

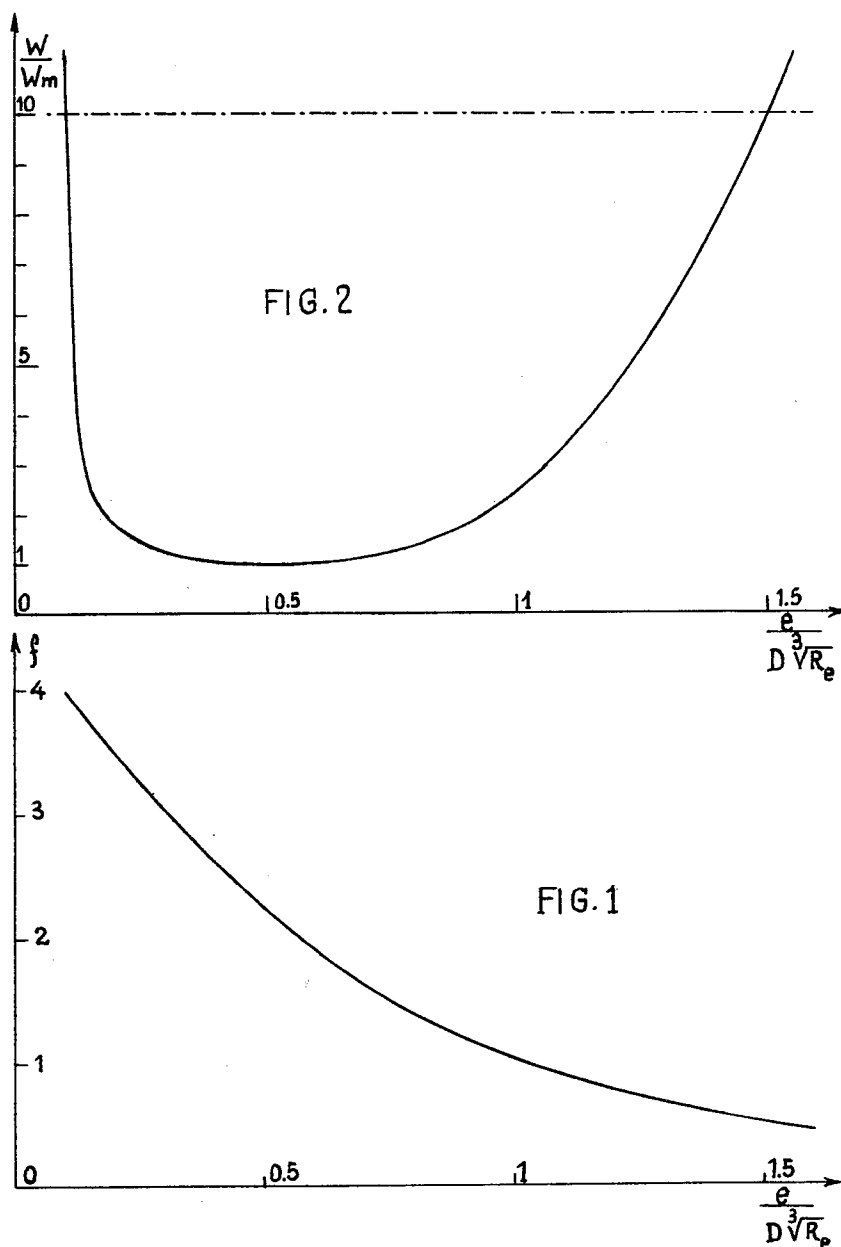
Nov. 5, 1963

A. FORTIER
HEAT EXCHANGER

3,109,485

Filed Feb. 20, 1959

2 Sheets-Sheet 1



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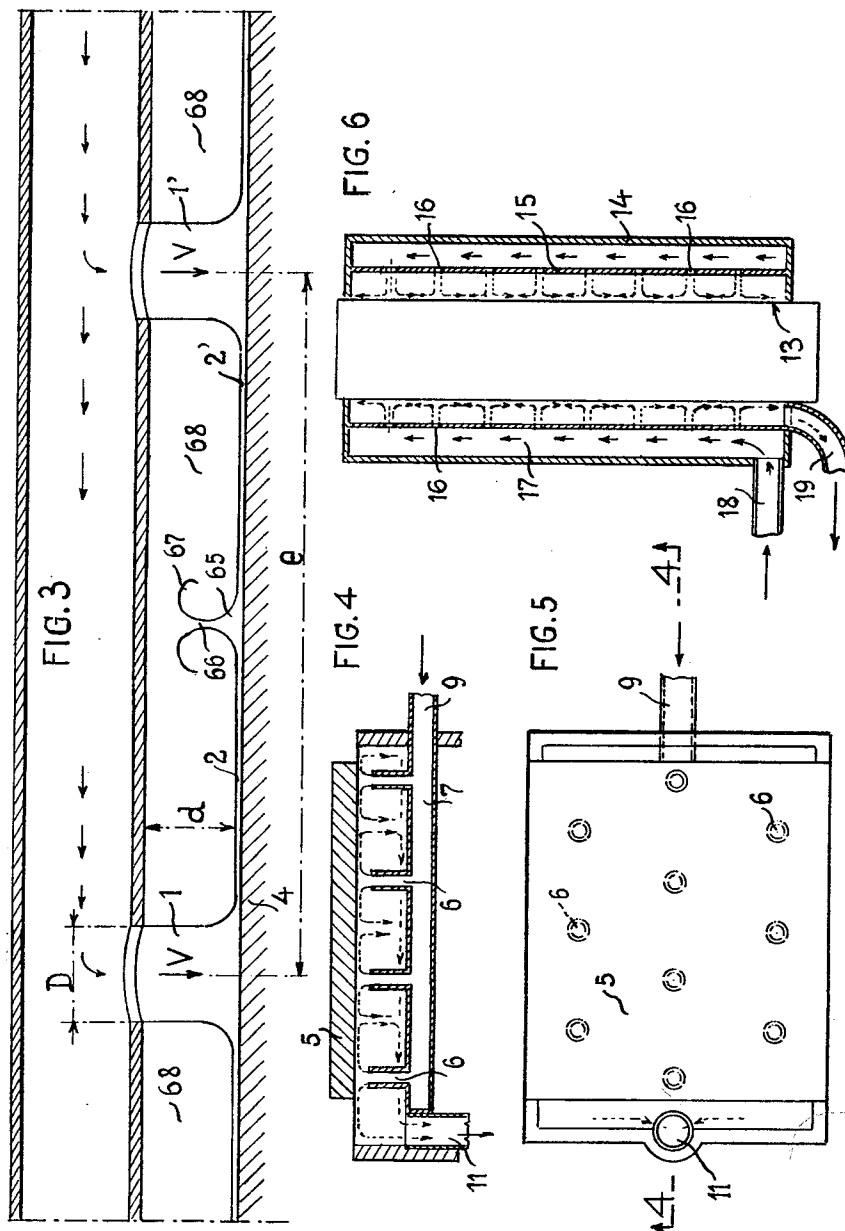
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HEAT EXCHANGER

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This invention relates to a heat exchanger of the type in which a fluid is continuously flowing in heat-exchange relationship with a solid surface.

Generally speaking, a heat-exchanger is characterized by a coefficient which will be called hereunder "heat exchange coefficient" and which is equal to the ratio of the amount of heat exchanged per unit of area of the solid surface and per unit of time to the difference between the maximum temperature of the surface and the fluid temperature as it enters the exchanger.

In conventional heat exchangers, the fluid flow takes place through a tubular channel at least one wall of which constitutes the heat exchange surface. It is then obvious that the conditions of flow and hence the heat exchange coefficient depend on the mechanical structure of the said tubular channel. Under these conditions, for a given fluid, it is impossible to obtain a heat exchange coefficient exceeding a certain limit which is, in particular, imposed by the minimum value of the dimensions of the tubular channel under which it would be impossible to design it, not only for obvious machining difficulties, but primarily due to unavoidable thermic deformations.

It is easy to realize, for example, that a passage narrower than 1 mm. between two metal surfaces, of say, 1 m.² area is liable to be obstructed as soon as the temperature reaches a few hundredths of degrees.

As a matter of fact, for all practical purposes, it has never been possible to overreach a heat exchange coefficient of about 10,000 kcal./m.²/h./° with liquid water.

For certain particular purposes such as quenching, wherein the only problem is to obtain quick cooling, it is usual to put the surface to be cooled in contact with a huge mass of water, either by suddenly immersing the part to be quenched into a water bath or by spraying said part with one or several jets of water without considering the pressure of projection nor the amount of water wasted.

It is obvious that such a method, if applied to continuous heat exchange, in particular with a very hot surface, would lead to prohibitive consumption of power as well as enormous installations.

In some quenching methods, it has been proposed to project water on a surface to be cooled through a multitude of closely spaced small orifices with a view to uniformly distributing the cooling effect on the whole area of the part to be quenched.

However, nobody seems to have attempted to apply these methods in the heat exchanger art proper.

This is due to the fact that the efficiency of a heat-exchange effected in such conditions depends upon a great number of parameters which have never been selected in such a manner as to meet simultaneously the hereunder defined conditions which happen to correspond to very complex thermo-dynamical relationships.

Another reason for which projection of water in small jets on a heat exchange surface has never been used in heat exchangers is that, surprisingly enough, the modification of certain of said parameters, whenever they are not chosen within the mathematically defined limits taught by the applicant, has results absolutely in contradiction with what is expected. Thus, for example, while it would seem logical to think that by increasing the diameter of

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the orifices, i.e. the size of the jets, the heat exchange surface would be more energetically cooled, surprisingly it is the contrary which occurs.

Finally, not only the heat exchange coefficient rapidly decreases towards the limits of the parameter range according to the invention, but as soon as the said limits are overreached the efficiency becomes lower than that of a conventional heat exchanger, all other things being equal. As, on the other hand, the said range is somewhat narrow, it is not surprising that nobody heretofore happened to find out the principles on which the invention is based.

This is why when very high heat exchange coefficients are to be obtained, there is resorted to special fluids such as metals which are liquid at the temperature of the exchange, e.g. sodium, potassium, or sodium and potassium alloys. However, the use of such metals offers serious difficulties and is extremely dangerous.

The main object of the invention is to provide a multiple jet heat exchanger in which the heat exchange coefficient can be raised to very high values, for example of the same order as those which would be obtained with the above cited metals, but in which usual fluids, such as water, are used, the heat exchange nevertheless requiring but a negligible consumption of power.

The following symbols will be used hereunder to show the difficulty of the thermo-dynamical problems which have been solved by the applicant:

h =heat exchange coefficient to be obtained

V =fluid velocity in a jet

D =diameter of an orifice

e =distance between the axes of two adjacent orifices

μ =dynamic viscosity coefficient of the fluid

λ =coefficient of thermal conductivity of the fluid

ρ =mass of one volume unit of the fluid

C_p =specific heat of the fluid at a constant pressure

R_e =Reynold's number

$$R_e = \frac{\rho V D}{2\mu}$$

P_r =Prandtl's number

$$P_r = \frac{\mu C_p}{\lambda}$$

L =length of the longest fluid outflow path (for example if the heat exchange surface has the shape of a cylinder, L is the length of that cylinder).

d =distance between the apertured wall and the heat exchange surface.

An object of the invention is to provide a multiple jet heat exchanger in which:

$$R_e < 60,000 \quad (1)$$

to obtain laminar flow within the boundary layer along the heat exchange surface.

A further object of the invention is to provide a multiple jet heat exchanger in which:

$$\frac{d}{D} < 30 \quad (2)$$

to prevent the fluid jets from spraying before impact. Still another object of the invention is to provide a multiple jet heat exchanger in which:

$$\frac{d}{L} > \frac{\left(\frac{D}{e}\right)^2}{1 - 1.15 \frac{D}{e}} \quad (3)$$

so that the fluid can flow out of the heat exchanger with a sufficiently low velocity so as to avoid any flow perturbation in the jets.

It is another object of the invention to provide a multiple jet heat exchanger in which

$$0.1\sqrt[3]{Re} < \frac{e}{D} < 1.5\sqrt[3]{Re} \quad (4)$$

When and only when this condition is met, the heat exchanger according to the invention requires but a lower consumption of mechanical power than a conventional heat exchanger, all other things being equal (and in particular for a same heat exchange coefficient and a same difference of temperatures).

The variations of the mechanical power consumed in a multiple jet heat exchanger according to the invention as a function of the parameter

$$\frac{e}{D\sqrt[3]{Re}}$$

are plotted in FIG. 2, in terms of the minimum consumption. Now, a conventional heat exchanger wherein the fluid flows at any point in a direction parallel with the heat exchange surface between the latter and another surface parallel therewith at a distance d requires a mechanical power consumption about ten times greater than the minimum power value on curve (2), all other things being equal.

Thus it will be easily understood that the heat exchanger according to the invention is advantageous only if the parameter

$$\frac{e}{D\sqrt[3]{Re}}$$

is comprised within an interval limited by the abscissae of the points having the ordinate 10 on curve (2) which defines the limits of condition (4).

When the four conditions (1), (2), (3), (4) are simultaneously met a given coefficient h may be obtained according to the following additional relation:

$$\frac{hD}{\lambda} = \sqrt[3]{Pr\sqrt[3]{Re}} f\left(\frac{e}{D\sqrt[3]{Re}}\right) g\left(\frac{d}{D}\right) \quad (5)$$

The function

$$f\left(\frac{e}{D\sqrt[3]{Re}}\right)$$

is plotted in FIG. 1 and the function

$$g\left(\frac{d}{D}\right)$$

may be considered as approximately equal to unity if

$$\frac{d}{D}$$

complies with condition (2).

It is clear that for a given use of the heat exchanger, the five conditions hereabove recited define the characteristics of the heat exchanger i.e. the diameter D of the orifices, the distance e between the axes of the orifices, the distance d between the two surfaces and the velocity V of the fluid in the jets.

Conditions (1) and (5) define a maximum diameter and conditions (2), (3), (5) a minimum diameter for each value of the parameter

$$\frac{e}{D\sqrt[3]{Re}}$$

that meets condition (4). On the other hand, in the considered application, a value of that parameter corresponding to a minimum consumption of power is obviously to be chosen, which finally determines the values to be given to D , e , d and V . It may be seen on curve (2) that the best results are obtained for

$$\frac{e}{D\sqrt[3]{Re}} = \text{about } 0.5$$

For example, if 8,600,000 kilocalories per hour are to be exchanged between water and a surface having the shape of a 1 m. x 1 m. square with an exchanger according to the invention, with outflow of the water across one side of the square surface with a difference of 80° C. between the maximum temperature of the heat exchange surface and the temperature of the water as it enters the heat exchanger, the above-mentioned conditions impose the following dimensions:

$$D = 0.35 \text{ mm.}$$

$$e = 3.5 \text{ mm.}$$

$$d = 10 \text{ mm.}$$

$$V = 24 \text{ m./sec.}$$

The power consumption is then of about 100 kw. instead of 1,000 kw. with a conventional heat exchanger all other things being equal.

In other words, it may be said that while it has been known to exchange heat between a surface and a fluid by projecting on said surface jets of said fluid, the invention makes it possible to obtain, with a same consumption of power, a heat exchange coefficient far higher than with any of the conventional methods by a suitable computation of the dimensions and spacing of the orifices, the length of the jets and the velocity of the fluid. Alternatively, for a same heat exchange coefficient, it is possible to reduce considerably the power consumption with respect to that required in the conventional heat exchangers.

The jets may be projected upon the heat exchange surface through a gas such as air as well as through the same liquid provided that the said jets have not to flow through a prohibitively thick mass of fluid.

In one embodiment, the heat exchange surface lies horizontal and jets of liquid are projected upon the lower face of the surface from where the liquid easily drops down under the action of gravity.

In another embodiment, the heat exchange surface is a cylindrical member upon which the jets are continuously projected through nozzles constituted by orifices drilled in the inner wall of an annular cylindrical member co-axial to the above-mentioned heat exchange surface and fed with pressure fluid, the exhaust of the fluid taking place through at least one end of the annular space comprised between the heat exchange surface and the apertured wall.

Other objects and advantages of the invention will be apparent from the following detailed description, together with the accompanying drawings submitted for purpose of illustration only and not intended to limit the scope of the invention, reference being had for that purpose to the sub-joined claims.

In the drawings:

FIG. 1 is the already mentioned curve of the function

$$f\left(\frac{e}{D\sqrt[3]{Re}}\right)$$

FIG. 2 shows the variations of the mechanical power actually consumed vs. its minimum as a function of

$$\frac{e}{D\sqrt[3]{Re}}$$

FIG. 3 is a much enlarged view showing the shape of a jet of liquid projected upon the heat exchange surface of a heat exchanger according to the invention.

FIG. 4 is a sectional view of an embodiment of the heat exchanger according to the invention.

FIG. 5 is a plan view corresponding to FIG. 4, and FIG. 6 is an axial sectional view of an alternative embodiment.

As shown at 1 and 1' in a heat exchanger according to the invention, jets of a usual fluid such as water, having a very small diameter D impinge from a very short distance d on a solid flat heat exchange surface 4 on which they are flattened into extremely thin sheets such as 2, 2' merg-

ing into each other as shown at 65 to be then reflected normally to surface 4 as shown at 66, whereupon the fluid forms vortices 67 which flow out through the gaps such as 68 between the jets. Thus, for example, with a jet of water about 5 mm. long projected through an orifice of 0.5 mm. diameter in a thin wall, the thickness of the liquid sheet is approximately 2/100 mm. for a velocity of the water in the jet of about 14 m./s. while the diameter of the area covered by the thin sheet is about 1 cm. The water velocity remains, within the whole free area of the sheet substantially equal to its value in the jet, i.e. 14 m./s. in this example. If the solid surface on which the jets are projected is an incurved one, the liquid sheet fits tightly the said surface and the above-described phenomena are not inherently modified. This is due to the negative pressure on the solid surface.

Now, assuming that the above-mentioned solid surface constitutes the heat exchange surface of a heat-exchanger, according to the invention, the liquid projected on said surface being intended to derive calories from the said surface or, conversely to yield heat thereto, due to the extreme thinness of the liquid sheet and the high velocity of the fluid at the surface of said sheet, the heat exchanges between the solid surface and the fluid are very intense and the heat exchange coefficient h calculated from relation (5) (cf. preamble) the fluid being liquid water is:

$$h=73,000 \text{ kcal./m.}^2/\text{h}^\circ\text{C.}$$

With a jet of water projected through an orifice of 0.3 mm. diameter and for the same velocity of 14 m./s. the heat exchange coefficient h reaches 100,000 kcal./m.²/h[°]C. Under these last conditions, a difference of temperature of 10° between the hot surface and the jet water thus suffices to evacuate a thermal power of 100 W./cm.². Now, to impart to the water a velocity of 14 m./s., it suffices to build upstream the orifice a pressure of 2 kg./cm.²; since the rate of flow in the jet is about 1 cm.³/s. and since three jets are required to cover 1 cm.², the power consumption is 0.3 W. per cm.² which is absolutely negligible as compared with the 100 W. which are exchanged.

If the surface to be cooled is disposed too far from the orifice, the jet spreads out in droplets and the heat exchange coefficient decreases; the distance between the orifice and the heat exchange surface has been defined precisely in the preamble. For a jet of 0.3 mm. diameter, the said distance should not overreach 4-5 mm. and may be as small as a few mm.

In the embodiment shown in FIGS. 4 and 5, the surface 5 to be cooled is a flat surface of 10 cm.² area, through which the thermal flux reaches 100 W./cm.². A plurality of small tubes 6 extending vertically under the surface 5 are fixed by their basis on a chamber 7 from which they are fed with cooling water. The upper end of each tube is provided with an orifice of 0.5 mm. diameter. These tubes are disposed at the intersecting points of the lines of a pattern formed by a plurality of adjacent equilateral triangles, the density of the heat flux of the surface to be cooled being assumed to be substantially uniform. The chamber 7 is provided with a pressure water inlet 9, and the space comprised between the surface 5 to be cooled and the chamber 7, i.e. the space in which the tubes 6 extend communicates with an outlet 11.

For an orifice having a given diameter and for a given velocity of the water in the jet, the heat exchange coefficient has a well defined value. Thus, if the thermal flux is increased the temperature of the liquid in contact with the hot surface increases and the diameter of the area covered by the liquid sheet also increases. This phenomenon results from the reduction of the surface tension of the liquid as the temperature increases. When the temperature of the liquid reaches the boiling point, the liquid in the liquid sheet is partly evaporated, but the vapour generated is carried away by the liquid flowing out at a high velocity which avoids any calefaction phenomena. It is

thus possible to vaporize a considerable proportion of the liquid without overreaching noticeably the boiling point which permits obtaining extremely high densities of thermal flux.

When the density of the thermal flux is not uniform on the whole area of the surface to be cooled, the number of the tubes per unit area is made proportional to the value of the said density.

In the embodiment of FIG. 6 the surface to be cooled is a cylindrical surface 13 extending for example vertically. This surface is surrounded by a coaxial annular cylindrical member 14 in which extends a coaxial cylindrical wall 15 provided with a multitude of apertures 16. The water feeding chamber 17 is constituted by the annular cylindrical space comprised within the walls 14 and 15; it is provided with an inlet 18. The annular cylindrical space comprised between the surface 13 to be cooled and the apertured wall is provided with an outlet 19 through which the water is exhausted. The pressure water jets projected out of the chamber 17 through the apertures 16 impinge radially upon the surface 13 to be cooled on which they are flattened in thin cylindrical sheets as explained above. The jets have to flow through the liquid layer comprised between the walls 13 and 15; the thickness of the said layer being of course reasonable, so that the linear velocity of the jets be not reduced too much.

While the invention has been described with particular reference to preferred embodiments it is not intended to limit the scope of the invention to the embodiments illustrated, nor otherwise than the terms of the subjoined claims.

What is claimed is:

1. The method of exchanging heat between a heat-conducting wall and a fluid, the temperature of which is different from that of said wall, which consists in conducting said fluid to one face of said wall in the form of a plurality of jets extending substantially perpendicular to said face and spaced with respect to each other by a distance e , imparting to said fluid by said jets a velocity V the Reynolds' number R_e of which is smaller than 60,000, given the trajectories of said jets a length d and giving each jet, at its origin, a diameter D , removing said fluid along a discharge path having a length L , and relating said distance e , said velocity V , said length d , said diameter D , and said length L so that

$$\frac{d}{D} < 30$$

$$\frac{d}{L} > \frac{\left(\frac{D}{e}\right)^2}{1 - 1.15 \frac{D}{e}}$$

$$0.1\sqrt{R_e} < \frac{e}{D} < 1.5\sqrt{R_e}$$

2. The method according to claim 1, wherein the ratio

$$\frac{e}{D\sqrt{R_e}}$$

is comprised between 0.4 and 0.6.

3. The method according to claim 2, wherein said ratio is equal to 0.5.

4. The method of removing heat from a heat-conducting wall, which consists in forming a plurality of jets of cooling liquid having each a diameter D , imparting to said jets a velocity V having its Reynolds' number smaller than 60,000, directing said jets to one face of said wall from a distance d , removing said liquid along a discharge path having a length L , distributing said jets uniformly in spaced relation to each other by a distance e , relating said

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diameter D, said distance d, said length L, and said spacing e so that

$$\frac{d}{D} < 30$$

$$\frac{d}{L} > \frac{\left(\frac{D}{e}\right)^2}{1 - 1.15 \frac{D}{e}}$$

and

$$0.1\sqrt[3]{R_e} < \frac{e}{D} < 1.5\sqrt[3]{R_e}$$

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and maintaining the ratio

$$\frac{e}{D\sqrt[3]{R_e}}$$

5 about 0.5.

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