July 15, 1969 C. A. BEURTHERET 3,455,376

HEAT EXCHANGER

Filed Aug. 22, 1967 4 Sheets-Sheet 1

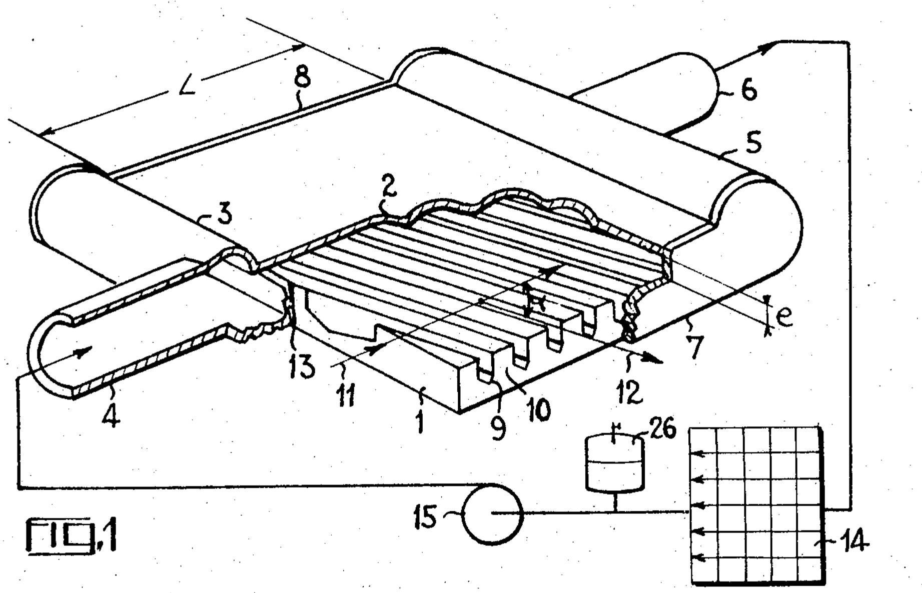

FIG. 1

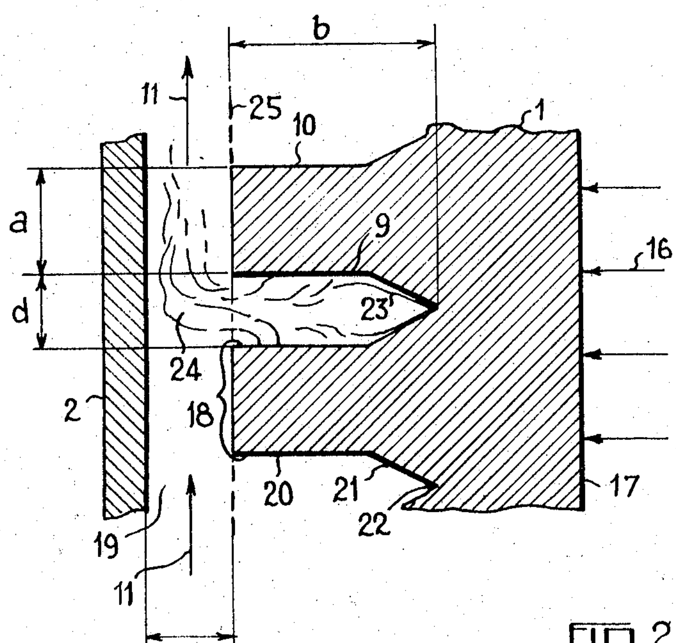

FIG. 2

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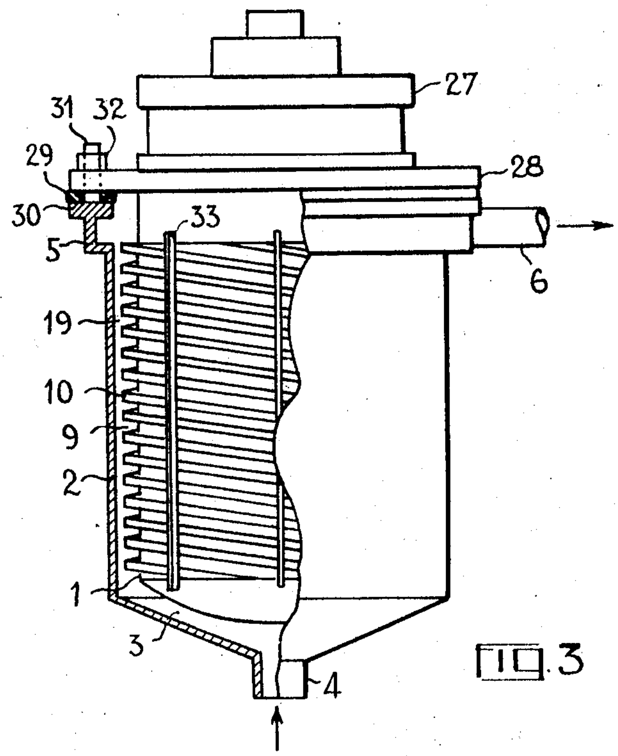

FIG. 3

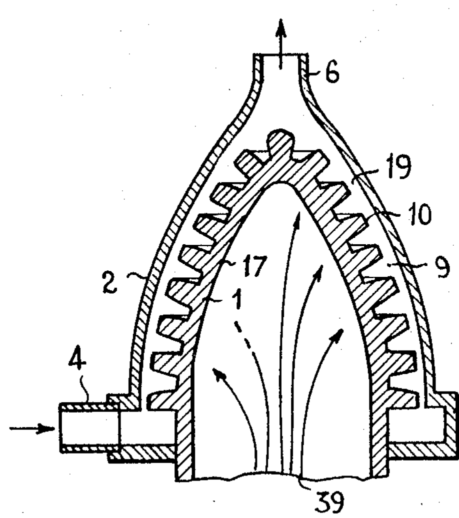

FIG. 5

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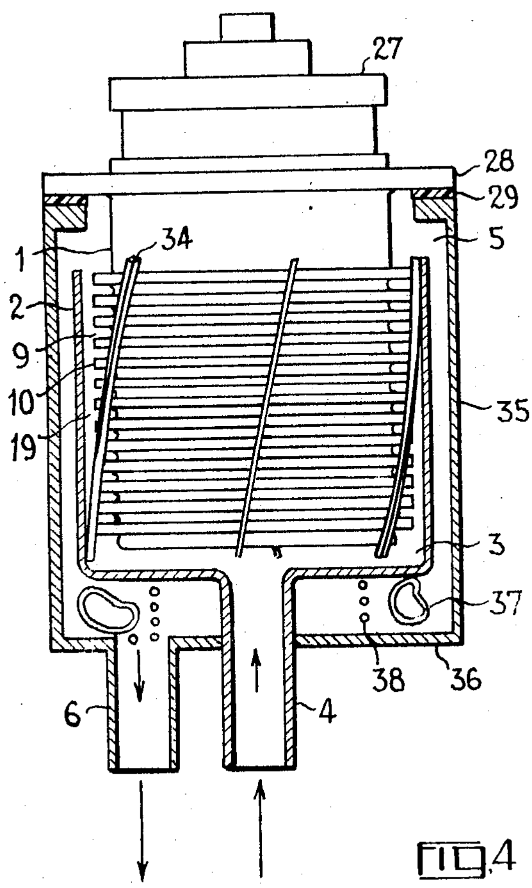

FIG.4

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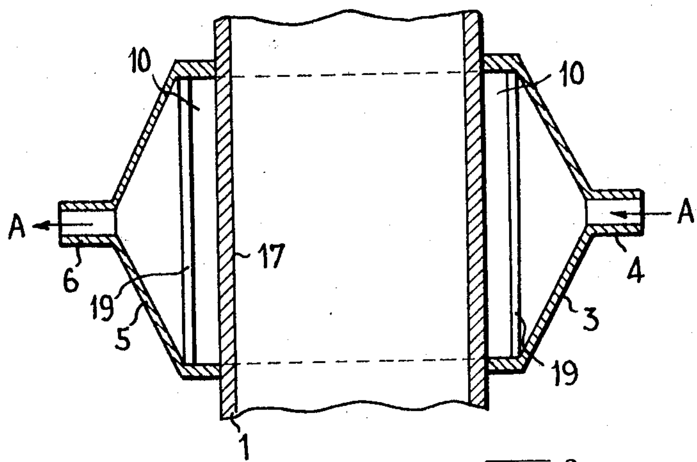

FIG.6

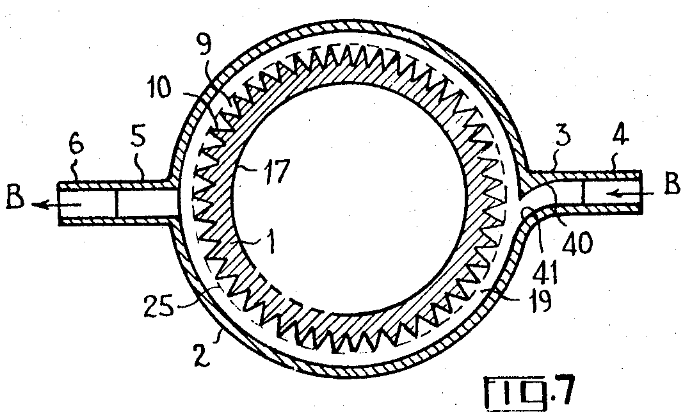

FIG.7

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HEAT EXCHANGER

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Claims priority, application France, Sept. 15, 1966, 76,446
Int. Cl. F28d *15/00;* F28f *7/00*
U.S. Cl. 165—1 16 Claims

ABSTRACT OF THE DISCLOSURE

A heat transfer surface having grooves formed therein has a forced circulating heat removing liquid confined between this surface and a guide wall, the flow of liquid being essentially in a direction transverse to the longitudinal direction of the grooves.

The present invention concerns improvements in methods and devices for heat exchange between a wall and a liquid circulated on contact therewith. It is more especially concerned with devices of this type in which the transfer of an intense thermal flux is effected essentially by local vaporization of liquid on contact with the hot wall, and recondensation of the vapor thus formed in the mass of the circulating liquid.

It is known that local vaporization is a fundamentally random and unstable phenomenon, susceptible to give rise to an irreversible overheating or burnout, leading normally to the destruction by local fusion of the exchanger wall. The burn out occurs if the source fo heat imposes a density of thermal flux which is greater than a critical flux, the vaule of which depends on the nature of the liquid used. By lowering the temperature of the liquid contacting the exchanger wall, the appearance of this destructive phenomenon will merely occur at a higher thermal flux; it will not eliminate it.

For a long time, the best procedure recommended to increase the thermal flux in exchanges having circulating liquid consisted in avoiding the occasional formation of boiling points on the hot wall. To this end, there has been employed overpressure, which increases the boiling point of the liquid, and moreover, turbulent flow, which quickly tears away the forming bubbles. This latter comprises the use of obstacles such as turbulence creators, secured either to the exchanger wall or the auxiliary wall, to guide the flow of liquid in a narrow space, tangential to the exchanger surface. Thus, on anodes of electronic tubes or on nuclear fuel elements cooled by circulation of liquid, obstacles in the form of ribs arranged transverse to the liquid current are formed. These ribs must be of small height and far enough away from each other so that the creation of nests of bubbles will not occur, which could be considered as establishing hot points capable of triggering the burnout phenomenon. If it is desired to designate by "throats" the intervals between ribs, these throats are thus wider than they are deep; recent research work has confirmed this practice in recommending, moreover even accentuating this ratio of dimensions in making the width of the throats up to ten or even twenty times their depth. In these conditions, the depth is then smaller than a millimeter, and the entire heat exchanger surface operates practically isothermically.

In the thus improved exchangers having induced liquid circulation, it is possible to permit a particular boiling operation, namely, "surface boiling." In this, the vapor bubbles which form are nearly immediately condensed in the cold liquid violently sweeping the exchanger surface. Especially under high pressure, this phenomenon contributes to increase the density of flux admitted before burn out appears; but it is only obtained on essentially isothermic surfaces, which prevents increasing, to any extent, the surface of the exchanger by ribs or wings participating in the transmission of the heat.

The increase in thermal dissipation by the improvements described does not satisfy all technical needs. Additionally, they have the disadvantage of requiring a high liquid pumping force due to the very strong turbulence.

In one field of research, in contradistinction to the above, the work of the applicant has resulted, since 1950, in very advantageous embodiments following from the systematic use of intense boiling, rendered possible by a particular shape of the exchanger wall. The active surface thereof is increased by the presence of dissipating extensions, and these extensions are dimensioned, as a function of the thermal conductivity, so that essentially anisothermic extensions of the exchanger surface are presented to the liquid. The temperature gradient which is established on these surfaces maintains a stable vaporization condition called "complex vaporization." It concerns an artificial phenomenon, resulting in the reciprocal stabilization of two phenomena which are naturally unstable: for example, the method of succession of different forms of vaporization as a function of the temperature of the wall (in accordance with the Nukiyama curve) and the evolution in time of the temperature itself in the presence of these forms of vaporization. This complex vaporization, as well as the means permitting it to form, has been described in an article by the inventor, published at Comptes Rendus Acad. Sc. Paris, volume 259, pages 519 to 522, July 20, 1964, Thermocinetic.

Improvements likewise due to the applicant have permitted exchangers using complex vaporization to become even more effective. In a particular type of exchanger, the exchanger wall comprises parallel ribs separated by narrow, deep throats. This exchanger is capable of transmitting a thermal flux several times greater than the critical flux of the liquid, and this even if the liquid is already at its boiling temperature. Its performance is even more increased in the presence of a cooler liquid, by natural convection. Rapid circulation forced on the liquid pushing the dissipation of this exchanger even higher as in exchangers having induced circulation.

Experiments have shown that increase in speed of circulation does not result in a predictable increase in the dissipation, particularly when the liquid flow is high in volume and speed. Test results do not correspond to supposed extrapolation of performances of the same device in its natural convection functioning.

The improvements proposed by the invention concern all devices for heat exchange between a wall and a liquid by local evaporation, accompanied by a recondensation in the mass of moving liquid, in which a heat transfer surface of the heat exchanger wall is formed with depressions, or grooves, in particular in the form of throats, which is exposed to a forced liquid circulation in a confined space, limited by a guide wall.

SUBJECT MATTER OF THE INVENTION

The depth of the depressions or throats is chosen greater than the distance separating their opposed edges; contrary to prior practice liquid is forcibly circulated in a direction of flow which is transverse to the longitudinal direction of the throats, that is to say, a direction forming locally angles comprised between 45° and 90°, preferably between 60° and 90°, with the longitudinal direction of the throats.

The invention is based on the resulting differing paths of flow presented to the liquid from the entry to the outlet of the device, namely a first path essentially external to the overall geometric volume of the exchanger wall and a second path essentially internal to this overall volume, and formed by the assembly of throats. Common sense suggests that the most effective heat exchange would be obtained when the liquid flow through the second path is as large as possible, the passage through the throats ensuring, at first blush, intimate contact between the liquid and the wall, whatever be the form of the throats. But, in an exchanger in which the exchanger wall comprises throats which are deeper than they are wide, the invention as defined above, proposes the contrary, namely, to create a main flow path exteriorly of the wall and the throats, in the confined space, limited by the overall surface of the wall of the exchanger and the guide wall. In other words, if the two flow paths which have just been defined are considered as being distinct, the path constituted by the overall assembly of throats in the heat exchanger in accordance with the invention, will have a large hydrodynamic resistance relative to the path defined by the confined space, between the heat exchanger wall and the guide wall.

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

FIG. 1 is a perspective view, the wall being shown partly removed to reveal the structure of the exchanger wall therebeneath, of a heat exchanger with a plane exchanger wall;

FIG. 2 is an enlarged section of a part of the exchanger of FIG. 1 in a plane perpendicular to the exchanger wall and arrow 11 of FIG. 1;

FIG. 3 is an elevational view of a heat exchanger comprising a cylindrical wall provided with helichordal throats, forming the anode of an electronic tube, a part of the casing of the exchanger being removed;

FIG. 4 is a part sectional elevation of a heat exchanger comprising an exchanger wall provided with circular throats, the exchanger wall and flow director elements being shown in elevation, the remainder of the device being shown in section;

FIG. 5 is a fragmentary section of a heat exchanger comprising a rounded conical exchanger wall provided with circular throats;

FIG. 6 is a section through a heat exchanger comprising a cylindrical exchanger wall provided with throats parallel to its axis, this section being taken on the line B—B of FIG. 7; and

FIG. 7 is a section through the heat exchanger shown in FIG. 6 taken on the line A—A of FIG. 6.

In the drawings, like reference numerals designate like parts.

The heat exchanger shown in FIGURES 1 and 2 is constituted by an exchanger wall **1**, a guide wall **2**, disposed parallel to the wall **1** and at a short distance *e* therefrom, a liquid distribution chamber **3**, provided with an inlet tube **4**, a collecting chamber **5**, provided with an outlet tube **6**, and two lateral closing walls **7** and **8**. The exchanger wall **1** comprises a rectangular metal plate, for example, in copper, and its heat transfer face which is visible in the figures comprises a network of depressions in the form of parallel throats **9**, separated by ridges or ribs **10**. As is shown in FIGURE 2, the depth *b* of these throats is clearly greater than the distance *d* separating their edges. The chambers **3** and **5**, which are arranged along the two opposite edges of the exchanger wall **1**, communicate with the space comprised between this wall **1** and the guide wall **2**. The throats form angles of less than 45° with each of these two edges. A flow of liquid entering through the tube **4** thus forms, between the exchanger wall **1** and the guide wall, a sheet flowing in one direction, designated by the arrow **11**, which is substantially that of the lines connecting the chambers **3** and **5** the shortest way. The direction of flow thus forms, with the longitudinal direction **12** of the throats, an angle comprised between 45° and 90°. Only a small proportion of the output of liquid is diverted in the direction **12** of the throats and flows thereinto. In the example shown, the throats do not themselves discharge into the chambers **3** and **5**, their entry being prevented by a part of a casing **13** of these chambers. As a varient of this arrangement, discharge can be effected into one or both chambers, but in both cases, the hydrodynamic resistance of the direct flow path from the distribution chamber to the collecting chamber is less than that offered by the assembly of the network of liquid flowing in the throats. In FIG. 1 there is also shown, in a very schematic manner, outer accessories, ensuring the circulation of liquid in a closed circuit. The liquid discharging from the outlet tube is cooled in the secondary chamber **14**, of known construction, and reinjected by a pump **15**, into the inlet tube **4**. The installation comprises, moreover, a reservoir **26**, if desired pressurized, and conventional safety devices not shown.

The heat exchange process being effected in the device of FIGURE 1 is explained with reference to FIG. 2. Arrows **16** symbolize the heat flux entering by the inlet surface **17** of the wall **1**. A small part of this heat is exchanged by direct conduction between the terminal parts **18** of the sides **10** and the liquid which circulates rapidly on contact therewith in the main flow path **19**, between the entire surface **25** of the exchanger wall and the guide wall **2**. However, for the larger part, the exchange is effected in the throats **9** by vaporization of the liquid which they contain, by a complex boiling operation stabilized by the gradient of temperature which is established on the sides of the ribs **10**, and this with a mean density of flux in the neighborhood of the critical flux, the value of which no longer constitutes a limit. Assuming that the liquid is water under an absolute pressure of two atmospheres (one atmosphere of overpressure) for which the boiling temperature is 120° C., that the temperature of this liquid is already up to 95° at the moment when it reaches the throat shown in the center of the figure, and that the mean speed in the space **19** is several meters per second. In such conditions there are found, for example, in the end zone **18** of the ribs a temperature in the neighborhood of 100° C. and on a zone extending along their sides, between the points **20** and **21**, temperatures ranging from 135° to 250° C. favoring complex boiling. Lower down, towards the base **22** of the throats, the temperature is even higher and gives rise to a purely film like vaporization. There is schematically represented the vapor film **23** which rejoins the zone **22–21** in which all types of vaporization coexist. The vapor **24** thus formed, at a temperature of 120° C., escapes at high speed out of the throat, transversely to the main flow direction of the liquid, symbolized by the arrows **11**. There results an almost immediate condensation by mixing in a turbulent manner, transferring to the liquid the latent heat of the vapor and, of course, the quantity of heat of smaller amount, corresponding to the difference between its own temperature and that of the liquid. The two successive exchange processes, vaporization-condensation, already co-exist in the throats themselves but, in fact, the boiling takes place essentially at the base and on the sides of the throats, while the condensation is produced near the opening, and essentially in the junction zone with the liquid flowing in the space **19**, on the outside of the exchanger wall.

In the present improved exchanger device, the liquid to be vaporized in the depressions of the exchanger wall is introduced thereinto by branching off from a small part of the main flow which is arranged to occur outwardly of the wall **1**. This branching results on the one hand from the component of speed which the flow can present in the preferred direction of the depressions—the longitudinal direction **12** of the throats in the example of FIGURE 1—and, on the other hand, in the turbulence of the liquid induced due to the component of its direction **11**, transverse to the depressions. This latter fact is only due from the necessary branching, if the flow direction no longer possesses a component in the longitudinal direction, which would be the case if, in the example of FIG. 1, the angle were to be 90°. It is above all the angular deviation between the direction of current of the liquid and that of the throats which ensures to exchangers, constructed in accordance with the invention, its superiority over those of the prior art, in which the throats in themselves conducted a large amount of liquid current.

To arrive at the concept of the present invention, the following reasoning was taken into consideration:

(a) The respective quantities of liquid necessitated by the two consecutive heat exchange processes, the vaporization and the recondensation, are very different. In fact, the boiling uses the vaporization heat of the liquid, while for the condensation, it is the specific heat of the liquid which is concerned. For example, if the liquid is water, its latent heat of vaporization is greater than 500 calories. while its specific heat is one calorie per ° C. It can be seen that, even if an overall heating of 50° C. of the liquid crossing through the exchanger is admitted, the boiling phenomenon only requires putting into use a tenth part of the total output. In the prior art exchangers, there has been circulated, through the throats, an output several times greater to that necessary for vaporization.

(b) A very intense circulation invoked in the throats of the boiling exchangers of the prior art is not only superfluous but detrimental. A too rapid flow of liquid longitudinally imposed in the throats, moreover, if it is turbulent, tends to tear out the bubbles of vapor which form on their walls. Such an effect has been sought in exchangers constructed for functioning with changing of phase; but, in exchangers in which a temperature gradient stabilizes boiling, this violent action of the liquid current on the surface of the exchanger is not only superfluous but undesirable, as it upsets the establishment of a complex vaporization operational condition.

In the devices constructed in accordance with the invention, these difficulties are avoided, because a minor part of the total output can pass longitudinally through the throats, and the major part of the liquid flows in directions which are essentially transverse to the throats. In the FIG. 1 embodiment, the main required direction of flow is obtained by the arrangement, along the two opposite edges of the wall **1**, of a distribution chamber **3** and of a collecting chamber **5**. Other means will be described with reference to the examples shown in FIGURES 3 to 7. It has been ascertained for all values of the angle α greater than 45°, an improvement in performance is felt with respect to prior art exchangers, that above $\alpha=60°$ the present device presents a notable improvement and that the best results are obtained for angles comprised between 80° and 90°. Even for $\alpha=90°$, the feed of the throats is still sufficient for the needs of vaporization up to operational conditions of dissipation which are extremely high. This latter result is explained by the fact that, in the present exchanger, high external circulation speed can be applied without effecting an excessive flow in the throats. The high turbulence of the liquid which results then makes a quantity of liquid, sufficient for the needs of the vaporization, divert towards the interior of the throats.

The external appearance of these exchangers is superficially similar to certain prior art devices having liquid circulation without phase change; but for the reasons which have been stated, they permit the transference of densities of flux five to ten times greater than those permitted for these prior art devices, and this with a reduction of the output of liquid and of the power of the pump, the liquid being able to be taken to a high output temperature, slightly less than its boiling temperature, under the applied pressure.

It is interesting to note that the present exchangers do not have the tendency to retain tartar from the use of calcinated water. It has been found experimentally that tartar deposits are spontaneously eliminated by the current of water. It would appear that this property is due to the combined effects of the gradient of temperature in the throats and of the turbulent flow in their immediate neighborhood.

The width e of the space in which the main flow is confined is not very critical but, when determining this width, there must be taken into account that the efficiency of heat exchange by condensation is increased with the speed and the turbulence of the liquid in circulation. A speed of several meters per second is already sufficient to ensure good performance. As the angle α approaches 90°, the speed may be increased. Keeping speed constant requires choice of a width e which increases proportionally to the length of the main flow path, since the necessary output is proportional to the power to be dissipated, and thus to the length of the exchanger wall in the main flow direction. There have been obtained good results with widths e equal to

$$k\frac{L}{100}$$

where L is the length of the main flow path and k a number between 0.3 and 3. Whatever be the value of k, the temperature received at the outlet of the exchanger can be in the neighborhood of its boiling temperature, for example, in the region of 100° C. for water under normal atmospheric pressure, and 120° C. for a pressure of 2 atmospheres. From this there result the following points of view for the choice of the factor k:

For relatively large values of k, the hydrodynamic resistance of the device and consequently the loss of heat are small; a high output of liquid is heated little in contact with the exchanger wall. For example, water with flow of 1 liter per minute and per kilowatt, is heated by 15° C. Moreover, such conditions are very advantageous for a circulation of distilled water in a closed circuit, as the temperature of the liquid can be fairly high in the entire circuit and the slight necessary lowering of the temperature is then obtained in a secondary, simple exchanger such as a ventilated radiator.

On the contrary, for relatively small values of k, the hydrodynamic resistance is high, thus compatible with small outputs which have high heating; for example, of 75° C. with flow of water of 0.2 liter/min./kw. The inlet temperature must, in this case, be fairly low, so that, for example, town water can be used without reuse, since only small quantities are needed; further, there is obtained as a byproduct some very hot water which can be put to various uses.

In so far as the dimension of the throats and of the ribs which separate them are concerned, the general relation $b>d$ is preferred. It is further desirable to obtain on a large portion of the surface of the sides and of the throats a gradient of temperature extending between suitable limits to provide for complex boiling, in other words, operation based on the Nukiyama curve should be on the first ascendent leg, the descendent leg and possibly even a portion of the second ascendent leg. This is achieved by using the following dimension formula:

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

where b is the depth of the throats, defined previously and measured in centimeters, a the mean width of the straight section of the ribs measured in centimeters, c the thermal conductivity of the constituent material of the exchanger wall, measured in watts/cm., degree centigrade, and m a numerical factor of the order of 1, preferably comprised between 0.7 and 1.8. For the width d of the throats, there are advantageously adopted values which are less than ⅓, preferably less than ⅕, of their depth b thus defined. Similar dimensions, although for heat exchangers with recondensation, have already been given previously by the applicant.

In the case of a relatively high width e of the main flow path, for example of several mm. an advantageous arrangement, likewise covered by the invention, consists in cutting off the ends of the ribs separating the throats in the form of crenellations. This arrangement locally increases the turbulence of the liquid in the opening region of the throats. Moreover, it accentuates the specific effect sought by the invention, namely, supplying a sufficient quantity of liquid in the throats without resulting in a uselessly high flow in the direction of their length. In the same way the ribs can be cut in a direction which is substantially orthogonal, by a reduced number of draining members or channels, which contribute to the feeding of the throats without effecting a flow in the direction of their length.

FIG. 3 shows a heat exchanger constructed in accordance with the invention, the exchanger wall **1** of which, in cylindrical form, constitutes the outer anode of an electronic tube **27**. The guide wall **2** forms, with the distribution chamber **3** and the collecting chamber **5**, a casing by the open end of which there is introduced the anode of the electronic tube until a flange **28** surrounding the anode abuts against a water-tight joint **29**, itself mounted on a ledge **30** in the said casing. Wing nuts, such as **31**, **32**, ensure the connection and the water-tightness of the assembly. The exchange device thus constructed presents again the following differences with respect to those of FIG. 1; the throats have a rectangular section and form parallel helices coaxial with the wall of **1**; the throats discharge not only into the distribution chamber **3** but also into the collecting chamber **5**; finally, the main direction of flow substantially transverse to the direction of the throats, is ensured by the joint effect of the chambers **3** and **5**, disposed at the ends of the cylindrical guide wall **2**, and of a series of direction blades **33**, disposed parallel to the axis of the device in the confined space **19**. These blades can be fixed either to the guide wall **2** or to the exchanger wall **1**.

The exchanger shown in FIG. 4, serving likewise to cool the anode of an electronic tube, presents with respect to the proceeding example the following differences: the ribs are disposed circularly around the anode, and the bases of the throats are rounded. The main flow is directed to be slightly inclined with respect to the generatrices of the cylindrical exchanger wall, by a series of blades **34**, extending, as seen obliquely, between the exchanger wall **1** and the guide wall **2**. In the example shown, the guide wall does not constitute the complete envelope of the device. It is surrounded by a spaced sleeve **35**. A dilatable elastic body **37**, for example, in an outlet tube **6** inserted in its flat base. The heated liquid, discharging from the chamber **5**, thus passes through the annular space between the wall **2** and the sleeve **35**. A dilatable elastic body **37**, for example, in the form of a hollow rubber toroid kept in place by a grid **38** is located in this space. Such an elastic body deadens and even eliminates certain abrupt pressure variations which can accompany the condensation in the turbulent system in the confined space.

In the exchanger shown in FIGURES 3 or 4, the throats can, for example, present a width of d of 2 mm. and a depth b of 7 mm. The width a of the ribs can likewise be 2 mm. and the distance e between the ribs and the guide wall **2** may be 0.3 to 3 mm. Under these conditions, the changer wall, present an inlet surface for heat of 150 cm.^{2}, is capable of continuously dissipating more than 250 kw., with a water flow in the order of one liter per second, the water temperature at the outlet being of the order of 100° C.

FIG. 5 shows an exchanger, the exchanger wall **1** of which constitutes the anode of an electron beam tube. In accordance with a known technique, the impact surface for the electrons, thus the heat receiving surface **17** of the wall **1**, is in approximately conical form, electrostatic or magnetic means being provided for diverging the electronic beams **39** so as to give a good distribution of the thermal load on the surface **17**. The wall **1** comprises outer throats **9** disposed in circular formation. The distribtution chamber **3** and the outlet tube **6** are arranged in such a manner that the liquid flows along the generatrices of the conical wall, thus at an angle α of 90° with respect to the direction of the throats. The distance between the exchanger wall **1** and the guide wall **2** increases from the base of the cone towards its apex, so that the flow speed is substantially the same along the whole exchanger surface. The circulation of the liquid can be established at random in the direction of the arrows or in the opposite direction.

The heat exchanger shown in FIGURES 6 and 7 possesses, as in the exchangers shown in FIGURES 3 and 4, a cylindrical exchanger wall but with the difference that in that exchanger, the two ends of the cylindrical body to be cooled are outside the heat exchange device, an arrangement which would be adopted, for example, for cooling the cylinders of internal combustion engines. In the exchanger shown in FIGURES 6 and 7 the throats **9** are arranged parallel to the axis of the wall **1** and present as do the ribs **10** which separate them, a section of triangular form. Around the wall **1** and at a constant distance from the spherical surface **25**, there is disposed a guide surface **2** with which there are associated a distribution chamber **3**, provided with an inlet tube **4**, and a collecting chamber **5**, provided with an outlet tube **6**. The lower lateral walls **40** and **41** of the distribution chamber are concave so as to inject the liquid tangentially into the confined space **19**. Due to this arrangement, the main flow of liquid is constituted by an annular sheet in rotational movement around the exchanger wall, thus perpendicular to the direction of the throats **9**. The flow of this liquid in circular motion can be greater than that entering by the chamber **3** and leaving by the chamber **5**. It is not necessary for these two chambers to be disposed in diametrically opposite regions of the guide wall **2** as in FIGURES 6 and 7 shown by way of example only.

The heat exchanger in accordance with the invention has been described in structures to cool electronic tubes and parts of thermal motors. The invention may also advantageously be applied to different apparatus, in which intense thermal energy must be removed, for example, to parts of chemical reactors and to nuclear fuel elements. In certain of its applications, there can be grouped together several of these parts in the same casing, for example, several cylinders of a thermal motor or clusters of nuclear fuel elements. The casing of the device can be used to constitute the guide wall or a part thereof, other means for guiding being disposed inside the assembly for directing the main flow in the required direction relative to the throats of each of the exchanger walls. In the case of curved walls, for example, cylindrical ones, the convex as well as the concave surface can form the outlet surface for heat. In particular applications where two heat exchange walls face each other, these can be disposed in such a manner that each serves as a guide wall for the other.

I claim:

1. A heat exchange arrangement to transfer heat from a heated wall to a circulating heat removing liquid by local evaporation accompanied by recondensation in the mass of the liquid, comprising

a heat transfer surface formed with a plurality of longitudinally extending throats having side walls;

a guide wall adjacent said heat transfer surface to confine said liquid between said heat transfer surface and said guide wall, the depth of each said throat being greater than the distance separating the edges between any one pair of side walls of said throats;

means to forcibly direct said liquid over said heat transfer surface in a direction of flow which, with respect to the longitudinal direction of said throats, forms an angle α of from 45° to 90°;

and an inlet and an outlet for said liquid forcibly circulated between said transfer surface and said guide wall in said direction.

2. A heat exchange arrangement according to claim 1, wherein said liquid flow directing means comprising a distribtuion chamber arranged adjacent said inlet and a collecting chamber adjacent said outlet, said chambers being contiguous with a confined space limited by the heat transfer surface and the guide wall, and extending in a direction which is between parallel and 45° with respect to said throats, whereby the direction of flow will be essentially over said throats.

3. A heat exchange arrangement according to claim 1, wherein the transfer surface is cylindrical and the throats are of helical form arranged between the distribution chamber and the collecting chamber.

4. A heat exchange arrangement according to claim 1, wherein said liquid flow directing means comprises guide blades disposed in the space confined by the heat transfer surface and the guide wall.

5. A heat exchange arrangement according to claim 1, wherein the angle α is from 80° to 90° whereby the direction of said flow will be essentially transverse to said grooves.

6. A heat exchange arrangement according to claim 1, wherein said heat transfer surface is a surface of revolution and the throats are formed circularly and coaxially with respect thereto.

7. A heat exchange arrangement according to claim 1, wherein said heat transfer surface is cylindrical and the throats are formed on a plurality of helices.

8. A heat exchange arrangement according to claim 1, wherein said heat transfer surface is cylindrical and said throats are formed longitudinally thereof.

9. A heat exchange arrangement according to claim 1, wherein said throats have straight sections, the mean width of which is less than ⅓ the depth of the said throats.

10. A heat exchange arrangement according to claim 9, wherein the depth of the throats b and the mean width of the sides a which separate them are defined by the relationship:

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

where b and a are expressed in centimeters, c designates the thermal conductivity of the constituent material of the heated wall expressed in watts/centimeter×° C., and m is a numeric factor of the order of 1 comprised between the limits 0.7 and 1.8.

11. A heat exchange arrangement according to claim 1, wherein the distance separating the guide wall from the overall surface of the exchanger wall is equal to:

$$k\frac{L}{100}$$

where L is the mean length of the main flow path and k a numeric factor of the order of 1 comprised between the limits 3 and 0.3.

12. A heat exchange device for heat transfer from a heated wall to a forced circulation heat removing liquid by local evaporation accompanied by recondensation in the mass of the circulating liquid, wherein the heat transfer surface of the heat exchanger wall is formed with throats, and exposed to the forced circulating liquid, confined by a guide wall, characterized in that:

the depth (b) of the throats (9) is greater than the distance (d) separating their two edges and the device comprises means forcibly circulating a major part of the liquid over said surface and in a direction of flow which, with respect to the longitudinal direction (12) of the throats (9) forms an angle α between 45° and 90°.

13. Heat exchange device according to claim 12 wherein the angle α is in the range of from 80° to 90°, whereby the direction of said flow will be essentially transverse to said throats.

14. In a method of removing heat from a heat-exchange surface having a plurality of substantially parallel deeper than wide grooves formed therein, including the step of circulating a heat exchange fluid in contact with said surface, the improvement comprising

the step of contacting the interior of said grooves with heat exchange fluid to remove heat from the walls defining said grooves by local evaporation, while forcibly circulating liquid heat-exchange fluid over the heat-exchange surface to re-condense said evaporating heat-exchange fluids into the mass of said circulating liquid fluid.

15. Method according to claim 14 wherein the steps of contacting the interior of said grooves and recondensing said fluid includes a step of forcibly circulating said liquid fluid over said surface in a direction essentially transverse with respect to said grooves and forming an angle with the major extent of said grooves of from between 45° to 90°.

16. Method according to claim 15 wherein said step of circulating said liquid fluid includes the step of circulating said liquid fluid at an angle with respect to the grooves of from between 80° to 90°.

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