ASSESSMENT OF THERMALLY ACTIVATED HEAT PUMPS WITH DESICCANT COOLING

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ABSTRACT

Two configurations of a thermally activated heat pump (TAHP) with a desiccant wheel for air drying are analyzed and compared to two conventional air conditioners for summer space cooling. Annual energy performance estimates are made for eight U.S. locations using historical data from the summer of 1986. Two system operational modes are considered: (1) ventilation mode and (2) recirculation mode. When the system operates in the ventilation mode-which pressurizes the house - the cooling load is reduced. The energy trade-offs and annual operating costs for each system in each of the eight locations are estimated.

SYSTEM DESCRIPTIONS

The Case 1 desiccant system is shown in Figure 1. An internal combustion engine (IC-E), fired by natural gas, powers the compressor of a conventional vapor compressor air conditioner (VC-AC). The system can draw air from the house at State Point 1 (SP-1) to be processed and returned (Recirculation Mode) or draw some ambient air to be combined with house air in the Mixing Box (Ventilation Mode). The ventilation fraction (B) can be regulated. In either case, the air is first dried in a desiccant wheel dehumidifier (DH), then cooled by an indirect evaporative cooler (I-DEC). For conditions where the ambient air is humid but cool, some useful heat can be recovered to the regeneration airstream by an air-to-air heat exchanger (A-HE, SPs 5 to 6).
The process air then goes to the VC-AC evaporator where it is finally cooled to meet the cooling load line. The process state points are depicted in Figure 2. Case 1 is further discussed by Kleiser (1987), Chen (1987), and Turner et al. (1987-2, 1987-3).

Case 2 is shown in Figure 3, and its psychrometric state point chart in Figure 4. Moist ambient air only is dried in a desiccant dehumidifying wheel (DH). Air for DH wheel regeneration (driving out the moisture) is taken from the house and heated. The dried ambientair is mixed with house air, and the entire supply air mixture is cooled in a conventional vapor compressor heat pump. Case 2 operates only in the ventilation mode (although the house is not pressurized) and has the potential advantage that only the most moist air is dehumidified. The heat transferred through the regenerator heat exchanger (H.E.) is at such high temperature that the rejected heat from the heat pump condenser is not available and is vented.

A conventional electric-driven vapor compressor air conditioner (VC-AC), which features reheat capability so that the interior humidity ratio can be controlled, is the standard of energy comparison. This is depicted in Figure 5. No energy penalty is assigned for the reheat because available rejected heat from the condenser unit is used for the supply air reheat. Instead of a motor powered by external electricity, the VC-AC can also be powered by a natural gas, thermally driven internal combustion engine (IC-E), also depicted in Figure 5. Considering a natural gas conventional air-conditioning case allows direct comparison with Cases 1 and 2.

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In every case, the fans necessary to move air are driven by externally supplied electricity, which quantity is calculated separately from natural gas consumption.

COOLING LOAD MODEL

The cooling load model is described by Kleiser (1987) and Turner et al. (1987-2, 1987-3), so only an overview is given here. The total house cooling load is composed of six separate elements: two are sources of latent load and four comprise the sensible load. One source of latent and of sensible load is infiltration of ambient air into the house. Infiltration-based loads are functions of the temperature and humidity differences between ambient and room air and of the infiltration flow rate.

If the house operates in the pure recirculation mode, or if the same quantity of air as the supply air is ducted from the house to the system in a ventilation mode (e.g., Case 2-Figures 3 and 4), then one air change per hour (1 ach) is assumed for infiltration. Here we let 1 ach be equivalent to 67 cfm/tonAC (Kleiser 1987; Turner et al. 1987-2, 1987-3). How the excess supply air ventilation, which can be interpreted as a pressurization factor (PF), impacts on the infiltration. Assuming a standard supply air flow rate of 400 cfm/tonAC, a ventilation excess of 216 cfm/tonAC [B=216/400=0.54] leads to zero infiltration and elimination of both the sensible and latent heat contributions due to infiltration. For 1 ach, the following normalized (to 1 tonAC = 12,000 btu/h = 12,560 kJ/h) house loads pertain for full sun when the ambient air is at ARIconditions (DBT = 95-F = 35-C and WBT = 75-F = 24-C) at sea level.

	Btu/h-tonAC
Latent Heat Internal Generation	700
Latent Heat Infiltration	1963
Sensible Heat Internal Generation	1200
Sensible Heat Infiltration	1447
Sensible Heat Solar Gain	2000
Sensible Heat Conductive Gain	4690

It is assumed that 500 square feet (46.5 square meters) is associated with 1 tonAC at ARI ambient conditions.

Each system must deliver supply air so that the house is maintained at DBT = 75-F and humidity ratio = 0.008 lbv/lbda. It is acceptable if either the house DBT or humidity ratio drifts below these criteria, provided that no extra energy is required.

Process air ducted into the house must have a relationship between its temperature and humidity such that its state point falls on the sensible heat ratio (SHR) line (the "load line") of the room (Clifford 1984). If it does not, then steady-state room conditions are not attained, as the room state point must then change until its SHR line does intersect the process air state point. We first assume an inlet supply air volume flow rate (say 400 cfm/tonAC) and then calculate the humidity ratio and temperature required to hit the load line. The SHR is shown in Figures 2 and 4.

SYSTEM ELEMENT MODELS

Both the desiccant assisted TAHPs and conventional heat pump air-conditioning systems are composed of a number of separate units, each of which is mathematically modelled, and whose interrelationships are mathematically described.

Jurinak's potential equations (Jurinak 1982; Howe 1983) are used in this analysis to model the behavior of the silica gel desiccant wheel.

Jurinak's F-potentials are:

$$F_{1,n} = \frac{-2865}{T_n^{1.490}} + 4.244\, W_n^{0.8624}$$

$$F_{2,n} = \frac{T_n^{1.490}}{6360} - 1.127\, W_n^{0.07969}$$

(n = 3, 5, or m: ports of the DH wheel)

and the effectivenesses relating them are:

$$e_i = \frac{Fi_{,5} - F_{i,3}}{F_{i,m} - F_{i,3}}$$

(i = 1 or 2, and refer to F_1 and F_2 above)

Jurinak (1982) suggests that effectiveness values of $e_1 = .08$ and $e_2 = .95$ be used.

Useful results of modeling the desiccant wheel can also be displayed graphically (Turner et al. 1987-1, 1987-3; Chen 1988).

The vapor compression heat pump is taken from an equation given by Howe (1983), which is a curve fit to data for a range of air-conditioning units. Howe's COP equation, converted to Fahrenheit, is:

$$COP = N1 + N2 + N3$$

where:

$$N1 = 3.68 + 0.162 * lr * tn * EXP(-0.183 * lr * tn)$$
$$N2 = - 0.753 * lr - 0.0073 * tn$$
$$N3 = - 0.02221 * (T_{cond} - T_{wb,evap} - 27)$$

and

lr = output as fraction of rated size
tn = rated size of VC-unit (tonAC)
T_{cond} = temperature (°F) of stream entering condenser
$T_{wb,evap}$ = WB temperature (°F) of stream entering evaporator

With the required engine shaft power known, a study by Segaser (1977) is used to determine the amount of fuel necessary to power a gas-fired IC-E and also to determine the ultimate distribution of energy from the fuel.

Fan power is estimated by calculating and assuming pressure drops through each component in the system. Then, knowing airflow rates (at sea level air density), a manufacturer's fan multi-rating table (1983) gives (dP = in-water)

$$\text{Specific Power} = A * (dp * * 2) + B * dP + C$$

with

$$\text{FAN POWER} = \text{Flowrate} * \text{Specific Power (kW)}$$

and

$$A = -1.067\ E\text{-}5 \quad B = 2.68\ E\text{-}4 \quad C = -1.233\ E\text{-}5$$

Mathematical performance estimates for the other system components, including the indirect and direct evaporative coolers, heat exchangers, gas auxiliary heaters, filters, and mixing boxes, are straightforward and are described elsewhere (Turner et al. 1987-2, 1987-3).

ANALYSIS

The component interactions described above have been programmed to simulate Cases 1 and 2 and the two conventional air-conditioner configurations. Referring to Figure 1 (Case 1), the unpressurized condition features no ventilation (B=0, thus pf = 0). Thus the same quantity of house air (M1) ducted to the house is also withdrawn from the house, and the net exfiltration relative to the HVAC system is zero. Then the model assumes a net ambient infiltration rate of 1 air change per hour (1 ach = 67 cfm/tonAC). Both the sensible and latent heat loads due to infiltration must be considered.

For the Case 1 pressurized case, ambient ventilation air (fraction B of M1) is mixed with the house return air. M1 is greater than the airflow rate withdrawn from the house by the system, and this difference causes a positive house pressure (pf > 0), causing net exfiltration from the house. For a ventilation flow rate of 216 cfm/tonAC (B = 0.54 and pf = 1), the net infiltration is zero, and there is no infiltration load imposed by the ambient conditions. This usually lowers both the sensible and latent cooling house loads. However, the air the system must condition is usually more warm and moist than the house air. Thus the two effects of load reduction and of higher enthalpy air entering the system impact in opposite directions and generally do not compensate equally.

Although Case 2 (Figures 3 and 4) operates only in the ventilation mode, the same quantity of air is removed from the house for regeneration as is supplied from the ambient. Therefore, the pressurization factor PF = 0 and 1 ach pertains, so the full latent and sensible infiltration cooling loads must be accommodated.

For each case and condition, Universal Energy Consumption Tables were produced to estimate the thermal and electrical power required to maintain the required cooling load. They are presented in Turner et al. (1987-2), along with other complete details of this study that cannot be included here. Figure 6 shows the energy consumption table plotted for Case 1 with pressure factor = 0 and solar fraction = 1. Similar Universal Energy Consumption Tables were produced for both configurations of the conventional air conditioner.

Annual energy consumption estimates are made using actual 1986 summer weather data (NOAA Local Climatological Weather Data). The eight locations are Miami, Atlanta, Philadelphia, Chicago, Memphis, Houston, Phoenix, and Sacramento. These data are divided into a discrete number of two-dimensional bins on a psychrometric chart. The number of hours throughout the 1986 summer season for which the ambient DBT and W fall within a given bin are recorded for both solar and no-solar gain conditions. Thus, 16 such tables are presented in Turner et al. (1987-2). A typical weather bin distribution psychrometric chart (for Miami, with solar fraction = 1) is presented as Figure 7.

Knowing how much energy is required per hour by each considered AC unit from the energy consumption tables, and knowing how many hours per (1986) summer season the system must provide cooling at different ambient conditions, it is a simple (but time-consuming) matter to take the dot product of the two tables to estimate how much energy will be required throughout the season. The bin performance maps show the total annual energy required in each bin, and the sum of all the bins is the total annual energy consumption (Chen 1987).

Sixty-four such maps are presented in Turner et al. (1987-2). Figure 8 shows a typical energy consumption sheet for the Case 1 desiccant cooling system in Miami, with pressure factor (PF) zero and solar fraction (sf) unity. A total of 1711.7 therms/year • tonAC of natural gas heat are required for this "day" condition; 180.2 therms/year • tonAC are required for night and cloudy conditions. Thus, a total of 351.9 therms/year • tonAC were required in Miami in 1986.

Knowing how many hours the system ran, airflow rates, and pressure drops, the electrical power is computed (993 kW-h/year • tonAC). These values of natural gas and electrical energy consumption appear in Table 1 (Case 1, PF = 0) for Miami. Table 1 also summarizes the 1986 energy consumption for all considered cases.

Assuming a natural gas heat cost of $0.50/therm and an electricity cost of $0.08/kW/h, the annual cost of cooling is estimated in Table 1 for all cases. Table 1 also indicates the total number of cooling hours recorded in 1986 for the eight locations.

RESULTS

Table 1 is a comparative assessment summary of the energy and money consumption for each city and machine through the 1986 summer season. The energy cost is estimated at $0.50/therm for gas heat and $0.08/kWh for electricity; 1 therm = 100,000 btu = 105,400 kJ. In Table 1, Case C-1 is the gas-fired conventional VC-AC unit, while Case C-2 is the all electric VC-AC found in most homes.

The most expensive U.S. location with any unit, in energy or in dollars, is Miami, while in every case Sacramento is the least expensive. This is because Miami requires the greatest number of hours of cooling during the season, while Sacramento requires the least.

The Case 1 desiccant system is less expensive to operate than either conventional system in every city. The main reason for this is that the desiccant system uses low-grade waste heat to meet a significant part of the latent load, while the conventional systems use high-grade gas or electric energy to meet every bit of both the latent and the sensible loads. Actually, a conventional machine, in order to meet the latent requirement, must usually overcool the process air in order to condense enough moisture, so that when the airstream is dry enough, it's too cold. In doing this, it has actually invested more in sensible cooling than is necessary to meet the sensible load. A reheat of the process stream will put the state point back onto the load line but does nothing toward recovering the excess energy invested in overcooling. It simply shows up as an increase in operating cost.

Another apparent reason that the conventional cases use more power is that they lack the indirect evaporative cooler (ID-EC). In a conventional system, an ID-EC would pre-cool house air before it entered the evaporator. However, reference to a psychrometric chart and bin weather data (Turner et al. 1987-2) shows that the ambient wet-bulb temperature is usually greater than 65 F for most of the cooling hours in the more moist climates. If we assume a reasonable "approach" of 80% for the ID-EC, then the lowest temperature into the evaporator will be 67 F. But in order to meet the latent load, the evaporator will have to cool the airstream considerably more than this while condensing moisture. The fraction of cooling below room temperature contributed by the ID-EC is relatively minor. In addition to this, the lowering of the evaporator inlet temperature lowers the COP of the VC unit, further abrogating any advantage of the ID-EC to the conventional VC unit. These factors lead to a factor of nearly 2 between Case 1 performance and the higher cost Case C-1 performance.

If free heat is available at the site, for example rejected heat from a commercial component such as a freezer, then either configuration of the desiccant-augmented heat pump will perform better than the conventional alternatives.

The two Case 1 situations, pressurization factor = 1 and PF = 0, show some interesting behavior. In every city but one, PF = 1 is better, though usually by less than 10%. In Phoenix, the pressurized case is more than 10% better. Because Phoenix gets so hot, any infiltration can significantly increase the sensible load. Although the latent load is actually reduced by dry infiltration, the effect is not enough to offset the increase in sensible load. Since sensible load is handled mostly by the VC unit, any additional sensible load impacts directly upon energy consumption. However, a small increase in latent load from pressurization (internally generated latent load is not offset by dry infiltration) could be easily handled by the desiccant wheel, which is liberally supplied with waste heat from the VC unit and IC-E. This means that in dry, hot conditions, pressurization is superior to nonpressurization. In fact, only in cool, moist conditions does nonpressurization excel. Miami spends many hours in just these conditions (see Figure 7), which is why the nonpressurized case is competitive there.

Table 1 shows that Case 2 (Figures 3 and 4) does not perform nearly as well as Case 1. In fact, in the moist southern cities, it is more expensive to operate than the gas-fired conventional heat pump, Case C-1. Figures 3 and 4 show why. The supply air (SP-9) must fall on the cooling load line and thus be dryer than the house air (which is maintained at W = 0.008 lbv/lbda). But since the mixing box (SPs-1, 5 and 7) admits house air (SP-1), the air at SP-5 must be extremely dry, which requires very a high regeneration temperature (SP-L). Because the condenser waste heat cannot be used and is thus vented, the only rejected heat from the system is from the IC-E, the heat from the water jacket and exhaust. Although these have a high temperature potential, there is not enough heat to raise the regeneration airstream to its required temperature. Thus additional gas must be burned in the auxiliary heater (SPs-J to L). This additional gas consumption represents an overwhelming energy penalty for Case 2. Furthermore, although operating in the ventilation mode, the house is not pressurized, so both latent and sensible infiltration loads must still be removed.

Figure 3 features a direct evaporative cooler (D-EC), intended to reduce the temperature at SP-5 as low as possible, prior to mixing. This, in turn, reduces the temperature at SP-7 to a minimum, so the minimum quantity of heat must be removed by the heat pump evaporator. But this logic is only valid if more expensive energy (e.g., electricity) is used to power the heat pump than is used for the additional heat in the auxiliary heater. Where the cost of both energies is the same, the D-EC is not a benefit, because the temperature at SP-L must be higher due to the greater moisture in the regeneration stream plus the reduced reject heat from the IC-Engine. Analyzing Case 2 without the D-EC improved the performance, but not enough to make it competitive with Case 1.

Table 1 shows how much natural gas (therms) and electricity (kWh) are required for the two conventional cases per tonAC. The energy quantities are fixed for each climate. For a natural gas cost of \$0.50/therm an electricity cost of \$0.08/kWh, the gas-fired heat pump (Case C-1) in every case costs much less to operate than the more common all-electric heat pump (Case C-2).

CONCLUSION

Mathematical simulations have estimated the annual energy requirements in different locations for different desiccant-augmented cooling systems as well as stand-alone heat pumps. At today's energy costs, the Case 1 (Figure 1) desiccant system can reduce summer cooling operational costs by a factor of 2.

Having the desiccant wheel dry only ventilation air (Case 2) is not advantageous because of the high regeneration temperatures required. This necessitates burning extra fuel in an auxiliary heater, since the temperature requirement is too high to be satisfied by the heat pump condenser's rejected heat.

A possible system improvement is depicted in Figure 9 - a two (or more) desiccant wheel configuration. Ambient ventilation air is predried by a first desiccant wheel, for which regeneration heat is provided by rejected heat from the VC-AC condenser. The temperature requirement and availability are compatible. The predried ambient air may have the same humidity ration as the house. Then house and ventilation airstreams are cooled and mixed, after which the total stream enters a second desiccant wheel. This ventilation arrangement also takes advantage of house pressurization to reduce the cooling infiltration load. The Figure 9 system attempts to capitalize on the best features of the considered systems. The idea could be extended to more than two cycles of desiccant drying and intercooling. We have not yet analyzed this multiple wheel system, although preliminary calculations indicate that the system's rejected heat should be adequate in most cases to accomplish the drying as in Case 1.

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TABLE 1

Annual Energy and Money Consumption Summary - All Cases

(Cooling hours)	CASE 1; PF=0	CASE 1; PF=1	CASE 2; PF=0	CASE C-1	CASE C-2
ENERGY	therms	therms	therms	therms	KW-hr
FAN POWER	KW-hr	KW-hr	KW-hr	KW-hr	KW-hr
TOTAL COST	$	$	$	$	$
MIAMI (4416 hrs)	351.936 993.600 $255.46	364.425 993.600 $261.70	698.448 719.808 $406.81	685.989 463.680 $380.09	6650.32 463.680 $561.12
ATLANTA (2412 hrs)	166.719 542.700 $126.78	157.488 542.700 $122.16	313.992 393.156 $188.45	329.253 253.260 $184.89	2903.19 253.260 $252.52
PHILADELPHIA (1560 hrs)	102.546 351.000 $79.35	94.140 351.000 $75.15	201.198 254.280 $120.94	213.063 163.800 $119.64	1846.44 163.800 $160.82
CHICAGO (1029 hrs)	67.929 231.525 $52.49	65.760 231.525 $51.40	137.019 167.727 $81.93	142.578 108.045 $79.93	1232.34 108.045 $107.23
MEMPHIS (3078 hrs)	245.502 692.550 $178.76	242.709 692.550 $176.76	456.513 501.714 $268.39	457.149 323.190 $254.43	4262.70 323.190 $366.87
HOUSTON (3573 hrs)	286.005 803.925 $207.32	281.487 803.925 $205.06	553.179 582.399 $323.18	545.544 375.165 $302.79	5193.84 375.165 $445.52
PHOENIX (3423 hrs)	230.934 770.175 $177.08	203.922 770.175 $163.58	347.487 557.949 $218.38	428.532 359.415 $243.02	4613.49 359.415 $397.83
SACRAMENTO (954 hrs)	53.814 214.650 $44.08	49.338 214.650 $41.84	87.684 155.502 $56.28	113.768 100.170 $64.90	1077.30 100.170 $94.20

NOTE:

1. ONE therm = $0.50
2. ONE KWh = $0.08
3. FAN POWER = 0.225 KW/ton AC FOR CASE 1
4. FAN POWER = 0.163 KW/ton AC FOR CASE 2
5. FAN POWER = 0.105 KW/ton AC FOR CASE C-1 & C-2

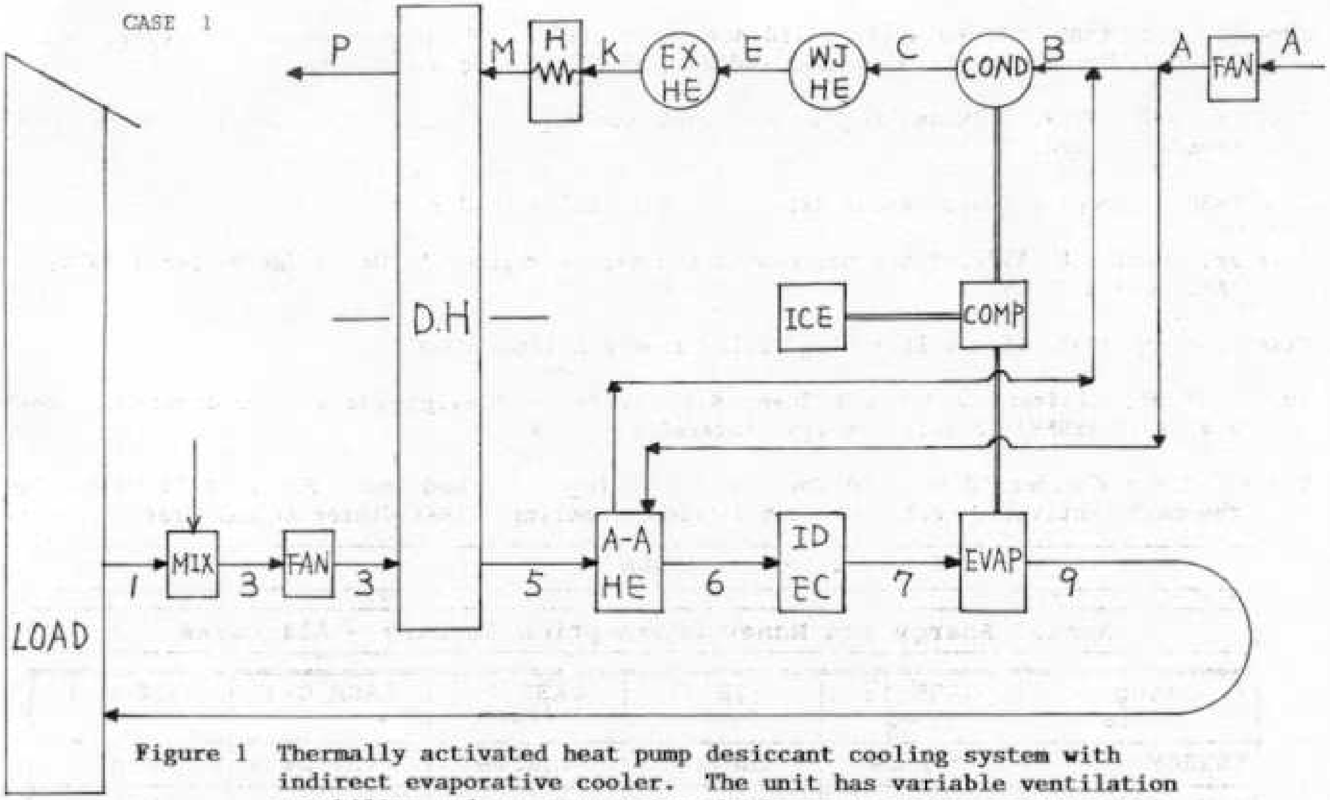
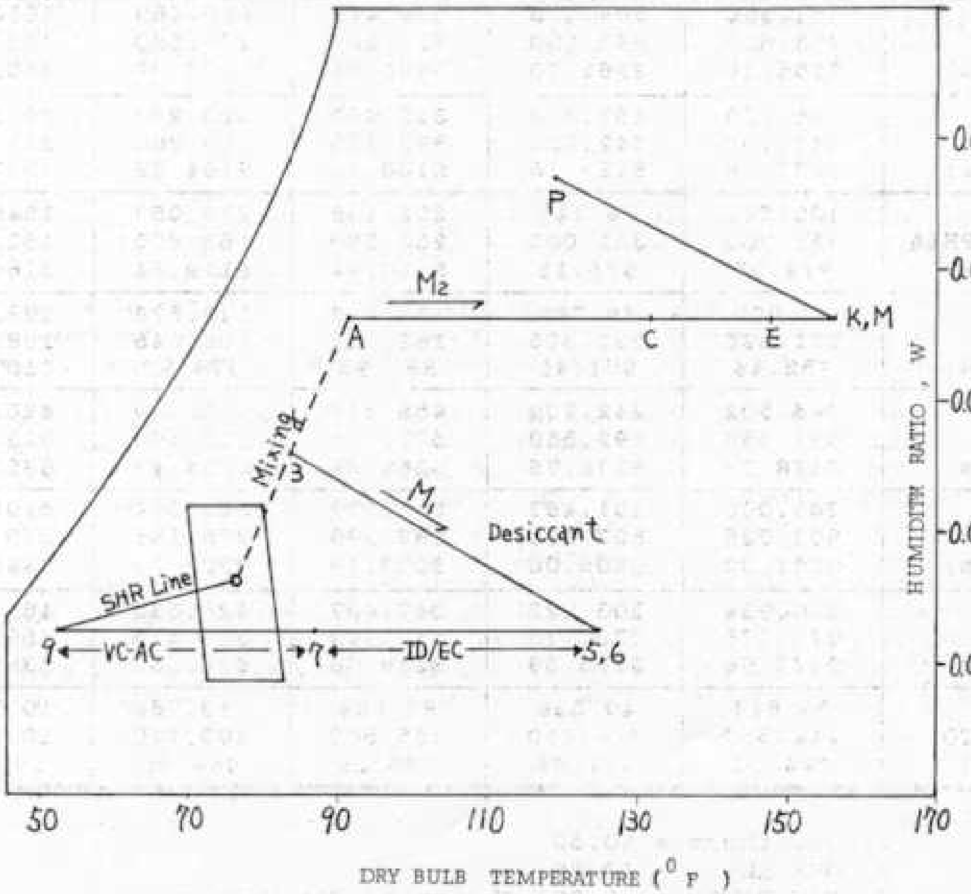


Figure 1 Thermally activated heat pump desiccant cooling system with indirect evaporative cooler. The unit has variable ventilation capability and can pressurize the house to reduce infiltration.



Note operation is shown for PP=1 and SP=1 for the Miami ASHRAE 1% Summer Design Day. No auxiliary heat is needed between S.P.s K and M.

Figure 2 Statepoints for the Case 1 TAHP-with-desiccant unit shown in Figure 1

CASE 2

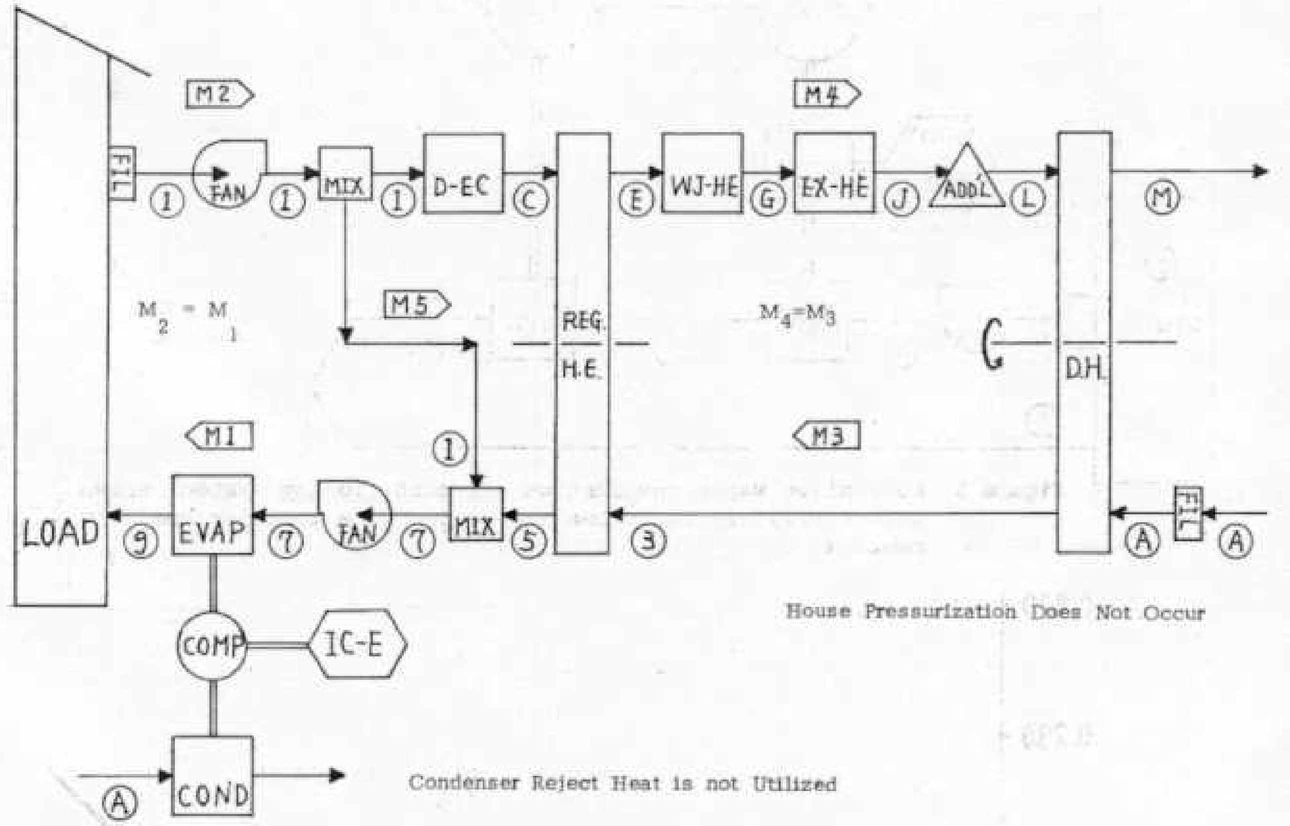
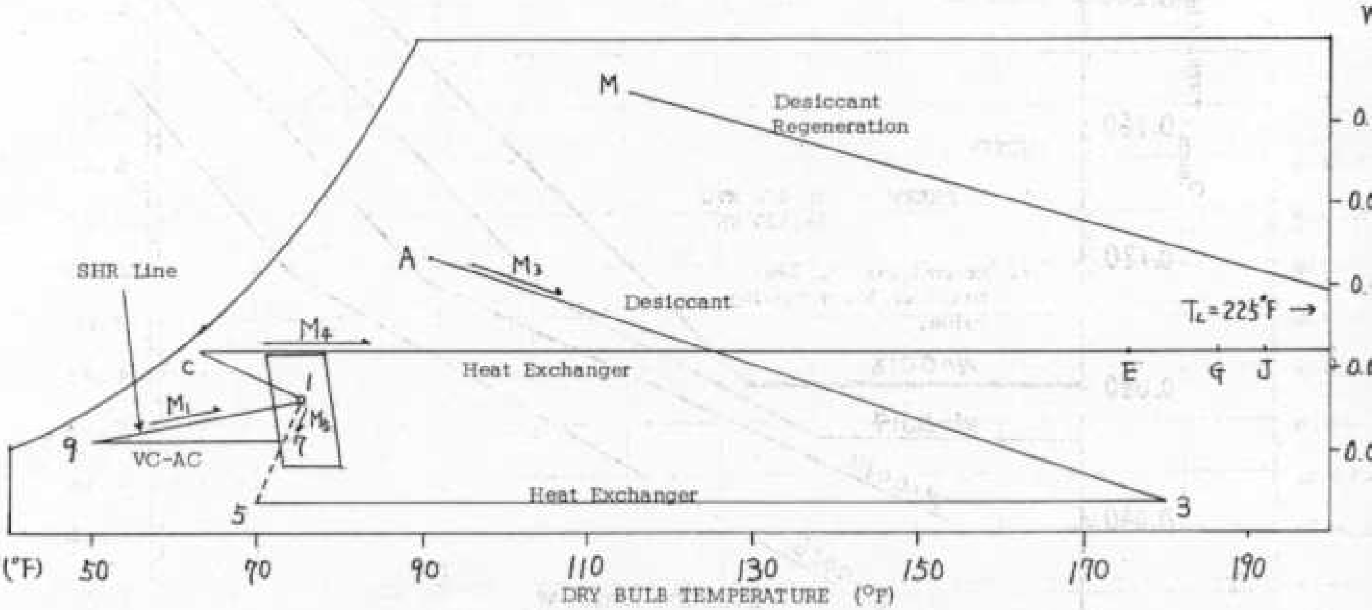


Figure 3 TAHP with variable ventilation and desiccation of only the ambient intake air



Note operation is shown for sf=1 for the Miami ASHRAE 1% Summer Design Day.
Auxiliary heat is needed to raise the regeneration air temperature from 192°F(89°C) to 225°F(107°C)
The heat exchanger effectiveness above was taken to be 95%.

Figure 4 Statepoints for the Case 2 TAHP-with-desiccant unit shown in Figure 3

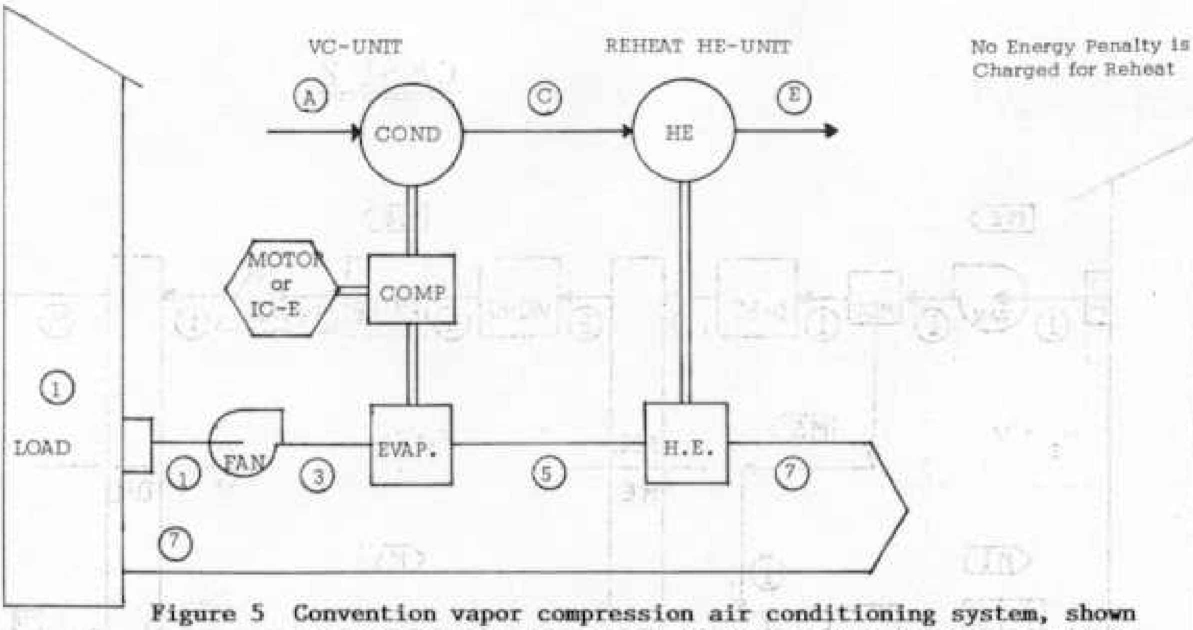


Figure 5 Convention vapor compression air conditioning system, shown with capability for reheat to accommodate interior humidity control

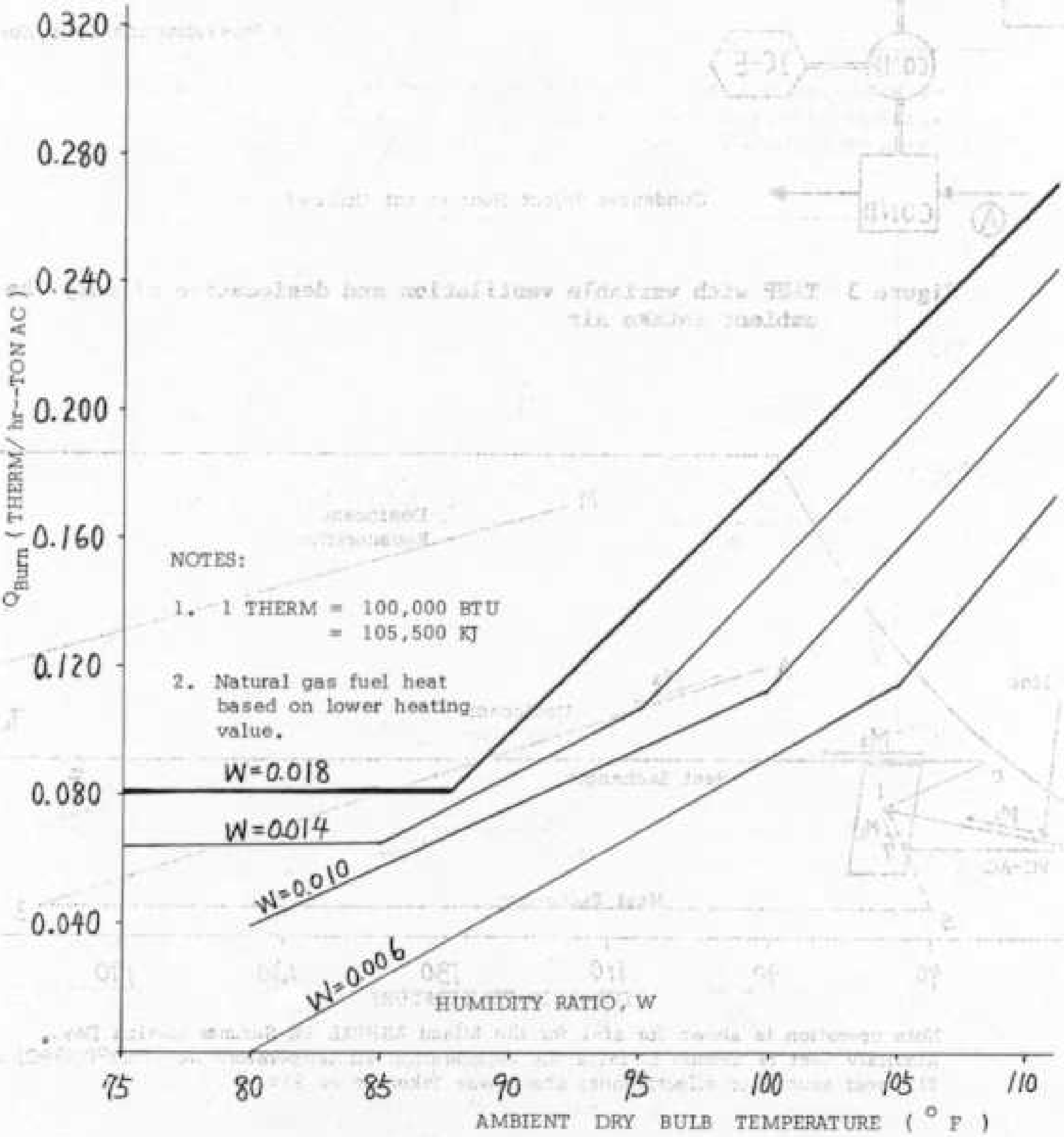


Figure 6 Natural gas fuel heat needed to power Case 1
Pressure factor = 0 Solar fraction = 1

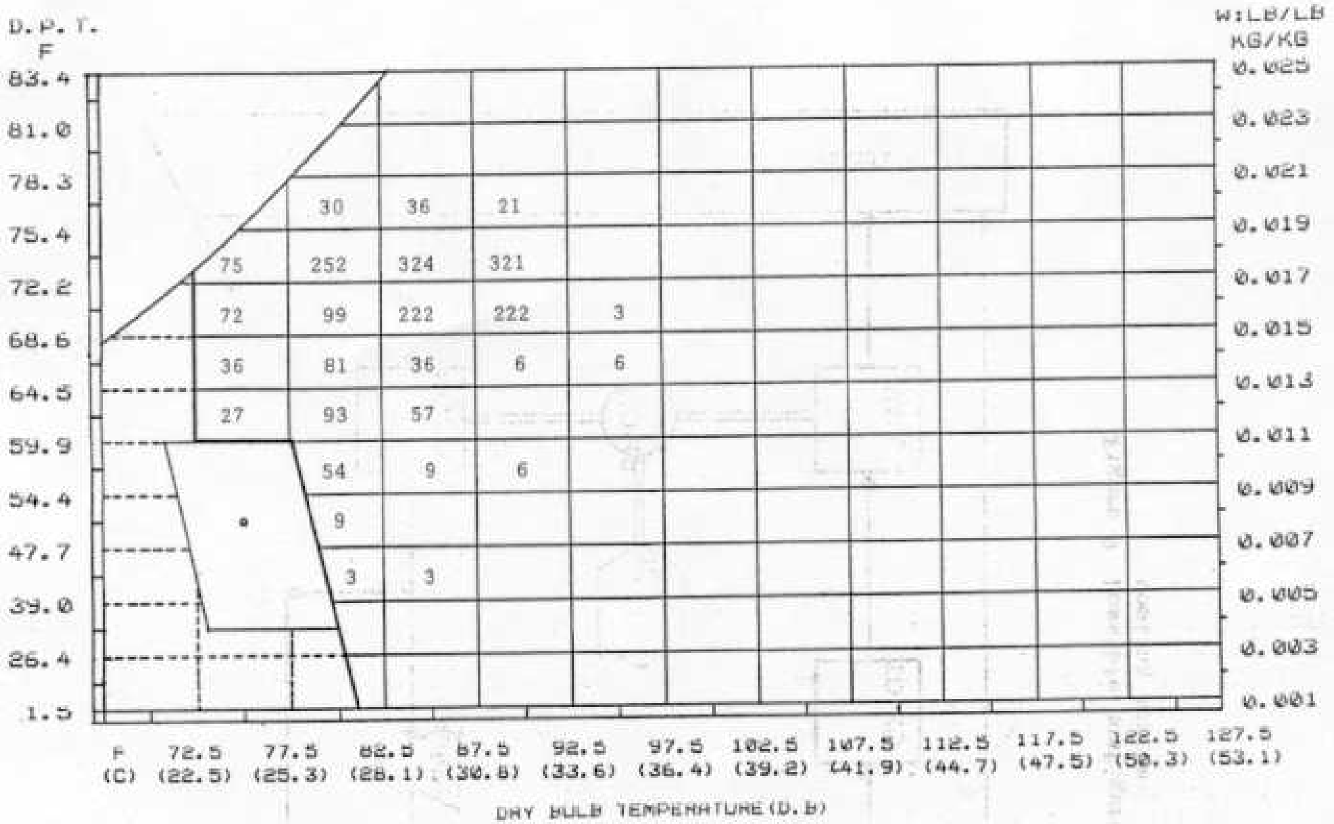


Figure 7 Miami weather bin data
Solar fraction = 1 2103 hours/year

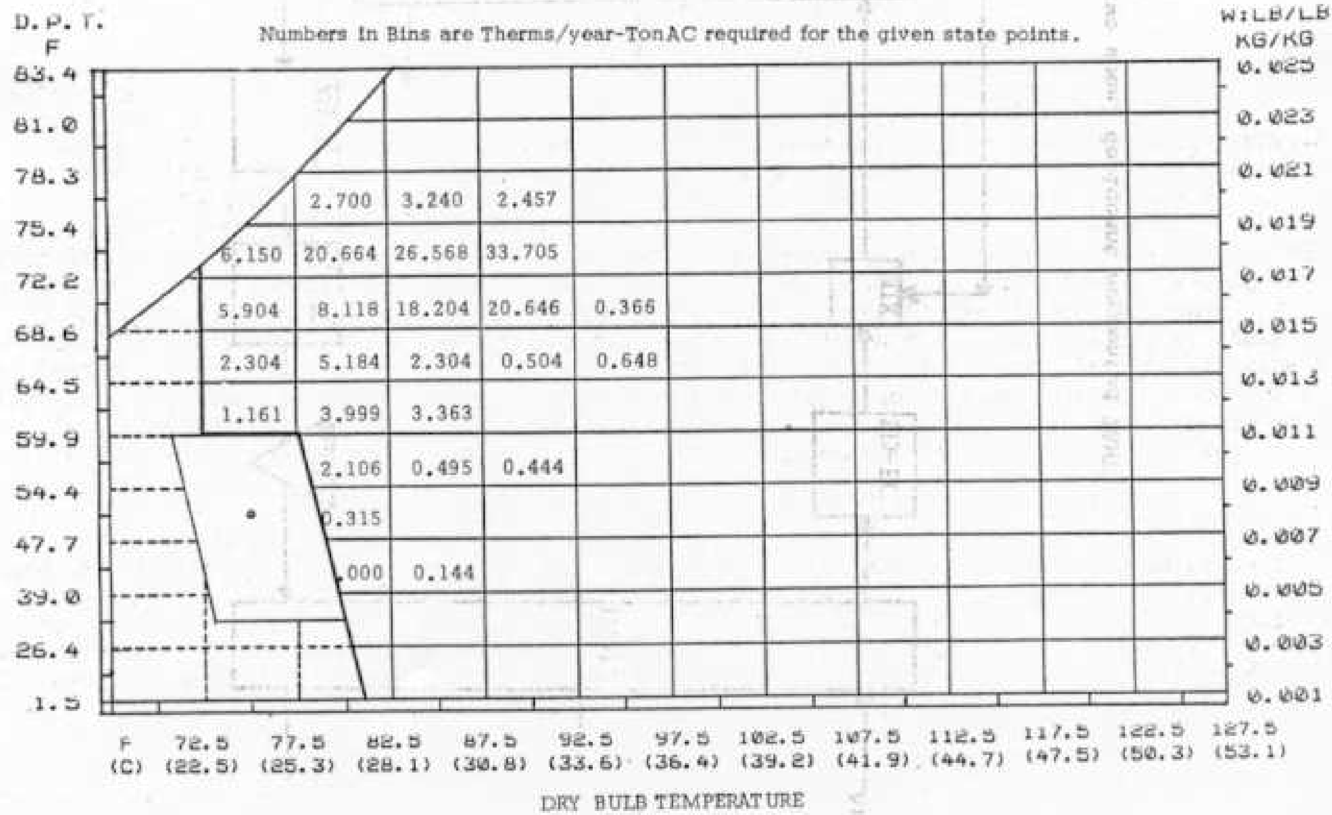


Figure 8 Miami energy consumption for case 1 PF = 0, SF = 1
171.7 Therms/year-ton AC

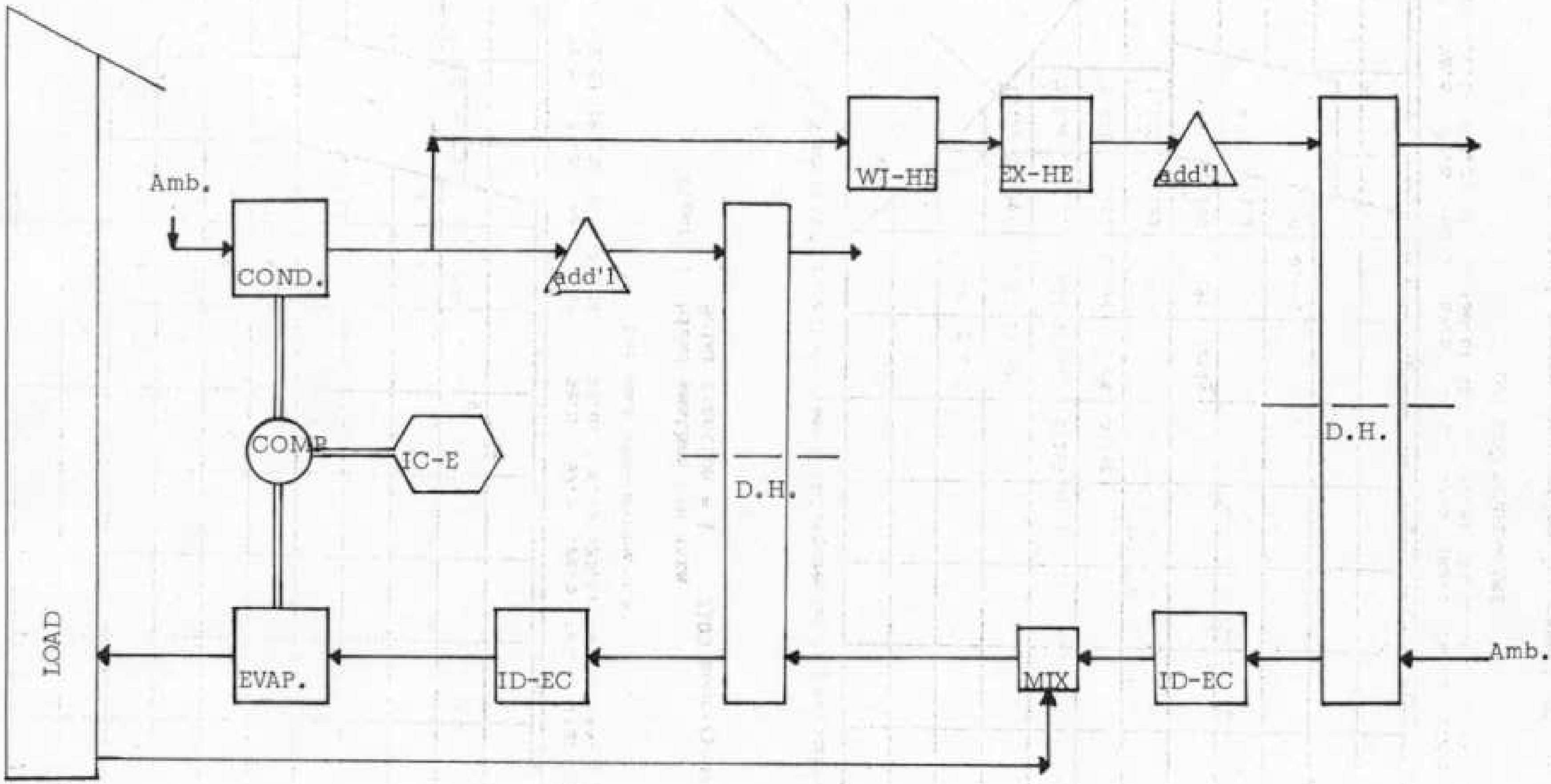


Figure 9 Possible configuration of a two wheel desiccant augmented TAHP cooling system