NREL/TP-254-4601 • UC Category: 350 • DE92001180

Impact of Ambient Pressure on Performance of Desiccant Cooling Systems

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Prepared for the ASME International Solar Energy Conference, Maui, Hawaii, April 4-8, 1992

National Renewable Energy Laboratory
(formerly the Solar Energy Research Institute)
1617 Cole Boulevard
Golden, Colorado 80401-3393
A Division of Midwest Research Institute
Operated for the U.S. Department of Energy
under Contract No. DE-AC02-83CH10093

December 1991

On September 16, 1991, the Solar Energy Research Institute was designated a national laboratory, and its name was changed to the National Renewable Energy Laboratory.

Printed in the United States of America
Available from:
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

Price: Microfiche A01
Printed Copy A03

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IMPACT OF AMBIENT PRESSURE ON PERFORMANCE OF DESICCANT COOLING SYSTEMS

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ABSTRACT

The impact of ambient pressure on the performance of the ventilation cycle desiccant cooling system and its components was studied using computer simulations. The impact of ambient pressure depended on whether the system was designed for *fixed-mass flow rate* or *fixed-volume flow rate* operation. As ambient pressure decreased from 1.0 to 0.8 atm, the system thermal coefficient of performance increased by 8% for both fixed-mass and fixed-volume flow rate, the cooling capacity of the system (in kW) was decreased by 14% for the fixed-volume flow rate system and increased by 7% for the fixed-mass flow rate system, the electric power requirements for the system with fixed-volume flow rate did not change, and the electric power requirement for the fixed-mass flow rate system increased by 44%. The overall coefficient of performance increased up to 5% for the fixed-volume flow rate system, and decreased up to 4% for the fixed-mass flow rate system.

NOMENCLATURE

ARI	American Refrigeration Institute
a	heat transfer area per unit volume (m^2/m^3)
atm	atmosphere, unit of pressure
CC	cooling capacity (kW)
CFC	chlorofluorocarbons
COP	coefficient of performance
COP_o	overall coefficient of performance
COP_t	thermal coefficient of performance (nondimensional)
c_p	specific heat of humid air (J/Kg °C)
D	moisture diffusivity (m^2/s)
D_h	passage hydraulic diameter (m)
E_{blower}	electrical energy requirement for blowers (kW)
E_{ec}	evaporative cooler effectiveness (nondimensional)
E_{hx}	heat exchanger effectiveness (nondimensional)
f	friction factor (nondimensional)
f'	defined as f * Re (nondimensional)
h	gas-side heat transfer coefficient (W/m^2 °C)
K	gas-side mass transfer coefficient (kg/s m^2)
k	air thermal conductivity (W/m °C)
L	length of transfer area parallel to the flow (m)
m	mass flow rate of humid air (kg/s)
Ntu_h	number of heat transfer units (nondimensional)
Ntu_m	number of mass transfer units (nondimensional)
Nu	Nusselt number, heat transfer (nondimensional)
Nu_m	Nusselt number, mass transfer (nondimensional)
Re	Reynolds number (nondimensional)
RH	relative humidity (nondimensional)
P	ambient pressure (Pa)
PD	pressure drop (Pa)
p_{sat}	water vapor saturation pressure (Pa)
p_v	water vapor partial pressure (Pa)
T	dry bulb temperature (°C)
T_{wb}	wet bulb temperature (°C)
V	face or passage air velocity (m/s)
v	volume flow rate of air (m^3/s)
w	absolute humidity ratio (kg water/kg dry air)
w_s	saturation humidity ratio at T_{wb} (kg water/kg dry air)
μ	air viscosity (Ns/m^2)
ρ	air density (kg/m^3)

INTRODUCTION

Desiccant cooling systems regenerated with a thermal source are gaining acceptance for air conditioning of spaces. Currently, because of its economic advantage, natural gas is being used as the thermal source. Heat from solar energy, when delivered at lower costs, is an attractive alternative for desiccant regeneration, particularly because the cooling load and the solar heat load match in summertime. Waste heat can also be used to regenerate desiccants. Desiccant cooling systems do not use chlorofluorocarbon (CFC) refrigerants and are suitable for electric summer peak load reduction.

In a desiccant cooling system, humid air is first dried in a desiccant dehumidifier; the air is then cooled by a regenerative evaporative cooler/heat exchanger to desired conditions to be supplied to the conditioned space. The desiccant in the dehumidifier is regenerated (reactivated or dried) by hot air to be used in the next cycle. Two commonly used desiccant cooling cycles are ventilation and recirculation. In the ventilation cycle (Figure 1), outside air is processed through the dehumidifier, but in the recirculation cycle, the return air from the conditioned space is processed through the dehumidifier. The components of these two cycles are a desiccant dehumidifier, a regenerative heat exchanger, two evaporative coolers, a regeneration heater, two air fans, filters, and associated controls.

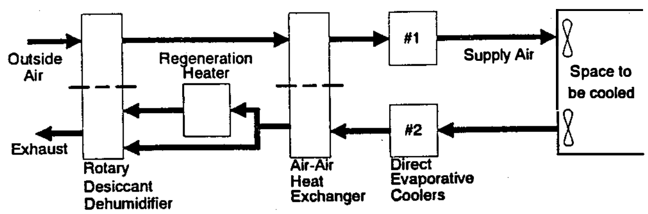

Figure 1. Schematic of Desiccant Cooling Ventilation Cycle

Over the last 15 years, the performance and reliability of the components of desiccant cooling systems have improved, and their costs have lowered. These goals have been achieved through improvements in materials, components, and system configurations. For example, the thermal coefficient of performance (COP) of the ventilation cycle has doubled from 0.5-0.6 to 1.0-1.2. The COP is the ratio of cooling load removed by the system to the thermal energy input to the system. New cycles have been proposed that can have a COP of over 2.0. Kini, Waugaman, and Ketteleborough (1990) described various desiccant cooling cycles and recent national research and development efforts.

There have been many system simulations of desiccant cooling cycles, but all of them were performed at 1 atm. However, desiccant systems may be installed at locations with higher elevations and thus lower pressures (see Table 1). Also, testing and field performance evaluation of components and systems are usually done under ambient pressures specific to the location of testing, which may be different than 1 atm. This pressure may be different than the actual pressure at the location where the system will be installed. Thus, the actual performance may deviate from the measured/predicted performance. Therefore, it is important to estimate how much the performance of a desiccant cooling system will change with a change in ambient pressure. *The purpose of this paper is to investigate the impact of ambient pressure on the performance of a ventilation cycle system and its components under various operating conditions and parameters.* Members of the ASHRAE Standards Committee (SPC 139 P) responsible for writing an industry standard testing method for desiccant dehumidifiers have expressed interest in the results of this study. Here, I will focus only on rotary solid desiccant systems.

PRESSURE EFFECT ON THERMOPHYSICAL PROPERTIES

Properties of Humid Air

The performance of rotary dehumidifiers, rotary heat exchangers, and evaporative coolers depends on geometry and size of air passages, heat and mass transfer characteristics in the gas and solid phases, thermophysical properties of the gas and solid phases, and rotational speed for the rotary devices. In the dehumidifier, the performance strongly depends on the type of desiccant used. All these parameters depend directly or indirectly on the thermophysical properties of the humid air, which may depend on ambient pressure.

For example, the performance of a dehumidifier depends on the number of heat transfer units, Ntu_h. Ntu_h is defined as

$$Ntu_h = h\,a\,L\,/\,m\,c_p\,, \qquad (1)$$

where

a = heat transfer area per unit volume (m^2/m^3),
L = length of heat transfer area parallel to the flow (m),
h = gas-side heat transfer coefficient (W/m^2 °C),
m = mass flow rate of humid air (kg/s),
c_p= specific heat of humid air (J/kg °C).

TABLE 1
Elevation and Ambient Pressure of Selected Cities

City	Elevation (ft)	Pressure (atm)
Albuquerque, NM	5311	0.80
Atlanta, GA	1010	0.96
Denver, CO	5283	0.81
Phoenix, AZ	1125	0.96
Chicago, IL	600	0.98
Las Vegas, NV	2178	0.92
Los Angeles, CA	270	0.99
Lubbock, TX	3254	0.88
Houston, TX	108	0.99
Greenville, TN	1319	0.95
Miami, FL	7	1.00
Orlando, FL	100	0.99
Salt Lake City, UT	4220	0.85
Tucson, AZ	2558	0.91
Washington, DC	14	1.00

The passage heat transfer coefficient, h, depends on the passage heat transfer Nusselt number, $Nu = h\,D_h\,/\,k$, where k is the thermal conductivity of humid air, and D_h is the passage hydraulic diameter. Substituting for h in Eq. 1, we can obtain

$$Ntu_h = Nu\,k\,a\,L\,/\,D_h\,m\,c_p\,. \qquad (2)$$

With fully developed laminar flow in the ducts, Nu is independent of the Reynolds number and is constant (Schultz, 1987; Edward, Denney, and Mills, 1977). Therefore, Ntu_h depends on thermal conductivity, k; specific heat, c_p; and mass flow rate of humid air. Mass flow rate will depend on density of air, ρ, for fixed air volume flow rate. As it can be seen, Ntu_h and, thus, the performance of a dehumidifier will depend on k, c_p, and ρ, all thermodynamic properties that in turn depend on ambient pressure.

In addition, the performance of the dehumidifier and evaporative cooler depends on inlet air relative humidity and wet bulb temperatures, respectively. These depend on ambient pressure for fixed absolute humidities. In the remainder of this section, I look at the impact of pressure on the thermophysical properties of humid air.

Density. Dry air at about atmospheric pressure can be considered an ideal gas. The humid ambient air for air conditioning applications contains, at most, 2% water vapor. Therefore, the air/water vapor mixture can also be considered an ideal gas for the conditions we are studying (Van Wylen and Sonntag, 1986). Obviously, the density of the humid air, assuming it is an ideal gas, is proportional to total pressure. For a fixed-mass flow rate system, the Ntu of the components does not change with pressure (see Eq. 2), so their thermal performance will not change. However, with laminar flow in passages and the fixed-mass flow rate, pressure drop across the components will change inversely with density and pressure: It can be shown that for fully developed laminar flow in ducts (assuming negligible entrance, exit, and acceleration effects), the pressure drop is

$$PD = V \mu f' L / 2 D_h^2 = m \mu f' L / 2 \rho D_h^2 A , \qquad (3)$$

where for laminar flows, the passage friction factor $f = f'/Re$, and $Re = \rho V D_h / \mu$.

Viscosity. According to the kinetic theory of ideal gases, the viscosity is independent of pressure (Edwards, Denney, and Mills, 1979). As the pressure increases, the number of molecular carriers (proportional to the density) increases; however, the number of paths that they can travel goes down. As a result, the viscosity remains unchanged because the resistance to the sliding motion of one layer of gas over another has not changed (Salsbersk, Acosta, and Hauptmann, 1971). This means that the Reynolds number and, thus, the friction factor of laminar flow passages do not change with ambient pressure.

Thermal Conductivity. The thermal conductivity of humid air is practically independent of pressure at near 1 atm (Reid, Prausnitz, and Sherwood, 1977). The thermal conductivity decreases by less than 0.3% by reducing pressure from 1 atm to 0.7 atm. Therefore, changes in thermal conductivity because of pressure changes will not have any measurable impact on components and system performance.

Specific Heat. The specific heat of humid air is also practically independent of pressure near 1 atm. The specific heat decreases by less than 0.05% when pressure changes from 1 to 0.7 atm (Bolz and Tuve, 1976). Therefore, the changes in specific heat because of pressure changes will not have any measurable impact on components and system performance.

Diffusivity. The density-diffusivity product, ρD, for a dilute ideal gas mixture is independent of pressure for reasons similar to why viscosity is independent of pressure (Edwards, Denney, and Mills, 1979). Therefore, the diffusivity of moisture in the air is inversely proportional to pressure. This may affect the mass transfer Ntu, Ntu_m, for the dehumidifier and evaporative coolers. However, Ntu_m depends on the mass transfer Nusselt number, Nu_m, which is inversely proportional to ρD ($Nu_m = K D_h / \rho D$). This product is independent of pressure; thus, the Ntu_m and moisture transfer performance of the dehumidifier and evaporative coolers are affected by changes in ρD because of pressure.

Relative Humidity. Relative humidity is defined as the ratio of partial pressure of water vapor, p_v, in a given moist air sample to the saturation pressure of water vapor, p_{sat}, at the same temperature, T. With the ideal gas law, it can be shown that for a fixed humidity ratio (w = ratio of *the mass of water vapor* to *the mass of dry air*), the relative humidity is proportional to total pressure, P:

$$RH = 0.622 \, P \, w / (1 - 0.622 \, w) / p_s , \qquad (4)$$

where 0.622 is the ratio of the molecular weight of water (18.01) to the molecular weight of air (28.96). Figure 2 is a plot of relative humidity versus humidity ratio for three pressures at 30°C. As pressure decreases from 1.0 to 0.8 atm, the relative humidity decreases by 20%. The relation between pressure, relative humidity, and absolute humidity ratio, w, is important for the moisture capacity of desiccants, as discussed in the next section.

Wet Bulb Temperature. The wet bulb temperature, T_{wb}, of air going through an evaporative cooler remains constant. The efficiency of evaporative coolers is expressed in terms of entering T_{wb}. It is important to know how T_{wb} changes with total pressure when other parameters are fixed. The relation between T_{wb} in °C, dry bulb temperature, T, in °C, humidity ratio, w, and total pressure, P, in atmosphere is (ASHRAE Handbook of Fundamentals, 1989):

$$w = \frac{(1074.9 - 1.02 \, T_{wb}) \, [0.622 \, p_{sat}(T_{wb})] \, / \, [P - p_{sat}(T_{wb})]}{(1057.2 - 0.799 \, T_{wb})} \qquad (5)$$

This equation is plotted in Figure 3 for two dry bulb temperatures to show the dependence of wet bulb temperature on pressure at various humidity ratios. For a given humidity ratio, as the pressure decreases (from 1.0 to 0.8 atm), the wet bulb temperature drops (by 2°C to 3°C or about 10% to 15%). For a given wet bulb temperature, as the pressure decreases (from 1.0 to 0.8 atm), the humidity ratio increases (by 30%). The performance of evaporative coolers is affected by change in wet bulb temperature because of changes in total pressure.

In summary, density, relative humidity, and wet bulb temperature are affected by change in pressure and may impact components and system performance. It should be noted that these and other properties are also functions of temperature. In this study, the components and system models incorporate temperature-dependent values for these properties.

Properties of Desiccants

Solid desiccants are materials that have a large internal surface area and can adsorb water vapor. The driving potential for adsorption is the difference between the vapor pressure of water vapor in the humid air and the vapor pressure of water in equilibrium at the internal surfaces of the desiccant (Ruthven, 1984). No moisture is adsorbed when the vapor pressure in the desiccant reaches the vapor pressure of the water vapor in the air. At this point, equilibrium has been reached between the desiccant and the humid air, assuming the same temperature for the desiccant and air. The amount of water adsorbed at equilibrium, W, depends on the type of desiccant; the system temperature, T; and the partial pressure of the water vapor, p_v. Many investigators have observed that the adsorption capacity (kg water/kg dry desiccant), when expressed in terms of relative humidity $[p_v/p_{sat}(T)]$, is a weak function of temperature (see Rojas, 1980, for a literature review).

The moisture capacities of two desiccants as a function of relative humidity (i.e., sorption isotherm) at 30°C are shown in Figure 4. Silica gel, a commercially available desiccant, is commonly used in dehumidifiers and can be regenerated with temperatures available from flat-plate solar collectors. The isotherm shape of the second desiccant is generally known as Type 1 moderate (1M) isotherm, which has been shown to be the desired isotherm shape for a desiccant for cooling applications (Collier, 1988). This isotherm shape provides higher thermal performance than other shapes. Desiccants with Type 1M isotherm shapes are not currently available commercially. Several organizations under funding from the Gas Research Institute and the U.S. Department of Energy are developing such desiccants.

The moisture capacity of a desiccant at a fixed relative humidity does not depend on pressure (Figure 4). However, the moisture capacity depends on pressure at a fixed humidity ratio (Figure 5). As ambient pressure decreases from 1.0 to 0.8 atm, the relative humidity decreases (Figure 2), and the moisture capacity decreases between 10% and 15%. This can adversely impact the performance of a desiccant dehumidifier.

It should be noted that at the pressure changes considered, the impact on moisture diffusivity of water into the desiccant is expected to be negligible; thus, the solid-side resistance to moisture diffusion will not be affected.

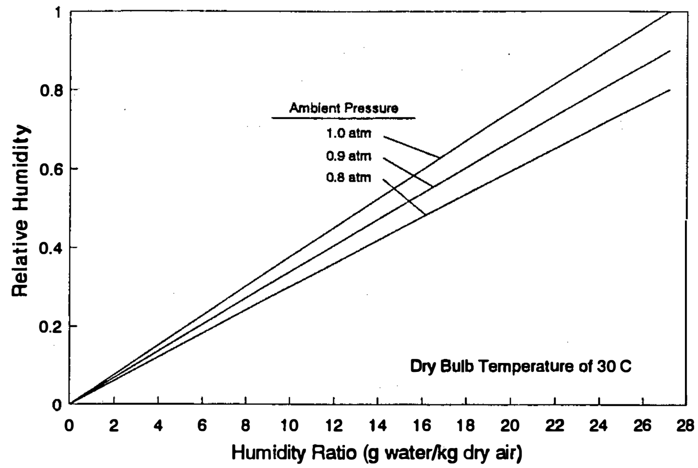

Figure 2. Impact of Ambient Pressure on Relation between Relative Humidity and Humidity Ratio (at 30°C)

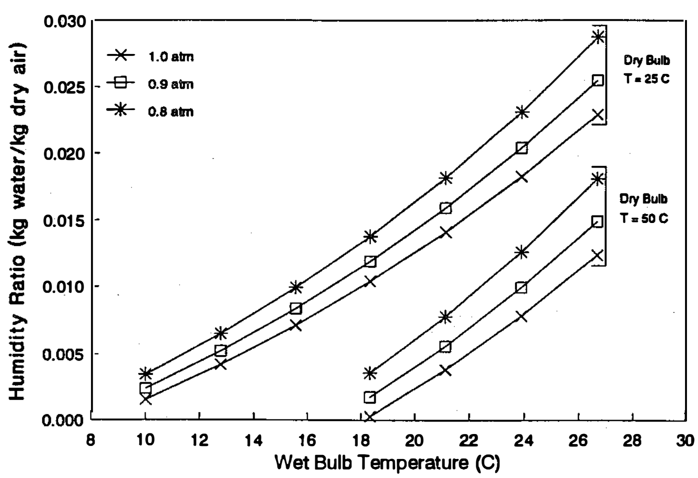

Figure 3. Impact of Ambient Pressure on Relation between Humidity Ratio and Wet Bulb Temperatures (at 30°C and 50°C)

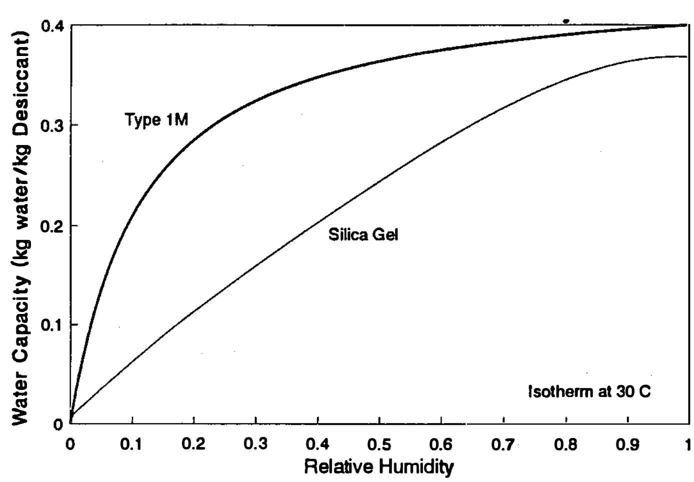

Figure 4. Moisture Capacity Isotherms for Silica Gel and Type 1M Desiccants (at 30°C)

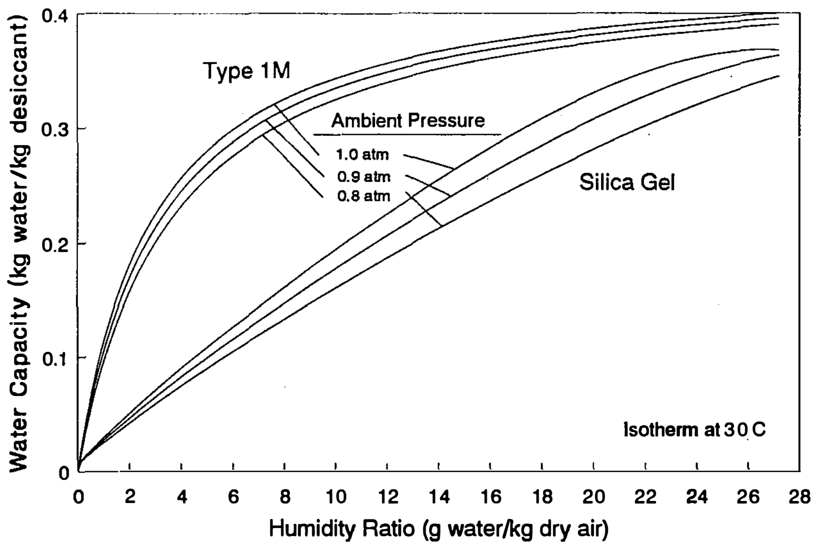

Figure 5. Impact of Ambient Pressure on Moisture Capacity of Silica Gel and Type 1M Desiccants (at 30°C)

COMPONENT MODELING AND RESULTS

The major components whose performance could be affected by the ambient pressure are the heat exchanger, evaporative coolers, the dehumidifier, the heater, and fans. Two parameters that are used here for evaluating performance of desiccant cooling systems are

- Cooling capacity (CC), defined as the *amount of cooling energy delivered to the space* (in terms of kW or tons), and
- Thermal COP, defined as the *amount of cooling energy delivered to the space* divided by the *thermal energy input for regeneration* of a desiccant dehumidifier.

Heat Exchanger

Rotary heat exchanger design has usually been used in desiccant cooling systems, although fixed counter-flow or cross-flow designs can be used. Because of pressure drop limitations, core geometries with laminar flow passages, such as corrugated (sinusoidal) passages, have been used for the heat exchanger. The effectiveness model is used to predict the performance of the heat exchanger. The heat exchanger effectiveness is defined as

$$E_{hx} = (m\ c_p)_{hot}\ (T_{hot,in} - T_{hot,out})/(m\ c_p)_{min}\ (T_{hot,in} - T_{cold,in})\ , \quad (6)$$

where $(m\ c_p)_{min} = \min\ [(m\ c_p)_{hot}\ , (m\ c_p)_{cold}\ ]$.

The effectiveness depends on the number of heat transfer units (Ntu_h) and the ratio of flow heat capacity of each stream ($m\ c_p$). For rotary heat exchangers, the effectiveness also depends on the ratio of the *heat capacity of the matrix * rotational speed* divided by *minimum flow heat capacity of the two streams.* For laminar flow cases, it can be shown that Ntu_h for the heat exchanger is

$$Ntu_h = Nu\ k\ a\ L\ /\ D_h\ m\ c_p\ . \quad (7)$$

Nu, a, L, and D_h are independent of ambient pressure; k and c_p are practically independent of ambient pressure. Therefore, for fixed-mass flow rate systems, Ntu_h is independent of pressure, but for fixed-volume flow rate systems ($m = \rho\ v$), Ntu_h is inversely proportional to ambient pressure. Thus, for a fixed-mass flow rate system, heat exchanger performance does not change with ambient pressure. However, for a fixed-volume flow rate system, the Ntu_h increases (up to 20%) as ambient pressure decreases (from 1.0 to 0.8 atm). Examination of E_{hx} - Ntu_h tables (e.g., Kays and London, 1964) for periodic heat exchangers reveals that a 20% increase in Ntu_h may result in a 2% to 4% increase in E_{hx} for the effectiveness range of 0.8 to 0.93; i.e., the heat exchanger performs better. This can result in a decrease (or *increase*) in air temperature between 0.5°C and 1.5°C, leaving the heat exchanger on the process (or *regeneration*) side. This can increase the cooling capacity and COP of the system slightly.

For the laminar flow heat exchanger, it can be shown (Eq. 3) that the pressure drop for fixed-mass flow rate increases (by 20%) with a decrease in ambient pressure (from 1.0 to 0.8 atm). However, for fixed-volume flow rate, the pressure drop will not change with ambient pressure.

Evaporative Cooler

Most common desiccant cycles use direct evaporative cooling. In the direct evaporative cooler, air undergoes an adiabatic saturation process. Therefore the air wet bulb temperature remains approximately constant through the device (Van Wylen and Sonntag, 1986). The effectiveness of an evaporative cooler, E_{ec}, defined as the extent to which the outlet dry bulb temperature reaches the inlet wet bulb temperature:

$$E_{ec} = (T_{db,in} - T_{db,out})\ /\ (T_{db,in} - T_{wb,in}) \quad (8a)$$

and

$$E_{ec} = (w_{in} - w_{out})\ /\ (w_{in} - w_{s,in})\ , \quad (8b)$$

where T_{db} is the dry bulb temperature, T_{wb} is the wet bulb temperature, w is the air humidity ratio, and w_s is the saturation hunidity ratio at T_{wb}.

For predicting the performance of evaporative coolers in the system, I used the effectiveness model. With commercially available structured packing, the effectiveness is a weak function of air face velocity (Munters, 1988), particularly at depths greater than 8 in. in the direction of air flow. For fixed-mass flow rate, as the pressure decreases (from 1.0 to 0.8 atm) the volumetric flow rate and face velocity increase (by 20%), and the effectiveness decreases only by 1% (Munters, 1988). For fixed-volume flow rates, the face velocity, thus the effectiveness, does not change with ambient pressure.

Although the effect of ambient pressure on evaporative cooler effectiveness is small or negligible, the pressure may affect the evaporative cooler performance because it affects the relation between humidity ratio, dry bulb temperature, and wet bulb temperature, as shown in Figure 3. The impact of ambient pressure on the outlet humidity and temperature from an evaporative cooler with E_{ec} of 0.93 is shown in Figure 6 for various inlet air humidities and dry bulb temperatures. It can be seen that as the ambient pressure decreases from 1.0 to 0.8 atm, the outlet air humidity increases by 5%, but the outlet temperature decreases by 10%. In other words, the evaporative cooler performs better. The reason is that as the ambient pressure decreases, the water evaporates easier to cool the air. As the evaporative coolers perform better, it is expected that the cooling capacity of the desiccant system will increase. However, the COP may increase because of an increase in cooling capacity or decrease because more regeneration heat is needed to heat the cooler air in the regeneration air stream.

The pressure drop across the evaporative cooler is proportional on face velocity (Munters, 1988) for laminar flow geometries (Eq. 3). For constant volumetric flow rate, the face velocity does not change with a change in ambient pressure, thus no change in pressure drop. For fixed-mass flow rate, the pressure drop is inversely proportional to ambient pressure and increases by 20% as pressure decreases from 1.0 to 0.8 atm.

Dehumidifier

The model used to simulate the performance of the dehumidifier was developed by Collier (1989); it is principally based on the combination of the pseudo-steady-state model of Barlow (1982) and the finite-difference algorithm of Maclaine-Cross (1974). The model solves the governing continuity, species, and energy equations for the air and desiccant. The model assumes gas-side controlled heat and mass transfer and uses an overall mass transfer coefficient that combines both solid- and gas-side resistances.

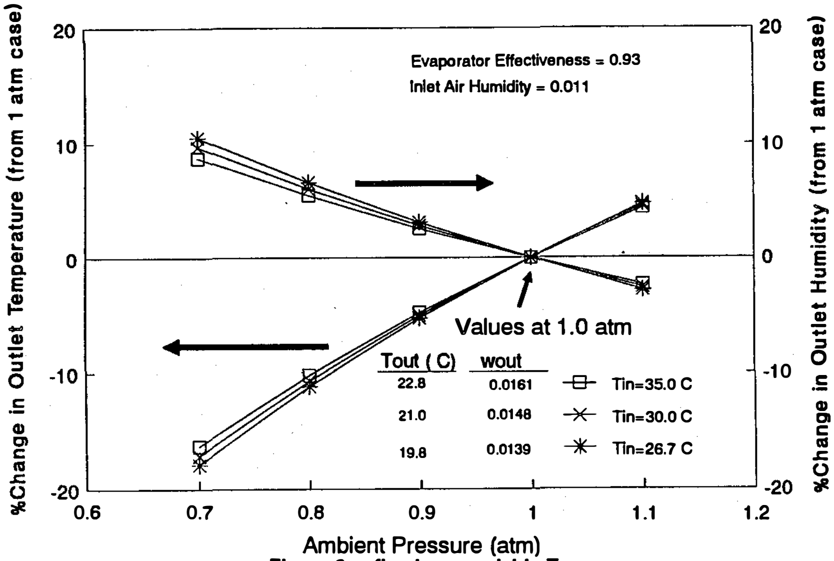

Figure 6.a: fixed w_{in}, variable T_{in}

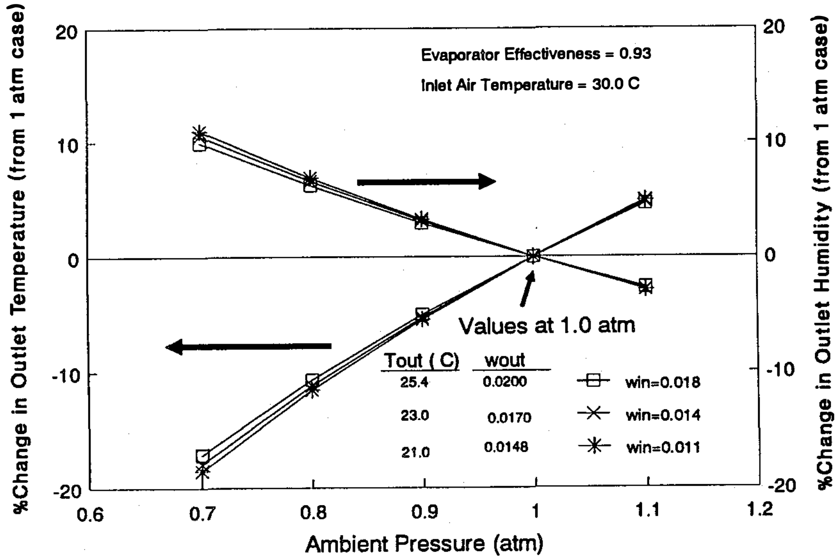

Figure 6.b: fixed T_{in}, variable w_{in}

Figure 6. Impact of Ambient Pressure on Performance of Evaporative Cooler with an Effectiveness of 93%

Table 2 summarizes the characteristics and conditions of the baseline rotary dehumidifier that was modeled. The physical dimensions of the modeled dehumidifier are based on those of a dehumidifier tested by Bharathan et al. (1987). Figure 7 shows the outlet air conditions from the process side of the silica gel dehumidifier as a function of dehumidifier rotational speed for three ambient pressures. As can be seen from Figure 7a, as the ambient pressure decreases, the process outlet air humidity increases, more for fixed-mass flow rate than fixed-volume flow rate. This is consistent with the previous discussion that as ambient pressure decreases, the moisture adsorption capacity of desiccants decreases; therefore, less moisture is removed from the process air, leading to higher outlet air humidity. For fixed-volume flow rate, the effect on outlet humidity is smaller because as the pressure decreases, a smaller air mass and, thus, less moisture flow through the dehumidifier.

For fixed-mass flow rate, the process outlet air temperature does not change a lot with a change in pressure, as seen from Figure 7b. However, for fixed-volume flow rate, the outlet temperature increases by 1° to 2°C when pressure changes from 1.0 to 0.8 atm. In removing the heat released by the adsorption process, for the fixed-volume flow rate system, the outlet temperature will rise more as less mass flows through the desiccant at lower pressure.

In summary, Figure 7 and other similar simulations for silica gel and Type IM isotherm dehumidifiers with and without staged regeneration indicate that as pressure decreases, dehumidifier performance degrades, and process air is dehumidified less for both fixed-volume and fixed-mass flow rate systems.

Similar to the heat exchanger under laminar flow conditions, pressure drop across the dehumidifier increases (by 20%) with a decrease in ambient pressure (from 1.0 to 0.8 atm) for fixed-mass flow rate. However, for fixed-volume flow rate, the dehumidifier pressure drop does not change with ambient pressure.

Regeneration Heater

The device that supplies the regeneration heat can be a solar collector, natural gas boiler or furnace, a waste heat recovery device, or even an electric heater. The dependence of performance on ambient pressure depends on the design of the regeneration device. For example, for a gas boiler, reduction in ambient pressure will reduce the amount of natural gas delivered for combustion if the boiler is designed to provide a fixed gas volume flow rate. As a result, less gas is burned and less heat is delivered for regeneration, resulting in lower dehumidifier performance. If the device is designed for fixed-mass flow rate, then the boiler performance is not affected. Considering the impact of pressure on different regeneration heaters was beyond the scope of the study. For system simulations, I assumed that the heater performance is not affected by ambient pressure, and the same amount of regeneration heat is delivered per mass of air passed through.

Air Blowers

The air blowers or fans are usually rated for delivery of fixed-volume flow rates for a given static pressure drop. If a blower is moved to lower ambient pressures, it still delivers the same amount of volumetric air flow rate but lower mass flow rate. To maintain the same mass flow rate at lower pressures, the volumetric flow rate should be increased, e.g., by increasing speed of the motor, which results in higher electric power consumption.

SYSTEM MODELING AND RESULTS

Table 3 summarizes the qualitative impact of ambient pressure on the performance of various components of the desiccant cooling system. As ambient pressure decreases, the performance of the heat exchanger and evaporative coolers may increase or remain unchanged, but the performance of the dehumidifier decreases. To obtain the quantitative impact of ambient pressure on ventilation cycle system performance, I used a desiccant cooling system simulation code, DCSSMX1, developed by Collier (1989). This code is based on the models discussed previously for heat exchangers, evaporative coolers, and dehumidifiers. For this

TABLE 2
Baseline Parameters for Modeling Dehumidifier

Parameter	Value
Matrix density	157 kg desiccant/m^3
Matrix heat capacity	1960 kJ/kg.K
Total frontal area	0.49 m^2
Nominal diameter	1.2 m
Matrix depth	0.2 m
Passage hydraulic diameter	2.3 mm
Total transfer area	95 m^2
Adsorption or regeneration air flow rate	0.2 kg/s, 0.174 m^3/s
Areas for adsorption or regeneration	Equal
Number of heat transfer units	28.2
Process Lewis number	1
Desiccant material	Silica gel or Type 1M with separation factor of 0.1
Inlet regeneration conditions	95°C and 0.014 kg water/kg air
Inlet process conditions	35°C and 0.014 kg water/kg air

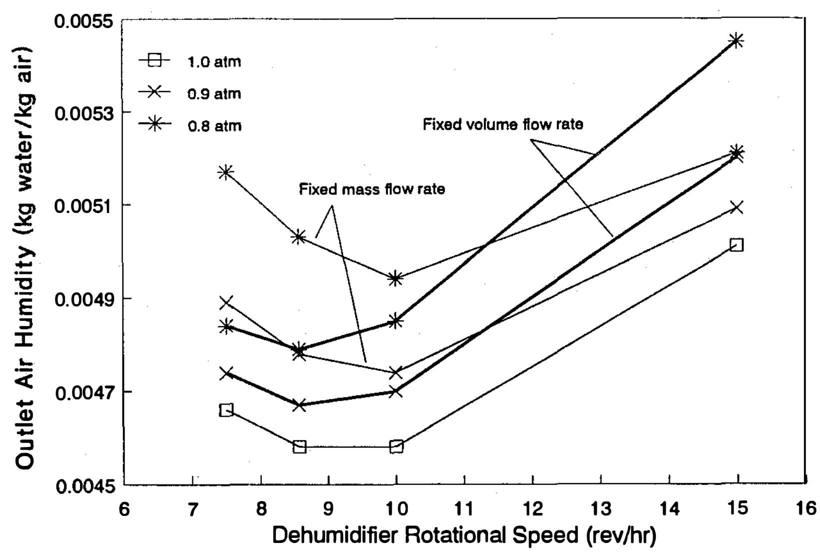

Figure 7.a: outlet air humidity

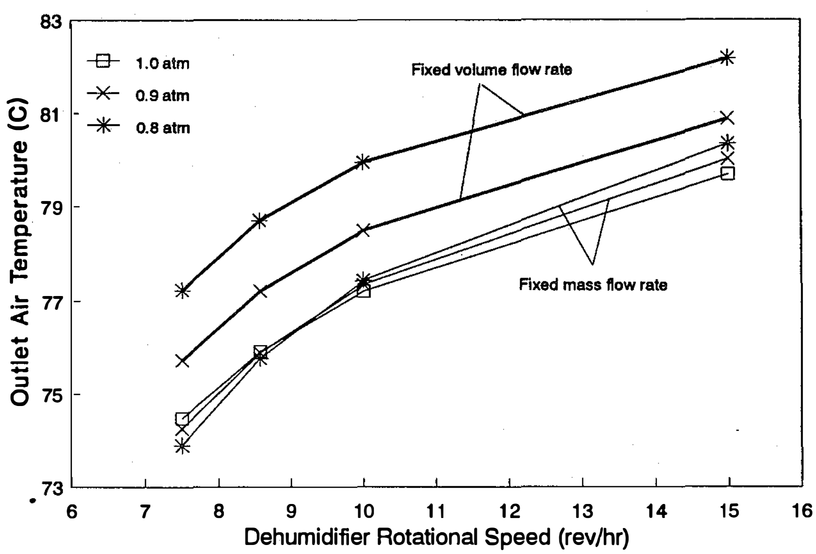

Figure 7.b: outlet air temperature

Figure 7. Impact of Ambient Pressure on Performance of Dehumidifier with Silica Gel without Staged Regeneration

TABLE 3
Impact of Reduction of Ambient Pressure on Various Components

Component	Thermal/Moisture Performance		Pressure Drop	
	Fixed-Volume Flow Rate	Fixed-Mass Flow Rate	Fixed-Volume Flow Rate	Fixed-Mass Flow Rate
Heat Exchanger	Increase	No change	No change	Increase
Evaporative Cooler	Increase	Increase	No change	Increase
Dehumidifier	Decrease	Decrease	No change	Increase
Blower	-------	-------	No change	Increase

study, I simulated a ventilation cycle desiccant cooling system with the following design parameters:

Dehumidifier:	As discussed in Table 2
Heat Exchanger:	Effectiveness of 0.93
Evaporative Cooler:	Effectiveness of 0.95
Outdoor Conditions:	ARI rating point (35.0°C, 0.014 kg moisture/kg air)
Indoor Conditions:	ARI rating point (26.7°C, 0.011 kg moisture/kg air)
Air Flow Rate:	0.2 kg/s at 1 atm
Nominal Capacity:	3.52 kW or 1 ton

Two types of desiccants were used for the simulations: microporous silica gel and a desiccant with Type 1M isotherm. Simulations were conducted with and without staged regeneration for the dehumidifier. Staged regeneration has been shown to be effective in improving the performance of the cooling system (Collier, 1989; Collier et al., 1990). In staged regeneration, the regeneration process consists of two stages. In the first stage, the air exiting from the warm side of the sensible heat exchanger is used for regeneration of the desiccant without adding external heat. In the second stage, the remainder of the air exiting the heat exchanger is used with additional external heat to regenerate the desiccant. In this study, the fraction the regeneration air that is heated in the regeneration heater was 0.5, i.e., only half of the regeneration air was heated.

The performance of a desiccant system depends on the rotational speed of the dehumidifier. The maximum cooling capacity and the COP do not usually occur at the same rotational speed. Figure 8 shows the changes in the cooling capacity or thermal COP as a function of dehumidifier rotational speed for a system using Type 1M desiccant *without staged regeneration* at three ambient pressures. As one can observe from Figure 8, for the pressure of 1 atm, the maximum thermal COP and cooling capacity occur at about 3 rev/hr. Another observation is that the thermal COP increases for both fixed-mass and fixed-volume flow rates with a decrease in ambient pressure. The increase is about 8% for 0.8-atm pressure. Although the dehumidifier performance decreases with a decrease in pressure, the overall system performance improves because of increases in the performance of the heat exchanger and evaporative coolers (Table 3). The cooling capacity increases (*by 9%*) as ambient pressure decreases (*to 0.8 atm*) for fixed-mass flow rate, but it decreases (*by 14%*) for a fixed-volume flow rate system. The major reason why the cooling capacity removed from the space decreases with a decrease in pressure for the fixed-volume flow rate system is that the mass flow rate of cool air supplied to the space has decreased; therefore, it has less capacity to cool the space. The above discussion about dependence of COP or cooling capacity on ambient pressure could also be applied to a system using Type 1M desiccant with staged regeneration, as the results of the simulations in Table 4 show.

We also simulated a ventilation cycle using silica gel dehumidifier *with* and *without* staged regeneration. Figure 9 shows the results of

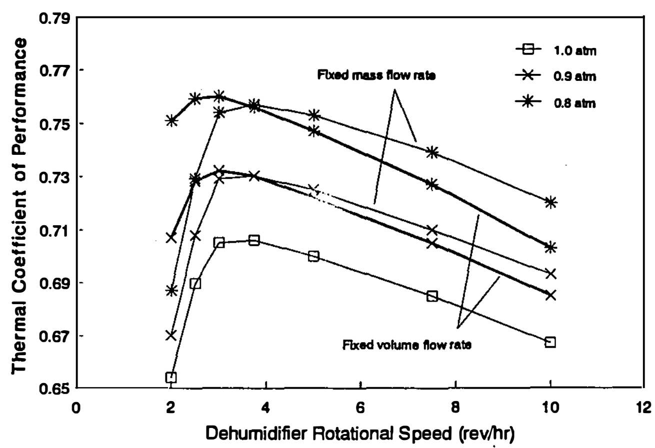

Figure 8.a: thermal COP

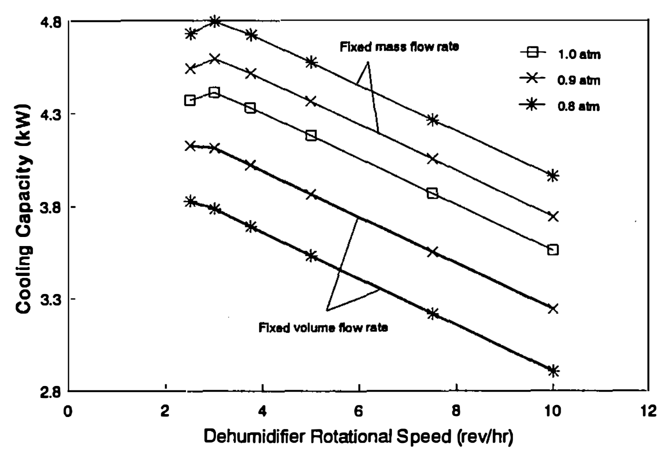

Figure 8.b: cooling capacity

Figure 8. Impact of Ambient Pressure on Performance of Desiccant Cooling System Using Type 1M Desiccant without Staged Regeneration

TABLE 4
Impact of Pressure on Performance of Cooling System
Using Type 1M Desiccant

Ambient Pressure, Flow rate	With Staged Regeneration Dehumidifier Speed = 2 rev/hr			Without Staged Regeneration Dehumidifier Speed = 3 rev/hr		
	COP_t	Capacity (kW)	COP_o	COP_t	Capacity (kW)	COP_o
0.8 atm, Fixed-Mass	1.12 (+7%)	4.01 (+8%)	0.71 (-4%)	0.76 (+8%)	4.79 (+9%)	0.57 (0%)
0.9 atm, Fixed-Mass	1.07 (+3%)	3.88 (+4%)	0.72 (-2%)	0.73 (+4%)	4.69 (+6%)	0.57 (0%)
1.0 atm, 0.2 kg/s, 0.174 m^3/s	1.04 (0%)	3.72 (0%)	0.74 (0%)	0.70 (0%)	4.41 (0%)	0.57 (0%)
0.9 atm, Fixed Volume	1.09 (+5%)	3.49 (-6%)	0.75 (+1%)	0.73 (+4%)	4.11 (-7%)	0.58 (+2%)
0.8 atm, Fixed Volume	1.13 (+8%)	3.23 (-13%)	0.75 (+1%)	0.77 (+10%)	3.78 (-14%)	0.60 (+5%)

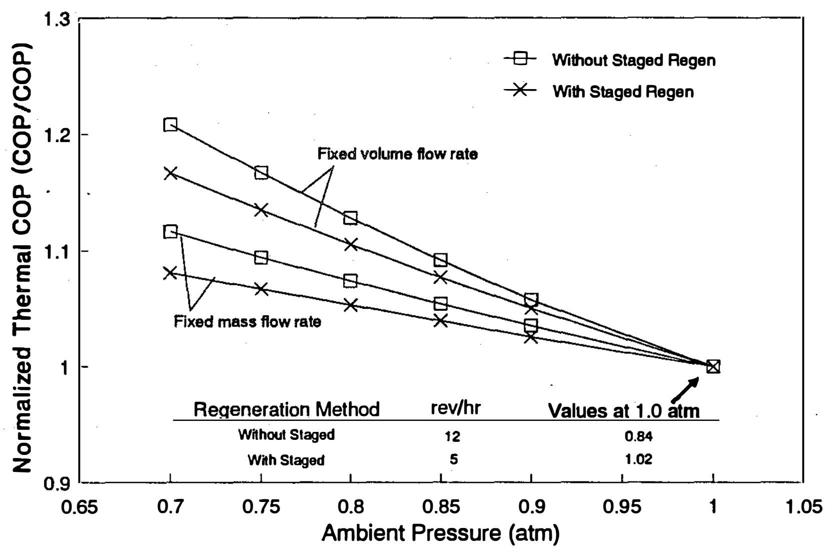

Figure 9.a: normalized thermal COP

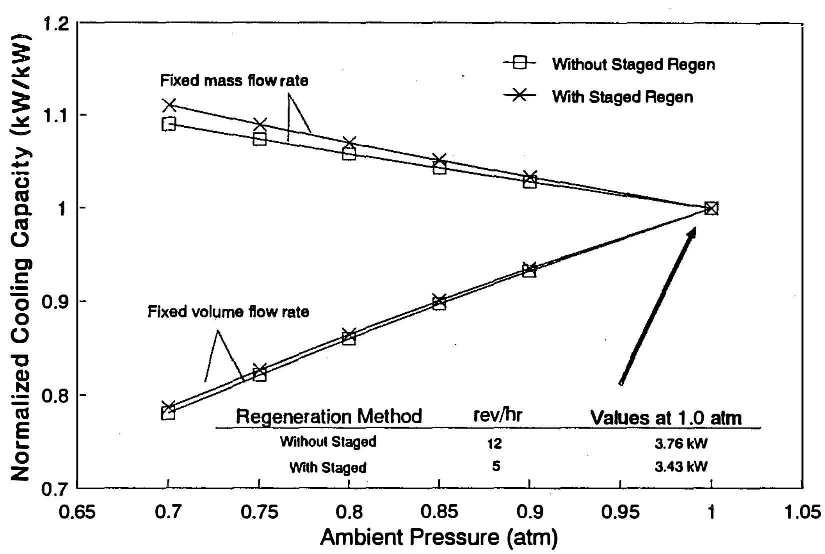

Figure 9.b: normalized cooling capacity

Figure 9. Impact of Ambient Pressure on Performance of Desiccant Cooling System using Silica Gel

simulations for COP and cooling capacity at optimum dehumidifier rotational speed. Similar to Type 1M isotherm desiccant, the COP of the system with silica gel increases (up to 11%) as the ambient pressure decreases (from 1.0 atm to 0.8 atm). The increase is more for fixed-volume flow rate than fixed mass flow rate. The increase in COP is also more for without staged regeneration than with staged regeneration. When ambient pressure is decreased from 1.0 atm to 0.8 atm, the cooling capacity increases by 7% for a fixed-mass flow rate system and decreases by 13% for a fixed-volume flow rate system. Another observation from the results is that tuning the rotational speed of the dehumidifier may improve the COP or cooling capacity with change in ambient pressure from its value at 1 atm, however, this increase in less than 1%.

For a fixed-volume flow rate system, the pressure drop across the components (and the system) does not change with changes in ambient pressure (see Table 3). Therefore, the parasitic power requirement for blowers does not change for a fixed-volume flow rate system. However, for a fixed-mass flow rate system, the pressure drop across all the components and thus the system increases as the ambient pressure decreases (see Table 3). Note that the parasitic power loss for blowers is proportional to the product of system pressure drop and volumetric flow rate (PD * v). Because both of these are inversely proportional to ambient pressure for the fixed-mass flow rate system, parasitic power loss is inversely proportional to square of ambient pressure. Therefore, as ambient pressure decreases from 1.0 atm to 0.8 atm, the electric power requirement for flowing air increases by 44% in the fixed-mass flow rate system. The electrical energy requirement for the fans is between 10% to 20% of the cooling capacity. Therefore, the increase in energy consumption of fans caused by a decrease in pressure would be between 4.4% to 8.8% of the cooling capacity. This may offset the increase in cooling capacity resulting from a decrease in pressure.

Here, the overall coefficient of performance, COP_o, is used to combine the effect of increases in both cooling capacity and electrical energy consumption. COP_o is defined as

$$COP_o = \frac{}{\text{regeneration heat input} + \eta(\text{electrical energy input})} \quad (9)$$

where η is the conversion factor from electricity to prime energy (fossil fuel) and is taken to be 3 here. Assuming negligible electrical energy consumption for pumps and controls, the COP_o can be calculated from

$$COP_o = CC/(CC/COP_t + \eta\, E_{blower}) \quad (10)$$

Assuming a pressure drop of 0.63 kPa (2.5 in. water) for the process airstream, 0.75 kPa (3.0 in. water) for the regeneration air stream, and a blower motor efficiency of 50%, the electrical energy requirement for the blowers would be 0.48 kW. Then, COP_o for the ventilation system using Type 1M desiccant would be 0.74 with staged regeneration and 0.57 without staged regeneration at an ambient pressure of 1.0 atm. Table 4 shows that for the fixed-mass flow rate, the overall COP decreases by 4% with staged regeneration and does not change without stage regeneration. Table 4 also shows that for the fixed-volume flow rate, the overall COP increases by 1% with staged regeneration and by 5% without staged regeneration.

CONCLUSIONS

The variation of thermophysical properties with ambient pressure as related to the performance of heat exchangers, evaporative coolers, and dehumidifiers was considered. Only air density, relative humidity, air wet bulb temperature, and desiccant moisture capacity were affected by changes in ambient pressure. The impact of ambient pressure on heat exchangers, evaporative coolers, and dehumidifiers was investigated under laminar flow conditions. Finally, the impact of ambient pressure on a ventilation cycle desiccant cooling system for silica gel and Type 1M isotherm desiccants with and without staged regeneration was determined.

As ambient pressure decreased from 1.0 to 0.8 atm, for the *fixed-mass flow rate*:

- The thermal performance of the heat exchanger did not change.
- The thermal performance of the evaporative cooler improved for fixed inlet humidity.
- The dehumidification ability of the dehumidifier decreased for fixed inlet air humidity.
- For both desiccants and both types of regeneration, the thermal COP and cooling capacity increased between 6% to 8%.
- The pressure drop across each component and the system increased (inversely proportional to pressure), by 20% and the power requirement for the blowers increased (inversely to square of pressure) by 44%. As a result, the overall COP decreased up to 4%.

As ambient pressure decreased from 1.0 to 0.8 atm, for the *fixed-volume flow rate*:

- The heat exchanger effectiveness improved by 2% to 4%.
- The thermal performance of the evaporative cooler improved for fixed inlet humidity.
- The dehumidification ability of the dehumidifier decreased for fixed inlet humidity.
- For both desiccants and both types of regeneration, the thermal COP increased by 8% and cooling capacity decreased by 14%.
- The pressure drop across each component and thus the system did not change. The overall COP increased up to 5%.

The results of this study can be applied when a desiccant cooling system is designed or tested for one elevation and is moved to another.

ACKNOWLEDGMENTS

This work was supported by the U.S. Department of Energy, Office of Buildings Energy Technology; Solar Cooling Program; Robert Hassett, Program Manager. The author wishes to thank Rick Heiden for performing some of the simulation runs.

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