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Hybrid Desiccant Cooling Systems in Supermarket Applications

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ABSTRACT

Supermarkets are major consumers of electrical energy. The presence of refrigerated cases in supermarkets makes moisture removal the primary air-conditioning requirement. Hybrid desiccant cooling systems have been proposed as an alternative to conventional vapor compression systems as a means of reducing electrical energy consumption and energy costs. This paper studies the performance of three possible hybrid system configurations in supermarket applications and compares their performance with the traditional vapor-compression system. Results presented suggest a total air-conditioning savings of 60% with hybrid systems for the design condition considered.

INTRODUCTION

Supermarkets consume nearly 4% of the U.S. electrical energy. Methods that reduce electrical energy use in these stores will have substantial impact. While the largest portion of this electricity is necessary to maintain low temperatures in the open refrigerated cases, significant reductions can be made in the operating costs of the air conditioning systems in these buildings.

The air-conditioning situation in a supermarket is substantially different from that of a standard commercial office building. The open refrigerated cases reduce the load that a cooling system has to meet. The cases provide more sensible cooling than latent cooling, so the remaining load tends to be largely of a latent nature. The size of the remaining load is strongly dependent on the ambient conditions, since the cooling effect of the cases will offset the internally generated load (lights, people).

The open refrigerated cases necessitate strict humidity control. The purpose of the refrigerated cases is to provide a reduced temperature environment for food products. Since the cases are open, considerable warm and humid store air is entrained, with resulting condensation in the cases. The higher the store humidity level, the more compression work required of the refrigerated cases. By industry standards, the design ambient condition for refrigerated cases is 75 F (24°C) and 55% rh (.0104 kg/kg). If store conditions exceed this design condition, the loads become greater than the cases can handle with the possible consequence of product spoilage. The air-conditioning system in a supermarket is thus required to maintain store conditions below 75 F (24°C) and 55% rh.

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Traditionally in commercial building air conditioning, the vapor-compression machine has been used to meet both the latent and sensible cooling requirements. To remove moisture, air must be cooled below its dewpoint. When only dehumidification is required, this air must be heated back up to the desired temperature. Vapor-compression air conditioners are designed to operate at moderate evaporator temperatures, which limits the amount of moisture that can be removed per unit mass of air. When large amounts of dehumidification are required it is necessary to process large amounts of air. Since supermarket loads are highly latent, larger amounts of air are circulated through the cooling system relative to the size of the cooling load than is standard in most commercial buildings. The need to maintain the evaporator temperature and the resulting larger airflow rates make maintaining the 55% rh requirement difficult and attempts to maintain even lower humidity levels unfeasible.

Open-cycle desiccant cooling systems have been considered as an alternative to vapor compression cooling. Jurinak (1982) presents an overview of this work. Sheridan and Mitchell (1982) and Howe, et al., (1983) have studied hybrid desiccant systems in commercial applications. A hybrid system has been installed in a Chicago supermarket (Cohen et al. 1983) under the sponsorship of the Gas Research Institute. Hybrid systems designed for supermarkets are now being marketed (Banks, 1984), with most of these installations in Texas.

In this paper, various hybrid system configurations are considered. Component models and interactions are discussed. Comparisons of performance between the various hybrid systems and in relationship to the standard vapor compression system are made. The ability of a hybrid system to reduce the refrigerating load in the store is discussed.

HYBRID DESICCANT CYCLES

Hybrid desiccant cooling systems combine a desiccant dehumidifier with a vapor-compression unit to meet the building air-conditioning load. A hybrid system utilizes the desiccant to meet the latent load, and a vapor-compression unit to handle the sensible portion. Heat exchangers and evaporative coolers can be added to handle a portion of the sensible load. Since the vapor-compression unit in a hybrid system only has to remove a portion of the load, the electrical energy consumption of the cooling system is substantially reduced over that for a conventional system. In addition, since the vapor-compression unit no longer has to cool air below dewpoint temperatures, the evaporator temperature may be raised. This increases the COP of the machine and further reduces vapor-compression work. This decrease in electrical energy use is not free; the moisture adsorbed by the desiccant must be removed with a high temperature airstream. Typically this regenerative heat is supplied with a gas burner. There is a trade-off between electrical energy and thermal energy consumption.

By separating the performance of dehumidification and sensible cooling into two different components, it is now possible to reduce humidity without simultaneously reducing supply air temperature below desired levels. Dehumidification is no longer dependent on the limits of the vapor-compression machine. This allows both lower store humidity levels and lower circulation flow rates to be maintained.

Various designs for desiccant dehumidifiers have been proposed and studied (Barlow 1983). The most promising configuration is a rotary wheel containing a porous matrix of desiccant material. Two separate airstreams pass through the matrix in a counterflow arrangement. The most common substances used are silica gel, lithium chloride, and molecular sieve. Fan power is required to move the air through the wheel, so the matrix should be designed in such a way as to minimize the pressure drop. With this in mind, Dunkle et al. (1980) have proposed parallel air flow passages. SERI (Barlow 1983) is developing a wheel consisting of spirally wound thin strips of tape lined with silica gel which form parallel passages.

Howe et al. (1983) studied a hybrid cooling system that processes all circulated air through the desiccant and utilizes condenser heat to preheat regeneration air. This system proved to be promising for commercial building applications with a projected electrical savings of 40% and will be evaluated for supermarket applications. A schematic diagram of this cycle, here called "recirculation/condenser", appears in Figure 1. In this particular configuration, air from the store is mixed with ventilating air (state 1) and processed through the desiccant. The adsorption process in the desiccant releases latent heat causing hot, dry air to leave the dehumidifier (state 2). The airstream is then cooled with an indirect evaporative cooler (state 3). The remainder of the sensible cooling is performed by the vapor-compression unit and the conditioned airstream is supplied to the store (state 4). To regenerate the desiccant, waste condenser heat is used to preheat ambient air (state 7). Any further heating necessary is provided by the auxiliary heat source (state 8). The regenerative airstream is cooled and humidified as it passes through the desiccant and then exhausted to the outside (state 9).

Another cycle to be considered (ventilation/condenser) is shown in Figure 2. This cycle is similar to the previous cycle with the exception that only ventilation air is processed. Since less air passes through the desiccant, lower humidity levels are required on the process side to provide the same store humidity level. These lower humidity levels and the higher moisture content of the air entering the desiccant mean that substantially higher regeneration temperatures are required than in the recirculation/condenser cycle.

The third system (ventilation/heat exchanger), shown in Figure 3, also processes ventilating air; however, a sensible heat exchanger is placed after the desiccant on the process side. The process air is cooled by heat exchange with the ambient. The ambient air is heated to the regeneration temperature in the auxiliary heater. The heat exchanger and the dehumidifier have different flow requirements for optimum performance. Flow should be balanced through the heat exchanger and unbalanced through the dehumidifier; therefore some air on the regeneration side is exhausted before entering the auxiliary heat source. This system is the simplest method of obtaining some free cooling and free heating.

Some interesting energy trade-offs exist in the first two cycles which utilize condenser heat. Cooling performed by the indirect evaporative cooler reduces the amount of electrical work required in the vapor compression unit. This, in turn, means there will be less condenser heat to preheat the regeneration stream, requiring more energy to be supplied by the auxiliary heater. Another trade-off occurs with the reclaiming of the condenser heat. To approach the regeneration temperature, the condensing temperature of the vapor compression unit will be higher than normal. This lowers the COP of the vapor compression unit and correspondingly increases the electrical energy consumption needed to meet the cooling load. In both of these situations there is a choice between more efficient cooling on the process side and more efficient heating on the regeneration side.

COMPONENT MODELS

Analysis of the different configurations has been performed using the transient simulation program, TRNSYS (Klein et al. 1983). The following is a brief description of the component models and store parameters used throughout this study.

Jurinak, et al., (1984) and Van den Bulck et al. (1984) have discussed in considerable detail the modeling of the performance of desiccant dehumidifiers, and the optimum operating conditions. The model used to simulate the performance of the desiccant is an effectiveness-NTU model developed by Van den Bulck. Based on the governing equations of heat and mass transfer, this model correlates effectivenesses to the resistances for heat and mass transfer. A moisture effectiveness is defined as

$$\varepsilon_w = (w_i - w_o)/(w_i - w_{ideal}) \tag{1}$$

where w_i is the inlet process humidity, w_{ideal} is the outlet process humidity assuming an ideal dehumidifier with zero resistance to heat and mass transfer, and w_o is the actual outlet humidity accounting for resistance. An enthalpy effectiveness is similarly defined. The wheel modeled is a high performance dehumidifier having a moisture effectiveness close to 0.9. The enthalpy effectiveness very closely approaches 1.0. Energy consumption can be reduced if the two flows through the wheel are unbalanced and an optimum wheel rotation speed is chosen. The minimum auxiliary energy requirements occur if the regeneration flow rate is 80% of the process flow rate and the dimensionless flow rate, Γ_1, is 0.15, where Γ_1 is defined as:

$$\Gamma_1 = \frac{\text{mass of desiccant/time in period}}{\text{mass flow rate of process airstream}} \tag{2}$$

These values were chosen as representative of the optimum values over the conditions expected to be encountered in supermarket applications.

The indirect evaporative cooler is a sensible heat exchanger using evaporatively cooled ambient air as a heat sink. The effectiveness of the evaporative cooling process is defined as:

$$\varepsilon_{EC} = (T_{cool} - T_{in})/(T_{wb} - T_{in}) \tag{3}$$

where T_{in} is the incoming air temperature, T_{wb} is the wet-bulb temperature of the incoming air, and T_{cool} is the temperature to which the air is cooled. This effectiveness is assumed to be 0.95. The effectiveness of the heat exchanger with the minimum capacity rate on the process side is defined as:

$$\varepsilon_{HX} = (T_h - T_o)/(T_h - T_c) \tag{4}$$

where T_o is the temperature of the exiting process air, T_c is the temperature of the entering cool-side air (after evaporative cooling, if applicable), and T_h is the temperature of the entering process (hot-side) air. The heat exchanger effectiveness will be varied form zero to one. The largest heat exchange occurs at high effectiveness; however some systems will not require this. A practical upper limit is an effectiveness of 0.9.

Performance data for vapor-compression machines are usually presented in terms of the ambient air temperature entering the condenser. This assumes that a standard flow rate passses through the condenser. Actually, the performance of a vapor-compression unit is dependent on the condensing temperature rather than the ambient air temperature. In utilizing condenser heat to preheat the regeneration stream, low airflow rates will be used, thereby increasing the condenser temperature. Some data are available relating COP to condensing temperature. Extrapolation to higher condenser temperatures has been made by relating these data to the Carnot COP:

$$COP = k * T_e/(T_c - T_e) \tag{5}$$

where the temperature group is the Carnot COP. The constant, k, is determined from

$$k = COP_{data}/COP_{Carnot} \tag{6}$$

T_e is the evaporator temperature, and T_c is the condensing temperature. The value for the constant, k, was found to be 0.46 for the units studied.

The overall conductance-area product, UA, for a condenser is assumed to stay constant for an individual unit and is determined from the data by assuming a 10°F (4.4°C) log mean temperature difference at ARI standard condition. This model can be extended to machines of different sizes by holding U constant and varying the area in proportion to the capacity of the unit.

An interative solution is used to determine the condensing temperature. The condenser heat rejection can be calculated from the energy balance for the unit. In terms of COP and evaporator heat flow, the condenser heat flow is

$$Q_{cond} = Q_{evap}(1+1/COP) \tag{7}$$

The outlet temperature of the airstream follows from an energy balance on the airstream

$$T_{out} = T_{amb} + Q_{cond}/(\dot{m}c_p) \tag{8}$$

The initial choice of condenser temperature is checked using the LMTD relation for heat transfer

$$Q_{cond} = UA\ (T_{out} - T_{amb})/\ln(T_c - T_{amb})/T_c - T_{out}) \tag{9}$$

where T_{amb} is the ambient temperture.

If the condenser heat rejection found in Equation 7 does not match that in Equation 9 a new condenser temperature is chosen and the iteration repeated.

The supermarket considered is loosely patterned after a store in West Chicago, IL, in which the hybrid system (Cohen, et al., 1983) was installed. The supermarket space contains 30,000 ft^2 (2800 m^2) of floor space and 50 tons (176kW) of installed refrigeration capacity. For purposes of system comparison, the base case, internally generated load adopted after refrigeration reductions are accounted for is 24.3 kW, 65% of which is latent. Typically a store will circulate 1 cfm air/ft^2 (.006kg/s-m^2) of floor space. This is approximately 3,000 cfm (16.7 kg/s) of circulation air. The necessary ventilation air is assumed to be 10% of this amount. It is assumed that all outside air that enters the store is ventilation air and that the store is pressurized to minimize infiltration. The store parameters are summarized in Table 1.

The hybrid systems are controlled such that the regeneration temperature is just sufficient to meet the dehumidification required. With a known load and known flow rates, the necessary humidity level exiting the process side of the dehumidifier is calculated from a mass balance. An iterative solution technique is then used to determine the regeneration temperature that will provide this humidity level.

Energy costs have been expressed in weighted units, with electrical energy consumption weighted twice that of thermal energy usage. This assumes a 35% efficiency in the conversion of fuel to electricity and a 70% efficiency in the auxiliary heater.

SYSTEM COMPARISONS

To illustrate the operation and the energy usage of a hybrid system, a base case example with the ventilation/condenser system is described in some detail. In this example, outdoor ambient condition is 86 F (30°C) and 0.016 kg/kg absolute humidity ratio (60% rh). Store conditions are maintained at 75 F (24°C) and 0.0104 kg/kg (55% rh). The typical amount of air, 30,000 cfm (16.67 kg/s) circulates through the store, with 3,000 cfm (1.67 kg/s) of outside ventilation air processed through the desiccant. The indirect evaporative cooler has an effectiveness of 0.8.

Given the latent load, the flow rates, and the ambient conditions, a mass balance finds the desired absolute humidity level of the process air to be .0066. The regeneration temperature providing this humidity level is 178.5 F (81.4°C). The process air has been heated in the adsorption process to 140 F (60°C). Indirect evaporative cooling provides 48.6 kW of free cooling, reducing the process temperature to 88.3 F (31.3°C). After mixing with the recirculated air, the remainder of the sensible cooling is performed by vapor compression. No further dehumidification is needed. In this case the amount of cooling is 20.5 kW which at a COP of 2.7 requires 7.6 kW of electrical energy consumption in the compressor. 28.1 kW are rejected to the condenser. The condenser heat raises the regeneration stream to a temperature of 123.8 F (51°C). 41.7 kW of auxiliary heat are needed to produce the regeneration temperature of 178.5 F (81.4°C). Figure 2 summarizes these energy flows on the schematic diagram. The total energy cost is 58.3 weighted units, substantially less than the 147 weighted units consumed by the standard vapor compression machine to meet this same load. Figures 1 and 3 provide energy flows and state points for the remaining two cycles.

As noted before, there is a trade-off between the amount of cooling performed by the indirect evaporative cooler (IEC) and the amount of heat available for regeneration from the condenser. By varying the effectiveness of the IEC from 0 to 1.0, which in effect regulates the amount of cooling done by that component, an optimum level of free cooling can be determined. Figure 4 illustrates the breakdown of auxiliary heat, electrical energy, and total energy consumption, expressed in weighted units for the ventilation/condenser configuration as a function of the IEC effectiveness. This figure indicates that the optimal amount of free cooling is just enough so that the condenser heat available can completely regenerate the desiccant. Any less free cooling and the vapor-compression unit performs more work without any further benefit on the regeneration side. If more free cooling is performed, the amount of auxiliary heat needed rises faster than the reduction in vapor-compression work. The rapid increase in the vapor compression work as heat exchanger effectiveness decreases is indicative of the penalty taken in reclaiming condenser heat. As more heat is rejected, the condenser temperature will have to rise. Analysis of the recirculation/circulation cycle show similar tradeoffs in the energy consumption of the system.

Figure 5 shows the total energy consumption of the three systems as a function of heat exchanger effectiveness. The ventilation/heat exchanger system is expected to perform best at a high effectiveness, since there is no free cooling trade-off in the system. The ventilation/condenser and recirculation/condenser systems both have optimal points at intermediate effectiveness. The COP penalty does not play as large a role in the recirculation/condenser system and the decrease in energy cost is significant as more heat is rejected to the regenerative airstream from the condenser. At this lower effectiveness, the recirculation/condenser system is competitive with the ventilation cycles.

A reduction in the recirculation/condenser energy consumption is possible at a lower system flow rates. Due to the limits of the vapor-compression machine discussed earlier, supermarkets have traditionally used a substantially larger amount of circulation airflow than is found in a standard commercial building. Since the hybrid desiccant systems do not require lower evaporator temperatures to remove large amounts of moisture, it is possible to circulate less air. This would have the effect of reducing the fan power required for air circulation. In addition, as the flow rate is reduced the energy consumption of the recirculation/condenser cycle decreases. Figure 6 shows the reduction in total energy at different flow rates for the recirculation/condenser cycle. As the amount of air through the desiccant is decreased both the amount of auxiliary heat required and the amount of vapor compression work decreases. Eventually, at low enough flow rates the recirculation/condenser cycle becomes better than the ventilation cycles.

At a flow rate one third of standard 10,000 cfm (5.6 kg/s), the recirculation/condenser configuration consumes 18% less energy than the ventilation/condenser cycle at optimum heat exchanger effectiveness. Except for fan power, the energy expenditure of the ventilation systems does not change as the circulation flow rate decreases. Since the required amount of ventilated air is assumed to remain the same, the loads and the desiccant performance remains the same.

Figure 7 provides a comparison of the energy breakdowns in the various systems at their optimum system parameters. Table 2 summarizes the operating parameters. The hybrid systems all consume considerably less energy than the standard vapor compression systems. They are all comparable in performance. At these optimum points, all systems utilizing condenser heat do not require auxiliary heat. The regeneration requirements are met in the condenser. At standard system flow rates, despite performing less compressor work, the fan power required to blow the large airflows through the desiccant make the recirculation/condenser cycle less favorable than the ventilation cycles. If the flow rate could be decreased both the vapor compression and the fan work go down. The recirculation/condenser cycle requires a regeneration temperature of only 100 F (38°C). If a large source of moderate-temperature heat were available (condenser heat from refrigerated cases or solar energy), the auxiliary heat requirement could easily be met by heat exchange with this source. With the heating requirement taken care of, the effectiveness of the IEC can be increased and more free cooling realized. The energy costs in this case would be quite low.

Since the electrical energy consumption of the refrigerated cases is the largest energy cost in the supermarket, one of the attractive ideas behind using a desiccant in this application is the ability to maintain lower store humidity levels than the vapor-compression system. Lowering the humidity level in the store reduces the amount of water removal the cases must perform in order to maintain desired case temperatures. Again there is a trade-off between the reduction in electrical energy consumption due to a lower humidity level and the energy cost of maintaining that lower humidity level. Figure 8 displays the total air conditioning energy consumption of the ventilation/condenser cycle at various store humidity levels. These calculations were made considering the standard circulation flow rate, 30,000 cfm (16.67 kg/s), and an ambient condition of 86 F (30°C) and .016 kg/kg absolute humidity ratio. Since the refrigerated cases do less work as humidity levels decrease, the internally generated load increases at a rate of 3.14 kW for every gram reduction in humidity ratio. As humidity levels are reduced regeneration temperatures increase considerably. The auxiliary heat required to meet these high regeneration temperatures, or the performance degradation of the vapor-compression unit if condenser heat is utilized, is so large that the energy consumption of these systems increases considerably as the humidity levels go down. The other two cycles show similar patterns.

CONCLUSIONS

The results presented indicate that considerable energy savings can be obtained by using hybrid desiccant cooling systems in supermarkets. At the design condition considered, a 63% reduction in air-conditioning costs could be realized. Further reductions are possible if the system flow rates are reduced. Of the various hybrid configurations studied none established itself as clearly superior to the others, though all of them do substantially better than the standard vapor-compression system. At standard flow rates, the ventilation cycles do slightly better due to smaller fan requirements through the desiccant. Hybrid desiccant systems are able to maintain lower store humidity levels; however, the increase in energy cost is quite large. In light of this, the benefits of holding lower store humidity levels must be closely scrutinized before utilizing these systems for that purpose. The large savings provided by the hybrid systems at standard store conditions might be lost at lower humidity levels.

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TABLE 1
STORE PARAMETERS

Floor Space	2800 m^2	30,000 ft^2
Generated Load	24.3 kW	83 MBtuh
Latent Load (65%)	15.8 kW	54 MBtuh
Circulation Flow	16.7 kg/s	30,000 cfm
Ventilation Air	1.67 kg/s	3,000 cfm
Refrigeration Capacity	176 kW	50 tons
Store Temperature	24 C	75 F
Standard Store Humidity	.0104 kg/kg	55% RH

TABLE 2
OPERATING CONDITIONS AT OPTIMUM $_{hx}$

	Std VC	Vent/HX	Vent/cond High Flow	Rec/Cond Low Flow	Rec/Cond
System Flow Rate	16.7 kg/s	16.7 kg/s	16.7 kg/s	16.7 kg/s	5.6 kg/s
Regeneration Temperature	---	81.4C	81.4C	38.1C	46.8C
Optimum ε_{hx}	---	0.9	0.44	0.18	0.27
System Energy Consumption (weighted units)	147	50	53	64	46

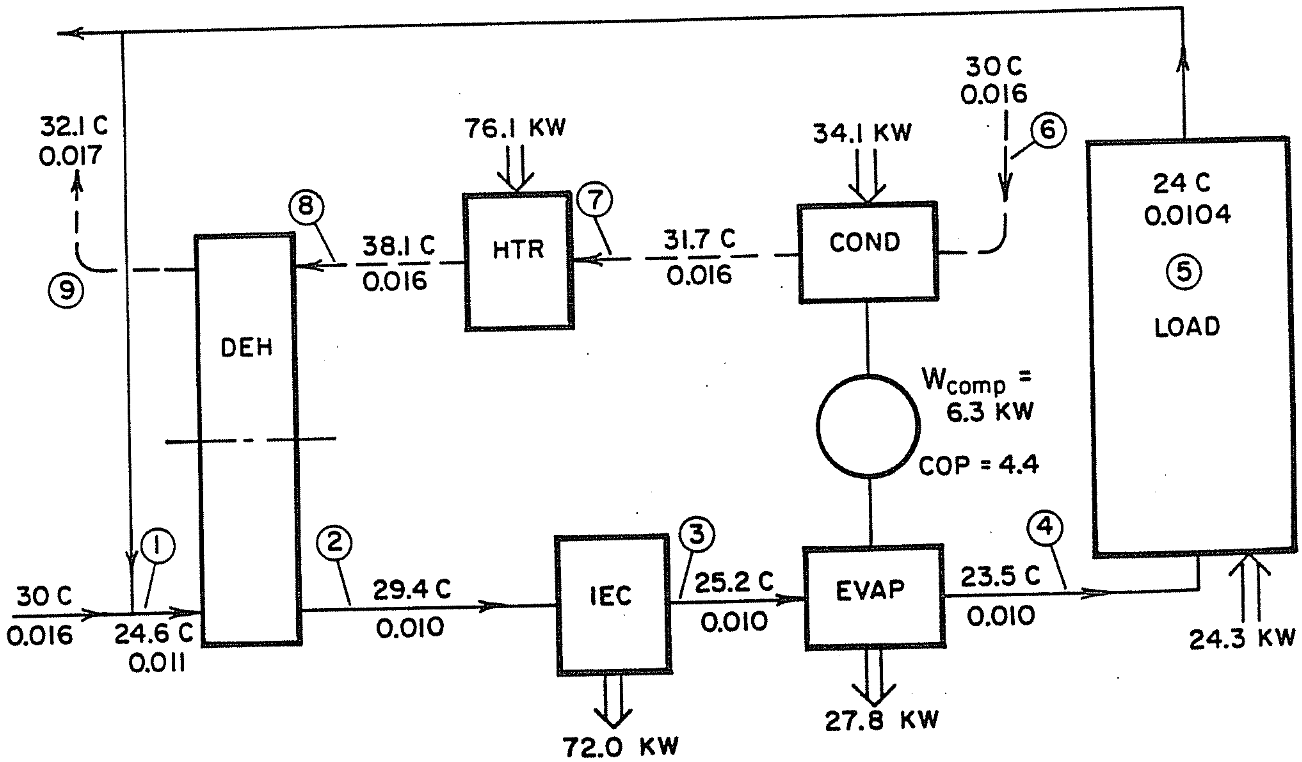

Figure 1. *Schematic diagram of the recirculation/condenser cycle with typical air state values and energy flows*

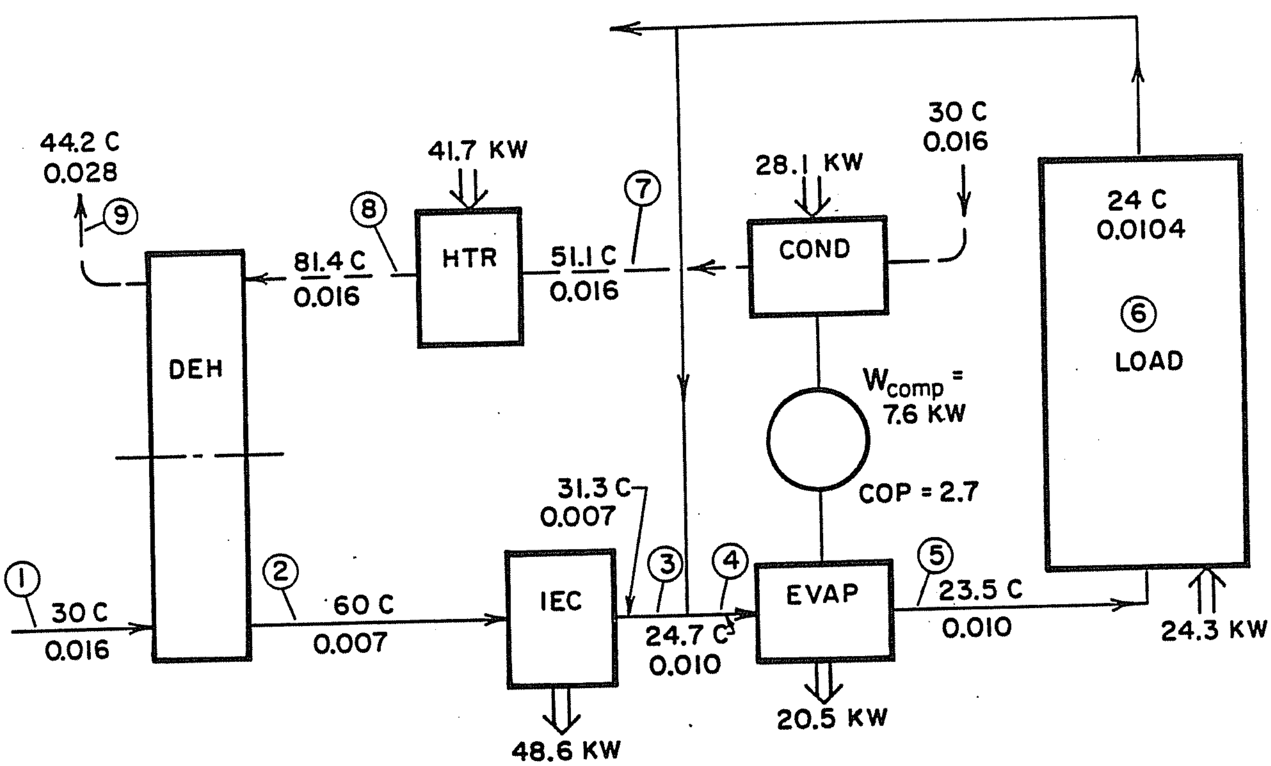

Figure 2. *Schematic diagram of the ventilation/condenser cycle with typical air state values and energy flows*

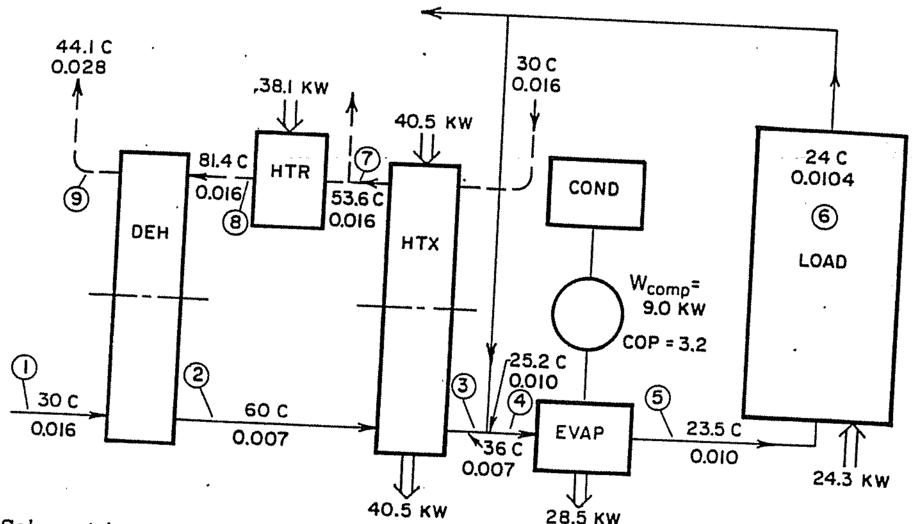

Figure 3. *Schematic diagram of the ventilation/ heat exchanger cycle with typical air state values and energy flows*

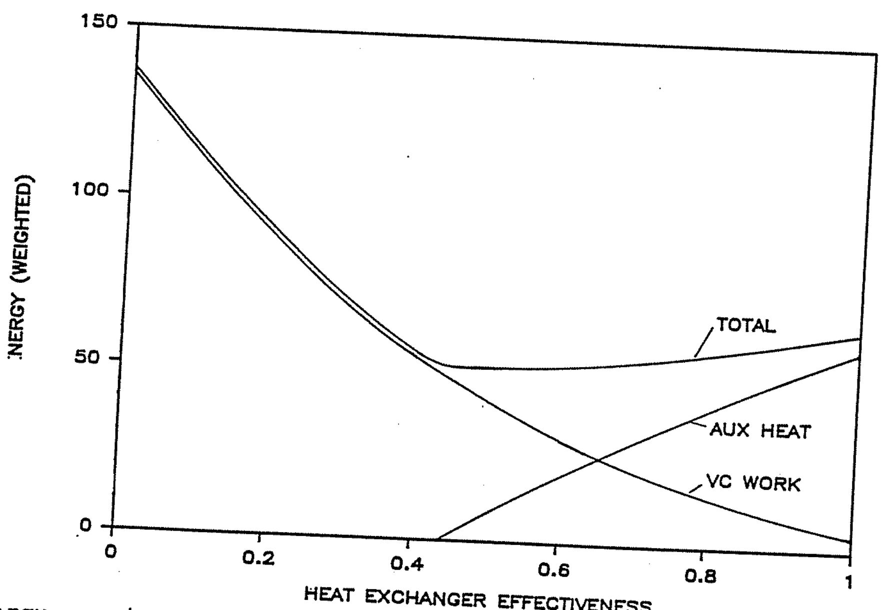

Figure 4. *Energy use in the ventilation/condenser cycle as a function of heat exchanger effectiveness*

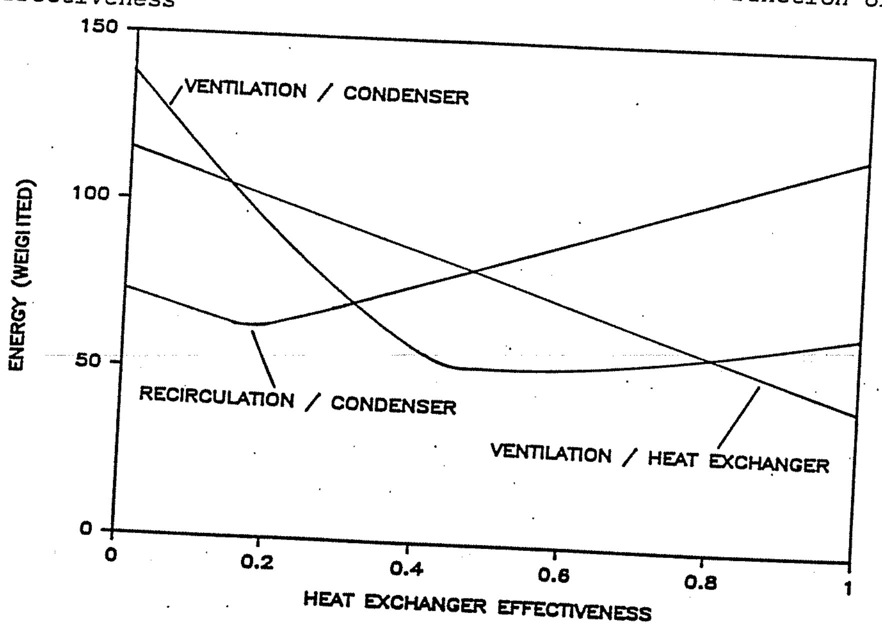

Figure 5. *System energy use as a function of heat exchanger effectiveness*

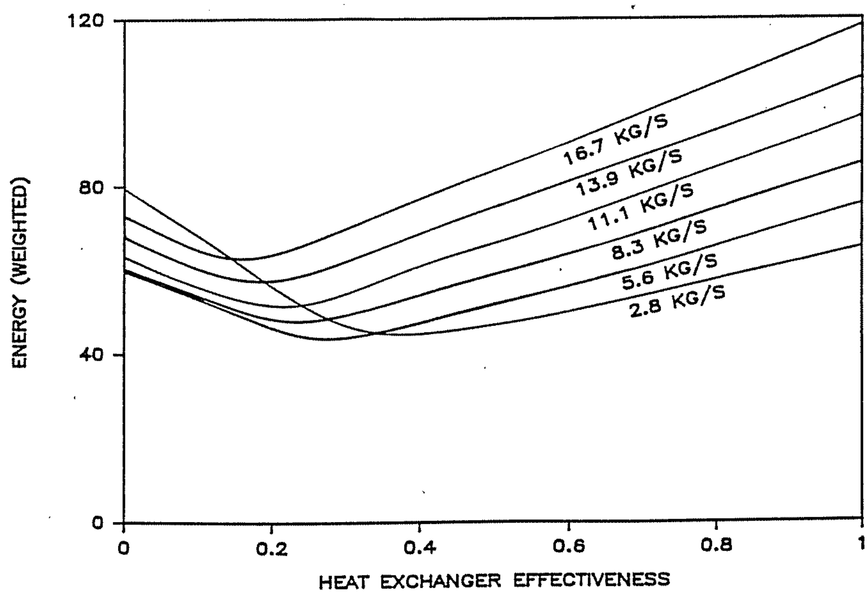

Figure 6. System energy use of the recirculation/condenser cycle at different circulation flow rates

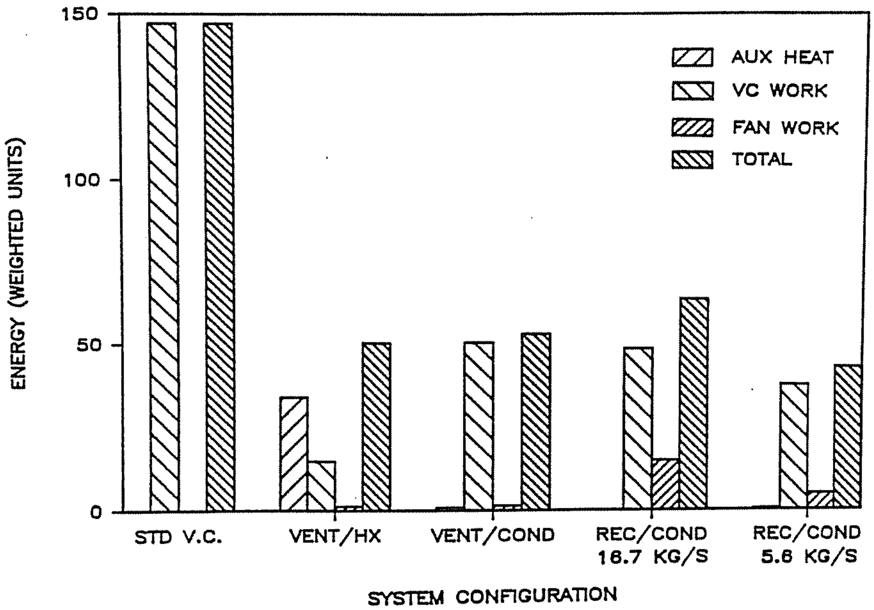

Figure 7. Energy breakdown of various systems at optimum heat exchanger effectiveness

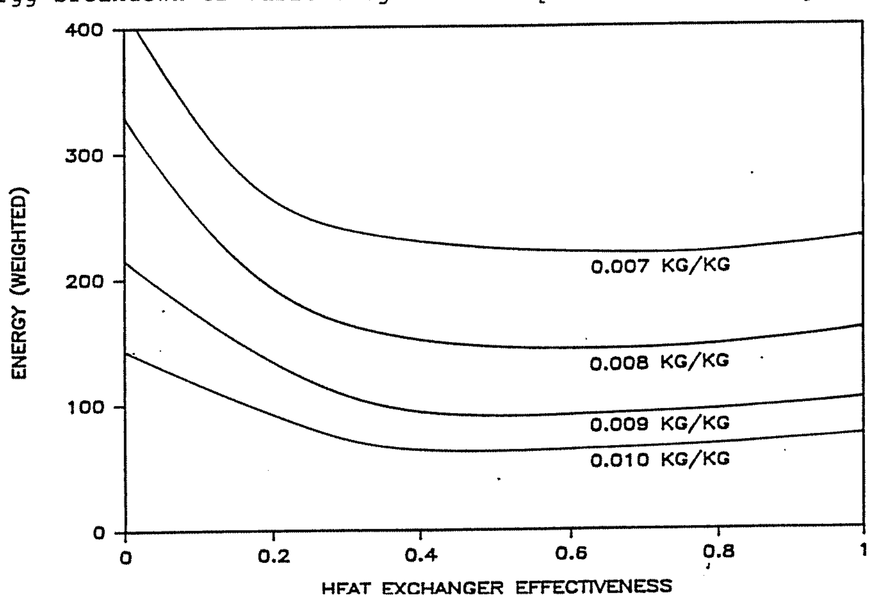

Figure 8. Total air-conditioning energy use of the ventilation/condenser cycle at reduced store humidity levels