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(54) **Improved efficiency indirect evaporative cooler**

(57) An indirect evaporative cooler wherein the exhaust air from the wet passages is drawn directly from the wet passages by a first fan (66) while a second fan (64 or 42) powers the movement of air to be cooled through dry passages of the cooler.

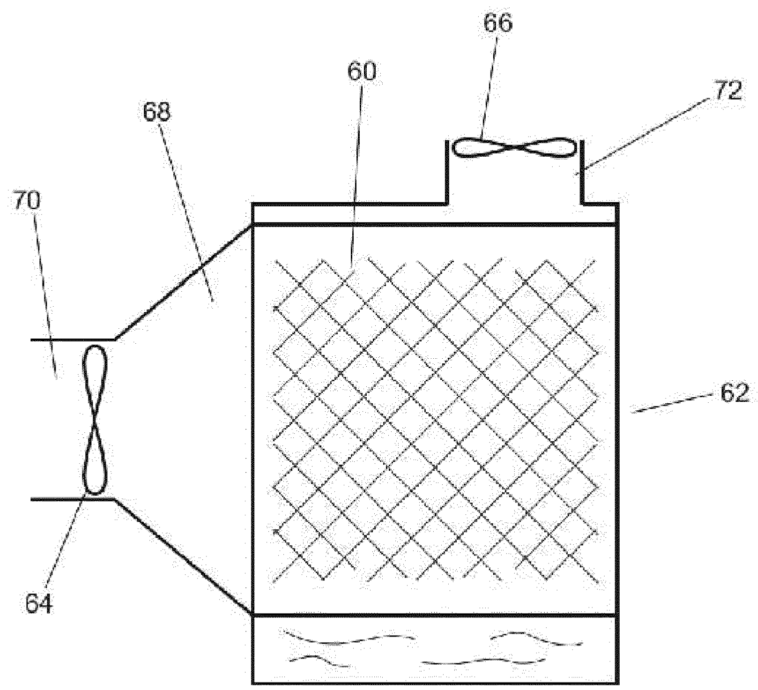

FIGURE 4

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Description

Technical Field

[0001] This invention relates to evaporatively cooled heat exchangers utilised in the cooling of air for the comfort cooling of buildings. These heat exchangers are generally constructed from adjacent wet and dry passages arranged such that air through the adjacent passages flows in relative counter flow.

[0002] In particular, the present invention concerns a method and means for significantly improving the operating efficiency of indirect evaporative cooling systems.

Background Art

[0003] Throughout this description and the claims which follow, unless the context requires otherwise, the word "comprise', or variations such as "comprises" or "comprising", will be understood to imply the inclusion of a stated integer or step or group of integers or steps.

[0004] The reference to any prior art in this specification is not, and should not be taken as, an acknowledgement or any form of suggestion that that prior art forms part of the common general knowledge in Australia.

[0005] The concept of indirect evaporative cooling, wherein a heat exchanger is arranged with alternating wet and dry passages with air flowing in counter flow within the adjacent passages has been well known for some time. One of the earliest references is SU 979796 by Maisotsenko illustrates the principle of counter flow indirect cooling.

[0006] More recently, practical indirect coolers have been built in accordance with this principle to produce useful quantities of air cooled to temperatures approaching the Dew point of the incoming air supply. An early device of this kind is described in US 4,977,753 by Maisotsenko which follows the principles of SU 979796 by using full counter flow of air between adjacent passages and restricting flow of water over the wetted surfaces by utilising wicking on the wetted surfaces to distribute water. Incoming air pressurised using a high pressure fan is directed firstly through the dry passages. Upon emerging from the dry passages, a proportion (generally about half) of the air is turned back through the wet passages through which it passes until emerging to atmospheric pressure as exhaust. The remainder of the air, not turned back through the wet passages, emerges as useful (or supply) air which has been cooled to a temperature approaching the Dew Point of the incoming air.

[0007] The pressure required to drive the proportion of air through the wet passages is produced by applying a restriction, or baffle, to the supply air exiting the dry passages. This represents a waste of energy supplied by the fan to the airstream, since energy in the supply air has been dissipated by being restricted via a baffle for no thermodynamic advantage.

[0008] A similar design is presented in US 2006/0124287 by Reinders. Air flow and subsequent energy losses are similar, although this design presents a variation to the design of the heat exchanger elements. WO2006074508 by James also uses a similar air flow configuration but with a variation in the construction of the heat exchanger.

[0009] A variation of the Maisotsenko design which attempts to make the indirect evaporative cooler more energy efficient is described in US 5,301,518 by Morozov et al. In this design, the indirect evaporative heat exchanger is divided into two parts, each with air flow patterns similar to the Maisotsenko design. The total air flow from the high pressure fan passes through dry passages of the first half of the heat exchanger. Upon exit, a proportion of the air is turned back into the wet passages of the first half heat exchanger and follows the wet passages to exhaust. The remaining air then flows through the dry passages of the second heat exchanger. Upon emerging from the dry passages of the second heat exchanger, a proportion of the air is turned back into the wet passages and follows the wet passages of the second heat exchanger to exhaust. Again, the pressure required to send the air emerging from the second heat exchanger back through the wet passages is achieved by baffling the supply air exiting the dry passages.

[0010] The Morozov design passes the full air flow only through the first heat exchanger dry passages, which are shorter in length to achieve the same cooling relative to the original Maisotsenko design and therefore result in less pressure drop through those dry passages. Air flow through the dry passages of the second heat exchanger dry passages is much less (generally about half) the air flow through the dry passages of the first heat exchanger thereby requiring less pressure drop associated with the air flow. Thus the pressure losses in this design are reduced, but the energy loss through the baffle on the supply air is still significant.

[0011] In US 2004/0061245, Maisotsenko describes an alternative design utilising a combination of counter flow and cross flow paths within the heat exchanger. In this design, air in the dry passages is progressively passed through to the wet passages via holes in the heat exchanger walls between adj acent dry and wet passages. Each hole dimensioned to allow only a pre-determined rate of flow between dry and wet passages. Once in a wet passage, air then travels in a cross flow direction to be exhausted at the end of the wet passage. Air flow within the exchanger is controlled by the geometry of the air passages and holes between wet and dry passages and no baffle is required to induce air to flow into the wet passages. Energy in the air flow is still dissipated when passing through the holes between wet and dry passages, but the design does not subject all of the airflow to pressure loss as in the former design. Energy losses are reduced, but at some compromise to thermodynamic efficiency of the cooler.

[0012] In WO2006074508 James describes an alternative construction of heat exchanger utilising the counter air flow principles of the prior art, but wherein improvements have been made to the construction of the heat exchanger to overcome previous problems of size limitation and the flushing of salts from the wet surfaces of the wet channels or passages. This design still has the problem of requiring air to be supplied to the heat exchanger dry passages at high pressure and subsequent inefficient loss of energy by regulating the ratio of supply air to exhaust air by means of throttling through a restriction or baffle.

Disclosure of Invention

[0013] In a first aspect of the present invention there is provided an indirect evaporative cooler comprising:

- a heat exchanger having alternating wet and dry air flow passages such that, in use, a first air flow passes in a first direction through the dry air flow passages and a secondary air flow, being a portion of the first air flow emerging from the dry passages, travels in counter flow to the first direction and through the wet passages to an exhaust;
- a plenum adapted to receive air emerging from the dry passages and from which a portion of the emerging air is, in use, said secondary air and the remainder is supply air;
- means of wetting said wet passages;
- a first fan adapted to move air into and through the dry passages;
  characterised in that a second fan is located downstream of the exhaust of the wet passages to draw said secondary air through the wet passages.

[0014] In a preferred embodiment of the first aspect the first fan is located upstream of the dry passages.
[0015] In a further preferred form the first fan is located downstream of the dry passages
[0016] In a second aspect the present invention provides a method of indirect evaporative cooling wherein a lower than ambient pressure is applied to exhaust outlets of wet passages of a heat exchanger having alternating wet and dry air flow passages and comprising passing a first air flow through the dry passages before drawing off a portion thereof as secondary air through the wet passages by said lower than ambient pressure.

Brief Description of Drawings

[0017] The present invention will now be described by way of example with reference to the accompanying drawings, in which:-

Figure 1 shows the schematic air and water paths through a prior art indirect evaporative cooler;

Figure 2 shows the basic construction of a practical prior art indirect evaporative cooler;

Figure 3 shows a longitudinal section view through the cooler of Figure 2;

Figure 4 is a schematic longitudinal section view through a counter flow indirect evaporative cooler in accordance with a first embodiment of the present invention; and

Figure 5 is a schematic longitudinal section view through a counter flow indirect evaporative cooler in accordance with a second embodiment of the present invention.

Best Modes

[0018] Figure 1 shows a known airflow configuration for an indirect evaporative cooler to function. Incoming air 10 is directed through the dry passages 12 of heat exchanger 20. Upon exiting the dry passages, the air stream is divided into supply air 18 and return air 22, directed into the wet passages 14. The wet passages have a hydrophilic inner surface 16 which is capable of being kept continuously wet. Air from the wet passages emerges through exhaust opening 22 where it is exhausted to atmosphere. Such an arrangement of indirect evaporative cooling is capable of producing supply air 18 at temperatures approaching the Dew Point of incoming air 10 without the addition of moisture to the air.
[0019] Figures 2 and 3 show perspective and section views, respectively, of a practical arrangement for a device exploiting the advantages of indirect evaporative cooling. Air enters from the external ambient through fan 42 which supplies high pressure air to the chamber 44. Heat exchanger 40 is manifolded such that high pressure air from chamber 44 can only flow through the dry channels of the heat exchanger, and air which flows through the dry channels must flow all the way through the dry channels, emerging into chamber 48.
[0020] A proportion of the air emerging from the dry channels into chamber 48 is required to be turned around to flow back through the wet channels spaced between the dry channels of heat exchanger 40. This requires a pressure in chamber 48 sufficient to overcome the flow resistance of the wet channels to leave exhaust 46 at the required flow rate. This pressure is achieved by applying a baffle or restriction 50 to the flow of air leaving chamber 48 through supply air duct 47, the pressure differential across baffle 50 at the required flow rate results in an increased static pressure in chamber 44.
[0021] Fan 42 is required to pressurise air to overcome the pressure loss associated with passing all of the air supplied through the dry channels, plus the static pressure in chamber 48. The static pressure in chamber 48 is sufficient to overcome the flow resistance of the proportion of air flowing through the wet channels to exhaust 46. The static pressure in chamber 48 is regulated by

adjusting baffle 50 thereby producing a static pressure differential across the baffle. The air flow through the baffle 50 at such a differential pressure represents a loss of power equal to the product of the air flow and pressure differential. This loss is an additional power load on fan 42 which provides no additional cooling or otherwise useful energy to the air flow. Although Fan 42 is shown schematically as an axial flow fan, in practice a centrifugal or combined flow fan is generally used due to the high pressures required.

[0022] In a typical indirect evaporative cooler, with a supply to exhaust ratio of 1:1, the fan is required to deliver air at around 600 Pa. If the supply air required is, say, n units, the power required will be 600 x 2n = 1,200n power units. This typically produces a static pressure of around 150 Pa in chamber 48 and thus a pressure differential of 150 Pa across the wet passages to exhaust. The pressure differential across the dry passages of the heat exchanger is 600 Pa - 150 Pa = 450 Pa.

[0023] Figure 4 shows a section through a first embodiment of an indirect evaporative cooler in accordance with the current invention. In this construction, the high pressure fan delivering air to the entrance of the heat exchanger has been replaced by separate fans 64, 66 on the air supply side and the exhaust, respectively.

[0024] External ambient air enters through face 62 of heat exchanger 60. A chamber or plenum (44 in Figures 1-3) delivering air to the entrance of the heat exchanger is no longer necessary since air enters the heat exchanger 60 at ambient air pressure. Fan 64 on the supply air duct 70 produces a negative (less than ambient) pressure in chamber 68 thereby providing the pressure differential across the dry passages of heat exchanger 60 for the required air flow. Fan 66 in the exhaust duct 72 of the heat exchanger provides a negative pressure relative to the pressure in chamber 68 sufficient to produce the required air flow through the wet passages of the heat exchanger. Thus the static pressure immediately before fan 66 will be the sum of the static pressure in chamber 68 and the pressure differential required for the air flow through the wet passages of the heat exchanger 60. The operation of the fans 64 and 66 can be controlled through electronic speed controllers or other means to produce a desired ratio of air flow between the supply air and exhaust air. Furthermore, the magnitude and/or ratio of these air flows can be readily adjusted by varying the speeds of the two fans 64, 66 thereby enabling optimisation of the performance of the indirect evaporative cooler. This allows the indirect cooler to operate under a wide range of conditions through direct control of the fans without the need to intervene and adjust mechanical baffles as in prior art designs, and also allows for automatic control of the operation of the indirect cooler, for example, under the control of a programmable electronic controller.

[0025] Such control is desirable, for example, in the initial cooling of, say, a living space which is initially at a high temperature. In this condition, it is desirable to operate the indirect cooler with a low flow rate of exhaust and high flow rate of supply air even though this condition results in a higher supply air temperature than is optimum. The relatively high delivery of supply air quickly purges the living space or premises of hot air. Once the hot air is purged, the indirect cooler can then be re-set for temperature and air delivery by adjustment of the speeds of the two fans 64, 66.

[0026] Construction of an indirect cooler and its overall size is improved in the Figure 4 embodiment by eliminating the need for a plenum or chamber 44 directing pressurised air from the fan of the prior art cooler to the entrance of the heat exchanger. Elimination of the chamber 44 will also generally improve the uniformity of air flow into the dry passages of the heat exchanger with a subsequent minor improvement in the overall performance of the heat exchanger.

[0027] In the case that the supply air to exhaust air ratio is 1:1 and the same pressure differentials and air flow across the dry and wet passages are required as in the prior art example, the required air pressure in chamber 68 would be -450 Pa to provide the necessary 450 Pa differential across the dry passages. In order to then provide a 150 Pa pressure differential across the wet passages, the static pressure immediately before fan 66 would need to be -450 Pa - 150 Pa = -600 Pa (all pressures relative to ambient atmospheric).

[0028] The power required to be imparted to the air flow by fan 64 at n units of supply airflow will be 450 X n. For fan 66, the power required to be imparted to the wet passage air flow will be 600 x n. The total power to be imparted by both fans is therefore 1,050 x n power units. This is less than the 1,200 x n power units required by the prior art design and therefore the design subject of the current invention provided for an increase in efficiency of the indirect evaporative cooler for the equivalent air flow and cooling.

[0029] In the Figure 5 embodiment, like integers are numbered the same as in Figure 4 and in Figures 1-3. For this version the size of unit is similar to prior art units (Figures 1-3) while its operating efficiency will be slightly higher than for the embodiment of Figure 4.

Claims

1. An indirect evaporative cooler comprising:

- a heat exchanger having alternating wet and dry air flow passages such that, in use, a first air flow passes in a first direction through the dry air flow passages and a secondary air flow, being a portion of the first air flow emerging from the dry passages, travels in counter flow to the first direction and through the wet passages to an exhaust;
- a plenum adapted to receive air emerging from the dry passages and from which a portion of the emerging air is, in use, said secondary air

and the remainder is supply air;
- means of wetting said wet passages;
- a first fan adapted to move air into and through the dry passages;
characterised in that a second fan is located downstream of the exhaust of the wet passages to draw said secondary air through the wet passages.

2. A cooler as claimed in Claim 1, wherein the first fan is located upstream of the dry passages.

3. A cooler as claimed in claim 1, wherein the first fan is located downstream of the dry passages.

4. A cooler as claimed in any one of the preceding claims wherein at least one of the fans is speed controlled.

5. A cooler as claimed in claim 4 wherein both fans are speed controlled.

6. A method of indirect evaporative cooling wherein a lower than ambient pressure is applied to exhaust outlets of wet passages of a heat exchanger having alternating wet and dry air flow passages and comprising passing a first air flow through the dry passages before drawing off a portion thereof as secondary air through the wet passages by said lower than ambient pressure.

7. A method as claimed in claim 6 wherein the lower than ambient pressure is variable.

8. A method as claimed in claim 6 or 7, wherein the first air flow rate is variable.

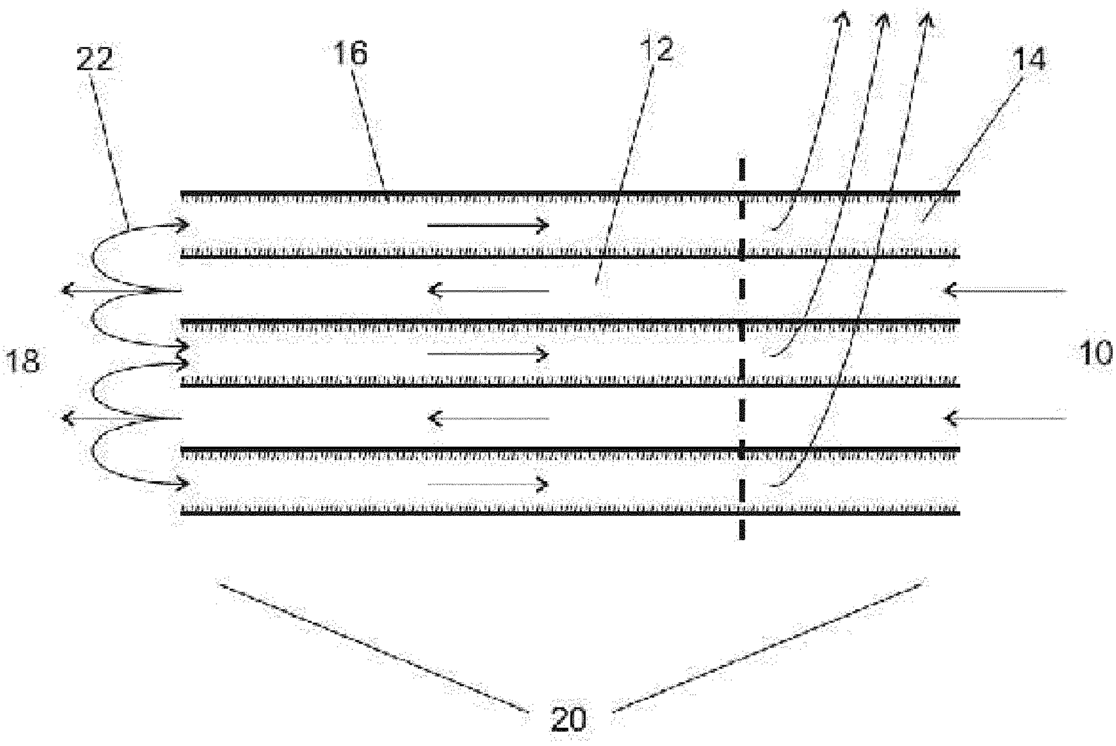

FIGURE 1

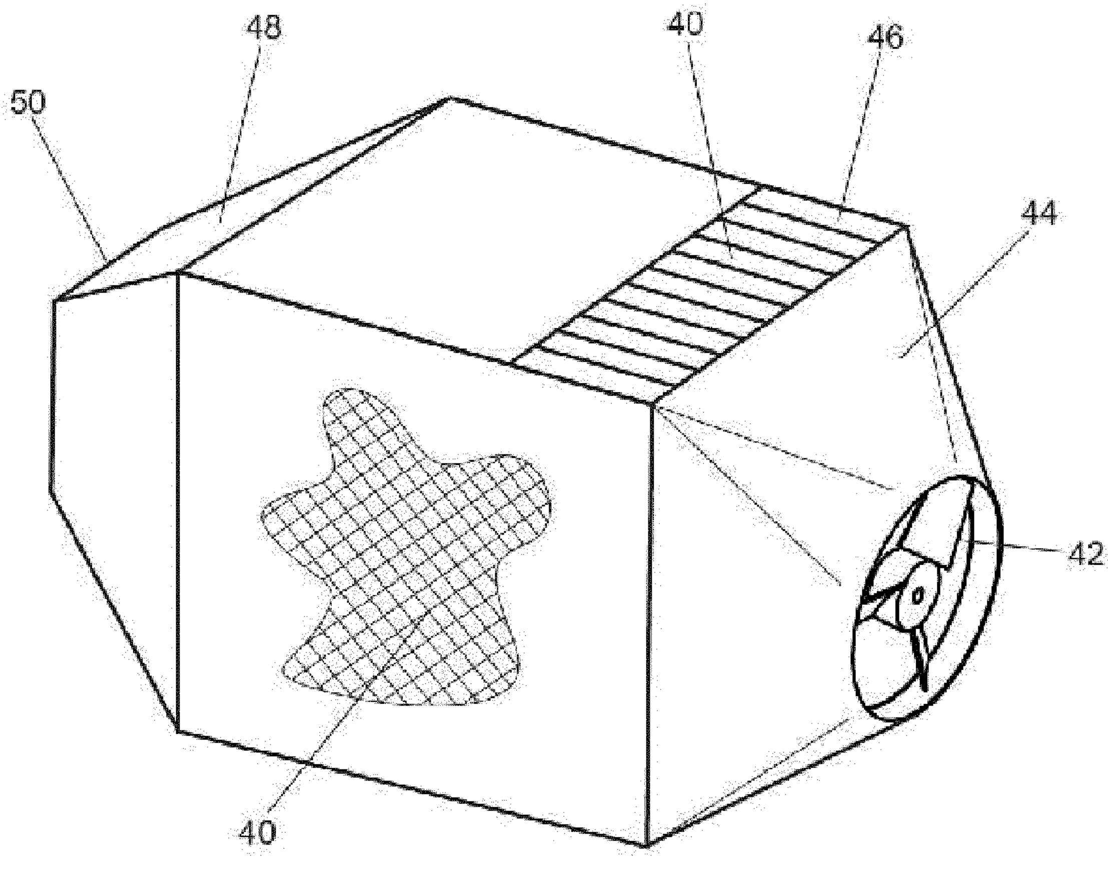

FIGURE 2

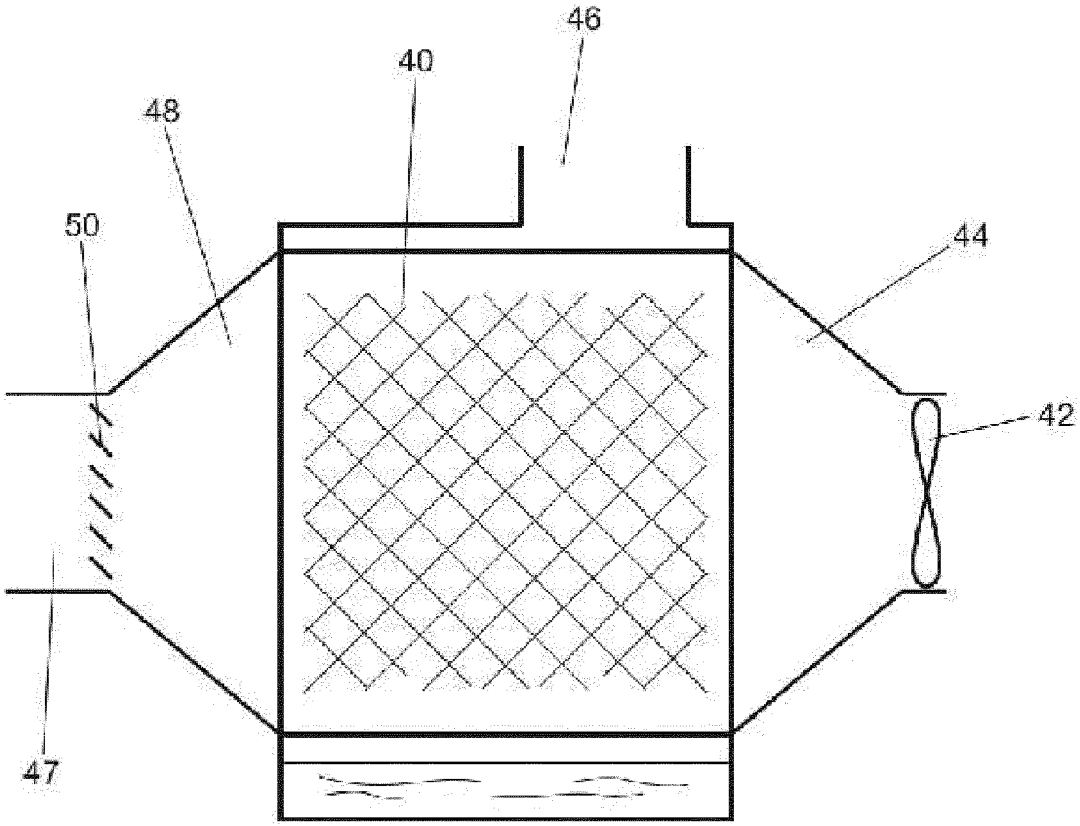

FIGURE 3

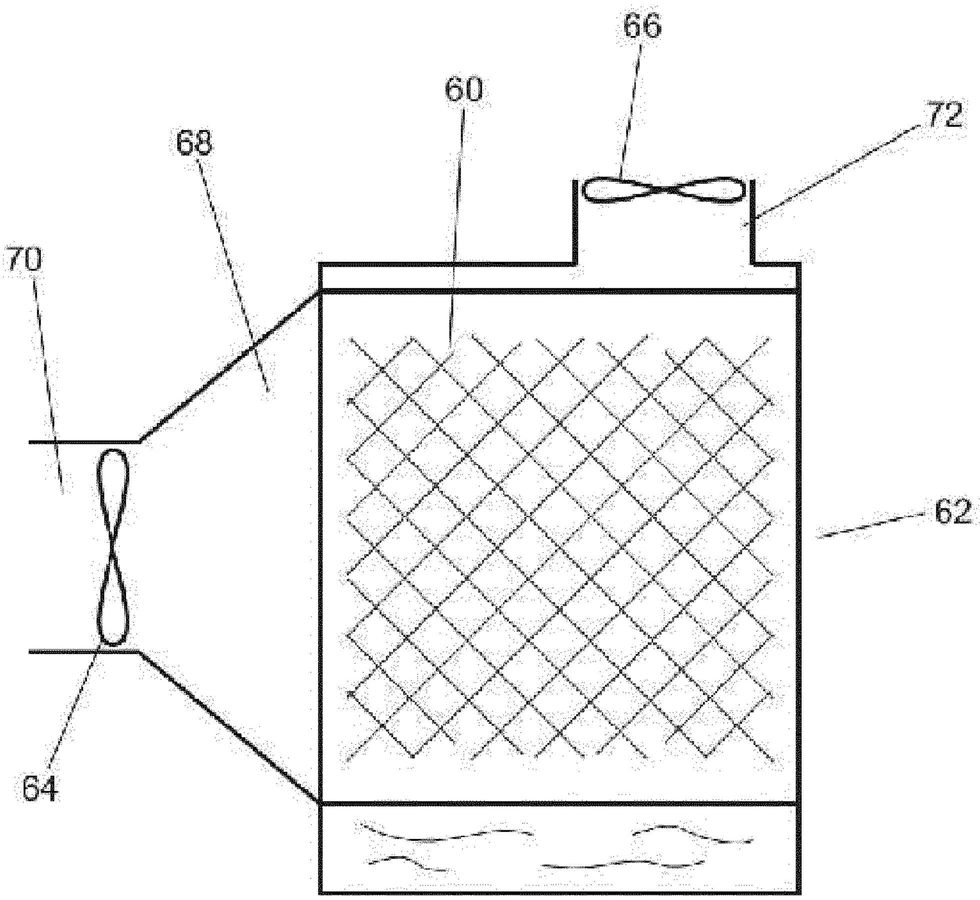

FIGURE 4

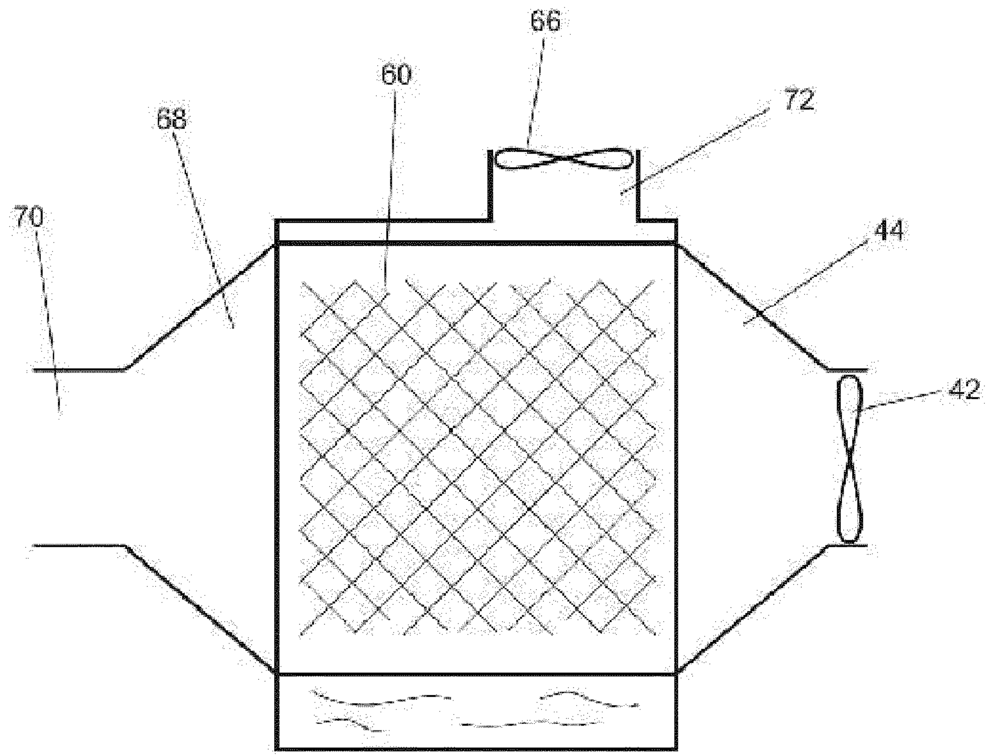

FIGURE 5

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EUROPEAN SEARCH REPORT

Application Number
EP 13 27 5152

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F28D
F24F
F28C

The present search report has been drawn up for all claims

2

Place of search	Date of completion of the search	Examiner
The Hague	15 November 2013	Berkus, Frank

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

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