ANALYSIS AND DESIGN OF DESICCANT COOLING SYSTEMS

by

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ABSTRACT

In the last decade, desiccant dehumidification has emerged as an alternative or as a supplement to conventional vapor compression systems for cooling and conditioning air in commercial and industrial buildings. It provides a method of drying air before it enters a conditioned space. When combined with conventional vapor compression systems, desiccant dehumidification systems are a cost-effective means of supplying cool, dry, filtered air. The object of this study is to investigate the performance of a desiccant cooling unit for a large building supermarket in Canfield, Ohio. Optimization of this system is done by computer simulation. Then a conventional vapor compression system is designed for the supermarket and the performances of the two systems are compared. This comparison shows that the desiccant unit performs better than a conventional unit in the environment of the supermarket. Another case study is to design and investigate the performance of a desiccant cooling unit in a more humid area, Tampa, Florida. This case study shows that the desiccant performs better in regions with large specific humidity. A further study deals with the design improvement on the desiccant units to enhance the units' performance. The improvement focuses on pre-cooling the make-up air and dehumidifying it with the desiccant before the air blends with return air from the zone. The study shows that significant improvement can be made for the cooling units used in Florida.

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CHAPTER I

STATEMENT OF THE PROBLEM

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 the moisture that may be generated by people, cooking, and other processes. The installation and operating costs of these units are usually high. Many such buildings, in recent years, are being equipped with desiccant cooling units that reduce the indoor moisture level with an expectation that these relatively new devices will increase the performance of the overall air-conditioning system, and thus reduce the unit capacity and the operation costs. However, the performance assessment of desiccant system have not been extensively reported or published.

The purpose of this thesis is to investigate the performance of the air conditioning system for a large supermarket building and to find an alternative design for improvement of the system.

A portion of the supermarket is considered for this study which contains an open-faced multi-deck meat, multi-deck dairy, produce and frozen food cases. The air conditioning system must supply treated air to the main sales area of the building. The supermarket requires additional latent cooling (dehumidification) to prevent condensation on frozen food. The only option available in a conventional air conditioning system is to turn down the thermostat, resulting in both latent and sensible cooling. This leads to air temperatures that are uncomfortably cold for the customers. Gas fired desiccant dehumidification breaks the link between sensible and latent cooling, allowing building operators to precisely control each to its required level.

In this study, the data relevant for routine operating conditions are shown in Table 3.2. These data were used to study and analyze the desiccant cooling unit. A conventional cooling unit was also designed for the same area in order to compare the coefficient of performance for the two units.

CHAPTER II

THEORY OF DESICCANTS AND DESICCANT COOLING SYSTEMS

Desiccants exhibit an affinity so strong for moisture that they can draw water vapor directly from the surrounding air. This affinity can be regenerated repeatedly by applying heat to 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 specialty air-conditioning systems. The systems have been more commonly applied in typical air-conditioning situations that involve large dehumidification load fractions. This situation often arises due to 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.

Desiccant dehumidification technology has been used in military storage and many industrial applications for more than 60 years. Continuous desiccant dehumidification can be achieved in a number of ways using liquid spray tower, solid packed tower, rotating horizontal bed, multiple vertical bed, and rotating wheel.

In the past, desiccant dehumidification was also integrated with mechanical refrigeration in early air-conditioning approaches for comfort in business and homes.

To the present day, though, in specialty applications, desiccant dehumidification still holds its economic advantage over dehumidification by mechanical refrigeration and reheat. In industrial air-conditioning, numerous moisture sensitive manufacturing and storage applications utilize desiccant dehumidification. The dramatic decrease in product rejection, and thus the direct increase in profitability of the product, yields a quick payback on the initial investment in depressed humidity control equipment.

Interest is now being revived in thermal-driven desiccant dehumidification in non-industrial air-conditioning applications to offset rising electricity prices. 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 the past, available desiccant dehumidification equipment has been considered too expensive, compared to assembly-line mechanical refrigeration equipment, for application outside the industrial field of use. But today, desiccant dehumidification technology, supported by ongoing research and development, is providing a cost-saving to reduce electric air-conditioning capacity and thus to lower electric-energy costs and power demand charges in certain nonindustrial air conditioning situations too.

2.1. Desiccants

Many materials are desiccants; that is they attract and hold water vapor. Wood, natural fibers, clays, and many synthetics attract and release moisture like commercial desiccants do, but they lack the holding capacity of some special desiccant materials. For example, woolen carpet fibers attract up to 23 % of their dry weight in water vapor, and nylon can take up almost 6 % of its weight in water. In contrast, a commercial desiccant takes up between 10 and 1100% of its dry weight in water vapor, depending on its type and the moisture available in the environment [1]. Furthermore, commercial desiccants continue to attract moisture even when the surrounding air is relatively dry, a characteristic that other materials do not share.

All desiccants behave in a similar way in that they attract moisture until they reach equilibrium with the surrounding air. Moisture is usually removed from the desiccant by heating it to temperatures between 120 and $500^{o}F$ and exposing it to a scavenger airstream. After the desiccant dries, it must be cooled so it can attract moisture once again. Sorption refers to the binding of one substance to another. It always generates sensible heat equal to the latent heat of water vapor taken up by the desiccant, plus an additional heat of sorption that varies between 5 and 25 % of the latent heat of the water vapor. This heat is transferred to the desiccant and the surrounding air.

The process of attracting and holding moisture is described as either adsorption or absorption, depending on whether the desiccant undergoes a chemical change as it takes on moisture. Adsorption does not change the desiccant except by the addition of the weight of water vapor, similar in some ways to a sponge soaking up water. Absorption, on the other hand, changes the desiccant. An example of this is table salt, which changes from a solid to a liquid as it absorbs moisture.

Sorbents are materials that have an ability to attract and hold gases or liquids. They can be used to attract gases or liquids other than water vapor, a characteristic that makes them very useful in chemical separation processes. Desiccants are subset of sorbents; they have a particular affinity for water.

2.2. Types Of Desiccants

Desiccants can be solids or liquids and can hold moisture through adsorption or absorption. Most absorbents are liquids, and most adsorbents are solids [1].

2.2.1. Liquid Absorbents

Liquid absorption dehumidification can best be illustrated by comparing it to the operation of an air washer. When air passes through an air washer, its dewpoint approaches that of the temperature of the water supplied to the machine. More humid air is dehumidified and less humid air is humidified. In a similar manner, a liquid absorption dehumidifier contacts air with a liquid desiccant solution. The liquid has a vapor pressure lower than water at the same temperature, and when the air passing over the solution approaches this reduced vapor pressure, then it is dehumidified. The vapor pressure of a liquid absorption solution is directly proportional to its temperature and inversely proportional to its concentration.

In standard practice, the behavior of a liquid desiccant can be controlled by adjusting its concentration, its temperature, or both. Desiccant temperature is controlled by simple heaters and coolers. Concentration is controlled by heating the desiccant to drive moisture out into a waste airstream or directly to the ambient.

As a practical matter, however, the absorption process is limited by the surface area of a desiccant exposed to the air being dehumidified and the contact time allowed for the reaction. More surface area and more contact time allows the desiccant to approach its theoretical capacity. Commercial desiccant systems reflect these realities either by spraying the desiccant onto an extended surface much like in a cooling tower, or holding a solution in a rotating extended surface with a large solution capacity.

2.2.2. Solid Adsorbents

Adsorbents are solid materials with a tremendous internal surface area per unit of mass; a single gram can have more than 50,000 ft^2 of surface area. Structurally, they resemble a rigid sponge, and the surface of the sponge in turn resembles the ocean coastline of a fjord. This analogy indicates the scale of the different surfaces in an adsorbent. The fjords can be compared to the capillaries in the adsorbent. The spaces between the grains of sand on the fjord beaches can be compared to the spaces between the individual molecules of the adsorbent, all of which have the capacity to hold water molecules. The bulk of the adsorbed water is contained by condensation into the capillaries, and the majority of the surface area that attracts individual water molecules is in the crystalline structure of the material itself [1].

Adsorbents attract moisture because of the electrical field at the desiccant surface. The field is not uniform in either force or charge, so it attracts polarized water molecules that have an opposite charge from specific sites on the desiccant surface. When the complete surface is covered, the adsorbent can hold still more moisture, as vapor condenses into the first water layer and fills the capillaries throughout the material.

As with liquid absorbents, the ability of an adsorbent to attract moisture depends on how much water is on its surface compared to how much water is in the air. That difference is reflected in the vapor pressure at the surface and in the air. The adsorption behavior of solid adsorbents depends on (1) their total surface area, (2) the total volume of their capillaries, and (3) the range of their capillary diameters. A large surface area gives the adsorbent a larger capacity at low relative humidities. Large capillaries provide a high capacity for condensed water, which gives the adsorbent a higher capacity at high relative humidities. A narrow range of capillary diameters makes an adsorbent more selective in the vapor molecules it can attract and hold; thus, some will fit and others will be too large to pass through the passages in the material.

2.3. Desiccant Life

The useful life of desiccant materials depends largely on the quantity and type of contamination in the airstreams they dry. In commercial equipment, desiccants last between 10,000 and 100,000 hours and longer before they need replacement [1].
Normally, two mechanisms cause the loss of desiccant capacity: change in desiccant sorption characteristics through reactions with contaminants, and loss of effective surface area through clogging or hydrothermal degradation. Liquid absorbents are more susceptible to chemical reaction with airstream contaminants other than water vapor than are solid adsorbents. For example, certain sulfur compounds can react with lithium chloride to form lithium sulfate, which is not a desiccant. If the concentration of such compounds in the airstream were below 10 ppm and the desiccant were in use 24 hours a day, the capacity reduction would be true; this may mean a 10 % reduction in capacity over the course of a year.

In air-conditioning applications, desiccant equipment is designed to minimize the need for desiccant replacement in much the same way that vapor compression cooling systems are designed to avoid the need for compressor replacement. Unlike filters, desiccants are seldom intended to be frequently replaced during normal service in an air-drying application.

2.4. The Desiccant Cycle

All desiccants function by the same mechanism-transferring moisture because of a difference between the water vapor pressure at their surface and that of the surrounding air. When the vapor pressure at the desiccant surface is lower than that of the air, the desiccant attracts moisture. When the surface vapor pressure is higher than that of the surrounding air, the desiccant releases moisture.

Figure 2.1 shows the relationship between the moisture content of the desiccant and its surface vapor pressure. As the moisture content of the desiccant rises, so does the water vapor pressure at its surface. At some point, the vapor pressure at the desiccant is the same as that of the air and the two are in equilibrium. Then moisture cannot move in either direction until some external force changes the vapor pressure at the desiccant or in the air.

Figure 2.1 also shows the impact of temperature on the vapor pressure at the desiccant. Both higher temperatures and increased moisture content increase the vapor pressure at the surface. When the surface vapor pressure exceeds that of the surrounding air, moisture leaves the desiccants process called reactivation or regeneration. After the desiccant is dried (reactivated) by the heat, its vapor pressure remains high, so that it has very little ability to absorb moisture. Cooling the desiccant reduces its surface vapor pressure so it can absorb moisture once again. The complete cycle is illustrated in Fig 2.1.

The operating economics of desiccants depends on the energy cost of moving a

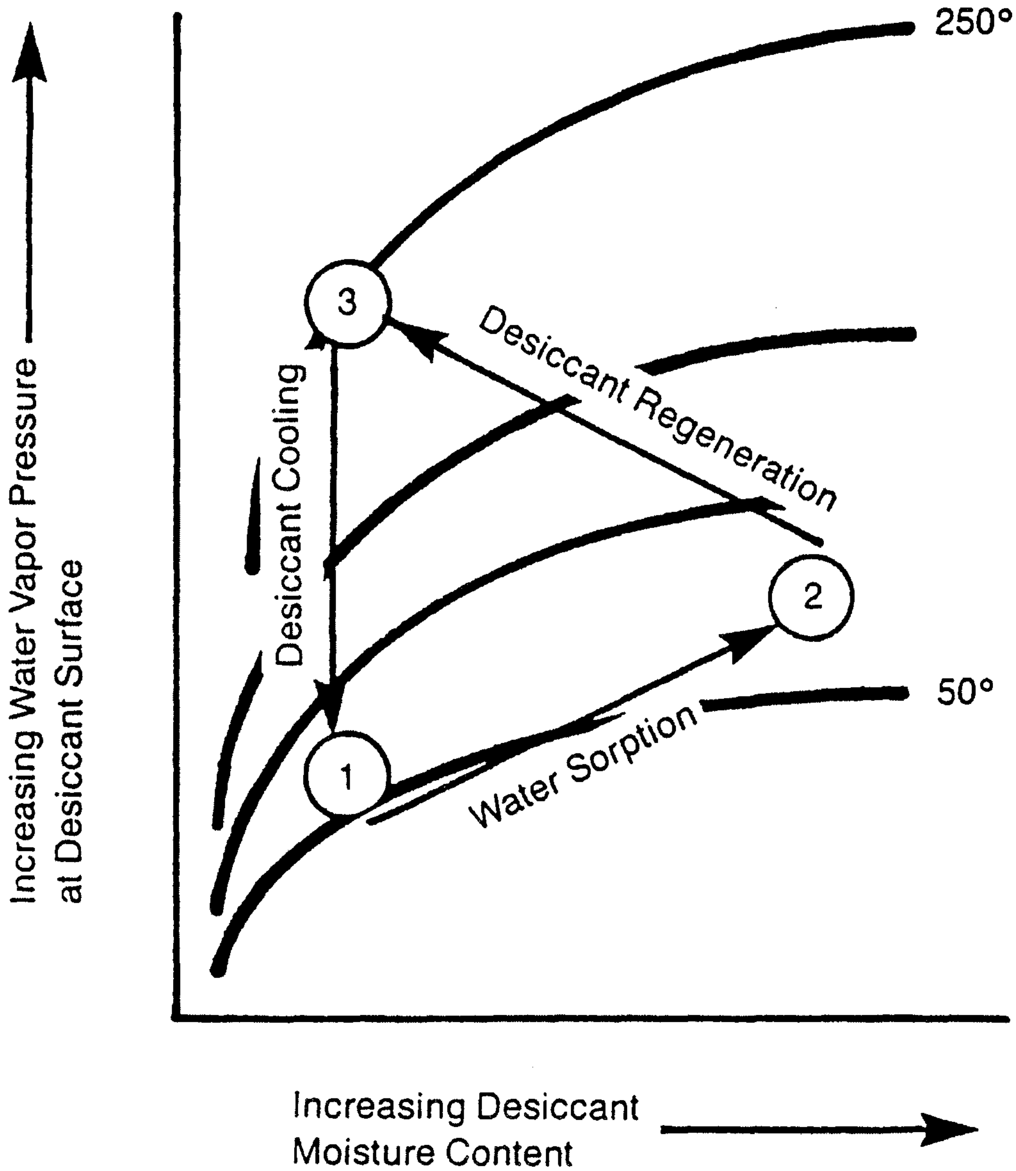


Fig. 2.1 The Desiccant Cycle

given material through this cycle. The dehumidification of air (loading the desiccant with water vapor) generally proceeds without energy input, other than fan and pump costs. The major portion of energy is invested in regenerating the desiccant (moving from point 2 to point 3) and cooling the desiccant (point 3 to point 1). Regeneration energy is equal to the sum of three variables [1]:

1. The heat necessary to raise the desiccant to a temperature high enough to make its surface vapor pressure higher than the surrounding air.
2. The heat necessary to vaporize the moisture it contains (1060 Btu/lb).
3. The small amount of heat from desorption of the water from the desiccant.

The cooling energy is proportional to the mass of the desiccant, and the difference between its temperature after regeneration and the lower temperature that allows the desiccant to remove water from the airstream once again.

The cycle is similar when desiccants are regenerated using pressure differences in a compressed air application. The desiccant is saturated in a high-pressure chamber, i.e., that of the compressed air. Then valves open, isolating the compressed air from the material, and the desiccant is exposed to air at ambient pressure. The vapor pressure of the saturated desiccant is much higher than ambient air at normal pressures, so the moisture leaves the desiccant for the surrounding air. An alternate desorption strategy uses a small portion of the dried air, returning it to the moist desiccant bed to reabsorb the moisture, then venting the air to the atmosphere at ambient pressures.

2.5. Desiccant Applications

Desiccants can dry either liquids or gases, including ambient air, and are used in many air-conditioning applications, particularly when [1]:

1) The latent load is large in comparison to the sensible load.

2) The cost of energy to regenerate the desiccant is low when compared with the cost of energy to dehumidify the air by chilling it below its dewpoint.

3) The moisture control level required in the space would require chilling the air to subfreezing dewpoints if compression refrigeration alone were used to dehumidify the air.

4) The temperature control level required by the space or process requires continuous delivery of air at subfreezing temperatures.

In any of these situations, the cost of running a vapor compression cooling system can be very high. A desiccant process may offer considerable advantages in energy, the initial cost of equipment, and maintenance. Since desiccants are able to absorb more than simply water vapor, they can remove contaminants from airstreams to improve indoor air quality. Desiccants have been used to remove organic vapors, and in special circumstances, to control microbiological contaminants.

Desiccants are also used in drying compressed air to low dewpoints. In this application, moisture can be removed from the desiccant without heat. Desorption is accomplished using differences in vapor pressures compared to the total pressures of the compressed and ambient pressure airstreams.

Finally, desiccants are used to dry the refrigerant circulating in air- conditioning and refrigeration systems. This reduces corrosion in refrigerant piping and avoids valves and capillaries becoming clogged with ice crystals. In this application the desiccant is not regenerated; it is discarded when it has adsorbed its limit of water vapor.

2.6. 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 seen in Fig. 2.2. After the desiccant wheel adsorbs moisture from the process air, this air exits the wheel hot and dry, due to the desiccant's heat of adsorption. For an effective cooling system, all or most of this heat must be rejected. This is typically accomplished 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 before it enters the conditioned space [12].

On the regeneration side, air is first reduced in temperature by passing it through an evaporative cooler. This cooled air provides a heat sink for the air-to-air heat exchanger. The hottest air exiting the heat exchanger is used for regeneration of the desiccant wheel. The remainder, if any, is normally rejected outside. The regeneration heat source for the desiccant can be the condenser coil of a boiler system, a direct-fired burner or a waste-heat source. The hot air exiting the heat source is passed through the regeneration section of the desiccant wheel, and the moisture released by the desiccant is rejected to the flue [12].

Desiccant cooling systems can be operated either in a recirculation mode or a ventilation mode. In a recirculation mode, the process inlet air is the return air from the building and the regeneration inlet air is outdoor air. In a ventilation mode, the process inlet air is outdoor air and the regeneration inlet air is either outdoor air (standard vent cycle) or building exhaust air (Pennington cycle) [12]. The state points of a typical desiccant cooling process operating in a recirculation mode are shown in Fig. 2.3. If the

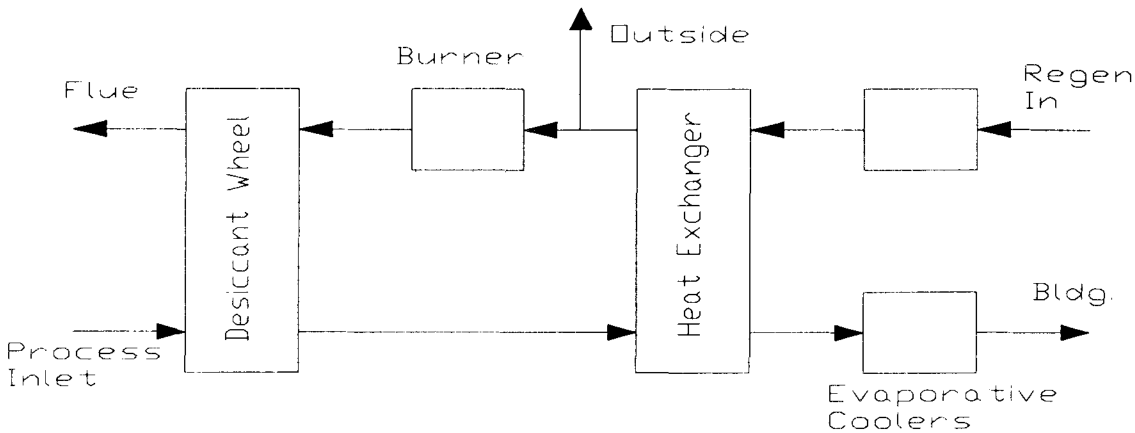


Fig. 2.2 Desiccant Cooling System

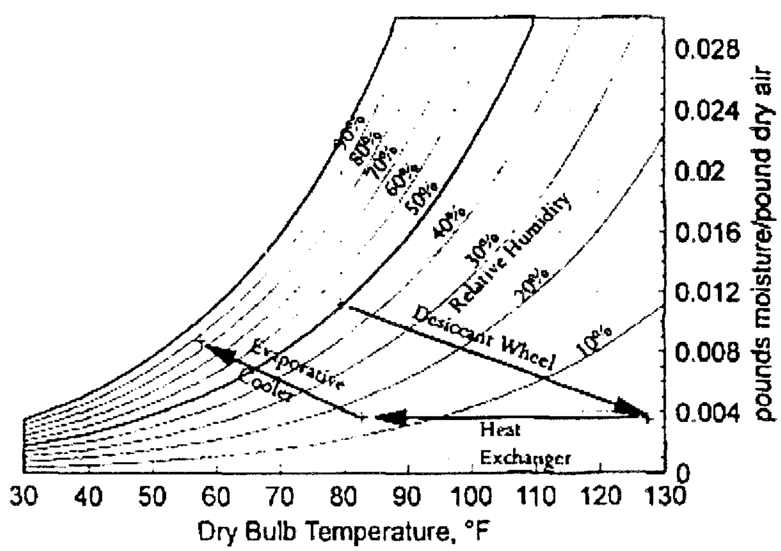


Fig. 2.3 Psychrometric Chart

system operates effectively, air entering the cooling system at ARI indoor condition, 80° F/ 50% relative humidity (RH), can be delivered to a building mostly saturated at 56-58° F. Alternatively, the degree of saturation achieved in the evaporative cooler can be reduced and the air will enter the building slightly warmer but lower in humidity.

Desiccant wheels used for cooling applications are designed with channels which provide laminar flow, maintaining the heat and mass transfer area as high as possible with a minimal pressure drop. Process and regeneration sections of the wheel are sealed to minimize cross-leakage. To achieve a small wheel size and optimum wheel efficiency, the following characteristics are desirable in the wheel [12]:

1. High surface area to volume ratio for fast heat and mass transfer.
2. High-temperature regeneration to minimize the size of the regeneration section.
3. Desiccant having a Type 1 M isotherm shape to enhance the containment of moisture wavefronts on adsorption and the temperature wavefront on regeneration
4. High desiccant/substrate loading ratio.
5. Desiccant having a low heat of adsorption or a high percentage of its capacity within a low temperature range.
6. Desiccant able to withstand direct-fired regeneration to eliminate losses in efficiency from a boiler
7. Desiccant properties which are stable over the projected wheel life

Characteristics listed in 6 and 7 above relate to the adsorption stability of the desiccant with time and exposure to combustion products. The performances obtained from a desiccant cooling system are obviously directly related to the intrinsic properties of the desiccant.

CHAPTER III

ANALYSIS OF THE COOLING SYSTEMS

3.1. Domain of the systems

The area of analysis is the main sales area of the Giant Eagle Store in Canfield, OHIO. This sales area contains open-faced multi-deck meat, multi-deck dairy, produce and frozen food cases. There are two types of open, refrigerated, display cabinet: one with a condenser at the bottom of the cabinet and the other with a remote condenser, outside the conditioned space. In the former all the power used by the compressors is an extra load on the room and there is no benefit from the heat absorbed by the frozen food in the cabinets themselves. Stores with this type of display cabinet do not suffer from the underheating sometimes experienced by those having the other type. The second type of open refrigerated cabinet, which is a subject of this study, has a big impact on the air-conditioning load. Heat transferred to the cabinets comes from the conditioned space and is therefore reducing the sensible heat gain because it is ultimately rejected at the remote condensers. This effect, plus the latent cooling also 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.

Refrigerated cabinets with remote condensers remove heat from the sales area over 24 hours of the day and 365 days of the year. Therefore, regardless of the room temperature, underheating is sometimes a difficulty at unexpected times.

Fig. 3.1, which was prepared using 'Encore 2100' software, shows a picture of the area concerned, the positions of the cabinets, the frozen food cases, and the temperatures that must be maintained inside them.

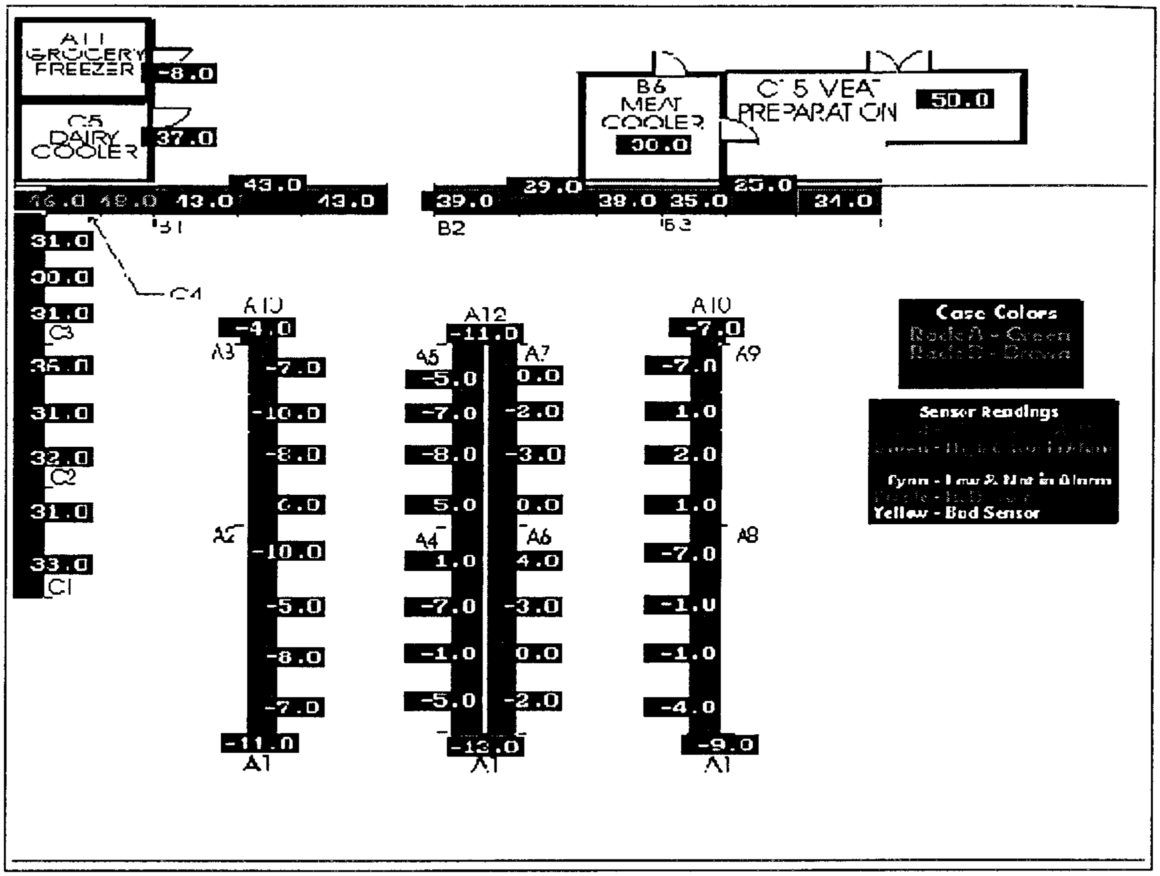


Fig. 3.1 Temperature Distrubution in the left side of the store

3.2. Computer Simulations

3.2.1. Desiccant Unit Selection

This Desiccant Unit Selection software is designed to choose the suitable desiccant unit required for any system. The desiccant units are designed from model DC 020, DC 030, to DC 130 and each unit is chosen according to the amount of air entering to the desiccant wheel. This software is not a tool for estimating cooling loads, so the engineer must use another software to find the loads for the building and, then, use the results as an input to the Desiccant Software.

Input data include the ambient and return air and the design conditions of the zone. Engineers can choose the components of the system to achieve the required results.

The Desiccant System is composed of these components:

1. Desiccant rotor
2. Heat exchanger rotor
3. Hydronic heater/hydronic loop
4. Regeneration heating coil
5. Process heating coil (optional)
6. Electrical control system
7. Process air blower
8. Regeneration air blower
9. Evaporative pad (optional)
10. Pre-cooling coil (optional)
11. Post-cooling coil (optional)

3.2.2. Hourly Analysis Program (HAP)

HAP is a computer tool that assists engineers in designing HVAC systems for commercial buildings. It is a tool for estimating loads and designing systems. HAP uses the ASHRAE-endorsed transfer function method for load calculations.

- **HAP System Design Features**

HAP 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. Specifically, the program performs the following tasks:

1. Calculates design cooling and heating loads for spaces, zones, and coils in the HVAC system.
2. Determines required airflow rates for spaces, zones and the system.
3. Sizes cooling and heating coils.
4. Sizes air circulation fans.

- **Using Hap to design systems and plants**

This section briefly describes how to use the HAP to design systems and plants in conceptual terms. All design work requires the same general five-step procedure:

1. Define the Problem; the scope and objectives of the design analysis are defined.
2. Gather Data; before design calculations can be performed, information about the building, its environment and its HVAC equipment must be gathered. This step involves extracting data from building plans, evaluating building usage and studying HVAC system needs.
3. Enter Data into HAP; data for climate, building and HVAC equipment must be

entered. When using HAP, the main program window allows easy access to the operation.

4. Use HAP to Generate Design Reports. Once weather, space, air system and plant data has been entered, HAP can be used to generate system and plant design reports.

3.2.3. Encore 2100

The Encore 2100 is able to monitor and control all HVAC and utility functions in the largest stores.

The Encore 2100 will:

- Monitor and control 256 input and 256 output devices
- Call personally by phone when an alarm occurs, even when the store is closed
- Provide historical information on system performance, using graphs and data

When the system is turned on, the main screen will appear. The entire program will be run from this screen by use of the function keys in the menu at the bottom of each screen. The main screen represents the basic information from which the Encore 2100 branches to provide more details as the user accesses other windows.

3.3. Systems Analysis

The design conditions for this analysis are selected from CARRIER CODES and they are shown in Table 3.1. Refrigerant R-12 is used as the working fluid in the direct expansion cooling coils and the system operates on an ideal vapor-compression refrigeration cycle. And the refrigerant enters the compressor as a saturated vapor and the vaporization pressure is 20 psi.

3.3.1 Desiccant Cooling System

In designing the desiccant cooling system, data gathered from the building were used as input. These data are shown in Table 3.2.

In order to get the best design using desiccant units, some of the parameters which affect the performance of desiccant dehumidification were considered in the analysis and these parameters are [2]:

1. Process moisture
2. Process air temperature
3. Reactivation air temperature
4. Reactivation air moisture
5. Amount of desiccant presented to the reactivation and process air streams

Since the model of desiccant wheel is chosen according to the amount of air entering to it [14], DC080 desiccant model is appropriate for this case, in which the process airflow is 7830 cfm.

Table 3.1 Design conditions- 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 effecting the Sales Area system

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, within the design area, shall be considered in load calculation, unless directed not to do so.

   d) Refrigeration case credits

      1) Case credits shall be used to in load calculations, unless directed not to do so

      2) Credits shall be 50% of total case load for open faced multi-deck meat, multi-deck dairy, produce, and frozen food cases. Of this 50%, 85% shall be sensible credits and 15% shall be latent credits

Table 3.2 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 were implemented to the ' Desiccant Unit Selection' software, and the process airflow leaves the desiccant wheel hot and dry, therefore an additional cooling will be 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. Therefore, indirect evaporative cooling coil can be used in this case. Consequently, in order to meet the load requirements of the conditioned space, the system must be composed of: desiccant rotor, heat exchanger rotor, boiler, regeneration heating coil, indirect evaporative cooler, process air blower, and regeneration air blower. Fig. 3.2 shows the desiccant unit selected and Table 3.3 shows the system data values for this design.

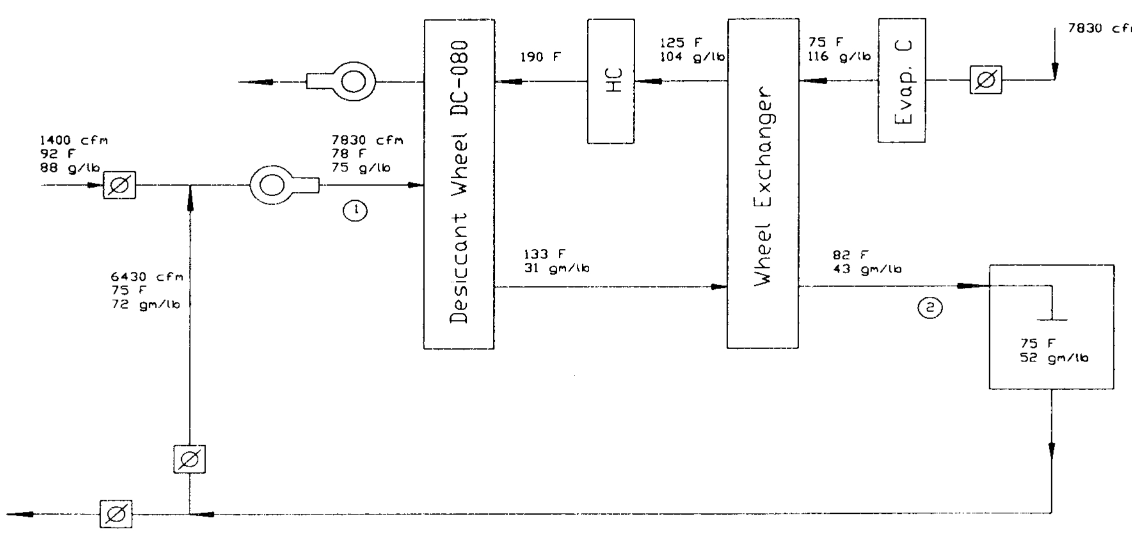


Fig. 3.2 Desiccant Cooling Unit, OH

Table 3.3 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

3.3.2. Coefficient of performance for the desiccant cooling system

The term coefficient of performance (COP) has been devised to measure the effectiveness of refrigerating machines and is usually defined as the ratio of the refrigeration produced to the network supplied. Usually the COP is used to analyze and compare cooling systems for their thermal performance. This definition of COP is applied mainly for conventional refrigerating machines. However, a definition provided by C. M. Shen and W. M. Worek [13] was used to define the COP for the all-cooling system including the system using desiccant, conventional or mixed systems.

COP for the desiccant system is defined as [13],

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

$$H_{out} = m_{a1}^{\cdot} h_1 \qquad \text{(Btu/hr)} \tag{3.2}$$

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

h_1 = enthalpy of mixed air

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

$m_{a2}^{\cdot}$ = 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 equals 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 [3]. 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, $m_a^{\cdot}$, known and the specific volume of the air at the point, we may calculate the volume flow rate in cubic feet per minute from

$$cfm = \frac{m_a^{\cdot} v}{60} \tag{3.5}$$

For uniformity in the manufacturing industry 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 [3]. Using the density, equal to $1/v_1$ in Eq. (3.5) and if we substitute the value of $m_a^{\cdot}$, we have

$$q_s^{\cdot} = (1.10cfm)(t_2 - t_1) \tag{3.6}$$

The power input to the compressor is calculated from:

$$W_{in} = m_r^{\cdot}(h_4 - h_3) \quad \text{(Btu/hr)} \tag{3.7}$$

$m_r^{\cdot}$ = mass flow rate of the refrigerant (lb/hr)

h_3, h_4 = enthalpy of the refrigerant at conditions 3 and 4 (Btu/lbm)

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 3.4 and also to graph the energy required to run the desiccant system as shown in Fig. 3.3. This energy is composed mainly of thermal energy required to run the heating coils and electrical energy to run the compressors and fans. Graphing the energy required to run the systems in this or in the following cases gives an idea about the operating costs to run the systems. However, detailed cost analysis is not included in this study.

Table 3.4 COP for Desiccant, OH

Energy Required to Run the Compressors

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

	Air Flow Rate (cfm)	Specific Volume (ft^3/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

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

Energy Required to Run the Fans

	Hp	Btu/hr
Desiccant Unit	2*10	50890

The Coefficient Of Performance

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

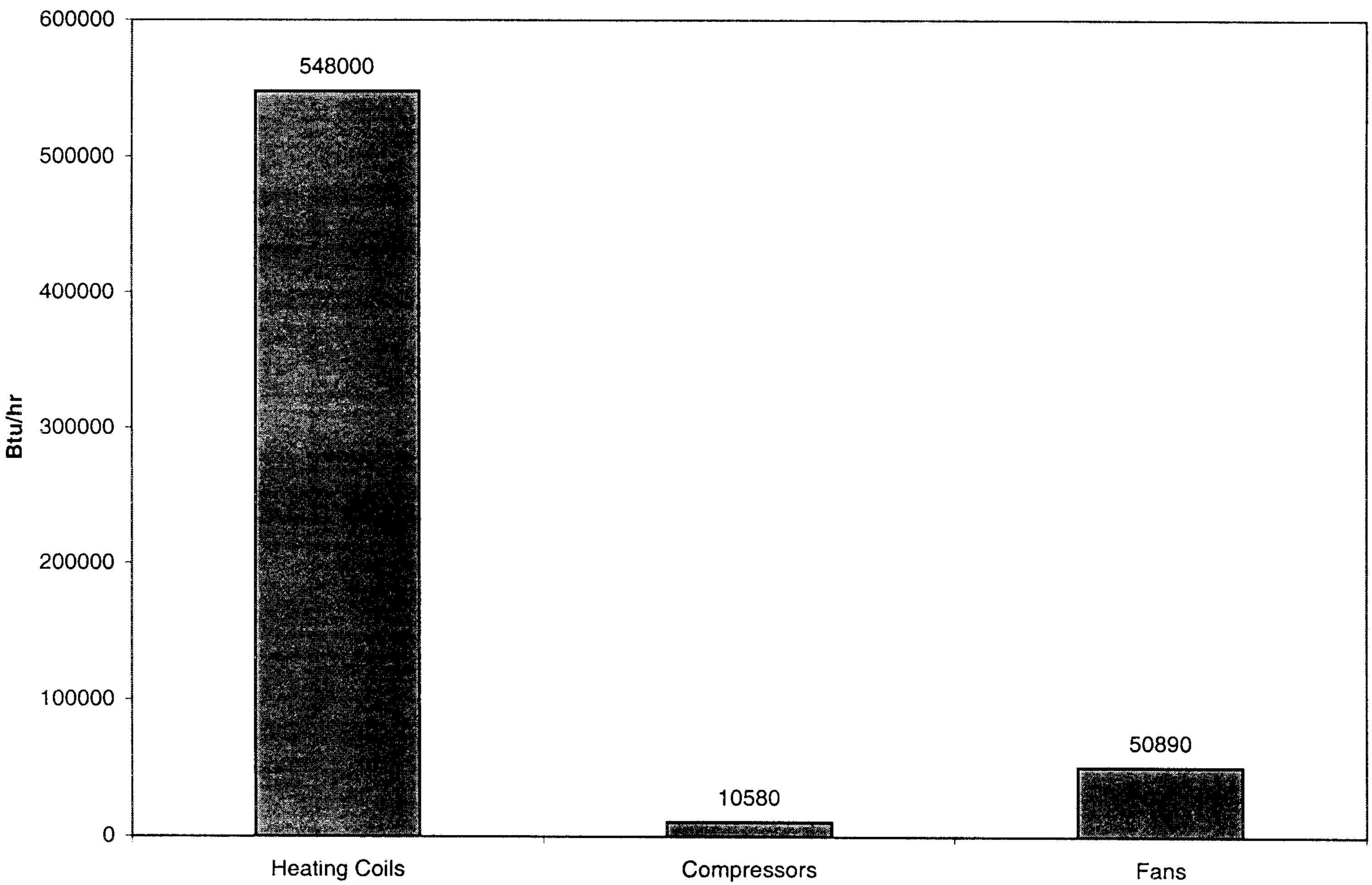


Fig. 3.3 Energy required to run the Desiccant System, OH

3.3.3. Conventional Cooling System

Most people are familiar with the principle of condensation. When air is chilled below its dewpoint temperature, moisture condenses on the nearest surface. 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 lower the temperature, the drier the air. The moisture removed from the air during any dehumidification process may be determined directly from the difference in specific humidities for the actual entering and leaving states, while the latent heat removal associated with this moisture removal can be calculated from the following equation [4]:

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

q_l = latent heat removal (Btu/hr)

ΔW = moisture removal (gr/lb)

This is the operating principle behind most commercial and residential air conditioning systems. A refrigeration system cools air, drains away some of its moisture as condensate and sends the cooler, drier air back to the space. The system basically pumps the heat from the dehumidified air to a different airstream in another location, using the refrigerant gas to carry the heat.

The actual hardware that accomplishes cooling dehumidification is exceptionally diverse. Literally thousands of different combinations of compressors are in use throughout the world. But there are three basic equipment configurations of interest to designers of humidity control systems [2], which include:

- Direct expansion cooling
- Chilled expansion cooling

- Dehumidification- reheat systems

For the analysis, 'HAP' software was used for designing this system. Weather data, the space conditions and the properties of the building were chosen from ASHRAE based on the specifications and the sheets of the location and the properties of the building. Tables 3.5 and 3.6 show the design weather parameters and the space input data for the main sales area that were used during the design.

The cooling of buildings is actually made up of two processes: sensible cooling, which is lowering air temperature, and latent cooling which is removing water vapor from the air. 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 [14]. These cooling coils cool the air to levels between 43 and $45^{o}F$. Below that point, frost begins to form on parts of the coil, spreading slowly through coil as the airflow becomes restricted. The frost insulates the refrigerant from the air passing through the coil, which reduces heat transfer and the frost physically clogs the coil, reducing the airflow. Eventually the frost blocks the airflow all together and dehumidification ceases. So in this case, reheating the air must be performed in order to dehumidify it again to get the required humidity of the air concerened.

In order to overcome this frosting problem and to meet the load requirements of the conditioned space, a dual air system was used. Air is supplied to the two air streams at different conditions (usually one hot, the other cold) and mixed by proportioning dampers upstream in a plenum. The entire air quantity for absorbing the load is conditioned centrally and distributed by the main fan. Mixing may be performed at the

Table 3.5 Design Weather Parameters for Youngstown, OH

Design Weather Parameters & MSHGs

Project Name: kamal projectmat — 05/31/00
Prepared by: AETOS — 10:18 AM

Design Parameters

City Name	**Youngstown**
Location	**Ohio**
Latitude	**41.3** Deg
Longitude	**80.7** Deg
Elevation	**1184.0** ft
Summer Design Dry-Bulb	**88.0** °F
Summer Coincident Wet-Bulb	**72.4** °F
Summer Daily Range	**20.6** °F
Winter Design Dry-Bulb	**-1.0** °F
Winter Design Wet-Bulb	**-2.5** °F
Atmospheric Clearness Number	**1.00**
Average Ground Reflectance	**0.20**
Soil Conductivity	**0.800** BTU/hr/ft/F
Local Time Zone (GMT +/- N hours)	**5.0** hours
Consider Daylight Savings Time	**Yes**
Daylight Savings Begins	**April, 1**
Daylight Savings Ends	**October, 31**
Simulation Weather Data	□□□□□□□□□□□□□□□□□□□□□ (□□□□□)
Current Data is	**User Modified**
Design Clg Months	**January to December**

Design Day Maximum Solar Heat Gains

(The MSHG values are expressed in BTU/hr/ft²)

Month	N	NNE	NE	ENE	E	ESE	SE	SSE	S
January	18.9	18.9	18.9	72.9	149.1	198.1	238.7	251.8	254.0
February	23.4	23.4	50.0	122.8	186.2	231.7	247.9	246.4	243.1
March	28.4	28.4	90.9	166.9	215.6	237.5	236.7	219.1	210.3
April	33.4	66.6	140.2	188.5	222.1	224.9	205.4	175.3	159.6
May	36.8	97.8	163.6	204.6	217.7	209.5	177.1	138.2	118.7
June	47.5	109.1	171.0	206.3	214.4	200.3	163.9	121.5	101.7
July	37.8	99.6	161.7	198.4	215.3	204.0	173.2	134.1	115.5
August	35.1	68.1	135.1	182.0	215.0	215.7	198.3	169.2	154.4
September	29.4	29.4	90.8	154.4	204.9	224.0	228.1	212.9	203.3
October	24.1	24.1	43.1	119.7	179.9	220.0	239.8	239.5	234.4
November	19.1	19.1	19.1	76.1	142.7	200.5	232.0	245.3	248.8
December	16.9	16.9	16.9	52.9	130.3	182.4	228.3	246.4	251.3
Month	**SSW**	**SW**	**WSW**	**W**	**WNW**	**NW**	**NNW**	**HOR**	**Mult.**
January	251.1	238.5	201.4	148.9	68.6	18.9	18.9	125.3	1.00
February	245.7	245.9	231.4	182.6	126.3	45.2	23.4	173.3	1.00
March	219.3	236.9	237.2	215.8	166.7	91.9	28.4	217.4	1.00
April	173.8	204.7	222.1	223.1	191.0	136.5	70.8	247.7	1.00
May	136.9	177.2	207.0	220.9	201.5	162.9	102.3	262.4	1.00
June	120.9	164.1	199.3	216.3	203.7	171.1	112.3	265.4	1.00
July	134.0	173.2	203.7	215.6	197.7	161.6	100.2	259.6	1.00
August	168.4	197.8	214.0	215.2	184.5	132.7	69.9	243.5	1.00
September	212.2	226.0	228.6	199.5	156.7	90.7	29.4	209.6	1.00
October	239.9	240.7	218.4	181.4	115.3	48.8	24.1	169.1	1.00
November	247.7	232.6	199.1	142.0	77.3	19.1	19.1	123.9	1.00
December	247.1	226.9	186.8	126.7	59.0	16.9	16.9	104.8	1.00

Mult. = User-defined solar multiplier factor.

Table 3.6 Space Input Data for the Giant Eagle 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).

apparatus with only a single duct extending to the zone. To maintain conditions at all the times, the cooling conditioner and duct must be sized for the maximum cooling load that exists when no heating is required from the other duct, and vice versa. This means that for any system where there is a wide variation in the cooling and heating requirements among the spaces served, each conditioner and duct will have to be sized to carry possibly 75% or more of the maximum load. At peak summer load, both air streams can carry the cooling effect, provided the equipment is arranged to furnish it [4].

Fig. 3.4 shows the conventional unit selected and conditions of air

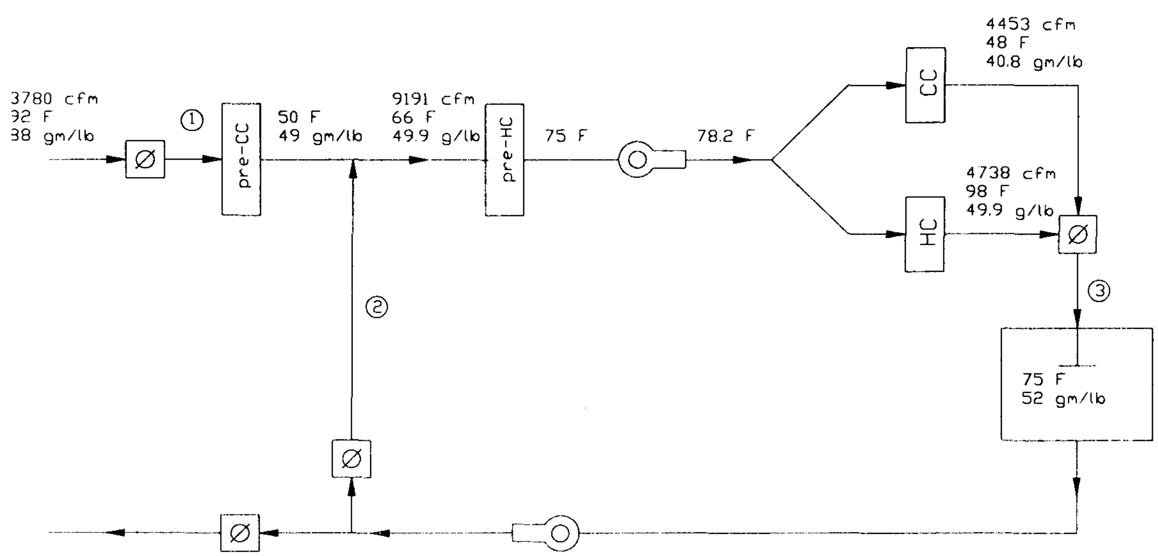


Fig. 3.4 Conventional Cooling Unit

In order to meet the load requirements of the conditioned space, the system as shown in Fig. 3.4 must be 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 that are shown in Fig. 3.4 were taken from Table 3.7 and the psychrometric analysis for the conventional cooling unit is shown in Fig.3.5.

Table 3.7 System Psychrometric for the Conventional Unit

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

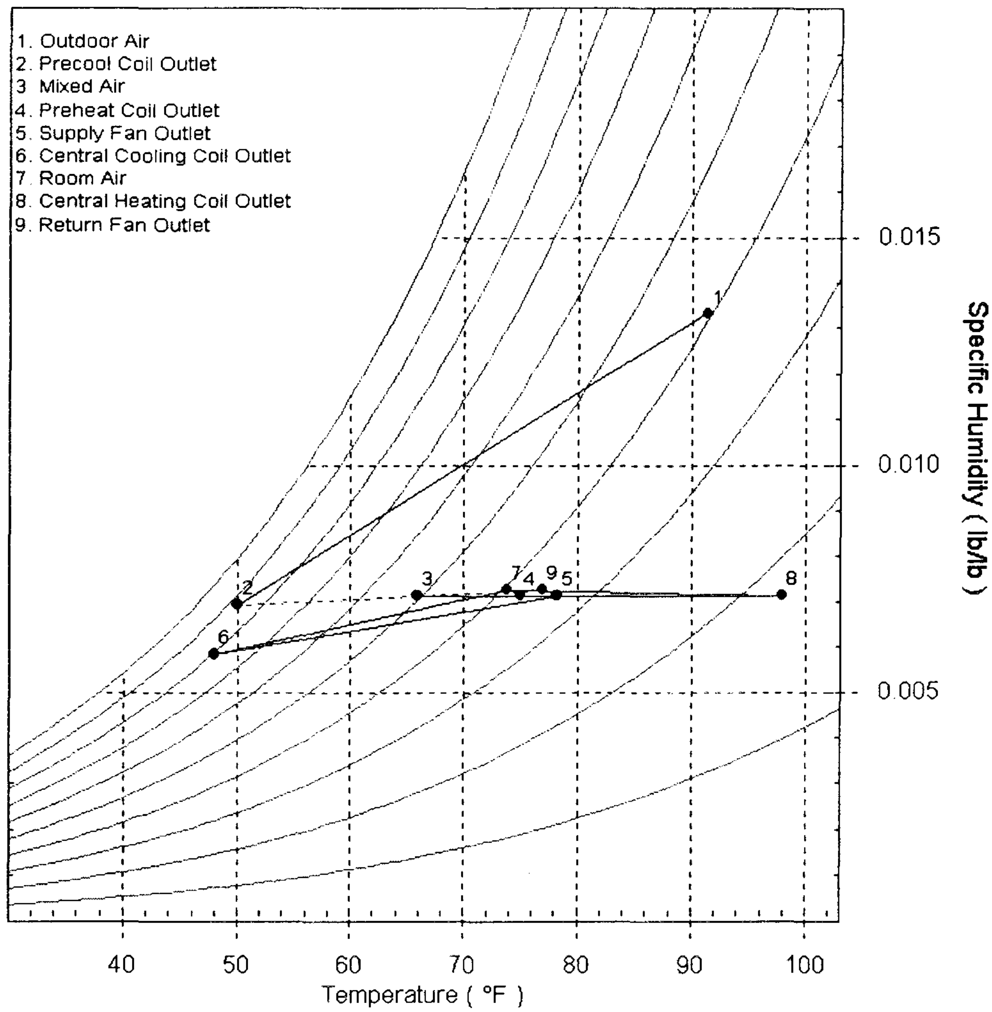
Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	15644	Deadband	0	73.8	9191	0	0

Fig. 3.5 Psychrometric Analysis for the Conventional Unit

Psychrometric Analysis for Packaged Rooftop AHU

Project Name: Giant Eagle project
Prepared by: AETOS

Location: Youngstown, Ohio
Altitude: 1184.0 ft.
Data for: July DESIGN COOLING DAY, 1700



3.3.4. Coefficient of performance for the conventional cooling system

COP for this system is defined as [13],

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

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

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

h_1, h_2 = enthalpy of ventilation and return air

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

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

h_3 = enthalpy of supply air

$w_{in1}^{\cdot}, w_{in4}^{\cdot}$ = power input to run 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. (3.4) 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 3.8 and also to graph the energy required to run the conventional system as shown is Fig. 3.6.

Table 3.8 COP for the Conventional Unit

Energy Required to Run the Compressors

	Total Coil Load (ton)	Refrigerant Flow Rate (Lb/hr)	Inlet Compressor Enthalpy (Btu/lb)	Inlet Evaporative Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Compressor Enthalpy (Btu/lb)	Working Input (Btu/hr)
Pre-Cool Coil	23	5200	76.4	23.3	82.5	87	55120
Cooling Coil	14.5	3000	76.4	18.4	55	84	22800

Conventional Unit	Air Flow Rate (cfm)	Specific Volume (ft^3/lb)	Enthalpy Of 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

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

Energy Required to Run the Fans

	Hp	Btu/hr
Conventional Unit	2*12	61068

The Coefficient Of Performance

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

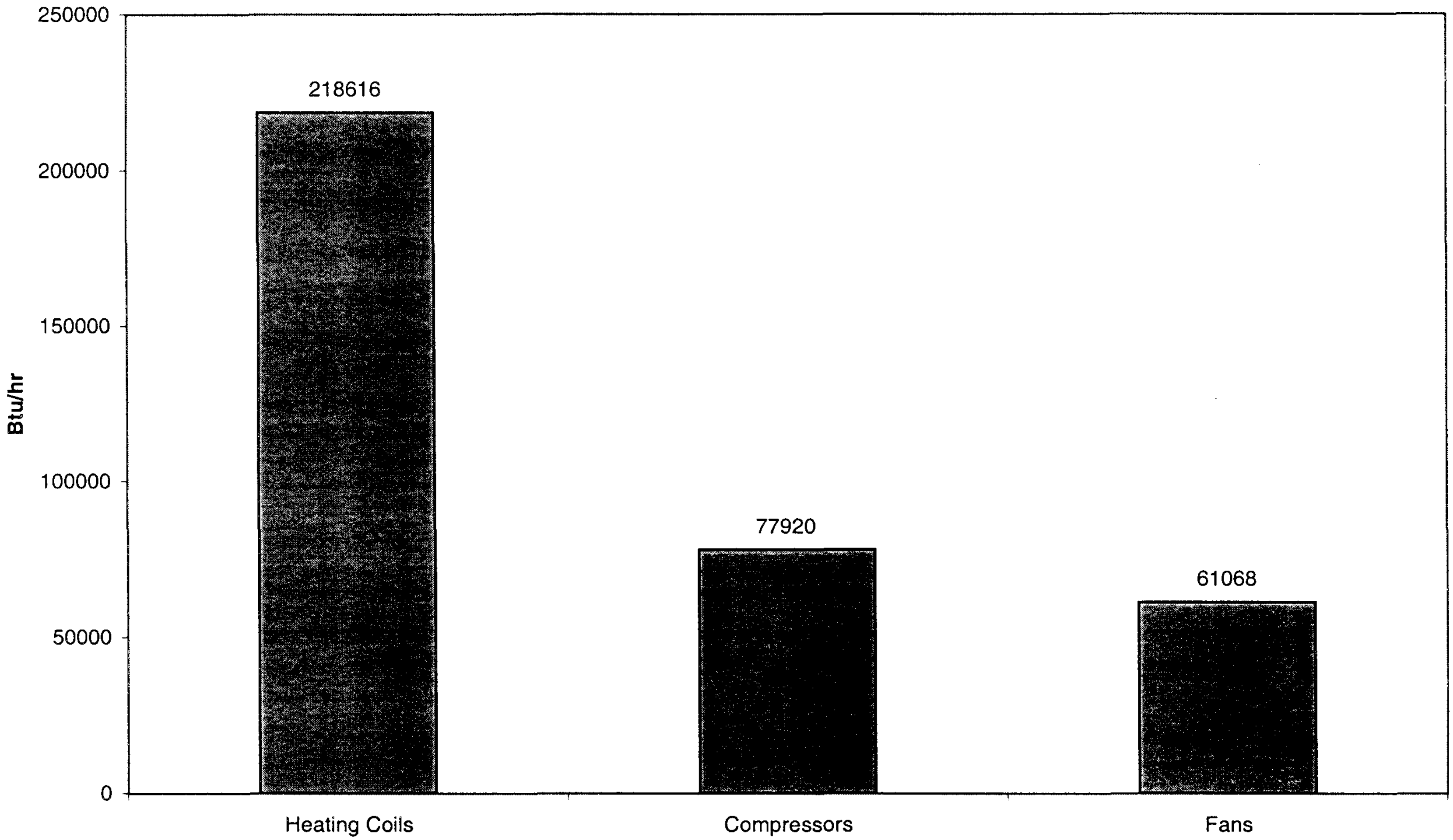


Fig. 3.6 Energy required to run the Conventional System

3.4. Comparison of the two systems:

The analysis described in the previous section showed that the values of COP for the desiccant and the conventional systems are 0.308 and 0.231 respectively. A spreadsheet was used to compare the energy required to run the two systems as shown in Fig. 3.7. This analysis confirms that using the desiccant system is better than using the conventional system, but that is not right for all cases of study or other different parts of the stores.

3.5. Recommendations for future choosing between Desiccant and Conventional units

1. Cooling and desiccant-based dehumidification systems are most economical when used together. The technologies complement each other. Each strength of desiccants covers a weakness of cooling systems and vice-versa.
2. 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. If thermal energy is cheap and power costs high, the economics will favor using desiccants to remove the bulk of the moisture. If power is not expensive and thermal energy for reactivation costly, the operating economics will favor using more cooling-based dehumidification in the project.
3. Cooling-based dehumidification systems are more economical than desiccants at high air temperatures and moisture levels. They are very seldom used to dry air below $40^{o} F$ dew point because condensate freezes on the coil, reducing moisture removal capacity.

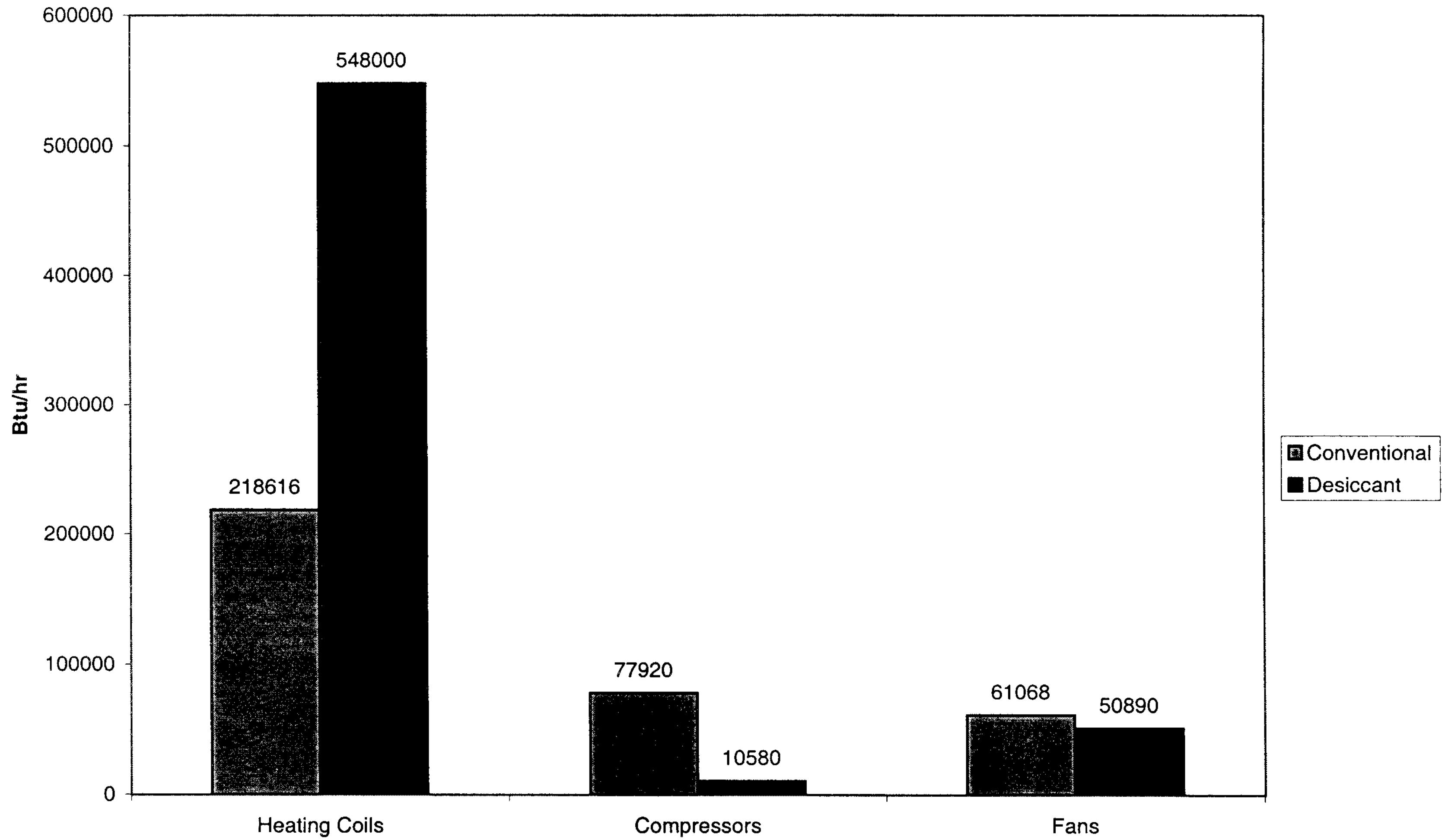


Fig. 3.7 Energy required to run the Desiccant and Conventional Systems, OH

4. Desiccants may have useful advantages when treating ventilation air for building HVAC systems with ice storage. Since these systems deliver air at moderately low dew points (40 to $45^{o}F$), dehumidifying the fresh air with the desiccant system decreases the installed cost of the cooling system, and eliminates deep coils with high air and liquid-side pressure drops. This saves considerable fan and pump energy.
5. Desiccants are especially efficient when drying air to create low relative humidities, and cooling-based dehumidification is very efficient when drying air to saturated air conditions. If the air should be drier than when it entered the machine, but still close to saturation at a lower temperature, cooling-based dehumidification would be a good choice. But if the desired end result is air at a condition far from saturation which means a low relative humidity, a desiccant unit would be ideal.

CHAPTER IV

SYSTEM DESIGN IMPROVEMENT

4.1. Additional Study on Desiccant Systems in Humid Areas

In an attempt to investigate the performance of a desiccant cooling system in humid areas and compare this system to the system considered earlier, additional study of desiccant systems in humid areas was performed. Changes in external parameters such as ambient temperature and humidity were made on a new system which combines desiccant and conventional units. Other design parameters such as building and space conditions were considered the same. The city of Tampa, Florida was selected for this case study DC 080 desiccant model is appropriate for this case, and the model processes the design air flow at 7830 cfm.

Since the air leaves the desiccant wheel drier but much warmer than the desired condition, an additional cooling is needed to meet the sensible and latent load requirements. The heat wheel provides part of the sensible and latent cooling. It meets the latent load but it does not provide sufficient cooling to meet the sensible load requirement of the conditioned space. For that purpose, a post cooling coil is used after the heat exchanger. The cooling coil is part of the conventional refrigeration unit.

The process airflow leaves the desiccant wheel hot and dry, therefore an additional cooling will be 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. Therefore, a post-cooling coil can be used in this case. Consequently, in order to meet the load requirements of the conditioned space, the system must be composed of: desiccant rotor, heat exchanger rotor, boiler, regeneration heating coil, process air blower, regeneration air blower and post cooling coil. A schematic diagram of the system with the design data is shown in Fig. 4.1.

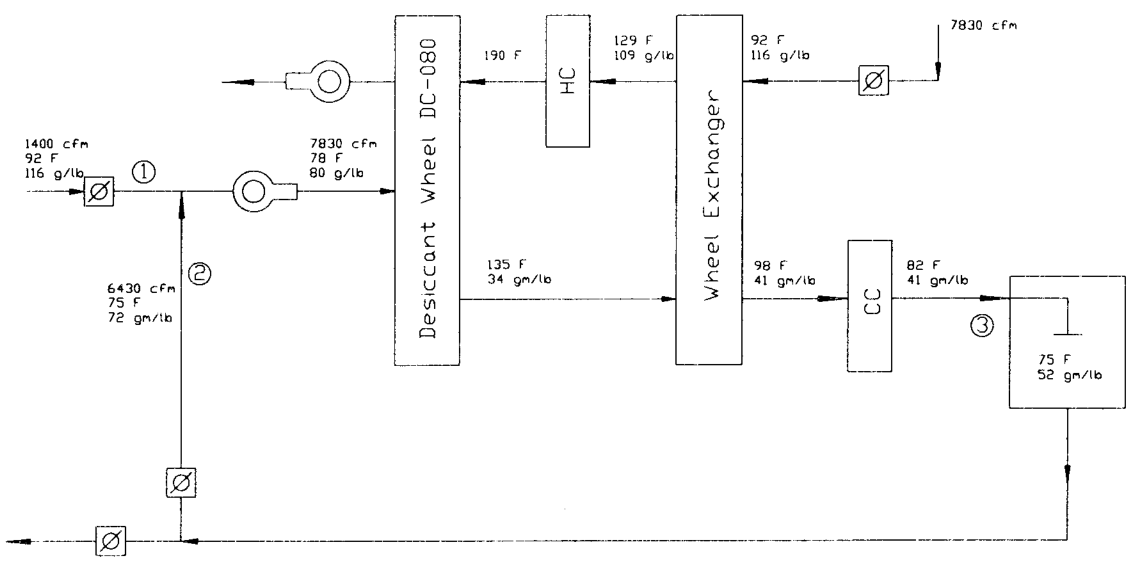


Fig. 4.1 Desiccant Cooling System, FL

Table 4.1 shows the system data values for this design.

Table 4.1 System Data Values for the Desiccant System, FL

Component	Dry Bulb Temp.	Specific Humidity	Airflow
Process Air	**FDB** (Degrees in Fahrenheit)	**GR/LB** (Grains/lb)	**CFM** (cubic feet/min)
Ventilation Air	92	116	1400
Vent-Return	78	80	7830
Desiccant Wheel (Mix)	135	34	7830
Wheel Exchanger	98	41	7830
Post-Cooling Coil	82	41	7830
Zone Air	75	52	7830
Regeneration Air			
Outside/ Ambient	92	116	7830
Wheel Exchanger	129	109	7830
Reg. Heating Coil	190	109	7830

The COP for this system is defined as [13],

$$COP = \frac{H_{out} - H_{in}}{w_1^{\cdot} + w_{in2}^{\cdot} + Q_3^{\cdot} + w_4^{\cdot}} \qquad (4.1)$$

$w_1^{\cdot}, w_4^{\cdot}$ *and* $Q_3^{\cdot}$ the same as defined in Eq. (3.1), (3.2) and (3.3).

$w_{in2}^{\cdot}$ = work input to run the compressor of the post-cooling coil

Based on Eqs. (3.4) to (3.8) and Eq.(4.1), the COP for this desiccant system is 0.3508. The calculation was done in Excel spreadsheet which is shown in Table 4.2.

Fig. 4.2. shows the energy required to run the entire system.

4.2. Comparsion between the desiccant units in Ohio and Florida

The value of COP for the desiccant system in Florida shows more than 10 percent higher than that of the similar system used in Ohio, which was 0.308. The analysis confirms the validity of the general assumption that the desiccant system performs better in more humid areas. The difference in the performance between the two systems is mainly attributed to more effective moisture removal capability of the desiccant and less use of the energy required to run the system when the system is used in more humid areas, as illustrated in Fig. 4.3.

Table 4.2 COP for Desiccant, FL

Energy Required to Run the Compressors

	Total Coil Load (ton)	Refrigerant Flow Rate (Lb/hr)	Inlet Compressor Enthalpy (Btu/lb)	Inlet Evaporative Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Compressor Enthalpy (Btu/lb)	Working Input (Btu/hr)
Cooling Coil	11.5	2178	76.4	13.0	36	80	7841

	Air Flow Rate (cfm)	Specific Volume (ft^3/lb)	Enthalpy Of Air (Btu/lb)	Energy Required (Btu/hr)
Desiccant Unit (FL)				
Mixed Air	7830	13.7	31.2	1069909
Supplied Air	7830	13.7	25.3	867587

Desiccant Unit (FL)	
Cooling Coil	11.5 tons
Heating Coil	518000 Btu/hr
Supply Fan	10 Hp
Return Fan	10 Hp

Energy Required to Run the Fans

	Hp	Btu/hr
Desiccant Unit (FL)	2*10	50890

The Coefficient of Performance

Mixed Air (Btu/hr)	Supply Air (Btu/hr)	Heating Coils (Btu/hr)	Compressors (Btu/hr)	Fans (Btu/hr)	COP
1069909	867587	518000	7841	50890	0.3508

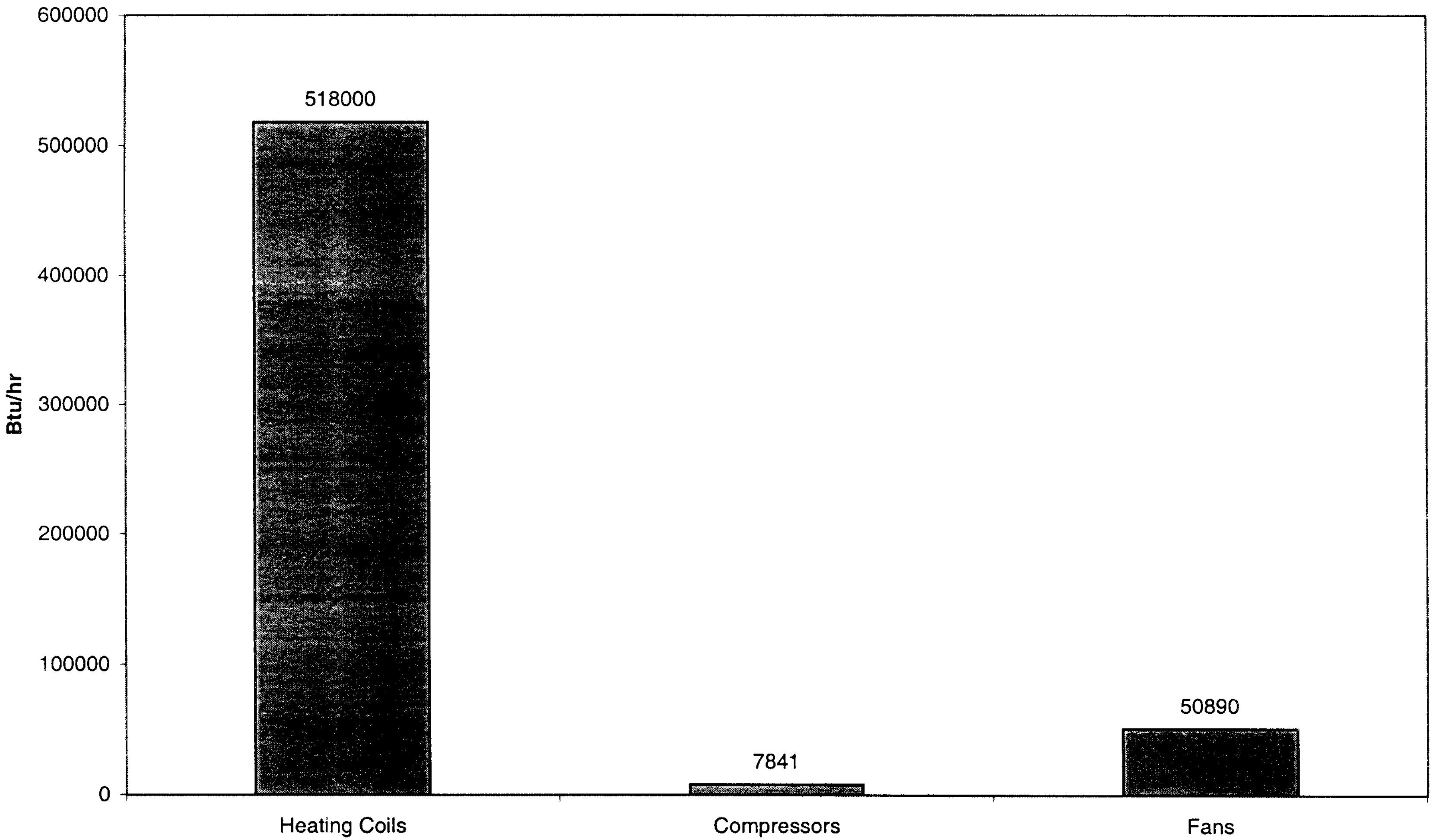


Fig. 4.2 Energy required to run the Desiccant System, FL

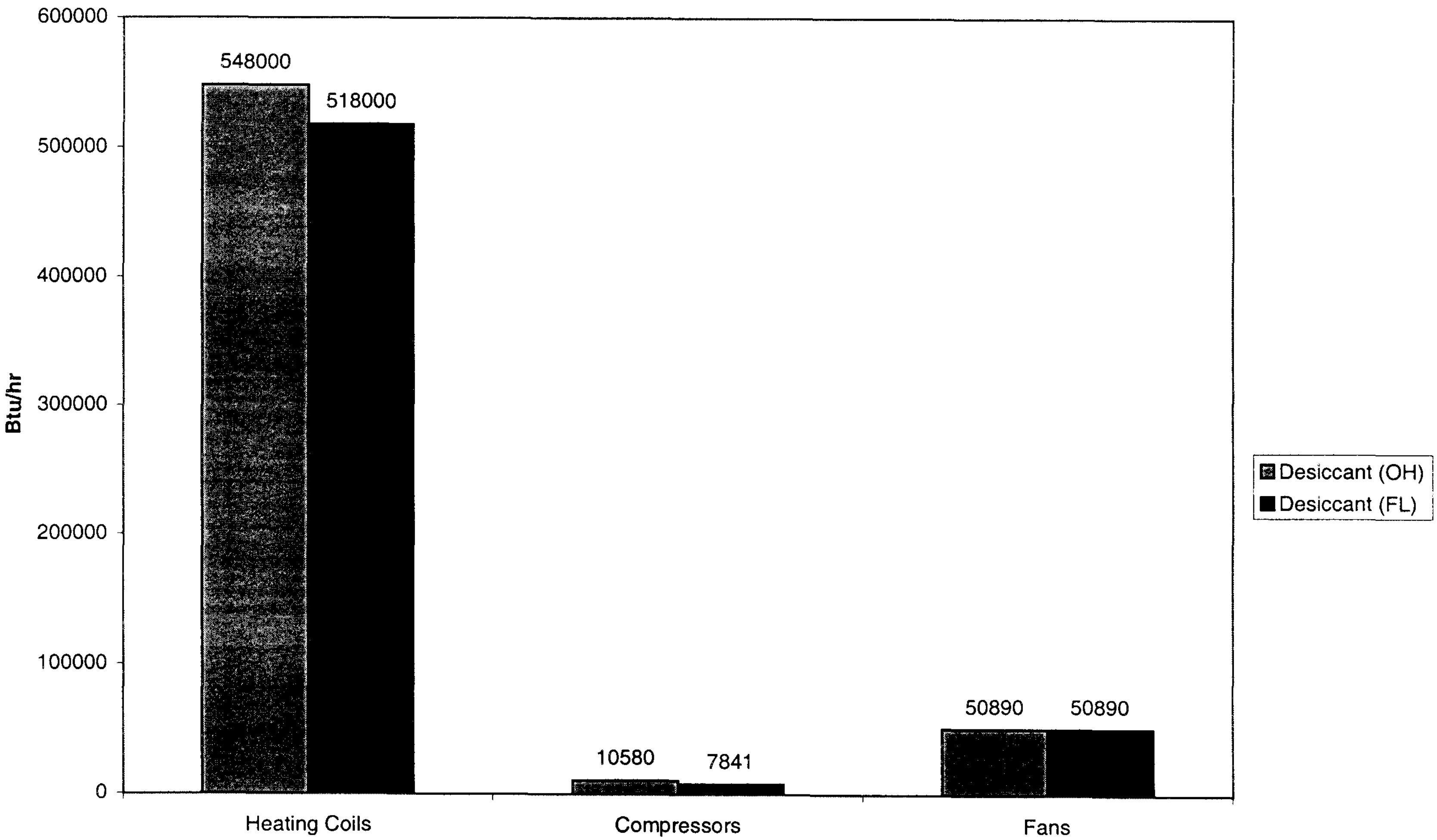


Fig. 4.3 Energy Required to run the Desiccant Systems in OH and FL

4.3. Parameters affecting System Performance

Three key parameters were considered for this design:

1. Air system design load
2. Process inlet moisture
3. Process inlet temperature

4.3.1. Air system design load

A large proportion of the latent load for the previous systems comes from the ventilation air as shown in Table 4.3. Therefore, it may give better results to dehumidify the ventilation air using the desiccant unit before the air blends with return air from the zone.

4.3.2. Process inlet moisture

The effect of changing the original moisture content of the air entering on the process side will be considered in this design. In other words, reduction of the moisture entering the desiccant wheel results in lower level of the moisture in the air leaving the process as shown in Fig.4.4.

4.3.3. Process air temperature

Vapor pressure on the surface of desiccant depends on the temperature of the material as well as on its water content as described in Fig. 2.1. Thus, it is not surprising that the desiccant performance is affected by the temperature of the incoming air. Lowering the process inlet temperatures improves moisture removal as shown in Fig. 4.5.

Table 4.3 Air System Design Load Summary

Air System Design Load Summary for Packaged Rooftop AHU

Project Name: Giant Eagle project
Prepared by: AETOS

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1700 COOLING OA DB / WB 91.4 °F / 72.2 °F			HEATING DATA AT DES HTG HEATING OA DB / WB -1.0 °F / -2.5 °F		
ZONE LOADS	**Details**	**Sensible (BTU/hr)**	**Latent (BTU/hr)**	**Details**	**Sensible (BTU/hr)**	**Latent (BTU/hr)**
Solar Loads	0 ft²	0	-	0 ft²	-	-
Wall Transmission	1582 ft²	2040	-	1582 ft²	9850	-
Roof Transmission	13608 ft²	36360	-	13608 ft²	39064	-
Glass Transmission	0 ft²	0	-	0 ft²	0	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Transmission	0 ft²	0	-	0 ft²	0	-
Floor Transmission	13608 ft²	0	-	13608 ft²	8333	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	34020 W	111333	-	0	0	-
Task Lighting	1955 W	6519	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	151	32667	30240	0	0	0
Infiltration	-	0	0	0	0	0
Miscellaneous	-	-174020	0	-	0	0
Safety Factor	5% / 5%	745	1512	15%	8587	0
>> Total Zone Loads	-	**15644**	**31752**	-	**65834**	**0**
Zone Conditioning	-	0	31752	-	0	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	9191 CFM	30533	-	9191 CFM	-30533	-
Ventilation Load	3780 CFM	56303	104130	3780 CFM	210347	0
Supply Fan Load	9191 CFM	30533	-	9191 CFM	-30533	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	**117370**	**135882**	-	**149281**	**0**
Central Cooling Coil	-	139188	26171	-	0	0
Central Heating Coil	-	-96995	-	-	0	-
Precool Coil	-	161834	109664	-	0	0
Preheat Coil	-	-86657	-	-	0	-
>> Total Conditioning	-	**117369**	**135834**	-	**0**	**0**
Key:	**Positive values are clg loads Negative values are htg loads**			**Positive values are htg loads Negative values are clg loads**		

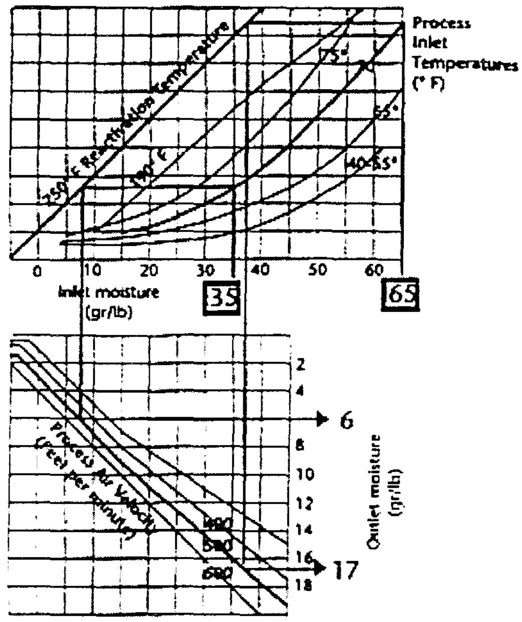


Fig. 4.4 Changing Process Air Moisture

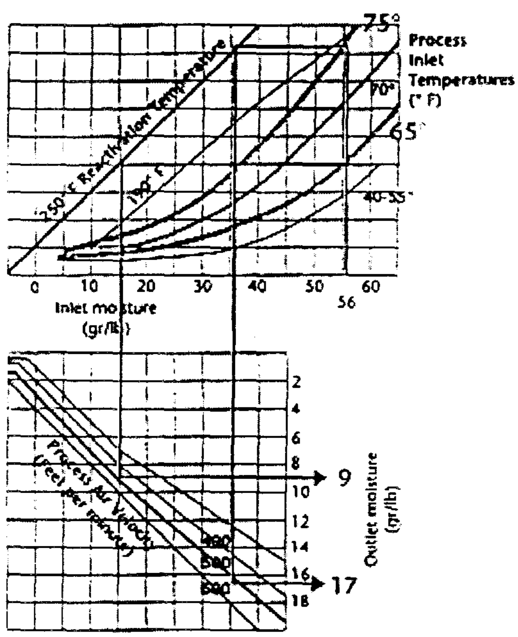


Fig. 4.5 Changing Process Air Temperature

The desiccant attracts more moisture since the desiccant becomes cooler and the surface vapor pressure also decreases.

Reduction of the inlet temperature and moisture can be achieved by adding a pre-cooling coil before the desiccant wheel.

This concept is used in a new alternative design to pre-cool the make-up air and then dehumidify it with the desiccant unit before the air blends with return air from the zone.

4.4. Alternative Designs for Improvement

4.4.1. Desiccant unit, OH

'Desiccant Unit Selection' and 'Hourly Analysis Program' were used for this design. Data of the make-up air were first implemented to the 'Desiccant Unit Selection', and, then the output of this software together with data of return air from the zone were used as input to the 'HAP'.

DC020 desiccant model was selected for this case based on the process air flow of 1400 cfm.

The make-up air enters the pre-cooling air where it is cooled and dehumidified, and leaves the desiccant wheel hot and dry. The air, then, enters the heat wheel that provides part of sensible load requirements. Subsequently, the air is blended with return air from the zone and passes through a cooling coil where it meets the latent load. Finally the air is heated sensibly in order to meet the space load requirements.

Fig. 4.6 shows the system selected and Table 4.4 shows the system data values for this design.

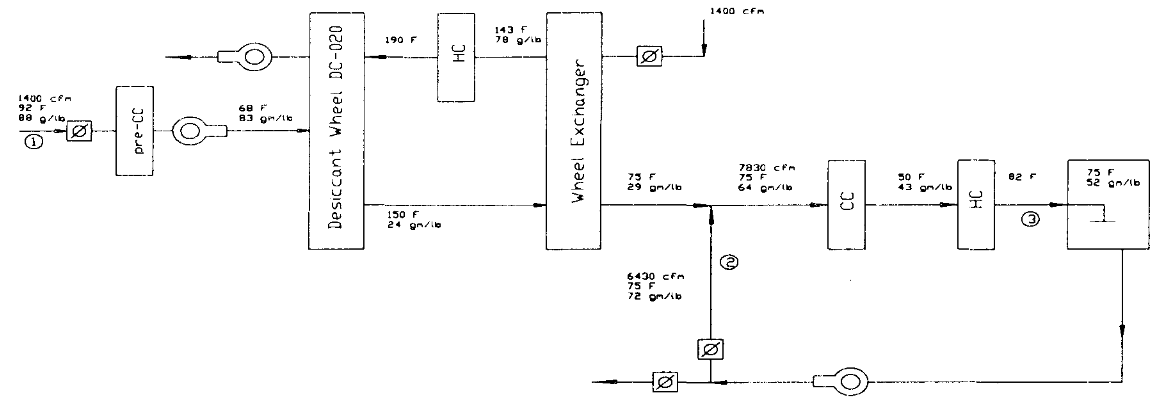


Fig. 4.6 Alternative System Design, OH

In order to meet the particular load requirements, the system must be composed of pre-cooling coil, desiccant rotor, heat exchanger rotor, boiler, regeneration heating coil, process air blower and regeneration air blower from the desiccant side. Also, it is composed of cooling coil, heating coil and a return air blower.

The COP for this system is defined as [13],

$$COP = \frac{H_{out} - H_{in}}{w_{in1}^{\cdot} + Q_2^{\cdot} + w_3^{\cdot} + w_{in4}^{\cdot} + Q_5^{\cdot}} \tag{4.2}$$

$$H_{out} = m_{a1}^{\cdot} h_1 + m_{a2}^{\cdot} h_2 \tag{4.3}$$

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

Table 4.4 System Data Values for the Design System, OH

Component	Dry Bulb Temp.	Specific Humidity	Airflow
Process Air	**FDB**	**GR/LB**	**CFM**
Ventilation Air	92	88	1400
Pre-Cooling Coil	68	83	1400
Desiccant Wheel	150	24	1400
Wheel Exchanger	75	29	1400
Mixing Air	75	64	7830
Cooling Coil	50	43	7830
Heating Coil	82	43	7830
Zone Air	75	52	7830
Regeneration Air			
Outside/ Ambient	92	88	1400
Wheel Exchanger	143	78	1400
Reg. Heating Coil	190	78	1400

h_1, h_2 = enthalpy of ventilation and return air

$$H_{in}^{\cdot} = m_{a3}^{\cdot} h_3 \tag{4.4}$$

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

h_3 = enthalpy of supply air

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

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

$w_3^{\cdot}$ = power input to run the fans

Based on Eqs. (3.4) to (3.8) and Eqs. (4.2) to (4.4), the COP for this system was found to be 0.302. The calculation was done on Excel spreadsheet which is shown in table 4.5.

Fig. 4.7. shows the energy required to run the system.

4.4.2. Desiccant unit, FL

For the desiccant unit, the same DC020 model was chosen based on the same air flow rate. New data were implemented using the same software used above.

Properties of the building and space data are the same as in Ohio, but weather data are different.

In order to meet the load requirements of the conditioned space, the system must be composed of the same elements as those in Ohio.

Fig. 4.8. shows the system selected and table 4.6. shows the system data values for this design.

Table 4.5 COP for the New Design System, OH

Energy Required to Run the Compressors

	Total Coil Load (ton)	Refrigerant Flow Rate (Lb/hr)	Inlet Compressor Enthalpy (Btu/lb)	Inlet Evaporative Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Compressor Enthalpy (Btu/lb)	Working Input (Btu/hr)
Pre-Cool Coil	3.5	704	76.4	18	54.8	82	3942
Cooling Coil	27	6270	76.4	24.8	89.4	87	66462

	Air Flow Rate (cfm)	Specific Volume (ft^3/lb)	Enthalpy Of Air (Btu/lb)	Energy Required (Btu/hr)
Ventilation Air	1400	14.15	35.9	213117
Returned Air	6430	13.7	29.3	825105
Supplied Air	7830	13.8	26.4	898748

Mixed Design Unit	
Pre-Cool Coil	3.5 tons
Cooling Coil	27 tons
Reg-Heating Coil	70000 Btu/hr
Heating Coil	270605 Btu/hr
Fans	20 Hp

Energy Required to Run the Fans

	Hp	Btu/hr
Mixed Design Unit	20	50890

The Coefficient Of Performance

Ventilation Air (Btu/hr)	Returned Air (Btu/hr)	Supply Air (Btu/hr)	Heating Coils (Btu/hr)	Compressors (Btu/hr)	Fans (Btu/hr)	COP
213117	825105	898748	340605	70404	50890	0.302

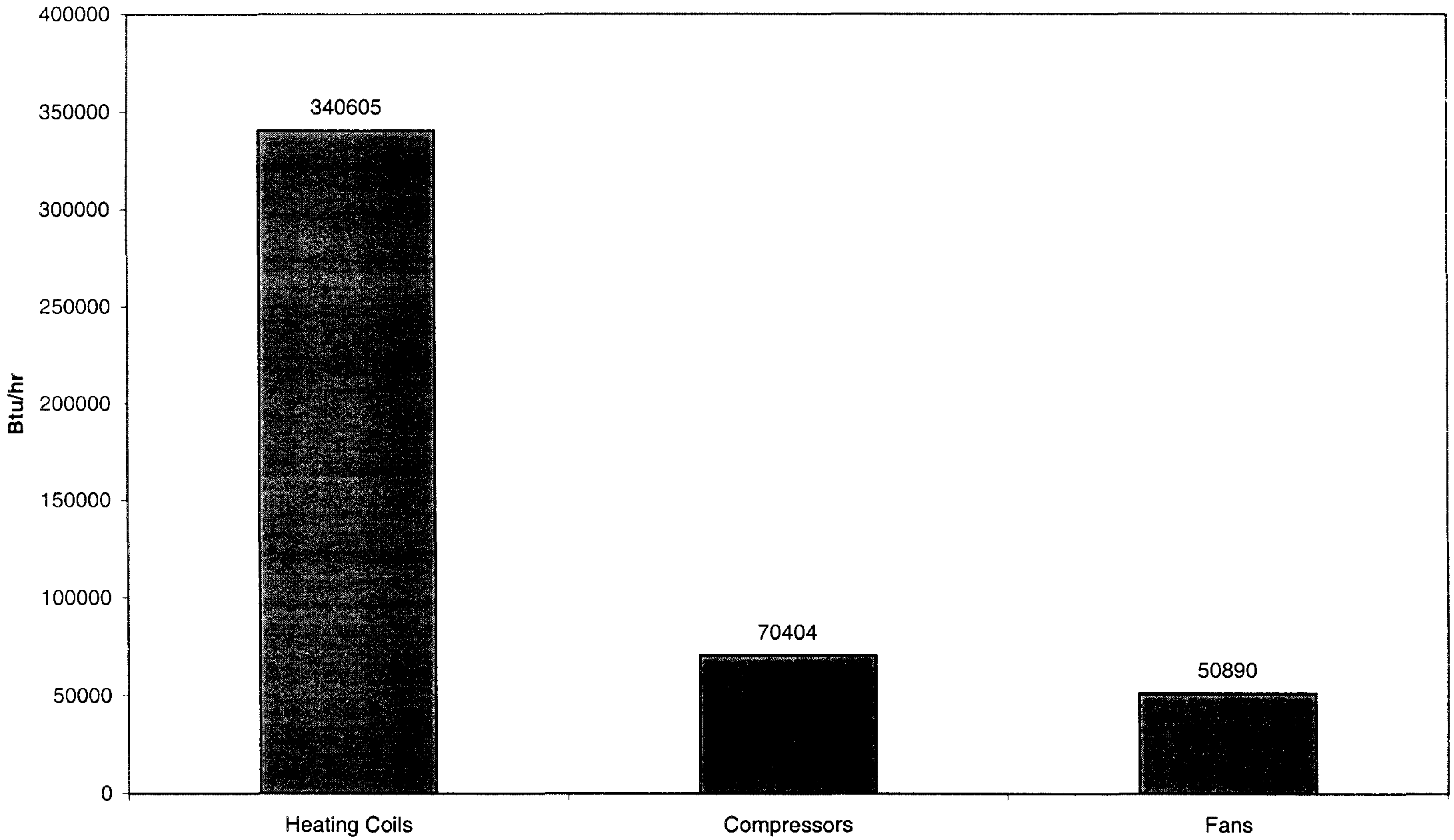


Fig4.7 Energy required to run the Alternative System, OH

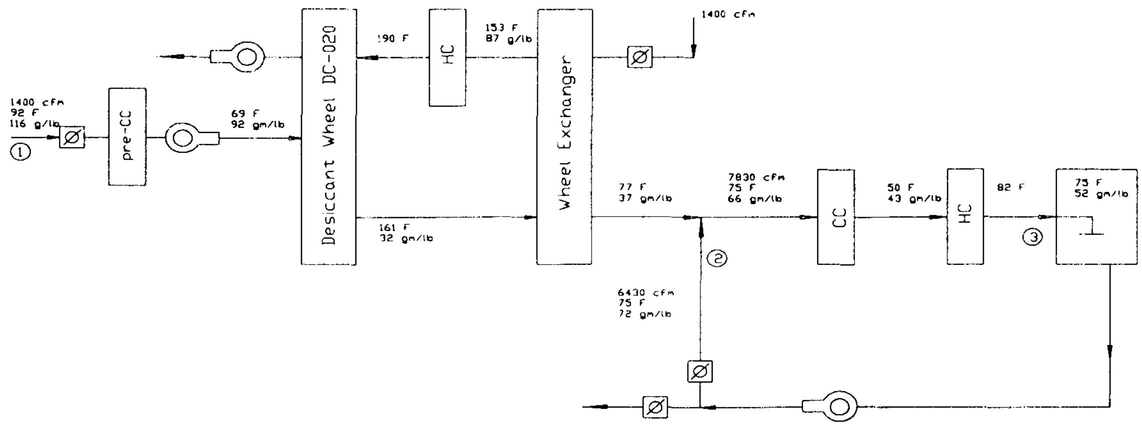


Fig. 4.8 Alternative System Design, FL

Based on Eqs. (3.4) to (3.8) and Eqs.(4.2) to (4.4), the COP for this system is 0.3573. The calculation was done on Excel spreadsheet which is shown in Table 4.7.

Fig. 4.9 shows the energy required to run the entire system.

Table 4.6 System Data Values for the Design System, FL

Component	Dry Bulb Temp.	Specific Humidity	Airflow
Process Air	**FDB**	**GR/LB**	**CFM**
Ventilation Air	92	116	1400
Pre-Cooling Coil	69	92	1400
Desiccant Wheel	161	32	1400
Wheel Exchanger	77	37	1400
Mixing Air	75	66	7830
Cooling Coil	50	43	7830
Heating Coil	82	43	7830
Zone Air	75	52	7830
Regeneration Air			
Outside/ Ambient	92	116	1400
Wheel Exchanger	153	87	1400
Reg. Heating Coil	190	87	1400

Table 4.7 COP for the New Design System, FL

Energy Required to Run the Compressors

	Total Coil Load (ton)	Refrigerant Flow Rate (Lb/hr)	Inlet Compressor Enthalpy (Btu/lb)	Inlet Evaporative Enthalpy (Btu/lb)	Condensation Pressure (psi)	Outlet Compressor Enthalpy (Btu/lb)	Working Input (Btu/hr)
Pre-Cool Coil	5	1029	76.4	18	54.8	82	5762
Cooling Coil	28	6504	76.4	25	90.6	88	75446

	Air Flow Rate (cfm)	Specific Volume (ft^3/lb)	Enthalpy Of Air (Btu/lb)	Energy Required (Btu/hr)
Ventilation Air	1400	14.25	40.3	237558
Returned Air	6430	13.7	29.3	825105
Supplied Air	7830	13.8	26.4	898748

Mixed Design unit (FL)	
Pre-Cool Coil	5 tons
Cooling Coil	28 tons
Reg-Heating Coil	56000 Btu/hr
Heating Coil	270605 Btu/hr
Fans	20 Hp

Energy Required to Run the Fans

FL	Hp	Btu/hr
Mixed Design Unit	20	50890

The Coefficient Of Performance

Ventilation Air (Btu/hr)	Returned Air (Btu/hr)	Supply Air (Btu/hr)	Heating Coils (Btu/hr)	Compressors (Btu/hr)	Fans (Btu/hr)	COP
237558	825105	898748	326605	81209	50890	0.3573

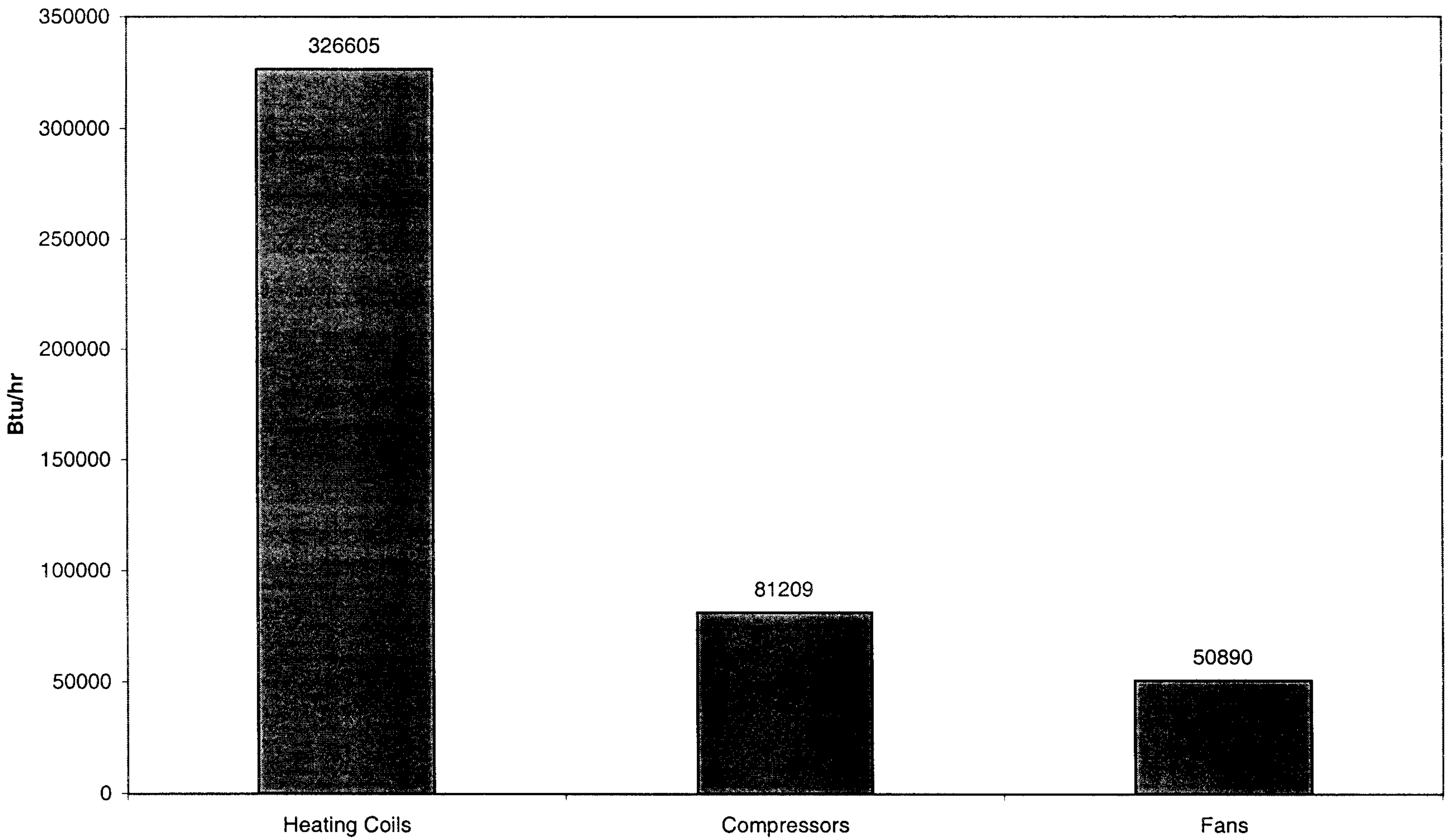


Fig. 4.9 Energy Required to run the Alternative System, FL

4.5. Comparison between the systems

4.5.1. The original desiccant system and the alternative design in Ohio

The value of COP using the new design system in OH is 0.302 and that value using the original desiccant system is 0.308. Therefore, the system will perform better using the desiccant system alone in these kinds of stores in OH.

The energy required to run the two systems is shown in Fig. 4.10 for easy comparison. This does not necessarily mean that the system using larger amount of energy costs more in the operation because the thermal energy is much less expensive than the electrical energy.

4.5.2. The original desiccant system and the alternative design in Florida

The value of COP using the new design is 0.3575 and is more than that using the desiccant unit alone which is 0.3508. Therefore, the new design system performs better than using the desiccant units in these kind of stores in humid areas.

Fig.4.11. shows the energy required to run the two systems

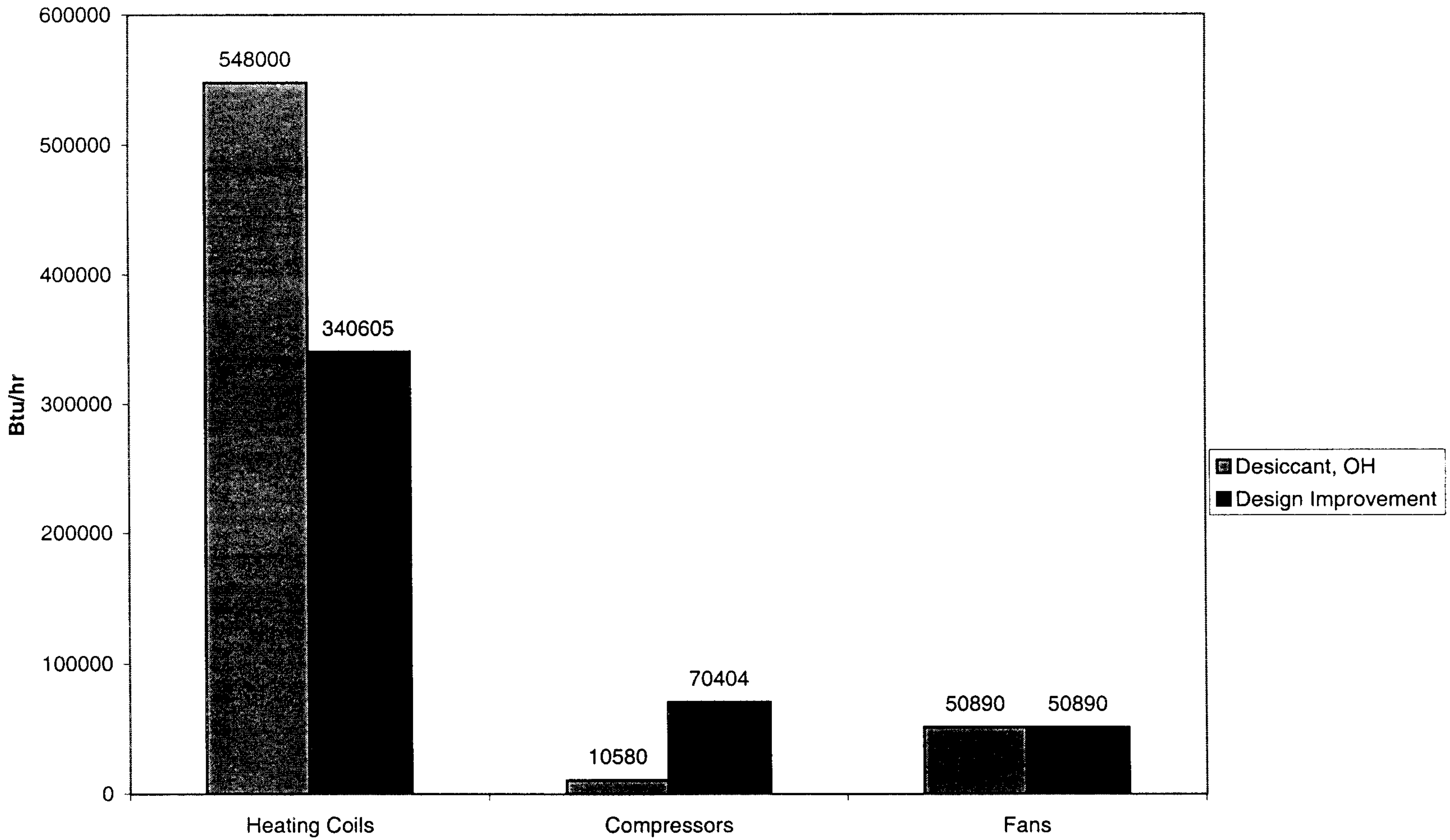


Fig.4.10 Energy Required to run Desiccant, OH and its Alternative Design

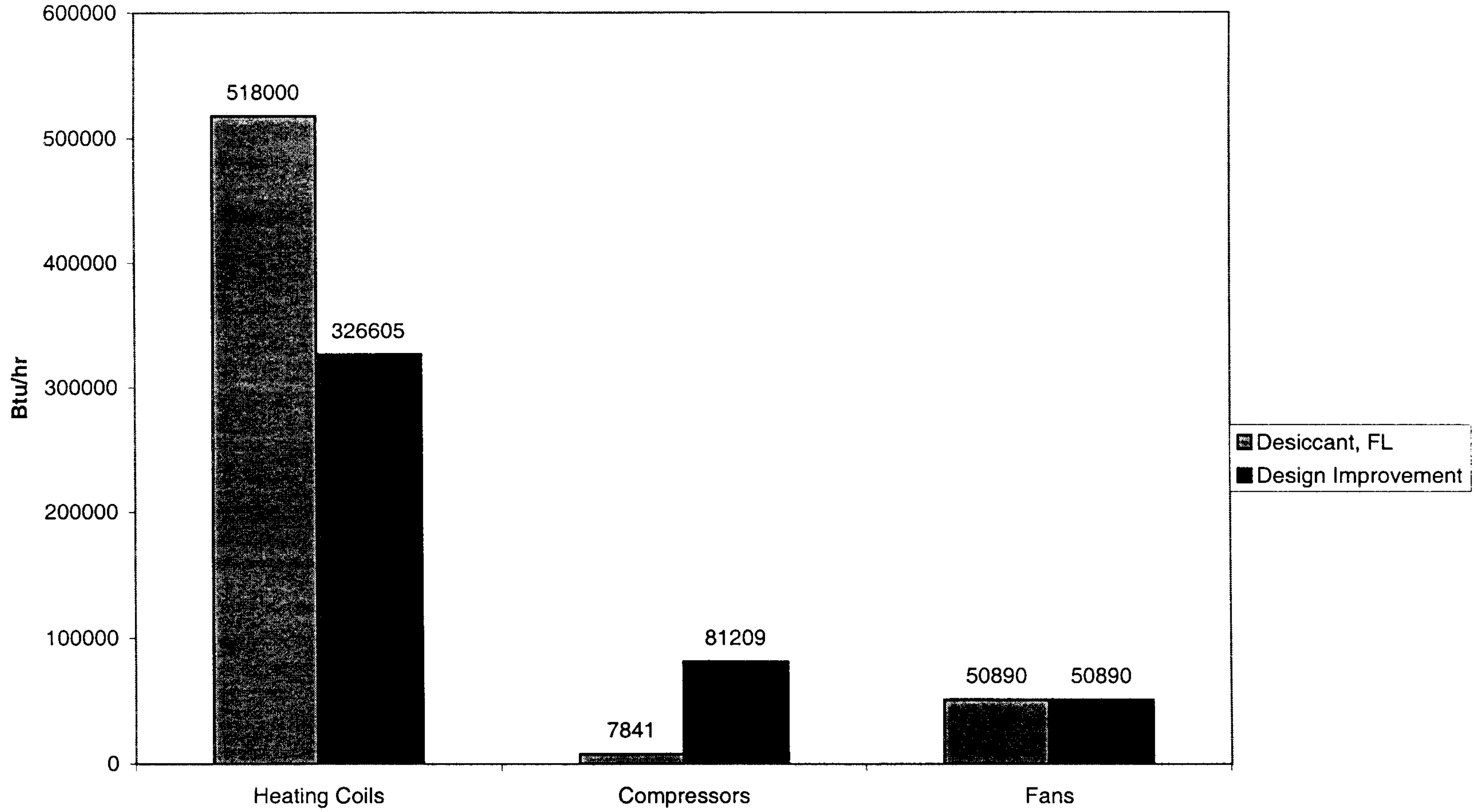


Fig. 4.11 Energy Required to run Desiccant, FL and its Alternative Design

CHAPTER V

RESULTS AND CONCLUSIONS

5.1. Results

The results of this analysis are summarized as follows:

1. The coefficients of performance for the system in OH, which is located in an area of 88 gr/lb specific humidity, is 0.308 using the desiccant unit and 0.231 using the conventional cooling unit, respectively.
2. The coefficient of performance of the desiccant unit, which is located in an area of 116 gr/lb in FL, is 0.351.
3. For the combined system of desiccant and conventional units, the coefficients of performance of the system in OH and FL are 0.302 and 0.357 respectively.

For a better illustration, the COPs are plotted in Fig.5.1.

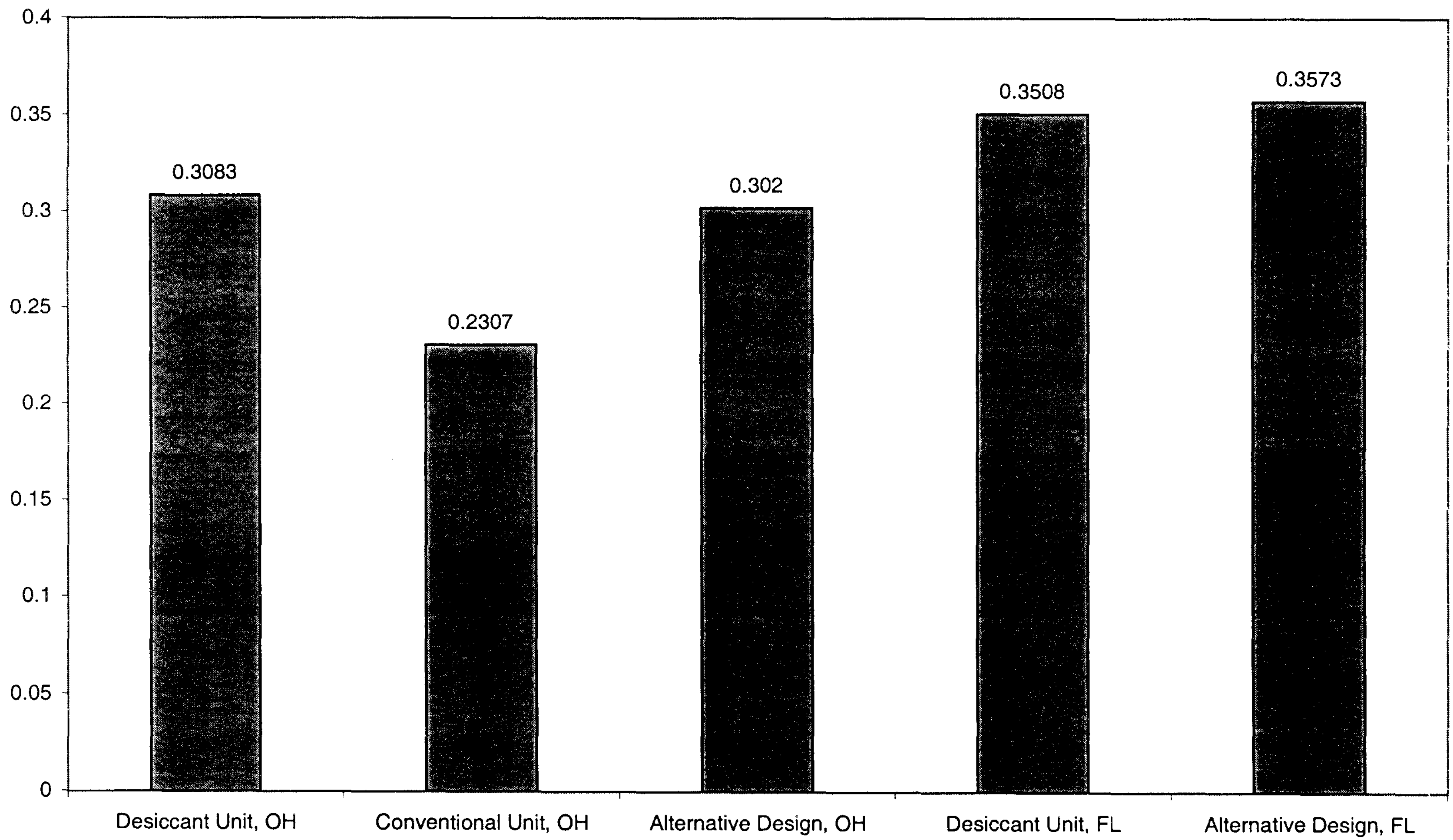


Fig.5.1 Coefficient of Performance

5.2. Conclusions

Evaluations made from the study draw general conclusions on the performance of the systems:

1. Desiccant units perform better than conventional units in an environment of large humidity such as supermarkets.
2. The desiccant units in humid areas perform better than those in less humid areas.
3. Pre-cooling the make-up air and dehumidifying it with the desiccant before the air blended with return air from the zone must be considered when:
   a) The latent load of the make-up air is much larger than that of the internal moisture load of the zone.
   b) The system requires a large proportion of make-up air.
4. Desiccants are especially efficient when drying air to create low relative humidities, and cooling-based dehumidification is very efficient when drying air to saturated air conditions. If the air passing through the desiccant process is still close to saturation point at a lower temperature, cooling-based dehumidification would be a good choice. But if the desired end result is the air at a condition far from saturation, which means a low relative humidities, a desiccant unit would be ideal.
5. Desiccant cooling units perform better than conventional cooling units when the latent load is large in comparison to the sensible load. The cooling coils often have low latent capacities, usually ranging 20% to 30%, which means higher sensible heat ratio (SHR). Therefore, this condition can create problems to the

cooling coil when the SHR of the load falls below 70%, since the coil will no longer have enough latent capacity to meet the latent load.

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APPENDIX

Fig. A.1 Picture of a Desiccant Cooling System

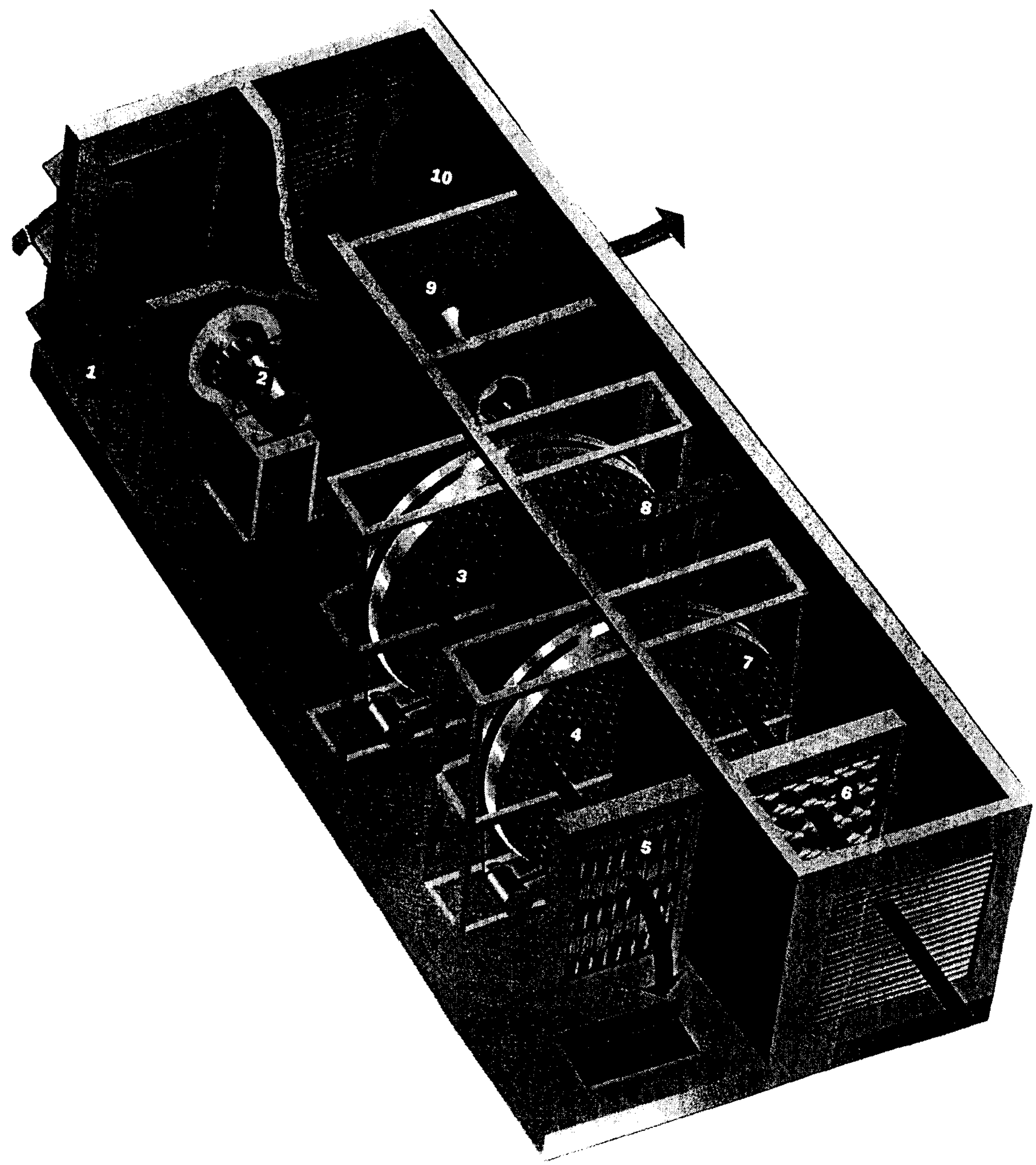

Fig. A.2 Process and Reactivation airflow temperature and humidity changes

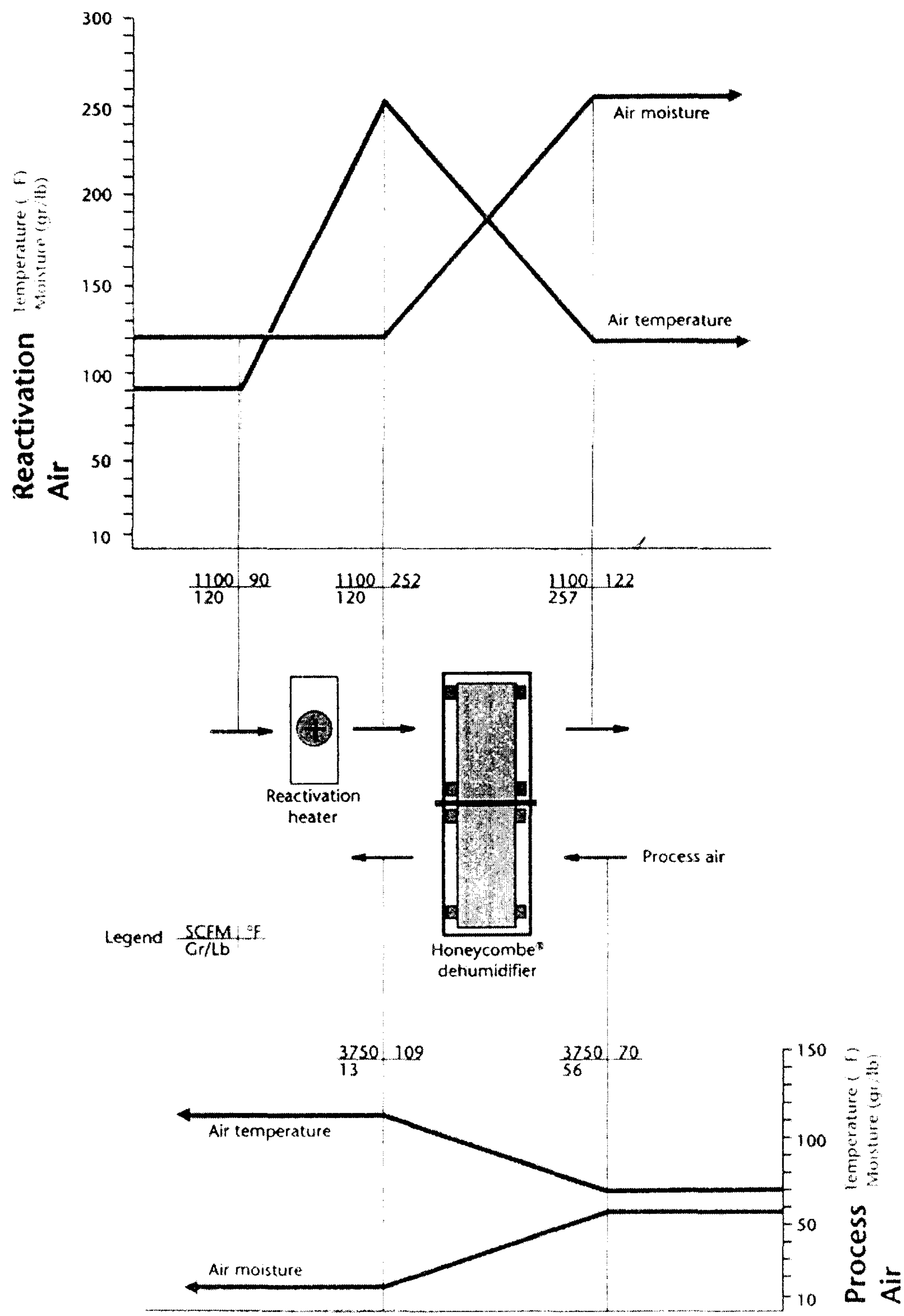

Table A.1 Cooling Design Temperature Profiles

Cooling Design Temperature Profiles

Project Name: Giant Eagle project
Prepared by: AETOS

Location: Youngstown, Ohio

(Dry and Wet Bulb temperatures are expressed in °F)

Hr	January		February		March		April		May		June	
	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
0000	30.5	30.0	34.5	34.0	46.1	45.6	57.3	56.8	66.3	62.4	73.3	65.7
0100	29.5	29.0	33.5	33.0	45.1	44.6	56.1	55.6	65.1	61.9	72.1	65.3
0200	28.4	28.0	32.4	31.9	44.0	43.6	55.1	54.6	64.1	61.6	71.1	65.0
0300	27.6	27.1	31.6	31.1	43.2	42.7	54.0	53.6	63.0	61.2	70.0	64.6
0400	27.0	26.5	31.0	30.5	42.6	42.1	53.2	52.7	62.2	61.0	69.2	64.4
0500	26.8	26.3	30.8	30.3	42.4	41.9	52.6	52.1	61.6	60.7	68.6	64.2
0600	27.2	26.7	31.2	30.7	42.8	42.3	52.4	51.9	61.4	60.7	68.4	64.1
0700	28.2	27.7	32.2	31.7	43.8	43.3	52.8	52.3	61.8	60.8	68.8	64.2
0800	30.1	29.6	34.1	33.6	45.7	45.2	53.8	53.3	62.8	61.2	69.8	64.6
0900	32.8	32.3	36.8	36.3	48.4	47.9	55.7	55.2	64.7	61.8	71.7	65.2
1000	35.9	35.4	39.9	39.4	51.5	51.0	58.4	57.2	67.4	62.7	74.4	66.0
1100	39.4	38.9	43.4	42.9	55.0	54.5	61.5	58.3	70.5	63.8	77.5	67.0
1200	42.7	42.1	46.7	46.2	58.3	56.6	65.0	59.6	74.0	64.9	81.0	68.0
1300	45.1	43.3	49.1	48.4	60.7	57.6	68.3	60.8	77.3	65.9	84.3	69.0
1400	46.8	44.1	50.8	49.1	62.4	58.2	70.7	61.6	79.7	66.7	86.7	69.8
1500	47.4	44.4	51.4	49.4	63.0	58.4	72.4	62.2	81.4	67.2	88.4	70.2
1600	46.8	44.1	50.8	49.1	62.4	58.2	73.0	62.4	82.0	67.4	89.0	70.4
1700	45.3	43.4	49.3	48.5	60.9	57.6	72.4	62.2	81.4	67.2	88.4	70.2
1800	43.1	42.3	47.1	46.6	58.7	56.8	70.9	61.7	79.9	66.8	86.9	69.8
1900	40.4	39.9	44.4	43.9	56.0	55.5	68.7	60.9	77.7	66.1	84.7	69.2
2000	37.7	37.2	41.7	41.2	53.3	52.8	66.0	60.0	75.0	65.2	82.0	68.4
2100	35.5	35.0	39.5	39.0	51.1	50.6	63.3	59.0	72.3	64.3	79.3	67.5
2200	33.4	32.9	37.4	36.9	49.0	48.5	61.1	58.2	70.1	63.6	77.1	66.9
2300	31.7	31.2	35.7	35.2	47.3	46.8	59.0	57.4	68.0	62.9	75.0	66.2

Hr	July		August		September		October		November		December	
	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
0000	76.3	67.9	76.3	67.9	70.3	64.6	60.3	59.0	48.5	48.0	36.5	36.0
0100	75.1	67.5	75.1	67.5	69.1	64.2	59.1	58.6	47.5	47.0	35.5	35.0
0200	74.1	67.2	74.1	67.2	68.1	63.8	58.1	57.6	46.4	46.0	34.4	33.9
0300	73.0	66.9	73.0	66.9	67.0	63.5	57.0	56.6	45.6	45.1	33.6	33.1
0400	72.2	66.6	72.2	66.6	66.2	63.2	56.2	55.7	45.0	44.5	33.0	32.5
0500	71.6	66.4	71.6	66.4	65.6	63.0	55.6	55.1	44.8	44.3	32.8	32.3
0600	71.4	66.4	71.4	66.4	65.4	63.0	55.4	54.9	45.2	44.7	33.2	32.7
0700	71.8	66.5	71.8	66.5	65.8	63.1	55.8	55.3	46.2	45.7	34.2	33.7
0800	72.8	66.8	72.8	66.8	66.8	63.4	56.8	56.3	48.1	47.6	36.1	35.6
0900	74.7	67.4	74.7	67.4	68.7	64.0	58.7	58.2	50.8	50.3	38.8	38.3
1000	77.4	68.2	77.4	68.2	71.4	64.9	61.4	59.4	53.9	53.4	41.9	41.4
1100	80.5	69.1	80.5	69.1	74.5	65.9	64.5	60.5	57.4	55.4	45.4	44.9
1200	84.0	70.2	84.0	70.2	78.0	67.0	68.0	61.7	60.7	56.6	48.7	47.3
1300	87.3	71.1	87.3	71.1	81.3	68.0	71.3	62.8	63.1	57.6	51.1	48.4
1400	89.7	71.8	89.7	71.8	83.7	68.7	73.7	63.7	64.8	58.2	52.8	49.1
1500	91.4	72.2	91.4	72.2	85.4	69.2	75.4	64.2	65.4	58.4	53.4	49.4
1600	92.0	72.4	92.0	72.4	86.0	69.4	76.0	64.4	64.8	58.2	52.8	49.1
1700	91.4	72.2	91.4	72.2	85.4	69.2	75.4	64.2	63.3	57.6	51.3	48.5
1800	89.9	71.8	89.9	71.8	83.9	68.8	73.9	63.7	61.1	56.8	49.1	47.5
1900	87.7	71.2	87.7	71.2	81.7	68.1	71.7	63.0	58.4	55.8	46.4	45.9
2000	85.0	70.5	85.0	70.5	79.0	67.3	69.0	62.1	55.7	54.7	43.7	43.2
2100	82.3	69.7	82.3	69.7	76.3	66.5	66.3	61.1	53.5	53.0	41.5	41.0
2200	80.1	69.0	80.1	69.0	74.1	65.8	64.1	60.4	51.4	50.9	39.4	38.9
2300	78.0	68.4	78.0	68.4	72.0	65.1	62.0	59.6	49.7	49.3	37.7	37.2

Fig. A.3 Design temperature profile

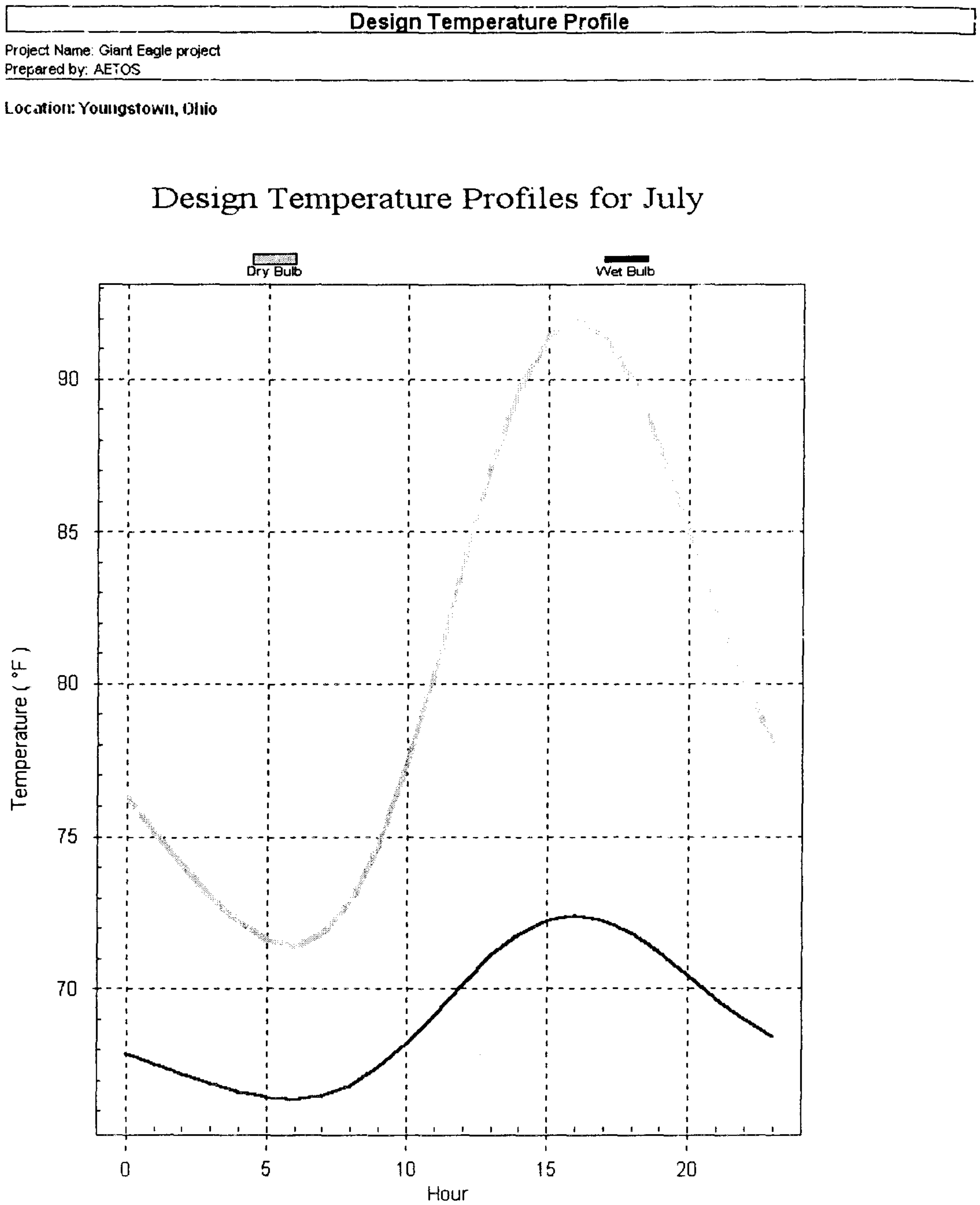

Table A.2 Air System Sizing Summary for the Conventional Unit

Air System Sizing Summary for Packaged Rooftop AHU

Project Name: Giant Eagle project
Prepared by: AETOS

Air System Information

System Name	Packaged Rooftop AHU		
Equipment Class	PKG ROOF	Number of Zones	1
System Type	DDCAV	Floor Area	13608.0 ft²

Sizing Calculation Information

Zone and Space Sizing Method:

Zone CFM	Peak zone sensible load	Calculation Months	Jan to Dec
Space CFM	Coincident space loads	Sizing Data	Calculated

Central Cooling Coil Sizing Data

Total coil load	14.6 Tons	Load occurs at	Jun 2200
Sensible coil load	12.0 Tons	OA DB / WB	77.1 / 66.9 °F
Coil CFM at Jun 2200	4595 CFM	Entering DB / WB	78.2 / 60.1 °F
Max possible CFM	9191 CFM	Leaving DB / WB	48.0 / 45.0 °F
Design supply temp.	48.0 °F	Coil ADP	40.4 °F
ft²/Ton	932.6	Bypass factor	0.200
BTU/hr/ft²	12.9	Resulting RH	41 %
Water flow @ 10.0 °F rise	- gpm	Zone T-stat Check	1 of 1 OK

Central Heating Coil Sizing Data

Max coil load	122547 BTU/hr	Load occurs at	Des Htg
Coil CFM at Des Htg	5986 CFM	BTU/hr/ft²	9.0
Max possible CFM	5986 CFM	Ent. DB / Lvg DB	78.2 / 98.0 °F
Water flow @ 20.0 °F drop	- gpm		

Precool Coil Sizing Data

Total coil load	22.9 Tons	Load occurs at	Jul 1600
Sensible coil load	13.7 Tons	OA DB / WB	92.0 / 72.4 °F
Coil CFM at Jul 1600	3780 CFM	Entering DB / WB	92.0 / 72.4 °F
Max possible CFM	3780 CFM	Leaving DB / WB	50.0 / 48.0 °F
Water flow @ 10.0 °F rise	- gpm	Bypass factor	0.200

Preheat Coil Sizing Data

Max coil load	96069 BTU/hr	Load occurs at	Des Htg
Coil CFM at Des Htg	9191 CFM	Ent. DB / Lvg DB	64.9 / 75.0 °F
Max possible CFM	9191 CFM		
Water flow @ 20.0 °F drop	- gpm		

Supply Fan Sizing Data

Actual max CFM at Jul 1700	9191 CFM	Fan motor BHP	12.00 BHP
Standard CFM	8804 CFM	Fan motor kW	8.95 kW
Actual max CFM/ft²	0.68 CFM/ft²		

Return Fan Sizing Data

Actual max CFM at Jul 1700	9191 CFM	Fan motor BHP	12.00 BHP
Standard CFM	8804 CFM	Fan motor kW	8.95 kW
Actual max CFM/ft²	0.68 CFM/ft²		

Outdoor Ventilation Air Data

Design airflow CFM	3780 CFM
CFM/ft²	0.28 CFM/ft²