Modeling a Thermally Activated Heat Pump with Desiccant Cooling

ROBERT H. TURNER, JAMES D. KLEISER and RUEY FENG CHEN
Department of Mechanical Engineering
University of Nevada—Reno
and
NORBERTO DOMINGO and FANG CHEN
Oak Ridge National Laboratory

This work was performed by the University of Nevada at Reno under contract to the Oak Ridge National Laboratory.

ABSTRACT

A thermally activated heat pump (TAHP) with a desiccant wheel for air drying is analyzed and compared to a conventional electrical alternative 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 operation 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 mutual interactions of the cooling load and the cooling system are described and quantified.

SYSTEM DESCRIPTIONS

The 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 air stream by an air-to-air heat exchanger (A-HE, SPs 5 to 6). The processed air then goes to the VC-AC evaporator where it is finally cooled to meet the cooling load line. The process statepoints are depicted in Figure 2.

In order for the system to maintain steady state operation, the desiccant in the dehumidifier wheel must be regenerated, that is, the collected moisture must be driven from the desiccant. Regeneration is accomplished by passing a hot secondary air stream through a part of the wheel. The heat for this secondary stream can often be derived largely from the system waste heat. Figure 2 shows that the regeneration heat is provided in part by the VC-AC condenser, as well as by the IC-E water jacket (WH-HE) and engine exhaust heat exchanger (EX-HE). For conditions where additional heat is needed (SPs K to M) the air-to-air HE (A to B) is also brought into service. Although this causes warmer air to enter the VC-AC condenser, decreasing the COP, if the heat is not obtained from the system waste then natural gas must be burned in the auxiliary heater. Calculations show it is more efficient to reduce the COP for these conditions than to burn additional fuel in the auxiliary heater. Although heat may still be needed between SPs K and M, for most conditions the reject heat is found adequate for regeneration air heating.

Two fans are used in the system, one in the main process loop and one in the regeneration steam, as shown in Figure 1.

The desiccant unit bifurcates the air cooling and air drying operations into two distinct and economical processes. This feature permits the separate control of the two processes, allowing the supply air to be easily matched to the load. The conventional air conditioning system must employ an energy extravagant reheat system to accomplish this, while in the desiccant system the air drying part of the process can be accomplished by inexpensive, low grade energy.

A conventional electric driven VC-AC, which features reheat capability so that the interior humidity ratio can be controlled, is the standard of energy comparison. No energy penalty is assigned for this reheat because available reject heat from the condenser unit is used for the supply air reheat.

Cooling Load Model

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 flowrate.

Infiltration Model

A model has been developed to estimate and quantify the effect of excess building ventilation pressure on infiltration and associated cooling loads [1]. Reducing infiltration reduces the sensible and latent cooling loads concomitant with infiltration. This reduction is associated with building air exfiltration caused by the cooling system operating in a net ventilation mode.

The model assumes that with no house pressurization (no net exfiltration) the building air infiltration rate is 1 air change per hour (1ACH). The complete analysis is described by Kleiser [1]. The equation developed by Kleiser is (V is in scfm/TonAC)

$$V_{infil} = A\,V_{vent}^3 + B\,V_{vent}^2 + C\,V_{vent} + D \quad (1)$$

Where:

A = -4.916 E-6
B = 2.248 E-4
C = -0.1294
D = 67 cfm/TonAC

Knowing the infiltration rate will allow the calculation of the total sensible and latent loads. Since load, and thus cost of operation, varies with net ventilation, the net ventilation provided by the system becomes one of the variables subject to optimization. Figure 3 is a plot of infiltration versus forced ventilation [1].

Notice that as a result of pressurizing the house both the sensible and latent loads are decreased, and that if V_{vent} is equal to 216 cfm/TonAC, loads due to infiltration become non-existent. We define a pressurization factor, pf, such that:

$$pf = V_{infil} / 216;\ pf \leqslant 1 \quad (2)$$

If forced exfiltration exceeds 216 cfm/TonAC, there is no increased benefit upon the load, while there is probably a negative impact upon operating costs. However, if pressurization is provided which is less than 216 cfm/TonAC, the beneficial effects fall very quickly, so that, for instance, infiltration is half its maximum value when forced ventilation is about 160 cfm/TonAC. This implies that for maximum benefit, the house must be fully pressurized, or not pressurized at all, but never partially pressurized.

House Loads

From the above infiltration flowrate the latent and sensible infiltration loads may be calculated. Knowing pressurization, Figure 3 may be used to estimate V_{infil}. Then we get:

-Latent load due to infiltration:

$$Qlinf = 60 * ro * h_{fg} * V_{infil} * (W_{amb} - W_{house}) \quad (3)$$

-Sensible load due to infiltration:

$$QsinF = 60 * ro * c_P * V_{infil} * (T_{amb} - T_{house}) \quad (4)$$

where: ro = .075 lb/ft3 standard air density
h_{fg} = 1050 Btu/lb latent heat of vaporization
C_P = .24 Btu/lb-F specific heat of air

The other sources of latent and sensible load include:

-Sensible radiant solar heat (Btu/hr-TonAC):

$$Qsol = sf * 2000 \quad (5)$$

where: $0 \leqslant sf \leqslant 1$ solar fraction

Thus during a sunny day sf = 1, while at night sf = 0. On an overcast or partly cloudy day, or during the time period which includes sunrise or sunset, sf = (a fractional value).

-Sensible heat conducted through the walls which is not due to solar radiation:

$$Qwall = UA\,(T_{amb} - T_{house}) \quad (6)$$

where: UA = 234.5 Btu/hr-F-TonAC overall conductance of building shell

-Sensible heat which is internally generated, such as by lights, cooking, and persons: This is assumed to be a constant "reasonable" value of 1200 Btu/hr-TonAC.

-Latent heat which is internally generated: This, again, is assumed to be a constant "reasonable" value of 700 Btu/hr-TonAC.

All calculations consider the normalized design load to be 12,000 Btu/hr (12560 kJ/hr = one TonAC) at ARI ambient design conditions, which are dry bulb temperature (DBT) of 95°F (35 C) and wet bulb temperature (WBT) of 75°F (24 C), at Sea level. This load is the sum of both the sensible heat load (Qsen) and the latent heat load (Qlat) at ARI ambient conditions, as this is the normalized load that the system will have to meet. For the purpose of calculating house conductance resistance, and also solar gain, it is assumed that 500 square feet (46.5 square meters) is associated with 1 TonAC at ARI ambient conditions. Thus all results will be expressed in a "per ton" basis.

Supply Air Characteristics

Each system must deliver supply air at a "reasonable" temperature and flowrate to the house, such that the house temperature of 75°F and humidity ratio of .008 lb_W/lb_A are maintained. It is acceptable that either the house DBT or humidity ratio drift below these criteria, provided that no extra energy is required. A reasonable supply air temperature is in the range of 45° - 65°F, while a rule-of-thumb flowrate for an air conditioning system is around 400 cfm/ton-cooling. The analysis assumes steady-state conditions.

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 [2]. If it does not, then steady state room conditions are not attained, as the room statepoint must then change until its SHR line does intersect the process air statepoint. For our purposes, we first assume a reasonable inlet volume flowrate (V1 = 400 scfm) and then calculate the humidity ratio and temperature required to hit the load line. The SHR line is shown in Figure 2.

System Element Models

Both the Desiccant-assisted TAHP and the conventional electrical air conditioning system are composed of a number of separate units, each of which is mathematically modelled, and whose interrelationships are mathematically described.

Silica Gel Desiccant Wheel Model

Jurinak's potential equations [3,4] are used in this analysis to model the behavior of the silica gel desiccant wheel. These equations provide a pair of potentials for each of two inlets and one outlet, and also an effectiveness equation relating the potential equations. Assuming a pair of effectivenesses for the desiccant wheel, and given the temperature and humidity at each of two of the three ports, the temperature and humidity at the third port can be found. Then heat and mass balances will yield the conditions at the fourth port. The situation is described in Figure 4.

The problem with this solution to a desiccant wheel which is incorporated into a system with other components is that normally the complete statepoint at each of two ports is not known. Rather, what is usually known are the temperature and humidity at one port, and the humidities at each of two other ports. The unknown is not simply a complete statepoint, but rather a pair temperatures at two separate ports.

It is possible to iterate a solution through Jurinak's equations, using Newton's method. However, we sought and found an explicit solution for the situation of interest [1].

As given, Jurinak's F-potentials are:

$$F_{1,n} = \frac{-2865}{T_n^{1.490}} + 4.244\, W_n^{0.8624} \qquad (7)$$

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

(n=a,b,or c, ports of the DH wheel)

and the effectivenesses relating them are:

$$e_i = \frac{F_{i,b} - F_{i,a}}{F_{i,c} - F_{i,a}} \qquad (8)$$

(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.

In our simulations we know that humidity and temperature of the process stream into the wheel, the (required) humidity of the process stream out, and the humidity of the regeneration stream in. We need the temperatures of the process-out stream and the regeneration-in stream (see Figure 4).

Thus we have two equations (e_1 and e_2) and two unknowns (T_b and T_c). By manipulating the equations algebraically, we arrive at a quadratic equation explicit in the unknown temperatures:

$$[N_1 e_2]x^2+[N_1 N + 2865(1-e_1 e_2)]x-[2865\, e_1 N_2] = 0 \qquad (9)$$

where

$$N_1 = (1 - e_1)\, F_{1,a} + 4.244\, (e_1 W_c^{0.8624} - W_a^{0.8624})$$

and

$$N_2 = (6360\, [1 - e_2) F_{2,a} + 1.127 (W_b^{0.7969} - e_2 W_c^{0.07969}$$

and

$$N_2 = 6360\, [(1-e_2)\, F_{2,a} + 1.127\, (W_b^{0.07969} - e_2 W_c^{0.07969})].$$

In use, values are installed into the N equations, which are solved and substituted into the quadratic. The quadratic is then solved for x using the quadratic formula, and since $x = T_c^{1.490}$

$$T_c = x^{0.67114} \qquad (10)$$

The other temperature, T_b, is (see Figure 5):

$$T_b = (N_2 + e_2 T_c^{1.490})0.67114 \qquad (11)$$

All tempertures in the desiccant solution are in Kelvin. Humidities are mass ratios.

Useful results of modelling the desiccant wheel are shown in Figures 5,6,and 7. In Figures 5 through 7, B is the ventilation ratio of ambient air introduced to the colling system. For example, B = 0.8 indicates that at SP 3 in Figures 1 and 2, 80% of the air is drawn in from the abmient (80% ventilation). These plots [5] are graphical solutions to Jurinak's potential model. They can be used for a fast solution of the desiccant wheel's state points. They allow an exploratory hand calculation to quickly yield acceptable results, with the benefit of visualization.

Heat Exchanger and Auxiliary Heater

Energy conservation requires that as moisture is removed from the process stream, the desiccant, and thus the airstream, becomes heated. The process stream temperature as it exits the dehumidifier is typically 90° to 120° F [5]. Under cool, moist ambient conditions, the waste heat from the IC- engine and VC- unit may be insufficient to regenerate the desiccant. In this case a heat exchanger from the process airstream, to the regeneration airstream, can become practical. It is located (see Figure 1) so that it recovers heat from the desiccating process and returns it to the regeneration stream.

It is practical if its use results in lower overall rate of energy consumption. If waste heat from the VC-unit and IC engine is enough to regenerate the desiccant without using the heat exchanger, then its use is unwarranted, as it wills erve only to reduce the COP of the VC-unit. But if the machinery's waste heat is insufficient to regenerate the desiccant, then the heat exchanger can preheat the incoming regeneration airstream.

The heat exchanger effect is three-fold: the added heat (SPs 5-6 in Figure 1) raises the temperature of the airstream directly, and the reduction in the VC-unit's COP causes greater heat rejection from both the VC-condenser and the IC engine waste heat exchangers. The overall effect of the heat exchanger is to increase the temperture available for regenerating the desiccant.

Under some cool, moist ambient conditions, waste heat, even with the heat exchanger, may not be enough to regenerate the desiccant. At these times additional heat from a gas burner may still be required.

Indirect Evaporative Cooler

To reduce the cooling requirement of the VC-AC evaporator, the process airstream is cooled by an indirect evaporative cooler (ID-EC) before it enters the evaporator. See Figure 1. The ID-EC consists of a device which cools a secondary stream of ambient air by evaporating water into it, and then transfers process airstream heat to this cool and moist secondary airstream. We can consider the device to be comprised of two coupled units: a direct evaporative cooler (D- EC), and a heat exchanger (HE). The effectiveness of the whole unit is the product of the effectivenesses of the two parts of the unit. This ID-EC component is itself subject to optimization [6]. For convenience, in this study we consider the indirect evaporative cooler to have a single effectiveness of 0.85 which is easily attainable with cellulose wetting fillers.

HVAC-Unit Cop and Power Requirement

In his Master's thesis Howe [4] presents an equation which he uses for the determination of the COP of a vapor compression heat pump. The equation is a curve fit to data for a range of air conditioning units.

For inputs, Howe's equation requires the dry bulb temperature of the airstream into the condenser, the wet bulb temperature into the evaporator, the rated size of the unit, and the operating load fraction. Howe's COP equation is used in our study to predict performance of the vapor compressor air conditioning unit. Knowing the (sensible) heat which must be removed from the process stream by the vapor compression unit, The COP value will allow determination of the shaft power required of the engine, as well as the amount of heat which must be rejected by the condenser. Howe's equation is useful to this study because it offers partial load data for life-cycle studies. However, there is opportunity for future work here.

Howe's COP equation, converted to Fahrenheit, is:

$$COP = N1 + N2 + N3 \qquad (12)$$

where:

$N1 = 3.68+0.162*1r*tn*EXP(-0.183*1r*tn)$

$N2 = -0.753*1r-0.0073*tn$

$N3 = -0.02221*(T_{cond}-T_{wb,evap}-27)$

and: 1r = output as fraction of rated size

tn = rated size of VC-unit (TonAC)

T_{cond} = temperature (^oF) of stream entering condenser

$T_{wb,evap}$ = WB temperature (^oF) of stream entering evaporator

Once the COP is known then power output required of the engine in Btu/hour) is:

$$POWER = Q_{evap}/ COP \qquad (13)$$

The heat rejected to the vapor compressor condenser is:

$$Q_{cond} = \frac{1 + COP}{COP} Q_{evap} \qquad (14)$$

Engine and Waste Heat Distribution

Once the required engine shaft power is known, a study by Segaser [7] is used to determine the amount of fuel necessary to power a gas-fired internal combustion engine (IC-E), and also determine the ultimate distribution of energy from the fuel.

Segaser's engine model gives engine efficiency and heat distribution in percent as a function of percent of maximum load. Different sets of curves are provided for naturally-aspirated gas engines, turbo-charged gas engines, engines with or without exhaust manifold cooling, diesel engines, etc. The cubic-polynomial curve-fit equations were derived from a least squares fit to data from over one hundred engines ranging in size from less than 10 to more than 10,000 horsepower. This study covers small gas engine powered desiccant cooling systems, and so uses the curves for a naturally-aspirated, gas fueled engine without exhaust manifold cooling. Heat distribution profiles for the engine used in this study are given in Figure 8.

For purposes of this model, it has been assumed that lube oil heat will be added to the water jacket heat. This might be accomplished with an oil to water heat exchanger in an oil supply line and is commonly done in practice.

All heat to the water jacket is fully recoverable due to first law considerations, and is useful due to the high water temperature. However, unlike heat to the water jacket or lubricating oil, heat to the exhaust stream is not fully recoverable. Some heat can be picked up with a heat exchanger, but practically, the temperature of the exhaust gas must not be allowed to fall below about 325^oF (=163^oC) or corrosion and deposition problems develop. Thus an exhaust gas heat exchanger will have an effectiveness of about .65 [7].

Once the heat from the various engine elements has been determined, the temperature rise of the airstream which receives the heat is easily found from a heat balance.

Fan Power

To move air through the pressure gradient of the system, two fans are assumed, as shown in Figure 1. Dampers are employed to yield the correct ventilation ratio. The fans are assumed to be powered by electric motors, whose electricity is externally provided by the power company. We assume motor efficiencies of 75 percent [8].

Pressure drop, or back pressure, for each circuit is the sum of the back pressures of each of the elements in series. For expected air flowrates, the pressure drops used were: 0.60 in-water (150 Pa) for the dehumidifier wheel [9]; 0.1 in-water (25 Pa) for each heat exchanger [10]; and 0.1 in-water for each filter [11]. The heat exchangers and evaporative coolers are modelled as water-to-air heat exchangers for purposes of finging their air flow resistance.

If the pressure drop and volume flowrate are known, then reference to a fan multi-rating table will yield the power required to deliver this flowrate. Sea level density was used here. The equation for fan power, developed from manufacturer's fan multi-rating tables [12], is (dP = in-water):

$$\text{Specific Power} = A*(dP**2) + B*dp + C \qquad (16)$$

with FAN POWER = Flowrate * Specific Power

and A = -1.067 E-5 B = 2.680 E-4 C = -1.233 E-5

Conventional Electric HVAC System

The conventional system doesn't separate its approaches to the different types of load, latent and sensible. Both are handled by the evaporator. In order to dry the process stream sufficiently to meet the latent load, the temperature of the air passing through the evaporator must be driven low enough so that condensation removes the excess water. In most cases the resulting airstream temperature is too low to precisely meet the sensible load. However, in some hot and dry ambient conditions the temperature is not low enough when the air is dried to the latent load requirements, and the room temperature will rise. The first case is controlled by the latent load while the second is controlled by the sensible load.

The model employed assumes that the air leaving the evaporator is at 90 percent relative humidity. It decides whether the load is controlled by the latent or sensible load, and solves for the heat removal required and the outlet stream state point. If the load is latent-controlled, then a reheat of the stream to meet load line is assumed, with free heat coming from waste condenser heat. If the load is sensible-controlled, the process stream is really too dry to meet the load line. This is ignored, as it is felt that excessive dryness with air conditioning is not usually a problem. In order to control the total output of the conventional electric air conditioner, pulse-width modulation is used, in which it is assumed that the machine is cycled on and off with the relative times in each phase controlling the effective output.

ANALYSIS

The component interactions described above have been programmed to simulate the system for two operational modes: (1) Pressurized; and (2) Unpressurized. Referring to Figure 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 pressurized case, ambient ventilation air (fraction B of M1) is mixed with the house return air. M1 is greater than the air flowrate withdrawn from the house by the system, and this difference causes a positive house pressure (pf > 0), causing net exfiltration from the house. Figure 3 indicates that for a ventilation flowrate 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 which the system must condition is usually warmer and more moist than the house air. Thus the two effects of load reduction and of higher enthalpy air into the system impact in opposite directions, and generally do not compensate equally.

For a given ambient condition the energy required to maintain the house at the design condition is calculated, for both pressurized and unpressurized cases. Universal Energy Consumption Tables were produced to estimate the thermal and electrical power requirement to maintain the required cooling load. They are presented in Turner et al [13], along with other complete details of this study which cannot be included here. A similar Universal Energy consumption Table was produced for the conventional electric air conditioner.

Annual energy consumption estimates are made using actual 1986 summer weather data (NOAA Local Climatological Weather Data). These 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, there are 16 such tables presented in Turner et al [13].

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 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 consumtion [14].

64 such maps are presented in Turner et al [13] Figure 9 shows a typical energy consumption sheet for the desiccant cooling system in Miami, with pressure factor (PF) zero and solar fraction (sf) unity. 171.7 Terms/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 Therm/year-TonAC were required in Miami in 1986.

Knowing how many hours the system ran, and air flowrates and pressure drops, the electrical power is computed (993 kW-hr/year-TonAC). These values of natural gas and electrical energy consumption appear in Table 1 (Case 1, PF = 1) for Miami.

Assuming natural gas heat cost of $0.50/Therm, and electricity cost of $0.08/kW-hr, the annual cost of cooling is estimated in Table 1 for both the desiccant cooling and conventional air conditioning systems. Table 1 also indicates the total number of cooling hours recorded in 1986 for the 8 locations.

Results and Conclusions

Table 1 is a summary of the seasonal results for each city and machine through the summer season. The table provides a comparative assessment of both energy and money annual consumption. The energy costs used are $0.50 per Therm for gas, and $0.08 per kWhr for electricity.

The most expensive location with either unit, in energy or in dollars, is Miami, while Sacramento is the least expensive. This follows from Miami's requiring the greatest number of hours of cooling (4416 hours), with Sacramento requiring the least (954 hours).

The desiccant cooling system is less expensive to operate than the conventional system in every city examined. The main reason for this is that the desiccant system, uses low grade waste heat to meet a significant portion of the total load, while the conventional system must use high grade, expensive electric energy to meet all the load. Actually, the conventional machine, in order to meet the latent load, 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 necessary. A reheat of the process stream will put the statepoint 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 air conditioner uses more energy is that it lacks the indirect evaporative cooler (ID-EC). The only path for heat to leave the house is through the vapor compressor.

All the factors taken together lead to the conventional system costing nearly twice as much to operate as the desiccant system.

REFERENCES:

[1] Kleiser, J.D., "Modelling a Desiccant Cooling System", M.S. Thesis, University of Nevada-Reno, 1987.

[2] Clifford, G.E., Heating, Ventilating, and Air Conditioning, Reston Publishing Co., Reston, Va. 1984.

[3] Jurinak, J.J., "Open Cycle Solid Desiccant Cooling--Component Models and System Simulations", University of Wisconsin-Madison, Ph.D. 1982.

[4] Howe, R.R., "Model and Performance Characteristics of a Commercially-Sized Hybrid Air Conditioning System Which Utilizes a Rotary Desiccant Dehumidifier", M.S. Thesis, University of Wisconsin-Madison, 1983.

[5] Turner, Kleiser and Chen (1), "Design Factors in Desiccant Cooling Systems", ASME-JSME Solar Energy Conference, 1987-1.

[6] Moosavy, Morteza, "The Optimization of Indirect Evaporative Coolers", M.S. Thesis, University of Nevada-Reno, 1987.

[7] Segaser, Charles L., "Internal Combustion Piston Engines", (ANL/CES/TE 77-1), Oak Ridge National Laboratory, 1977.

[8] Marks Handbook, eighth ed.; McGraw Hill, 1978.

[9] Rousseau, J., "Development of a Solar Desiccant Dehumidifier (82-18944)", Airesearch Manufacturing Company, 1982.

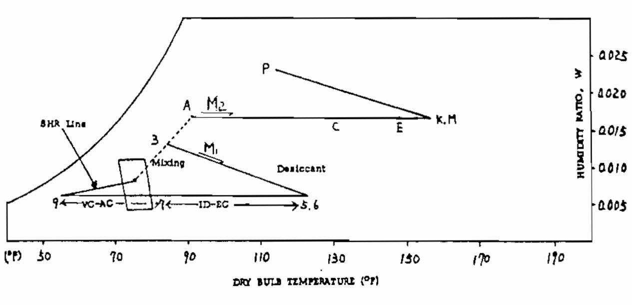
[10] Airmax Commercial Heat Exchanger Literature, 1984.

[11] Granger Catalogue of Commercial Hardware, 1987.

[12] Trane Company, "Fan Multirating Tables from Fan Literature", 1983.

[13] Turner, Kleiser and Chen (2), Report to Oak Ridge National Laboratories, May, 1987-2.

[14] Chen, Ruey-Feng, "Assessments of Desiccant Cooling Systems", M.S. Thesis, University of Nevada-Reno, 1987.



Note operation is shown for PF = 1 and af = 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 : TAHP-with-Desiccant Unit Shown in Figure I

	TAHP;PF=0	TAHP;PF=1	CONVENTIONAL (MOTOR)	
ENERGY	therms	therms	kW-hr	COOLING
FAN POWER	kW-hr	kW-hr	kW-hr	HOURS
TOTAL COST	$	$	$	
MIAMI	351.936 993.600 $255.46	364.425 993.600 $261.70	6650.32 463.680 $561.12	4416
ATLANTA	166.719 542.700 $126.78	157.488 542.700 $122.16	2903.19 253.260 $252.52	2412
PHILADELPHIA	102.546 351.000 $79.35	94.140 351.000 $75.15	1846.44 163.800 $160.82	1560
CHICAGO	67.929 231.525 $52.49	65.760 231.525 $51.40	1232.34 108.045 $107.23	1029
MEMPHIS	245.502 692.550 $178.76	242.709 692.550 $176.76	4262.70 323.190 $366.87	3078
HOUSTON	286.005 803.925 $207.32	281.487 803.925 $205.06	5193.84 375.165 $445.52	3573
PHOENIX	230.934 770.175 $177.08	203.922 770.175 $163.58	4613.49 359.415 $397.83	3423
SACRAMENTO	53.814 214.650 $44.08	49.338 214.650 $41.84	1077.30 100.170 $94.20	954

NOTE: 1 ONE therm = $0.50
2 ONE kW-hr = $0.08
3 FAN POWER = 0.225 kW/TonAC FOR TAHP
4 FAN POWER = 0.105 kW/TonAC FOR CONVENTIONAL

TABLE I: Annual Energy and Money Consumption Summary - All Cases

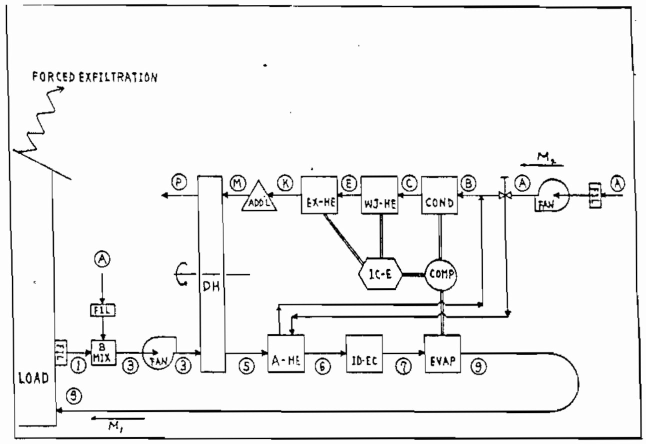


Figure I Thermally Activated Heat Pump Desiccant Cooling System with Indirect Evaporative Cooler. The Unit has Variable Ventilation Capability and can Pressurize the House to Reduce Infiltration.

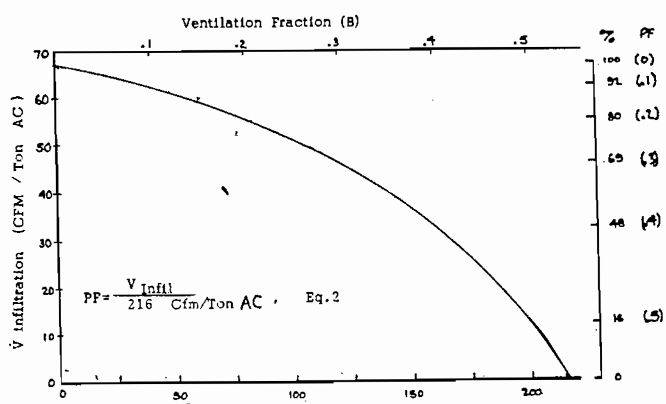


FIGURE 3 INFILTRATION vs FORCED VENTILATION

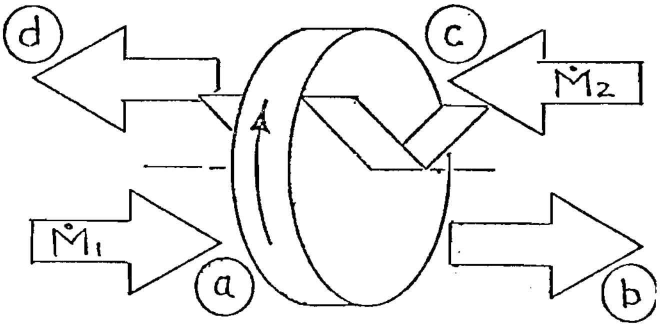


Figure 4 : Flow Through Desiccant Wheel

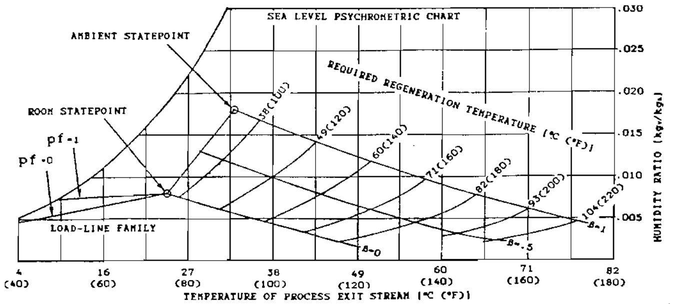


Figure 5: PROCESS HUMIDITY RATIO vs. PROCESS EXIT TEMPERATURE
REGENERATION HUMIDITY RATIO = .18

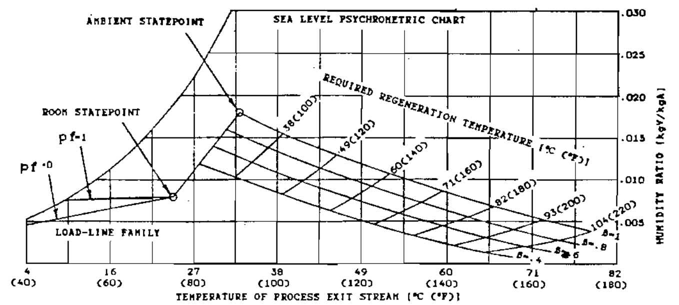


Figure 6: PROCESS HUMIDITY RATIO vs. PROCESS EXIT TEMPERATURE
REGENERATION HUMIDITY RATIO = .14

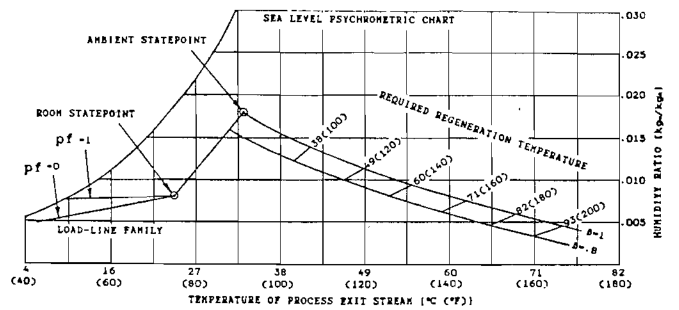
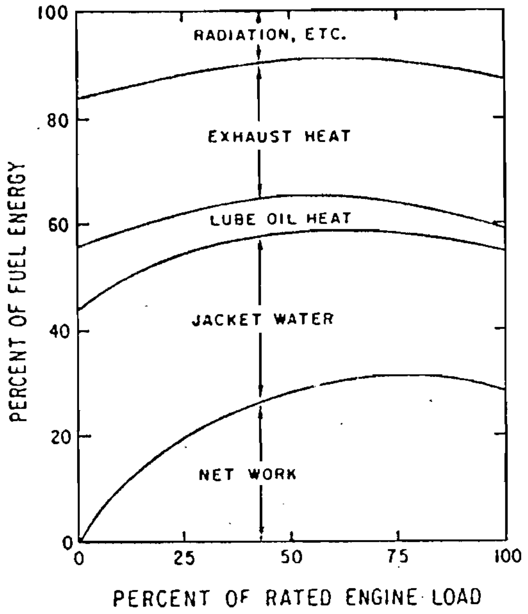


Figure 7: PROCESS HUMIDITY RATIO vs. PROCESS EXIT TEMPERATURE
REGENERATION HUMIDITY RATIO = .10



Source: Segaser, C. L., "Internal Combustion Piston Engines"; Report ANL/CES/TE 77-1; July 1977

Heat balance for naturally aspired spark ignition gas engine with hot exhaust manifold. The energy input in the fuel appears as brake horsepower as heat rejected to the jacket cooling water, as heat rejected to the lube oil, as heat rejected to the exhaust gas and as heat lost by radiation and natural convection.

FIGURE 8

Heat Balance for IC-Engine Used in this Study

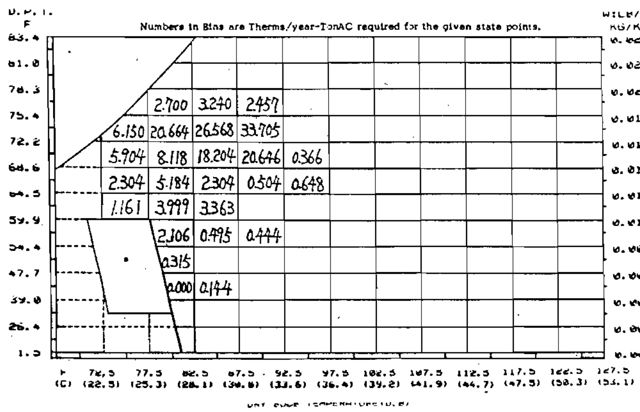


FIGURE 9 MIAMI Energy Consumption for Case 1

PF=0 sf=1 171.7 Therms/year-Ton A C