Desiccant Dehumidification in Air Conditioning for a Large Commercial Building

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Abstract

A case study was done to investigate the performance of air conditioning systems for a large newly built supermarket. Data for routine operating conditions were gathered through an automatic data monitoring system and were downloaded for analysis of the system. An alternative conventional cooling system is designed and analyzed for comparison with the desiccant cooling system. The study showed that incorporating the desiccant dehumidification unit improved the performance of the air conditioning system, reduced operating costs, and increased the levels of human comfort, in addition to a simple reduction of humidity. The study also revealed the possible design improvement on the desiccants units. The improvement focused on pre-cooling the make-up air and dehumidifying it with the desiccant before the air blends with return air from the zone. The investigation produced significant information on design and selection criteria of the system and improvement on the system operation of the desiccant unit used for air conditioning systems for large commercial buildings. It is intended to incorporate this study into the thermal design or HVAC course in the mechanical engineering program.

I. Introduction

Large commercial buildings, such as supermarkets or restaurants, require large capacity air conditioning units to maintain a comfortable environment for their customers and employees. These air conditioning units control, not only the indoor temperature, but also excessive moisture that may be generated by people, cooking and other operating processes. The installation and operating costs of these units are usually high. In order to reduce the unit capacity and operation costs, many buildings, in recent years, are being equipped with desiccant cooling units that reduce the indoor moisture level. These relatively new devices in commercial or residential air conditioning are expected to increase the performance of the overall system, and, thus, reduce the costs of installation and operation.

Desiccants exhibit an affinity so strong for moisture that they attract and hold water vapor directly from the surrounding air. This affinity can be regenerated repeatedly by heating the desiccant material to drive off the collected moisture. Desiccants are placed in dehumidifiers, which have traditionally been used in tandem with mechanical refrigeration in air-conditioning systems. The systems have been more commonly applied in typical air-conditioning environments that involve large dehumidification load fractions, such as low humidity levels required for operations in many industries. Lower humidity levels, below the level necessary for comfort, are generally unattainable, cost-effectively, with mechanical refrigeration and reheat. Numerous moisture sensitive manufacturing, thus, utilize desiccant dehumidification in industrial air-conditioning.

Interest is now being revived in thermal energy driven desiccant dehumidification in non-industrial air-conditioning applications to offset the rising cost of electricity. Lower cost thermal energy, including natural gas, waste heat, solar energy, and other sources, is substituted for electric energy to meet the dehumidification load on the air-conditioning system. In addition, the cost of desiccant dehumidification equipment has decreased considerably to warrant wide use of this method for application outside the industrial field of use. Desiccant dehumidification provides a cost saving to reduce electric air-conditioning capacity and, thus, to lower electric-energy costs and power demand charges in many non-industrial air conditioning situations [1].

II. Scope of the study

An analysis is made to investigate the thermal performance of a desiccant cooling system that is to be installed for air-conditioning of a newly built supermarket building. An alternative conventional cooling system is designed and analyzed for the purpose of comparison with the performance of the desiccant unit. The area considered for this analysis is a part of the main sales area of the supermarket store located in northeast Ohio. This sales area contains open-faced multi-deck meat, multi-deck dairy, produce, and frozen food cases/cabinets, which have remote condensers located outside the conditioned space. Fig. 1 shows the area considered, the positions of the cabinets, the frozen food cases, and the temperatures that must be maintained inside them. The figure was prepared using 'Encore 2100' software that monitors and controls all HVAC and utility functions in the store [2].

The area is to be conditioned by a desiccant cooling system while the proper air-conditioning for the store is maintained by a vapor-compression refrigeration system. On the air-conditioning load, the refrigerated open display cabinet/cases located in the area have a large impact. Heat transferred to the cabinets comes from the conditioned space and is rejected at the remote condensers, effectively reducing the sensible heat gain. This effect, plus the latent cooling done at the cabinets, is very significant and must be taken into account when calculating the heat gains, the cooling load and the ratio of sensible to total heat in the central air-handling plant. The design conditions are selected from 'CARRIER CODES' as shown below [3].

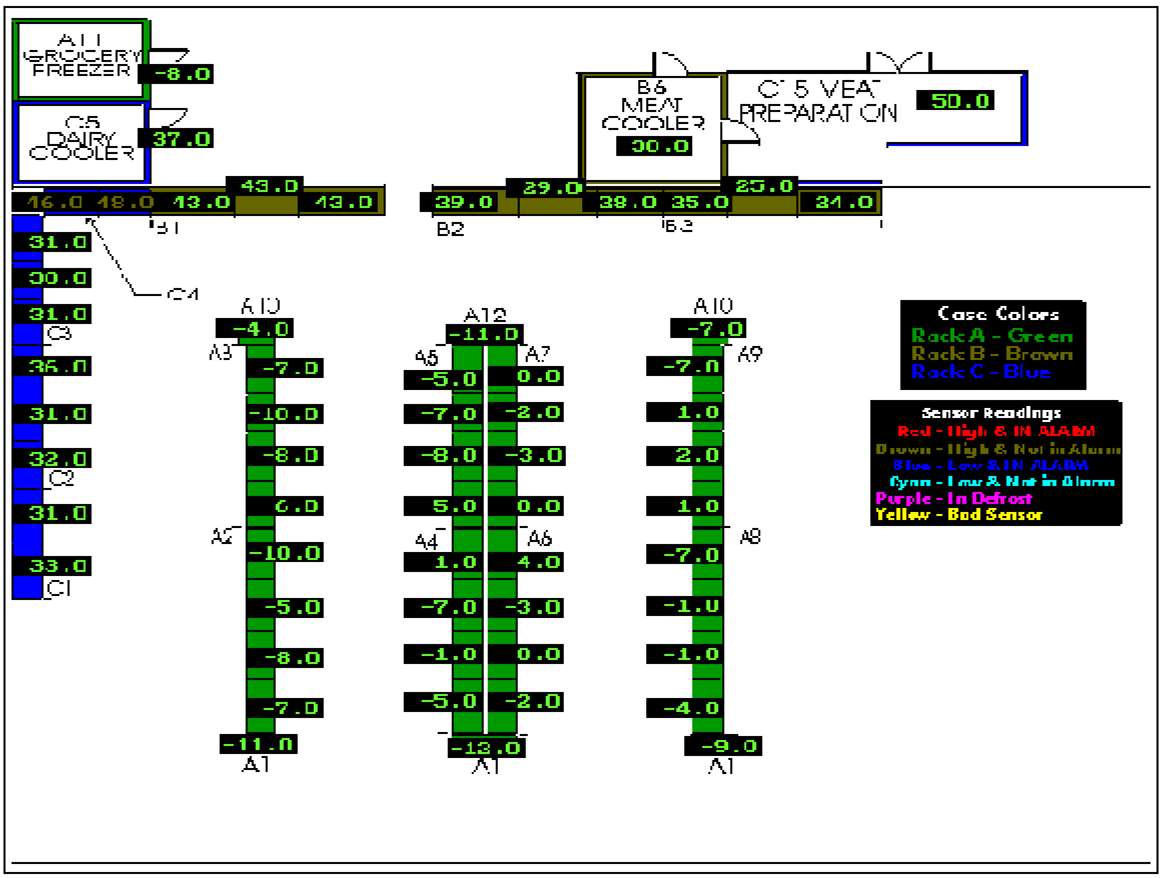
Design conditions for the main sales area

1. Summer
   a) Outdoor design- Per State Energy Codes
   b) Indoor design- $74^o F$ .D.B.- $50^o F$ Dewpoint
2. Winter
   a) Outdoor design- Per State Energy Codes
   b) Indoor design- $70^o F$ .D.B.
3. People loads: 120 sq.ft./person of sales area (sensible and latent loads per ASHRAE)
4. Ventilation
   a) 15.0 C.F.M. per person minimum

b) Exact quantity of ventilation air shall be at least 10% greater than the total of exhaust air affecting the sales area

5. General
   a) Conventional system- 0.75 to 0.85 C.F.M. per square foot of floor area
   b) Desiccant dehumidification System- 0.65 to 0.75 C.F.M per square foot of floor area
   c) Miscellaneous heat producing equipment shall be considered in load calculation.
   d) Refrigerated cases/cabinets
   Credits shall be given 50% of total caseload for open faced multi-deck meat, multi-deck dairy, produce, and frozen food cases.

Fig. 1 The Area of Study and Temperature Distribution



III. Desiccant cooling system

A typical desiccant cooling system consists of a desiccant wheel, a heat exchanger, two evaporative coolers and associated blowers for air movement. As the moisture of the process air is adsorbed by the desiccant wheel, the air becomes drier and warmer, due to the desiccant's heat of adsorption. For an effective cooling system, all or most of this heat is typically rejected by transferring the heat to regeneration air using an air-to-air stationary heat exchanger or a rotary heat wheel. The cooler, dry air leaving the heat exchanger is then passed through an evaporative cooler, which adds moisture to the air, reducing its temperature further before it enters the conditioned space [4].

In designing the desiccant cooling system for this study, data gathered from the building by 'Encore 2100' software were used as input data as shown in Table 1. The cooling load for the store is estimated using 'HAP' (Hourly Analysis Program) that estimates design cooling and heating loads for commercial buildings in order to determine required sizes for HVAC system components. Ultimately, the program provides information needed for selecting and specifying equipment. The data from HAP was fed into the 'Desiccant Unit Selection' software to select the suitable desiccant unit required for the system. The desiccant units considered for the system are ENGELHARD/ICC models. The unit is chosen according to the amount of air entering to the desiccant wheel [5]. DC080 desiccant model is selected for the desiccant system, in which the process airflow is 7830 cfm.

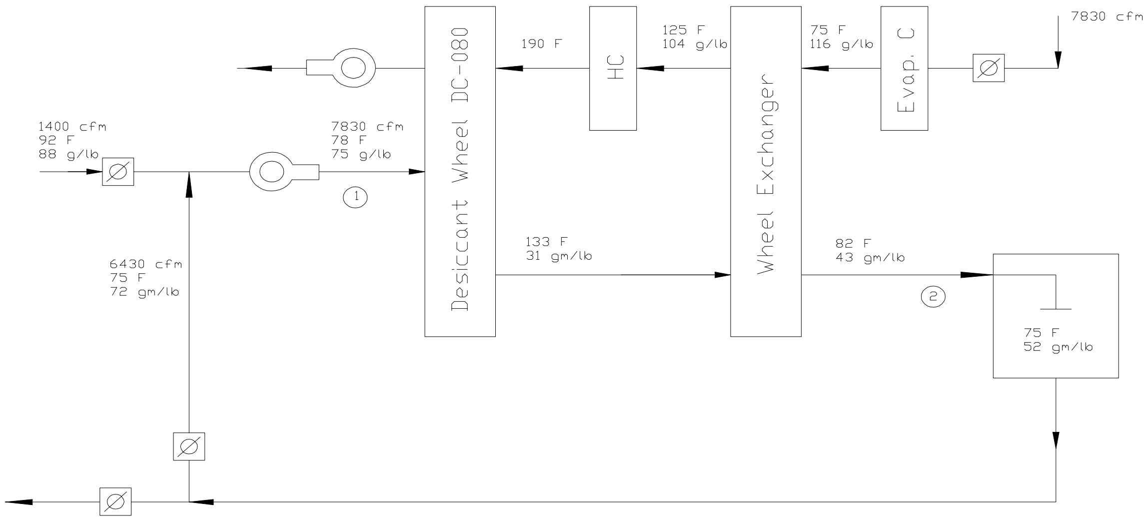
Table 1 Design Airflow Schedule/ Airflow Configuration

Inlet Conditions						
Process Air	CFM	FDB	FWB	GR/LB	FDP	RH
Outside/Ambient	1400	92	72.4	88	63.6	39
Inside/Bldg Return	6430	75	64.1	72	58.1	55.6
Process Mixture	7830	78	65.7	74.9	59.2	52.4
Regeneration Air						
Outside/Ambient	7830	92	72.4	88	63.6	39
Inside/Bldg Exhaust	0	75	64.1	72	58.1	55.6
Regeneration Mixture	7830	92	72.4	88	63.6	39
Supply Air Conditions						
After Heat Exchanger	7830	82	60	42	44	27

The data that were implemented to the ' Desiccant Unit Selection' software show that the process airflow leaves the desiccant wheel at 133 °F DB and 31 gr/lb. An additional cooling is needed to meet the sensible and latent load requirements. The heat wheel provides part of the sensible cooling. However, it is not sufficient to meet the sensible load requirement of the conditioned space. In order to meet the load requirements of the conditioned space, indirect evaporative cooling coil is incorporated in the system. Consequently, the system consists of: desiccant rotor, heat exchanger rotor, boiler, regeneration heating coil, indirect evaporative cooler, process air blower, and regeneration air blower as shown in Fig 2. Table 2 shows the system data values for this design.

The performance of the two systems considered in this study is evaluated by calculating COP. The term coefficient of performance (COP) has been devised to measure the effectiveness of refrigerating systems and is usually defined as the ratio of the refrigeration produced to the net work supplied. This definition of COP is applied mainly for conventional refrigerating systems [5]. However, the definition is not suitable for other cooling systems such as desiccant or mixed cooling systems due to additional energy input. A new definition provided by Shen and Worek was used to formulate the COP for all cooling systems including the desiccant, conventional or mixed systems [7].

Fig. 2 Desiccant Cooling System



COP for the desiccant system is defined as [7],

$$COP = \frac{cooling\ capacity}{energy\ input} = \frac{H_{out} - H_{in}}{w_1^{\cdot} + w_{in2}^{\cdot} + Q_3^{\cdot} + w_4^{\cdot}} \tag{3.1}$$

$$H_{out} = m_{a1}^{\cdot} h_1 \quad \text{in Btu/hr} \tag{3.2}$$

where $m_{a1}^{\cdot}$ = mixed air flow rate before the desiccant wheel

h_1 = enthalpy of mixed air

$$H_{in} = m_{a2}^{\cdot} h_2 \quad \text{in Btu/hr} \tag{3.3}$$

where $m_{a2}^{\cdot}$ = supplied air flow rate

h_2 = enthalpy of supplied air

$w_1^{\cdot}, w_4^{\cdot}$ = power input to run the supply and the regeneration fan

$w_{in2}^{\cdot}$ = power input to run the compressor of the evaporative cooler coil

$Q_3^{\cdot}$ = power required to run the heating coil

The energy required to run the heating coil is equal to the energy required to heat the air sensibly and is calculated from

$$q_s^{\cdot} = m_a^{\cdot}(h_1 - h_2) \tag{3.4}$$

Air-handling components in the systems such as fans, ducts, and so forth, are selected on the basis of *volume* flow rather than mass flow of air [8]. Therefore, if the air volume flow rate is to be determined, it is necessary to specify the point in the system where flow volume rate is to be determined and find the specific volume of the air at that point. With the mass flow rate, $\dot{m}_a$, known and the specific volume of the air at the point, the volume flow rate in cubic feet per minute can be calculated from

$$cfm = \frac{\dot{m}_a v}{60} \qquad (3.5)$$

Air-handling equipment is normally rated on the basis of "standard air" which has been defined as dry air at $70^o F$ and 29.92 in. Hg barometric pressure, which gives a density of 0.075 lbm/ft^3. Using the density and substituting the value of $\dot{m}_a$ in Eq. (3.5), the sensible heat becomes

$$\dot{q}_s = (1.10 cfm)(t_2 - t_1) \quad \text{in Btu/hr} \qquad (3.6)$$

The power input to the compressor is calculated from:

$$\dot{W}_{in} = \dot{m}_r (h_4 - h_3) \quad \text{in Btu/hr} \qquad (3.7)$$

where $\dot{m}_r$ = mass flow rate of the refrigerant

h_3, h_4 = enthalpy of the refrigerant at conditions 3 and 4

Based on Eqs. (3.1) to (3.7), the value of COP for the desiccant system is calculated. Spreadsheets were used to find the value of COP as shown in Table 3. The energy required to run the desiccant system is broken down as: 548,000 Btu/hr for cooling coil, 10,580 Btu/hr for compressors, 50,890 Btu/hr for fans. This energy is composed mainly of thermal energy required to run the heating coils and electrical energy to run the compressors and fans. The breakdown gives an idea about the operating costs to run the system. However, detailed cost analysis is not included in this study.

Table 2 System Data Values for the Desiccant System

Component	Dry Bulb Temp.	Specific Humidity	Airflow
Process Air	FDB	GR/LB	CFM
Ventilation Air	92	88	1400
Vent-Return (Mix.)	78	75	7830
Desiccant Wheel	133	31	7830
Wheel Exchanger	82	43	7830
Zone Air	75	52	7830
Regeneration Air			
Outside/ Ambient	92	88	7830
Evap. Cooling Coil	75	116	7830
Wheel Exchanger	125	104	7830
Reg. Heating Coil	190	104	7830

Table 3 COP for the Desiccant Cooling System

Energy required to run the compressors

	Total Coil Load (ton)	Refrigerant Flow Rate (Lb/hr)	Inlet Enthalpy (Btu/lb)	Inlet Evaporative Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Enthalpy (Btu/lb)	Work Input (Btu/hr)
Evaporat. Cooler	12	2300	76.4	13.8	39	81	10580

	Air Flow Rate (cfm)	Spec. Volume (ft3/lb)	Enthalpy of Air (Btu/lb)	Energy Required (Btu/hr)
Mixed Air	7830	13.75	30.5	1042102
Supplied Air	7830	13.75	26.5	854182

Energy required to run the fans

	Hp	Btu/hr
Desiccant Unit	2*10	50890

Desiccant Unit	
Evaporative Cooler	12 tons
Heating Coil	548000 Btu/hr
Supply Fan	10 Hp
Return Fan	10 Hp

Mixed Air (Btu/hr)	Supply Air (Btu/hr)	Heating Coils (Btu/hr)	Compressor (Btu/hr)	Fans (Btu/hr)	COP
1042102	854182	548000	10580	50890	0.308

VI. Conventional Cooling System

When air is chilled below its dew point temperature, moisture condenses and, thus, the air is dehumidified by the process of cooling and condensation. The amount of moisture removed depends on how cold the air can be chilled. The moisture removed from the air during any dehumidification process may be determined directly from the difference in specific humidity for the actual entering and leaving states. The cooling of buildings is actually made up of two processes: sensible cooling that reduces air temperature, and latent cooling that removes water vapor from the air. The latent heat removal associated with this moisture removal can be calculated from the following equation [9]:

$$q_l = 0.68(cfm)(\Delta W) \tag{3.8}$$

where q_l = latent heat removal (Btu/hr)

ΔW = moisture removal (gr/lb)

Cooling coils often have low latent capacities, usually ranging from 20% to 30%. This high coil sensible heat ratio can create problems when the SHR of the load falls below 70 %, since the coil will no longer have enough latent capacity to meet the latent load [5]. These cooling coils cool the air to levels between 43 and 45 ^oF. Below that point, frost begins to form on parts of the coil, spreading slowly through coil as the airflow becomes restricted. Eventually the frost blocks the airflow all together and dehumidification ceases. Reheating the air must be performed in order to dehumidify it again to get the required humidity of the air concerned. In order to overcome this frosting problem and to meet the load requirements of the conditioned space, a dual air system is usually employed. Air is supplied to the two air streams at different temperatures and mixed by proportioning dampers upstream in a plenum [10]. The entire air quantity for absorbing the load is conditioned centrally and distributed by the main fan. The idea is adopted in this study.

The 'HAP' software was used for designing this conventional cooling system. Weather data, the space conditions and the properties of the building were chosen from ASHRAE, based on the specifications and the properties of the building[11]. Table 4 shows the space input data for the main sales area of the store.
Fig. 3 shows the conventional unit selected and conditions of air. The system is composed of pre-cooling coil, pre-heating coil, a supply air-blower, a return air-blower and a dual duct system that contains heating and cooling coils and a mixing box. System data values were taken from Table 5 and the psychrometric analysis for the conventional cooling unit is shown in Fig.4.
COP for the conventional cooling system is also defined similarly as [7],

$$COP = \frac{H_{out} - H_{in}}{w_{in1}^{\cdot} + Q_2^{\cdot} + w_3^{\cdot} + w_{in4}^{\cdot} + Q_5^{\cdot} + w_6^{\cdot}} \tag{3.9}$$

$$H_{out} = m_{a1}^{\cdot} h_1 + m_{a2}^{\cdot} h_2 \tag{3.10}$$

where $m_{a1}^{\cdot}, m_{a2}^{\cdot}$ = ventilation and return air flow rate

h_1, h_2 = enthalpy of ventilation and return air

$$H_{in}^{\cdot} = m_{a3}^{\cdot} h_3 \tag{3.11}$$

where $m_{a3}^{\cdot}$ = supply air flow rate

h_3 = enthalpy of supply air

$w_{in1}^{\cdot}, w_{in4}^{\cdot}$ = power input to the compressors of the pre-cooling and cooling coils

$Q_2^{\cdot}, Q_5^{\cdot}$ = energy required to run the pre-heating and heating coils

$w_3^{\cdot}, w_6^{\cdot}$ = power input to run the supply and return fans

Based on Eqs. (3.8) to (3.11), the COP for the conventional cooling system is calculated. Spreadsheets were used to find the value of COP as shown in Table 6. The energy required to run the conventional system is broken down as: 218,616 Btu/hr for heating coils, 77,920 Btu/hr for compressors, and 61,068 Btu/hr for fans. Compared to the desiccant cooling system, larger fraction of the energy comes from the electricity.

Fig. 3 Conventional Cooling System

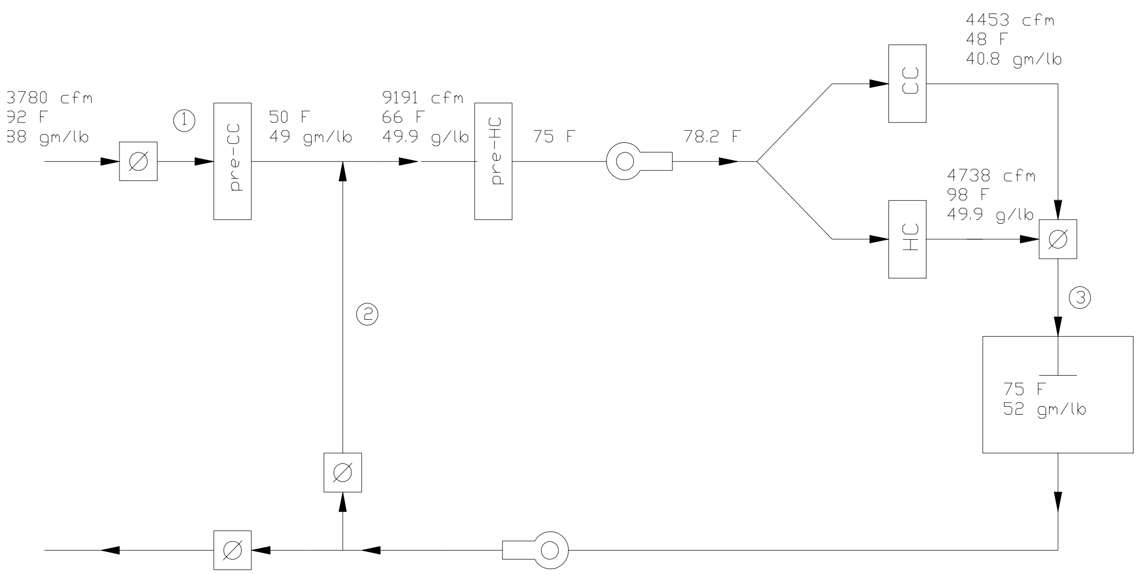


Table 4 Space Input Data for the Store

Space Input Data

Project Name: Giant Eagle project
Prepared by: AETOS

Main Sales Area

1. General Details:

- Floor Area ______ **13608.0** ft²
- Avg. Ceiling Height ______ **14.000** ft
- Building Weight ______ **70.000** lb/ft²

2. Internals:

2.1. Overhead Lighting:

- Fixture Type ______ **Recessed (Unvented)**
- Wattage ______ **2.50** W/ft²
- Ballast Multiplier ______ **1.00**
- Schedule ______ **Lighting_overhead**

2.2. Task Lighting:

- Wattage ______ **1955.0** Watts
- Schedule ______ **Lighting_Task**

2.3. Electrical Equipment:

- Wattage ______ **0.00** W/ft²
- Schedule ______ **None**

2.4. People:

- Occupancy ______ **90.00** ft²/person
- Activity Level ______ **User defined**
- Sensible ______ **250.0** BTU/hr/person
- Latent ______ **200.0** BTU/hr/person
- Schedule ______ **Occupents**

2.5. Miscellaneous Loads:

- Sensible ______ **-174020** BTU/hr
- Schedule ______ **Cabinets**
- Latent ______ **0** BTU/hr
- Schedule ______ **Cabinets**

3. Walls, Windows, Doors:

Exp.	Wall Gross Area (ft²)	Window 1 Qty.	Window 2 Qty.	Door 1 Qty.
S	1582.0	0	0	0

3.1. Construction Types for Exposure S

- Wall Type ______ **Default Wall Assembly**

4. Roofs, Skylights:

Exp.	Roof Gross Area (ft²)	Roof Slope (deg.)	Skylight Qty.
H	13608.0	0	0

4.1. Construction Types for Exposure H

- Roof Type ______ **Assembly Roof**

5. Infiltration:

- Design Cooling ______ **0.0** CFM
- Design Heating ______ **0.0** CFM
- Energy Analysis ______ **0.0** CFM

Infiltration occurs only when the fan is off.

6. Floors:

- Type ______ **Slab Floor On Grade**
- Floor Area ______ **13608.0** ft²
- Total Floor U-Value ______ **1.200** BTU/hr/ft²/F
- Exposed Perimeter ______ **113.0** ft
- Edge Insulation R-Value ______ **7.0** hr-ft²-F/BTU

7. Partitions:

(No partition data).

Fig. 4 Psychrometric Analysis for the Conventional Unit

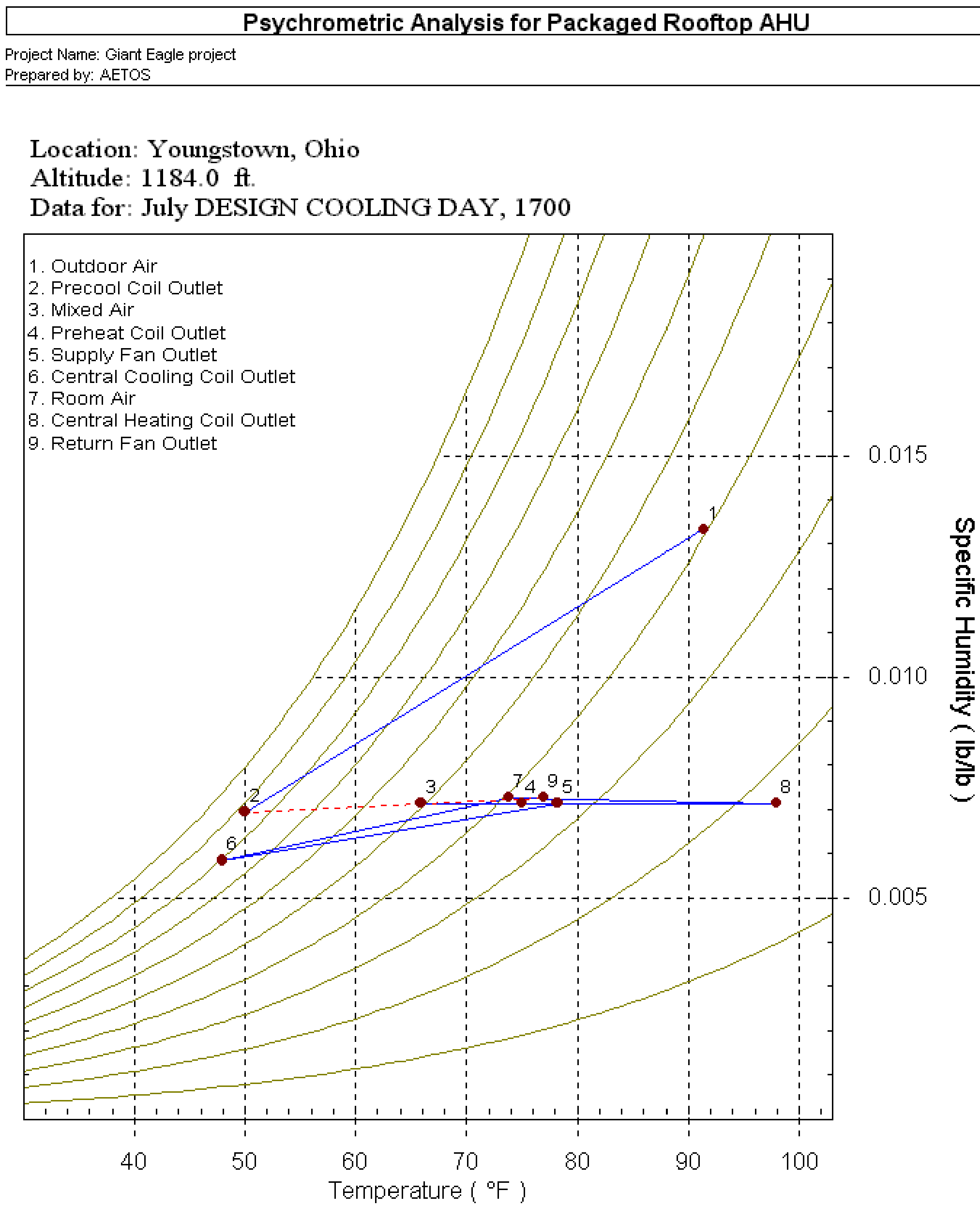


Table 5 System Psychrometrics for the Conventional Cooling System

System Psychrometrics for Packaged Rooftop AHU

Project Name: Giant Eagle project
Prepared by: AETOS

July DESIGN COOLING DAY, 1700

TABLE 1: SYSTEM DATA

Component	Location	Dry-Bulb Temp (°F)	Specific Humidity (lb/lb)	Airflow (CFM)	Sensible Heat (BTU/hr)	Latent Heat (BTU/hr)
Ventilation Air	Inlet	91.4	0.01332	3780	56303	104130
Precool Coil	Outlet	50.0	0.00694	3780	161834	109664
Vent - Return Mixing	Outlet	65.9	0.00713	9191	-	-
Preheat Coil	Outlet	75.0	0.00713	9191	86657	-
Supply Fan	Outlet	78.2	0.00713	9191	30533	-
Central Cooling Coil	Outlet	48.0	0.00584	4453	139188	26171
Central Heating Coil	Outlet	98.0	0.00713	4738	96995	-
Cold Supply Duct	Outlet	48.0	0.00584	4453	0	-
Hot Supply Duct	Outlet	98.0	0.00713	4738	0	-
Zone Air	-	73.8	0.00727	9191	0	31752
Return Plenum	Outlet	73.8	0.00727	9191	0	-
Return Fan	Outlet	77.0	0.00726	9191	30533	-

TABLE 2: ZONE DATA

Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	15644	Deadband	0	73.8	9191	0	0

Table 6 COP for the Conventional Cooling System

Energy required to run the compressors

	Total Coil Load (ton)	Refriger. Flow Rate (Lb/hr)	Inlet Enthalpy (Btu/lb)	Inlet Evapor. Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Enthalpy (Btu/lb)	Work Input (Btu/hr)
Pre-Cool Coil	23	5200	76.4	23.3	82.5	87	15520
Cooling Coil	14.5	3000	76.4	18.4	55	84	22800

Convent. Unit	Air Flow Rate (cfm)	Spec. Volume (ft3/lb)	Enthalpy, Air (Btu/lb)	Energy Required (Btu/hr)
Ventilation Air	3780	14.15	35.9	575415
Returned Air	5411	13.65	26.5	630292
Supplied Air	9191	13.6	27.7	1123194

Energy required for the fans

	Hp	Btu/hr
Conven. Unit	2*12	61068

Conventional Unit	
Pre-Cool Coil	23 tons
Cooling Coil	14.5 tons
Pre-Heat Coil	96069 Btu/hr
Heating Coil	122547 Btu/hr
Supply Fan	12 Hp
Return Fan	12 Hp

Ventilation Air (Btu/hr)	Returned Air (Btu/hr)	Supply Air (Btu/hr)	Heating Coils (Btu/hr)	Compressors (Btu/hr)	Fans (Btu/hr)	COP
575415	630292	1123194	218616	77920	61068	0.231

V. Conclusion

The analysis described in the previous sections showed that the values of COP for the desiccant and the conventional systems are 0.308 and 0.231, respectively. This result confirms that the desiccant system has considerable advantages over the conventional system for the application considered for commercial air-conditioning. Further observations made on desiccant-based and cooling-based dehumidification systems are [1,10]:

1. Cooling and desiccant-based dehumidification systems are most economical when used together since they complement each other. The difference in the cost of electrical power and thermal energy will determine the optimum mix of desiccant to cooling-based dehumidification in a given situation.
2. Desiccants are especially efficient when drying air to create low relative humidity, and cooling-based dehumidification is efficient when drying air to saturated air conditions.
3. The analysis dealt with an actual application of air-conditioning methods and the material produced in this study is adequate to be incorporated in senior-level design-intensive courses in mechanical engineering.

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