United States Patent [19]

Laing

[11] **3,877,515**

[45] **Apr. 15, 1975**

[54] **TEMPERATURE-CONTROL SYSTEM WITH ROTARY HEAT EXCHANGERS**

[76] Inventor: **Nikolaus Laing,** Hofener Weg 35-37, 7141 Aldingen, Germany

[22] Filed: **Sept. 5, 1972**

[21] Appl. No.: **286,569**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 847,771, June 17, 1969, abandoned.

[52] **U.S. Cl.** **165/86;** 62/499; 165/125; 165/107

[51] **Int. Cl.** **F28f 5/00;** F28d 11/00

[58] **Field of Search** 165/122, 125; 62/499

[56] **References Cited**

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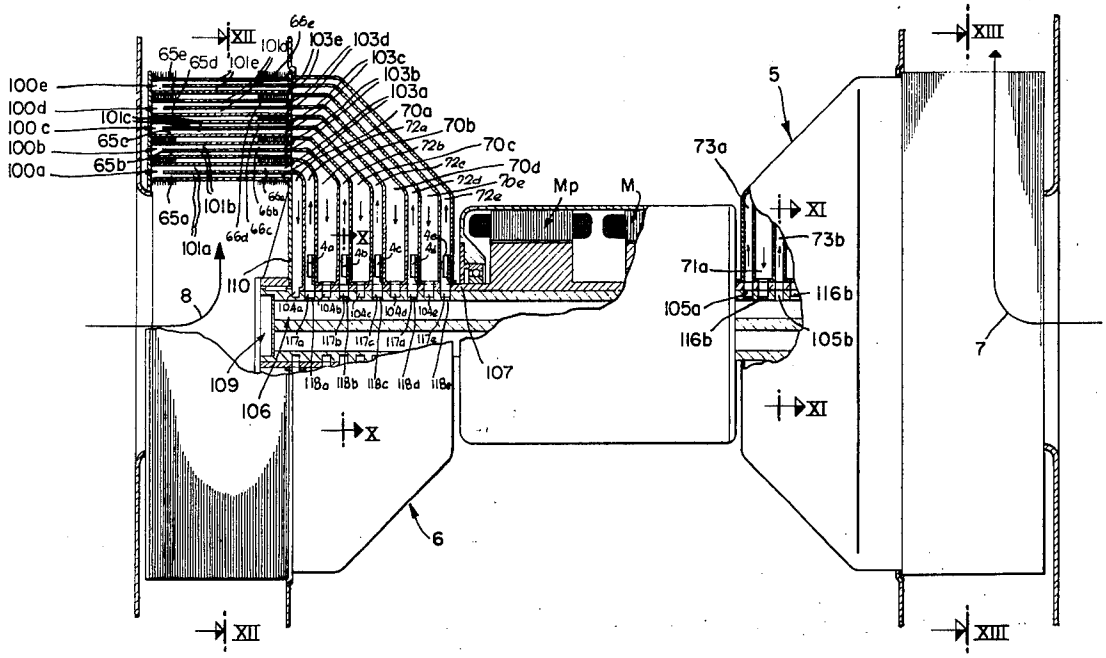
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Primary Examiner—Charles J. Myhre
Assistant Examiner—Theophil W. Streule
Attorney, Agent, or Firm—Karl F. Ross; Herbert Dubno

[57] **ABSTRACT**

An independently heated or cooled room is linked with an environment of different temperature through a fluid-circulation system including a closed path traversed by a heat-carrier medium. The medium travels between two heat exchangers, respectively located in the warmer and in the colder air, under a pressure differential low enough to maintain a coherent flow at substantially uniform speed in one or more pairs of channels extending between the two heat exchangers.

9 Claims, 21 Drawing Figures



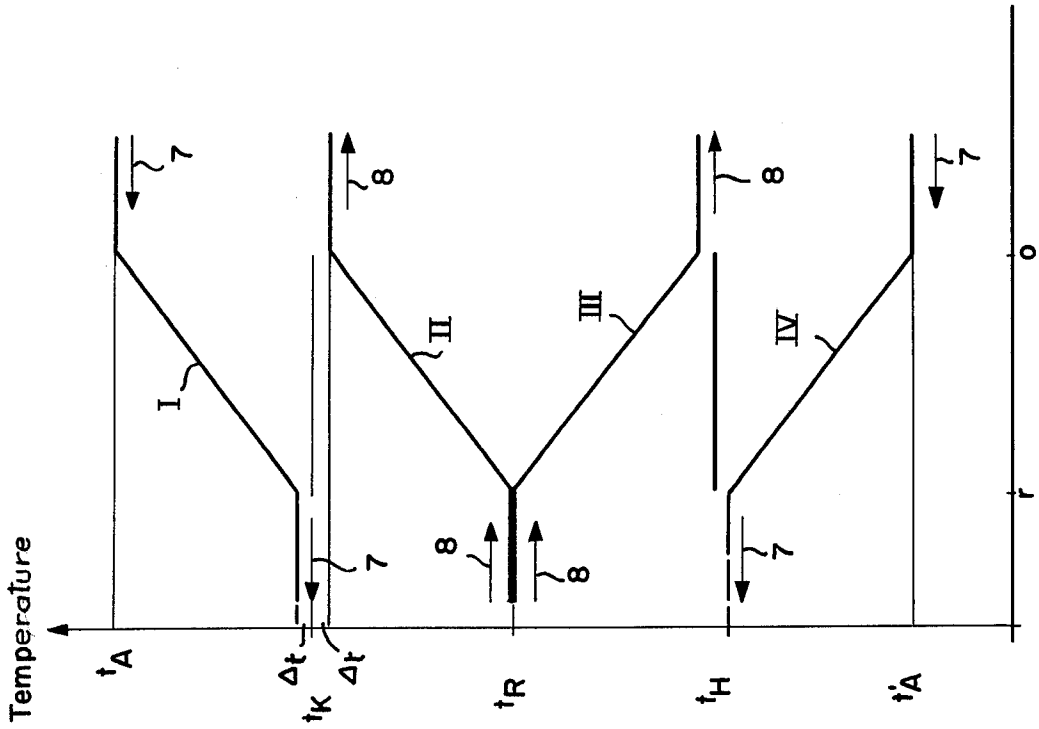

Temperature
t_A
Δt
Δt
t_K
t_R
t_H
t'_A
I
II
III
IV
7
8
o
r

FIG. 2

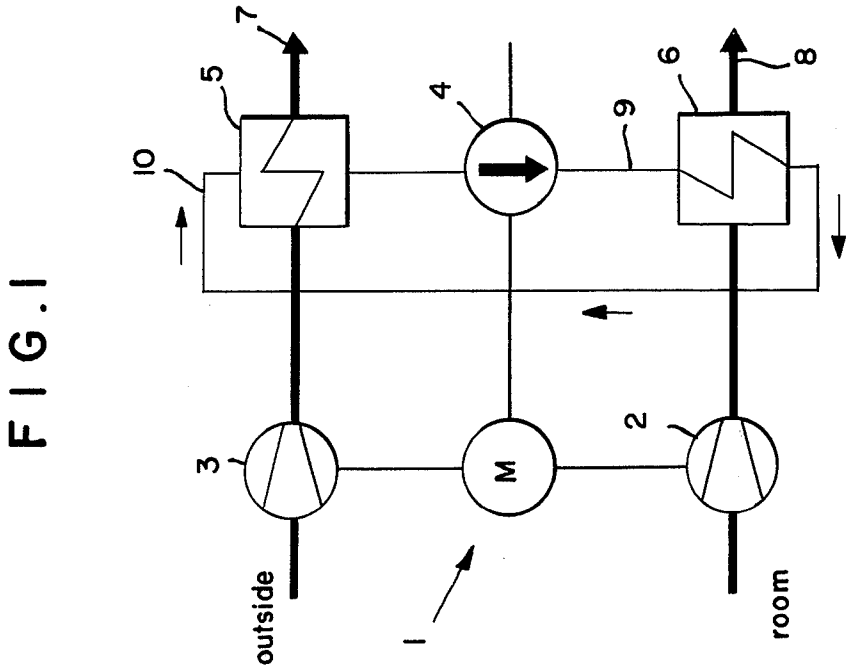

outside
room
M
1
2
3
4
5
6
7
8
9
10

FIG. 1

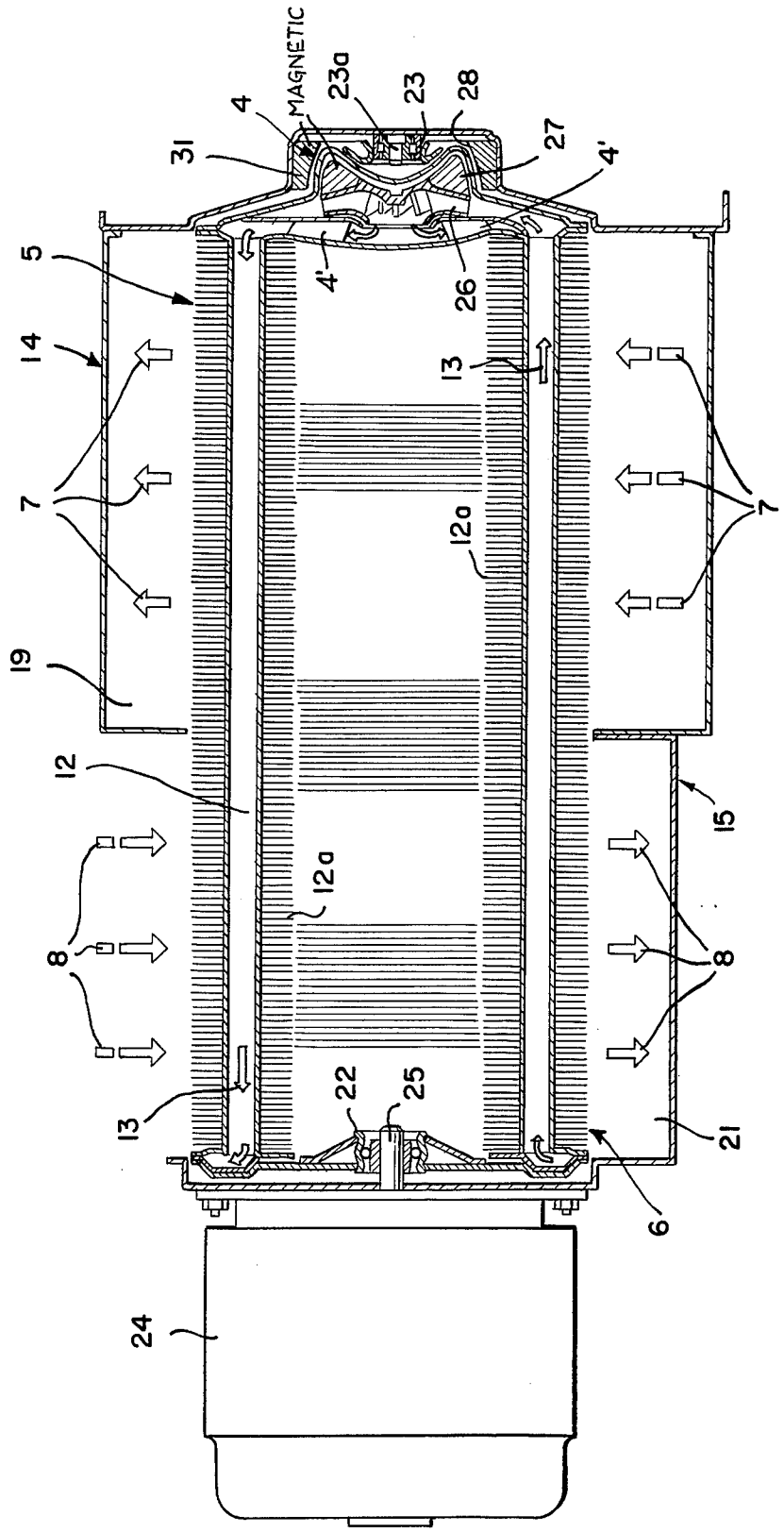


FIG. 3

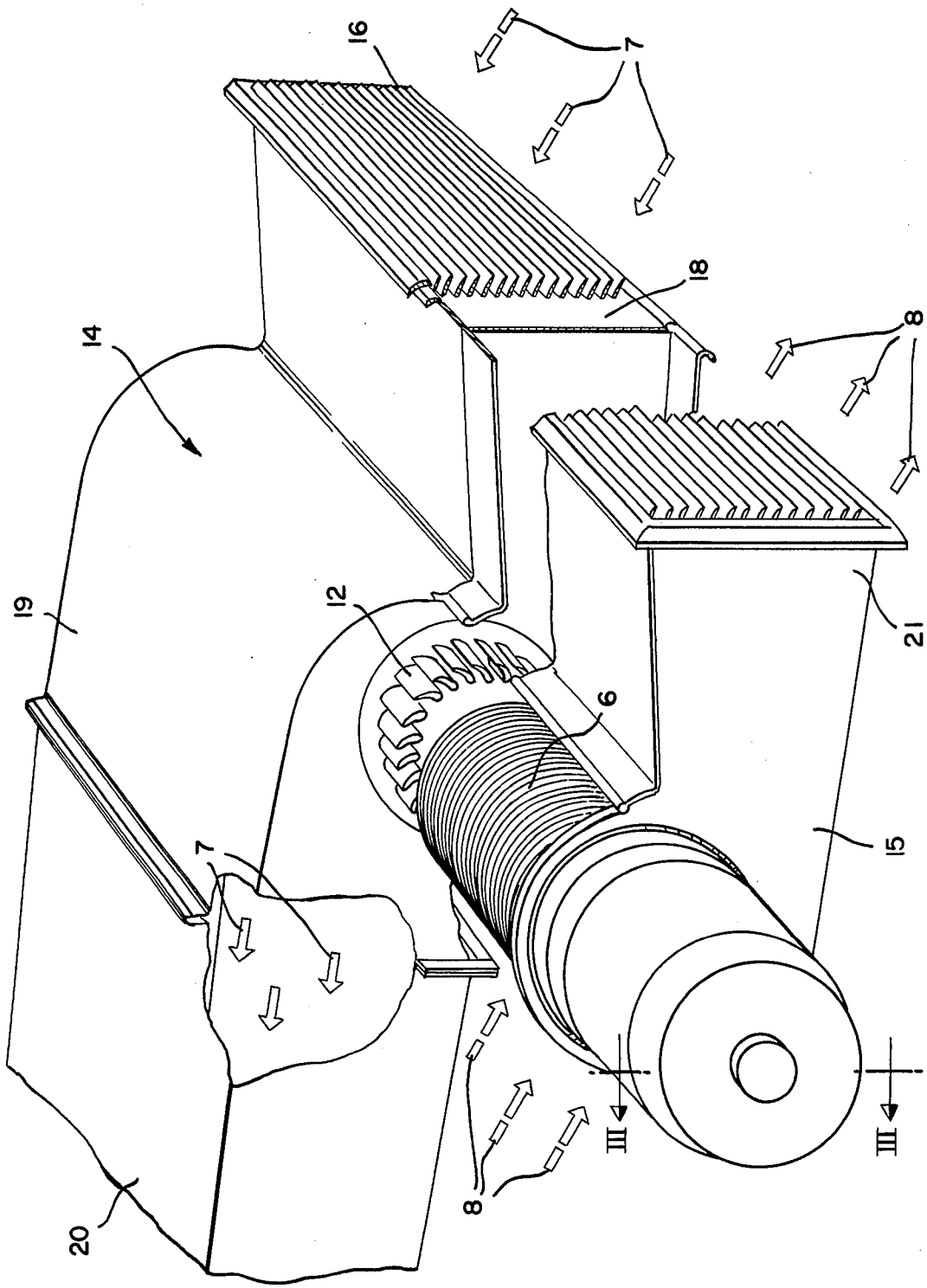


FIG. 4

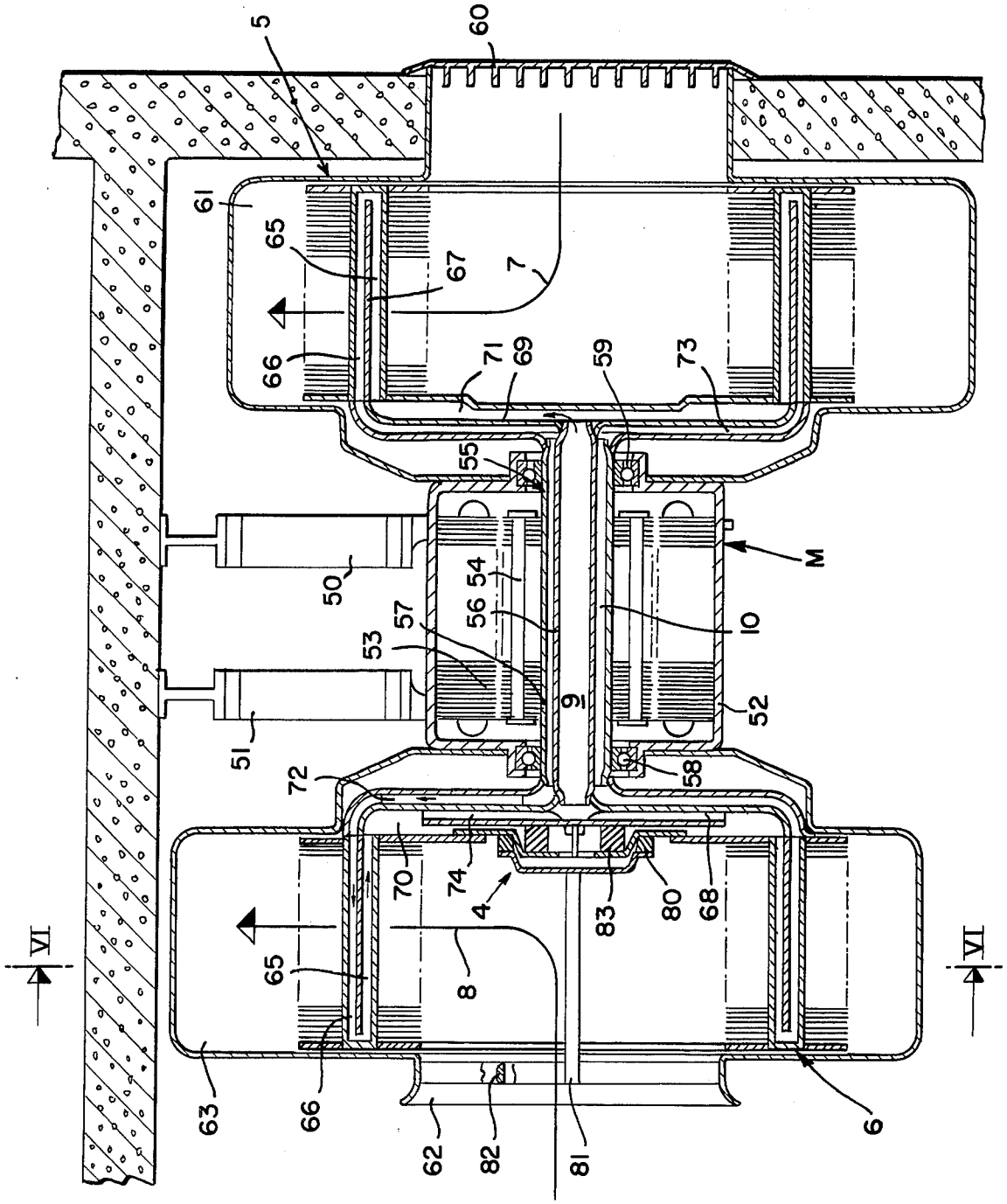

5
60
61
65
67
7
66
71
69
59
73
55
M
50
54
56
57
10
53
9
52
51
72
58
70
74
4
83
80
68
VI
VI
65
8
63
66
62
82
81
6

FIG. 5

FIG. 6

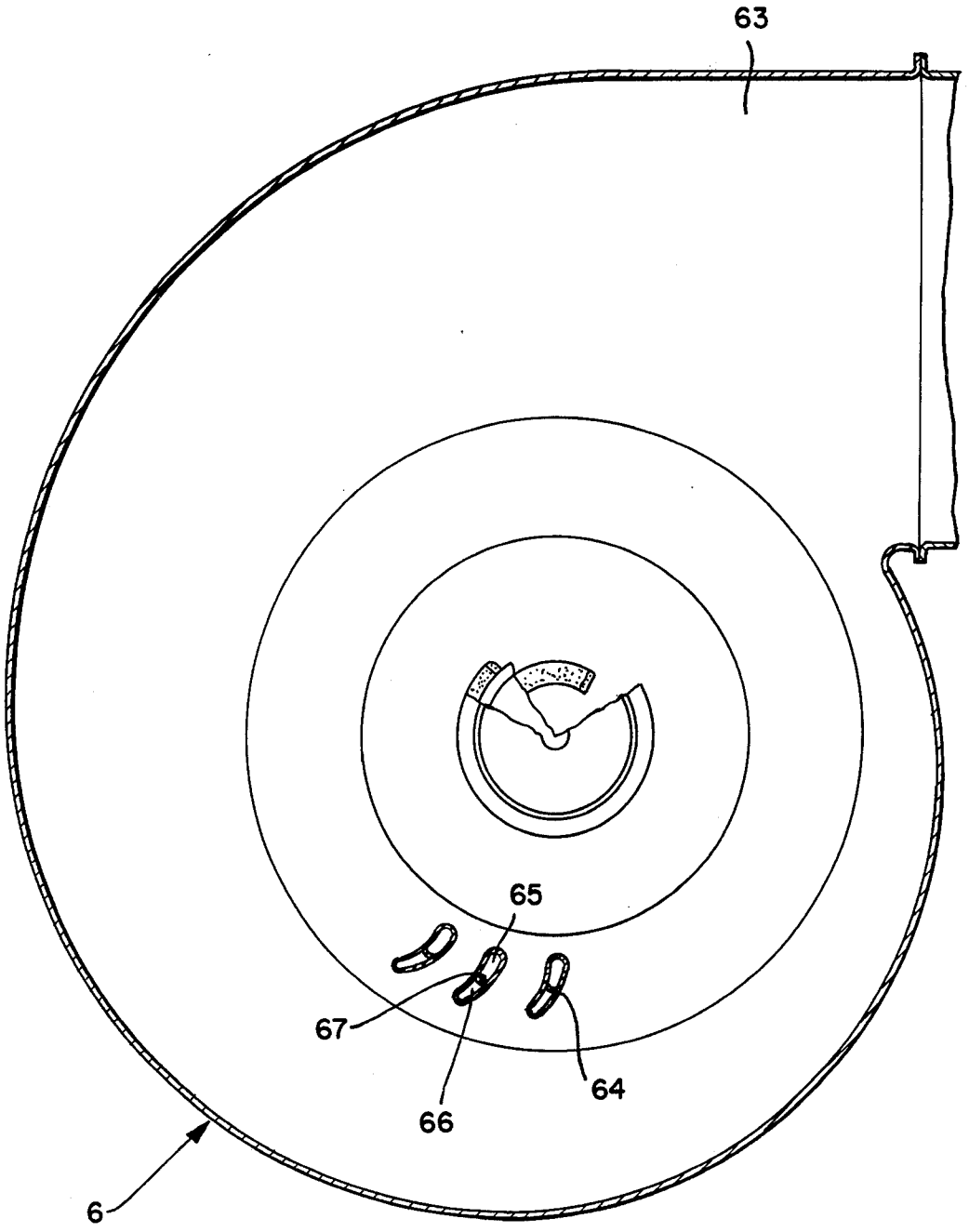


FIG. 8

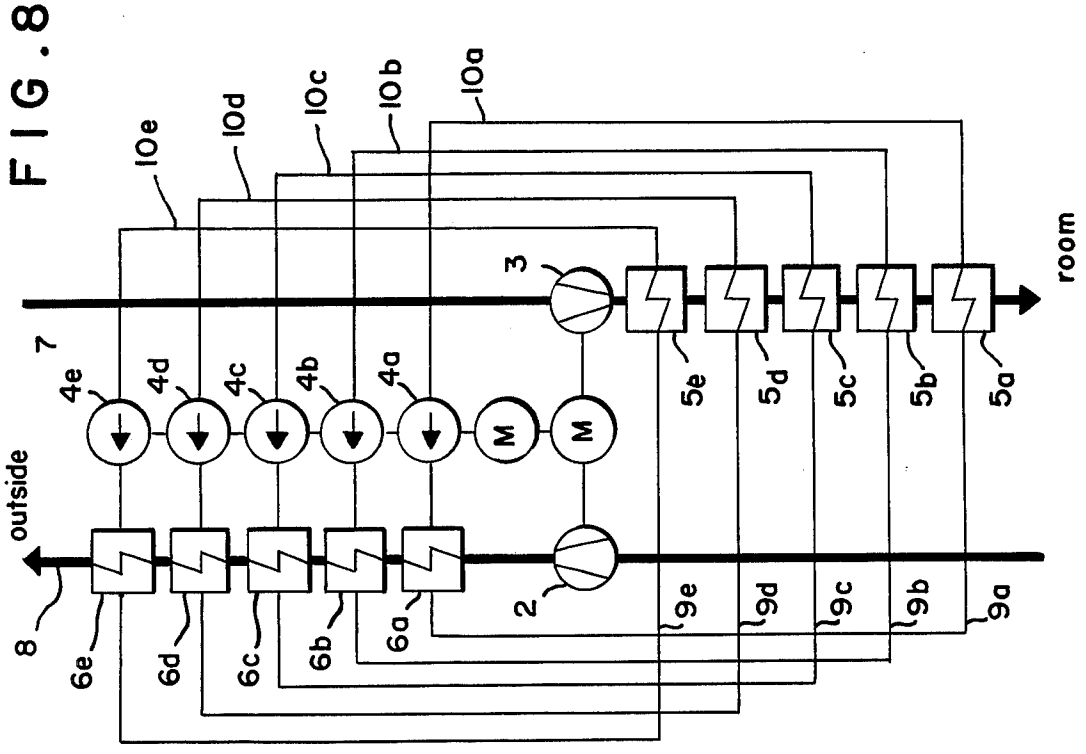
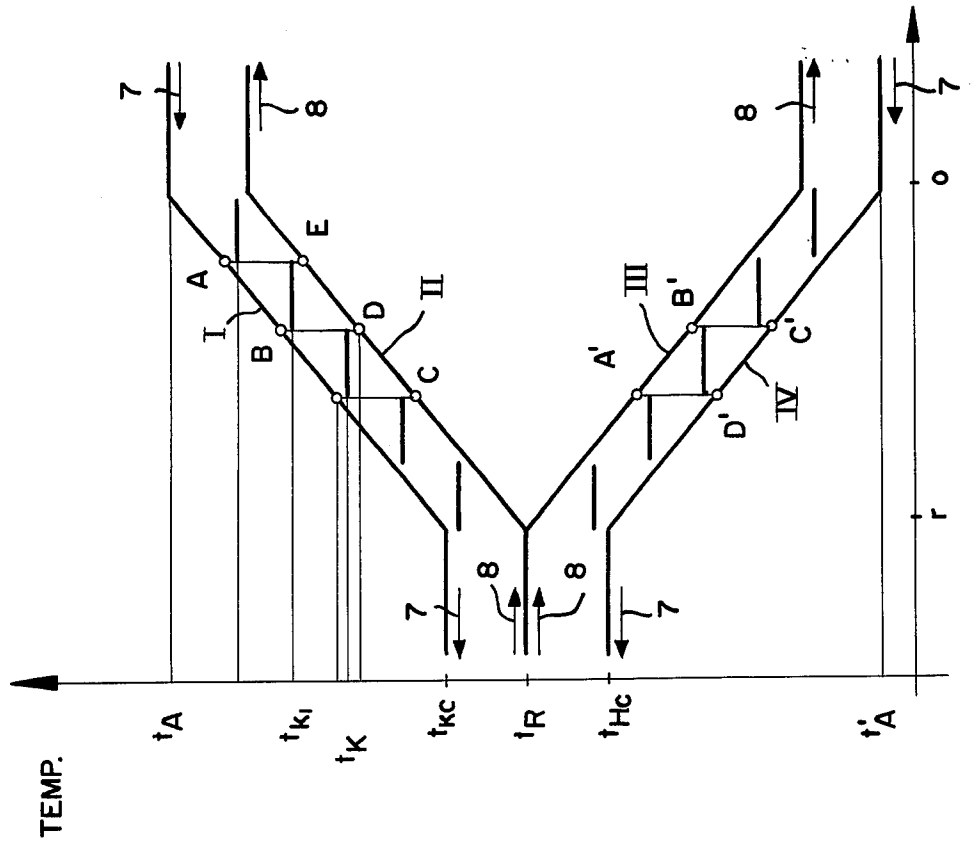

10e
10d
10c
10b
10a
3
7
4e
4d
4c
4b
4a
M
M
5e
5d
5c
5b
5a
room
outside
8
6e
6d
6c
6b
6a
2
9e
9d
9c
9b
9a

FIG. 7


7
8
A
E
I
B
D
II
C
III
B'
A'
C'
IV
D'
8
7
8
8
7
o
r
TEMP.
t_A
t_{K_I}
t_K
t_{KC}
t_R
t_{HC}
t'_A

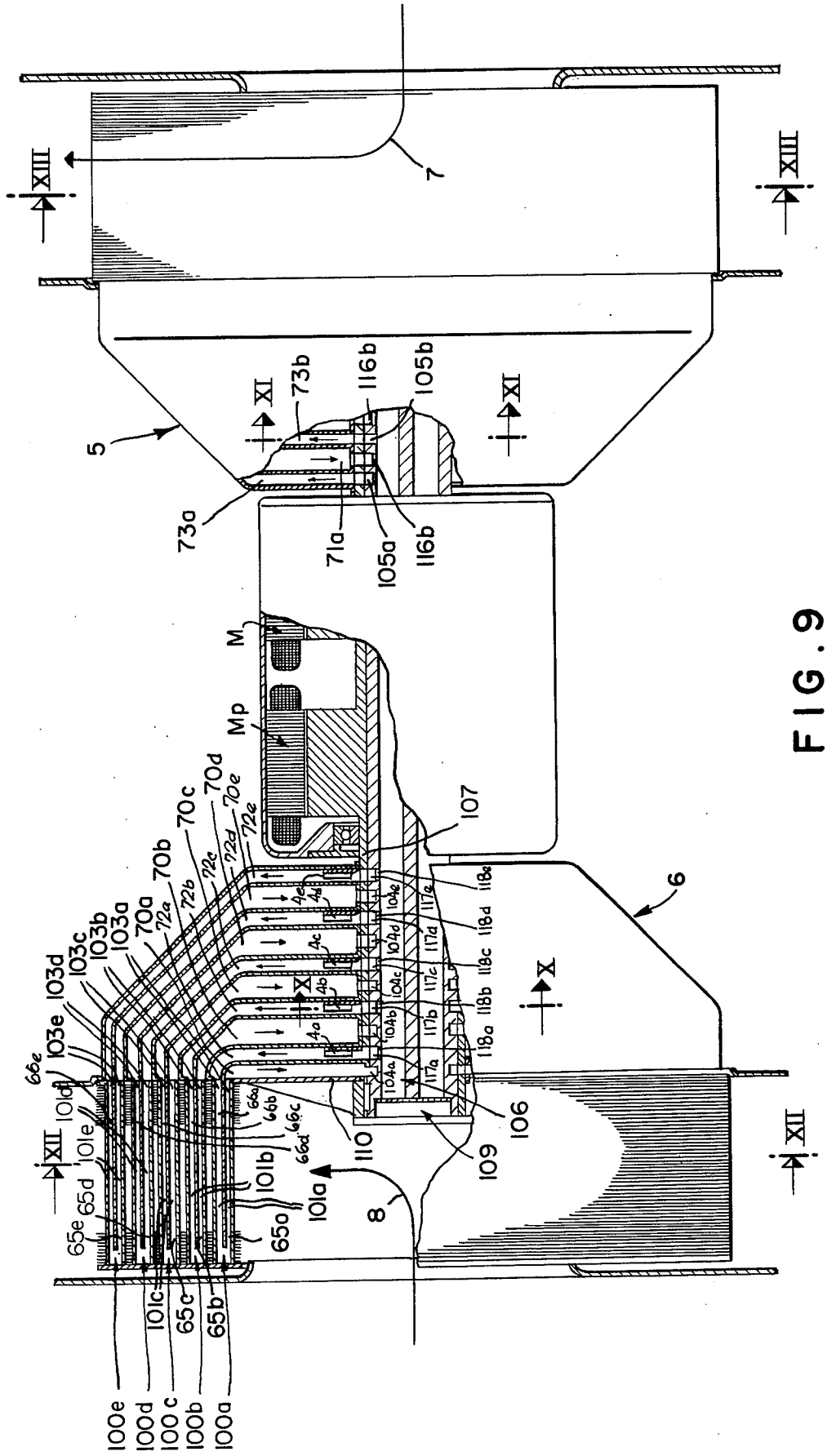
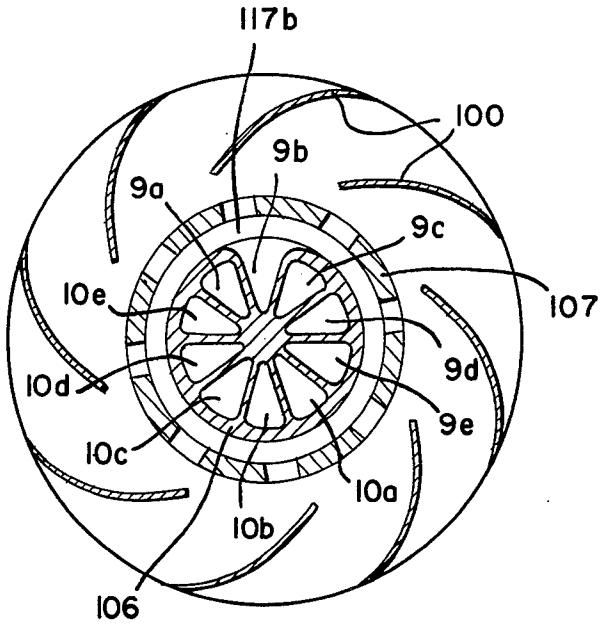

7
XIII
XIII
XI
XI
73b
116b
105b
5
73a
71a
105a
116b
M
Mp
107
6
X
X
XII
XII
110
109
106
8
101a
101b
101c
101d
101e
65a
65b
65c
65d
65e
100a
100b
100c
100d
100e

FIG. 9

FIG. 10



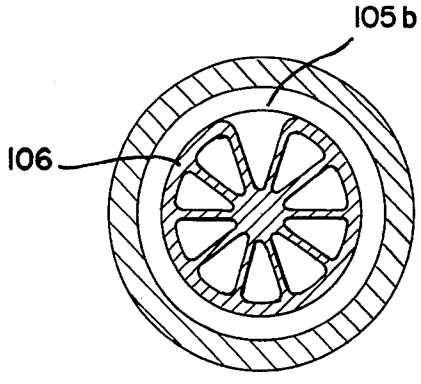


FIG. 11

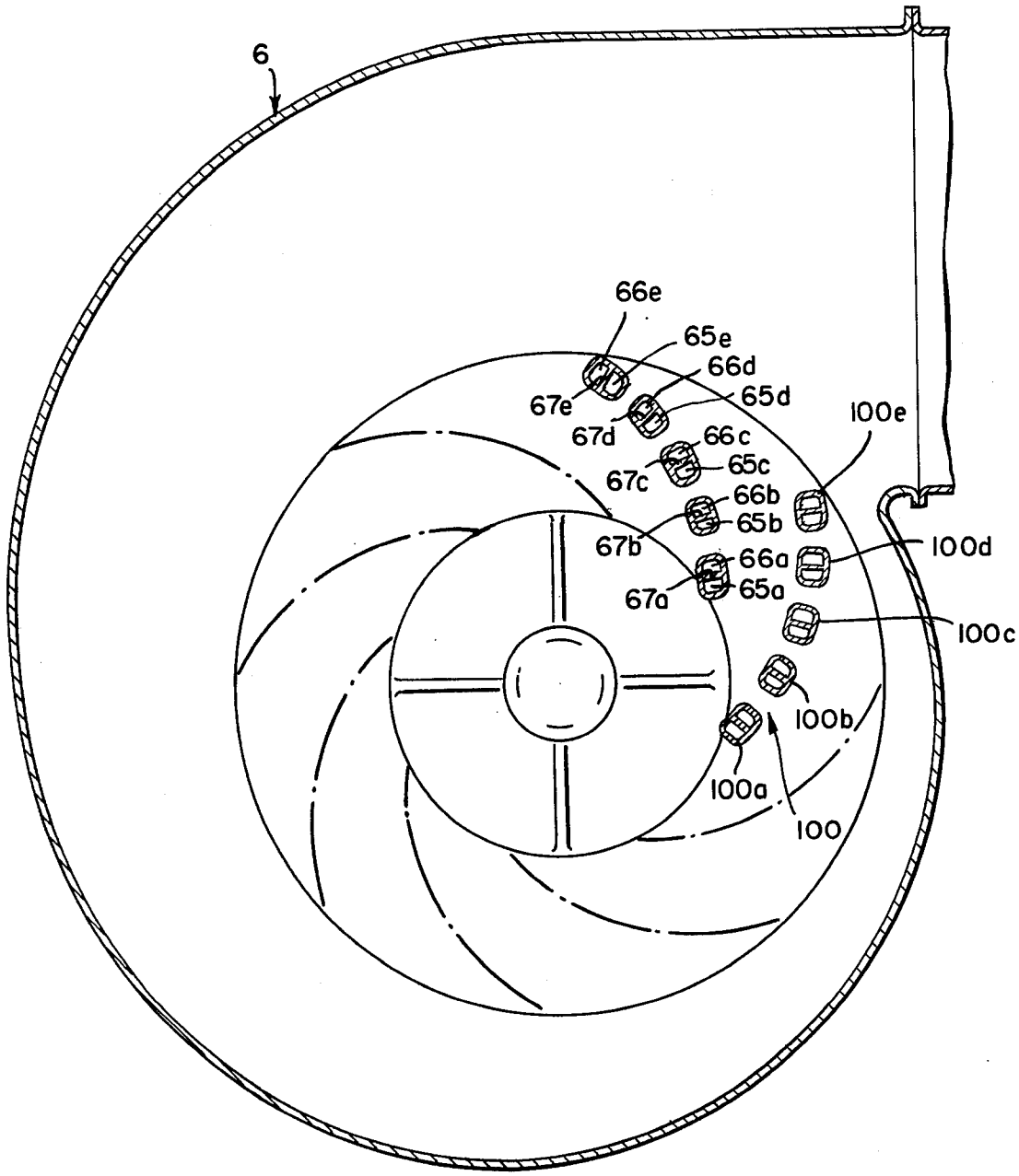

6
66e
65e
66d
65d
67e
67d
66c
65c
67c
66b
65b
67b
66a
65a
67a
100e
100d
100c
100b
100a
100

FIG. 12

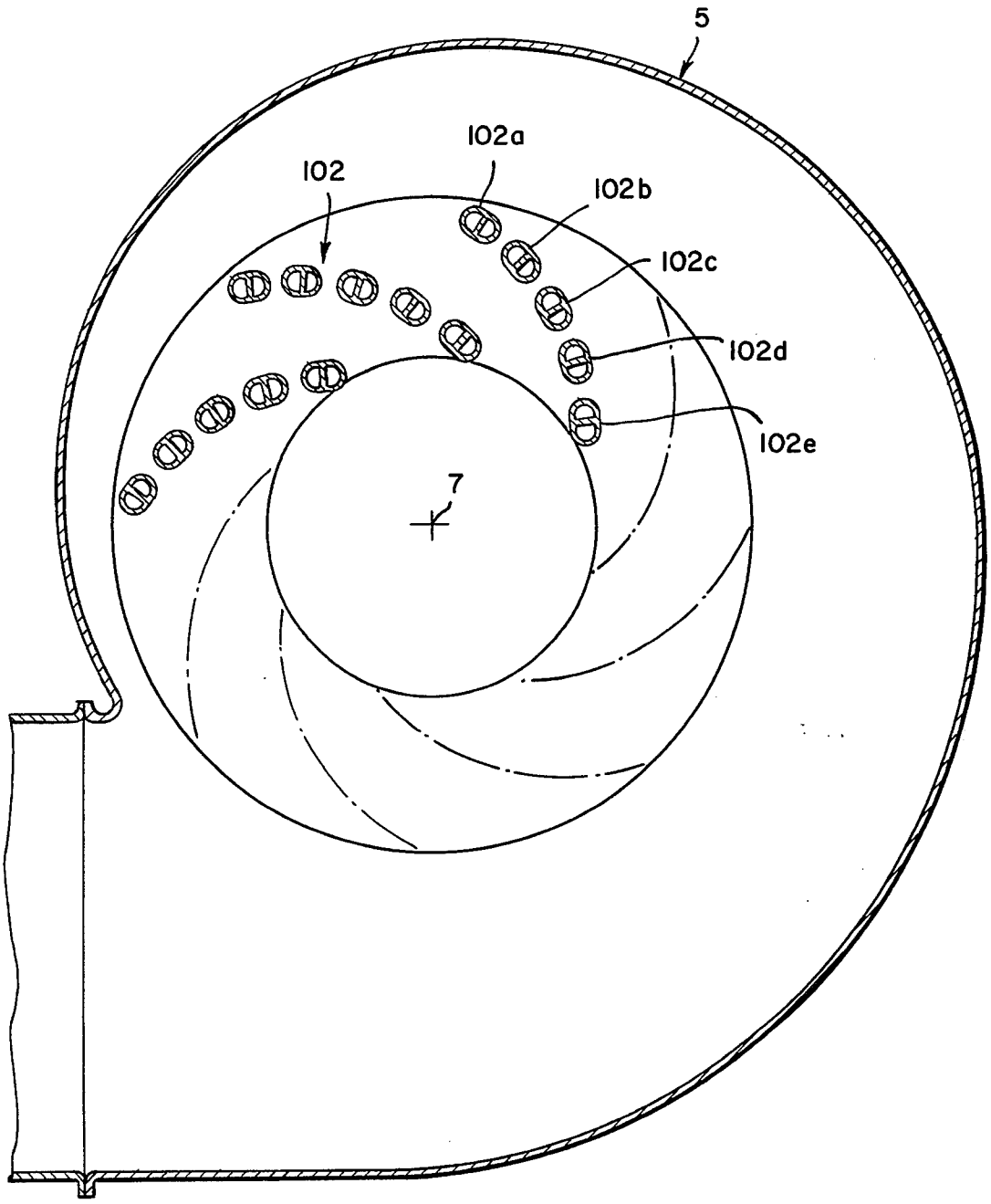


FIG. 13

FIG. 14

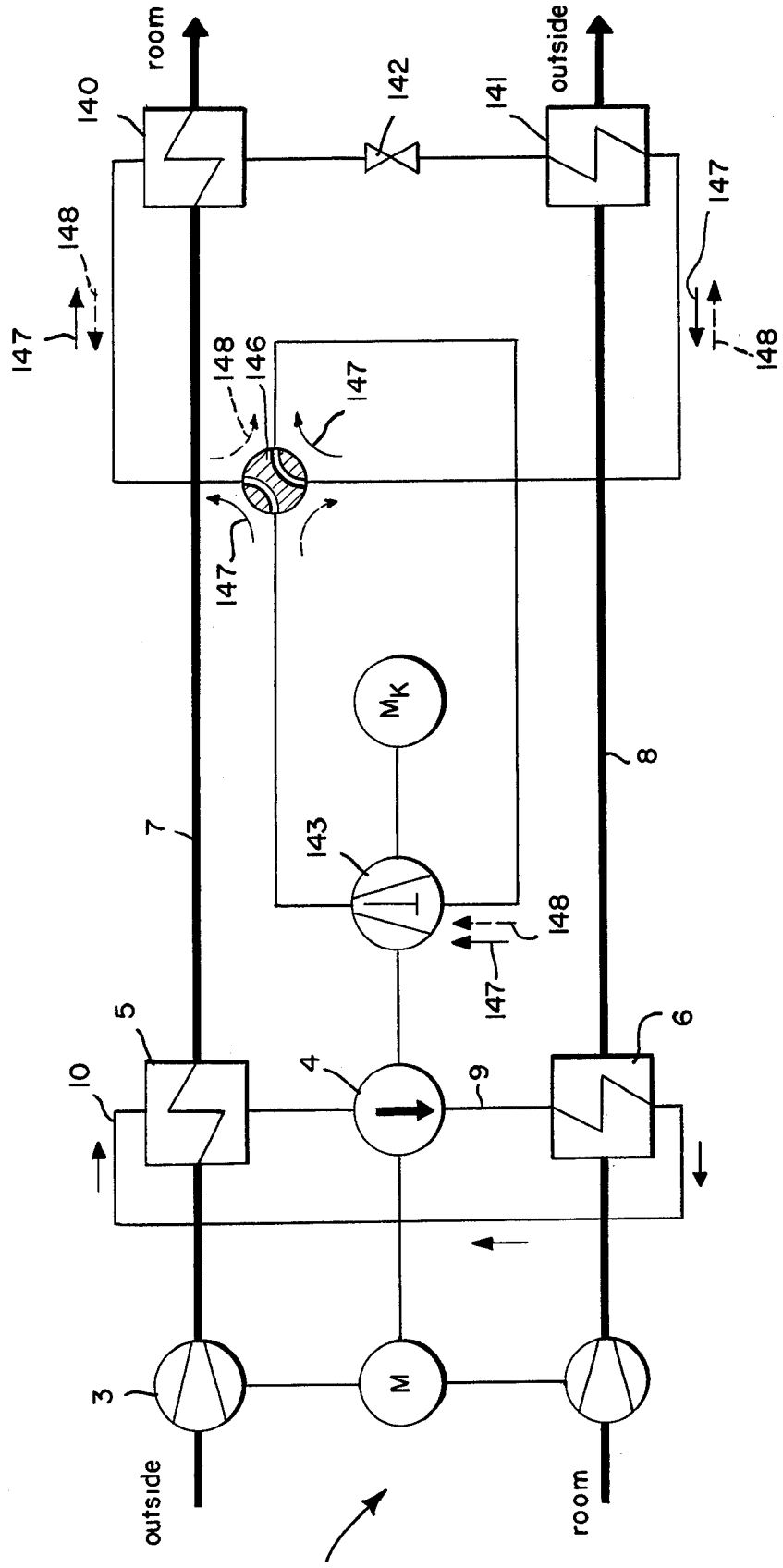
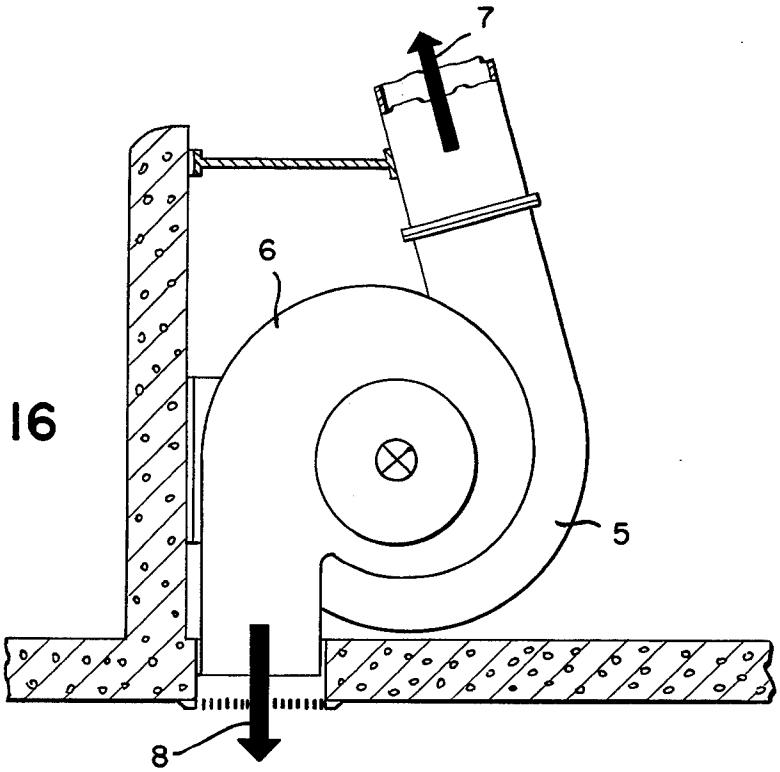

room
outside
140
142
141
147
148
146
143
Mk
M
8
7
5
6
4
9
10
3
outside
room

FIG.16



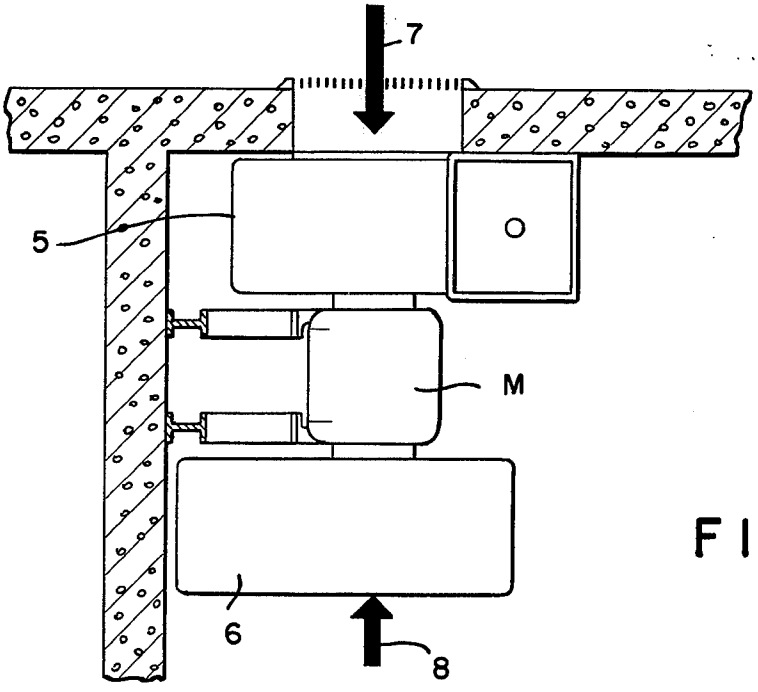
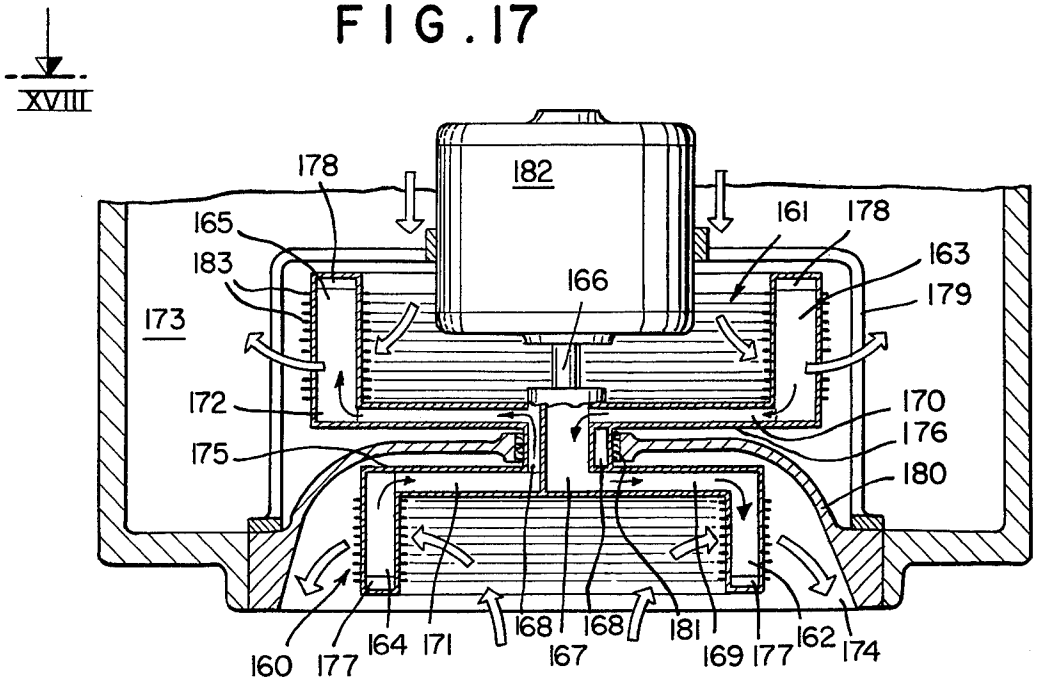


FIG.15

FIG. 17


XVIII
XVIII
178
182
165
161
178
183
163
166
179
173
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170
175
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180
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174

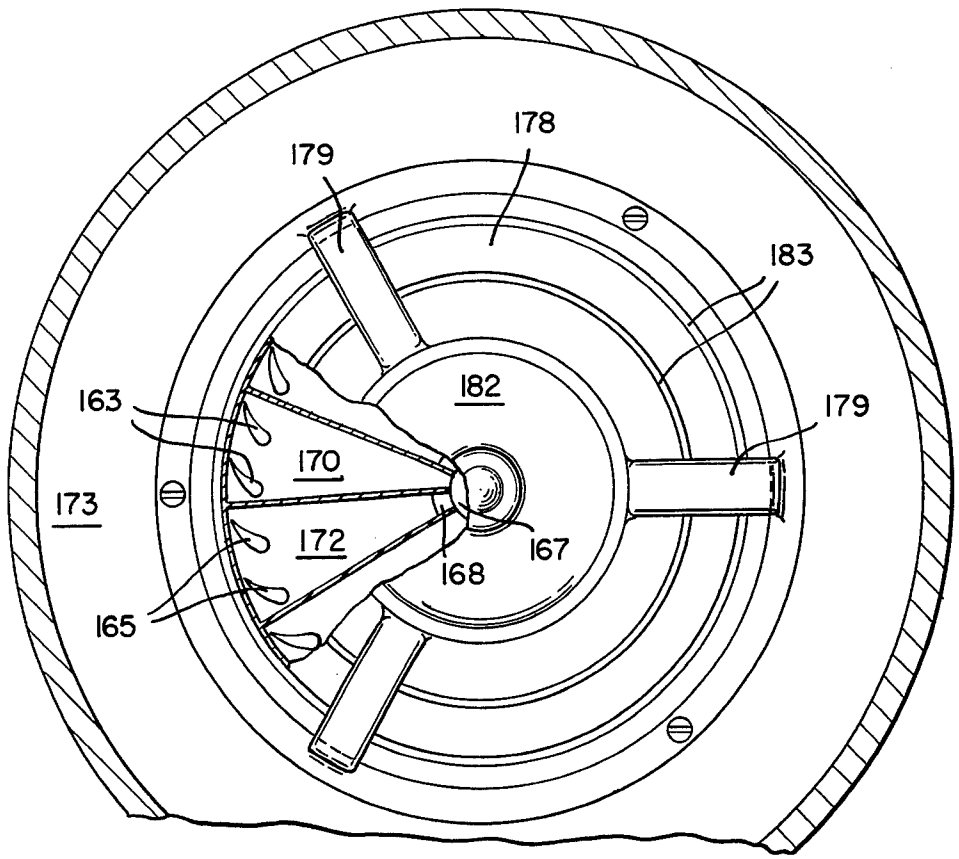
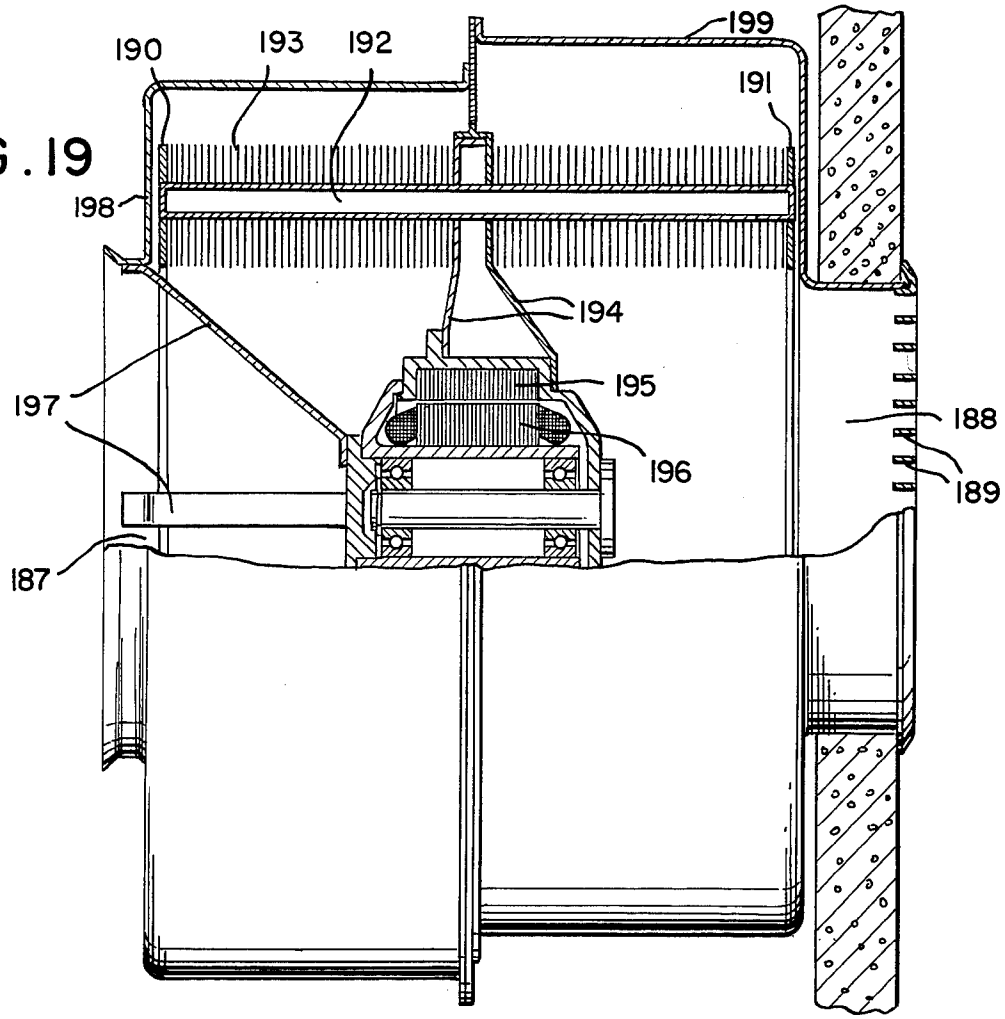

178
179
183
182
163
179
170
173
172
167
168
165

FIG. 18

FIG. 19


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193
192
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188
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189
187

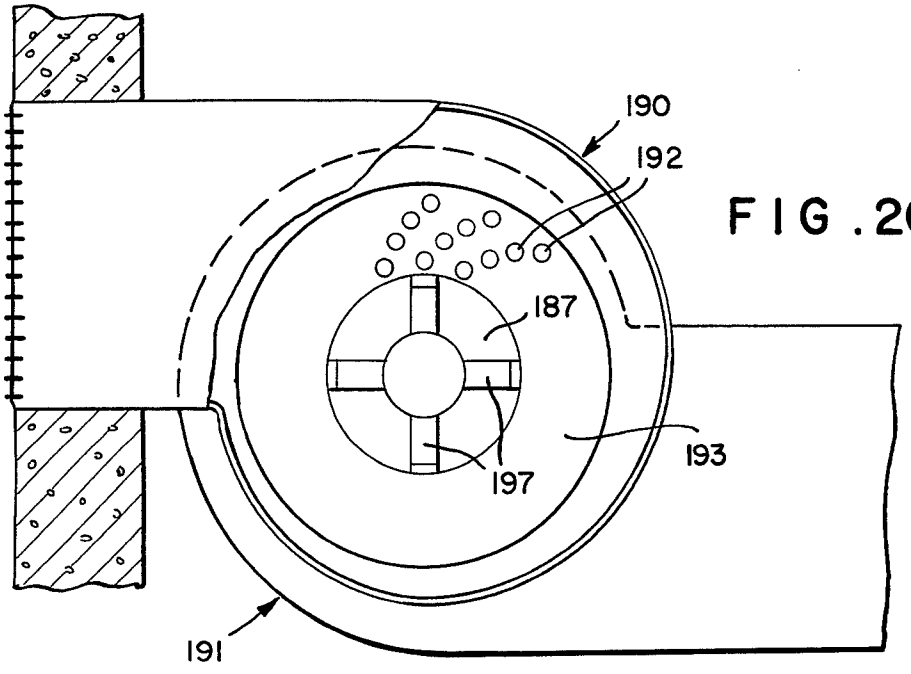
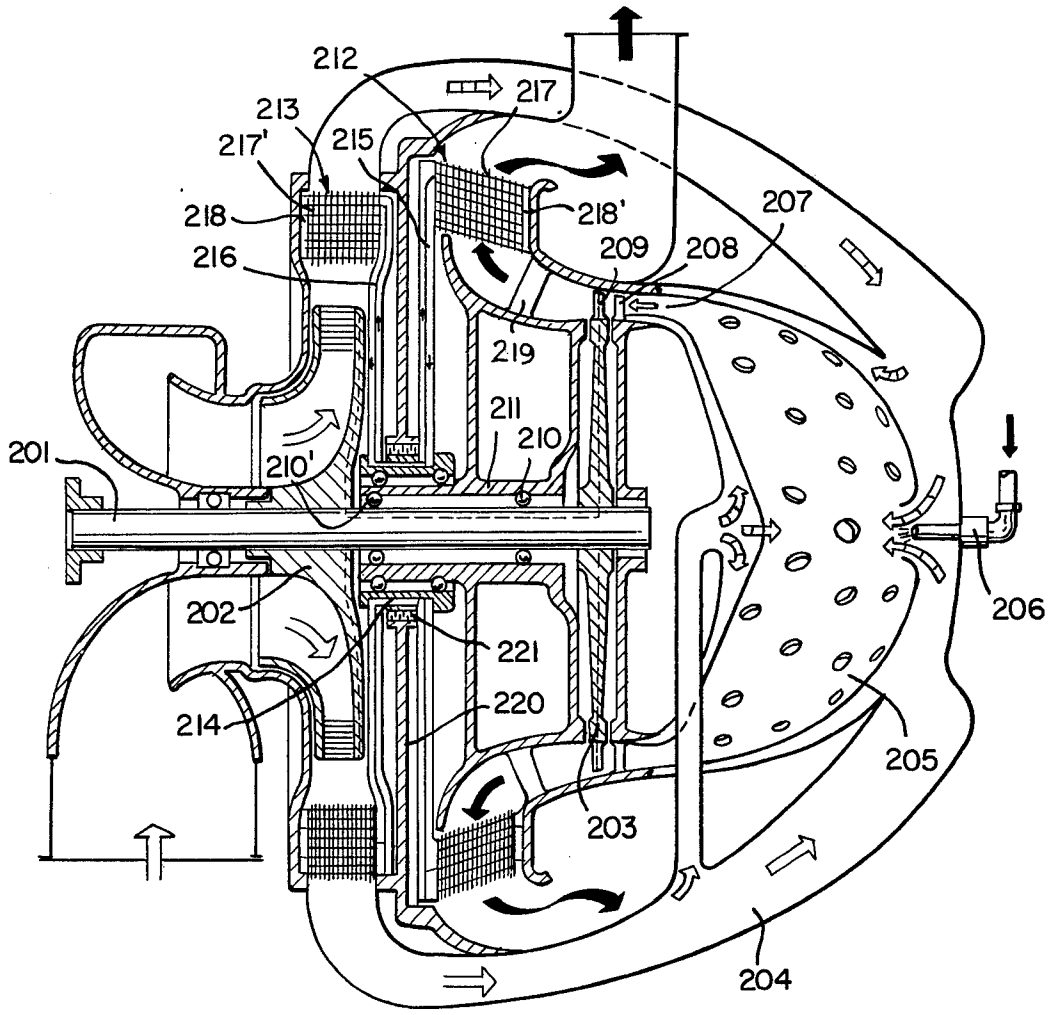

190
192
187
193
197
191

FIG. 20

FIG. 21


212
217
213
217'
215
218
218'
207
216
209
208
219
211
210
201
210'
202
221
220
214
206
205
203
204

TEMPERATURE-CONTROL SYSTEM WITH ROTARY HEAT EXCHANGERS

This application is a continuation-in-part of my copending application Ser. No. 847,771 filed June 17, 1969 and now abandoned. The latter application partly discloses subject matter of other applications filed by me earlier and now abandoned, i.e.: Ser. No. 105,056 filed April 24, 1961, included in Ser. No. 452,622 filed May 1965, included in turn in Ser. No. **562,880** filed July 5, 1966 and its continuation Ser. No. 802,704 filed Dec. 11, 1968; and Ser. No. 427,323 filed Jan. 22, 1965, with continuation Ser. No. 566,088 filed July 18, 1966 (now U.S. Pat. No. 3,347,059) and further continuation Ser. No. 585,819 filed Oct. **11, 1966,** also since abandoned. This application also includes subject matter disclosed in my pending application Ser. No. 84,097 filed Oct. 26, 1970 now U.S. Pat. No. 3,811,495 granted May 21, 1974.

My present invention relates to a system for controlling the temperature of a space subjected to extraneous heating or cooling. Such a system has utility, for example, in a heated room into which colder outside air is admitted continuously or intermittently for ventilating purposes or, conversely, in the cooling of the interior of a housing for electronic equipment or the like generating excess heat which must be dissipated.

The general object of my invention is to provide simple and effective means for maintaining a desired thermal equilibrium between two bodies of air or other fluids of different temperature, i.e., a first fluid in the space to be thermally controlled and a second fluid in an adjoining environment.

A more particular object is to provide means for increasing the efficiency of existing heating or air-conditioning equipment in a room or building by using available excess heating or cooling power to preheat or precool fresh air entering from the outside instead of wastefully discharging a like quantity of heated or cooled internal air directly into the atmosphere.

These objects are realized, in accordance with my present invention, by the provision of a first heat exchanger exposed to the fluid in the controlled space and a second heat exchanger exposed to the fluid in the adjoining environment, the two heat exchangers being interlinked by sealed conduit means forming a closed path for a heat-carrier medium (referred to hereinafter simply as "carrier") abstracting heat from the warmer fluid and giving off heat to the colder one. In contradistinction to conventional refrigeration systems in which the circulating medium is alternately compressed and expanded while moving back and forth between a heat sink and a heat source, the carrier in a system according to my invention is subjected to a pressure differential only sufficient to drive it through the circuit with a coherent flow and at substantially constant speed in each of the stretches extending between the two heat exchangers. These stretches, therefore, may be constituted by tubes or other channels of more or less constant cross-section, e.g., axial passages in a shaft supporting a pair of rotors respectively forming part of the two coaxially mounted heat exchangers.

This substantial constancy of the flow rate in each stretch is maintained even if the carrier undergoes a change of its state of aggregation on passing through the heat exchangers. Thus, the invention can be realized with a carrier which remains either gaseous or liquid (preferably the latter) throughout its path or which is alternately vaporized and condensed by the absorption and release of heat at the two terminal points of the circuit.

For use at substantially ambient temperatures, the carrier could be ethyl or methyl alcohol, or freon; at elevated temperatures, sodium or a eutectic mixture of sodium and potassium is preferred.

The circulation of the carrier along its closed path may be effected by various means, e.g., by relative rotation of a set of vanes and a surrounding duct in one of the heat exchangers. In order to insure absolute fluid-tightness, the vanes may be entrained (or arrested with reference to the rotating duct) by magnetic force, e.g., as disclosed in commonly owned U.S. application Ser. No. 42,018 filed June 1, 1970, now U.S. Pat. No. 3,696,634. In the case of an electrically conductive carrier it is also possible to induce the desired relative motion by eddy currents. Finally, a thermosiphon effect may be utilized by letting the carrier circulate in two heat-exchanger rotors of different effective diameters through radial extensions of the axial channels, with the colder and therefore specifically heavier portion of the carrier moving outwardly over a larger radius than the warmer portion.

According to a further feature of my invention, each heat exchanger (or at least one of them) may be subdivided into a plurality of cascaded heat-exchange stages successively traversed by the carrier in counterflow to the external fluid, i.e., with a variation in carrier temperature generally paralleling the change in the temperature of that fluid. Thus, in a heating cycle, the hottest portion of the carrier traverses the stage last encountered by the external flow which has therefore already been preheated by the other units to nearly the same temperature; in a cooling cycle, of course, the relationship is reversed. In either case, the cascaded stages are advantageously constituted by a plurality of interconnected branch channels in the heat-exchanger rotor communicating with respective channel pairs in the shaft, these channel pairs being formed by peripherally offset sectoral passages.

The motive power for driving the rotors of the heat exchangers may be obtained either from a separate motor or from one of the fluids involved in the exchange, the latter in particular where the system is used for abstracting heat from high-pressure gases or vapors issuing from a combustion chamber or a boiler.

According to another feature of my invention, an especially efficient heat transfer is achieved if the carrier passes through axially extending tubes forming part of the rotor body, these tubes being provided with transverse fins which are sufficiently closely spaced to develop a shear force which entrains the surrounding fluid in a generally radial direction so as to create an outward flow at a rate determined by the rotor speed. I have found that it is highly advantageous to dispose the tubes substantially along segments of an archimedean spiral whose pitch should be so correlated with the rotor speed, or more generally with the flow rate, that the spiral array of tubes exerts substantially no reaction (in either an accelerating or a retarding sense) upon the flow, i.e., that this array does not act as either an impeller blade or a turbine blade. The tubes in such a case are preferably round but could also have an elongate cross-section extending in the direction of the spiral; the tubes may be internally partitioned to let the

carrier of an associated channel pair pass therethrough in opposite directions.

The above and other features of my invention will be described in detail hereinafter with reference to the accompanying drawing in which:

FIG. 1 is a block diagram of a ventilating and heat-recuperating system embodying my invention;

FIG. 2 is a schematic view of the temperature pattern in the system of FIG. 1 for heating and cooling operation;

FIG. 3 is an axial sectional view of an apparatus, taken in the direction of arrows as seen from the line III—III of FIG. 4, to be used in the system of FIG. 1;

FIG. 4 is a perspective view (parts broken away) of the apparatus shown in FIG. 3;

FIG. 5 is a sectional view similar to FIG. 3, showing a modification;

FIG. 6 is a cross-sectional view taken on the line VI—VI of FIG. 5;

FIG. 7 is a schematic view similar to FIG. 2, showing the temperature pattern in a system using cascaded heat-exchanging stages;

FIG. 8 is a block diagram of an apparatus according to the invention with cascaded heat-exchange stages operating with a temperature pattern as shown in FIG. 7;

FIG. 9 is a partly elevational and partly sectional view of an apparatus representing a further embodiment;

FIGS. 10, 11, 12 and 13 are cross-sectional views respectively taken on the lines X—X, XI—XI, XII—XII and XIII—XIII of FIG. 9;

FIG. 14 is a block diagram similar to FIG. 1, representing a modified system;

FIG. 15 is an overall plan view of an apparatus according to my invention, shown installed in the corner of a room;

FIG. 16 is an end view of the apparatus shown in FIG. 15;

FIG. 17 is a sectional top view of yet a further embodiment;

FIG. 18 is a cross-sectional view taken on the line XVIII—XVIII of FIG. 17;

FIG. 19 is a partly elevational and partly sectional view of an apparatus representing still another embodiment;

FIG. 20 is an end view (partly in section) of the apparatus shown in FIG. 19; and

FIG. 21 is an axial sectional view of a gas turbine embodying a system according to my invention.

In FIG. 1 I have schematically illustrated a ventilating and heat-recuperating system 1 serving to transfer heat from or to a stream of used air, which is being withdrawn from a body of spent air in a room, to or from a stream of fresh air which enters the room and is being withdrawn from a body of atmospheric air, i.e., from the external environment. The system comprises a prime mover, here shown as an electric motor M, which drives two air-circulating blowers 2, 3 and a pump 4 for displacing a liquid or gaseous heat-carrier medium. The blower 2 drives used air from the room (arrow 8) to the outside while the blower 3 draws fresh air from the outside into the room (arrow 7). Used air passes through a heat exchanger 6 and thermally interacts with the carrier which is circulated by the pump 4 to flow in a closed circuit including ducts 9 and 10. The stream of fresh air thermally interacts with the same carrier in another heat exchanger 5. After passing through the duct 10 the carrier is cooled in the heat exchanger 5 if, as assumed for the present, the outside temperature is below room temperature, i.e., if the room is independently heated. The thus-cooled carrier flows through the duct 9 and abstracts heat from warmer air in the heat exchanger 6 whereupon it reenters the duct 10. It will be seen that the cooler stream (arrow 7) cannot be contaminated by the warmer stream (arrow 8), or vice versa, and that the two streams cannot contaminate, or be contaminated by, the carrier.

If the temperature of the air stream which leaves the room (arrow 8) is lower than the temperature of the inflowing air stream (arrow 7), i.e., if the room is independently cooled, the inflowing air is cooled in the heat exchanger 5 by the carrier which transfers its heat to the outflowing air in the heat exchanger 6. Elements 4, 5, 6, 9 and 10 together form a sealed circuit for the carrier.

FIG. 2 illustrates the temperature pattern for the heat exchangers 5, 6 of the apparatus shown in FIG. 1 when the room temperature is lower (curves I, II) or higher (curves III, IV) than the outside temperature. The temperature to be maintained in the room is marked t_R. Curves II and III show the temperature pattern in the heat exchanger 6 during the cooling phase and the heating phase, respectively. Curves I and IV show the temperature pattern in the heat exchanger 5 for the cooling and heating phases, respectively.

In summer, as shown by the curve I, the carrier in the heat exchanger 5 absorbs heat from the fresh-air stream represented by the arrows 7 so that incoming air is cooled from the outside temperature t_A to an intermediate temperature $t_K+\Delta t$. The carrier, which has a mean temperature t_K, is transported by the pump 4 from the heat exchanger 5 to the heat exchanger 6 where it transfers heat to the outgoing air stream, represented by arrows 8, whose temperature pattern is given by curve II; thus, the temperature of this outgoing air stream is raised from room level t_R to an intermediate level $t_K-\Delta t$ slightly below the temperature of air stream 7.

In winter, as shown by curves IV and III, the operation is analogous with maintenance of a mean carrier temperature t_H.

As can be readily seen from the temperature diagram of FIG. 2, the heat-recuperating system according to my invention is able to recover 50percent of the heat otherwise lost, this representing an ideal case with $\Delta t = 0$ (i.e., with the heat-exchange surface assumed to be infinite). Thus, the temperature difference $t_R - t_H$ exists in the heating phase and the temperature difference $t_K - t_R$ exists in the cooling phase; these differences have to be balanced by a supplemental heating or cooling system not shown. Naturally, the outside temperatures t_A and t'_A (and thus also the intermediate temperatures t_K and t_H) are subject to temperature fluctuations of the ambient atmosphere.

It is also evident from FIGS. 1 and 2 that a continuous transition from heating to cooling operation and vice versa can be effected without changing the direction of air flow and without reversing the carrier flow in the sealed circuit. In switching between heating and cooling operation, the heat exchangers 5 and 6 merely change functions.

The abscissa segment between points *r* and *o* in FIG. 2 shows the temperature rise or drop along the flow path of the individual heat exchangers.

FIGS. 3 and 4 show the structure of a ventilating and heat-recuperating apparatus included in the system of FIG. 1. The apparatus comprises rotary heat exchangers 5 and 6 of the type hereinabove referred to as cross-flow blowers, with hollow blades 12 traversed by a liquid heat carrier, such as an alcohol, in the direction indicated in FIG. 3 by arrows 13. The heat exchangers 5 and 6 are coaxial and are directly interconnected by the hollow blades 12. The air is guided through the heat exchangers 5 and 6 by a housing including a first shell 14 for the heat exchanger 5 and a second shell 15 for the heat exchanger 6. The fresh-air stream flowing through the rotary heat exchanger 5 is aspirated through a grill 16 and an inlet port 18 in the direction of arrows 7 and transported through an outlet port 19 to a distributing manifold 20 whence it is conducted to the room or rooms to be ventilated. The used-air stream is aspirated in the direction of arrows 8 through an inlet port of heat exchanger 6 and discharged through an outlet port 21 by way of the same grill 16.

The physically interconnected rotary heat exchangers 5 and 6 form a single rotor body mounted at its ends in ball bearings 22 and 23, the bearing 22 being for instance of the self-aligning tapering type whereas the bearing 23 is a simple journal bearing for a stub shaft 23*a* rigidly fixed to the rotor body. The latter is driven by a motor 24 with shaft 25 via reduction gearing not shown. The carrier-circulating pump 4, here shown to be of the radial type, feeds the carrier to every other blade and draws it from the intervening blades so that adjacent rotor blades are tranversed by the carrier in opposite directions. As more fully illustrated in subsequent Figures, the blades are surrounded by annular cooling fins 12*a* to enhance the heat transfer.

The pump 4 has an impeller with radial vanes 4' and a casing with return guide vanes 26. These return guide vanes are rigidly fixed to a ferromagnetic ring 27 which is held in place by a similar outer ring 31 separated therefrom by a thin metal sheet 28 hermetically sealing the carrier circuit (communicating with the interior of the rotor blades 12) against the outside. At least one of the rings 27, 31 is permanently magnetized. Thus the casing with vanes 26 is immobilized while the rotating impeller vanes entrain the carrier to drive it along its closed path. As explained in connection with FIGS. 1 and 2, the temperature level of the heat carrier in the hollow blades 12 adjusts itself automatically, so that the temperature of the heated or cooled air stream is likewise adjusted. The air volume is easily controlled by butterfly valves (not shown) located in the suction and-/or pressure channels, or by changes in rotor speed. The channels defined by blades 12 represent the stretches 9 and 10 of FIG. 1 along which the carrier flows at substantially uniform speed between the two heat exchangers 5, 6.

The return guide vanes 26 need not be completely stationary but, with their mounting ring 27, could also rotate at a speed different from that of the impeller 4'. Thus, the magnetic field linking the rings 27 and 31 need only exert a retarding effect upon the pump casing.

FIGS. 5 and 6 show an embodiment of my invention wherein, in contradistinction to the embodiments of FIGS. 3 and 4, the two rotary heat-exchanger units are traversed by an axially entering and radially exiting flow.

As shown in FIG. 5, a motor M is centrally mounted on brackets 50 and 51 which are rigidly connected to a housing 52 containing the stator 53 and the armature 54 of the motor.

Armature 54 includes a set of coils would on a hollow shaft 55 consisting of two coaxial cylinders 56 and 57 defining a cylindrical inner passage 9 and an annular outer passage 10 which represent the two stretches referred to above (cf. FIG. 1). The shaft 55 is mounted in bearings 58 and 59 with both of its extremities protruding from the motor housing 52. These extremities carry the two rotary heat exchangers 5 and 6, again constituted by rotary blowers, traversed by air streams flowing in the direction of arrows 7 and 8, fresh air being axially aspirated from the outside through an inlet port provided with a grill 60 and passing through a scroll-type outlet channel 61 of the blower 5 into the room. Similarly, the stream of used air is axially aspirated by the blower 6 through an inlet port 62 and fed to a scroll-type outlet channel 63 (see FIG. 6) leading to the outer atmosphere.

As best seen in FIG. 6 for the rotor 6, each of these rotors comprises an array of hollow blades 64 (illustrated only in part), the interior of each blade being longitudinally subdivided into two passages 65 and 66 by an axially extending partition 67. If desired, each hollow blade 64 may consist of two discrete tubes which form the two passages 65, 66. The hollow end plates supporting the blades of the rotors 5 and 6 and facing the motor M (FIG. 5) are subdivided by radial partitions 68 and 69 to define inner chambers 70, 71 and outer chambers 72, 73, respectively. The passages 66, which are nearer to the tips of the hollow blades 64 of each of the two rotors, respectively lead to the outer chambers 72, 73, communicating with each other via the annular outer channel 10 of the hollow shaft 55. Similarly, the passages 65 nearer to the centers of the two rotors respectively communicate with the inner chambers 70, 71 which, in turn, open into the cylindrical inner channel 9 of shaft 55. As the passages 65 and 66 are interconnected at the suction-side ends of the rotors, the result is a hermetically sealed circuit in which the carrier moves in the directions indicated by arrows in FIG. 5.

As in the embodiment shown in FIGS. 3 and 4, the carrier in the circuit of FIGS. 5 and 6 is moved by a radial pump 4 having return guide vanes 74 on a ferromagnetic ring 83 which, in the manner described above, is held against rotation by a system of magnets on an outer ring 80 which is fixed to the housing of the rotor 6 by a central rod 81 and spider legs 82.

FIG. 7 shows the temperature pattern of an arrangement in which the carrier circuit is divided into several (here five) cascaded subcircuits, each of the subcircuits comprising a heat-exchange stage with two branches wherein heat is respectively absorbed and given up by the carrier. The heat exchange between the two air streams 7 and 8 and the blower rotors, traversed by the carrier in countercurrent thereto, results in higher thermodynamic efficiency.

Curve I of FIG. 7 shows at A – B the temperature drop of the carrier in a heat-exchange unit of a subcircuit in the cooling phase. The carrier moving in this subcircuit (arrows 7) cools the air to a temperature $t_K+\Delta t$ and transfers the absorbed heat in the other heat-

exchange unit to the air stream (arrows **8**) which has been drawn from the room to be ventilated. The temperature pattern in this latter unit is shown by the section C–D of curve II. The air flows then to the corresponding heat-exchange unit of the next subcircuit where its temperature is raised to $t_{Kl} - \Delta t$, according to section D–E of curve II. The individual subcircuits are cascaded in both directions of flow.

The lower curves III and IV of the temperture pattern represent the heating phase, with the air temperature rising in one subcircuit between points of A′ and B′ (curve III) while the carrier temperature drops between points C′ and D′ (curve IV). The heat to be replaced in this instance as defined by the temperature difference $t_{Kc} - t_R$ represents the necessary energy input. These temperature differences are much smaller than the corresponding temperature differences in the single-stage systems of FIGS. **1** to **6**; thus, the cascading as per FIG. **7** affords better heat recuperation.

FIG. **8** is a flow diagram of a cascaded temperature-control system whose mode of operation follows the temperature pattern depicted in FIG. **7.** The reference numerals used in FIG. **8** are analogous to those employed in the basic block diagram of FIG. **1** for a single carrier circuit. Thus, the heat exchanger **5** for the incoming stream **7** circulated by the blower **3** has been replaced by the heat-exchanger stages **5***a* to **5***e* of the individual subcircuits. The same applies to pumps **4***a* to **4***e*, heat-exchange units **6***a* to **6***e* for the outgoing air stream **8** circulated by blower **2,** and associated conduits **9***a* to **9***e* and **10***a* to **10***e*. The individual subcircuits, distinguished by the postscripts *a* to *e*, are so arranged that the stages **5***a* to **5***e* are traversed by the air flow in a direction opposite that of stages **6***a* to **6***e*. The carrier passing through stage **6***a*, closest to blower **2** and therefore cooled (or heated) to a level near the room temperature, passes immediately thereafter through stage **5***a*, farthest from blower **3**, so as to provide final cooling (or heating) for the incoming air already precooled (or preheated) by the four preceding stages.

Thus, the fresh-air stream is cooled along section A–B of curve I in stage **5***c* and the heat so extracted is transferred to the used-air stream in the same subcircuit along the portion C–D of the curve II in the stage **6***c*. With the total temperature difference between room temperature and outside temperature $t_A - t_R$ in the cooling phase and $t_R - t'_A$ in the heating phase subdivided into several steps, the residual temperature differential $t_{Kc}\, t_R$ or $t_R - t_{Hc}$ is reduced in proportion to the number of stages. The cascaded system, therefore, combines the higher thermodynamic efficiency of the countercurrent circuit with the high heat-transfer coefficients of rotary heat exchangers.

In FIGS. **9 – 13** I have shown an adaptation of the apparatus of FIGS. **5** and **6** to the cascaded system described with reference to FIGS. **7** and **8.** Similar reference numerals have been used, with the addition of postscripts *a – e* where necessary to distinguish the several subcircuits. Pumps **4***a – 4e* have been symbolized in FIG. **9** only by their radial vanes.

Rotors **5** and **6** have respective sets of blades **102** and **100** extending along segments of archimedean spirals, each blade **100** being composed of five tubes **102***a – 102e* and **100***a – 100e* which (as illustrated for blades **100**) are subdivided by longitudinal partitions **67***a – 67e* into paired passages **65***a – 65e* and **66***a – 66e*. Passages **65***a – 65e* communicate with generally frustoconical manifolds **72***a – 72e* in rotor **6** and **73***a*, **73***b* etc., in rotor **5**; passages **66***a – 66e* communicate with similar manifolds **70***a – 70e* in rotor **6** and **71***a* etc., in rotor **5.** The manifolds are connected to the axially extending blade passages at junctions **103***a – 103e* as illustrated in FIG. **9** for the rotor **6.** Thus, the tubes of each rotor form several (here five) concentric tiers, the tubes of each tier being served by a common manifold.

The two nested cylinders **56** and **57** of shaft **55** (FIG. **5**) have been replaced, as illustrated in FIGS. **10** and **11,** by a central shaft **106** and a sleeve **107** spacedly surrounding same. Shaft **106** is formed with ten sectoral channels **9***a – 9e* and **10***a – 10e*, communicating with respective manifolds of each rotor, conducting the heat carrier in opposite directions; the annular space between shaft **106** and sleeve **107** is axially subdivided for this purpose into transverse compartments **104***a – 104e* and **117***a – 117e* within rotor **6** and into similar compartments **105***a*, **105***b* etc., and **116***a* **116***b* etc., within rotor **5.** These compartments, in turn, open into respective channels **9***a – 9e*, **10***a – 10e* as illustrated in FIGS. **10** and **11** for the channel **9***b* and compartments **117***b*, **105***b*. Shaft **106** traverses an end wall **110** of rotor **6** and terminates in an end cap **109.**

Advantageously, as explained above, the pitch of the archimedean spirals defined by blades **100** and **102** is so correlated with the rotor speed that the blades exert substantially no reaction upon the surrounding air flow.

FIG. **14** shows the flow diagram of FIG. **1** expanded to include an external source of thermal energy, specifically a heat pump, in direct heat-exchanging relationship with the air streams **7** and **8.** This heat pump, which may be of the construction described in my prior U.S. Pat. No. 3,347,059, is reversible so as to provide either heating or cooling. The pump circuit includes a pair of additional heat exchangers **140** and **141** traversed by the incoming and outgoing air streams **7** and **8,** respectively, these heat exchangers lying in circuit with a compressor **143** driven by a motor M_K and an expansion valve **142.** A four-way valve **146** allows the selective circulation of a heating/cooling fluid in either of two directions as indicated by arrows **147** (for heating of the room) and **148** (for cooling).

Heat-pump motor M_K is advantageously mounted on a common shaft with the motor M of the temperature-control system **1.**

In FIGS. **15** and **16** I have shown an apparatus according to my invention installed in an upper corner of a room to be cooled or heated. Incoming air (arrow **7**) enters the heat exchanger **5** from above, either through a distributing manifold or directly through the roof, whereas outgoing air (arrow **8**) is aspirated upwardly into heat exchanger **6** and then discharged laterally into the atmosphere.

FIGS. **17** and **18** illustrate a temperature-control apparatus according to my invention mounted in a housing **173** for electronic equipment (not shown) whose operation generates waste heat. The interior of housing **173** is sealed against the outside with the aid of an end cap **180** forming an outwardly open well **174** for an externally mounted rotary heat exchanger **160** of the centrifugal-blower type. A similar second heat exchanger **161** is linked with rotor **160** by a hollow shaft **166** journaled by a bearing **181** in cap **180.** The rotors **160, 161** respectively comprise hollow blades **162, 164** and **163,** 165 arranged in pairs as shown in FIG. **18.**

The hollow shaft **166** provides two concentric channels **167, 168** which respectively communicate with separate compartments **169, 170** and **171, 172** in the facing end walls **175, 176** of the rotors **160, 161.** The axial bores of two adjacent blades **162, 164** and **163,** 165 of the rotors **160, 161** respectively communicate with the compartments **169, 171, 170, 172**; thus, the blades **164** and **165** intercommunicate by way of the outer channel **168** of the shaft **166** while the blades **162** and **163** communicate with each other by way of the inner channels **167.** The blades of each rotor are interconnected at the open ends of the rotors **160, 161** by short segments of annular ducts **177** and **178.** Thus each blade has one end which communicates with one compartment in the hollow end walls of its rotor and another end connected to a blade communicating with an adjacent compartment. This establishes a closed circuit for the flow of a heat carrier through the rotors. The flow path begins at a blade **162** of the rotor **160** and extends via duct **177,** blade **164,** compartment **171,** outer shaft channel **168,** compartment **172** of rotor **161,** blade **165,** duct **178,** blade **163** and back by way of the compartment **170,** shaft channel **167** and compartment **169** to the blade **162.** Arrows in FIG. **17** indicate the direction of flow of the warm air stream in the housing **173** and of the cool air stream in the well **174.** The apparatus of FIGS. **17** and **18** preferably employs a heat-carrier fluid of low viscosity. The blades of the two rotors are provided with fins **183** to enhance the exchange of heat.

The rotor **161** in housing **173** absorbs heat, thereby cooling the air circulated by it. The rotor **160** in the well **174** is cooled by the external air it circulates. The colder carrier entering the compartment **172** is subjected to a radial force due to centrifugal action. As the diameter of the rotor **161** in housing **173** is larger than the diameter of the rotor **160** in well **174,** the radial force urging the colder carrier outwardly in rotor **161** outweighs the countervailing forces acting upon the fluid in rotor **160** where the hot carrier has to displace the cooled carrier present in the blades **162, 164** near the rotor periphery. This thermosiphon effect has been claimed in my copending application Ser. No. 234,433 filed March 13, 1972, as a division of application Ser. No. 847,771 and now abandoned.

A drive motor **182** for rotors **160, 161** is supported in housing **173** by spider legs **179** integral with cap **180.** Journal bearing **181** forms a fluidtight seal around the hollow shaft **166.**

It will be noted that the inlet and outlet ports for the circulation of the colder airstream in well **174** communicate with the body of air surrounding the housing **175** whereas the inlet and outlet ports for the warmer air stream within housing **173** communicate with the body of fluid which is entrapped in the housing; this is in contrast to the construction shown, for example, in FIG. **4.**

FIG. **19** is a partly elevational and partly sectional view of a further embodiment which comprises two rotary heat exchangers **190, 191** of the radial-flow type. The rotors **190, 191** circulate the air by shear force with the aid of axially extending tubes **192,** traversed by the heat carrier, which are provided with annular fins **193** more clearly illustrated in FIG. **20.** Tubes **192** may be round, as shown in FIG. **20,** or may have an aerodynamically shaped profile as shown in FIG. **6,** preferably so arranged as to form a set of reactionless blades. Rotors **190, 191** are mounted on a hollow disk **194** of a rotor **195** which forms part of an external-rotor motor **196.** The motor **196** is supported on the housing **198** of rotor **190** by means of spider legs **197.** The housings **198, 199** of rotors **190, 191** have a convoluted cross-section relatively offset in the peripheral direction to facilitate their connection to external air ducts. Air enters from the room by way of an inlet port **187** and from the outside by way of another inlet port **188** covered by a grill **189.**

The two heat-exchange rotors **190, 191** are thermodynamically interlinked by the pipes **192** which are substantially equispaced along their circumference. The pipes **192** are filled with a fluid heat carrier which is maintained at a pressure insuring evaporation at the hot-air side and condensation at the cool-air side. The heat carrier advantageously consists of a combination of liquid and gaseous phases of the same fluid which therefore is present partly as a saturated vapor and partly as a condensate. Heat transfer within the pipes **192** is enhanced by centrifugal forces which reduce the thickness of the condensate layer at the cool-air side. The transport of condensate to the hot-air side, due to these differences in layer thicknesses, can be enhanced by a capillary tube lining as is well known per se.

In FIG. **21** I have shown a gas turbine embodying the principles of my invention. A shaft **201,** connected to a load not shown, carries a radial-compressor wheel **202** and an axial-turbine wheel **203.** A coaxial and approximately rotationally symmetrical combustion chamber **205** is supplied with compressed air via conduits **204** and with fuel via a nozzle **206.** The hot combustion gases, expanded in an annular channel **207,** flow through a guide-blade ring **208** and then traverse a set of hollow axial blades **209** of the turbine wheel **203.** The shaft **201** is journaled in anti-friction bearings **210** and **210'** is a housing portion **211**; a heat-exchanger unit, consisting of a first heat exchanger **212** constituting the heat sink and a second heat exchanger **213** through which the compressor air flows, is also carried by the same stationary housing portion **211** by means of a hub **214** having channels **215** and **216** radiating therefrom. The heat exchangers **212** and **213** comprise tubes **217** and **217'** with substantially parallel axes, the tubes communicating with one another at their outlying ends by rotary annular chambers **218** and **218'** and with the channels **215** and **216** at their opposite ends. The entire channel system is filled with a thermally stable liquid, preferably a eutectic of sodium and potassium. A guide-blade ring **219** gives the stream of exhaust gas emerging from the axial blading **208,** 209 a whirl whose thrust serves the purpose of driving the unit **212, 213** by its passage through the heat exchanger **212.** As a result of the rotation of the heat exchangers and the heating up of the wall of the channels facing the turbine, the enclosed liquid heat carrier is set in circulation (arrows) by the aforedescribed thermosiphon principle since in the larger-diameter heat exchanger **212** the inwardly flowing liquid stream is of lower density than the portion of liquid flowing outwardly from the interior. In this way the heat which is withdrawn from the exhaust gas in the heat exchanger **212** is transferred to the compressor air via the heat exchanger **213.** The two heat exchangers are separated by a partition **220** which is split into two halves and carries a shaft seal **221.**

I claim:

1. A temperature-control system for a space provided with means for maintaining a first fluid therein at a temperature differing substantially from that of a second fluid in an adjoining environment, comprising:

a first rotary heat exchanger centered on an axis and exposed to said first fluid in said space;

a second rotary heat exchanger centered on said axis and exposed to said second fluid in said adjoining environment, each of said heat exchangers including a set of peripherally spaced tubes parallel to said axis, the tubes of each set being disposed along at least one segment of an archimedean spiral to define a hollow blade;

drive means coupled with said heat exchangers for rotating same about said axis;

sealed conduit means including said tubes forming a closed path for a heat-carrier medium abstracting heat from the warmer one of said fluids and giving off heat to the colder one of said fluids; and

forced-circulation means in said path for driving said medium through said conduit means under a pressure maintaining a coherent flow of said medium at substantially uniform speed between said heat exchangers.

2. A system as defined in claim wherein the tubes of each heat exchanger are divided into a plurality of concentric tiers interconnected in cascade for successive traversal by said medium.

3. A system as defined in claim 1 wherein each of said tubes is longitudinally subdivided into two parallel conduits traversed by said medium in opposite directions.

4. A system as defined in claim 3 wherein said heat exchangers are axially spaced apart and provided with a common shaft, said conduit means including axially extending channels in said shaft and manifolds connecting the conduits of the tubes of each tier with a respective channel pair.

5. A system as defined in claim 4 wherein said channel pairs are formed by peripherally offset sectoral passages in said shaft.

6. A system as defined in claim 1 wherein said tubes are provided with closely spaced transverse fins entraining the surrounding fluid by shear force and imparting thereto a centrifugal acceleration proportional to the speed of said drive means.

7. A system as defined in claim 6 wherein the pitch of said archimedean spiral is correlated with the speed of said drive means to exert substantially no reaction upon the centrifugally accelerated fluid flow.

8. A system as defined in claim 6 wherein said fins are annular and common to all the tubes of a heat exchanger.

9. A temperature-control system for a space provided with means for maintaining a first fluid therein at a temperature differing substantially from that of a second fluid in an adjoining environment, comprising:

a first rotaty heat exchanger centered on an axis and exposed to said first fluid in said space;

a second rotary heat exchanger centered on said axis and exposed to said second fluid in said adjoining environment;

drive means coupled with said heat exchangers for rotating same about said axis, each of said heat exchangers including a set of peripherally spaced hollow blades parallel to said axis and a multiplicity of closely spaced transverse fins entraining the surrounding fluid by shear force and imparting thereto a centrifugal acceleration proportional to the speed of said drive means;

sealed conduit means including said hollow blades forming a closed path for a heat-carrier medium abstracting heat from the warmer one of said fluids and giving off heat to the colder one of said fluids, said blades defining at least one segment of an archimedean spiral whose pitch is correlated with the speed of said drive means to exert substantially no reaction upon the centrifugally accelerated fluid flow; and

forced-circulation means in said path for driving said medium through said conduit means under a pressure maintaining a coherent flow of said medium at substantially uniform speed between said heat exchangers.

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