DESICCANT MOISTURE EXCHANGE FOR DEHUMIDIFICATION ENHANCEMENT OF AIR CONDITIONERS

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ABSTRACT

This paper discusses the potential of an improvement to the typical vapor compression air conditioning cycle used to cool and dehumidify air. This improvement uses a desiccant to remove moisture from the saturated air exiting from the chiller coil of an air conditioning system, and then revaporating this moisture into the air prior to entering the chiller coil. When a high proportion of the overall load on the chiller coil is latent or moisture removal, this new process provides the opportunity for operational energy savings as well as the potential for reduced equipment cost.

THE PROBLEM OF MOISTURE (LATENT) LOAD

One function of buildings, commercial and residential, is to maintain a controlled balance of temperatures and humidity. In humid areas of the U.S. and for many commercial operations, the removal of moisture necessary for the humidity requirement represents a significant portion of the energy consumed. The reduction of nonrenewable energy consumption is a major national goal.

In occupied spaces, improvements in building design, insulation and equipment can reduce the demands of the environment, but the heat and moisture generated by occupants depend on building function and cannot be significantly altered (1). In addition, it is recognized that certain minimal levels of fresh air exchange are required to maintain health in occupied spaces. In humid environments, this fresh air may bring with it significant levels of moisture, upsetting the temperature moisture balance and reducing comfort. The natural response to this moisture discomfort is to reduce the temperature setting of the thermostat. In the residential market, as the structures are improved to reduce the sensible load, a need has been recognized to shift the air conditioner equipment operational heat removal from sensible to more latent as to better match the load and thus maintain a better balance for comfort. This is especially true in the hot-humid southeast. A high EER air conditioner for residential use is presently manufactured by the Dinh Corporation of Alachua, Florida, that uses heat pipes to assist moisture removal. There is a residential market for the incorporation of desiccants on air conditioner units to outperform heat pipes at a lower cost, however the first installed use of this improved air conditioner cycle with desiccants will most likely occur in the commercial sector where desiccants are commonly known and the potential for savings is greater. One potential commercial application where the humidity problem is familiar to most of us, is that of supermarkets - our comfort is affected because many times the markets are too cold. Supermarket loads have been studied in detail by Thermo Electron Corporation for the Gas Research Institute and some of their results are helpful in quantifying the problem.

"The space-conditioning loads on a supermarket are unique in that the latent-to-total load ratio is higher than for most other commercial buildings. This is because the refrigerated cases act as open air coolers, absorbing predominantly sensible energy from the store and rejecting it to the outside of the building through the condensers. Figure 1 presents the typical load profile for a supermarket in Miami which is open 24 hours a day. The latent-to-total ratio is at least 0.4 most of the time (2)".

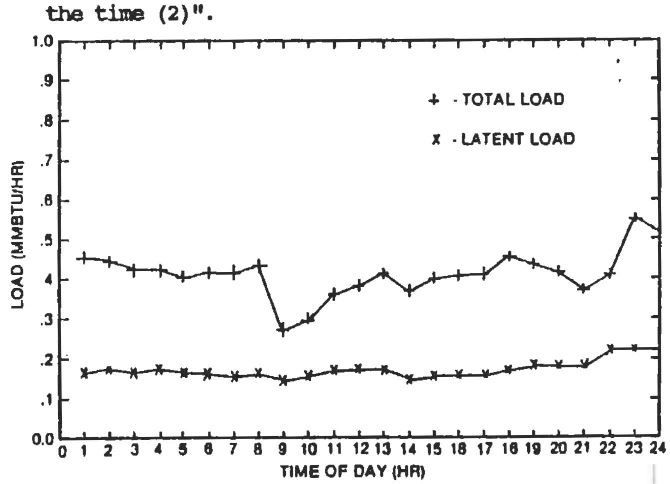

Figure 1 Hourly Latent and Total Load Variations on Space-Conditioning System, Miami-June

Engineers involved with the design of HVAC and food refrigeration systems for supermarkets are well aware of the load patterns resulting from the interaction of these systems. The space-cooling effect of the refrigerated cases can provide all of the sensible cooling required on the hottest days, and may result in a need for heating during periods when other conditioned spaces may require cooling.

REDUCED COIL TEMPERATURE

In all evaluations, we will use the ARI condition as state point 1, i.e., 26.7C (80°F) at 50% RH (11.0 g/kg) as shown in Fig. 2. Given the entering air condition to the coil as state point 1, as the apparatus due point (ADP) becomes colder, the slope of the line between 1 and 3 becomes steeper, producing a lower Sensible Heat

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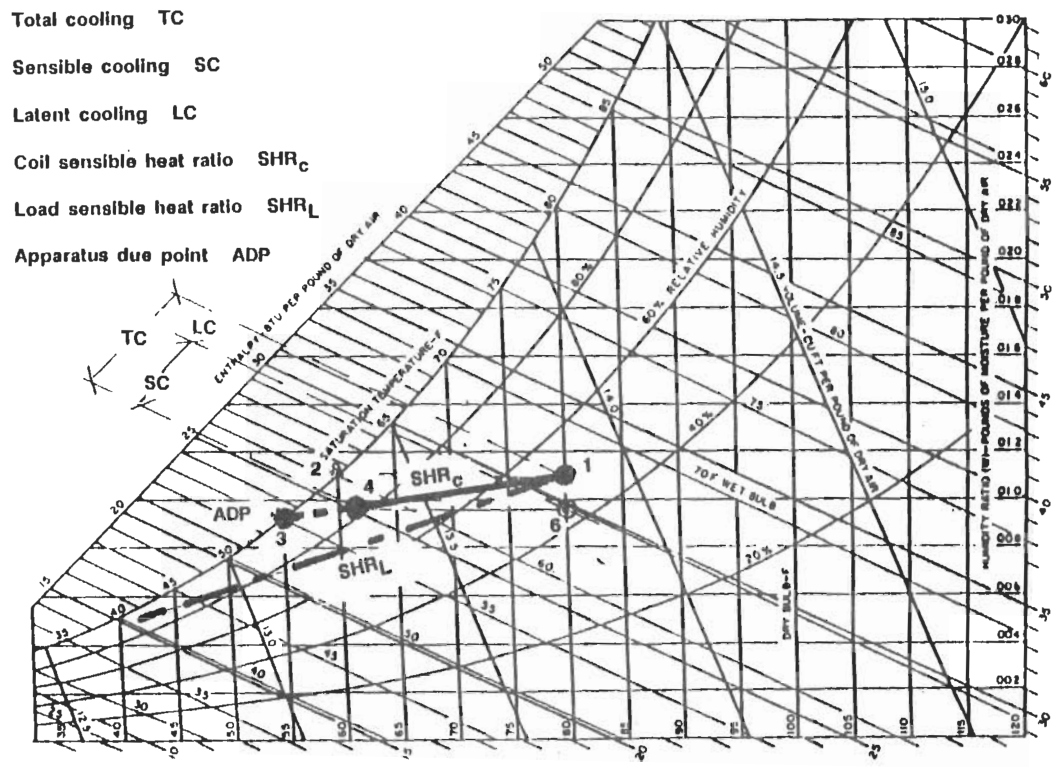

Fig. 2 Psychometric chart of cooling coil process

Ratio (SHR). Thus, colder ADP improves dehumidification performance of the coil.

The most common method of reducing the effective coil temperature or apparatus dew point temperature of the coil is to increase the pressure drop across the expansion valve of the vapor compression machine by increasing the head pressure on the compressor, or reducing the suction pressure, or both. Because the gas expansion of the expansion valve is the irreversible process of the Carnot cycle, this method of improving dehumidification must reduce energy efficiency. Indeed, a lowering of the ADP by any external means to the air conditioner cycle, also has the effect of reducing the energy efficiency or coefficient of performance (COP) of that cycle.

In addition to higher pressure drops, reduced air flow over the coil also reduces the ADP. The improvement in dehumidification as a result of lowering the air speed across the coil is a "reduced coil temperature" option because of the lowered ADP. A similar COP penalty must be paid. Figure 3 from Khattar (4) shows the effect of lowering the ADP temperature for a typical coil on return air (80°F, 50% RH). The shape of the curve tells us much about this option. For example, reducing the ADP from 60°F to 50°F produces a major improvement in dehumidification; however, the next 10° reduction (from 50°F to 40°F), does little additional dehumidification. Every reduction in ADP costs COP, so clearly, the continued reduction of ADP temperature will at some point be too costly from an energy or capacity standpoint.

The ADP cannot operate below freezing without the risk of condensate freezing and potential coil damage. The psychometric chart of Figure 2 makes the point clear. Given state point 1 entering the coil, there is a limit to the SHR_C (slope of 1 to 3) that can be obtained by lowering the ADP, and with a SHR_L = .6 (from GRI supermarkets) as shown on the chart, these slopes will match only with an ADP at freezing (32°F). The option to reduce the ADP for improved dehumidification is an action on the coil operation itself. The other three options of this review; reheat, heat transfer and

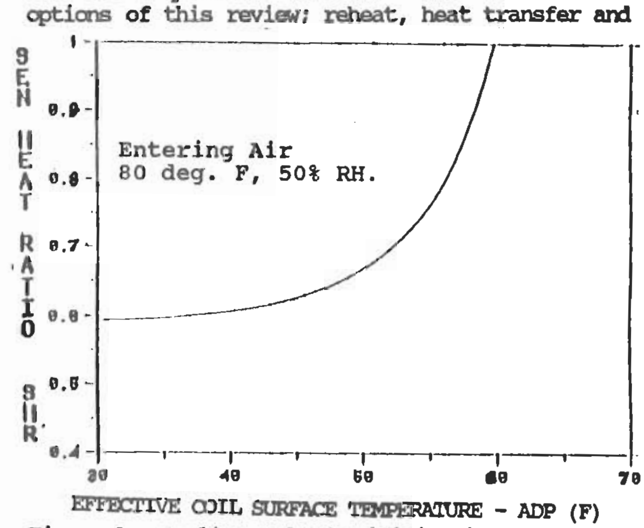

Figure 3. Cooling and Dehumidification process SHR vs. Cooling Coil ADP temperature.

moisture transfer, are options to change the state of the air after it exists the coil or before it enters the coil and after it exists. Any modification to the air conditioner system to improve performance of the coil can be equally applied to the other three options.

Before the other options are reviewed however, it is useful to consider what happens to an air conditioner coil when it is operated with a different SHR_C from the SHR_L.

For the evaluations actual system product data (ARI rating conditions) for a typical central air conditioning unit (split system)will be used:

Total Net Capacity (TC)	=	22800 Btu
Air flow	=	850 CFM
Compression Power	=	2370 watts
I.D. Fan Power	=	310 watts
O.D. Fan Power	=	220 watts
S.E.E.R.	=	8.00 Btu/watt
Sensible cooling (SC)	=	17,300 Btu
Latent cooling (LC)	=	5500 Btu
SHR_C	=	.76

If a constant space load of 10,000 Btu/hr is used, and assumed it has a SHR_L of .76 (it falls along the SHR_C), i.e., 7600 Btu/hr are sensible and 2400 Btu/hr are latent; then, with the capacity of 22,800 Btu/hr, our equipment operates at a 0.439 duty cycle and would use 30,554 watts per day.

But if the SHR_L is equal to .6, as in the GRI study of supermarkets, then of the 10,000 Btu/hr load, 6000 are sensible and 4000 are latent. If we operate the air conditioner at the previous duty cycle (.439) it will produce 2400 Btu/hr latent cooling against 4000 Btu/hr latent load. The moisture load is not met. Moisture builds up in the space - begins to condense on the chilled products. The meats, milk, cheese, eggs all get wet - frozen products get covered with frost where you can't read the labels. On top of this, the air conditioner is producing 7600 Btu/hr sensible cooling against a load of 6000 Btu/hr. We have a moisture problem and we are already overcooling.

In supermarkets as in many operations, the moisture load must take precedence. To meet the moisture load of 4000 Btu/hr moisture removal, the air conditioner also does 12,667 Btu/hr sensible cooling for a total of 16,667 Btu/hr. This is a 0.73 duty cycle and would use 50,808 watts/day, a 66% increase in energy use to meet the latent load. In addition, our air conditioner unit must produce 12,667 Btu/hr sensible cooling against a sensible load of 6,000. We are really overcooling the space (6,667 Btu/hr overcooling).

THE AIR REHEAT OPTION

Air reheat is the most common method of matching the conditioned space SHR_L with the sensible heat ratio of the overall conditioning system (SHR_S). We can pay for the reheat energy in the form of gas or electricity, or we can use any free heat that is available - which is typically hot gas bypass.

If we add electrical resistance heat to the air leaving the cooling coil, we can solve the overcooling problem. This is represented on the psychometric chart of Figure 4. The reheat energy is represented by the change of the air going from state point 4 to state point 5. At state point 5, the SHR_S equals the SHR_L.

For our example, to provide supply air from our system to the space to match the load, we must reheat the air by 6,667 Btu/hr. If electrical resistant reheat at 100% efficiency is used, it will cost 46,898 watts/day additional energy. If it is assumed the heating coils do not require additional fan power, 50,808 watts/day cooling and 46,898 watts/day reheat is used for a total energy consumption of 97,706 watts/day to meet the load.

Hot gas bypass is the method of using heat available from the compression side of the air conditioner cycle as reheat energy. This heat would normally be passed to the outside by the condenser coil. Instead, part of the hot gas leaving the compressor is passed to a set of coils placed in the supply air stream after the cooling coil. The energy used for reheat is free, i.e., it does not add to the cooling energy requirement.

The addition of coils to the supply plenum does add approximately 10% to the inside fan power requirement. For the comparison equipment this is an additional 31 watts or 744 watts/ day. We can thus meet the latent and sensible load with free reheat (hot gas bypass) consuming 50,808 + 744 = 51,552 watts/day.

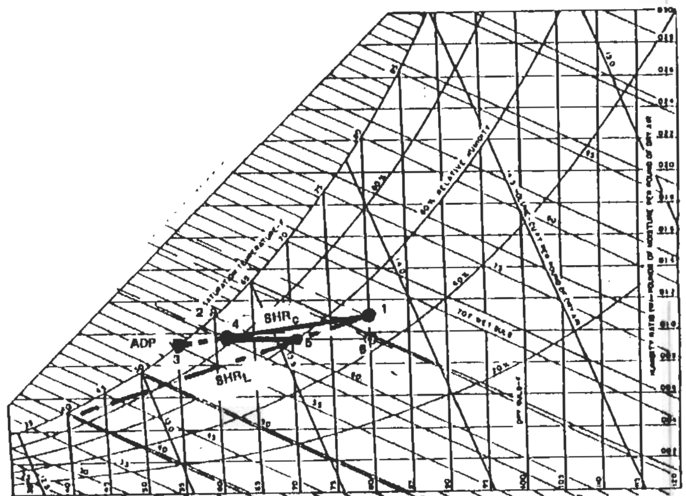

Figure 4. Psychometric chart of cooling and reheat.

THE SENSIBLE HEAT EXCHANGE OPTION

The alternative of sensible heat exchange uses the heat available in the return air, to reheat the supply air. This is shown on the psychometric chart of Figure 5 and diagrammatically in Figure 6. The supply air needs to be heated from state point 4 to state point 5.

The cooling coil now operates between state point 2 and state point 4. The precooling of the air before it reaches the cooling coil, reduces the ADP of the coil, improving its dehumidification. In addition, the reheat energy does not add to the energy requirement of the system.

There are three mechanisms used to provide this heat exchange between the return air and the supply air: 1) run-around coil, 2) plenum exchange, and 3) heat pipe.

The run-around coil consists of two fin-coil stacks with a small pump between them. The working fluid (usually water) is circulated in a closed loop between the two coils, removing heat from one coil and adding it to the other, thus providing the necessary heat exchange. This mechanism has been used since the 30's on very large systems. The amount of heat exchange is controlled by controlling the flow of heat exchange fluid. Plenum exchange, developed by Doderer (5), provides for the heat exchange across the surfaces of the ducting going to and from the cooling coil. Large surface area is needed to accomplish the required heat exchange, so lamberenthian and corrugated surfaces have been used. The amount of heat exchange is controlled by ducting & bypassing varying amounts of air such that all air does not contact all the heat exchange surfaces.

Heat pipes use two fin-coil stacks as in the run-around coil. However, the working fluid of the heat pipe is a condensing fluid such as freon. The amount of heat exchange is fixed by initial design or varied by valving the working fluid. In some systems, the freon may be pumped as in the run-around coil.

These three mechanisms provide the same thermodynamic process to the air conditioner system, i.e., sensible heat exchange between the return and supply air stream. Referring to Figure 5 and assuming that the proper amount of heat is transferred from the return air stream, our return air is precooled by 6,667 Btu/hr to state point 2, and our supply air is reheated by 6,667 Btu/hr to state point 5. The coil ADP is shifted down to 50° F from 55° F and the SHR_C is equal to the SHR_L.

If our total capacity and COP stay the same, the coil would have a sensible cooling capacity of 13,680 Btu/hr and a latent cooling capacity of 9,120 Btu/hr (SHR_C =.6). To meet the 6000 Btu/hr sensible load and the 4000 But/hr latent load, the equipment would operate at a 0.439 duty cycle and would use 30,554 watts per day total (compressor, ID fan, and OD fan).

However, we have reduced the ADP of the coil and must pay a COP penalty. Khattar, in the simulation and study of heat pipe systems, provides that the COP reduction is one-half percent per degree reduction in ADP (6). Thus, the compressor would consume 2½% more power.

In addition, we must increase the ID fan power to maintain flow. Khattar has measured this requirement to be from 40 to 50% increase in ID fan power (actual power depends on configuration and fintube design). A 10% ID fan power penalty was used in the evaluation of the hot gas bypass coil - but the heat pipe option uses two coils, and the temperature differences are much lower. To assure our design will meet the full heat transfer requirement, we will assume a 50% ID fan penalty. Full capacity power use is:

Compressor:	2370 + 60	=	2430 watts
ID fan :	310 + 155	=	465 watts
OD fan :	220	=	220 watts
			3115 watts

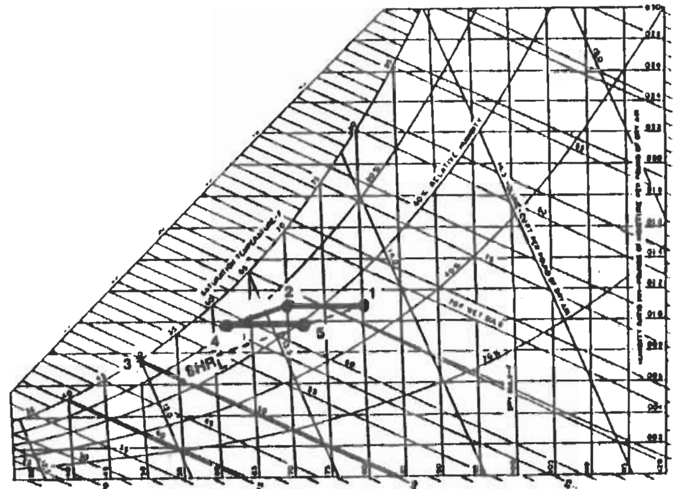

Figure 5. Psychometric chart of cooling coil with heat exchange.

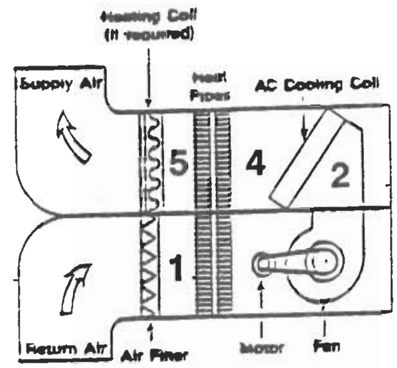

Figure 6. Diagram of heat exchange using heat pipes.

Operating at the 0.438 duty cycle, the daily power requirement to meet the moisture and sensible load using the heat exchange option is 32,820 watts per day.

THE MOISTURE EXCHANGE OPTION

Moisture exchange uses the return air to act on the supply air, but the thermodynamic process is different from that of simple heat exchange.

The process is shown on the psychometric chart of Figure 7 and diagrammatically in Figure 8. A desiccant is used to remove moisture from the high humidity air exiting the cooling coil. This sorption of moisture drys the supply air and reheats it by the heat of sorption. This reheat energy is free. The process very closely follows a line of constant enthalpy moving from state point 4 to state point 5.

The moisture taken from the supply air by desiccant, is then evaporated into the return air prior to it reaching the cooling coil. The evaporation precools this air and brings it close to saturation. In the evaporation process, the air follows closely the line of constant enthalpy up from state point 1 to state point 2.

Control of moisture exchange in this option is provided by control of the presentation of the desiccant. In a wheel design, the speed of the wheel is varied. In a liquid design, the flow of the desiccant is varied. There are several constraints associated with this process. First, the moisture moved in 1 to 2 must equal the moisture moved in 4 to 5; and second, the affinity for moisture of the desiccant must be greater at point 4 than it is at point 2.

The affinity for moisture of most desiccants vary in this way: higher affinity with higher RH, higher affinity with lower temperature and higher affinity with higher pressure. Point 4 easily meets all these criteria in respect to point 2. A more detailed evaluation of this process has been provided by Cromer (7). In the evaluation of this option, it will be assumed (as in the heat transfer evaluation) that the state points shown are achievable by the process such that the $SHR_C = SHR_L$.

A second and important factor of this process is that state point 4, air exiting the cooling coil, is very close to the saturation line. This is the operational effect of loading the air with moisture before it reaches the cooling coil. The higher the RH of the air entering the coil, the higher the RH of the air exiting the coil. Figure 9 presents data taken on a DX cooling coil in a Cocoa, FL townhouse.

Notice the remarkable effect of this process. The moisture load is met ($SHR_C = SHR_L$) but the ADP has been <u>increased</u> by 12°F to 62°F from the heat exchange option, and increased 7°F from the base case (55°F). Thus an improvement in COP of 3.5% would be obtained by operating at a higher ADP.

Simulation work by Collier's modification of the SERI desiccant model (8), predicts a desiccant wheel with a frontal area of two square feet and two inches thick will very nicely provide the moisture exchange needed. These wheels use a parallel passage technology that has a low pressure drop, nevertheless such wheels require additional ID fan power. For the evaluation, a 50% ID fan penalty will be assumed. Full capacity power use is:

Compressor:	2370 − 83	=	2287 watts
ID fan :	310 + 155	=	465 watts
OD fan :	220	=	220 watts
			2972 watts

Operating at the 0.439 duty cycle, the daily power requirement to meet the moisture and sensible load using the moisture exchange option is 31,313 watts/day.

SUMMARY OF COMPARISON OF SYSTEMS

The best or most energy efficient option is moisture exchange. The heat exchange option is also very good. The following TABLE 1 summarizes the results of the two ton air conditioner unit comparison, and provides the energy consumption required to meet the sensible heat ratio of the load of 0.6.

The use of moisture exchange through desiccants as a means to improve the dehumidification of a typical air conditioner

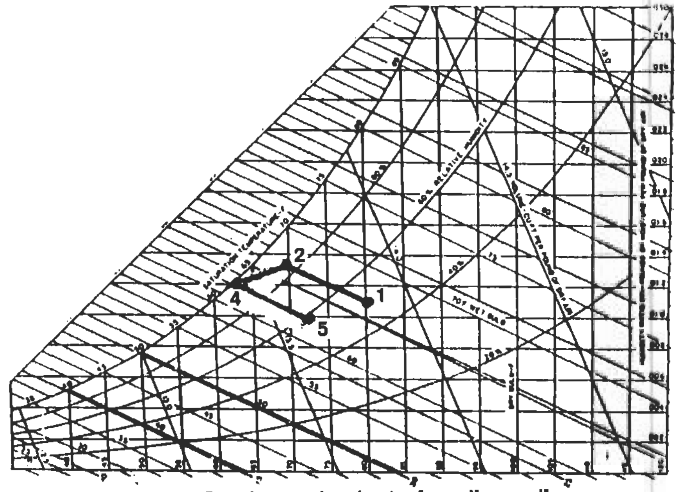

Fig. 7 Psychometric chart of cooling coil with moisture exchange.

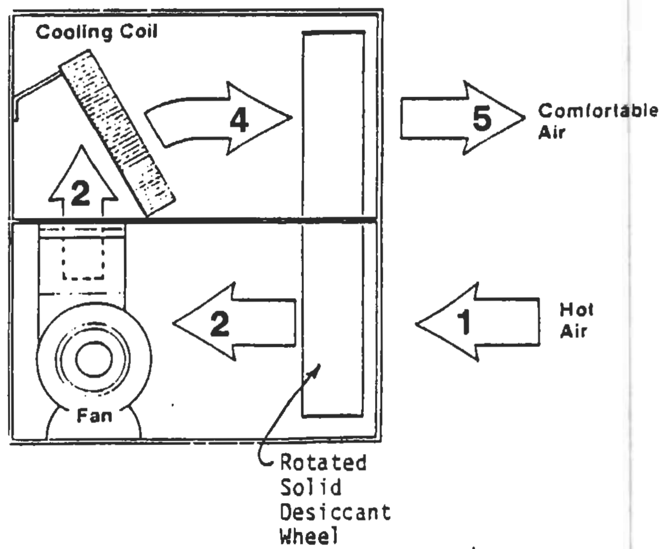

Fig. 8 Dehumidifier/air conditioner using desiccant moisture exchange.

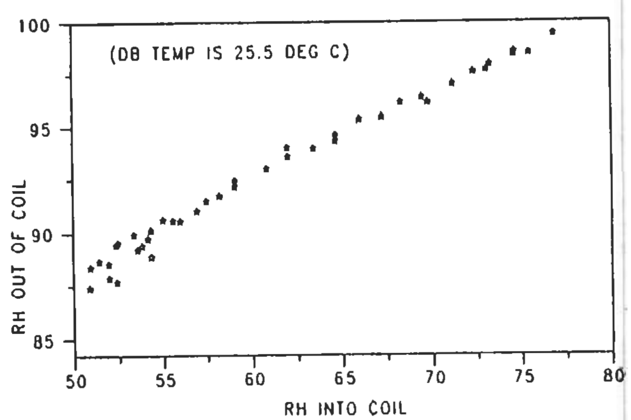

Fig. 9 RH OUT OF COIL AS A FUNCTION OF INLET AIR RH

system is presently unproven. However, psychometric analysis and simulations of desiccant wheels both suggest that the concept has the potential to provide increased dehumidification at reduced first cost and reduced operating cost when compared to existing alternatives.

TABLE 1 COMPARISON OF ENERGY REQUIREMENTS TO MEET LATENT LOAD OF 40%

Option	KWh/day	equipment duty cycle
1. Moisture exchange (desiccants)	31.31	0.44
2. Heat exchange (heat pipes)	32.83	0.44
3. Free reheat (hot gas bypass)	51.55	0.73
4. Electrical reheat	97.71	0.73

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