG. 3 — Isometric view of an array of truncated conical projected channels with outwardly projected buttons.

FIG. 3A — Cross-sectional view of channels in FIG. 3 taken along line 3A—3A.

FIG. 3B — Sectional side view of channels in FIG. 3 taken along line 3B—3B.

FIG. 4 — Isometric view of an array of truncated conical projected channels with inwardly projected buttons.

FIG. 4A — Sectional side view of channels in FIG. 4 taken along line 4A—4A.

For general heat exchange application, a truncated conical projected surface, as illustrated in FIG. 1A, having a repeatable wall-supporting projection spacing D of between about 0.2 and about 2.5 inch; a D/d ratio between about 3 and about 10, a H/D ratio between about 0.05 and about 0.2, a cone angle θ less than about 35°, a R/D ratio of greater than about 0.075; and a sheet or wall thickness between about 0.003 and about 0.25 inch will be quite suitable. As used above and as shown in FIGS. 1A, H equals the maximum height measured perpendicularly from a surface which contains the extremities of the wall-supporting projections to the plane containing the undeformed surface of the wall adjacent the projection; D equals the spacing between the center of the closest adjacent wall-supporting projections on the surface of the wall, d is equal to the equivalent diameter of the projection defined by the ratio 4a/p whereby a equals the area of the load bearing segment of the wall-supporting projection and p equals the perimeter of said load bearing segment, i.e., for a circular load bearing segment d is equal to the diameter of the circle as shown in FIG. 1A; θ is the acute interior angle measured between the horizontal undeformed surface of the wall adjacent the projected protrusion and the substantially straight segment along the sloped side of the protrusion; and R equals the radius of curvature of the surface segments on both sides of the bounding line of intersection formed by the protrusion and the undeformed surface of the wall adjacent the protrusion. The limitation on the D spacing
is imposed because spacing less than 0.2 inch results in very small clearance passages on the projected side of the wall thereby being very susceptible to fouling, i.e., trapping of foreign matter between adjacent wall, which if excessive, would clog the passages for one of the fluid mediums. A high external fluid pressure drop per unit length of fluid flow path would also result. Spacing D above 2.5 inches would result in a small heat exchange area per cubic foot of heat exchange volume thus resulting in excessive manufacturing cost and decreased efficiency. Also the ability for the material to withstand a differential pressure across its wall thickness would be decreased for a constant wall thickness because of the large spacing between wall supports.

For \( D/d \) ratio of less than 3, the allowable differential pressure across the wall of a channelized heat exchange element would go up, but a very large percentage of the surface area would be lost for heat exchange purposes. On the other hand, a \( D/d \) ratio greater than 10 would require tight manufacturing tolerance to insure the mating of bearing segments (buttons) on abutting walls and would also localize and concentrate the load at the contact point of the bearing segments and produce stresses sufficient to cause rupture or excessive deformation of the walls.

A heat exchanger composed of truncated conical projected channels with a \( H/D \) ratio less than 0.05 would be susceptible to fouling and have a high external fluid pressure drop per unit length of fluid flow path. For a \( H/D \) ratio of greater than 0.2, a small heat exchange area per cubic foot of heat exchange volume would result thereby resulting in excessive manufacturing cost and decreased efficiency. A \( R/D \) ratio lower than 0.075 would result in excessive stresses along the bounding line of intersection formed by the conical projection and the undeformed surface of the wall adjacent the projection. In effect, the bounding line of intersection becomes a line of high stress concentration.

A heat exchange wall having truncated conical projections with a cone angle of greater than 35° would result in excessive deflections of the unsupported areas disposed between the projections when pressure is applied to the wall. If such deflections are imposed repeatedly in service, then the material may be fatigued and crack after a relatively short service life. Additionally, deflections reduce the available space between heat exchange walls in the lower pressure passages, and result either in higher fluid pressure drop or in reduced rate of fluid flow.

A material thickness \( t \) of less than about 0.003 inch would be unsuitable due to local imperfections in the metal, produced during rolling or as a result of pitting (corrosion) or erosion. A material thickness to about 0.25 inch is not suited to this invention when employed within the imposed limits of \( D, H \), and \( d \), because full or near-full utilization of the material strength implies extremely high pressure differentials. Embodiments wherein pressure forces are not balanced within the channels require massive external structures to absorb the loads, while force-balanced embodiments wherein wall-supporting projections are bonded together and loaded in tension would be characterized by severe stress concentration in such bonded areas.

To meet the specific heat exchange requirements for radiators of internal combustion engines, the allowable ranges expressed above have to be narrowed to the following: a repeatable distance \( D \) between about 0.2 and about 0.6 inch, a \( D/d \) ratio of between about 3 and about 7; a \( H/D \) ratio of between about 0.05 and about 0.12; an angle \( \theta \) of less than about 35°, preferably less than about 30°, \( R/D \) ratio of greater than about 0.075; and a sheet or wall thickness between about 0.003 and about 0.02 inch. The preferred dimensions of a truncated conical projected surface for automobile radiator applications are a repeatable \( D \) of about 0.4 inch, a height \( H \) of about 0.035 inch; a button dimension width \( d \) of about 0.09; a \( D/d \) ratio of about 4.8; a \( H/D \) ratio of about 0.08; an angle \( \theta \) less than about 30°; a \( R/D \) ratio of greater than about 0.075; and a sheet or wall thickness of about 0.008 inch.

As an illustration of this invention, the following example will be directed to an automobile radiator employing the primary-surface heat exchange elements described above.

The deflection of truncated conical projected surfaces suitable for automobile radiator applications was investigated by plotting curves of applied pressure (lb/sq. inch) versus surface deflection (inches) for 0.008 inch thick aluminum sheets having truncated conical projections with cone angles of 30° and 45°.

The sheets were stamped with wall-supporting truncated conical projections arranged in a square pattern. The \( D \) spacing between the projected supports was 0.4 inch and the height \( H \) was 0.035 inch as shown in FIG. 1A. Pressure was applied to the surface of the aluminum sheet of the indented side of the sheet such as to place the cone material under compression, and the deflection at the center of the diagonals of the square pattern was measured. This data for cone angle surfaces of 30° and 45° is shown plotted as curves on the graph of FIG. 1.

Deflections of the material tending to distort the wall are objectionable and should be minimized even though such deflections may be safely below the buckling point of the material. Moreover, the material is usually stressed in bending and shear as it deflects, and when deflections are excessive the material may experience stresses approaching the yield point in localized areas. If such deflections are imposed repeatedly in service, the material may be fatigued and crack after a relatively short service life. Additionally, deflections reduce the available space between the heat exchange walls in the lower pressure passages, and result either in higher fluid pressure drop or in reduced rate of fluid flow. With reference to FIG. 1, it is seen that the 30° cone surface used in the tests exhibited a relatively small deflection at the center of the diagonals of the square pattern of the truncated conical projections for pressure differentials as high as 35 psi. In contrast, the deflection of the 45° cone surface was approximately double the deflection of the 30° cone surface pressure differentials up to about this 35 psi level. For environments wherein pressure differentials across a heat exchange surface will be as high as 35 psi, a surface with 35° cone angle truncated conical projections, sized and spaced within the ranges specified above, will be suitable.

In the foregoing tests of the 30° and 45° cone angle surfaces, the stress in the material was also measured directly by means of strain gauges at 30 psi differential pressure. The stress was measured on the diagonal at the point where the inclined surface of the conical indentations met the flat undeformed segment of the ma-
The data shows the increase in stress resulting from use of the 45° cone surface over the 30° cone surface thus demonstrating the high stress in the radius R arc of the 45° cone surface.

It is essential that all the surface area exclusive of the wall-bearing supports be unrestricted so as to be free to deflect and therefore be devoid of local mechanical loading. As can be seen from FIGS. 3A and 4A, if the undeformed areas were bonded together than the passages between such bonded surfaces would disappear, and if stays were employed to provide suitable spacing therebetween, a primary advantage of this type construction would vanish.

Once the dimensions of, and the dimensional relationship between, the desired wall-supported truncated conical projections of a heat exchange element are determined, a die or the like can be prepared by conventional techniques. The die can be used in conventional type apparatus to impart by pressing, stamping or rolling the desired truncated conical projections onto a thin-walled thermally conductive sheet, such as aluminum. For radiator applications, a rectangular aluminum sheet can be stamped or the like with the truncated conical projections. If the sheet is to be folded, then the central folding area shall be left free of the projections. The sheet, which may have any desired thickness, as specified above, although about a 0.008 inch thick sheet is preferable, can then be longitudinally folded at the center forming a flattened tube-like configuration with the wall-supporting truncated conical projections facing inward or outward. Instead of preparing one large sheet and folding it, two sheets may be prepared and formed appropriately at the longitudinal edges for bonding and then spaced apart by suitable means to form a flattened tube-like configuration. If desires, the longitudinal edges of the sheets could be flared a specific amount so that when said longitudinal edges of two sheets are juxtaposed in touching relationship, they will provide the desired spacing within the channel. The edges of the sheets can be "potted" as with epoxy resin to seal the sheets leak-tightly together to form tube-like configurations, an array of which can also be sealed leak-tightly into a header to form a radiator assembly.

As shown in FIGS. 2, 2A and 2B, flattened tube-like heat exchange elements 1 can be air-tightly sealed along their edges 2-3 using a lock-seam joint filled with an adhesive 14, such as a suitable epoxy type adhesive. The heat exchange elements 1 having an horizontal surface 4 with spaced apart wall-supporting projections 5, can be superimposed with the surface extremities 17 (buttons) in touching relationship to form a multiple layer heat exchanger. As shown in FIG. 2B, the projecting buttoned 17 provide passages 15 between adjacent heat exchange elements 1 defined by the horizontal surfaces 4 of the adjacent elements 1, and in addition, the contacting buttons 17 act as a restraint against internal pressure in the heat exchange elements 1. The projected button 5' could be offset or non-symmetrically disposed on opposite sides of each element 1', as shown in FIG. 2C, thereby altering the passage area of element 1'. The ends 6 of elements 1 are slightly depressed, if necessary, to provide a clearance for the teeth 7 of comb-shaped members 8. Members 8 retain elements 1 in proper relationship and provide an outer plate segment 9 adaptable for securing header 10 thereto. In addition, members 8 must also produce a leak-tight seal to header 10 and to the channel elements 1 so that in the operational mode a fluid fed through the elements 1 via the header 10 will not leak into the space between adjacent elements 1. As shown, header 10 can be secured to members 8 by using an adhesive type joint arrangement 11. A suitable resin for use in adhesive type joints for aluminum is Resin Type EA-914, manufactured by Hysol Division of Dexter Corporation, California. However, this resin must be used in conjunction with an Alodine process for pre-treating the surfaces to be bonded. An Alodine pre-treatment process would basically consist of the following steps:

a. soaking and rubbing the surfaces to be bonded in acetone to degrease;

b. immersing the surfaces in weak H₃PO₄ acid for 10-15 seconds at room temperature;

c. washing the surfaces in water;

d. immersing the surfaces in Alodine 1200 at room temperature for 5 to 20 minutes (Alodine 1200 is manufactured by Amchem Products, Inc., Fremont, California, and contains acidic chromates and fluorides);

e. washing the surfaces with water; and

f. drying the surfaces.

The dried surfaces can thereafter be bonded with Hysol preferably within a 4 hour period. Elements 1 can then be retained together by employing a tension type channel 12 which can be secured to either members 8 and/or to a separate structure member 13. Channel 12 must also be designed rigidly with sufficient cross-sectional moment of inertial to absorb a bending load and to permit a small expansion of elements 1. Members 8 and/or 13 can further be secured to a frame of the automobile for better support. To better illustrate the dual set of passages of an array of elements of this invention, FIGS. 3, 3A and 3B show an array of elements 21 with outwardly protruding wall supports 22. Passages 23 in elements 21 define a set of confined passages independent of and separate from a second set of passages 24 formed between adjacent elements 21. One fluid, shown as solid line arrows, can be fed through passages 23 in elements 21 while simultaneously a second cooler fluid, shown as broken line arrows, can be fed through passages 24 to effectively cause a transfer of heat from the hotter fluid to the cooler fluid without having them intermixed. For this type embodiment, a rigid frame or support similar to support 12 of FIG. 2 is required so as to constrain the stack of elements 21 along the sides. FIGS. 4 and 4A illustrate a similar array of elements 30 except that the wall-supporting projections 31 are inwardly projected. Passages 32 within elements 31 are independent of and separate from passage 33 formed between adjacent elements 30. One fluid, shown as solid line arrows, can be fed through passages 32 while simultaneously a second cooler fluid, shown as broken line arrows, can be fed through passages 33 to effectively cause a transfer of heat from the hotter fluid to the cooler fluid without having them intermixed. For this type of element arrangement, spacers 34 are required to space the ele-
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ments 30 sufficiently apart so as to define passages 33. It is to be understood that the spacer 34 could be similar to the comb-like structure 8 as shown in FIG. 2, which in turn could be coupled directly to a header similar to header 10 illustrated also in FIG. 2.

In operational mode of an automobile radiator, as shown in FIGS. 2, 2A and 2B, hot water from an internal combustion engine is fed through elements 1 while cool air is circulated through the passages 15 formed between adjacent elements 1. To increase the efficiency of the heat exchange elements 1, one or both of the edges 2 and 3 may be extended to provide a secondary surface heat dissipating fin 16 as shown in FIG. 2D. The fin, which could also be added to the elements by conventional securing means, can be provided with dimples to promote turbulence, or provided with slots or any other desirable geometric configuration which would enhance the performance of the heat exchange elements. Also side bars could be used to separate the elements as shown in U. S. Pat. No. 3,291,206 or edge ribs, as shown in U. S. Pat. No. 3,106,242.

Although the above illustrated was directed to automobile radiators, the primary-surface heat exchange element of this invention can be employed in any type heat exchanger wherein a heat transfer between a heated medium and a coolant medium is to be accomplished without an intermixing of the media occurring. The design flexibility of the primary-surface heat exchange elements of this invention makes them admirably suited for complex type heat exchanger applications including preheaters for gas turbines and low grade heat rejecters for atomic power plants.

As used herein, Mylar is a tradename of E. I. DuPont Company, and Alodine is a tradename of Amchem Products, Inc.

What is claimed is:

1. A primary surface heat exchanger comprising at least one channel element bound by two thermally conductive walls being spaced by edge portions, said channel element having an entrance opening at one end, an exit opening at the opposite end and on at least a portion of each wall surface substantially uniformly disposed only outwardly extending truncated conical wall-supporting projections formed from the wall, said projections having load bearing segments at their extremities which are shaped for mating with and abutting against similar type load bearing segments on wall-supporting only outwardly extending projections of a second channel element; said wall-supporting outwardly extending truncated conical projections having a dimensional size and a dimensional relationship therebetween defined by a \( H/D \) ratio of between about 0.05 and about 0.2, a \( D/d \) ratio of between about 3 and about 10, a \( D \) dimension of between about 0.2 and 2.5 inches and a wall thickness between about 0.003 and about 0.25 inch; a \( R/D \) ratio greater than about 0.075; and a cone angle \( \theta \) less than about 35°; wherein \( H \) equals the maximum height measured perpendicularly from a surface which contains the extremities of the wall-supporting outwardly extending projections to the plane containing the undeformed surface of the wall adjacent the projection; \( D \) equals the spacing between the centers of the closest adjacent wall-supporting outwardly extending projections on the surface of the wall; \( d \) equals the equivalent diameter defined by the ratio \( 4a/p \) whereby \( a \) equals the area of the load bearing segment of the wall-supporting outwardly extending projection and \( p \) equals the perimeter of said load bearing segment; \( \theta \) equals the acute angle measured between the horizontal undeformed surface of the wall adjacent the outwardly extending projection and the substantially straight segment along the sloped side of the projection; and \( R \) equals the radius of curvature of the surface segments on both sides of the bounding line of intersection formed by the projection and the undeformed surface of the wall adjacent the projection.

2. The heat exchanger of claim 1 wherein said thermally conductive heat exchange element is made from at least one material selected from the group consisting of metals, metal alloys, metal clads, plastics and plastic coated metals.

3. The heat exchanger of claim 2 for use in conjunction with internal combustion engines wherein said \( H/D \) ratio is between about 0.05 and about 0.12; wherein said \( D/d \) ratio is between 3 and about 7; wherein said \( D \) dimension is between about 0.2 and about 0.6 inch; wherein said angle \( \theta \) is less than about 35°; wherein said \( R/D \) ratio is greater than about 0.075; wherein said wall thickness is between about 0.003 and about 0.02 inch; wherein at least two of said elements are juxtaposed with the mating load bearing segments on the wall-supporting projections of said elements in touching relationship to form a first set of passages defined by said channel elements and a second set of passages defined by, and between, adjacent channel elements.

4. The heat exchanger of claim 3 wherein the heat exchange element is made of aluminum having a wall thickness of about 0.008 inch; said \( H/D \) ratio is about 0.08; said \( D/d \) ratio is about 4.8 and said \( D \) dimension is about 0.4 inch.

5. The heat exchanger of claim 2 wherein secondary surface heat dissipating fins are added to said channel element.

6. The heat exchanger of claim 3 wherein secondary surface heat dissipating fins are added to said channel elements.

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