

Image Cover Sheet

CLASSIFICATION

SYSTEM NUMBER

502976

UNCLASSIFIED



TITLE

A STUDY OF A NON-REGENERATING DRY AIR MICROCLIMATE SYSTEM

System Number:

Patron Number:

Requester:

Notes:

DSIS Use only:

Deliver to:

A STUDY OF A
NON-REGENERATING
DRY AIR MICROCLIMATE
SYSTEM

April 1992

Paul Browne
Exotemp Ltd.

SSC CONTRACT NO. W7711-1-7136

Index

1. Foreword
2. Cooling Performance Versus Air Flow Characteristics
3. Estimated Performance of a Torso Cooling Garment
4. Means of Drying air
 - 4.1 Solid Desiccants
 - 4.1.1 Heat Evolution from Adsorption
 - 4.1.2 Conditions Within the Adsorbent Bed
 - 4.1.3 Bed Dimensions
 - 4.1.3.1 Volume
 - 4.1.3.2 Configuration
 - 4.1.3.3 Pressure Drop
 - 4.2 Liquid Desiccants
 - 4.3 Auxiliary Components
 - 4.3.1 Heat Exchanger
 - 4.3.2 Fans
5. A Design Exercise
6. Conclusions
7. Comments

1. FOREWORD

There are some circumstances where it would be desirable to be able to entirely eliminate mass transfer between defence personnel and their environment. A butyl suit would do the job well enough, but in warm conditions heat stress would severely limit endurance.

This study evaluates the practicality of reactivating the user's natural evaporative cooling by drying and distributing the air inside such a suit through a closed loop configuration. The concept is not to cool the air below-ambient, but rather to dry it well with a desiccant, hopefully eliminating the need for refrigerating the air. Most of this study focuses on systems which do not regenerate their desiccant, but rather have an endurance limited by the amount of desiccant carried.

2. COOLING PERFORMANCE VERSUS AIR FLOW CHARACTERISTICS

The net cooling rate of a person subjected to a flow of hot dry air will be the difference between the heat transfer rates resulting from two competing mechanisms, convective and evaporative heat transfer.

If the incoming air temperature is above skin temperature, heat will be transferred to the body by convection. In this case:

$$C = h_c f_{CL} A (T_a - T_s)$$

where

C = convective heat transfer rate, W

h_c = convective heat transfer coefficient, $W m^{-2} \cdot C^{-1}$

$$= 8.3 v^{0.6} \text{ (for } v > 0.2 \text{ ms}^{-1}\text{),}$$

where v = velocity, ms^{-1}

Note that this equation gives the heat transfer coefficient for the whole body subjected to a flow of air from one direction. It may be somewhat pessimistic for a case where special effort is taken to achieve air flow over the skin surface through cooling garment design.

f_{CL} = multiplier to h_c for clothing between body and air stream

$$= \frac{1}{1 + 0.45 I_{CLO}}$$

Note: Reference is made to "Personal Cooling Garments:
A Review" - S.A. Konz, Ph.D.

ASHRAE AT-84-09 No. 4

2

I_{CLO} = insulation value of clothing between body and air stream, CLO

A = skin surface area, m^2

T_a = air temperature, $^{\circ}C$

T_s = skin temperature, $^{\circ}C$

If the vapour pressure of the water on the skin is above the partial pressure of water vapour in the adjacent air, water will evaporate from the skin surface, resulting in evaporative cooling. In this case:

$$E = h_e f_{PCL} A w (P_s - P_{wa})$$

where

E = evaporative heat transfer rate, W

h_e = evaporative heat transfer coefficient

$$= 2.2 h_c$$

F_{PCL} = multiplier to h_e for permeable clothing between body and air stream

$$= \frac{1}{1 + 0.415 I_{CLO}}$$

A = skin area, m^2

w = ratio of actual evaporation to maximum, possible. Also termed ratio of wetted to dry skin.

P_{wa} = vapour pressure of water in the air, mm of Hg

P_s = vapour pressure of water on skin, mm of Hg

To estimate the actual performance of the proposed system, we will assume that the hot, dry air is introduced under any clothing, directly on the skin. In addition the following values are assumed:

w = 0.7, reputedly a reasonable maximum for someone sweating profusely

T_s = $35^{\circ}C$, a skin temperature at which sweating will commonly be substantial

P_s = 42.2 mm of Hg

Net Cooling is thus

$$\begin{aligned}
 Q &= E - C \\
 &= h_c A (0.7) (P_{wa} - 42.2) \\
 &\quad - h_c A (T_a - 35) \\
 &= [2.2(0.7)(P_{wa} - 42.2) - (T_a - 35)] h_c A \\
 &= 8.3 v^{0.6} A [1.54 P_{wa} - 64.988 - T_a + 35] \\
 \frac{Q}{A} &= 8.3 v^{0.6} (1.54 P_{wa} - T_a - 29.99)
 \end{aligned}$$

This net heat transfer rate may be taken as the local rate achieved for the humidity, velocity, and temperature of air over a specific point on the skin. To model an actual application, we need a simplified model of the air distribution garment and the human's surface.

3. ESTIMATED PERFORMANCE OF A TORSO COOLING GARMENT

For this exercise we assume that a transverse flow of air is applied across the upper torso only, such as might be used to cool aircrew. The cooled chest is taken as a cylinder 1 m in circumference, 30 cm high. Air is introduced through a ring plenum at the top, and collected through a ring plenum at the bottom, per figure 1.

The velocity will be,

$$v = \frac{Q}{1 \times t}, \text{ where } Q = \text{air flow in m}^3 \text{ s}^{-1} \\
 t = \text{vest air channel thickness, m}$$

The air temperature and humidity will change as the air moves downward. For a ring around the chest at a distance l from the top of the garment,

$$\begin{aligned}
 dA &= dl \\
 dC &= 8.3 v^{0.6} dl (T_a - 35) \\
 \text{and } dT_a &= - \frac{dC(0.239)}{\dot{m} C_p} \\
 &= 0.24 \frac{\text{Cal}}{\text{g } ^\circ\text{C}}
 \end{aligned}$$

where \dot{m} = mass flow of air in g s^{-1}

C_p = specific heat of air at constant pressure

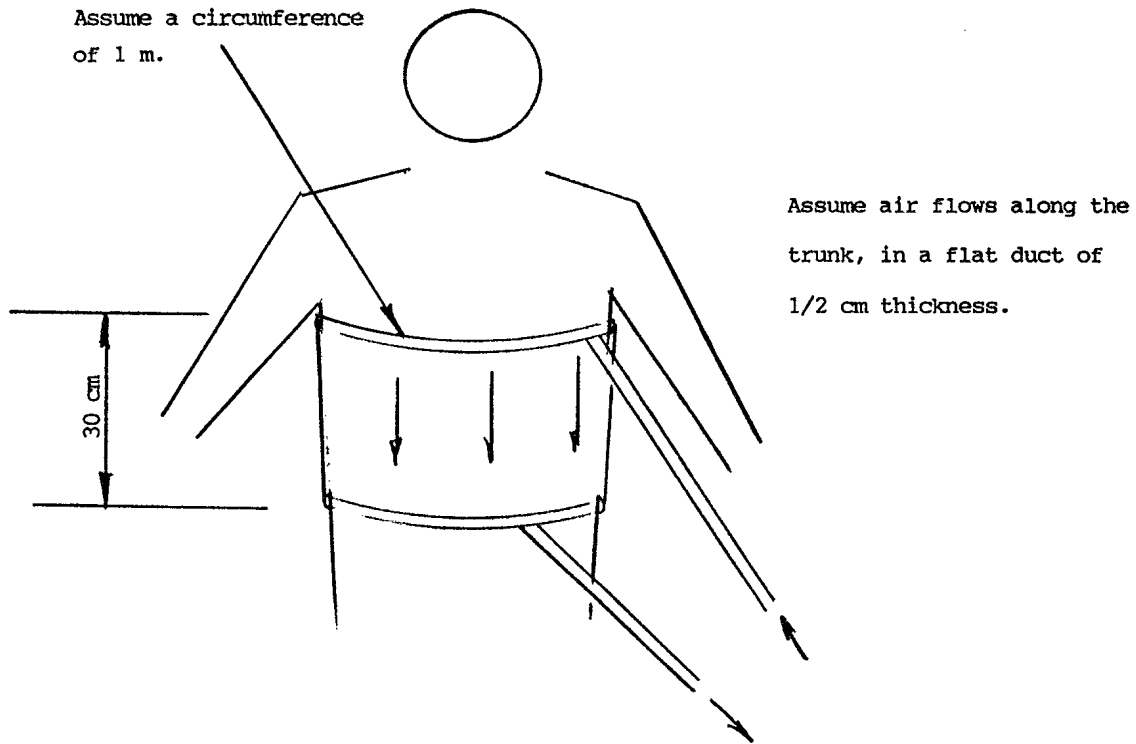


FIGURE 1 - AIR VEST MODEL

4

and $\dot{m} = Q \gamma$, where $\gamma =$ specific weight of air
 $\approx 1121 \text{ g m}^{-3}$ at 40°C and 1 atmosphere

$$\begin{aligned} \therefore dT_a &= - \frac{(0.239)(8.3)(T_a - 35)}{(11.21)(0.24)Q} \left(\frac{Q}{t}\right)^{0.6} dl \\ &= - \frac{0.00734}{Q^{0.4} t^{0.6}} (T_a - 35) dl \end{aligned}$$

$$l = \int_0^l dl = \int_{T_{ai}}^{T_a} - \frac{136.24 Q^{0.4} t^{0.6}}{T_a - 35} dT_a$$

$$l = - 136.24 Q^{0.4} t^{0.6} \ln \left(\frac{T_a - 35}{T_{ai} - 35} \right)$$

$$\frac{T_a - 35}{T_{ai} - 35} = e^{- \frac{l}{136.24 Q^{0.4} t^{0.6}}}$$

$$T_a = (T_{ai} - 35) e^{- \frac{l}{136.24 Q^{0.4} t^{0.6}}} + 35$$

So at $l = 0.3 \text{ m}$,

$$T_a = \frac{(T_{ai} - 35)}{e^{\frac{l}{136.24 Q^{0.4} t^{0.6}}}} + 35$$

$$\text{and } C = (T_{ai} - T_a) \times \frac{0.24}{0.239} \times Q (1121)$$

$$= 1126 Q \left[T_{ai} - \frac{T_{ai} - 35}{e^{\frac{l}{136.24 Q^{0.4} t^{0.6}}}} - 35 \right]$$

$$= 1126 Q (T_{ai} - 35) \left(1 - \frac{1}{e^{\frac{l}{136.24 Q^{0.4} t^{0.6}}}} \right)$$

5

So if

$$t = 0.5 \text{ cm} = 0.005 \text{ m}$$

$$l = 0.3 \text{ m}$$

$$Q = 20 \text{ cfm} = 0.0094 \text{ m}^3/\text{sec}$$

$$T_{ai} = 60^\circ\text{C}$$

$$C = 1126 \times 0.0094 (60 - 35) \left(1 - \frac{l}{\frac{0.3}{136.24 \times 0.0094^{0.4} \times 0.005^{0.6}}} \right)$$

$$= 76.67 \text{ Watts}$$

$$T_{ao} = \frac{60 - 35}{\frac{0.3}{136.24 \times 0.0094^{0.4} \times 0.005^{0.6}}} + 35$$

$$= 52.176^\circ\text{C}$$

The humidity of the air will also increase,

$$\begin{aligned} dE &= 2.2 h_c dA w (P_s - P_{wa}) \\ &= 2.2 (8.3) \frac{Q^{0.6}}{t^{0.6}} (0.7) (42.2 - P_{wa}) d\ell \end{aligned}$$

How will P_{wa} change with E?

The change of enthalpy from liquid water on the skin to vapour in the air stream is calculated as follows:

$$h_v = 1062 + 0.44 t_a \quad , \text{ where}$$

$$\begin{aligned} h_v &= \text{enthalpy of vapour, BTU/lb} \\ t_a &= \text{air temperature, } ^\circ\text{F} \end{aligned}$$

$$h_f = t_f - 32 \quad , \text{ where}$$

$$\begin{aligned} h_f &= \text{enthalpy of liquid, BTU/lb} \\ t_f &= \text{skin temperature, } ^\circ\text{F} \end{aligned}$$

6

$$\begin{aligned}
 \text{if } t_s &= 55^\circ\text{C} = 131^\circ\text{F} \\
 t_f &= 35^\circ\text{C} = 95^\circ\text{F} \\
 \Delta h &= h_v - h_f \\
 &= 1119.6 - 63 \\
 &= 1056.6 \quad \text{BTU/lb.} \\
 &= 2459.5 \quad \frac{\text{Watt sec}}{\text{g}}
 \end{aligned}$$

This will be taken as a constant for the temperatures we will see in the air stream. The vapour pressure can also be expressed as proportional to the quantity of water evaporated per unit volume:

At say 100°F , saturated steam has a specific volume of

$$350.4 \frac{\text{ft}^3}{\text{lb}} \Rightarrow 45.716 \text{ g/m}^3$$

It also has a vapour pressure of 1.9325 in Hg \Rightarrow 49.09 mm Hg.

So an estimate of dP_{wa} as a function of E and Q would be

$$\begin{aligned}
 dP_{wa} &= \frac{49.09 \text{ mm Hg}}{45.716 \text{ g/m}^3} \frac{dE \text{ Watts}}{2459.5 \frac{\text{Watt sec}}{\text{g}} \text{ Qm}^3/\text{sec}} \\
 &= \frac{dE}{2290.46 \text{ Q}} \quad , \text{ mm Hg}
 \end{aligned}$$

and if we assume an initial relative humidity of 10%,

$$\text{ie, } P_{wai} = 14.94 \text{ mm Hg}$$

7

$$\begin{aligned} \therefore dP_{wa} &= \frac{(2.2)(8.3)(0.7)Q^{0.6}(42.2 - P_{wa})}{2290.46 Q t^{0.6}} d\ell \\ &= \frac{1}{179.2 Q^{0.4} t^{0.6}} (42.2 - P_{wa}) d\ell \end{aligned}$$

$$\ell = \int_0^{\ell} d\ell = 179.2 Q^{0.4} t^{0.6} \int_{P_{wai}}^{P_{wa}} \frac{dP_{wa}}{42.2 - P_{wa}}$$

$$\ell = 179.2 Q^{0.4} t^{0.6} \left[-\ln \left(\frac{42.2 - P_{wa}}{42.2 - P_{wai}} \right) \right]$$

$$-\frac{\ell}{179.2 Q^{0.4} t^{0.6}}$$

$$\frac{42.2 - P_{wa}}{42.2 - P_{wai}} = e$$

$$-\frac{\ell}{179.2 Q^{0.4} t^{0.6}}$$

$$P_{wa} = (P_{wai} - 42.2) e^{-\frac{\ell}{179.2 Q^{0.4} t^{0.6}}} + 42.2$$

if $\ell = 0.3 \text{ m}$

$$Q = 0.0094 \text{ m}^3/\text{sec}$$

$$t = 0.005 \text{ m}$$

$$P_{wai} = 14.94 \text{ mm Hg, at 10\% RH and } 60^\circ\text{C,}$$

$$P_{wao} = \frac{14.94 - 42.2}{\frac{0.3}{179.2 \times 0.0094^{0.4} \times 0.005^{0.6}}} + 42.2$$

$$= 21.18 \text{ mm Hg,}$$

This is about 20% RH for the exit temperature of 52.7°C .

$$\begin{aligned} \text{and } E &= 2290.46 Q (P_{wao} - P_{wai}) \\ &= 2290.46 \times 0.0094 (21.18 - 14.94) \\ &= 134.35 \text{ Watts} \end{aligned}$$

$$\text{So Net Cooling} = E - C$$

$$= 134.35 - 76.67$$

$$= 57.68 \text{ Watts}$$

Clearly not enough cooling for aircrew.

Note that velocity over skin is:

$$v = \frac{Q}{t} = \frac{.0094}{0.005} = 1.88 \text{ m/s}$$

This should be about as high as we should realistically go. We probably won't gain anything by increasing velocity. However, if the relative humidity of the inlet air is lower, the performance improvement is dramatic.

Suppose

$$P_{wa_i} = 0 \text{ mm Hg (0\% RH)}$$

$$\text{then } P_{wa_o} = 9.66 \text{ mm Hg (9.4 RH)}$$

$$E = 207.98 \text{ Watts}$$

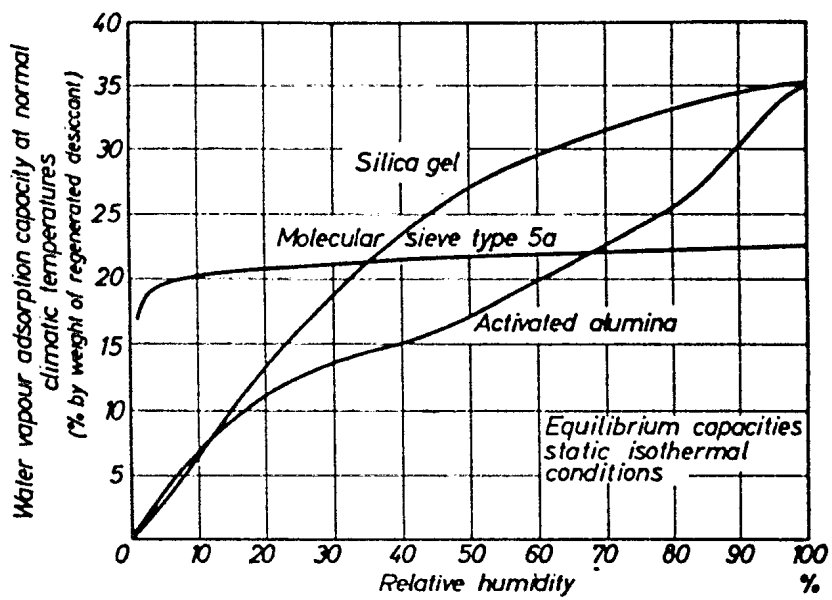
Net Cooling = 131.31 Watts, probably close to enough.

4. MEANS OF DRYING AIR

4.1 Solid Desiccants

Solid adsorbents do not change chemically or physically during the sorption process. There are many such materials, for example alumina, activated carbon, certain clays, silica gel and molecular sieves. Apparently only one of these types, molecular sieve is capable of achieving the relative humidities indicated in the preceding section. Figures 2 and 3 show that other solid adsorbents can hold more water per unit weight than molecular sieves, at temperatures below 25°C and for relative humidities in excess of 35%. However for the conditions we expect, 15% to 5% RH and 50 - 70°C, molecular sieves are much better. In fact their capacity of 18% to 26% water by weight is rather encouraging.

Absorbents do change physically or chemically during the sorption process. There are also examples of solid absorbents. CaO is one. It reacts with water to produce CaH₂O₂. This reaction will proceed to quite low vapour pressures, but unfortunately a large amount of heat is evolved, much more than the latent heat of condensation of the water. The stuff also increases in volume dramatically, and the speed of reaction is normally quite slow when used in desiccant applications. Mg (ClO₄)₂ is a possibility, but unfortunately it is a strong oxidizing agent and can form explosive mixtures. Examples of other solid absorbents are deliquescent salts, like CaCl₂. Since these form a liquid when exposed to moist air, they are considered later under



Static water vapour adsorption capacities of activated alumina, silica gel and 5A molecular sieve

FIGURE 2

EQUALIBRIUM WATER ADSORPTION CAPACITY AS FUNCTION OF TEMPERATURE AT 10% AND 50% OF RELATIVE HUMIDITY

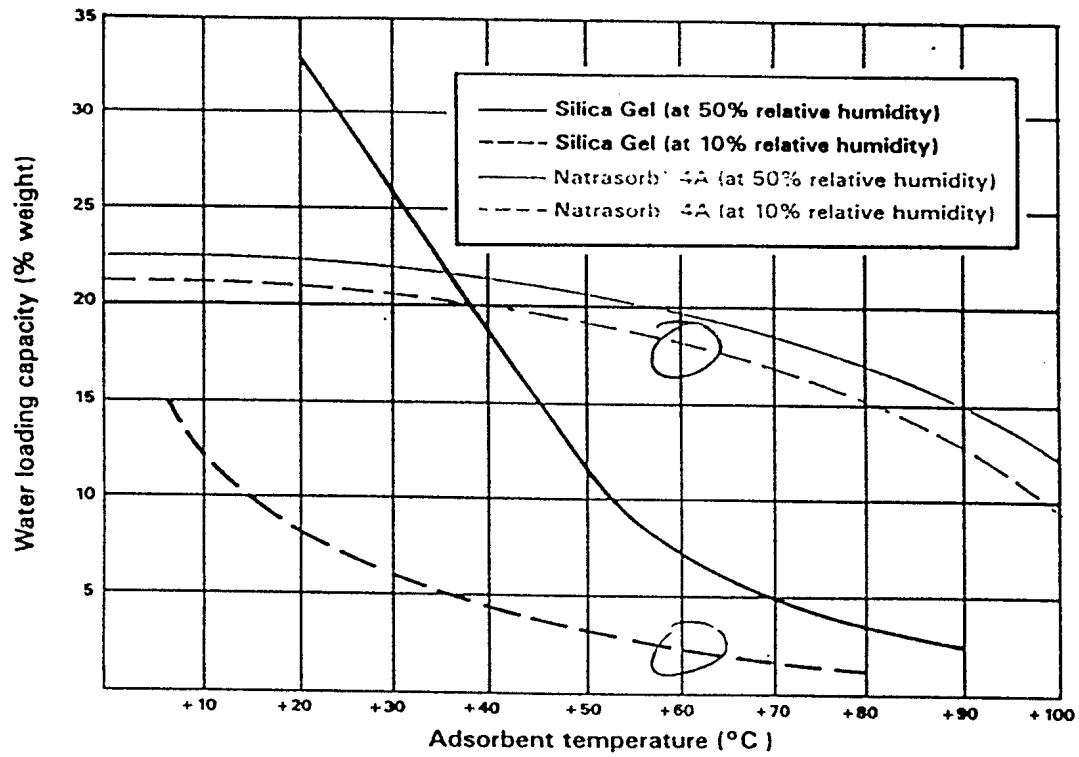


FIGURE 3

liquid absorbents.

I have been unable to find a solid absorbent which appears more promising than molecular sieves for this application.

4.1.1 Heat Evaluation from Adsorption

When water vapour is captured by a molecular sieve, heat equivalent to about 170% of the heat of condensation is evolved. In a bed of solid granular material, it is difficult to cool the bed by conduction to the ambient. We can assume that the total heat evolved is deposited into the air stream as sensible heat.

In a single bed, if 20 cfm of air is dehumidified from 9.4% RH to 0, RH, and if the inlet air temperature is taken as,

$$T_{ao} = 53^{\circ}\text{C}$$

the rate of heat evolved from sorption of water vapour is about 70% more than the rate of heat removed from the body,

$$\begin{aligned} S &= 1.7 E, & \text{where} & & S &\text{ is the rate of heat} \\ & & & & & \text{evolved from} \\ & & & & & \text{adsorption} \\ &= 1.7 \times 207.98 \\ &= 353.57 \text{ Watts} \end{aligned}$$

The air temperature rise across the bed is,

$$\begin{aligned} \Delta t &= \frac{353.57 \text{ Watts} \times 0.239 \frac{\text{Cal/sec}}{\text{Watt}}}{0.24 \frac{\text{Cal}}{\text{g}^{\circ}\text{C}} \times 1121 \frac{\text{g}}{\text{m}^3} \times 0.0094 \frac{\text{m}^3}{\text{sec}}} \\ &= 33.414 \text{ C}^{\circ} \end{aligned}$$

Although this increased air temperature will impede the action of the molecular sieve somewhat, it will not be a show stopper, especially early on in the cycle. Since the desiccant nearest the inlet will saturate first, dry desiccant downstream will be able to achieve very low vapour pressures well on into the cycle, even at the higher temperature. In fact, a study of the adsorption isotherms for type 13X molecular sieve presented in figure 4, suggests that very low humidities (less than 1%) might be achieved in an unsaturated, uncooled bed of 13X molecular sieve.

WATER ADSORPTION ISOTHERM NATRASORB® 4-A

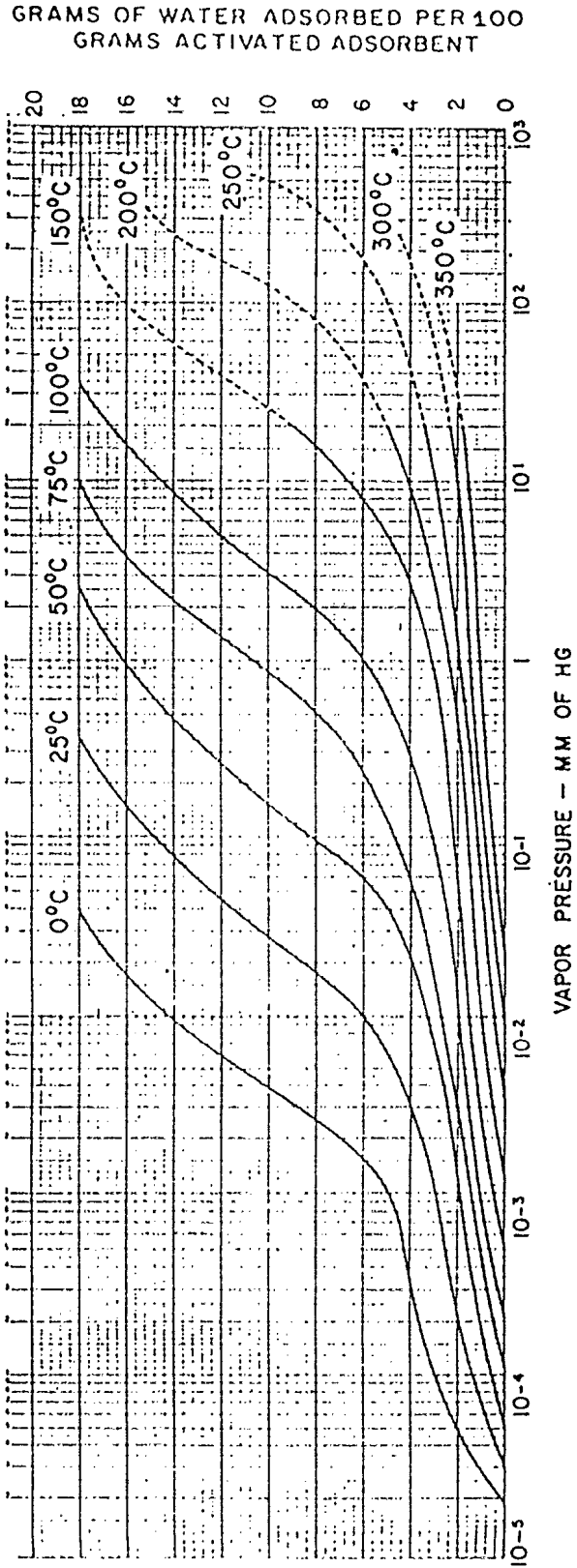


FIGURE 4

4.1.2 Conditions within the Adsorbent Bed

Mass transfer within a packed bed of adsorbent is not uniform along the bed depth. As the cycle progresses saturation of the desiccant is seen first at the farthest point upstream. Downstream from this is the mass transfer zone. The thickness of the mass transfer zone depends on the characteristics of the gas mixture being dried, the properties of the adsorbent, and the velocity of approach to the bed. The more rapid the adsorption, and the slower the approach velocity, the smaller is the mass transfer zone. For some arrangements, particularly for adsorbents like alumina or silica gel, the mass transfer zone will be a substantial portion of the bed depth, and "breakthrough" of the water vapour at lower concentrations at first, but gradually increasing to unacceptable levels later, will terminate the useful life of the bed. Fortunately molecular sieves tend to adsorb very quickly, and the water they adsorb is sharply, not gradually, affected by vapour pressure. The net result is that the active zone for molecular sieves tends to be thinner. There will commonly be an inactive zone of dry desiccant downstream of the active zone. The useful life of the adsorbent bed will finish when the active zone has moved from farthest point upstream to farthest point downstream.

Such a bed will produce very dry air until breakthrough occurs, after which the exit humidity will rise quite rapidly to equal the inlet humidity. The "breakthrough capacity" is closer to the equilibrium capacity.

4.1.3 Bed Dimensions

4.1.3.1 Volume

The bed volume can be estimated based on the total amount of water to be adsorbed.

Assume the adsorbent must last for 4 hours, if

$$E = 207.98 \text{ Watts}$$

$$\text{and } \Delta h = \frac{2459.5 \text{ Watts sec}}{\text{g}}$$

then in 4 hours,

$$\text{Total Water Adsorbed} = \frac{4 \times 3600 \text{ sec} \times 207.98 \text{ Watts}}{2459.5 \frac{\text{Watts sec}}{\text{g}}}$$

$$= 1217.7 \text{ g}$$

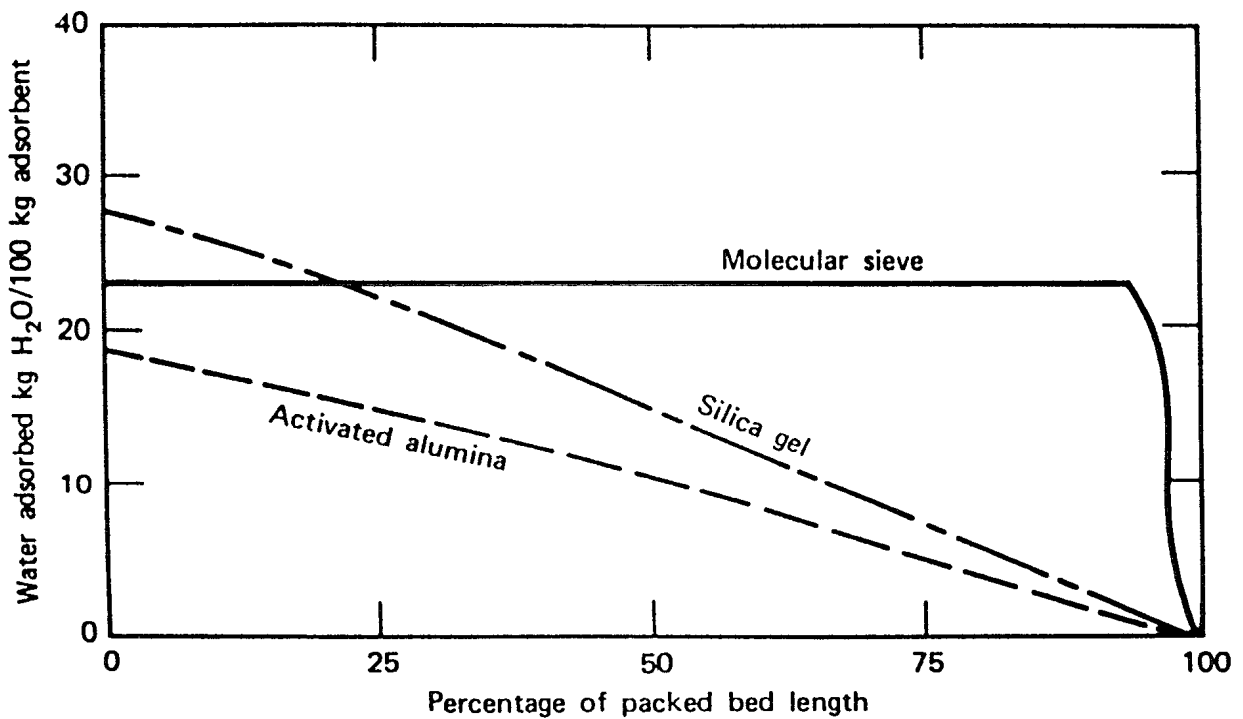


FIGURE 6 Position of water front in packed bed of adsorbent during dynamic dehydration. Conditions: 50% rh; 10.2 cm/s air; particle size = ca 0.167 cm; temperature = 25°C; contact time = 1.7 s.

molecular sieve 13X will hold about 22% of its dry weight in water, so,

Weight of Dry Molecular Sieve = 5535 g (12.2 lbs)

at 45 lbs/ft³, the bed volume is 0.271 ft³, 468 in³, 7.68ℓ.

The contact time will be 0.81 seconds, a reasonable value.

4.1.3.2 Configuration

To reduce pressure drop, the bed thickness should be kept small. If bed thickness is 4 cm, the total bed area will be 1920 cm². The bed might be arranged in 4 layers per the attached sketch. In this case each layer would be 21.9 cm square.

4.1.3.3 Pressure Drop

For a mass flow of air of

$$20 \frac{\text{ft}^3}{\text{min}} \times 0.067 \frac{\text{lb}}{\text{ft}^3} \times 60 \frac{\text{min}}{\text{hr}} = 80.4 \frac{\text{lb}}{\text{hr}}$$

and a bed area of 2.07 ft², the mass velocity will be,

$$\frac{80.4}{2.07} = 38.9 \frac{\text{lbs}}{\text{hr ft}^2}$$

Figure 5 shows that the bed resistance for 1/8" diameter molecular sieve will be

$$0.005 \text{ lbs/in}^2 \text{ per ft of bed.}$$

In our case the bed pressure drop will be

$$0.005 \times 27.68 \times \frac{4}{30.48} = 0.018" \text{ H}_2\text{O} \quad \text{quite low!}$$

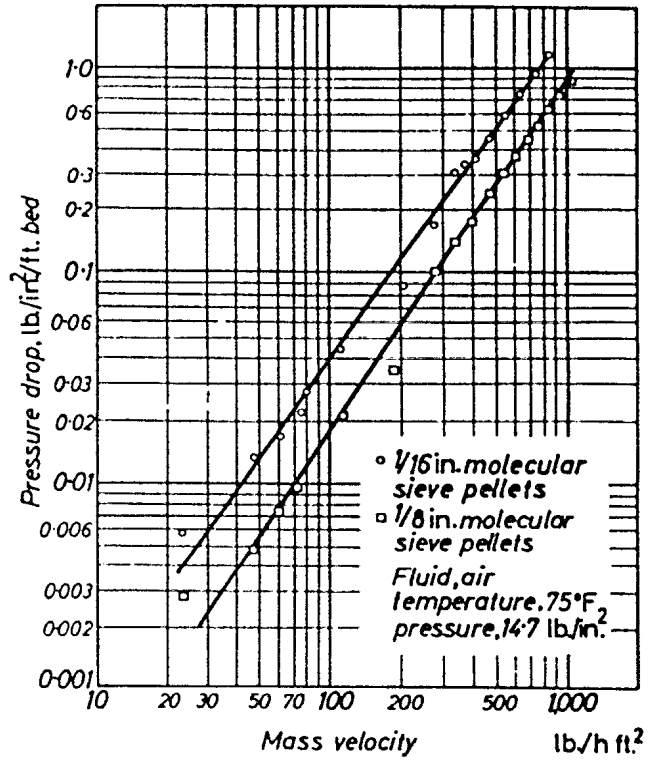
4.2 Liquid Absorbents

The vapour pressure of water over solutions can be lowered dramatically by many organic and inorganic compounds. For example, glycols have been used as desiccants to dry air. Other possible materials are rather nasty; sulphuric acid and sodium hydroxide work well. The more benign chemicals to consider are inorganic salts, like LiCl, CaCl₂, LiBr, and LiI.

Of these, LiBr and LiI exhibit the greatest hygroscopic depression. In fact LiI will allow vapour pressures of less than 5 mm Hg over saturated solutions at less than 80% LiI concentration, if the temperature can be 90°C. In general though, liquid absorbent solutions have trouble producing vapour pressures as low as we would like for this

FIGURE 5

Resistance to air flow of molecular sieves. [By courtesy of Union Carbide Ltd. (The term 'Union Carbide' is a trade mark of Union Carbide Corporation)]



application.

One theoretical advantage of liquids is that they may be cooled by circulating them through a heat exchanger, which makes the absorption process more isothermal. Since for many sorbents, the equilibrium quantity of water sorbed and the resultant vapour pressure is quite adversely affected by increased temperature, cooling in this way should result in a smaller amount of desiccant required and a drier exiting gas.

A second potential advantage concerns regeneration. As liquids, such solutions can be moved through evaporators and heat exchangers for reconditioning.

Another possible advantage is that contact with the air stream might be made in through a liquid spray or in a counterflow packed bed. This would increase the surface area, and hence the rate of absorption.

The first advantage, that of a more isothermal absorption, might be significant in comparison with amorphous adsorbents like carbon, silica gel, or alumina. However molecular sieves are far less sensitive to increased temperature and this property pretty well makes adiabatic sorption fully acceptable. In fact, adiabatic sorption is probably preferable with molecular sieves since it results in a larger temperature differential to drive heat transfer to the ambient.

Concerning regeneration, solid adsorbents can also be regenerated, usually on a batch basis through a number of valves to direct air flow.

The last supposed advantage is more than offset by the relatively high surface areas of molecular sieves, and the difficulty of removing liquid droplets from the air stream. We have found no promising means of accomplishing this. Experiments have been conducted with hydrophobic porous plastic sheets. The results did not inspire confidence for an application where the equipment might be inverted. Tests were also done with a solid membrane of perfluorosulfonic acid polymer, a material which can incorporate water into its structure on one side and out again on the other. Complete separation of the absorbent from the air stream is thus achieved. Unfortunately the surface area would be necessarily small and the resultant reaction slow. The membrane is also too expensive at \$750/m².

I do not recommend pursuing the design of systems which use liquid absorbents. They are more likely to be difficult to make reliable. Their performance will not likely be as good as systems based on molecular sieves.

4.3 Auxiliary Components

The auxiliary components are discussed below assuming that a molecular sieve is used.

4.3.1 Heat Exchanger

In the proposed system, this component cools dry air before it flows to the body, taking the dry air temperature from 86.4°C to 60°C. The heat transferred to the ambient is,

$$\begin{aligned} & 10.537 \frac{\text{g}}{\text{sec}} \times (86.4 - 60) \text{ C}^\circ \times 0.24 \frac{\text{cal}}{\text{gC}^\circ} \\ & = 66.8 \frac{\text{cal}}{\text{sec}} \\ & = 279.68 \text{ Watts} \end{aligned}$$

This suggests a fairly small and light heat exchanger. A good arrangement would be a counter flow or perhaps a cross flow flat plate design, with thin metal plates, essentially aluminum foil.

One would expect that an overall heat transfer coefficient of 10 BTU/hr ft²F° (0.0057 Watts/C° cm²) could be achieved with low pressure drop. To illustrate the size,

$$\text{Heat Transferred} = 279.68 \text{ Watts}$$

If the temperature differential averages only 10C°, the area needed is

$$\frac{279.68}{10 \times .0057} = 4907 \text{ cm}^2$$

For a spacing of 3 mm between plates, a cube 11.4 cm on each side would do.

Since it will likely be practical to make the pressure drop through the heat exchanger very small, an axial fan would be called for on the ambient air side. To make for a small heat exchanger and to make cross flow a possibility, the ambient air flow might be set at twice the dry air flow, ie. 50 cfm. Ambient air would enter at 50°C and exit at 63.2°C.

4.3.2 Fans

The ambient air fan is simply dealt with. Against say less than 1/8" H₂O pressure, a propeller type fan should do the job while consuming less than 1/4 Amp at 28 VDC.

The dry air circulating fan will be more of a problem. In fact it's power consumption might spoil the system. For example, if the total pressure drop around the system were

5" H₂O (1.26 kPa), the power consumption of a Technofan EVCT0620A centrifugal blower would be 3.5 A at 28 VDC. The thing is also 145 mm x 150 mm dia. and weights 4.8 lb. If 60% of its power consumption went to heating air, 58.8 watts of heat would be added to the circulating air stream. Clearly it is quite important to reduce the total circuit pressure drop.

It should be possible to reduce the total circuit pressure drop to 1/4" H₂O. 20 cfm in a 2" hose results in a velocity of 917 ft/min and a velocity pressure of only 0.05" H₂O. If it were a smooth duct the pressure loss would be 1" H₂O per 100 ft. The velocity in the air vest is only 376 ft/min, for a velocity pressure of 0.01" H₂O. The loss through the desiccant bed is only 0.02" H₂O. The pressure drop through the heat exchanger should be smaller.

If 20 cfm at 1/4" can be achieved, the dry air flow could be handled by a fan like the Rotron Spartan, which consumes only 280 mA at 24 VDC. The Spartan is 4-3/4" square x 1-1/4" thick.

5. A DESIGN EXERCISE

Figure 7 shows one potential arrangement for a 4 hour air drying unit.

6. CONCLUSIONS

It should be practical to develop a closed circuit dry air microclimate system which will produce useful cooling in very high ambient temperatures. Such a system will produce essentially zero relative humidity air using 13X molecular sieve operating adiabatically. The heat of adsorption can be transferred to ambient air by a simple air to air heat exchanger following the absorber. The resulting warm, but bone dry air can be returned to the user where it will evaporate sweat with a useful net cooling effect. Somewhat surprisingly, the return air is not expected to be very moist, only about 10% RH.

If the adsorbent is not to be regenerated, then at 50°C ambient and 131 Watts net cooling, the weight of adsorbent necessary for a 4 hour endurance is 5.5 kg. During this time 1.2 kg of sweat will be adsorbed. Electrical power consumption might be as low as 14 Watts. At lower ambient temperatures the system's performance will improve dramatically.

7. COMMENTS

I am quite encouraged by these results. The degree of dryness achieved by molecular sieves was a surprise to me. The stuff will deliver air almost entirely devoid of moisture, and it hardly cares how hot it gets doing it.

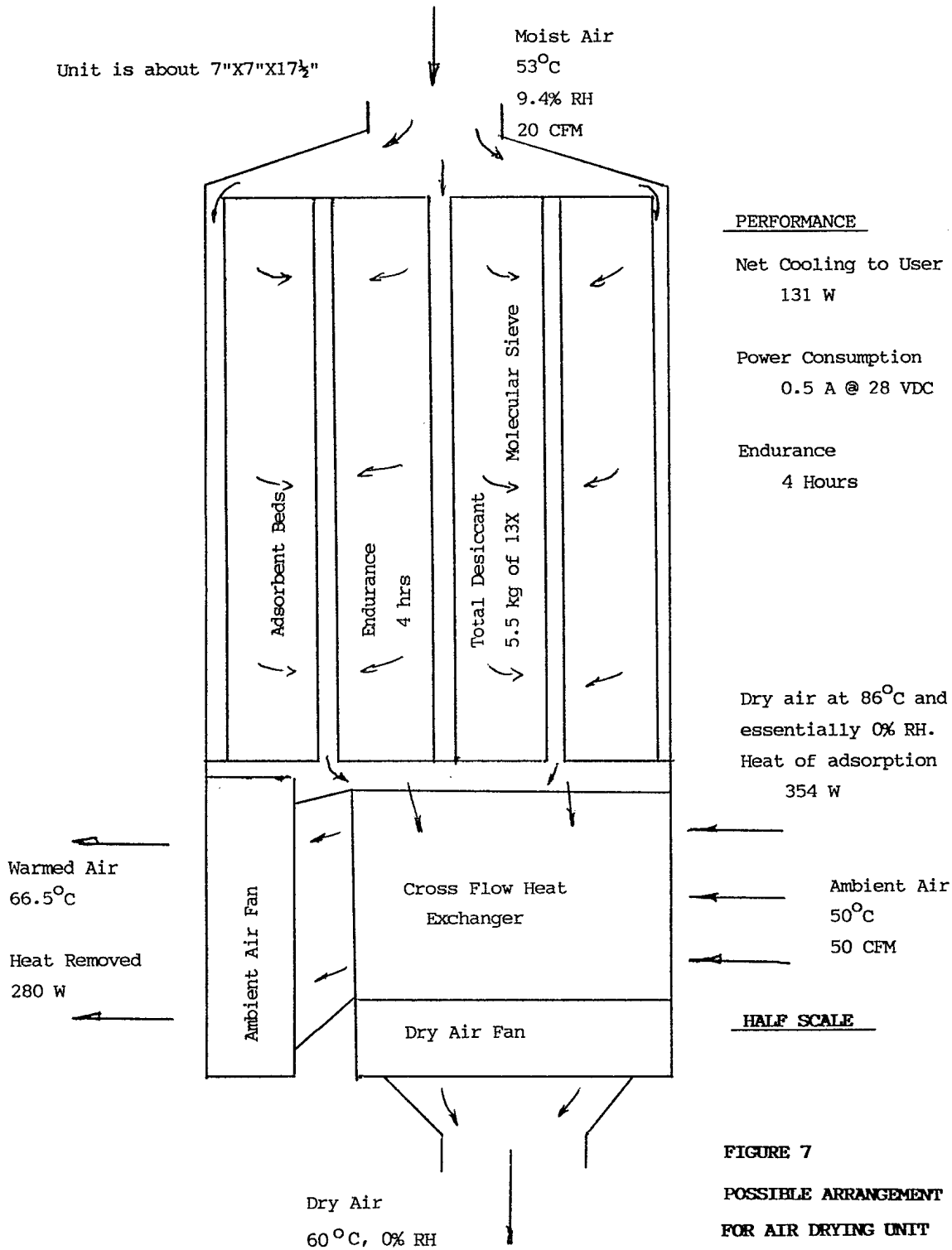


FIGURE 7
POSSIBLE ARRANGEMENT
FOR AIR DRYING UNIT

Humans don't usually experience a stiff breeze of really dry air at 60°C. An experiment should certainly be undertaken. The exercise which projects the performance of an air vest is also interesting. The technique used should be a useful design tool. Although this study assumes the air is collected directly after flowing through the vest, while it is still not very moist, there would be no harm and obvious benefit to allowing it to escape into the rest of the suit, where it would pick up additional moisture before collection.

If the weight of desiccant can be accepted, then non-regenerating systems are obviously simpler. A desiccant cartridge could be thrown away, or regenerated at some central site in a simple rig which would heat and evacuate it.

I suggest however that a closer look be taken at regenerating systems, not only for aircrew use, but perhaps for man-portable applications. In this case a fuel like diesel oil might provide the heat, and thermoelectrics might produce the small amount of fan power needed.

The water from the regeneration could be drunk. The numbers in this study indicate a diesel fuel consumption of about

$$\frac{3}{70\% \text{ efficiency of combustion heat capture}} \times \frac{353 \text{ Watts}}{12300 \text{ Watts hrs}} = 0.12 \text{ kg/hr}$$

at 3 x 121 Watts = 393 Watts of cooling.

Karl Chuang and Associates Inc.
14717 - 45 Avenue
Edmonton, Alberta
T6H 5R4
February 23, 1992

Mr. Paul Browne
Exotemp Ltd.
320 Boundary Road
Pembroke, Ontario
K8A 6W5
FAX: (613) 735-3814

Dear Paul,

Review of Processes for Cooling Suit Applications

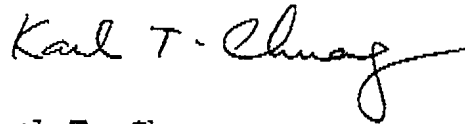
1. Liquid desiccants contained in membrane: This system is very inefficient at near ambient temperatures because the driving force at these conditions is too low to transport a large amount of water. The driving force is the difference in water vapour pressure across the membrane. The value at 100% R.H. is less than 0.12 atm at 50°C. It is, however, possible to transport the desired amount of water if a large membrane surface area and a high gas velocity are provided. In this case the weight of equipment can be excessive. Besides, when water vapour is absorbed by the liquid desiccant, the heat of absorption (equals the latent heat of vapourization for water) must be removed. If you feel that this system warrants further study, please provide me with the details of the desiccant and the air blower capacity (i.e. flowrate and pressure drop). I can estimate the required membrane area.

2. Solid reactants: Your suggestion on the use of CaO or BaO was evaluated. The absorption capacity is about 30 kg H₂O/100 kg reactant. However, the heat of reaction is 26 kcal/mole water, which is much higher than the heat of condensation (9.7 kcal/mole water). The feasibility of this system seems to depend on the efficiency of heat removal.

3. Solid adsorbents: Water vapour can be adsorbed on several solid materials. The most efficient adsorbents are zeolites, silica gels and activated carbons. After reviewing the characteristics of these adsorbents, I have come to the conclusion that you can use activated carbon for removing moisture down to 50% R.H. and then use 5A zeolite for 50% to 10% R.H. Please see the adsorption capacity shown in the figure I sent you two weeks ago. For zeolite, the adsorption capacity may drop 20% when the temperature is raised from 25°C to 50°C. Since the adsorption on activated carbon may take a different mechanism than the zeolite, you should check with the supplier for capacity at 50°C. I may also measure the capacity for you if design data are not available. Please note that you need to remove the heat of adsorption in this system so that the adsorbent temperature will not increase significantly.

I hope that the above summary is sufficient for your preliminary evaluation. Please let me know if you need additional information.

Sincerely Yours,



Karl T. Chuang

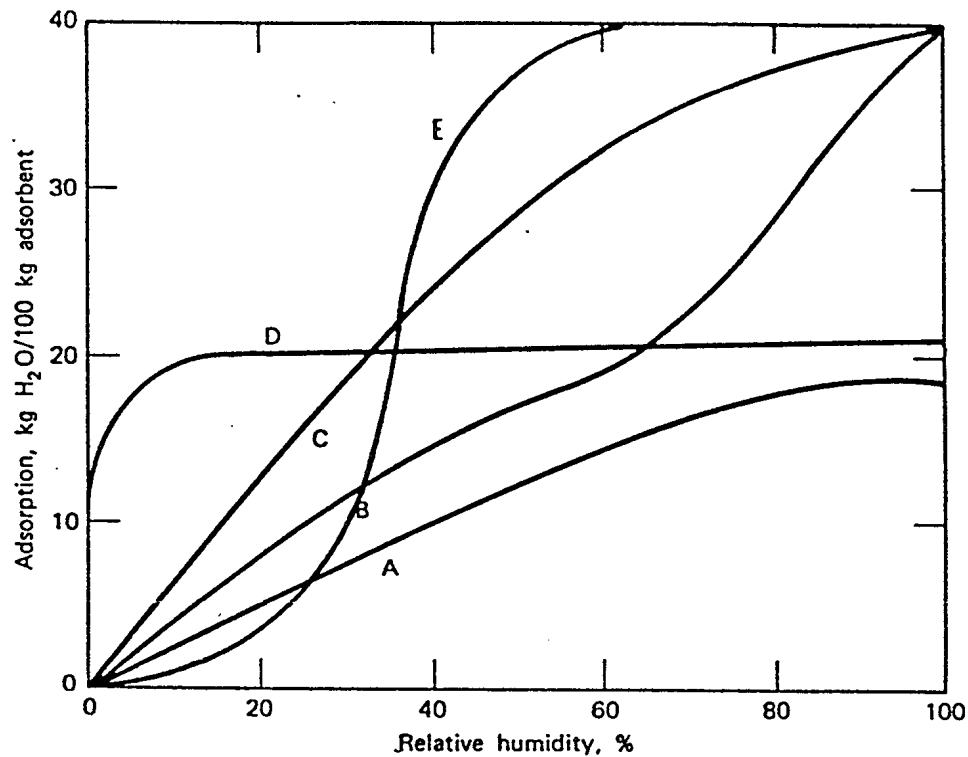


Figure 2.2 Equilibrium sorption of water vapor from atmospheric air at 25°C on (A) alumina (granular); (B) alumina (spherical); (C) silica gel; (D) 5A zeolite; (E) activated carbon. The vapor pressure at 100% R.H. is 23.6 Torr. Sources: (D) Breck [14]; (E) Stoeckli; Kraehenbuehl, and Morel [15].

↑

Carbon, vol. 31, p 589 (1983)

#502976