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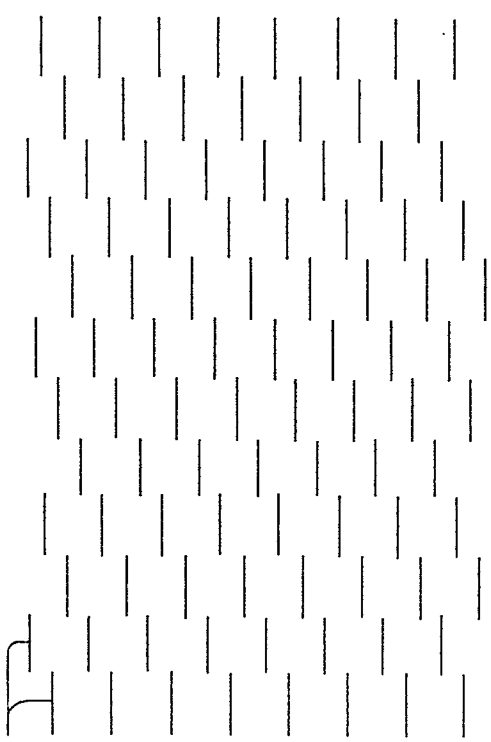
(54) **Heat exchanger.**

(57) A heat exchanger for transferring heat from a body through heat transfer elements to streamlines of a laminar flowing coolant fluid has fin elements arranged in a row and column pattern, with elements staggered transversely to the flow so that no fin is aligned with its nearest or next nearest neighbours in either the row or column direction. During fluid flow across the heat exchanger, each streamline contacts one heat transfer element and is directly heated exactly once before any streamline contacts more than one element and is directly heated twice. Thus, every streamline of the fluid is used for heat transfer. After an upstream heat transfer element heats its wake-core, enough time elapses before these streamlines flow close to a downstream element to allow heat to diffuse sideways. Thus, no downstream heat transfer element is in the immediate thermal wake-core of any upstream heat transfer element.

FIG. 1



101

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HEAT EXCHANGER

Technical Field of the Invention

This invention relates to heat exchangers for transferring heat between the exchanger and a fluid.

Definitions

The term "heat transfer coefficient" shall hereinafter refer to the amount of thermal power transferred from a hot solid substrate to a fluid divided by the temperature difference between the hot solid substrate and the fluid.

The "volumetric heat transfer coefficient" will refer to the heat transfer coefficient per unit volume of an array of heat transfer elements. "Areal heat transfer coefficient" will refer to the heat transfer coefficient per unit area of heat transfer element. "Basal heat transfer coefficient" will refer to the thermal heat transfer coefficient per unit area of substrate on which the heat transfer elements lie.

The "hydraulic diameter" of a passage used for forced-convection heat transfer is two times the groove width between heat transfer elements. The thermal conduction across the hydraulic diameter defines a scale, or measure, or the heat transfer coefficient per unit area.

Hereinafter, the term "Nusselt number" will be used to refer to the normalised heat transfer characteristics of a system. The Nusselt number measures the areal heat transfer coefficient along the scale defined by the hydraulic diameter.

The term "heat exchanger" will refer to a device that serves to transfer heat from solid substrate to fluids as well as between two or more fluids. A "heat transfer element" will be used to describe a solid object, such as a tube, a fin or the like, which can be used to transfer heat between a substrate and a fluid or between an interior fluid and an exterior fluid. Heat transfer may take place between a cool element and a hot exterior fluid or between a hot element and a cool exterior fluid.

A "row" of heat transfer elements will be used to describe a plurality of elements spaced transversely across the direction of fluid flow. A "column" of heat transfer elements will be used to describe a plurality of fins spaced along the direction of fluid flow. The term "downstream" will be used to describe the direction along the fluid flow away from the origin of flow.

The term "thermal boundary layer" will hereinafter refer to the relatively warm fluid layer that develops adjacent to a heat transfer element as fluid flows past the element. The thermal boundary layer is conventionally defined by an isotherm very close to the fluid temperature, usually the isotherm having a temperature approximately equal to that of the fluid, ie 99% of the temperature difference from the heat transfer element to the fluid. Hereinafter, the thermal boundary layer will refer to the layer bounded by the isotherm approximately one-half the temperature difference between the heat transfer element and the fluid. Defining the thermal boundary layer according to this 50% isotherm gives an average heat transfer coefficient. The streamlines of that thermal boundary layer traced downstream will be referred to as a "wakecore" of a heat transfer element.

Streamlines which impinge on the thermal boundary layer of an element will be referred to as "flowing close" to the heat transfer element.

Background Art

Forced-convection cooling has long been used to transfer heat between solids and fluids. Many heat transfer designs have employed fluid flowing past multiple, closely-spaced heat transfer elements to transfer a large power density from a high-temperature substrate to a low-temperature fluid. Many prior designs have minimised the width of the grooves, or distances, between heat transfer elements as well as the distance between the centre lines of the elements, or transverse pitch of the groove.

The volumetric heat transfer coefficient of prior designs is proportional to the Nusselt number associated with the designs divided by the groove width, and divided by the transverse pitch. The minimum groove width is dictated by practical problems of fabrication and the possibility of fouling the heat exchanger. Transverse pitch is limited by the size of the heat transfer elements used and the groove width. Prior designs achieve a limited Nusselt number, and therefore a low volumetric heat transfer coefficient. "Compact Heat Exchangers", Second Edition, W M Kays and A L London - (McGraw-Hill, 1964) sets forth many conventional prior designs.

One conventional design employs a set of parallel long fins. At any fluid flow speed where fluid flow is laminar, a characteristic, relatively small Nusselt number can be calculated. A low Nusselt number is characteristic of a low volumetric heat transfer coefficient. Thus, this design is not an efficient method to transfer large amounts of thermal power.

Another conventional design uses a rectangular array of heat transfer elements in orthogonal rows in which each element is directly aligned with its downstream neighbour. This design also results in a relatively low Nusselt number.

Yet another conventional design employs an array of heat transfer elements in which alternate rows of elements are offset by one-half the transverse pitch between elements. The Nusselt number of this array is also relatively small.

US-A-3,421,578 (Marton) describes a heat exchanger in which fins are arranged at substantially equal distance from each other along diagonal lines. US-A-3,421,578 proposes the use of a conventional offset pattern to improve heat transfer.

Thus, in laminar-flow forced convection heat transfer devices, very dense configurations could achieve large thermal power transfer per temperature difference per unit area. However, they require very large pressure drops and high pump power. Furthermore, dense designs are difficult to fabricate and are easily debilitated by fouling.

Another solution proposed by the prior art is the creation of turbulent fluid flow in order to mix the fluid and to promote contact between all heat transfer elements and all parts of the fluid. This solution, however, requires a disproportionately large pressure gradient drop and a disproportionately large expenditure of pump power. Further, since turbulence is inefficient, for a given limited pressure gradient, designs using it achieve a relatively low Nusselt number.

The poor performance of prior art practices is theoretically due to the geometric arrangement of the heat transfer elements. Efficient heat transfer to a fluid requires a substantial temperature difference between the higher-temperature object and the fluid.

Laminar convection through closely spaced long fins leads to "fully developed laminar convection". Upstream parts of each fin have adjacent streamlines, which flow along downstream parts of the fin. If the fin has constant power density, the downstream part of the fin becomes very hot. Other streamlines flow near the centre of the gap between the fins, but do no impinge closely on the thermal boundary layer of the fins. These streamlines, therefore, cannot directly remove heat from the fins. They can only remove heat indirectly, from the streamlines nearer the fins.

A rectangular pattern of short fins causes "fully developed periodic laminar convection", and is similarly inefficient. All fins in a column share a wake-core. An upstream fin heats a wake-core, which immediately impinges on a downstream fin. By contrast, other streamlines, such as those flowing between the fins, never impinge on any fin. Therefore, they remove thermal power only indirectly, after it diffuses outside a wake-core.

A similar problem occurs in a conventional staggered array. As a fluid flows past an element, a thermal boundary layer develops. The thickness of the thermal boundary layer determines the local heat transfer coefficient. In arrays of heat transfer elements such as those contemplated for use in this invention, the thermal boundary layer grows in thickness along the downstream length of the element. Some fluid streamlines passing by the element impinge on the thermal boundary layer of the element.

When heat transfer elements are arranged in a rectangular pattern the streamlines flowing through the thermal boundary layer of a first element pick up heat from that element. The streamlines then quickly impinge on the thermal boundary layer of a second heat transfer element which is directly downstream of the first element. The temperature difference between the second element and the streamline is not as great as the temperature difference between the first element and the streamline. Thus, the heat transfer between the second element and the streamline is less efficient than between the first element and the streamline. The overall efficiency in a conventional array therefore decreases as the stream continues along the flow path. Meanwhile, other streamlines may never impinge on the thermal boundary layers of any heat transfer element because they flow through the open areas in the array. These streamlines do not effect heat transfer, and therefore some of the heat transfer capacity of the fluid is lost. In a conventionally offset array, there is a similar inefficiency because the elements in alternative rows are directly aligned. The wake-core from one element soon impinges on another element, two rows downstream.

Disclosure of the Invention

Thus, the prior art has not solved the problem of promoting efficient heat transfer with a convective heat transfer apparatus using laminar fluid flow for a given density of heat transfer elements in a small area, and for a given limited pressure gradient.

Accordingly, the present invention provides a heat exchanger for transferring heat between the exchanger and a fluid, the heat exchanger comprising at least one surface provided with a plurality of heat exchanging fins positioned on a grid defined by intersecting row and column lines such that each fin lies on a row line and in longitudinal alignment with a column line, characterised in that no fin is aligned, in either the row or column directions, with its nearest or next nearest neighbours.

This overcomes the problems presented in the prior art by increasing the Nusselt number, or thermal power transfer coefficient, and significantly improving heat transfer for a given pressure gradient.

From an alternative viewpoint, the invention provides an apparatus for convective heat transfer to a fluid stream comprising a substrate and heat transfer elements, said heat transfer elements being arranged in a configuration on said substrate such that when said fluid stream having streamlines flows across the substrate, substantially every streamline flows close to some heat transfer element before many streamlines flow close to more than one heat transfer element.

Preferably said heat transfer elements are arranged in a plurality of rows transverse to the fluid flow direction, the heat transfer elements in said rows being separated along a direction transverse to the fluid flow by irregular distances.

Preferably again, in said columns, heat transfer elements located further away from the fluid source are downstream of said heat transfer elements located closer to said fluid source, said heat transfer elements being offset from each other in a direction transverse to the fluid flow such that a streamline flowing close to the thermal wake-core of a first heat transfer element does not flow close to the thermal wake-core of a downstream heat transfer element until the heat received from the first heat transfer element has been substantially diffused away from said wake-core. Said transverse separation may be random.

Preferably said heat transfer elements are fins and have a base, a tip, a leading edge and a trailing edge wherein said fins have a wedged thickness decreasing from base to tip, a tapered chord decreasing from base to tip, a curved leading edge and a straight trailing edge.

In general, each heat transfer element is preferably surrounded by a thermal boundary layer having a thickness N_Z times thinner than the transverse gap between elements, each row successively downstream being transversely offset such that each streamline of the fluid passes through the thermal boundary layer of only one heat transfer element per N_Z rows downstream of each other.

More specifically each successive row of heat transfer elements is displaced transversely by a factor (W times S) wherein W is the width of the transverse gap between heat transfer elements, and S is a number such that positive integers A and B having magnitudes less than N_Z satisfy the inequality:

$$| A - B \times S | \geq 1/N_Z$$

More precisely still, S = P/M, wherein $M = 1/N_Z$ and P is a number relatively prime to M, wherein M is an integer between $N_Z + 1$ and $N_Z - 1$. S may satisfy the inequalities given there for most positive integers B less than N_Z and may equal (3 -5)/2 = 0.381,966.

Preferably the numbers P and M are alternate Fibonacci numbers.

In general, it is preferable that said heat transfer elements are further arranged such that most streamlines flow through the thermal boundary layer of at least one heat transfer element.

From yet another viewpoint, the invention comprises an apparatus for convective heat transfer comprising a substrate having a plurality of heat transfer elements, said heat transfer elements being arranged on said substrate such that any two elements located a small distance apart along the flow direction are misaligned transverse to the flow direction by a distance greater than the thickness of the thermal boundary layer formed by fluid flowing past said elements.

In general, it is not necessary for fluid flow to be parallel. It may instead be radial.

Thus the heat exchange apparatus of this invention has rows of heat exchange elements organised in a pattern on a plate or substrate in such a way that the thermal boundary layers of the fluid are thin in comparison with the groove width, resulting in the efficient transfer of thermal power. Accordingly, the rows of heat transfer elements are transversely offset from each other such that the fluid streamlines impinging on any heat transfer element will not impinge on any other element until they have given up a substantial amount of transferred heat.

When a fluid stream flows across the substrate, each streamline contacts one heat transfer element and is directly heated exactly once before any streamline contacts more than one element and is directly heated twice. Thus, every streamline of the fluid is directly used for heat transfer. After an upstream heat transfer element heats its wake-core, enough time elapses to allow heat to diffuse sideways before these streamlines flow close to a downstream element. Thus, no downstream heat transfer element is in the immediate thermal wake-core of any upstream heat transfer element. Because there is a heat transfer element close to every streamline, such an array is referred to as "omnipresent staggering".

In addition, because the fluid does not contact elements in rapid succession the temperature difference between the fluid and the heat transfer element it contacts is sufficiently great to provide good heat transfer.

This invention, therefore, uses aperiodic laminar convection or long period developed convection to avoid the inefficiency of conventionally staggered heat transfer elements. Thus, although the array is dense, each element functions as efficiently as an isolated heat transfer element, thus giving the best heat transfer for a given pressure gradient.

Brief Description of the Drawings

Figure 1 is a plan view of an arrangement of fins on a substrate forming part of a heat exchanger according to the present invention;

Figure 2 is a plan view of an alternative arrangement of fins in a heat exchanger according to the present invention on an enlarged scale and with one dimension exaggerated for clarity;

Figure 3 is a diagram of isotherms representing the temperatures of streamlines impinging on a set of fins in a heat exchanger according to the present invention; and

Figure 4 is a plan view of a further arrangement of fins in a heat exchanger according to the present invention.

Detailed Description of the Invention

Before discussing the embodiments of Figures 1 to 4 in detail, it is desirable to derive a set of mathematical design conditions for heat exchangers according to the invention.

A heat exchange apparatus according to this invention may be made having heat transfer elements in many configurations as long as the following requirements are met: (a) substantially every streamline of the fluid flows close to some heat transfer element in the array and (b) substantially every streamline flows close to one heat transfer element before many streamlines flow close to more than one heat transfer element.

Thus, many configurations of heat transfer elements can be used for heat transfer according to this invention as long as they are not locally regular rectangular nor conventionally staggered patterns.

In one preferred embodiment, the heat transfer elements can be randomly placed in an irregular manner. In such an embodiment, fluid streamlines would randomly impinge on the elements. In this configuration, very few streamlines would avoid impinging the thermal boundary layer of a heat transfer element. Also, very few wake-cores flow immediately from one element to another nearby element.

In another more preferred embodiment, heat transfer elements can be transversely offset such that each stream passes through the thermal boundary layer of only one heat transfer element per a certain number of successive rows, represented by the term N_Z. $N_Z$is the ratio between the width, W, of the transverse gap between the heat transfer elements in each row and the thickness of the wake-core immediately beyond the downstream end of each element. Preferably, N_Z is larger than 3. This will allow the streamline to flow past at least three rows of heat transfer elements before impinging directly on another element. Further, this configuration creates areas in which other streamlines can impinge on other heat transfer elements. This configuration affords considerable time for thermal power to diffuse sideways before the wake-core impinges on another heat transfer element.

In a more preferred embodiment, each successive row is offset by a distance of S times W, where S is a suitable constant to be described below. For example, an element (A, B) can be found by starting at any element, and counting A elements transversely and B rows downstream to reach another element. The starting element is the "zeroth" element and the element reached is the - (A, B) element. The centre of the wake-core trailing the zeroth element will avoid the element (A, B) by a transverse distance Y defined as follows:

$$Y = AW - BSW$$

In order to achieve a configuration of heat transfer elements such that the requirements of this invention are met, the wake-core of the zeroth element should pass outside of the thermal boundary layer of element (A, B).

The thickness of the thermal boudary layer is W/N_Z. Therefore, the desired situation can be represented mathematically in the following way:

$$|Y| \geq W/N_Z$$

$$\text{ie } |AW - BSW| \geq W/N_Z$$

In a preferred embodiment of this invention, the value S must satisfy these inequalities for every (A, B) element in N_Z rows downstream of the zeroth element. Algebraically, this can be represented by the following inequality:

$$|A - BS| \geq 1/N_Z \quad (I)$$

for all integers B between O and N_Z and all positive integers A.

Values of S can be found which meet these requirements. M is an integer between N_Z-l and N_Z+1. P is another integer, relatively prime to M. S can then be determined by the following equation:

$$S = \frac{P}{M}$$

This pattern repeats itself every M rows. For example, if M is 13, then S can be 5/13 or 0.384,615... The resulting pattern is illustrated partly in Figure 2.

An S value which solves inequality (I) for one N_Z value may not be suitable for faster flow and larger N_Z. For example, if N_Z exceeds 13, 5/13 is not the best value for S, although it can enhance the normalised heat transfer to an intermediate degree independent of flow speed. The mathematical art of "Diophantine approximations" leads to another class of solutions. There, a classical problem is to find numbers which are especially hard to approximate by rational fractions, such as, Phi, the "Golden Ratio" of classical mathematics. (See, for example, "Elementary Number Theory", J Roberts, 1977, Chapter XIII, MIT Press, pgs 116-119). When S is the compliment of Phi:

S = 1 -Phi = (3 -Root 5) / 2 = 0.381,967...

the following relationships are satisfied:

$| A - BS | > 1/B$ (II)

for all positive integers (A,B) where B IS NOT a Fibonacci number, and

$$\left| A - BS \right| > 1/(B - \sqrt{5})$$

for all positive integers (A,B) where B IS a Fibonacci number.

A Fibonaci number sequence is 1, 1, 2, 3, 5, 8, 13, 21 ... which is widely known in the field of mathematics. Inequality (II) implies that Inequality (I) is solved for most positive integers B less than N_Z.

In this case, the wake-core from an upstream element will miss elements in almost all of N_Z downstream rows. For a few downstream rows, given by the Fibonacci sequence, the wake-core will slightly impinge on a downstream fin. If N_Z is 34, the wake-core of an original fin will miss every fin in the next 34 rows by more than 100% of (W/N_Z) except in rows 21 and 34, where the miss distances are 72% and 44% of W/N_Z.

Figure 4 shows an arrangement of fins in which S = 0.382. Streamlines downstream on the fins illustrate that no fin is aligned with any other and that the wake cores of each fin will thus miss every other fin.

Some theorems in Diophantine approximation theory prove that for this solution, and for certain related solutions, such as S = Phi, the offset Y will be as large as possible. One skilled in solving Diophantine approximations can find many other satisfactory values of S involving irrational numbers. These values satisfy Inequality (I) for most values of B. They can be translated into arrays of heat transfer elements according to this invention. Of course, these patterns will not repeat and no two fins will be in the same column line.

Figure 1 is an example, however, of a repetitive pattern of heat transfer elements 101 arranged on a substrate in accordance with the invention. It can be seen that the pattern repeats every 8 rows so that fins which are longitudinally aligned are spaced by 8 rows. In some column lines, only one fin 101 is found.

Figure 2 shows a local section of another repetitive pattern of fins which has been distorted by transverse exaggeration. Elements 201 are surrounded by representative isotherms 203. Fluid flows in the "x" direction. Element 211 can be considered the "zeroth" element and has X and Y coordinates of (0,0). W is the distance along the "y" axis, transverse to the flow, between Element 211 and Element 212, the next element having an x-coordinate of 0. The tranverse width of the thermal boundary layer following each of the elements is 1/13 of the width W. M, the number of rows of elements in the offset pattern is thus 13. In Figure 2, the integer P, which has been selected to be relatively prime to M, is 5. Thus S, the offset factor, is P/M or 5/13. Each successive row is therefore displaced transversely across the flow in the y-direction by 5/13 of W. Thus, Element 207 is displaced from Element 209. Likewise, along the x-direction, Elements having the same y-coordinate occur only every 13 x-units. Thus, a streamline impinging on the zeroth Element 211 will not again impinge an element until the 13th row farther downstream. During the intervening time, much of the heat will diffuse away sideways. Therefore, heat transfer from a downstream element will be more efficient than if the subsequent element were located only one or two rows away.

Figure 3 shows further isotherms 203 around the fin positions 201 in the heat exchanger of Figure 2 and demonstrates more clearly the fact that streamlines do not impinge on more than one heat transfer element.

A heat exchanger made according to this invention can be used in a variety of applications, including cross-flow and tube fluid-fluid heat exchangers, in fin-and-fluid heat exchangers and the like. Fin-and-fluid heat exchangers can be used to dissipate heat from integrated circuits on a small scale. The heat transfer element can be fins, tubes, wires or many other structures useful for convective heat transfer.

Preferably, in fin-and-fluid heat exchangers, the fins should be streamlined. Where the fin is heated only at the bottom, it should have a cross-section like a symmetric airfoil, have a wedged thickness decreasing from base to tip, and tapered chord decreasing from base to tip, a curved leading edge and a straight trailing edge. However, any type of fin known to those of skill in the art which is appropriate to the application may be used.

Although the above discussion described embodiments of this invention in which fluid flowed in only one direction, there can be embodiments of this invention having radial flow between central and cicumferential fluid terminals. In general, the offset pattern of this invention can be applied to any fluid flow or topology known to those of skill in the art.

For example, the novel offset pattern of this invention can be used in three-dimensional heat exchangers. For example, metal wire emerging from the substrate can be randomly crushed and used as convective fins. Alternatively, fins in a conventional fin-and-fluid transfer device can be bent along random planes which thereby offset the fins at random.

The heat exchangers of this invention can be made according to methods known to those of ordinary skill in the art. For example, a fin-and-fluid type heat exchange apparatus can be made by the "lance and offset" process. In this process, a large metal sheet is lanced to produce slots. The resulting sheet is then pressed parallel and perpendicular to its base plane. The result is a dense array of fins.

Claims

1. A heat exchanger for transferring heat between the exchanger and a fluid, the heat exchanger comprising at least one surface provided with a plurality of heat exchanging fins (101, 201) positioned on a grid defined by intersecting row and column lines such that each fin lies on a row line and in longitudinal alignment with a column line, characterised in that:-

no fin is aligned, in either the row or column directions, with its nearest or next nearest neighbours.

2. A heat exchanger as claimed in claim 1, in which the fins are arranged at uniform separation - (W) transversely to their length in uniformly pitched rows and adjacent rows are staggered by an offset which is a proportion (S), other than $\frac{1}{2}$, of the uniform separation (W) of the fins so that any longitudinally aligned fins must be spaced by more than twice the row pitch.

3. A heat exchanger as claimed in claim 2, in which the proportion (S) defining the offset is a rational fraction (P/M) so that any longitudinally aligned fins must be spaced by a corresponding integral multiple (M), greater than two, of the row pitch.

4. A heat exchanger as claimed in claim 3, in which the numerator (P) of the fraction defining the offset is greater than one.

5. A heat exchanger as claimed in claim 4, in which the fraction defining the offset is 5/13.

6. A heat exchanger as claimed in any preceding claim, in which at least one column includes at least two longitudinally aligned fins.

7. A heat exchanger as claimed in claim 2, in which the proportion (S) defining the offset approximates to an irrational number so that no two fins occur in the same column line.

8. A heat exchanger as claimed in any preceding claim, in which the plurality of fins are parallel to each other.

9. A heat exchanger as claimed in any one of claims 2 to 7, in which the columns of fins diverge radially.

10. A heat exchanger as claimed in any preceding claim, in which each of the fins has a base, a tip, a leading edge and a trailing edge, each fin decreasing in both width and length from base to tip and each fin having a curved leading edge and a straight trailing edge.

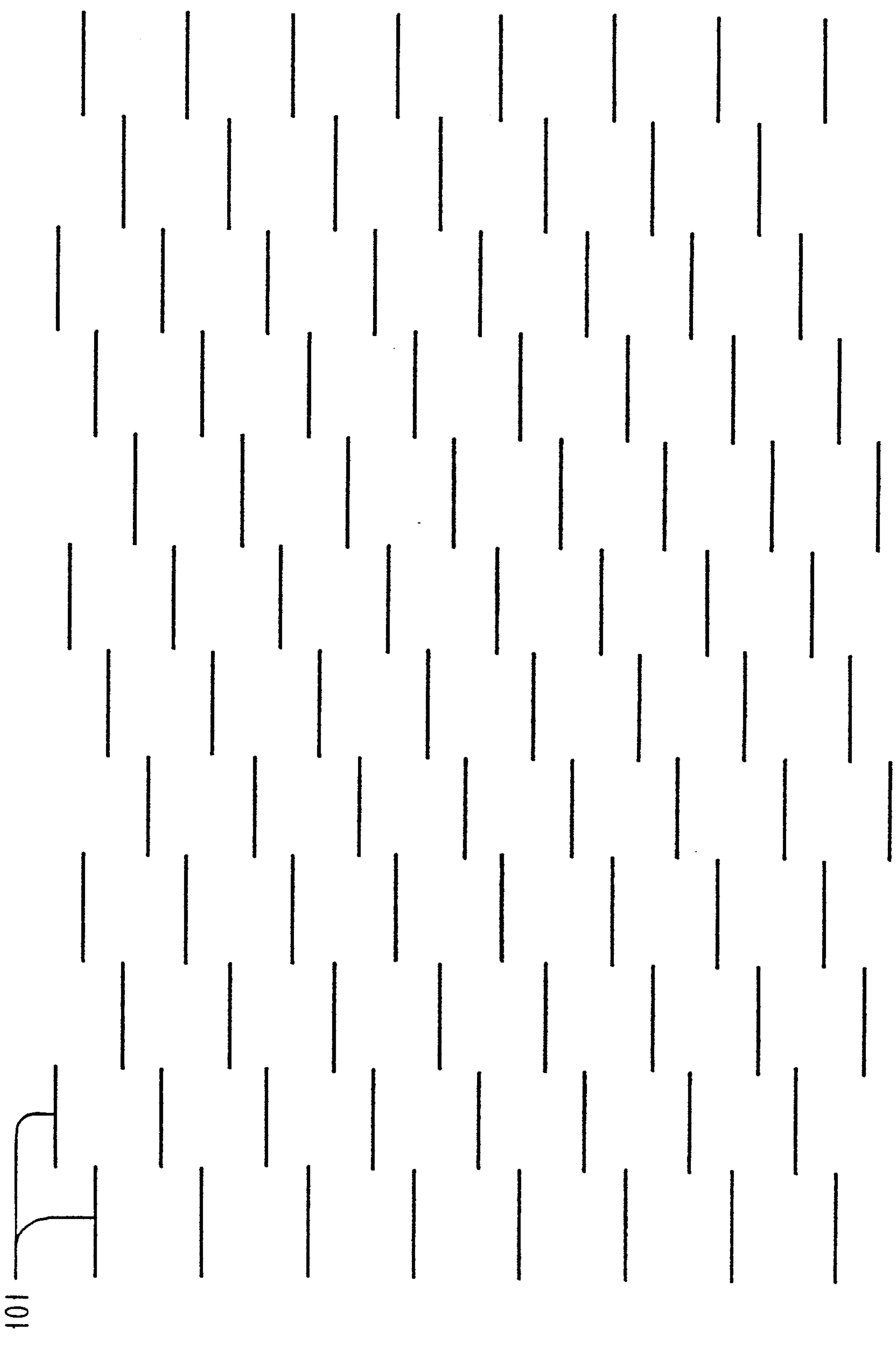

FIG. 1

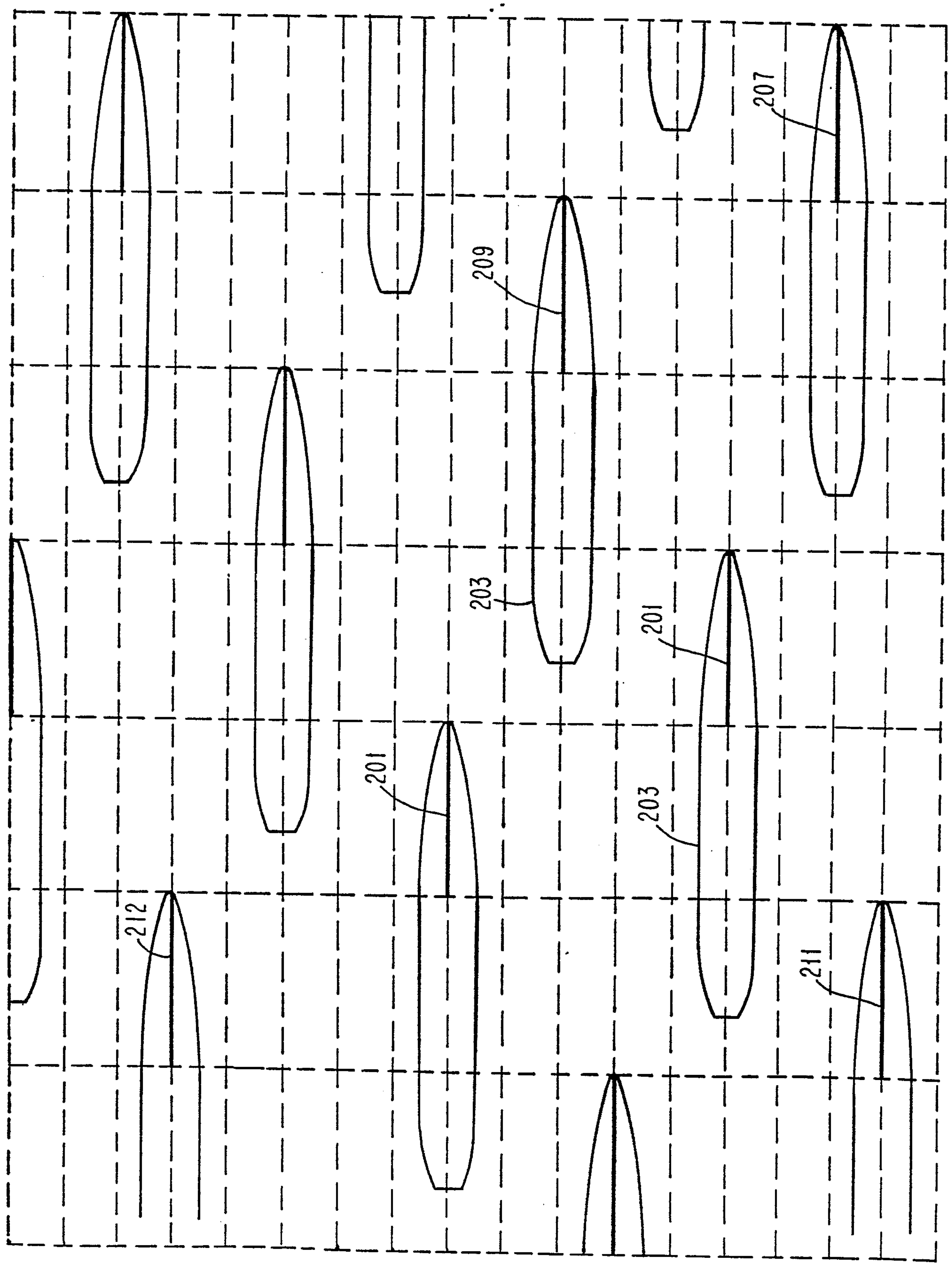

FIG. 2

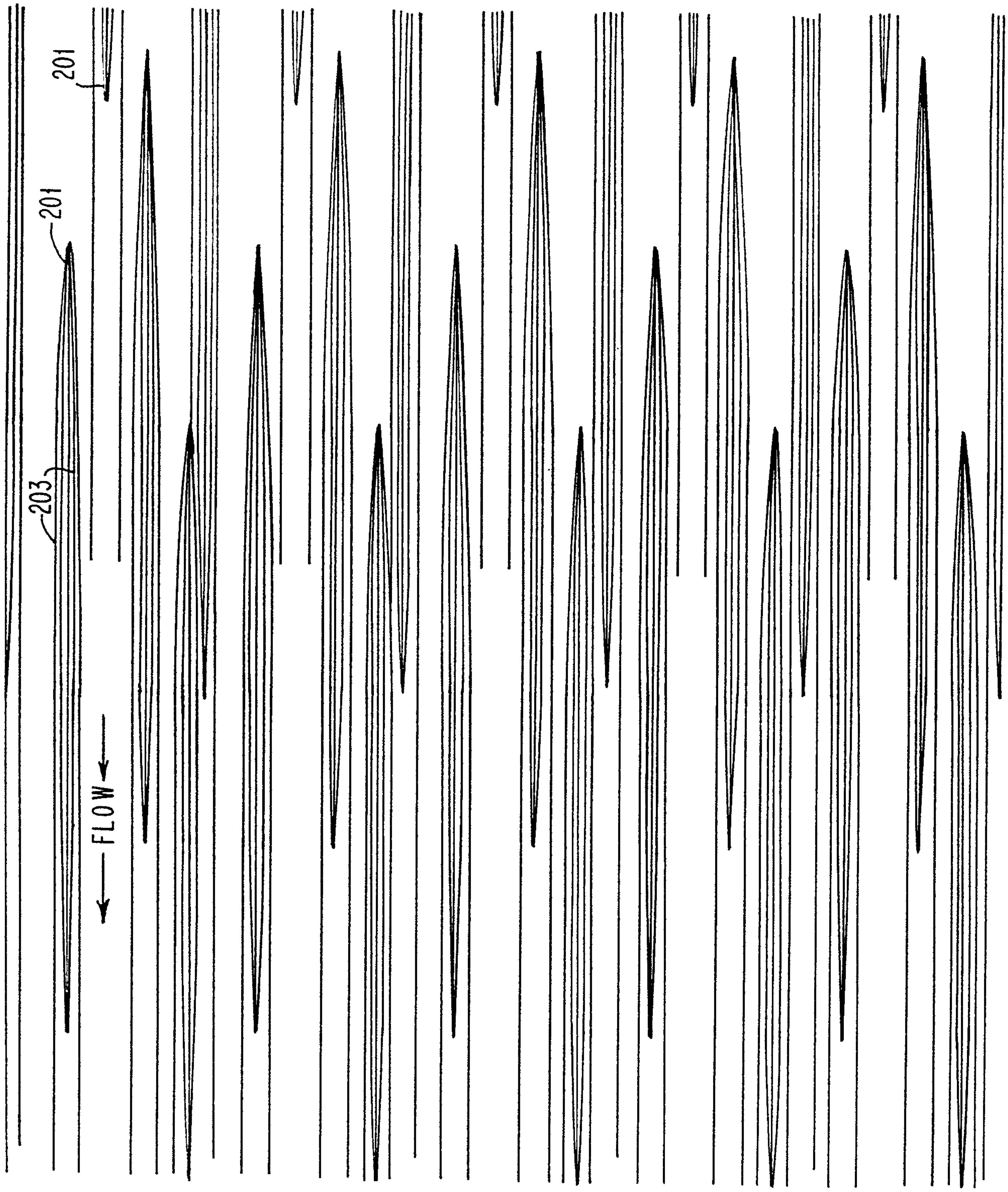

FIG. 3

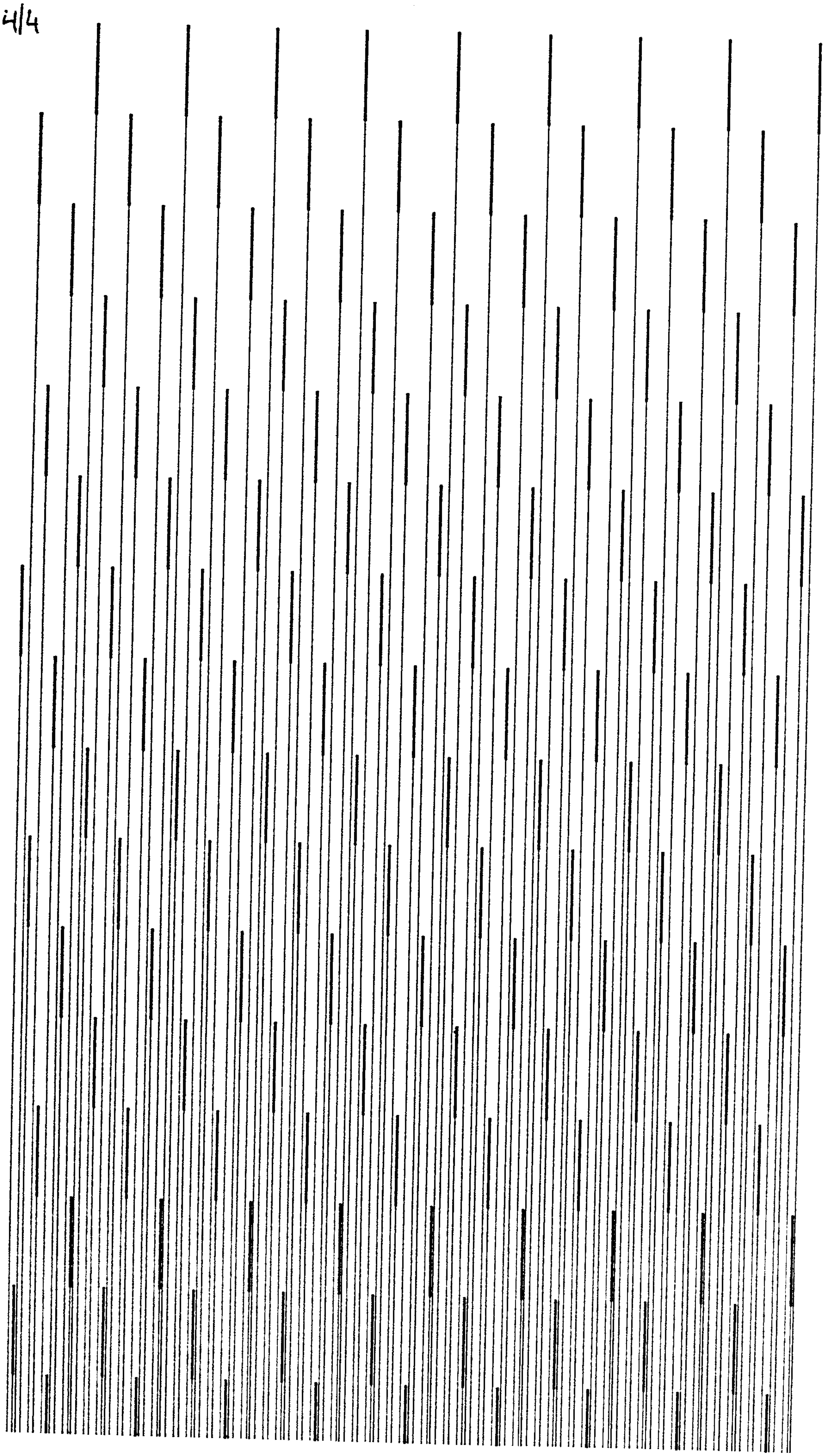

FIG. 4

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EUROPEAN SEARCH REPORT

Application number

EP 86 30 4626

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
X	US-A-4 019 494 (SAFDARI) * Abstract; figures 1,2,5; column 1, lines 35-43,55-68; column 2, lines 1-11,23-27; column 3, lines 13-21; claims 1-5 * ---	1-4,6-8	F 28 F 3/04 F 28 F 1/14
X	US-A-2 789 797 (SIMPELAAR) * Column 2, lines 51-73; column 3, lines 1-55; column 4, lines 65-75; column 5, lines 1-64; claims 1-3,12-14; figures 1,2,6-9 * ---	1-4,6-8	
A	GB-A- 627 859 (FRENKEL) * Claims 1,2; figure 2 * ---	1-4,6,8	TECHNICAL FIELDS SEARCHED (Int. Cl.4)
A	GB-A- 883 547 (LAUGHTON) * Figures 1,4,5 * ---	1,8	F 28 F F 28 D F 15 D
A	US-A-2 647 731 (LUDLOW) * Figures 1,3,4 * ---	1	
A	US-A-2 133 502 (EMMONS) * Figures 3,4 * ---	1	
A	DE-A-2 508 727 (RANDELL) * Claim 1; figures 4,5 * --- -/-	9	

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	26-09-1986	HOERNELL, L.H.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
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EUROPEAN SEARCH REPORT

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DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
A	DE-C- 325 832 (NORDDEUTSCHE KÜHLER FABRIK) * Figures 2,3 * ---	9	
A	FR-A- 536 145 (SOCIETE MULLER) * Figures 1-3 * ---	10	
A	CH-A- 234 859 (SULZER) * Figures 1,2 * -----	10	
			TECHNICAL FIELDS SEARCHED (Int. Cl.4)

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