

Feb. 15, 1966

C. BEURTHERET

3,235,004

HEAT DISSIPATING STRUCTURE

Filed Feb. 21, 1963

4 Sheets-Sheet 1

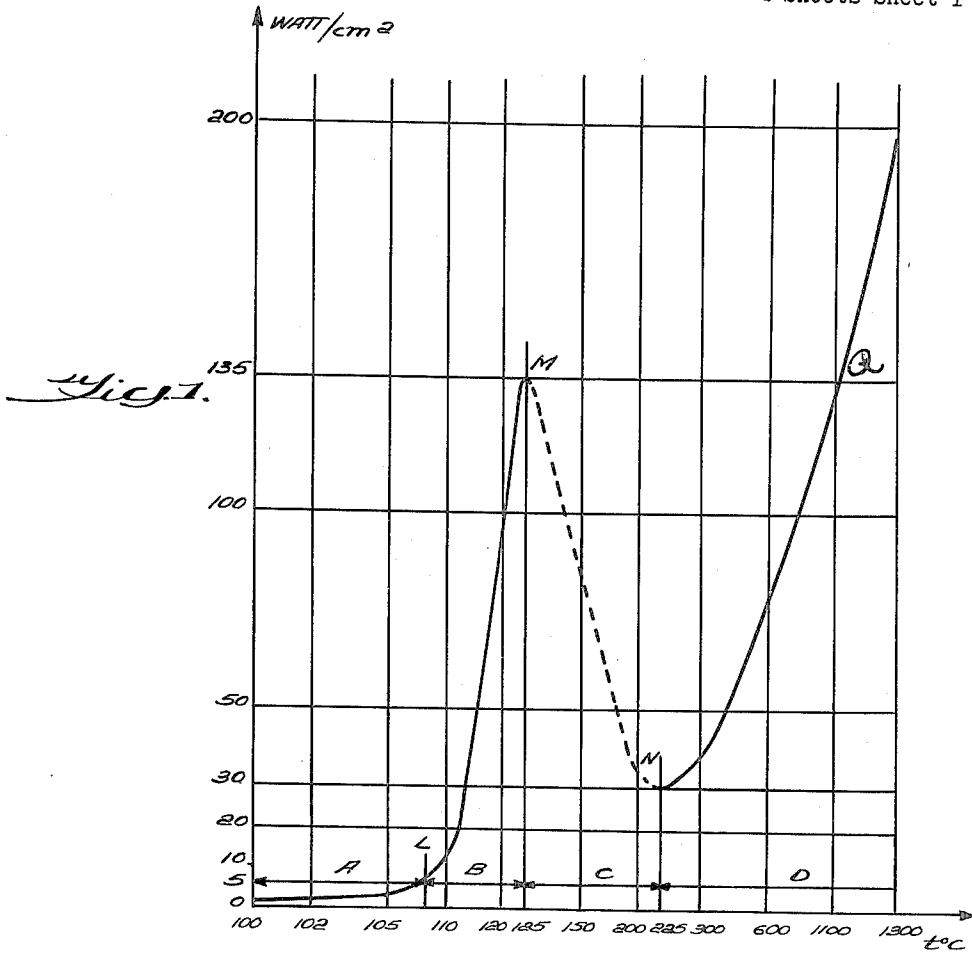
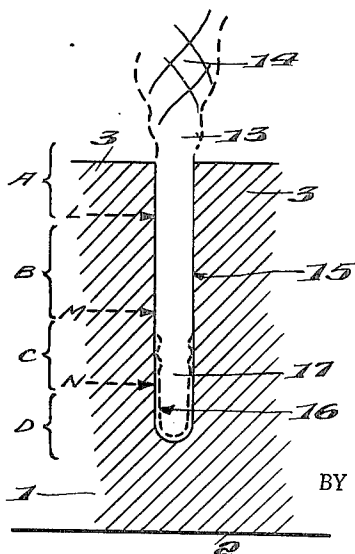


Fig. 4.

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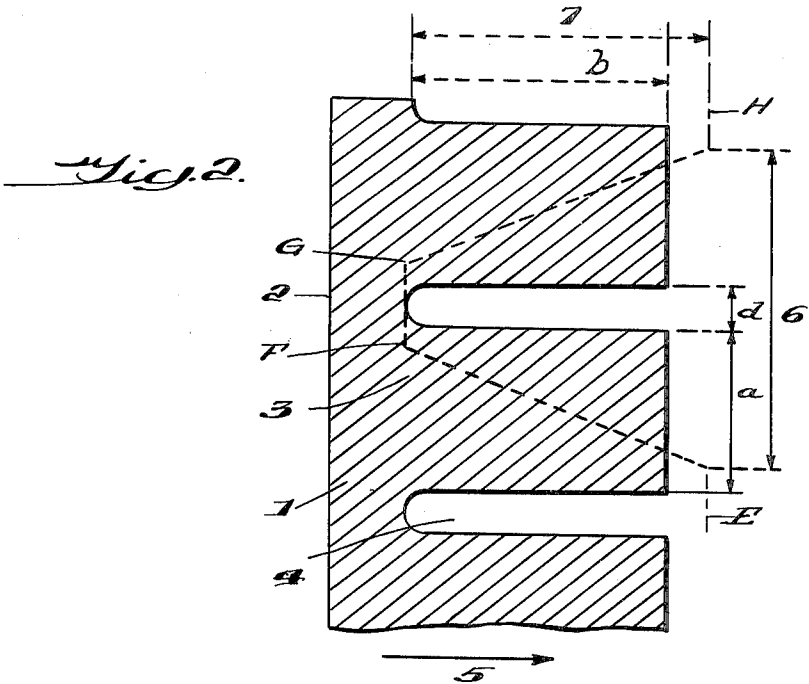
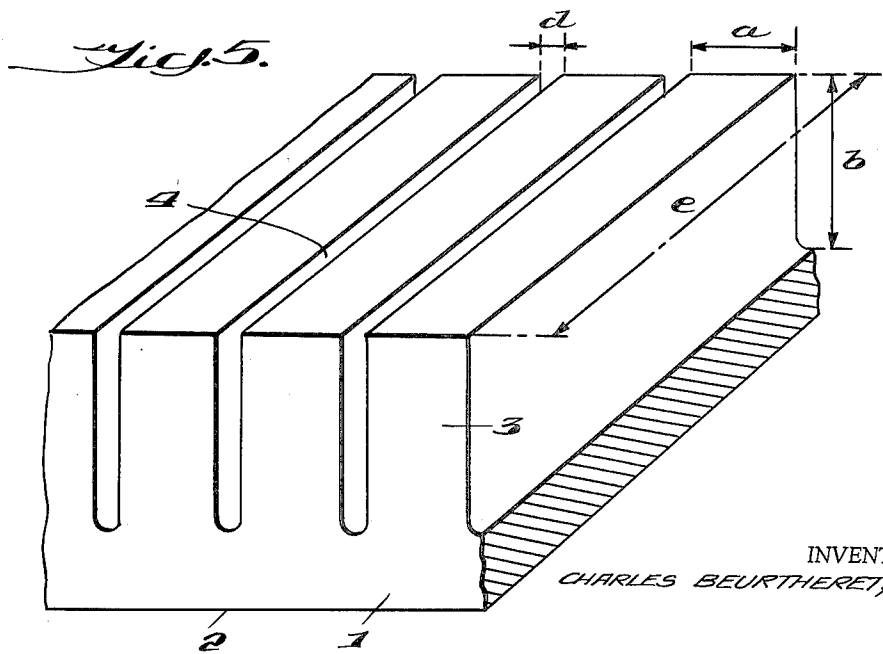
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Fig. 1.

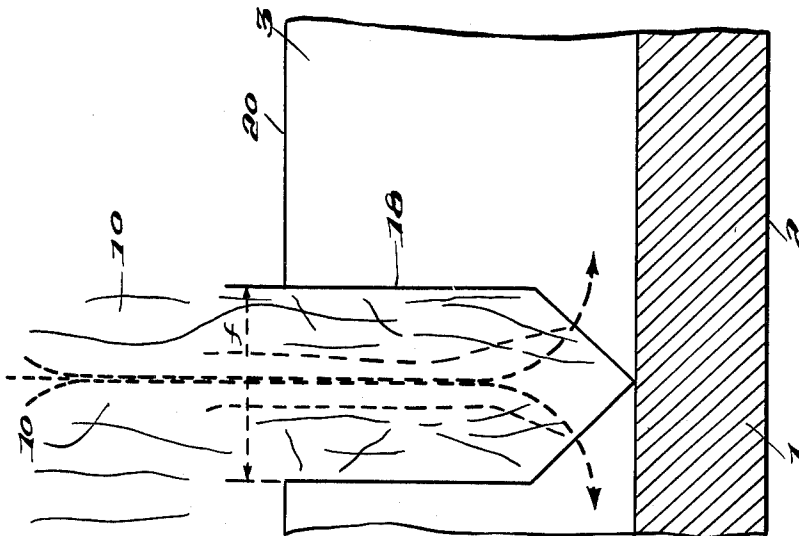
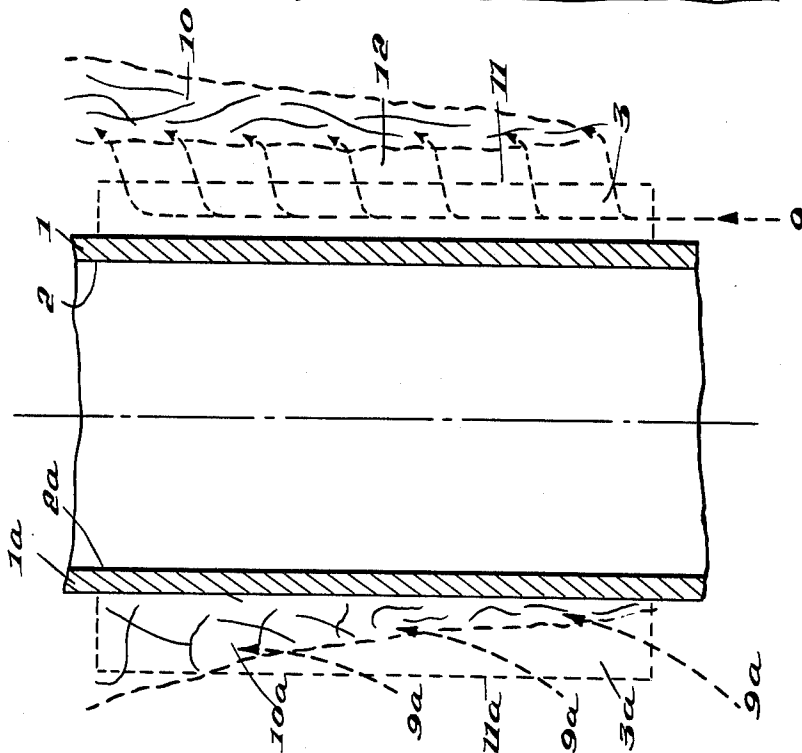


Fig. 3.



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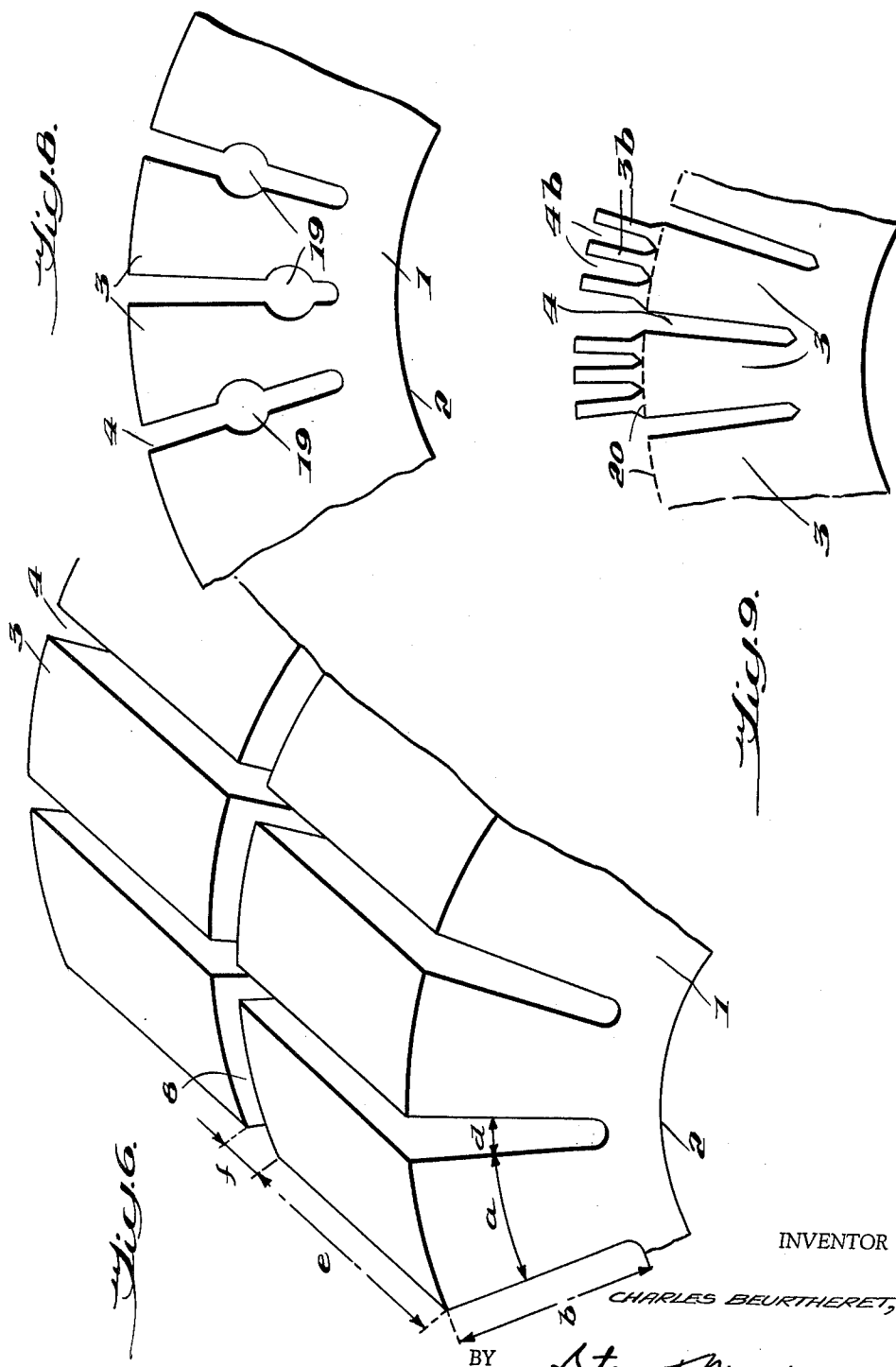
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HEAT DISSIPATING STRUCTURE

Filed Feb. 21, 1963

4 Sheets-Sheet 4



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3,235,004

HEAT DISSIPATING STRUCTURE

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Filed Feb. 21, 1963, Ser. No. 260,245

Claims priority, application France, Feb. 23, 1962,

889,025

13 Claims. (Cl. 165—185)

This invention relates to heat exchange structures, especially those used to dissipate heat from one side of a wall the other side of which is exposed to intense heat, by means of a vaporizable liquid supplied to said one side of the wall. Evaporation-cooling heat dissipating structures of this sort are used in many kinds of apparatus in various fields of engineering, including inter alia high-power electronic tubes, nuclear fuel elements, etc.

The dissipation of heat from a wall surface through partial evaporation of a liquid, such as water, supplied thereto, involves a variety of quite complex fluid-dynamics phenomena which have only relatively recently formed the subject of systematic investigation. Considerable difficulties were experienced in the past due to the formation of hot points on the surface, by an effect known as spheroidal state, which led to destructive burn-out, thereby severely limiting the permissible rate of heat dissipation and hence the power rating of the apparatus units that were to be cooled.

Early attempts at avoiding the difficulties due to hot points and burn-out in evaporation-cooling systems, generally relied on maintaining a large supply of non vaporized water at the surface thereby continuously removing the layers of steam as they were formed. A high rate of water circulation under substantial overpressure was provided so as to eliminate and/or remove all steam bubbles. With this technique the heat dissipating surface had to be consistently held under substantially isothermal conditions, since any point of it if allowed to become even slightly hotter than the average surface temperature would immediately initiate an enormous rise in local temperature leading to destructive burn-out, as will be explained in greater theoretical detail hereinafter.

The applicant has for a number of years been conducting extensive research in the field of evaporation-cooling, and has developed a type of heat-exchange surface in which the formation of hot points and destructive burn-out has been greatly reduced or delayed, thereby improving the performance of the cooling system. As disclosed inter alia in United States Patent 2,935,305 and United States patent application, Serial Number 25,799, filed April 29, 1960, this known type of heat dissipating surface as earlier developed by the applicant for evaporation-cooling systems, was essentially characterised in that the surface was formed with certain structural contours which resulted in establishing and maintaining, over predetermined areas of the surface, a stable progressive, temperature gradient (rather than maintaining the surface isotherm throughout as was the usually accepted previous practice), which steady gradient was in turn found to stabilize the temperature at each point of the surface in such a manner that the initiation of sharp local temperature rises, at any point of the surface was positively prevented. The theory on which this stabilizing effect is based will be outlined later herein.

The present invention represents a further advance along the same line of development. Where the applicant's earlier patents cited above taught the provision of certain structural patterns, in the general form of ribs, protuberances and/or channels, in a heat dissipating surface ex-

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posed to a vaporizable liquid, whereby stable, steady, temperature gradients were created effective to reduce hot point formation, it is the object of the present invention to disclose specific dimensional relationship whereby the operation of said structural patterns will be modified in a manner to maximize their effectiveness.

Broad objects of this invention are to provide an improved heat exchange structure, especially for evaporation-cooling systems; to provide such structure in which the rate of heat exchange can be considerably increased over what was possible heretofore without encountering dangerous hot-point and burn-out conditions; to provide more effective evaporation-cooling systems applicable to a variety of heat-generating equipment, including especially high-power electronic tubes and other electrical apparatus, nuclear fuel elements and related nuclear equipment, etc., whereby the power rating of such apparatus units can be safely increased over what is considered feasible at the present time. Other objects will appear as the disclosure proceeds.

The invention will now be further described, by way of example, with reference to the accompanying drawings in which:

FIG. 1 is a graph showing the variations of heat dissipation per unit surface area as a function of surface temperature, where the surface is swept by boiling water at atmospheric pressure;

FIG. 2 is a schematic sectional view of a heat dissipating structure according to a basic form of the present invention, illustrating the dimensional parameters involved in the relations taught hereby;

FIG. 3 is a schematic view in longitudinal cross-section of a heat dissipating member, of e.g. tubular form, constructed on the left-hand side with a cooling structure in accordance with the applicant's prior patents and on the right-hand side with a structure in accordance with the present invention, in order schematically to illustrate the comparative behaviour of the cooling fluid in the respective cases;

FIG. 4 is a transverse section through a groove in a cooling structure according to the invention, and serves to explain a further feature of operation;

FIGS. 5 and 6 are perspective views of two modified forms of cooling structure according to the invention;

FIG. 7 is a view of a further modification, showing a longitudinal section through a groove;

FIGS. 8 and 9 illustrate two further modifications showing a section transverse to the grooves of the structures.

Before the specific teachings of this invention can be intelligibly disclosed, it is necessary to consider the theoretical explanations outlined above somewhat more closely with reference to the drawings.

When a wall of heat-conductive material is exposed on one side to a heat source and on the other to a liquid at vapour-saturation or boiling point, e.g. water at 100° C. at ordinary pressure, the law of heat transfer from each element of the surface (assumed to be under isothermal conditions) exposed to the fluid is expressed by a curve of the type shown in FIG. 1, and known as Nukiyama's curve. In FIG. 1 the ordinates represent dissipated heat per unit area, expressed in watts, per sq. cm., and the abscissae represent temperature of the surface element in ° C. The curve is seen to comprise four main sections: a first low-slope initial section "A" from the origin to a knee point L, over which the heat exchange between the surface and the liquid occurs through normal convection, without the liquid boiling; a second steep-rising section "B," from knee L to peak M, in which the water undergoes normal boiling, and the rate of heat transfer through convection is greatly increased; a third drooping section "C" from peak M to valley N, which corresponds to a

region of transition; and a fourth rising final section "D" from valley N onwards, in which the vaporization of the water proceeds by so-called film-boiling (or spheroidal state) rather than the ordinary ebullition as at lower surface temperatures.

The meaning of the graph is that, when any surface element is held at any strictly constant temperature, selected on the abscissae axis, as by appropriately regulating the heat applied from the source to the opposite side of the wall, then the rate of heat dissipation remains generally constant at the corresponding ordinate value of the curve. In practice however, the heat source usually imposes an increasingly high temperature on the wall and under these conditions it is found that the operating point first follows the first two rising sections A and B of the curve, as far as peak point M, and then, assuming the surface temperature continues to increase, as is usually the case, the operating point jumps from point M to a point Q positioned at a corresponding ordinate on the fourth section D, resulting in a quasi-instantaneous increase in the surface temperature from about 125° C. to more than 1000° C. usually causing irretrievable damage to the surface material (burn-out). This enormous and sudden increase in the temperature of an element of the cooling surface, occurring wherever the temperature of the cooling surface is allowed to exceed at some point the comparatively moderate temperature of the peak point M (125° C. at atmospheric pressure), constitutes the so-called irreversible overheating effect or "calefaction" due to the spheroidal state of the liquid, leading to the aforementioned hot-spot formation. The critical point M on Nukiyama's curve has accordingly been known as the burn-out point. Wherever the heating rate from the source is so high as to cause the burn-out point to be locally exceeded in any portion of the cooling surface by even an exceedingly small amount, destructive burn-out of the surface occurs.

For many years the only known technique for avoiding such burn-out was to increase both the rate of circulation, and the pressure, of the cooling liquid, thereby rapidly eliminating the layer of steam that tended to form along the surface and removing steam bubbles. This is equivalent to increasing the ordinate value of the burn-out point M, so that the operating point of the heat exchange process would remain on the rising portion B of Nukiyama's curve for greater rates of heat supply. In yet other words the procedure tended to prolong the state of normal ebullition and retard the onset of the transitional conditions. The flow rate and pressure of the cooling liquid were adjusted to values great enough to ensure that the operating point would remain sufficiently below the burn-out point M to provide an adequate safety margin against local hot-spot formation and burn-out. It is to be noted that as earlier mentioned herein, in the procedure just described, the heat-dissipating surface can be regarded as being held substantially isotherm, since measures are taken at no point thereof to exceed the critical burn-out temperature which in turn is only very slightly higher than the boiling point of the liquid at the pressure under consideration, as shown by the steep slope of the curve from L to M. A moderate local increase in temperature beyond the over-all average isothermal temperature would result in pushing the operating point past the burn-out point M with the disastrous consequences stated above.

In contrast to this well-known technique, the applicant has discovered, and disclosed in the aforementioned United States Patent 2,935,305, that it is possible to construct the heat dissipating surface in an evaporation-cooling system in such a manner that the transitional condition of the cooling fluid between its normal boiling state and skin vaporization or spheroidal state can be stabilized adjacent the cooling surface. This means that should the surface temperature be allowed to increase somewhat beyond the temperature corresponding to peak point M (about 125° C. at ordinary pressure) due to an

increase in the heat supply rate, the operating point, instead of jumping at once the point Q with a corresponding over-heating of the surface to more than 1000° C., as explained above, will simply follow the drooping portion "C" or M-N of Nukiyama's curve, bringing about a moderate and entirely permissible rise in temperature accompanied by a drop in heat transfer rate, without damaging burn-out. The cooling surface construction required to bring about these remarkable results, as disclosed in the aforementioned patent, essentially comprised providing a pattern of massive ribs or protuberances of short, thick configuration, separated by complementary grooves or channels of relatively shallow depth. Preferably these channels are vertical. The outer ends of the ribs or projections extend to a region of the body of the cooling liquid beyond the area filled with the vapour forming on contact of the liquid with the hot surface at the root of the projections. Thus the steam gathers in the vertical channels between the ridges and rises rapidly through them by thermosiphon effect. The channels may be open, or they may be enclosed laterally by longitudinal partitions to channelize the liquid entrained with the steam.

A heat-dissipating surface thus constructed in accordance with the teachings of the applicant's prior patents is essentially non-isotherm. Experiments have shown that continuous, relatively stable, temperature gradients become established along the sides of the massive ridges or projections. The temperature ranges in these gradients extend continuously both below and above the temperature corresponding to the peak point M of Nukiyama's curve (125° C. at ordinary pressure), from cooler areas corresponding to the arc "B" of the curve, in which ordinary ebullition prevails to hotter regions within the transition arc "C." In this manner the transition phase becomes stabilized in temperature and the operating point can be safely positioned within the region M-N without danger of burn-out. In other words, the peak point M no longer behaves as a critical point that cannot be exceeded along the curve without the surface temperature immediately jumping to a burn-out value of over 1000° C. as in more conventional evaporation heat-exchange surfaces, and the peak point M no longer deserves the name of "burn-out point."

According to an improvement disclosed in applicant's copending U.S. patent application, Serial Number 25,799, filed April 29, 1960, the ridges or projections are provided with extensions serving to improve the thermal contact and heat exchange between the liquid and the tips of the ridges.

Cooling structures constructed in accordance with the applicant's earlier patents are now widely used for the cooling of the anodes in high-power electronic transmitter tubes. Their cooling efficiency is surprisingly high. Using them, it is possible to employ a simple bath of boiling water at ordinary atmospheric pressure as the cooling medium, while still obtaining a greatly increased cooling rate as compared to earlier evaporation-cooling systems employing forced circulation of cool water under pressure.

The applicant's continued research in the field of evaporation cooling surfaces of the deliberately non-isotherm, stabilized temperature-gradient type described above, has now revealed the existence of certain specific mathematical relationships able to maximize the efficiency of such structures while somewhat modifying their manner of operation. It is to these relationships that the present invention is primarily directed.

Referring to FIG. 2 of the drawings, there is illustrated in cross-section a fragment of a wall of an evaporation-cooling system made of a suitable heat-conductive metal. The generally smooth surface to the left-hand side of the wall (as shown) is exposed to a source of heat to be dissipated. Its opposite right-hand side is swept by a body of vaporizable liquid, such as water at ordinary

pressure. The side of the wall exposed to the cooling liquid is formed with parallel spaced grooves or channels 4 defining between them complementary ridges or solid wall elements 3. As indicated in the drawing, the principal geometrical dimensions defining the structure are: a , the intergroove spacing or width of solid ridges 3 between adjacent grooves; b the depth of the channels or grooves as measured in the direction of the heat flux to be dissipated, indicated by arrow 5; d , the average width of a channel.

In accordance with the present invention, it has been found that optimum performance of such a heat-dissipating structure, in regard to stabilization of the temperature gradients and hence maximum permissible cooling rate, is obtained when the following relations are satisfied:

Firstly, the channel width d should be less than one third the depth b , and preferably within the range of from one-twelfth to one-fifth said depth, i.e.:

$$d < \frac{1}{3}b, \text{ preferably } \frac{1}{12}b < d < \frac{1}{5}b \quad (1)$$

Secondly, the width a and depth b of the solid wall elements separating the channels should be related to the thermal conductivity factor c of the wall material by the equation

$$b = m\sqrt{ac} \quad (2)$$

where m is a numerical constant which is of the order of unity when the dimensions a and b are expressed in centimetres and the thermal conductivity c is expressed in watts per cm. per degree C. In the case of the usual heat conductive metals used, such as copper and bronze, or non-metallic materials of appreciable conductivity such as graphite, and where the cooling liquid is water at atmospheric pressure, a preferred value for the factor m is 1.25. More generally, a satisfactory range for the value of m is from 0.7 to 1.8 preferably from 1 to 1.5. This broad range of values for the factor m appears to hold for vaporizable liquids other than water also.

The geometrical relationship expressed by inequality (1) broadly states that the channels are relatively deep, rather than being relatively shallow as in the earlier structure referred to above. This is a first significant difference of the present invention over said earlier structure. Equation 2, which is especially fundamental to the teachings of the present invention, brings in a mathematical relationship between the geometric dimensions of the cooling wall elements or ribs and the thermal conductivity factor of the material from which said elements are made, which relationship has not to applicant's knowledge been taught or foreshadowed in the prior art. This equation provides a means of optimizing the geometry of the cooling surface in the case of any metallic or other material that may be arbitrarily selected for use in the cooling structure according to circumstances. Conversely it makes it possible, in cases where the geometry of the structure is more or less pre-ordained, usually due to dimensional limitations in either sense, to select the cooling wall material so as to obtain maximum cooling rate despite such geometrical limitations. Thus Equation 2, especially when taken in conjunction with inequality (1), is of very great value in the design of novel heat-dissipating structures according to the invention and represents a remarkable advance in the art.

It may be noted that in Equation 2 the value of b represents an optimum when the values of a and c are given. That is, should b be taken less than the value derived from the equation, the resulting cooling rate is found to drop off sharply. If b is selected greater than the optimum value, the cooling rate may increase somewhat, but by an amount so small as to make the necessary increase in wall material unrewarding.

While it is not desired to limit the invention to any explanatory theory concerning the law expressed by Equation 2 above, some insight into the significance of the equation may be gained from the following explanation.

Experiments using essentially analogue (rheographic) methods have shown that for a given constant value of the dimensional ratio b/a , a given temperature differential across the cooling wall, and a given rate of heat flow through the wall surface 2 at the base of the elements, the desired stabilization of the continuous temperature gradient across the wall can be achieved throughout the full extent of the depth b of the element, in elements made of different materials, provided the said depth b is related to the thermal conductivity factor c of the material by a proportional relation, i.e. $b_1/c_1 = b_2/c_2 \dots$, wherein the subscripts 1 and 2 relate to two different materials. In other words, the relative dimensioning of two elements made of differently-conductive materials, should be in the same ratio as the ratio of the conductivity factors of the materials.

Hence, assuming the dimension b of an element is to be reduced in a certain ratio $1/p$, both its width dimension a and its conductivity factor c should each be simultaneously reduced by the same factor $1/p$ if the optimum cooling rate is to be retained. Again, if it is desired to make the element out of the same material as before, i.e. keep the factor c unchanged, the element of width a/p and depth b/p would be too conductive to ensure the desired gradient stabilization, and its thermal resistance should therefore be increased by further reducing its width by the same factor $1/p$, i.e. imparting to it the width a/p^2 . From a consideration of both the transformations just mentioned, it becomes clear that the relationship between the factors b , a and c , required for maintaining the desired stabilized temperature gradient, should be of such form as to hold true, both when each of the three variables b , a and c is divided (or multiplied) by a common factor p , and when variable b is divided by p and variable a simultaneously divided by p^2 . It is immediately verified that a relationship of the form (2) is the simplest that can satisfy these conditions. The truth of the relation has been confirmed by extensive tests.

The above specified relations (1) and (2), the result of extensive theoretical and experimental investigations conducted by the applicant, and his assignees, ensure when verified that the desired steady temperature gradients will become established throughout the depth b of the heat-conductive wall portions 3 even at extremely high rates of heat transfer. It should be noted that owing to the non-linear form of the relation between the dimensions a and b , no similitude is present as between structures of different size. The width/depth ratio is greater for larger elements than for smaller ones. Thus, with $m=1.25$, using copper as the conductive material ($c=3.7$), then for a width $a=1$ cm. Equation (2) specifies $b=2.5$ cm. ($a/b=0.40$), whereas with $b=1$ cm. the equation specifies $a=0.16$ cm. ($a/b=0.16$). It can therefore happen that in the case of very small-size structures, the thickness a specified for the inter-channel ridges by Equation 2 may become too small for practical considerations. In such cases it may be advantageous, somewhat paradoxically, to make the heat-dissipating wall structure out of a metal or other material having a relatively low heat conductivity factor c , since this, according to Equation 2, will increase the optimum thickness a specified for ridges 3 of given depth or length b .

It is important to note that when the mathematical relationships specified by the present invention are verified, the construction and operation of the resulting heat-dissipating structure differ significantly from any of the structure disclosed in the applicant's prior patents and applications mentioned above. In those of applicant's prior structures including parallel spaced grooves and ridges and to that extent generally comparable to the structures of the present invention, the width and depth of the grooves were disclosed as being generally of the same order of magnitude, contrasting markedly with the relatively narrow, deep channels of this invention. In FIG. 2, the dotted outline EFGH indicated a typical

cross-section of the wall structure disclosed in the aforementioned United States Patent 2,935,305. The differences in operation between the two types of structures will be explained with reference to FIG. 3, the left-hand side of which is a section through the longitudinal mid-plane of a channel according to the earlier construction, e.g., having a cross-sectional shape as at EFGH in FIG. 2, while the right-hand side is a similar longitudinal section through a channel according to the present invention, such as that shown in full lines in FIG. 2. Corresponding elements in the two structures shown in FIG. 3 are designated by the same numerals, followed by suffix *a* for the structure according to applicant's earlier patents. In each case the structure comprises a heat dissipating wall **1a** or **1**, having an inner face **2a** or **2** exposed to an intense source of heat as by electron bombardment in a high-power electron tube, and an outer side exposed to a cooling liquid such as boiling water. Numeral **3a** or **3** designates a solid wall portion or ridge between adjacent channels. The arrows **9a** or **9** indicate the flow of the cooling water. The body of steam that forms on contact of the water with the heat-dissipating elements **3a**, **3**, is indicated at **10a**, **10** respectively. In the case of the earlier structure, it is found as earlier described that this body of steam collects predominantly along the bottoms of the channels, substantially throughout the full length thereof, and that the outer regions of the ridges **3a** are in contact with liquid water as at **11a**, except towards the outlet end of the channels where the sheet of steam **10a** billows outward as indicated. With this configuration of the liquid and vapour phases, the continuous temperature gradient earlier mentioned as responsible for the effectiveness of the cooling structure, is established externally of the conical sheet of steam, and is liable to some instability at the water-steam interface, nor does it generally extend the full length of the structure as far as its output end (the upper end in the drawing). With the cooling structure constructed in accordance with the present invention, a significantly different liquid-vapour configuration is found to prevail, as is shown at the right of FIG. 3. Here the channels, due to the dimensioning used, act as steam generators and ejectors to expel the steam from their bottom outwards in directions generally normal to the wall surface. As a result the channels are substantially completely filled with liquid water and the sheet of water may even extend some distance outward beyond the outer tips **11** of the ridges **3**. The inner surface of the conical sheet of steam **10** thus lies along a surface spaced from said outer ends of the ridges and provides a positive boundary confining the liquid in the channels throughout their length. Whereas liquid water is supplied to the channels in the earlier structure through their outer sides, as indicated by arrows **9a**, in the structure of the invention the channels are generally fed with liquid water exclusively from the inlet end (preferably the lower end in the case of vertical channels) as shown by arrow **9**. The resulting conditions within a channel according to the invention are better illustrated in the cross-sectional view of FIG. 4. The channel is completely filled with a turbulent flow of liquid outwardly confined by the water-steam interface **13** of the external sheet of steam **14**, which prevents the ingress of liquid laterally into the channel. The side walls of the channel are thus subjected to temperature gradients of remarkably high stability. The temperature decrease steadily from the bottom of the channel outwards. In other words, the thermal conditions characterising the four regions A, B, C and D of Nukiyama's curve are distributed over the cross-sectional depth of the channel in the manner indicated by the letters A, B, C and D in FIG. 4, with the lowest-temperature region A (normal ebullition) being outermost. The peak point M representing the higher limit of the normal boiling range oc-

curs somewhat beyond mid-depth of the channel, as indicated, so that the intermediate area of the channel is filled with boiling water. Beyond this area (looking into the bottom of the groove) is the transitional region C from point M to point N, while beyond this again there may be a short extent, down to the base of the groove, in which skin vaporization obtains. This innermost region and partly also the transitional region MN, are therefore coated with a thin film of vapour **16** which does not impede the smooth flow of liquid water through the channel at **17**.

The flow configuration of the two-phase water-steam system thus obtained is, as already stated, extremely stable. Therefore the full extent of Nukiyama's curve, including the allegedly unstable transitional region MN, can be utilized at increased rates of heat dissipation without any danger of hot points and burn-out.

It will further be noted from FIG. 3 that due to the fact that virtually the entire capacity of the channels according to the invention is filled with liquid, instead of being partly filled with steam as in the earlier structures described, the invention achieves increased heat dissipating efficiency for a given volume.

FIG. 5 illustrates in perspective a generally flat heat dissipating wall according to what is the simplest form of embodiment of the invention. It is noted from this figure that the length dimension of the ridge and channel formations, designated *e*, should preferably be substantially greater than the width dimension *a*. In cases where the dimension *e* is relatively short with respect to dimension *a*, the relations of the invention still remain applicable, provided the numerical coefficient *m* is somewhat decreased correspondingly.

FIG. 6 illustrates a heat dissipating wall according to the invention which differs from the preceding one firstly in that its general shape is arcuate, e.g., cylindrical. The side walls of the channels are shown radial, and hence somewhat diverging. The channel width dimension *d* should in this case be measured as the average width of the channel over its depth. The same applies to all cases where the channels are not uniform in width over their depth, it being understood that the invention is applicable to such embodiments. An additional difference embodied in FIG. 6 is that there are provided in this case two sets of spaced channels, intersecting at right angles, and shown at **4** and **8** respectively. The equations of the invention may apply to both sets of channels and corresponding ridge dimensions. In this respect it should be noted that in the case of a two-dimensional array of channels as in FIG. 6, the length dimension *e* may or not be large with respect to the width dimension *a*, and in the latter case the coefficient *m* in Equation 2 should be somewhat decreased as earlier mentioned.

Where the channels of the invention have considerable length, and in view of the fact that as previously indicated the liquid water is prevented from entering the channels from their outwardly opening sides due to the surrounding sheet vapour, means may advantageously be provided for facilitating the entry of the liquid into the innermost areas of the channels such as **17** (FIG. 4). For this purpose inlet recesses such as **18** (FIG. 7) may be provided at spaced points in the side walls of the channels, extending from the outer surfaces **20** thereof down to a depth corresponding substantially to the boundary of the boiling zone (as previously described) and therebeyond tapering in section down to the bottom of the channel. The cross-sectional width *f* of such inlet recesses should be large enough to enable the liquid to flow in against the outward flow of steam **10**; the general flow path of the incoming liquid is indicated by the arrows.

In the modification of FIG. 8, the channels are formed with enlargements **19** defining cylindrical ducts extending the length of the channels and facilitating the in-

take of liquid as earlier indicated. The ducts 19 may, as shown, be alternately offset as between adjacent channels to avoid weakening the intervening ridges excessively; however, they are preferably positioned so as to communicate generally with the boiling region in each of the channels.

According to the further modification shown in FIG. 9, the inter-channel ridges 3 are in turn capped with heat-dissipating structures of generally similar character to the main structures, including secondary channels 4b defined between secondary ridges 3b. Desirably, the dimensioning of the secondary channels and ridges is in accordance with the same mathematical relations described herein in connection with the primary structures 3 and 4. The provision of the secondary structures increases the surface contact area with the fluid and improves the local cooling of the tips 20 of the main ridges 3, thereby further increasing the stability of the temperature gradients, as described per se in applicant's copending U.S. patent application, Serial Number 25,799, filed April 29, 1964. With this arrangement the liquid-vapour pattern of the system is somewhat altered from that shown in FIGS. 3 and 4 earlier described, in that some local boiling of the liquid will usually occur at the tips 20 of the main ridges. It is found that with this arrangement the overall performance of the heat-dissipating system can be further increased without danger of burn-out.

Heat-dissipating structures constructed in accordance with the present invention have made it possible to use rates of heat dissipation of the order of one kilowatt per square centimetre of heat surface in a continuous manner, using as the cooling liquid water at atmospheric pressure without forced circulation, the general temperature of the water remaining at about 100° C. This remarkable result can be further enhanced by the use of such conventional expedients as pressurizing the cooling fluid, forced circulation, external cooling of the water to a temperature below its boiling point, and use of cooling liquids other than water, such as fluorine compounds.

The invention is susceptible to a great variety of applications. In addition to its use in high-power electron tubes already mentioned, it can be used in cooling structures for combustion engines and other heat machines and chemical reactors.

Various other applications of the invention to the field of cooling systems and more generally to heat exchange structures will occur to those familiar with the art and should be deemed as lying within the scope of the present invention.

I claim:

1. In a heat dissipating structure comprising a wall of heat-conductive material having one side exposed to a source of heat to be dissipated and its opposite side exposed to a vaporizable liquid and wherein said opposite side of the wall is formed with massive protuberances defining therebetween spaced narrow channels, the improvement characterized in that said protuberances and narrow channels are so dimensioned as to verify substantially the relations

$$d < \frac{b}{3}$$

and

$$b = m\sqrt{ac}$$

wherein d represents the average transverse width of a narrow channel, b represents the depth of a narrow channel, a the transverse width of a solid wall portion defined between adjacent narrow channels, c the heat conductivity factor of the wall material, and m a numerical coefficient within the range from about 0.7 to about 1.8 when a and b are expressed in centimeters and c in watts transmitted heat per centimeter and per degree centigrade.

2. Heat dissipating structure as claimed in claim 1, wherein the channel depth is greater than about one half the total wall thickness.

3. Heat dissipating structure as claimed in claim 1, wherein the channels are parallel to each other.

4. Heat dissipating structure as claimed in claim 1, wherein said liquid is water and said numerical coefficient m is within the range from about 1.0 to about 1.5.

5. In a heat dissipating structure comprising a wall of heat-conductive material having one side exposed to a source of heat to be dissipated and its opposite side exposed to a vaporizable liquid and wherein said opposite side of the wall is formed with massive protuberances defining therebetween spaced narrow channels, the improvement characterized in that said protuberances and narrow channels are so dimensioned as to verify substantially the relations

$$\frac{1}{12}b < d < \frac{1}{5}b$$

and

$$b = m\sqrt{ac}$$

wherein d represents the average transverse width of a narrow channel, b the depth thereof, a the transverse width of a solid wall portion defined between adjacent narrow channels, c the heat conductivity factor of the wall material, and m a numerical coefficient of the order of unity when a and b are expressed in centimetres and c in watts transmitted heat per centimetre and per degree centigrade.

6. Heat dissipating structure as claimed in claim 5, further including inlets spaced along the length of the narrow channels for feeding liquid into the narrow channels to areas adjacent the bottom walls thereof.

7. Heat dissipating structure as claimed in claim 5, wherein the narrow channels are formed with longitudinally extending enlargements in an area adjacent the bottom walls thereof.

8. Heat dissipating structure as claimed in claim 5, wherein the solid wall portions between the narrow channels are provided along their outer surfaces with extensions defining secondary parallel spaced narrow channels.

9. Heat dissipating structure as claimed in claim 5, wherein the narrow channels are parallel to each other.

10. In a heat dissipating structure having a wall of heat conductive material exposed on one side to a heat source and on its other side to a vaporizable liquid and wherein said other side is formed with massive protuberances defining therebetween parallel spaced narrow channels, the improvement characterized in that said narrow channels are provided in two sets angularly disposed to one another, with the narrow channels in at least one set so dimensioned and spaced as to substantially verify the relations

$$d < \frac{1}{3}b$$

and

$$b = m\sqrt{ac}$$

wherein d represents the average transverse width of a narrow channel, b the depth thereof, a the transverse width of a solid wall portion between adjacent narrow channels, c the heat conductivity factor of the wall material, and m a numeric coefficient within the range from about 0.7 to about 1.8 when a and b are expressed in centimeters and c in watts transmitted heat per centimeter and per degree centigrade.

11. Heat dissipating structure as claimed in claim 10, wherein the narrow channels of at least one set are substantially parallel to each other.

12. Heat dissipating structure as claimed in claim 5, wherein said liquid is water and said numerical coefficient m is in the range of from about 1.0 to about 1.5.

13. Heat dissipating system comprising a wall of heat conductive material having one side exposed to a source of heat to be dissipated, massive protuberances defining therebetween at least one set of spaced narrow channels

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formed on the opposite side of said wall, and means for supplying a vaporizable liquid to the one end of all said narrow channels, said protuberances and narrow channels being so dimensioned as to verify the relations

$$d < \frac{b}{3}$$

and

$$b = m\sqrt{ac}$$

where d represents the average transverse width of a channel, b the depth of a channel, a the transverse width of a solid wall portion defined between adjacent channels, c the heat conductivity factor of the wall material, and m a numerical factor which is of the order of unity when a , b and d are expressed in centimeters and c is expressed in watts per centimeter and per degree centigrade, whereby in operation said liquid will fill substantially the full volume of said narrow channels in the liquid phase while vapour of said liquid will form a substantially continuous sheet outwardly covering said narrow channels, the side

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walls of said narrow channels being subject to continuous, stable, temperature gradients increasing steadily towards the bottom of the narrow channel, with the temperature in said gradients covering a range extending continuously both below and above the peak point in Nukiyama's curve and whereby local burn-out through spheroidal state effect will be positively prevented.

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