

Substitution of  $M=M_1=0.0355907$  and the rest of the values used previously in the example under consideration, gives  $\rho_{ave}=0.915817 \text{ kg/m}^3$  and  $\Delta p_h=13462.5 \text{ Pa}$ . The exit pressure is then:  $p_2=p_1-\Delta p_h-\Delta p_a=89920.2-13462.5-19.6=76438.1 \text{ Pa}$ . The difference between it and the accurate value of  $p_2=76438.2 \text{ Pa}$  from Eq. (5) is  $-0.1 \text{ Pa}$ . Substitution of  $M=M_{ave}=0.0382492$  changes the sign of the difference but does not reduce its magnitude.

**No-Flow Average Density.** When  $M=0$  and  $T=t$ , Eq. (17) simplifies to:

$$\frac{\rho_{ave,M=0}}{\rho_1} = \frac{1}{\left(\frac{\gamma}{\gamma-1}\right)} \frac{c_p t_1}{g \delta z} \left(1 - \left(\frac{t}{t_1}\right)^{(\gamma/\gamma-1)}\right)$$

$$\rho_{ave,M=0} = \frac{\rho_1 R t_1}{g \delta z} \left(1 - \left(\frac{p}{p_1}\right)\right) = \frac{p_1 - p}{g \delta z} \quad (18)$$

This is in fact the well-known hydrostatic equation:  $\Delta p = -\rho_{ave} g \Delta z$ . Equation (17) gives the equivalent equation when a compressible fluid accelerates vertically at low Mach numbers.

## Conclusion

This note develops a method for finding all the thermodynamic properties for compressible frictional flow through tall vertical chimneys. The method finds the stagnation temperature distribution directly from the altitude, by applying the energy equation. It then finds the Mach number distribution from the vertical distributions of stagnation temperature, friction factor, bracing drag loss coefficient, and flow area. The static pressure at each altitude follows from the continuity equation. It turns out that the generally used adiabatic temperature lapse rate equation in principle applies to the stagnation temperature, and applying it to the static temperature is valid only when the Mach number is equal to zero. The note presents two equations for the vertical pressure and density distributions in terms of Mach number. One of these is a generalization of the adiabatic pressure lapse ratio equation to include flow at small Mach numbers. The other is analogous to the hydrostatic relationship between pressure and density, extended to small Mach numbers. Its integration leads to an equation for the average density in the chimney. A very accurate value of the average density is exactly what the commonly used incompressible flow approach to the problem requires to calculate the hydrostatic pressure drop in the chimney.

## Nomenclature

$A$  = flow area [ $\text{m}^2$ ]  
 $c_p$  = specific heat [ $\text{J/kgK}$ ]  
 $D$  = chimney inside diameter [ $\text{m}$ ]  
 $F$  = force [ $\text{N}$ ]  
 $f$  = friction coefficient -  
 $g$  = gravitational acceleration [ $\text{m/s}^2$ ]  
 $K$  = pressure drop coefficient -  
 $L$  = sum of loss terms -  
 $M$  = Mach number -  
 $\dot{m}$  = mass flow [ $\text{kg/s}$ ]  
 $P$  = stagnation pressure [ $\text{Pa}$ ]  
 $p$  = static pressure [ $\text{Pa}$ ]  
 $R$  = gas constant [ $\text{J/kgK}$ ]  
 $T$  = stagnation temperature [ $\text{K}$ ]  
 $t$  = static temperature [ $\text{K}$ ]  
 $V$  = velocity [ $\text{m/s}$ ]  
 $z$  = height [ $\text{m}$ ]

## Greek

$\gamma$  = specific heat ratio -  
 $\mu$  = dynamic viscosity [ $\text{Ns/m}^2$ ]  
 $\rho$  = density [ $\text{kg/m}^3$ ]

## Prefix

$\Delta$  = change over chimney height  
 $\delta$  = change over interval

## Subscript

$a$  = acceleration  
 $ave$  = average  
 $D$  = drag  
 $f$  = friction  
 $h$  = hydrostatic  
 $i$  = integration interval number  
 $1$  = inlet (or coefficient number)  
 $2$  = exit (or coefficient number)

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# A Study of the Performance of a Hybrid Liquid Desiccant Cooling System Using Lithium Chloride

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*This paper presents field test of a hybrid solar liquid desiccant cooling system conducted at a test house at the University of Florida's Energy Research and Education Park. These tests consisted of operating the air conditioning system at the test house in two configurations: the conventional vapor compression system and the hybrid desiccant system. Experiments were conducted to study the influence of the air mass flow rate, temperature of the inlet air, temperature of the desiccant, and desiccant mass flow rate on the performance of both system configurations. Based on the field test results, it was found that the hybrid desiccant system improves the air conditioning performance in the field test house by decreasing the outlet humidity and temperature of the air.*  
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## Introduction

In hot and humid regions where the relative humidity is high, the latent load becomes a big problem. Properly sized conventional vapor compression systems sometimes cannot even meet this load. Consequently, oversized compressors are installed to dehumidify the incoming air. To meet humidity requirements, vapor-compression systems are often operated for long cycles and at low temperatures, which reduce their efficiencies and require reheating the dry, cold air to achieve comfort conditions. Both consequences are costly. However, a combination of a desiccant dehumidification system and a vapor compression system (also known as hybrid desiccant cooling system) may not only meet the load but also save energy.

Commercially available desiccants include silica gel, activated alumina, natural and synthetic zeolites, titanium silicate, lithium chloride, calcium chloride, triethylene glycol, and synthetic polymers. In this study, an aqueous solution of lithium chloride was used as the desiccant. Fumo and Goswami [1,2] had earlier studied the heat and mass transfer performance for this desiccant for air dehumidification. A detailed review of liquid desiccant cooling system was given by Oberg and Goswami [3].

This paper compares experimental results for the performance of a hybrid desiccant system with a vapor compression system.

**Experimental Facilities.** Field tests of a hybrid solar liquid desiccant cooling system were conducted at a test house at the University of Florida's Energy Research and Education Park. These tests consisted of operating the air conditioning system of the house in two configurations—the conventional vapor compression system (properly sized existing system), and the hybrid desiccant system. Figure 1 shows the air conditioning system. The system was operated in the two configurations (vapor compression system with and without the liquid desiccant system) and the data was collected to compare the performance of the two arrangements. For the *vapor compression system only* mode, the desiccant system was bypassed as shown by the dotted line in Fig. 1.

The desiccant system was constructed as a tower of 56 cm dia and 60 cm height. The tower was packed with Raschig rings made with polypropylene and with a specific surface area of  $210 \text{ m}^2/\text{m}^3$ . To distribute the desiccant over the packings, three spray heads were used. The system had two tanks and two pumps; one tank stores the unused desiccant, which is pumped to the tower, and the other tank stores used desiccant, which is pumped by the second pump from the bottom of the tower. This arrangement ensures constant temperature and concentration condition of the desiccant going into the tower. Temperatures were measured using copper-constantan thermocouples, with an accuracy of  $\pm 0.5^\circ\text{C}$  and relative humidities were measured by humidity transmitters, with an accuracy of  $\pm 2\%$  RH (0-90% RH) and  $\pm 3\%$  RH (90%-100%). Humidity ratio of air was determined from the relative humidity and temperature measurements. The uncertainty of the humidity ratio is  $\pm 0.0009 \text{ kg/kg}$ . The air velocity was measured using a Velometer with an accuracy of  $\pm 2\%$ . Air velocity was measured at a cross section of the duct using a matrix of points from which the average air velocity was

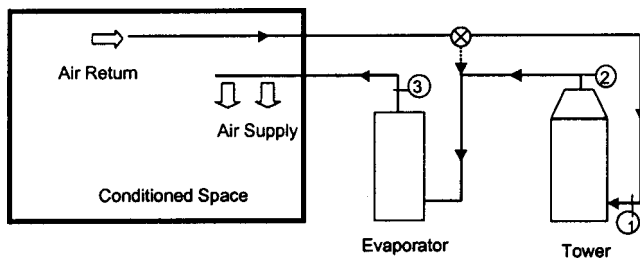


Fig. 1 Air conditioning system showing measurements locations

calculated. The mass flow rate was calculated from the average velocity, the cross sectional area and the density of air.

Desiccant concentration was determined by the Karl Fischer titration method. This is a quantitative method for determining water concentration by titrating the desiccant solution with anhydrous alcohol containing iodine and sulfur dioxide. The uncertainty of this method is  $\pm 0.003 \text{ kg LiCl/kg Solution}$ .

**Experimental Procedure.** Both experiments were conducted under the same air inlet conditions for approximately one hour each. The following data were measured using a PC-based data acquisition system:

- Temperature and relative humidity of the air at the inlet of the tower (Point 1, Fig. 1)
- Temperature and relative humidity of the air at the outlet of the tower or inlet of the evaporator (Point 2, Fig. 1)
- Temperature and relative humidity of the air at the outlet of the evaporator (Point 3, Fig. 1)
- Temperature of the liquid desiccant in the tower
- Time
- Air flow rate

To ensure steady state before recording the data, air and desiccant conditions (temperature and RH of inlet and outlet air, temperature of the desiccant) were monitored. It took approximately 45 min to achieve steady state conditions. Samples of the liquid desiccant were taken before and after the experiment to measure the concentration of the liquid desiccant to determine if regeneration was needed.

The maximum airflow rate was the maximum that the air handler of the actual system could attain. Lower flow rates were obtained using dampers. Valves were used to regulate the mass flow of the liquid desiccant.

## Results

Experiments were conducted to study the influence of air mass flow rate, temperature of the inlet air, temperature of the desiccant, and desiccant mass flow rate on the performance of both system configurations. The air conditioning system was operated in both hybrid and conventional configurations, and for recirculating air and 100% fresh air modes. Each variable was studied for three different values while the others were held constant. After reaching a steady state, the data listed in the previous section were taken each minute during the experiment, which was continued for about 15 min. The minute-by-minute data was averaged to give one experimental data point for this study. The experiment was repeated for each condition three times to give three experi-

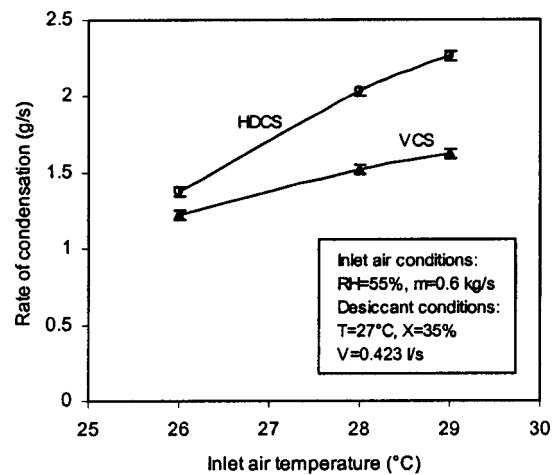
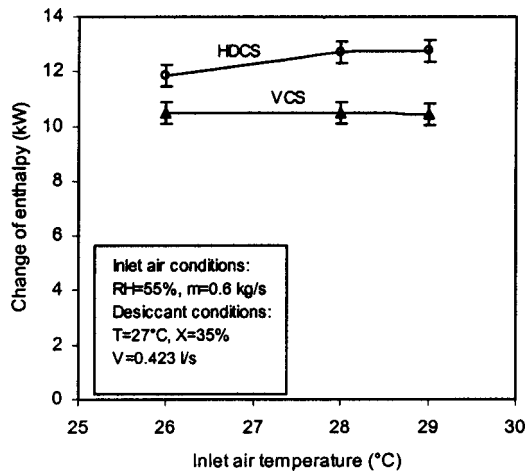


Fig. 2 Influence of inlet air temperature with constant relative humidity on the water condensation rate



**Fig. 3 Influence of inlet air temperature on change of enthalpy of the system**

mental data points. These three experimental data points for the same set were averaged to provide a data point to plot in the figures. A standard deviation was also calculated. The average of the standard deviation of three points for a curve were plotted as the error bars in each plot. Complete results of the experiments are given in Mago and Goswami [4]. A sample of the results is shown in Figs. 2 and 3.

### Conclusions

The following conclusions may be drawn from the complete study although just a sample of the results is shown in this paper:

- In the hybrid liquid desiccant system, rate of condensation increases with: airflow rate, inlet air temperature or desiccant mass flow rate; and it decreases when the inlet desiccant temperature is increased.
- In the hybrid liquid desiccant system, outlet air temperature increases with: airflow rate, inlet air temperature or inlet desiccant temperature; and decreases when the desiccant mass flow rate is increased.
- In the hybrid liquid desiccant system change in enthalpy of air is almost constant with the change in airflow rate and with the desiccant mass flow rate or the inlet air temperature. However the change of enthalpy decreases when the inlet desiccant temperature increases.

### Nomenclature

- HDCS = hybrid desiccant cooling system  
 RH = relative humidity (%)  
 T = temperature (°C)  
 V = desiccant volumetric flow rate (l/s)  
 VCS = vapor compression system  
 X = concentration in mass of the desiccant (%)

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