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(54) HEAT EXCHANGER AND METHOD FOR EXCHANGING HEAT

(57) A heat exchanger according to the invention comprises a shell (7) and at least one channel, located in its interior, with at least one perforated wall (5) adapted to carry a first fluid. The perforation of the wall is provided on at least a part of the surface of this perforated wall (5) so that, as a result of the flow of the first fluid, microjets are formed in the perforation openings. The shell (7) is adapted to carry a second fluid, and the said at least one channel has an additional solid wall (6) surrounding the perforated wall (5).

A method for transferring heat between the first fluid and the second fluid according to the invention consists in that these fluids are introduced into the heat exchanger in which they are separated from each other by a partition, wherein the exchanger according to the invention is used therefore.

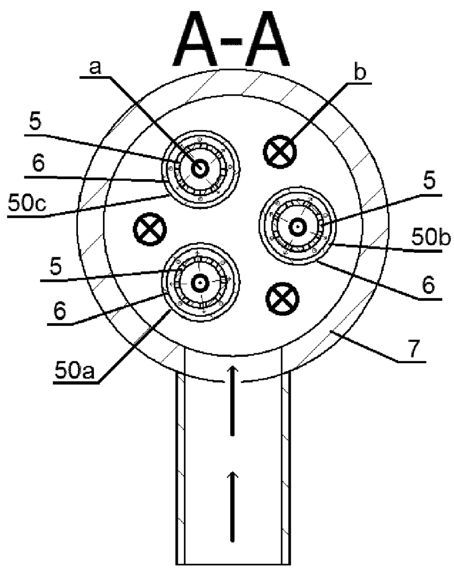

Fig. 2

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Description

Field

[0001] The invention concerns a shell-and-tube heat exchanger designed to perform a heat exchange, both in single-phase two-phase convection mechanisms, in general applications in the area of a broadly defined heat technology, and in particular power industry, in industrial processes, as well as in the systems for the recovery of waste heat from technological and energetic or power engineering processes.

Prior art

[0002] In the state-of-the art, numerous mechanisms for the hydrodynamic intensification of heat exchange are known. They include, for example, the use of jet impingement onto the surface of heat exchange. This mechanism allows reducing or eliminating one of the factors limiting the speed of heat exchange, namely a boundary layer of fluid at the surface of heat exchange. As a result of the occurrence of the boundary layer of fluid, an undesirable share of thermal conductivity in the mechanism for transferring heat through the partition increases. Thermal conductivity is less efficient than convection. By reducing or eliminating the boundary layer, the gradient of temperature between the fluid and the heat exchange surface, i.e. the partition surface, is reduced. The use of fluid jet of sufficiently high speed, rinsing the heat exchange surface makes the boundary layer disturb and locally disappear. Therefore, by using jets rinsing uniformly the entire surface of heat exchange, the boundary layer can be almost completely eliminated.

[0003] In international application published under the number WO 2009/095896A1, entitled "Fluid microjet system", a device for generating microjets with which solids are cooled is disclosed. This device consists of a set of plates with a common channel, and microchannels reaching it. In US Patent no 5 329 994, entitled "Jet impingement exchange", an exchanger consisting of two plates separated by an additional sieve plate which, at a fluid flow, generates jets rinsing the heat exchange surfaces is disclosed.

[0004] It is difficult to use the jet impingement in shell-and-tube heat exchangers due to the fact that in such heat exchangers a uniform coverage, with jets, of the entire surface of heat exchange is very troublesome, and not using the entire surface of heat exchange has a negative effect on the heat exchange efficiency and nullifies the benefits arising from the use of jet impingement.

[0005] In US Patent no US 8051900 B2, a shell-and-tube heat exchanger, wherein inside the shell there is a bundle of tubes, is disclosed. In this bundle of tubes, there are tubes with a solid wall and tubes with a perforated wall. In openings of the tubes with a perforated wall, jets of a first fluid are formed and rinse the tubes with a solid wall in which a second fluid flows.

[0006] Although the solution disclosed in document US 8051900 was designed to increase the use of heat exchange surface rinsed with jets of the first fluid, it is, however, not perfect due to the fact that by increasing the number of tubes with a perforated wall, with the same cross section of the exchanger, the number of tubes with a solid wall has to be reduced. Actually, the heat exchange surface is the wall surface of the solid tubes. Therefore, by increasing the utilisation of the heat exchange surface, its size is reduced, and consequently intensification of heat exchange is limited.

[0007] The aim of the present invention is to solve the above-described problems occurring in the in the state-of-the art, in particular by providing an exchanger structure allowing the use of the heat exchange intensification by rinsing with jets, while ensuring a large heat exchange surface in relation to dimensions of the heat exchanger and making full use of the surface.

Summary the invention

[0008] The heat exchanger according to the invention comprises a shell and at least one channel, located in its interior, with at least one perforated wall adapted to carry the first fluid, wherein the perforation of the wall is provided on at least a part of the surface of this perforated wall so that, as a result of the flow of the first fluid, microjets are formed in perforation openings. The exchanger according to the invention is distinguished in that the shell is adapted to carry a second fluid, and at least one channel has an additional solid wall surrounding the perforated wall. Consequently, the additional solid wall constitutes a partition between the first fluid and the second fluid and this partition is rinsed with microjets of the first fluid. As a result, the rinsing with microjets takes place on the entire surface of the partition and is very uniform. Thanks to this, the heat exchange surface, that is constituted by the partition, is more fully used, and simultaneously an advantage of desired thermal effects of the rinsing with microjets is taken.

[0009] The perforation openings in the perforated wall have dimensions preferably smaller than 500 micrometres, which ensures the formation of microjets in case of most of the fluids used in a large range of pressures and flow rates used. In particular, these dimensions may be diameters.

[0010] The perforation openings in the perforated wall are constituted by circular openings made in a staggered configuration, which ensures a more uniform rinsing.

[0011] Alternatively, the perforation openings in the perforated wall are made in a checkerboard configuration.

[0012] Preferably, the ratio of the distance between the solid wall and the perforated wall to the average diameter of the perforation openings is in the range of 0.5 to 25.

[0013] The shell has preferably a cylindrical shape, and the perforated wall and the solid wall are concentric tubes. The use of such shapes allows obtaining advantageous mechanical properties - in particular resistance to high pressures of fluids, with a relatively small wall thickness. Thanks to this, the exchanger may be lighter.

[0014] Preferably, in the shell, there is a bundle comprising at least 3 pairs of concentric tubes with perforated and solid walls.

[0015] Preferably, the shell is provided with auxiliary partitions positioned so as to force cross and parallel-current flow, i.e. so-called mixed flow of the second fluid. Such flow also promotes acceleration of heat exchange.

[0016] Preferably, at least one solid wall of at least one channel is externally finned on at least a part of its surface. Such solution also accelerates heat exchange. Ribbing on individual channels, in particular on individual tubes, can be used, and common ribbing for all or a part of the solid walls of the channels can be also used.

[0017] The method for transferring heat between a first fluid and a second fluid according to the invention consists in that these fluids are introduced into the heat exchanger in which they are separated from each other by a partition. A characteristic feature of this method is that as an exchanger the exchanger according to the invention is used. The partition through which the heat is exchanged is then constituted by solid walls.

[0018] As a first fluid and a second fluid, virtually any fluid used for transferring heat, heating or cooling can be used, including in particular water, alcohol, flue gas, air, oil, low-boiling fluids or nanofluids.

[0019] The first fluid is preferably a gas, and the second fluid is a liquid. The heat exchange between the gas and the partition is usually worse, and the use of rinsing of the partition with gas microjets improves the process.

[0020] Also preferably, the first fluid is a condensing vapour, and the second fluid is a liquid. This is also the situation in which the use of the exchanger according to the invention allows compensation of differences in heat exchange rate between the partition and the first fluid, and between the partition and the second fluid.

[0021] Also preferably, the first fluid is a gas, and the second fluid is a vapourising liquid. This is also the situation in which the use of the exchanger according to the invention allows compensation of differences in heat exchange rate between the partition and the first fluid, and between the partition and the second fluid.

[0022] Preferably, the first fluid and the second fluid are introduced into the exchanger in the countercurrent manner.

[0023] Preferably, the first fluid and the second fluid are introduced into the exchanger cocurrently when one of the fluids is a vapourising liquid , or when one of the fluids is a condensing vapour.

[0024] The advantage of the heat exchanger with a heat exchange intensification by microjets according to the invention is that its construction allows for obtaining a high efficiency of energy transport while maintaining external dimensions and mass of the exchanger thanks to the good use of the heat exchange surface while intensifying the heat exchange by rinsing of this surface with jets. With fixed thermal parameters, this solution results in a significant reduction in external dimensions and mass of the exchanger, as well as in material consumption and costs of its production. Heat exchangers made according to the invention may be combined into larger networks of exchangers. Additionally, an advantage of the method according to the invention is that it allows compensation of differences in heat exchange rate between the partition and the first fluid, and between the partition and the second fluid.

Description of Figures

[0025] The object of the invention is shown in embodiments in the drawing in which Fig. 1 shows a general view of an embodiment of the shell-and-tube exchanger according to the invention, Fig. 2 shows a sectional view of this embodiment of the shell-and-tube heat exchanger according to the invention, Fig. 3 shows a schematic diagram of flow organisation of a heating fluid in a bundle and of a heated fluid rinsing the bundle, Fig. 4a shows a perforated wall with a perforation in a staggered configuration, Fig. 4b shows a perforated wall with a perforation in a checkerboard configuration, Fig. 5 shows a distribution of tubes in a bundle inside the shell in an embodiment of the exchanger according to the invention.

Detailed description of embodiments

[0026] In Fig. 1, a general view of the shell-and-tube heat exchanger with a microjet intensification of heat exchange in the bundle tubes is shown. Fig. 2 shows this exchanger in cross-section.

[0027] The channels carrying the first fluid are tubes 50a, 50b, 50c, with double walls. Walls of these tubes are concentric. A perforated wall 5 forms an inner tube, and a solid wall 6 forms an outer tube.

[0028] A port 1 for the inlet of the heating agent, i.e. the first fluid, is used to supply the fluid to the bundle of tubes 50a, 50b, 50c, more specifically to the interior of the tubes with perforated walls 5. The tubes with perforated walls 5 are enclosed in the tubes with solid walls 6 covering them. The cooled first fluid is discharged through a port 2 for the outlet of the heating agent. A heated agent, i.e. the second fluid, is introduced through a port 3 for the inlet of the heated agent, and, after heating, is discharged through a port 4 for the outlet of the heated agent.

[0029] The first fluid from the port 1 for the heating agent inlet is distributed to three tubes with perforated walls 5. The perforation openings in these walls have a diameter of 500 micrometres or smaller. As a result of the first fluid flowing through the tube with a perforated wall 5, jets rinsing the inner surface of the tube with a solid wall 6 are generated. Thanks to this, the whole inner surface of the solid wall 6 is used as a heat exchange surface, subject to intensification due to the jet impingement. The solid walls 6 constitute a partition between the first fluid and the second fluid. Their total surface constitutes a heat exchange surface. The heat exchange surface is the accumulated surface of the walls 6.

[0030] The second fluid is carried inside the shell 7 and isolated from the first fluid by solid walls 6 of the bundle tubes 50a, 50b, 50c.

[0031] In the cross-section shown in Fig. 2, flow direction "a" of the first fluid and flow direction "b" of the second fluid are indicated. They are in opposite directions. The tube bundle covered with the shell is constituted by three pairs/sets of concentric tubes 50a, 50b, 50c carrying the first fluid. The space between the inner perforated wall 5 and the outer solid wall 6 of each of the tubes 50a, 50b, 50c is not filled with the first fluid. This fluid flows out from the perforation openings 5 with jets onto the wall 6. Therefore, the space between the walls 5 and 6 is partially occupied by the first fluid jets. The staggered configuration of the perforation openings ensures that the wall 6 is rinsed with jets in a very similar manner to the completely uniform one. A tube with the staggered distribution of the perforation openings is shown in Fig. 4a. Alternatively, any other distribution of openings, e.g. checkerboard configuration illustrated in Fig. 4b, can be used, although it provides a somewhat slower heat exchange than the staggered distribution.

[0032] The heat exchanger according to the invention can be used with virtually any fluids used for transferring heat, heating and cooling, including in particular water, alcohols, flue gases, air, oil, low-boiling fluids or nanofluids. Terms nanoliquids and nanofluids are used interchangeably to describe a fluid being a suspension of solid particles, in particular metals e.g. aluminium.

[0033] The exchanger according to the invention is particularly useful in heat transfer between the gas and the liquid. Generally, gases are worse at transferring heat, and also the influence of the boundary layer on the heat exchange with a gas is more significant than in the case of a liquid. Therefore, in most known heat exchangers between the gas and the liquid separated from each other by a metal partition, a major limitation to the heat exchange rate is the intensity of the heat exchange between the partition and the gas.

[0034] The use of the exchanger according to the invention for the heat exchange between a gas as the first fluid and a liquid as the second fluid allows a very effective intensification of the exchange by the elimination of "bottleneck" i.e. by intensification of the heat exchange between the partition, constituted by the solid walls 6, and the gas, by eliminating the boundary layer being smashed with jets.

[0035] For example, with a single microjet, heat transfer coefficient for the air is approx. 1000 W/(m^{2}K) with an impingement of air under the pressure of 300 kPa and for the temperature difference between the partition and the fluid of 150 K.

[0036] For water, the coefficient is 25 000 W/(m^{2}K) with the temperature difference of 60 K and other conditions unchanged.

[0037] The diameter of the perforation opening which is used in the heat exchanger according to the invention is associated with the pressure of the first fluid. This problem is complex because the choice of this diameter is also dependent on the mass flow of the first fluid. These parameters should be adjusted so that as a result of fluid flow under a predetermined pressure through a tube having a wall perforated with openings with a predetermined diameter, microjets are formed over its entire length. The diameter of the openings of up to 500 micrometres ensures the formation of microjets in a wide range of pressures. The pressure drop generated during the formation of a first fluid jet in an opening of the perforated wall 5 is dependent on the opening diameter and the wall thickness. A small thickness of the wall and a parallel flow distribution through the perforation openings in an embodiment of the exchanger according to the invention are very advantageous because they minimise the total hydraulic resistance associated with the flow of the first fluid through the heat exchanger.

[0038] The distance between the perforated wall (5) and the solid wall (6) is usually determined after determining the diameter of the perforation openings. This value affects the heat exchange rate under conditions in which the formation of microjets is ensured. It can be chosen experimentally or by numerical analysis. The best results were obtained under conditions in which the ratio of the distance between the solid wall and the perforated wall to the diameter of the perforation opening is in the range of 0.5 to 25.

[0039] Generally, the aim is to maximise the number of tube pairs 50a, 50b, 50c forming a tube bundle in the shell because, along with the increase in this number, also the maximum possible heat exchange surface for predetermined dimensions of the shell 7 increases. Good packing of tubes within the shell improves compactness of the heat exchanger. Compactness is determined by a parameter defined as the ratio of heat exchange surface of the exchanger to volume occupied by the fluid having worse properties for heat exchange. An example of tube distribution in the cylindrical shell is shown in Fig. 5. A tube bundle arranged on a geometric grid is shown therein, with the omission of inner perforated tubes. Together with the diameter of the outer tube and the diameter of the shell D, distance k between the outer tubes and the walls of the shell and the distance between the centres of the tubes, i.e. so-called scale S, are provided. Distance S is determined from the formula $S = A * dz$, where A is within the range of 1.2 to 5.0. Distance k is approximately equal to half the diameter of the outer tube.

[0040] In the embodiment illustrated in the drawing and discussed above, the exchanger according to the invention is used in a countercurrent configuration in which the first fluid flows through the heat exchanger in a direction substantially opposite to the flow direction of the second fluid. This configuration is preferable in most cases. Sometimes, however, it is advisable to direct the flow of the two fluids in the same direction, for example, in cases where phase transformations, i.e. boiling or condensation, take place in the exchanger. Naturally, in this situation, the intensification of heat exchange will also occur, as a result of the rinsing of the heat exchange partition with jets of the first fluid.

[0041] Alternatively, in the shell of the exchanger, auxiliary partitions which will locally change, and even reverse the flow direction of the second fluid can be used. They will lead to turbulisation in the flow of the second fluid, thereby forcing a cross and parallel-current flow, i.e. so-called mixed flow of the second fluid. This modification further accelerates the heat exchange between the second fluid and the partition. This solution is appropriate if as a result of the heat exchange intensification between the first fluid and the partition, the rate of this exchange exceeds the rate of the exchange between the second fluid and the partition. The use of partitions will accelerate the resultant heat exchange.

[0042] In addition, the solid tubes 6 can be externally finned. Selected ones or all of the tubes can have individual finning on part or on the whole surface, or they can have a common finning resembling a lamella.

[0043] It is clear that the above embodiments and drawings illustrating them were provided only for ease of explanation of the essence of the invention, the proper scope of which is defined by the patent claims. This scope includes methods and heat exchangers in which various shapes of openings and their sizes in the perforated wall, and various densities and distributions, as well as various distributions and number of tubes in a bundle, or different geometrical dimensions of the exchanger can be used. There is not any reason to interpret these given examples as limiting in any way.

Claims

1. A heat exchanger comprising a shell (7) and at least one channel, located inside the shell and having at least one perforated wall (5) adapted to carry a first fluid, wherein the perforation of the wall is provided on at least a part of the surface of this perforated wall (5) so that, as a result of the flow of the first fluid, microjets are formed in perforation openings,
   characterised in that
   the shell (7) is adapted to carry a second fluid, and at least one channel has an additional solid wall (6) surrounding the perforated wall (5).

2. The heat exchanger according to the claim 1, **characterised in that** the perforation openings in the perforated wall have dimensions smaller than 500 micrometres.

3. The heat exchanger according to the claim 1 or 2, **characterised in that** the perforation openings in the perforated wall are constituted by circular openings made in a staggered configuration.

4. The heat exchanger according to the claim 1 or 2, **characterised in that** the perforation openings in the perforated wall are constituted by circular openings made in a checkerboard configuration.

5. The heat exchanger according to any of the claims 1 to 4, **characterised in that** the ratio of the distance between the solid wall (6) and the perforated wall (5) to the average diameter of the perforation openings is in the range of 0.5 to 25.

6. The heat exchanger according to any of the claims 1 to 5, **characterised in that** the shell has a cylindrical shape, and the perforated wall (5) and the solid wall (6) form a pair of concentric tubes (50a, 50b, 50c).

7. The heat exchanger according to the claim 6, **characterised in that** in the shell (7), there is a bundle comprising at least 3 pairs of concentric tubes with perforated walls (5) and solid walls (6).

8. The heat exchanger according to the claim 7, **characterised in that** the shell (7) is provided with auxiliary partitions positioned so as to force cross and parallel-current flow of the second fluid in the shell (7).

9. The heat exchanger according to the claim 7, **characterised in that** the solid wall (6) of at least one channel is externally finned on at least a part of its surface.

10. A method for transferring heat between the first fluid and the second fluid, wherein these fluids are introduced into the heat exchanger in which they are separated from each other by a partition, **characterised in that** as an exchanger, the exchanger as defined in any one of the claims 1 to 9 is used.

11. The method according to the claim 10, wherein the first fluid is a gas, and the second fluid is a liquid.

12. The method according to the claim 10, wherein the first fluid is a condensing vapour, and the second fluid is a liquid.

13. The method according to the claim 10, wherein the first fluid is a gas, and the second fluid is a vapourising liquid.

14. The method according to any of the claims 10 to 13, **characterised in that** the first fluid and the second fluid are introduced into the exchanger countercurrently.

15. The method according to claim 12 or 13, **characterised in that** the first fluid and the second fluid are

introduced into the exchanger cocurrently.

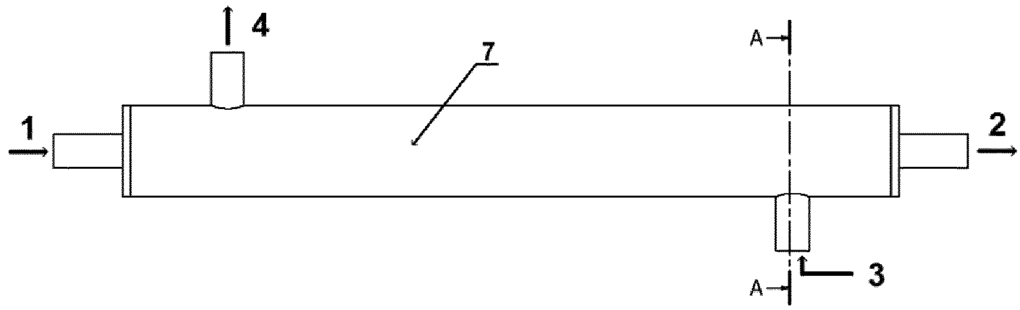

Fig. 1

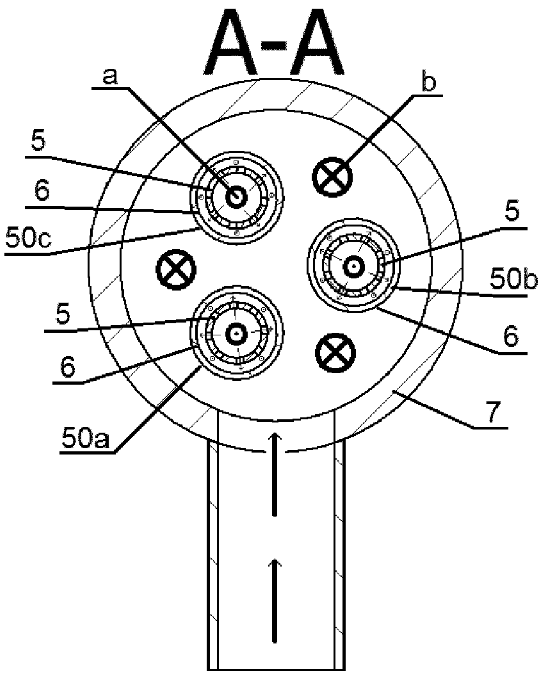

Fig. 2

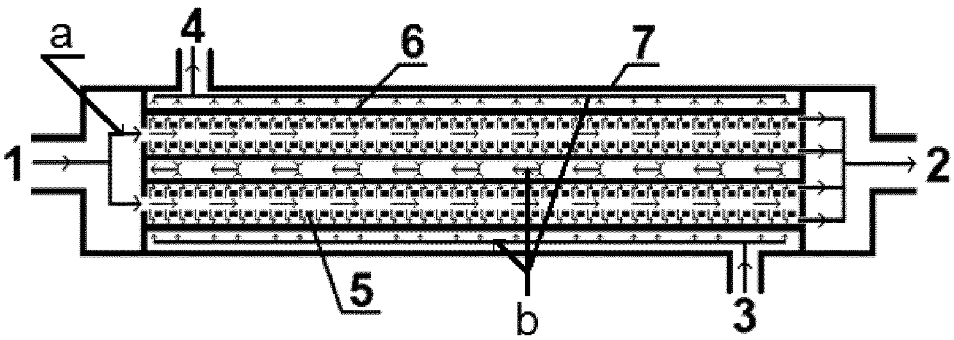

Fig. 3

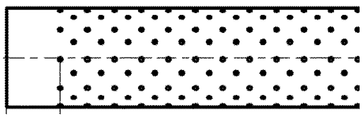

Fig. 4a

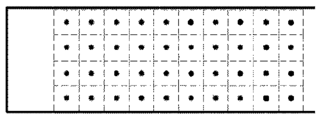

Fig. 4b

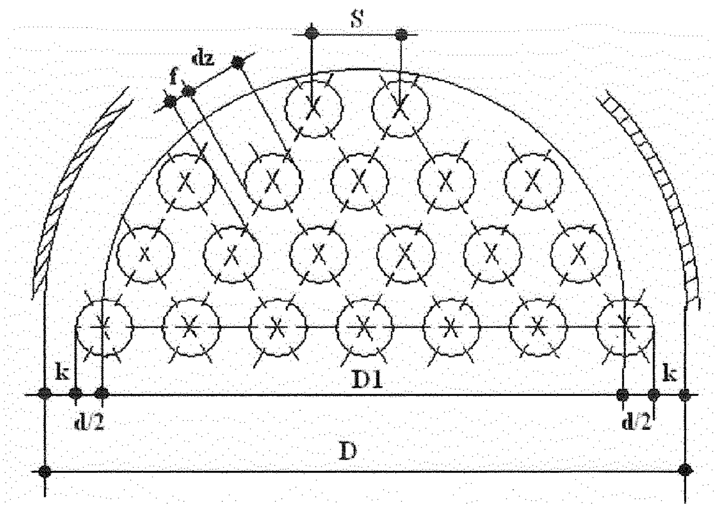

Fig. 5

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EUROPEAN SEARCH REPORT

Application Number
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The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
Munich	1 September 2015	Leclaire, Thomas

CATEGORY OF CITED DOCUMENTS

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EUROPEAN SEARCH REPORT

Application Number
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1 The present search report has been drawn up for all claims

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Munich	1 September 2015	Leclaire, Thomas

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