

Nov. 27, 1951

N. A. PENNINGTON
AIR-CONDITIONING APPARATUS

2,576,140

Filed June 15, 1946

3 Sheets-Sheet 1

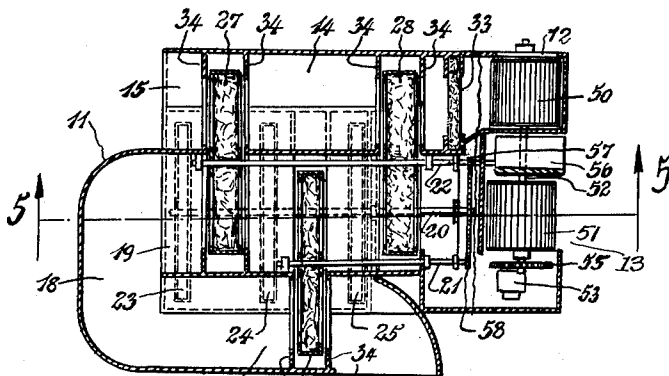


Fig. 1.

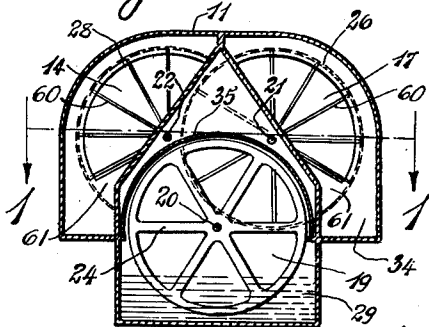


Fig. 2.

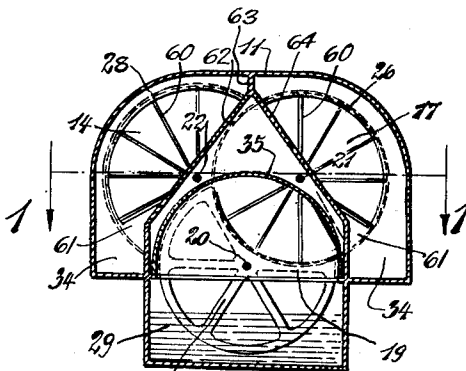


Fig. 3.

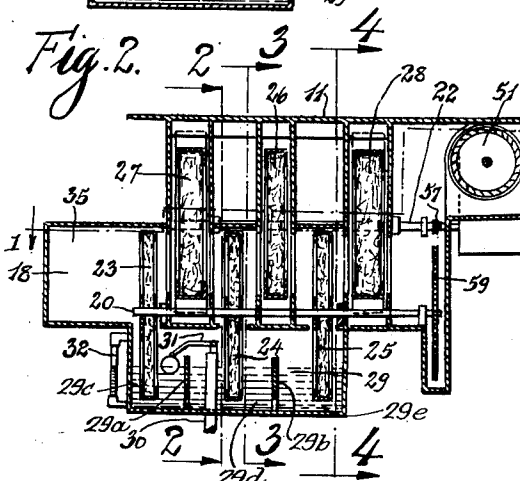


Fig. 4.

Fig. 5.

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3 Sheets-Sheet 2

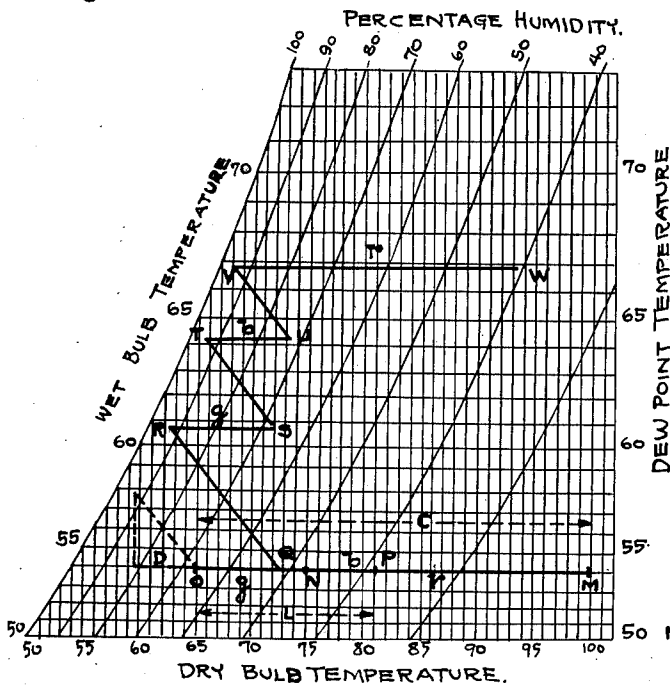
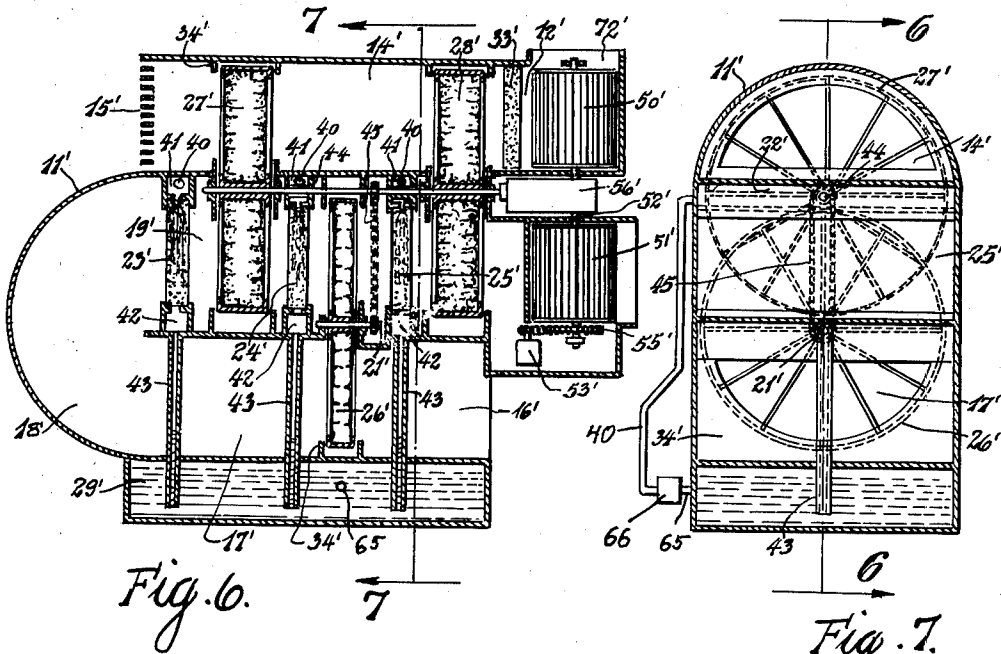


Fig. 8.

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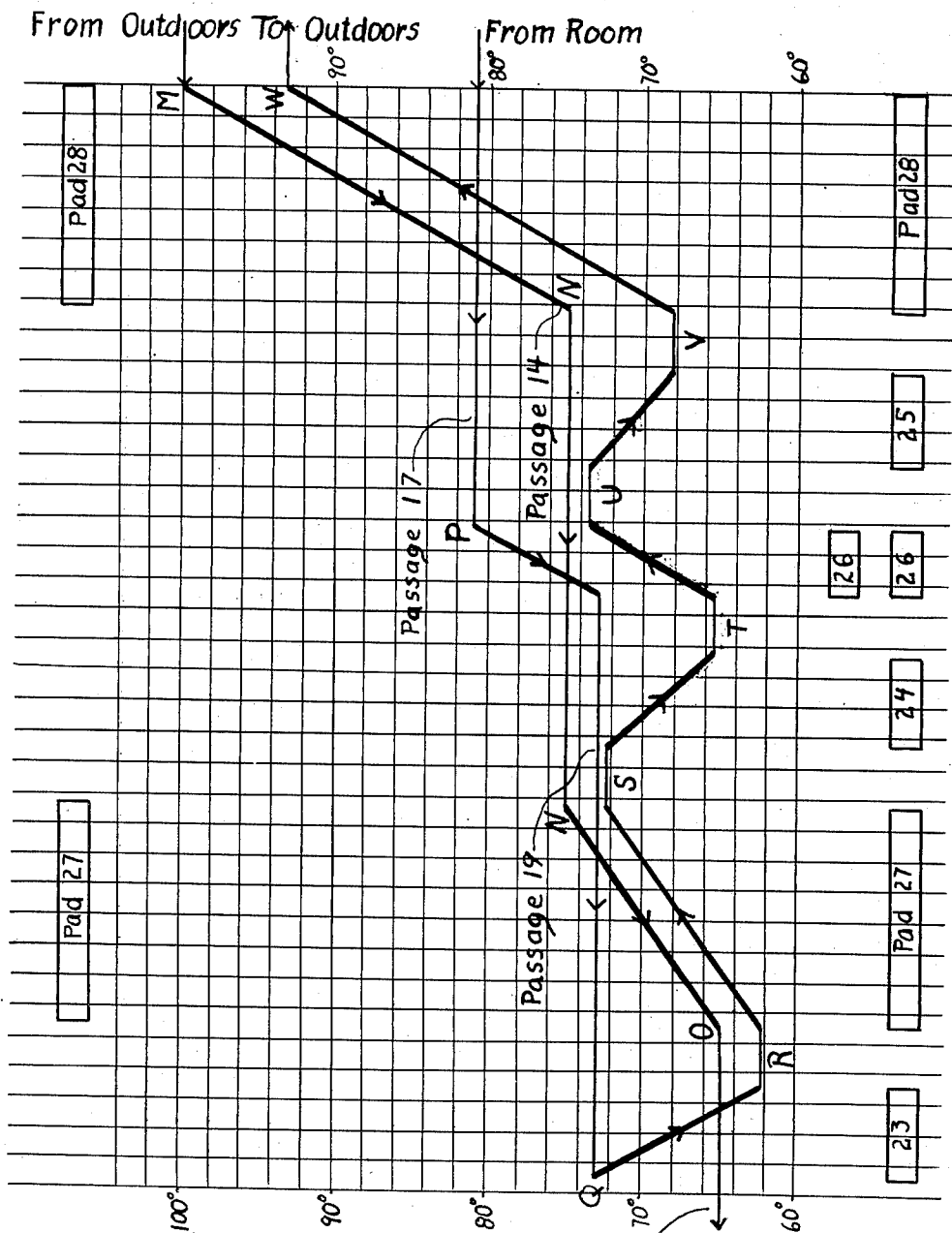
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AIR-CONDITIONING APPARATUS

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Filed June 15, 1946

3 Sheets-Sheet 3



UNITED STATES PATENT OFFICE

2,576,140

AIR-CONDITIONING APPARATUS

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Application June 15, 1946, Serial No. 676,962

26 Claims. (Cl. 62—139)

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My invention relates to new and useful improvements in air-conditioning apparatus, and more particularly to self-contained units for cooling individual rooms, although my invention could equally well be used in a central air-conditioning plant for a building or residence.

The principal object of my invention is to devise a unit for merely cooling the air, without appreciable incidental change in the moisture-content thereof; although by adding appropriate further elements (either conventional or inventive per se) my unit would serve to humidify, dehumidify, or otherwise change the characteristics of the air which is being treated.

In my copending application, Serial No. 640,792, filed January 12, 1946, now Patent No. 2,464,766, granted March 15, 1949 (which application may be referred to for further details as to all common subject-matter), I show and describe two species of the genus to which belong the two species to be shown and described herein. But as to those two prior variants, it was there pointed out that neither of them could reduce the dry-bulb temperature of the incoming air to below one degree higher than the wet-bulb temperature which the outgoing air has when it leaves the room. Also that, in order to accomplish a reduction closely approaching this absolute limit, would require the use of rotating heat-exchange pads of almost prohibitive thickness.

But this is frequently no drawback. A situation in which the "load" on the machine (i. e., the difference in dry-bulb temperature between the incoming air and the outgoing air) is excessive, can always be met by enlarging the capacity of the machine, by increasing the diameter of the pads and conduits, while maintaining the same rate of flow of air per unit of cross-sectional area (i. e., multiplying its diameter by the square root of 2) should halve the load, by admitting to the room twice as much cool air per unit of time.

But this solution is unsatisfactory whenever it requires the admission of such a large quantity of air per unit of time as to create a drafty condition. Air-conditioning experts are in general agreement that a change of air more often than once every three minutes is excessive. Accordingly if the air in the room heats up at such a rate that an excessive supply of cool air is necessary to compensate for this heating-up, then the use of a large air-conditioner is not the solution—there is need instead for a machine which will cool less air to a lower temperature.

And, even in cases in which the use of a large air-conditioner would not introduce excessive

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quantities of air, it may be preferable to use a machine which is smaller and yet capable of a greater degree of cooling.

Accordingly it is the principal object of my present invention to devise a machine which will effect a cooling of the incoming air to below the limit inherent in my earlier machine; and which, even above that limit, will cool the incoming air with the use of less total pad-thickness for the same diameter of pads.

More specifically it is my object to accomplish this end by an anhydrous heat exchange between the outgoing stream of air at one stage of its exit and the same stream at another stage of its exit.

This may sound like "lifting oneself by one's own boot-straps," and it is exactly and absurdly that unless the two stages are properly selected: Accordingly it is the final object of my invention to make not only a feasible, but the best possible selection of these two stages.

In addition to my principal objects, above stated, I have worked out a number of novel and useful details, which will be readily evident as the description progresses.

My invention consists in the novel parts and in the combination and arrangement thereof, which are defined in the appended claims, and of which two embodiments are exemplified in the accompanying drawings, which are hereinafter particularly described and explained.

Throughout the description the same reference number is applied to the same member or to similar members.

Figure 1 is a horizontal section of one variant of my present invention, taken along the lines 1—1 in Figures 2, 3, 4 and 5.

Figure 2 is a vertical cross-section of the same, taken along the lines 2—2 of Figure 5.

Figure 3 is a vertical cross-section of the same, taken along the lines 3—3 of Figure 5.

Figure 4 is a vertical cross-section of the same, taken along the lines 4—4 of Figure 5.

Figure 5 is a vertical longitudinal section of the same, taken along the lines 5—5 of Figure 1.

Figure 6 is a vertical longitudinal section of a second variant, taken along the lines 6—6 of Figure 7. The close apparent similarity between Figures 1 and 6 do not indicate that these are comparative sections of the two variants.

Figure 7 is a vertical cross-section of the second variant, taken along the lines 7—7 of Figure 6.

Figure 8 is a psychrometric chart, the same for both variants.

Figure 9 is a chart of the complete air-circuit, the same for both variants, dry-bulb temperature

being plotted against depth within pad for the successive passage of the air through the pads.

Before describing the device of my present invention, let us for a moment consider a set of data which cannot be met by the device of my former invention. Let us suppose outside air of 100° dry-bulb and 54° dew point, and a desired room temperature of 81°; and that, even with a machine which is delivering cooled air at a rate sufficient to change the air in the room once every three minutes, the room-load is 16°. This means that, in order to hold the room temperature at 81°, we must deliver the incoming air of 65°. But, inasmuch as the wet-bulb of air at 81° dry-bulb, and 54° dew point, is 64°, we cannot cool this outgoing air to below 65°, and hence an infinite total pad-thickness would be necessary to render our machine operative in such conditions, no matter how many pads we were to employ.

I shall now proceed to discuss a slight, but very ingenious, modification of my former device, whereby we can escape this seeming impasse, with a total pad-thickness of only 33 inches.

Turning now to Figures 1 to 5, we see that 11 is the main container of the first variant of my present invention. 12 is an air-inlet from outdoors. 13 is an air-outlet to outdoors.

Inasmuch as my present invention is an improved modification of the invention of my already-mentioned copending application, Serial No. 640,792, some of the conventional parts shown and described in said prior application may be omitted therefrom or insufficiently described, but reference may be had to said prior application for a fuller description of such parts.

In inlet 12 there is a fan 50 (hereinafter more fully described), or other appropriate means, to impel air from right to left in Figure 1, into passage 14, and thence into the room through opening 15.

In outlet 13, there is a fan 51 (hereinafter more fully described), or other appropriate means, to suck air from the room through opening 16, thence through passage 17, thence around curve 18, and thence through passage 19 into the outside air, or the attic or other exhaust space.

Shaft 20 rotates at a rate of about 3 R. P. M. Shafts 21 and 22 rotate at a rate of about 30 R. P. M. The reasons for these rates is explained hereinafter.

Shaft 20 carries, keyed thereto, three circular pads 23, 24 and 25, which may be of the same sort as the "excelsior pads" of my said copending application, or the equivalent. Herein they will be referred to as "water-holding pads." I prefer to pack the excelsior in, at about 2.5 pounds per cubic foot, which density of packing I have found to be somewhat preferable to the density suggested in my earlier application. This seems to interpose practically no resistance to airflow, and yet to provide a maximum of evaporative cooling. This is about the density customarily employed in ordinary evaporative coolers. Note also that the spokes have an I cross-section, and the rim has a channel cross-section, all so as to prevent the draining of water, from the excelsior, out of either face of the pad.

Since the conception of my said copending application, I have devised a further refinement of my water-holding pads, which renders them much more efficient, and which will now be described. This refinement consists in pretreating the excelsior, or other water-absorbing packing,

with minute quantities of that type of wetting agent (i. e., water-soluble surface-active agent) characterized by its special ability to lower the surface tension between water and air. This agent should preferably be odorless, non-poisonous, and non-evaporating at the air-temperatures with which it is to be used. Such an agent is di-octyl sodium sulfosuccinate. For example, I have found that, by soaking the pads in a bath of water containing one part in 10,000, by weight, of this agent, for 24 hours before placing them in my device, a thinner pad can be used with the same cooling effect, thus reducing the already negligible resistance to airflow, and furthermore even increasing to over 95% the percentage of humidity which can be imparted to the air.

When my pads are thus treated, the agent need not be renewed until either the stuffing is replaced, or the tanks are cleaned out and a completely new change of water is added.

Instead of pre-treating the pads, they can be treated by impregnating the tank-water with the agent.

Either of these methods of treating the pads is equally applicable to the variants of my said copending application, to either of the variants of this present application, or for that matter to the pad of an ordinary evaporative cooler.

Shaft 21 carries, keyed thereto, one circular pad 26, containing a packing of non-hygroscopic air-permeable highly heat-absorbent material, of such sort and so arranged that there will be no appreciable heat conduction or convection within the body of the pad parallel to the axis of rotation of the pad during the time of one cycle of rotation, and no appreciable airflow within the body of the pad perpendicular to the axis of the pad. These two criteria will be detailed a bit later herein, just preceding and following my description of the air-cycle of my device. It is also important that this material be not subject to deterioration under the influence of moisture. Such a pad as shown and described in my already-mentioned copending application, will satisfy these criteria.

High heat-absorption is a combination of: proper surface conditions (such as ratio of surface to volume, and local individual coefficient of heat transfer), so as to admit heat quickly into the material; high heat-conductivity, so as to spread the heat quickly through the material; and high specific heat per volume, so as to store the heat in large quantities without such a rise in temperature as would unduly interfere with the flow of heat between air and pad by reducing the temperature difference between air and pad to too small a figure. The exact part which each of these desirable qualities plays in what might be called "specific thermal absorption" has not yet been completely theoretically worked out, but empirically it may be said that filamentous aluminum, zinc, or copper appear to be ideal. I prefer to pack this metal wool in, at about 3.5 to 4.0 pounds per cubic foot for coarse grade aluminum wool; other materials proportionally to their specific gravity. For medium or fine grade metal wool, the density of packing would be somewhat less.

Any other sort of pad, having the characteristics above described, could be substituted. Pads of this generic sort will hereinafter be referred to as "rotating heat-exchange pads."

Shaft 22 carries, keyed thereto, two rotating heat-exchange pads 27 and 28, similar to pad 26,

already described. The relative thickness of these three heat-exchange pads, and the three water-holding pads, will be discussed later herein.

Water-holding pads 23, 24 and 25 rotate partly in water-tank 29, and partly in outgoing air-passage 19. It will be noted that my water-tank is divided, by two perforated partitions, 29a and 29b, into three separate compartments 29c, 29d, and 29e. This has the advantage of having the three bodies of water interconnected so that they can be filled from a common source and kept at a common level, and yet have them sufficiently separated from each other, so that each of them can remain at practically the evaporation temperature of the three water-holding pads 23, 24 and 25, which three temperatures will be somewhat different. The water-tank can be kept filled to a proper level by water-pipe 30 and ball-cock 31, its level being ascertainable by gauge 32. For the proper level I prefer an immersion of the lower one-quarter (by height) of the pad.

I have found that, with this degree of immersion in water, and with a rotation of about 3 R. P. M., the radially inward gravity flow of the water from the soaked upper portion of the periphery of the pad, distributes the water very evenly and at just about the right degree of saturation for the best evaporation. Flanges on the rim and radial ribs confine this water and prevent it from flowing out of the pad.

Slower rotation would permit too much inward draining, and too much drying out due to evaporation, and so would reduce the total amount of evaporation. Faster rotation would not give time for the desired evenness of distribution, and so would reduce the total amount of evaporation.

Too fast rotation would also result in the entrainment of water particles by the outgoing air-stream. Although, of course, we do not care whether or not this stream is thus contaminated, these particles would become carried over into the incoming stream by the aluminum-wool pads, about to be discussed.

Heat-exchange pad 26 rotates partly in outgoing air-passage 19, and partly in outgoing air-passage 17.

Heat-exchange pads 27 and 28 rotate partly in outgoing air-passage 19, and partly in incoming air-passage 14.

I have found that, for these three heat-exchange pads, a rotation of about 30 R. P. M. is optimum for an air velocity of about 600 feet per minute.

This rate represents a compromise between the following offsetting considerations. Due to the fact that rate of heat flow is proportional to the difference in temperature between pad and air, and to the fact that each particle of the pad, as it passes across each air-stream, becomes less and less different in temperature from that stream, these two facts combine to produce the result that the change in temperature of each particle is exponential rather than linear. At an infinite speed of rotation, the temperature of each particle would remain constant, half way between the temperatures of the two air-streams, thus attaining ideal pad-efficiency. At any reasonably high speed, the change in temperature of each particle would be practically linear. At very slow speeds, this change in temperature would be very curved, with corresponding great loss in pad-efficiency. Also too slow a rotation of the pad would give time for heat to flow along

the individual filaments, thus violating the desideratum that there should be no appreciable conduction of heat within the pad parallel to the airflow, which desideratum is hereinafter discussed in considerable detail. And yet, if the pad be rotated too fast, air from one stream would be entrained and mixed with the other stream, which is undesirable. Still faster rotation would also interfere with the cross-flow of the air through the pad.

33 is a removable dust-filter of any appropriate sort, either conventional or not.

The by-passing of air around the rotating heat-exchange pads 26, 27 and 28, is prevented by baffles 34, each of which has openings as shown, to permit the passage of air to and from a portion of each pad. These baffles (except for the bridge portions thereof, to be hereinafter mentioned) could be supplanted by arched partitions similar to 35, hereinafter mentioned.

The by-passing of air around the rotating water-holding pads 23, 24 and 25, is prevented by the arched partition 35, which could be supplemented or supplanted by apertured baffles similar to 34.

To attain the already-mentioned desired absence of appreciable airflow perpendicular to the axis of rotation of the pad, which airflow would cause an undesired mixing of the air-streams of passages 14, 17 and 19, I divide each of the rotating heat-exchange pads 26, 27, and 28, by radial ribs 60, each as wide (in a direction parallel to the axis of rotation) as is the pad itself. These ribs cooperate with the bridge portions 61 of baffles 34, which bridge portions are of at least the width of one sector between two successive radial ribs.

In Figures 1 and 4, I have shown the power-plant of my machine. 50 is a centrifugal fan, which sucks outdoor air in through inlet 12, and impels it through passage 14, at about 600 feet per minute through the pads. 51 is a centrifugal fan, which sucks room air from passage 19, through the pads at about 600 feet per minute, and impels it out through outlet 13. As shown, the two fans are separate, although keyed to a common shaft 52; but they could be one double fan. Different air-impelling means could likewise be used.

Shaft 52 is driven by motor 53, through pinion 54 and gear 55.

On shaft 52, there is a gear-reduction 56, which drives shaft 22 at about 30 R. P. M.

On shaft 22 there is a small sprocket 57. On shaft 21 there is a similar sprocket 58. On shaft 20 there is a sprocket 59, ten times the diameter of sprockets 57 and 58. Through a chain (omitted in the figures, in order to expose the three sprockets to view) sprocket 59 drives sprockets 57 and 58.

The air-cycle of my device is as follows:

Exhaust air, leaving the room through passage 17, is precooled anhydrously by rotating heat-exchange pad 26. It then passes around curve 18 into passage 19, where it is further cooled adiabatically by the evaporation of water from water-holding pad 23. It then successively cools anhydrously pad 27, is cooled adiabatically by pad 24, cools anhydrously pad 26, is cooled adiabatically by pad 25, cools anhydrously pad 28, and passes out of the system.

Note that pad 26 is located between pads 27 and 28. Psychrometric charting of the operation of the system with pad 26 variously located,

indicates that this location is very important, if not essential, to the efficient operation of my present invention.

Meanwhile fresh outdoor air is entering at the same rate through opening 12. Here it is filtered by passing through dust-filter 33. It then passes through two successive anhydrous cooling stages in rotating heat-exchange pads 28 and 27, and hence into the room.

It should be noted that the incoming air-stream in passage 14 flows through each of the heat-exchange pads 28 and 27, and that the outgoing air-stream in passage 17 flows through heat-exchange pad 26, in exactly the opposite direction from the flow of the air-stream in passage 19 through each of these pads. Also that the speed of flow is practically identical. Also that the two flows through pads 28 and 27 are in inverse order.

It should be noted that the three parallel air-passages 14, 17 and 19, are separated from each other by inner walls 62, 63 and 64.

I have already referred, earlier herein, to the desideratum of there being no appreciable conduction or convection of heat by any of the heat-exchange pads parallel to the axis of rotation thereof. This desideratum could be attained by a laminated arrangement of the filamentous metal in the pad, this fact being the discovery of another. I have found, however, that even unlaminated metal wool is satisfactory for my purpose, provided the pad be rotated at proper speed; for at such speed, on account of the very small thickness of the individual aluminum filaments, the heat absorbed by the wool in the hot part of the pad's cycle of rotation does not have time to travel by conduction longitudinally of the individual filaments, or from filament to filament, an appreciable distance, before being given up in the cold part of the cycle (although it does have time to flow from the surface of each filament to its center), and hence remains at substantially the same point in the pad in each of the two air-streams. And the fact that the pad rotates so that each particle thereof is carried in a plane perpendicular to the airflow, prevents any conveying of heat by the pad except perpendicular to the airflow.

As the result of these characteristics the pad can be regarded as though made up of an infinite number of infinitesimal discs or laminae. These characteristics result in the mean temperature difference between either air-stream and the wool at any given depth being equal to one half the temperature difference between the two streams at either surface of the pad: i. e., practically ideal counterflow.

It should be emphasized that, in order to attain the advantages which I obtain from counterflow, it is necessary not only to flow my two air-streams in opposite directions, but also to flow them in opposite directions through a means of heat-exchange in which, as just explained, there is no appreciable conduction or convection of heat by the heat-exchange means except perpendicular to the airflow.

The importance of such counterflow cannot be overemphasized, for such counterflow enables each anhydrous heat-exchange between the air to be cooled and adiabatically cooled outgoing air to remove heat from the first-mentioned air to an extent impossible in any system in which there is any averaging of temperature in the heat-exchange means parallel to the direction of flow of the air. Such averaging can occur by the

use of cooling or heating coils, or by the use of random flow of non-hygroscopic cooling-heating fluid in pads, or even (although such flow of fluid in the pads be strictly perpendicular to the flow of the air therethrough, and hence no averaging takes place within either pad), by commingling the fluid as it passes from pad to pad.

In a thoroughly averaging system, the absolute limit of cooling the incoming air by any one heat-exchange is half of the difference between the temperature at which the air to be cooled enters the heat-exchanger and the temperature at which the adiabatically cooled outgoing air enters the heat-exchanger, and even this theoretic limit is not practically attainable. Even if the averaging of temperature takes place only as the cooling-heating fluid passes from pad to pad, the absolute limit is increased merely to two-thirds the above-mentioned difference. Whereas in a counterflow system of the sort contemplated by me, the limit is the temperature at which the adiabatically cooled outgoing air enters the heat-exchanger, and it is quite practical to cool the incoming air to within a very few degrees of this limit.

Figure 8 is a psychrometric chart of my air-cycle, which I outlined just prior to the above dissertation on counterflow.

Fahrenheit temperatures are used throughout this patent.

Outside air at 100° dry-bulb, 54° dew point, is represented at M. Pad 28 anhydrously cools it from M to N, and pad 27 further cools it from N to O, whereupon it enters the room.

The room load of 16° heats the air from O to P, at which point the air reenters the apparatus. Pad 26 anhydrously precools it from P to Q. Pad 23 adiabatically cools it further from Q to R. It then anhydrously absorbs heat from pad 27, from R to S. Pad 24 adiabatically recools it from S to T. It then anhydrously absorbs heat from pad 26, from T to U. Pad 25 adiabatically recools it from U to V. Finally it anhydrously absorbs heat from pad 28, from V to W, and then passes out of the system.

It will be noted that RS equals ON, that TU equals QP, and that VW equals MN.

Points M, O and P are respectively fixed by the dry-bulb and dew-point of the outdoor air, the dry-bulb at which it is desired to feed air to the room, and the room load. Any given choice of points N and Q, fixes points R, S, T, U, V, and W.

As stated in my already-mentioned copending application, I have empirically worked out the following rough rule of thumb for determining what thickness of rotating heat-exchange pad is necessary to accomplish any given desired heat transference: namely that I have found that, at an air velocity of about 600 feet per minute, and a pad rotation of about 30 revolutions per minute, an anhydrous cooling of about one degree Fahrenheit per inch of pad thickness per degree of mean temperature difference between pad and each air stream can be expected. This formula is, however, presented here not so much as a quantitative guide to the designing of machines of the sort invented by me, but rather for purposes of comparison in discussing the patentable differences between my present invention and the two-pad variant of my prior copending application. This formula is predicated upon the assumption of practically perfect counterflow, which is attainable by my invention.

In connection with the two-pad variant of said prior application, it was stated as mathemati-

cally demonstrable that, for any given set of data, a minimum of total heat-exchange pad thickness would suffice, if the two pads were of equal thickness; and a formula was given for computing this minimum.

Somewhat similarly, it is mathematically demonstrable that, with the present invention, for any given set of data, a minimum of total heat-exchange pad thickness will suffice, if pads 27 and 28 are of the same thickness, pad 26 being considerably thinner.

The most feasible mathematical procedure thus far devised for determining the optimum pad thickness for the present invention is a mixture of formula-solving and tabulation.

My formula involves the following quantities:

C=The absolute value of total dry-bulb cooling: OM on the chart.

D=The dry-bulb temperature of the cooled air as it enters the enclosure, less the dry-bulb temperature which this air would have if then cooled adiabatically to 95% humidity. D must be scaled very carefully from the chart.

L=The load imposed on the machine by the enclosure: OP on the chart.

K=An empirical constant, approximately equal to 3. K is almost exactly the absolute value of the dry-bulb difference between O and N, divided by the wet-bulb difference between those two points. By comparing the values of the S's (hereinafter defined) as scaled from a psychrometric chart with the computed values for the first two or three computations, K can then be adjusted so as to secure exact correspondence.

My formulas involve the following quantities which are to be juggled until the optimum result is obtained:

g=The absolute value of the anhydrous dry-bulb change effected in each air-stream by heat-exchange pad 27: ON or RS on the chart.

b=The absolute value of the anhydrous dry-bulb change effected in each air-stream by heat-exchange pad 26: QP or TU on the chart.

r=The absolute value of the anhydrous dry-bulb change effected in each air-stream by heat-exchange pad 28: NM or VW on the chart. r is expressed in my formulas as C-g.

My formulas are solved to obtain the following quantities:

S_g=Thickness of pad 27.

S_b=Thickness of pad 26.

S_r=Thickness of pad 28.

S=Total pad thickness: i. e., 2S_g+2S_b+2S_r.

My formulas are:

$$S_g = \frac{2Kg}{KD-L+b}$$

$$S_b = \frac{2Kb}{KD+(K-1)(L-b)-g}$$

$$S_r = \frac{2K(C-g)}{KD-L+(K-1)g}$$

The procedure is to give b various successive values, at intervals of one or two degrees. For each value of b, give g various successive values and compute S. Determine, by plotting or interpolation, the minimum S for each b, and the minimum of these minima, thus getting the optimum value for b and g. Compute pad thicknesses for this value of b and g. It should come out with S_g almost exactly equal to S_r, and with

S_b about one-third to two-thirds thereof. For any given b, the change in S per change in g is so slight that S_r can always be taken as exactly equal to S_g.

It may help to appreciate the psychrometric chart which constitutes Figure 8, if Figure 9 be considered in connection therewith. Figure 9 is a chart of the complete air-circuit, dry-bulb temperature in degrees Fahrenheit being plotted as ordinates against inches as abscissas. For convenience in comparing the chart of Figure 9 with the representations of my two variant machines in Figures 1 and 6, Figure 9 is so oriented that the flow of air in each air passage is in the same direction in all three figures. Each square represents 2° F., and 1". The inch-measurements relate only to distances within pads (represented by heavy lines on the chart) and not to distances between pads (represented by lighter lines on the chart). The temperature which is plotted against each depth-within-pad, is the average temperature of the air stream in question throughout the theoretical lamina lying at that depth.

Above the chart is a symbolical representation of the two pads encountered by the incoming air-stream in passage 14. Below the chart is a symbolical representation of the one pad encountered by the outgoing air in passage 17; and below that is a symbolical representation of the six pads encountered by the outgoing air in passage 19.

It will be noted that each pair of parallel lines slanting downwardly to the left represents an anhydrous heat-exchange between air and air in one of the rotating heat-exchange pads. The parallelism indicates that, at any given depth within one of these pads, the difference between the average temperature of one air stream and the average temperature of the other air stream is the same as at any other depth within the same pad.

Each line slanting downwardly to the right represents an adiabatic cooling of air, effected by one of the three water-holding pads.

Turning now to Figures 6 and 7, we see a variant of my present invention, psychrometrically identical to the first variant described herein, and differing from it merely by substituting fixed pads and a water circuit for the three rotating water pads 23, 24 and 25 of my first showing.

This substitution, although introducing the added complication and expense of a water-pump, offsets this by enabling me to employ less complicated air-passages, and to reduce the number of shafts to two.

In comparing Figure 6 of my second variant with Figure 1 of my first variant, it should be remembered that Figure 6 is a vertical section, whereas Figure 1 is a horizontal section. But otherwise they correspond, and are perhaps the best two views for comparing my two variants. Insofar as possible, each part of the second variant will bear the same reference number as the corresponding part of the first variant, but primed in the second.

In Figures 6 and 7, 11' is the main container of my second variant. 12' is the air-inlet from outdoors, and 13' is the air-outlet to outdoors.

Air entering through inlet 12', passes through air filter 33', and rotating heat-exchange pads 28' and 27', in passage 14', being successively anhydrously cooled an amount of r and g (see Chart 8) by these two pads. It then enters the room through opening 15'.

In the room it receives a load of L.

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It then leaves the room by opening 18' and passage 17', where it is anhydrously cooled an amount of *b* by rotating heat-exchange pad 26'.

It then passes around curve 18', into passage 19', where it alternately is adiabatically cooled (as shown by the three iso-wet-bulb lines on the chart) by water-holding pads 23', 24', and 25', and anhydrously cools rotating heat-exchange pads 27', 28', and 28', by respective amounts of *g*, *b*, and *r*. Then it is discharged through opening 13'.

Water-holding pads 23', 24', and 25' are identical in composition to corresponding pads 23, 24 and 25, of the first variant, but are non-rotating. They are fed by water from tank 29' through pipe 65, pumped through pipes 40 by a pump 66, into troughs 41, from which the water seeps down through the respective pads 23', 24', and 25', into troughs 42, thence passing by pipes 43 back into tank 29'.

Shaft 22', which drives pads 27' and 28', passes through two of the three troughs 41, by means of sleeves 44. Shaft 22', and shaft 21' which drives pads 26', are synchronized by chain 45.

There are appropriate fans 50' and 51', and an appropriate motor 53' to drive the fans and shafts 21' and 22'.

Except as otherwise described, variant two operates exactly like variant one. Their functioning is identical.

Having now described and illustrated two forms of my invention, I wish it to be understood that my invention is not to be limited to the specific forms or arrangements of parts herein described and shown, or specifically covered by my claims.

I claim:

1. Apparatus for conditioning air for use in an enclosure, comprising: an inlet conduit, for conducting air to be conditioned into said enclosure; an outlet conduit, for conducting air away from said enclosure; means for heat-exchange, circulating alternately through the two conduits, and having anhydrous heat-exchange relationship with the air in each conduit; means in the outlet conduit, for precooling the outgoing air stream by anhydrous heat-exchange with itself; and means in the outlet conduit, for adiabatically further cooling the outgoing air prior to its heat-exchange relationship with the first-mentioned heat-exchange means; whereby the thus cooled outgoing air anhydrously cools said heat-exchange means, which means in turn anhydrously cools the incoming air.

2. Apparatus for conditioning air for use in an enclosure, comprising: an inlet conduit, for conducting air to be conditioned into said enclosure; an outlet conduit, for conducting air away from the enclosure; a first and second means in the outlet conduit, each for adiabatically cooling the outgoing air; means for heat exchange, circulating alternately through the outlet conduit at a place between the enclosure and the first adiabatic cooling means, and through the outlet conduit at a place beyond the second adiabatic cooling means from the enclosure, and having anhydrous heat-exchange relationship with the outgoing air in both places; and a second means for heat-exchange, circulating alternately through the inlet conduit, and through the outlet conduit at a place between the two adiabatic cooling means, and having anhydrous heat-exchange relationship with the air in each conduit; whereby the outgoing air is anhydrously precooled

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by the first heat-exchange means, is further cooled by one of the adiabatic cooling means, and then anhydrously cools the second heat-exchange means, which in turn anhydrously cools the incoming air.

3. Apparatus for conditioning air for use in an enclosure, comprising: three circulating means for anhydrous heat-exchange; three means for adiabatically cooling air; an outlet conduit, for conducting air from said enclosure, successively into relationship with the second heat-exchange means, the first cooling means, the first heat-exchange means, the second cooling means, the second heat-exchange means, the third cooling means, and the third heat-exchange means; and an inlet conduit, for conducting air to be conditioned, successively into relationship with the third heat-exchange means and the first heat-exchange means, and then into the enclosure.

4. Apparatus according to claim 3, further characterized by the fact that the adiabatic cooling means are all evaporative.

5. Apparatus according to claim 3, further characterized by the fact that each adiabatic cooling means comprises an evaporative-liquid-holding pad and means to constantly renew the evaporative liquid in said pad.

6. Apparatus according to claim 3, further characterized by the fact that each adiabatic cooling means comprises a tank and a rotatable evaporative-liquid-holding pad, rotatable partly in the tank, and partly in the outlet conduit.

7. Apparatus according to claim 3, further characterized by the fact that each adiabatic cooling means comprises a fixed evaporative-liquid-holding pad, located in the outlet conduit, and means for constantly circulating evaporative liquid through said pad.

8. Apparatus according to claim 3, further characterized by the fact that each anhydrous heat-exchange means comprises a rotating pad containing a packing of non-hygroscopic air-permeable highly heat-absorbent material of such structure that no appreciable conduction of heat thereby will take place during one rotation of the pad, and by having means to prevent the by-passing of air from passage to passage through or around this pad, and by having the passages so arranged that the air in each passage passes through this pad in a sense opposite to that of the air in the other passage.

9. Apparatus according to claim 3, further characterized by the fact that each adiabatic cooling means comprises an evaporative-liquid-holding pad and means to constantly renew the evaporative liquid in said pad; and that each anhydrous heat-exchange means comprises a rotating pad containing a packing of non-hygroscopic air-permeable highly heat-absorbent material of such structure that no appreciable conduction of heat thereby will take place during one rotation of the pad, and by having means to prevent the by-passing of air from passage to passage through or around this pad, and by having the passages so arranged that the air in each passage passes through this pad in a sense opposite to that of the air in the other passage.

10. Apparatus according to claim 3, further characterized by the fact that each anhydrous heat-exchange means involves counterflow of the two air streams therethrough and comprises a rotating pad containing a packing of non-hygroscopic air-permeable highly heat-absorbent material, so contrived as to prevent appreciable heat-conduction or convection by the heat-ex-

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change means parallel to the axis of rotation of the heat-exchange means during one rotation thereof, there being also means to prevent appreciable airflow through the heat-exchange means from either air conduit to the other.

11. Apparatus according to claim 3, further characterized by the fact that each adiabatic cooling means comprises a water-holding pad and means to constantly renew the water in said pad; and that each anhydrous heat-exchange means involves counterflow of the two air streams therethrough and comprises a rotating pad containing a packing of non-hygroscopic air-permeable highly heat-absorbent material, so contrived as to prevent appreciable heat-conduction or convection by the heat-exchange means parallel to the axis of rotation of the heat-exchange means during one rotation thereof, there being also means to prevent appreciable airflow through the heat-exchange means from either air conduit to the other.

12. Apparatus for conditioning air for use in an enclosure, characterized by having the following psychrometric circuit; a stream of air, anhydrously reduced in dry-bulb temperature a given amount, and then passed into the enclosure; said air then being raised in dry-bulb and wet-bulb temperature within the enclosure; said air being then withdrawn from the enclosure, and then having its dry-bulb temperature anhydrously reduced, and then successively adiabatically reduced and anhydrously increased a number of times; said anhydrous reduction being effected by heat exchange with one of the succeeding anhydrous increases, said other anhydrous increases effecting the anhydrous reduction in the first step.

13. Apparatus according to claim 12, further characterized by the fact that the two chart-lines of each anhydrous heat-exchange overlap each other more than one-half.

14. Apparatus for conditioning air for use in an enclosure, characterized by having the following psychrometric circuit; a stream of air, anhydrously reduced in dry-bulb temperature a given amount, and then passed into the enclosure; said air then being raised in dry-bulb and wet-bulb temperature within the enclosure; said air being then withdrawn from the enclosure and then having its dry-bulb temperature anhydrously reduced, and then successively adiabatically reduced and anhydrously increased three times each; said anhydrous reduction being effected by heat exchange with the second succeeding anhydrous increase; said other two anhydrous increases effecting the anhydrous reduction of the first step.

15. Apparatus according to claim 14, further characterized by the fact that the two chart-lines of each anhydrous heat-exchange overlap each other more than one-half.

16. Apparatus for conditioning air for use in an enclosure, comprising: means for introducing a stream of air into the enclosure; means for extracting a stream of air from the enclosure; a first heat-exchange means for precooling the outgoing air by anhydrous heat-exchange with the outgoing air itself; means for then adiabatically reducing the dry-bulb temperature of the outgoing air to the greatest practicable extent; a second heat-exchange means for then increasing the dry-bulb temperature of this air by anhydrous heat-exchange with the incoming stream; means for then adiabatically reducing the dry-bulb temperature of the outgoing air to the greatest prac-

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ticable extent, and then passing it through the first heat-exchange means; means for then adiabatically reducing the dry-bulb temperature of the outgoing air to the greatest practicable extent; and a third heat-exchange means for then increasing the dry-bulb temperature of this air by anhydrous heat-exchange with the incoming stream; the second heat-exchange means being in each stream indoorward of the third heat-exchange means.

17. Apparatus according to claim 16, further characterized by the fact that each of the three anhydrous heat-exchange means reduces the dry-bulb temperature of the air passing through it in one direction by an amount equal to the dry-bulb temperature increase thereby effected in the air passing through it in another direction, and to a dry-bulb temperature more than two-thirds of the way from the dry-bulb temperature at which the hotter air enters that means toward the dry-bulb temperature at which the cooler air enters that means.

18. Apparatus according to claim 16, further characterized by the fact that each of the three anhydrous heat-exchange means reduces the dry-bulb temperature of the air passing through it in one direction by an amount equal to the dry-bulb temperature increase thereby effected in the air passing through it in another direction, and to a dry-bulb temperature more than two-thirds of the way from the dry-bulb temperature at which the hotter air enters that means toward the dry-bulb temperature at which the cooler air enters that means; each anhydrous heat-exchange means involving counterflow of the two air streams therethrough, and being so contrived as to prevent appreciable heat conduction or convection by the heat-exchange means parallel to the axis of rotation of the heat-exchange means during one rotation thereof, and including means to prevent appreciable airflow through the heat-exchange means in a plane perpendicular to its axis.

19. Apparatus for conditioning air for use in an enclosure, characterized by having the following psychrometric circuit: a stream of air, anhydrously reduced in dry-bulb temperature a given amount, in two successive steps, and then passed in to the enclosure; said air then being raised in dry-bulb and wet-bulb temperature within the enclosure; said air then leaving the enclosure and having its dry-bulb temperature anhydrously reduced a certain amount; said air then having its dry-bulb temperature adiabatically reduced to the greatest practicable extent; said air then having its dry-bulb temperature increased an amount equal to the reduction involved in the second of the two successive steps, by anhydrous heat-exchange with that step; said air then having its dry-bulb temperature again adiabatically reduced to the greatest practicable extent; said air then having its dry-bulb temperature increased an amount equal to the reduction occurring immediately upon the air leaving the enclosure, by anhydrous heat-exchange therewith; said air then having its dry-bulb temperature for the third time adiabatically reduced to the greatest practicable extent; said air finally having its dry-bulb temperature increased an amount equal to the reduction involved in the first of the two successive steps, by anhydrous heat-exchange with that step.

20. Apparatus according to claim 19, further characterized by the fact that the two chart

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lines of each anhydrous heat-exchange overlap each other more than one-half.

21. In an air-conditioning unit, the combination of two air-passages; liquid-containing means; a plurality of air-permeable evaporative-liquid-holding pads, rotatable partly in the liquid-containing means and partly in the first passage; a plurality of air-permeable highly heat-absorbent non-hygroscopic pads, alternating with the other pads, one of these non-hygroscopic pads rotating partly in one portion and partly in another portion of the first passage, and the rest rotating partly in each passage; means for impelling air in the first passage first through the first-mentioned non-hygroscopic pad, and then alternately through the evaporative-liquid-holding pads and the non-hygroscopic pads, there being at least one other non-hygroscopic pad between the two occurrences of the non-hygroscopic pad which twice enters this same passage; and means for impelling air in the second passage through the successive non-hygroscopic pads in that passage.

22. In an air-conditioning unit, the combination of two air-passages; liquid-containing means; a plurality of air-permeable fixed evaporative-liquid-holding pads, located in the first passage; means for constantly circulating evaporative liquid through said pads from and to the liquid-containing means; a plurality air-permeable highly heat-absorbent non-hygroscopic pads, alternating with the other pads, one of these non-hygroscopic pads rotating partly in one portion and partly in another portion of the first passage, and the rest rotating partly in each passage; means for impelling air in the first passage first through the first-mentioned non-hygroscopic pad, and then alternately through the evaporative-liquid-holding pads and the non-hygroscopic pads, there being at least one other non-hygroscopic pad between the two occurrences of the non-hygroscopic pad which twice enters this same passage; and means for impelling air in the second passage, through the successive non-hygroscopic pads in that passage, in the opposite direction from that in which the air in the other passage passes through these same pads.

23. In an air-conditioning unit, the combination of: two air-passages; means for impelling a separate stream of air through each passage; means for adiabatically cooling the air-stream in the first passage by the evaporation of water

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therein, said cooling means comprising a pad of air-permeable water-absorbing material treated with a member of the class consisting of water-soluble surface-active agents characterized by their special ability to lower the surface tension between water and air; an air-permeable non-hygroscopic highly heat-absorbent pad, rotatable partly in each air-passage; means for rotating this pad; and means for continually supplying water to the cooling pad; whereby the rotating pad is cooled by the cooled air in the first passage, and in turn anhydrously cools the air in the second passage.

24. Evaporative air-cooling means for an air-conditioning unit, comprising: a pad of air-permeable water-absorbing material treated with a member of the class consisting of water-soluble surface-active agents characterized by their special ability to lower the surface tension between water and air; means for continually supplying water to this pad; and means for impelling through this pad the air to be cooled thereby.

25. Evaporative air-cooling means according to claim 24, further characterized by the fact that the treating agent is odorless, and has at atmospheric temperature a vapor pressure well below atmospheric pressure.

26. Evaporative air-cooling means according to claim 24, further characterized by the fact that the treating agent is dioctyl sodium sulfosuccinate.

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