

THE PERFORMANCE OF A DESICCANT-BASED AIR CONDITIONER ON A FLORIDA SCHOOL

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1.0 INTRODUCTION

Indoor air quality has become a major public health issue in recent years. ASHRAE standard 62-1989—which is an attempt to improve indoor air quality by increasing building ventilation rates—greatly increases the latent loads on many buildings. In more humid climates, the Sensible Heat Ratio (SHR) of a building's air conditioner (which is the fraction of total delivered cooling that is sensible) is too high to meet the existing latent loads. The implementation of ASHRAE 62-1989 will only exacerbate this problem.

One of the more sensitive populations to indoor air pollution is the country's school-aged children. As stated in Reference 1*:

The U.S. government's General Accounting Office issued reports stating that one in five schools in the United States has problems with indoor air quality (IAQ). According to the same studies, 36% of the schools surveyed listed HVAC systems as a "less-than-adequate building feature."

Furthermore, Reference 1 notes the connection between poor IAQ in schools and the health of the students:

There are many reasons that IAQ should be considered a top priority in the school environment. One is that children are still developing physically and are more likely to suffer the consequences of indoor pollutants. Another is that the number of children suffering from asthma is up 49% since 1982, according to the American Lung Association...Asthma is the principal cause of school absences, accounting for 20% of lost school days in elementary and high schools...

Desiccant air conditioners can be an important part of a strategy to improve IAQ in all buildings, including schools. Desiccants are materials that have a strong affinity for water vapor. This property allows a desiccant air conditioner to dry air without first cooling the air to below its dew point. The inefficient over-cooling followed by reheating that must frequently be used by conventional air conditioners when processing very humid air can therefore be avoided.

However, desiccant air conditioners are a relatively new technology. Their performance has not been sufficiently documented in the field to permit a meaningful comparison with conventional systems. Also, the engineering community is not familiar with them, and so they are reluctant to specify them.

The work reported here was conducted to obtain the field performance data that is needed to assess the ability of an air conditioner that uses both a DX cooling coil and a desiccant rotor to serve the large latent loads at a school with high ventilation rates and determine how the desiccant system affects indoor comfort and air quality. The work also compares the operating costs of the desiccant system with those for a conventional vapor-compression system. It is hoped that the results from this project will guide the HVAC engineering community to the proper selection of equipment needed to maintain the health of the country's students.

* 1) Bayer, Charles W., Sidney A. Crow, and John Fischer, "Causes of Indoor Air Quality Problems in Schools," US DOE Oak Ridge National Laboratory, ORNL/M-6633, January, 1999.

2.0 METHODOLOGY

The objective of this project was to determine the impact on both operating costs and classroom comfort/IAQ of replacing a school's existing vapor-compression air conditioner with one that uses a solid desiccant rotor to increase its latent cooling capacity. This task of quantitatively comparing the performance in the field of different HVAC systems under nearly identical operating conditions is always a challenge. Two approaches are possible—each having its advantages and disadvantages. In this project the approach that would have produced the most definitive results would first document the performance of the conventional air conditioner that was already at the school. Following this initial monitoring period, which ideally would have lasted for an entire cooling season, the desiccant-assisted air conditioner (DAAC) would have been installed and then monitored for the following cooling season. By recording both the outdoor weather conditions and the occupancy patterns within the school, it would have been possible to compare the performance of the two air conditioners under nearly identical conditions.

In this project, time constraints prevented the implementation of the previous approach. As an alternative, two school buildings were selected that were similar in construction, ventilation rates and occupancy patterns. Since the two buildings were on the same campus of the Tampa Bay Technical High School, weather conditions were nearly identical.

Unfortunately, the side-by-side approach to comparing two HVAC systems can have its own set of problems. In this project, it was very difficult to measure the ventilation rate to either building in its original configuration. For the building that was retrofitted with the DAAC, an accurate measurement of the ventilation rate was possible after the new system was installed. However, for the building that was to be monitored with its existing air conditioner, the configuration of the air inlet to the building prevented an accurate measurement of the ventilation rate.



The methodology that was finally used in this project was a compromise. The DAAC that was tested could operate in a mode in which only the DX vapor-compression subsystem operated. In this mode, the performance of the DAAC is essentially identical to that of a conventional air conditioner. Thus, by operating the air conditioner several weeks with the desiccant subsystem functioning (DA mode) and without it functioning (DX mode) the performance of the fully functional DAAC could be compared with that of a conventional system. (As noted above, this methodology was a compromise; it did not allow the performance of the DAAC to be compared to that of the conventional air conditioner that originally served the school.)

3.0 SITE DESCRIPTION

The Tampa Bay technical high school where the test was performed trains 1,800 students in work skills that include health care services, dental technician, business education, electronics, auto mechanics, HVAC technician, drafting, industrial arts, and journalism. The school's campus consists of 18 permanent buildings that were built

between 1969 and 1989. A photograph of the high school is shown in Figure 1.

The DAAC was tested on Building T of the high school. Building T is a 12,084 square feet, two-story classroom facility. The walls consist of concrete block with ½" rigid insulation and a 4" brick face. Glazing is 4.9% of floor area; it is single pane, non-tinted with overhang. The roof is a steel deck with 4" rigid insulation and built-up roofing material. The bottom floor is a concrete slab without insulation. The HVAC system that was replaced was a nominal 40-ton DX split system with a variable volume air delivery.

4.0 GENERAL OPERATION OF THE DESICCANT UNIT

The DAAC that was installed on Building T processes both the ventilation air and recirculation air for the building. The unit includes a silica-gel desiccant rotor with a nominal water removal capacity of 226 lb/h, a 40-ton two-compressor DX air cooler and a 160,000 Btu/h air heater.

For this project, the DAAC was modified so that it could operate both with and without the desiccant rotor active. As shown in the system layout (Figure 2) and psychrometric chart (Figure 3), when in the desiccant-assisted mode (DA mode), the Process Fan draws approximately 2000 cfm of outdoor air through Air Damper #1 (Point A) which is located on the side of the DAAC and 3000 cfm of return air through a damper in the floor of the DAAC (Point B). These streams mix (Point C) and are dehumidified as they move through the process side of the desiccant rotor (Point C to D). After leaving the desiccant rotor, the hot, dry process air is cooled in a heat-pipe heat exchanger (HPHX, Point D to E). It then mixes with approximately 5,000 cfm of return air (Point G) before being cooled by the DAAC's DX coil (Point G to H) and then delivered to the school (Point I).

Since the building is kept under positive pressure, the quantity of air that is exhausted through the DAAC will be slightly less than the ventilation air that is introduced through Air Damper #1. Slightly less than 2,000 cfm of return air is diverted to the exhaust stream that also becomes the regeneration air for the desiccant rotor. The temperature of this air stream is lowered in an evaporative cooler (Point K to L) before it becomes the heat sink for cooling the process air in the HPHX (Point L to M). After gaining energy in the HPHX, the regeneration air is further heated using desuperheat energy from the vapor-compression system (Point M to N). As discussed in Section 6, this use of desuperheat energy reduces slightly the amount of gas that must be burned to regenerate the desiccant. The regeneration air is finally heated in a direct-fired burner (Point N to O) after which it passes through the regeneration

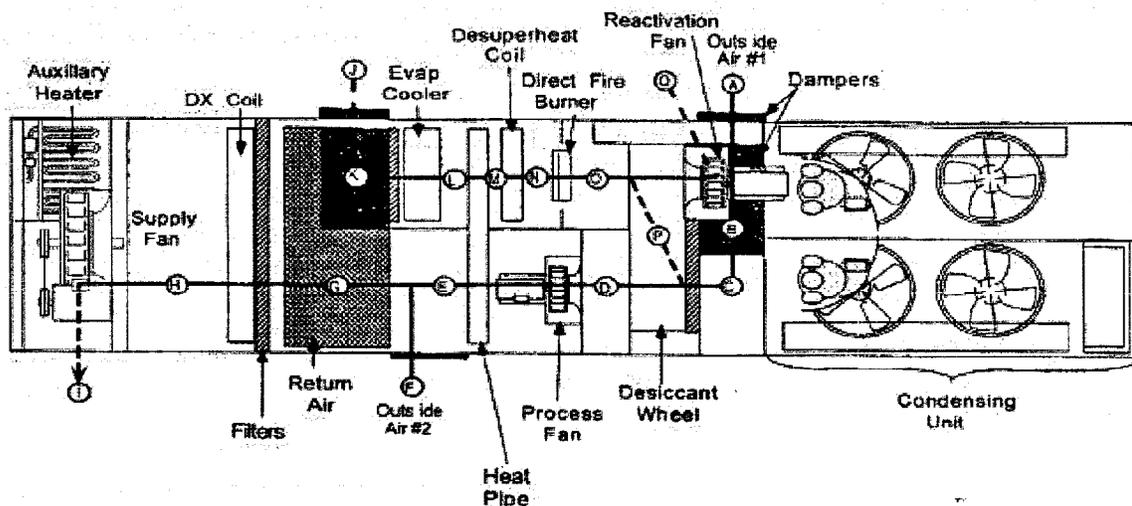


Figure 2 - Air Flow Paths through the Munters HR30GG Desiccant Air Conditioner

side of the desiccant rotor. The regeneration air along with the moisture that it strips from the rotor exits the unit through a damper in the roof (Point Q).

One-quarter of the face area of desiccant rotor is used for regeneration and three-quarters for dehumidification. The partitions that direct air into and out of the rotor are configured so that approximately 6.5% of the process air is "purged" through the hottest portion of the rotor (i.e., the process air that flows through the approximately 17.5 degree sector of the rotor immediately after the regeneration sector is split from the process air stream and exhausted with the regeneration air). This purge reduces the temperature of the rotor before it begins drying the process air, thus minimizing the heat "dumped" into the process air.

The regeneration burner operates whenever the humidistat that is located in the return air duct upstream of the DAAC senses a need for dehumidification. The burner is modulated so that the temperature of the regeneration air downstream of the desiccant rotor is maintained at 118°F. This temperature control also reduces the heat transferred from the hot desiccant rotor to the process air.

When there is no call for dehumidification, both the regeneration fan and the process fan are turned off. Outdoor Air Damper #1 is closed and Outdoor Air Damper #2 is opened so that ventilation air enters the DAAC immediately upstream of the DX coil.

The air distribution system is a VAV design with dampered VAV boxes located in each classroom. The VAV boxes are controlled so that they deliver conditioned air to the classrooms that need cooling (based on the control signal from the classroom's thermostat). The speed of the DAAC's supply fan is controlled to maintain a set-point pressure in the main supply duct to the building.

When the DAAC operates in the DX mode, 2000 cfm of ventilation air enters through Air Damper #2 (Point F). This air then mixes with the return air from the building and is cooled and dehumidified by the DX coil. (When in the DX mode, the DAAC operates as if it were in the DA mode, but with the humidistat always satisfied.)

5.0 INSTRUMENTATION AND DATA LOGGER

Ventilation Air Volume

Hot-wire anemometers were originally used to measure the ventilation-air volumetric flow through Outdoor Air Dampers #1 and #2. (The damper locations are shown in Figure 2). These anemometers were mounted in ducts that were installed on the DAAC and that provided a relatively well-behaved flow region upstream of the dampers in which to make a velocity measurement. The anemometers had a full-scale reading of 2,500 fpm and a measurement accuracy of 2% of reading plus 0.5% of full scale.

A third hot-wire anemometer measures the return air that is delivered upstream of the desiccant rotor. Its full-scale reading is 1,000 fpm and its measurement accuracy is 2% of reading plus 0.5% of full scale.

The hot-wire anemometer that measured the ventilation air through Outdoor Air Damper #1 began to operate erratically several months after its installation. The problem was traced to unusually high levels of particulates in the outdoor air generated by a local industrial plant. Particulate loadings in the air were sufficiently high that they coated the probe with a thin layer.

Since the hot-wire anemometers would require unusually high maintenance, rotating cup anemometers were installed as redundant measurements for the two ventilation airflows. These anemometers have a full-scale reading of 7,500 fpm and an accuracy of 2% of reading.

CO₂ Sensors – Classrooms

Wall-mounted CO₂ sensors were located in one first-floor and one second-floor classroom at a height of about 5 feet and within 1 foot of the classroom's thermostat. The sensors use a non-dispersive IR technology. The sensors were factory calibrated to read 0 to 5,000 ppm with an accuracy of 1% full-scale plus 1.5% of reading.

CO₂ Sensors – Ducts

Duct-mounted sensors measured the CO₂ concentration in the ventilation air, supply air and return air. The ventilation CO₂ sensor was located in duct that delivered air to Outdoor Air Dampers #1 and #2. Both the supply-air and return-air CO₂ sensors were located in the mechanical room that was below the roof-mounted DAAC. All three CO₂ sensors used non-dispersive IR technology. The sensors were factory calibrated to read 0 to 2,000 ppm for the ventilation air and 0 to 5,000 ppm for the supply and return air with an accuracy of 1% full-scale plus 1.5% of reading.

Classroom Temperature and Relative Humidity

Wall-mounted T/rh transmitters measured the temperature and relative humidity in the classrooms. Their temperature accuracy was $\pm 1^{\circ}\text{F}$ and relative humidity accuracy, $\pm 2\%$ for readings between 20% and 95% rh. They were located next to the room CO₂ sensors.

Outdoor Temperature and Humidity

Outdoor air temperature and relative humidity were monitored with a probe that contained a capacitive rh sensor and a 1000-ohm platinum RTD. The rh sensor had a range of 0 to 100% with an accuracy of $\pm 2\%$ rh. The temperature sensor had a range of -40°F to 140°F with an accuracy of $\pm 0.5^{\circ}\text{F}$. The outdoor T/rh sensor was located at the corner of the walkway that runs around the desiccant air conditioner approximately 7 feet above the roof and mounted in a radiation shield.

Process, Ventilation, Supply and Return Air Dew Points

Duct-mounted chilled-mirror hygrometers were used to measure the dew-point of the ventilation, supply and return air streams, as well as the dew-point of the process air downstream of the process air blower. They have an accuracy of $\pm 1^{\circ}\text{F}$ and a mirror depression of 70°F at an ambient temperature of 77°F .

The hygrometer that monitored the ventilation air was located in the external ventilation duct immediately downstream of the anemometer. The hygrometers that monitored the supply and return air were located in the mechanical room.

Ventilation, Supply and Return Temperatures

Thermistor probes were used to measure the temperature of the ventilation, supply and return air. The thermistors have an accuracy of $\pm 0.2^{\circ}\text{F}$ over a range of -40°F to 250°F .

Temperatures within the Desiccant Air Conditioner

Thermistors were used to measure temperatures at the following five locations within the desiccant air conditioner:

- (1) Process air entering the heat-pipe heat exchanger
- (2) Process air leaving the heat-pipe heat exchanger
- (3) Regeneration air leaving the saturator before it enters the heat-pipe heat exchanger
- (4) Regeneration leaving the heat-pipe heat exchanger before it enters the desuperheater
- (5) Regeneration leaving the desuperheater before it enters the burner

The thermistors had an accuracy of $\pm 0.2^{\circ}\text{F}$.

Classroom Occupancy

Passive-infrared motion detectors provided a qualitative indication of the activity within the classrooms. A detector was located in the corner of each classroom near the ceiling.

Electrical Power

Power to the desiccant air conditioner was monitored with a variable-frequency watt transducer. The transducer has a full-scale reading of 160 kW and an accuracy of 1% of full scale.

Regeneration and Air Heater Gas Consumption

Diaphragm gas meters were used to monitor gas consumption for both the regeneration burner and the auxiliary air heater. The meters have a 1 CF resolution, and a 415 CFH maximum reading.

The gas meters are temperature compensated but not pressure compensated. Changes in gas density due to changes in the supply pressure must be compensated for as part of the data reduction. A 30-psia pressure transmitter with an accuracy of 0.1 psia was used to measure the gas pressure.

Pressure Drop Across the Heat-Pipe Heat Exchanger

The pressure drops across both the process-side and the regeneration-side of the heat-pipe heat exchanger were monitored with pressure transducers. The transducers had a range of 0 to 5 inches w.c. and an accuracy of 1% full scale.

Damper and Dehumidification Status

The status of the two ventilation air dampers on the desiccant air conditioner and its control signal for dehumidification are monitored with industrial I/O modules.

Data Logger

A programmable data logger performed the A-to-D conversion and recording of the signals from the instrumentation. A 16-channel relay multiplexer and a 8-channel switch-closure module were used to increase the available inputs to the data logger. The data logger included 2 MB of memory that provided about 14 days of on-site data storage. The data channels were sampled every five seconds and average or integrated values were stored once per minute. Data was downloaded off-site to a data-storage computer every night.

6.0 VERIFICATION OF COMPONENT PERFORMANCE

Rotor Performance

During the monitoring period, the DAAC operated with two different desiccant rotors. Prior to March 16, 2000, the DAAC used the manufacturer's standard silica-gel rotor. On that date, an advanced rotor was installed.

Figure 4 compares the measured performance of the original silica-gel rotor with the manufacturer's predicted performance. The square data points are the design operation for the unit and the circle data points are the measured operation. As shown in this figure, the circle and square data points at B (entry conditions to the rotor) and at C (exit conditions from the rotor) are close to each other, indicating that the rotor performed very close to its specifications.

Blowers and Compressors

The power consumption of the process air, regeneration air and supply air blowers and the DX system's compressors were checked to verify that measured values were consistent with the catalog ratings for these motors. (Although only one power transducer monitors the desiccant air conditioner, it is possible to infer the power of individual motors by measuring the changes that occur as motors cycle on and off.) The measured and approximate catalog powers are as follows:

	<u>Measured</u>	<u>Catalog</u>
Process Fan	4 kW	3.9 kW
Regeneration & Supply Fan	14 kW	14.4 kW
One Compressor Stage	20 kW	20 kW (assuming 1 kW per ton)

Gas Consumption

The gas burner for regeneration modulated, so it was difficult to compare its measured gas consumption rate with its catalog value. During the first six months of monitoring, the highest 15-minute average for gas consumption by the regeneration burner was 742 kBtuh on November 14, 1998, a day when maintenance was performed on the unit. This value is close to the 750 kBtuh catalog value for this burner.

Direct Evaporative Cooler

Throughout the monitoring period, the operation of the direct evaporative cooler (DEC) was problematic. Some of the problems were "self-imposed". (E.g., when the DAAC was installed, the bleed from the DEC's sump was not turned on. This led to severe mineral accumulation within the DEC that eventually blocked the air flow.) These performance problems are described in a later section.

The data that were collected yielded a mixed picture for the DEC's performance. Some early data showed effectivenesses as high as 90% (where the effectiveness is defined as the ratio of the actual temperature change of the air divided by the change that would have occurred if the air had been cooled to its wet-bulb temperature). However, throughout most of the test period, the DEC's measured effectiveness was between 30% and 70%.

Heat-Pipe Heat Exchanger

The performance of the DAAC was monitored with two different heat-pipe heat exchangers (HPHX). On-site temperature traverses of the process air leaving the HPHX that was initially installed on the DAAC showed temperature variations of 25°F. Since the process air that flowed over the outer section of the HPHX exited at a much higher temperature than the air that flowed over the inner section, there appeared to be a problem with the distribution of working fluid within the heat pipes. The estimated effectiveness for the first HPHX when it was operating near the maximum observed heat fluxes was about 60%.

In March, 2000, a new HPHX was installed. This HPHX performed much better than the first. Its effectiveness at high heat fluxes was between 70% and 73%.

The data plotted on the psychrometric chart in Figure 4 were taken from the summer of 1999, when the original heat HPHX was still in place. It is clear from this data that the original heat pipe did not perform as expected: the measured data point at D is almost 20°F higher than the temperature based on design operation.

Desuperheater

The DAAC had a desuperheater that increased the unit's efficiency by using reject heat from the DX air cooler to preheat the regeneration air. In the manufacturer's design of the DAAC, the desuperheater supplies 19.1 kBtuh to the regeneration air. This is about 2.5% of the total regeneration heat input. The calculated heat recovery from the desuperheater was 25.9 kBtuh, or 3.4% of the total. The desuperheater also increases the efficiency of the vapor compression system by reducing the load on the condensers. This effect was not quantified in this test.

7.0 DAAC SYSTEM PERFORMANCE

DAAC Performance with and without the Desiccant Rotor Enabled

The DAAC that was installed at the high school was able to serve much larger latent loads than a conventional DX air conditioner. However, this added capability increased the costs for cooling and dehumidifying the school.

To determine the impact that the desiccant subsystem had on both classroom conditions and operating costs, we compared these performance parameters when the DAAC operated as a conventional air conditioner (i.e., DX mode) and a desiccant-assisted unit (DA mode). Since both operating costs and classroom conditions will be a strong function of the outdoor weather, this comparison was made as a function of outdoor conditions.

The two test periods for which the DX and DA comparison was made were April 19 through December 17, 1999 for the DX mode and September 15 through December 1, 2000 for the DA mode[†]. State point parameters (e.g., temperatures, humidities) were averaged from 6:30 AM to 4:00 PM for each day that the school was in session. Other parameters such as electricity and gas consumption, and operating time were totaled over the same time interval.

For both the DA and the DX modes, 2,000 cfm of outdoor air was supplied to the school. This air flow corresponded to 15 cfm per student. It is important to note that the DAAC was designed to dehumidify approximately 5,000 cfm of outdoor air. (As explained in a later section, the student population in the building that was the test site was only 40% of the number that was expected at the time that the DAAC was designed.) It is possible that the DAAC in DA mode is being penalized because it is operating at loads very much below its design loads—e.g., the energy needed to heat the rotor as it passes from the process side to the regeneration side, which is a parasitic loss for the system, will not decrease linearly with the latent load served by the rotor. Also, it is likely that in DX mode, the air conditioner could not handle the latent load that would accompany 5,000 cfm of outdoor air for ventilation without overcooling the air and then reheating it. This energy penalty on the DX mode could not be quantified in this test.

Also, it should be noted that we were not able to verify that the direct evaporative cooler in the regeneration air stream was operating properly. If this component was not operating at a 90% effectiveness, the DAAC in DA mode will have higher operating costs than it would otherwise.

In Figure 5, the daily averages for the outdoor dry-bulb temperature is plotted as a function of the outdoor wet-bulb temperature for the days included in the DA and DX test periods. This figure shows a reasonably good correlation between the two temperatures for both test periods. Since these two temperatures did correlate well, we compared the performance of the DAAC in the DA and DX modes as a function only of outdoor wet-bulb temperature.

Figure 5 also shows that both test periods have several days with very high wet-bulb temperatures (i.e., above 75°F).

In Figure 6, the fraction of the day that the regeneration burner operates (i.e., the humidity of the return air to the DAAC is above the humidistat's setpoint) is plotted as a function of average outdoor wet-bulb temperature. As shown in this figure, there was no need for dehumidification by the desiccant rotor at outdoor wet-bulb temperatures below 65°F. At outdoor wet-bulb temperatures above 75°F, the dehumidifier operated almost continuously.

[†] It was not possible to include data for the DAAC in the DA mode during summer months because of a combination of problems at the site. However, there were days with very high wet-bulb temperatures in the test period for the DA mode.

When the outdoor wet-bulb temperature was below 65°F, the operation of the DAAC in either the DA or the DX mode looked the same. In both modes, the supply fan and the DX system's compressors operated to cool and dehumidify the school, but the regeneration burner and the regeneration and process fans did not. This behavior is shown in Figure 7 where the data for daily electricity use for the two modes scatter about the same curve when the outdoor wet-bulb temperature is below 65°F. Above this wet-bulb temperature, the DA mode's electricity use is higher.

At an outdoor wet-bulb temperature of 75°F, the regeneration burner operated almost continuously when the DAAC was in the DA mode. Since the process and regeneration fans also operated when the burner was on, the DA mode should use approximately 100 kWh more than the DX mode whenever the outdoor wet-bulb temperature was at or above this value. (The additional fan power was 11.2 kW and the fans operated for 9.5 hours.) The data in Figure 7 shows this behavior.

The gas consumption for regenerating the desiccant rotor when the DAAC is in the DA mode is shown in Figure 8, again as a function of outdoor wet-bulb temperature. Gas consumption is shown to increase approximately linearly when outdoor wet-bulb temperatures were above 65°F.

Figures 9 and 10 show the indoor conditions that are maintained by the DAAC when it was operating in either the DA or DX modes. For both figures, the state of the return air from the school to the DAAC was assumed to reflect the conditions within the school. In both operating modes, Figure 9 shows that the DAAC maintained indoor temperatures within a range between 70°F and 75°F.

Although indoor temperatures were maintained at comparable values by both operating modes, the same cannot be said for indoor humidity. As shown in Figure 10, DX mode operation during high outdoor wet-bulb days produced indoor humidities that tended to be 10 to 20 grains higher than when the DAAC operated in DA mode.

Seasonal Operating Costs for DAAC

The preceding comparison between the DA and DX modes of operation for the DAAC shows that classroom conditions will be more comfortable with the desiccant subsystem operating. However, this improved humidity control will entail higher operating costs.

To estimate the added cost to operate the desiccant system for one school year, we assume that the operation of this system depends on outdoor wet-bulb temperature as shown in Figures 5 through 10. Then, starting with TMY2 hourly weather data for Tampa, Fl, we calculate the average wet-bulb temperature for each day of the year between the hours of 6:30 AM and 4:00 PM. The occurrence of daily wet-bulb temperatures are summed to wet-bulb bins that are 3°F "wide" and start at 65°F.

As shown in Figure 6, the fraction of the hour that the desiccant wheel is fired (and the process and regeneration fan operate) is an approximately linear function of the day's average wet-bulb temperature with the zero occurring at 65°F. Using this linear function for desiccant wheel's operating time, the gas use shown in Figure 8 can be approximated by an average firing rate of 380,000 Btuh for the regeneration burner.

The additional electricity use for the DAAC when it is dehumidifying is dominated by the power for the regeneration and process fans. These fans have a combined electrical load of 11.2 kW.

To complete the calculation of seasonal operating costs, we assume that the school is occupied (and hence the dehumidifier is enabled) only during weekdays when the school is in session, and that the school is in session for 45 weeks per year. With this schedule for classes, the school would be occupied and fully conditioned for 62% of the days in the year. This percentage is multiplied by the number of days in each wet-bulb bin. Finally, assuming that the dehumidifier is enabled for 9.5 hours of each occupied day, the number of operating hours for the dehumidifier is calculated and converted to a gas and electricity usage using the 380,000 Btuh firing rate for the regeneration burner and the 11.2 kWh load

for the two fans. The result of this analysis is that the operation of the desiccant wheel increases annual gas and electricity use by 353 MBtu and 10,400 kWh.

DAAC Performance with High Ventilation Rates

As noted earlier, the DAAC that was installed on Tampa Bay Technical High School was designed to process 5,000 cfm of outdoor air for ventilation. However, once the project was underway, it became evident that this ventilation rate was approximately 2.5 times too high given the actual occupancy of the school.

The ventilation rate through the DAAC was reduced from 5,000 cfm (nominal) to 2,000 cfm (nominal) in October 1999. Figure 11 compares the daily maximum hourly-averaged CO₂ concentration in one of the classrooms both before and after the ventilation rate was changed. As shown in this figure, the maximum values tended to scatter around a value of 1000 ppm for the lower ventilation rates with values peaking at about 1150 ppm. At the higher ventilation rate the CO₂ concentration is significantly lower: the data scatters around 800 ppm with values peaking at about 930 ppm.

8.0 PROBLEMS ADDRESSED IN THE FIELD TEST

It would not be reasonable to expect that a project as complex as this one would proceed without problems. To the extent that the problems addressed in this project are general, they are documented here to help others working with DAACs.

These problems fell into the one of the following three categories: (1) instrumentation, (2) building controls, and (3) DAAC hardware. The instrumentation problems tend to be specific to the project. While they may be interesting to other researchers that execute field tests, they do not effect the applications of DAACs, and so they will only be briefly discussed.

Instrumentation

The most vexing instrumentation problem occurred with the CO₂ sensors. Almost immediately after their installation, almost all of the classroom sensors began to drift badly. The manufacturer of the sensor attributed the problem to contamination of the sensor by water vapor. Replacement sensors were installed and they operated without problems for the duration of the test.

As noted in Section 5.0, the hot-wire anemometer that was installed in the duct that directed ventilation air to the DAAC began to behave erratically about four months after it was installed. A site visit to fix the problem showed that the probe was caked with a powder, most probably from a cement plant that operated in the vicinity of the school.

The transducer that measured the pressure to the gas meter failed at the start of the project and had to be replaced.

Building Controls

The original test plan specified that the DAAC would operate under the control of the building's energy management system (EMS), a unit manufactured by Andover Controls. The EMS was to provide a signal to the DAAC to identify periods when the building was occupied and unoccupied. Ventilation rates were to be reduced to zero during unoccupied periods.

For reasons that could not be determined, the control signal from the EMS to the DAAC was very erratic. During one visit to the site in November 1998, it was observed that the DAAC received several signals to switch between occupied and unoccupied modes, each signal separated by only a few seconds.

The short-cycling of the control signal to the DAAC probably caused the fuse for the process fan to blow several times during the first few months of the test.

Despite several attempts to correct the problem, a reliable interface between the DAAC and the EMS could not be established. The problem was circumvented by installing a programmable timer at the site and using it to supply an occupied/unoccupied signal to the DAAC.

During periods that lasted many weeks, the DAAC was controlled to operate in the occupied mode 24 hours per day, 7 days per week. This operation greatly increased both gas and electricity usage since the school was fully ventilated during periods when it was unoccupied.

DAAC hardware and Its Controls

During the two years that the DAAC was monitored, several hardware problems occurred. These were as follows:

- The humidistat that controls the operation of the DAAC was initially installed wrong: it was both wired incorrectly and oriented incorrectly within the duct. These problems led to almost continuous operation of the desiccant unit during most of October 1998. The school was significantly “over dried” during this period with room relative humidities frequently below 20% during unoccupied periods.
- Towards the end of the monitoring period, the controller that modulates the regeneration burner failed. This controller adjusts the firing rate of the desiccant rotor so that the regeneration air leaving the rotor is 118°F. When this controller failed, air temperatures leaving the regeneration sector of the rotor exceeded 160 °F. As shown in Figure 12, this failure almost doubled the gas usage for the DAAC.
- The DAAC was mistakenly switched by school personnel into its manual dehumidification mode on several occasions over the duration of this test. This mode was intended to allow service technicians to force the desiccant components of DAAC to operate for diagnostic purposes. Unfortunately when left in this manual dehumidification mode, the reactivation burner runs continuously increasing energy consumption and over drying the space.
- When the DAAC was installed, the bleed for the evaporative cooler was not set up. Without this bleed operating, the minerals that are dissolved in the make-up water for the evaporative cooler built up within the unit. After several months of operation without the bleed, the regeneration air flow became severely restricted. A service call was required to replace the core of the evaporative cooler and set up the bleed.
- The heat-pipe heat exchanger (HPHX) that was initially installed in the DAAC did not meet its performance specifications. As shown in Figure 13, the effectiveness of this heat exchanger, which should have been above 70%, degraded as it operated at higher heat fluxes. At the highest heat fluxes, the effectiveness was 60%. A new HPHX was installed in the DAAC in March 2000. As shown in Figure 13, this HPHX maintained an effectiveness of at least 70% at high heat fluxes.

9.0 CONCLUSION

The work reported here demonstrated the ability of a desiccant-assisted air conditioner to deliver high volumes of ventilation air to a school in a very humid climate while maintaining very comfortable classroom conditions. At ventilation rates that kept classroom CO₂ concentration levels below 1000 ppm most of the time, with a few excursions to about 1150 ppm, the DAAC kept humidity levels in the classroom below 72 grains. Based on the field test data, a conventional DX air conditioner is projected to keep classroom humidities at levels that are between 10 and 20 grains higher. Assuming that the building is ventilated for 9.5 hours on days when school is in session, the additional gas and electric usage for the DAAC compared to a conventional DX air conditioner is estimated to be 10,400 kWh and 353 Mbtu per year.

The field test also uncovered several problems that are likely to be encountered in other DAAC installations. The DAAC introduces two components that many HVAC engineers will not be familiar with: the desiccant rotor and the evaporative cooler. The desiccant rotor did not present problems during the field test. Problems with the evaporative cooler, however, did compromise the performance of the DAAC. During one period in the test, the evaporative cooler operated without a bleed from its sump to control mineral build up. During another period, the water supply to it was inadvertently turned off.

It is important to note that several problems within the DAAC caused higher than normal operating costs, but since the DAAC continued to deliver cool and dry air, they might have gone undetected if the unit was not extensively instrumented for the field test. These problems included (1) a heat-pipe heat exchanger that did not meet its performance specifications, (2) a faulty controller for the regeneration burner that led to firing rates that were too high, and (3) an evaporative cooler that became inoperable due to improper maintenance. The incorporation of internal diagnostics that could detect problems such as these may be a useful addition to a future DAAC.

10.0 ACKNOWLEDGEMENTS

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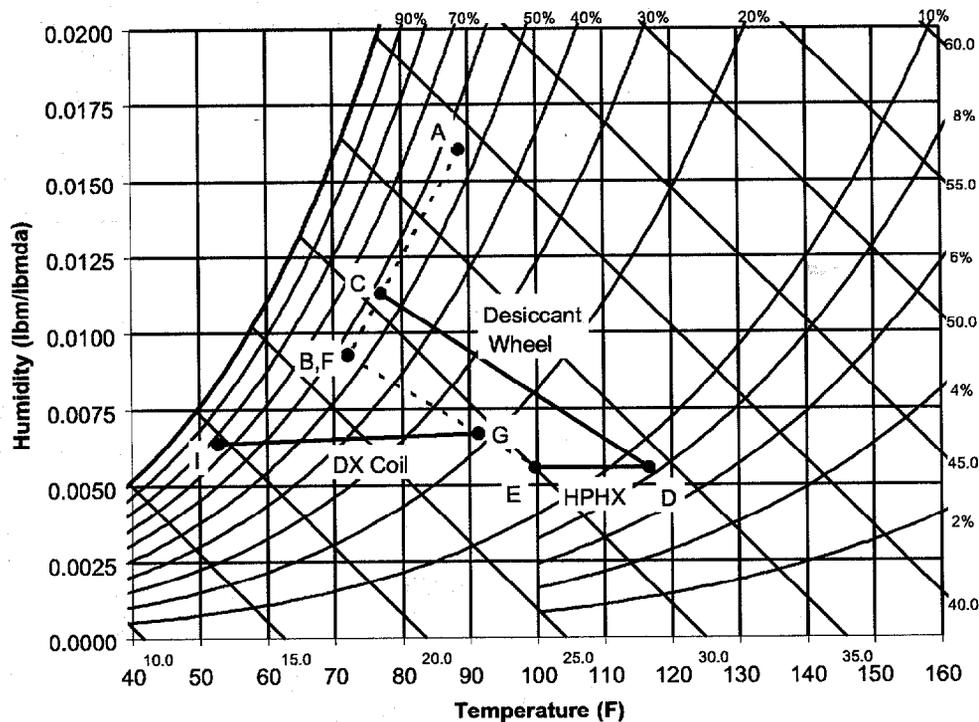


Figure 3 - Psychrometric chart showing typical performance of the DAAC.

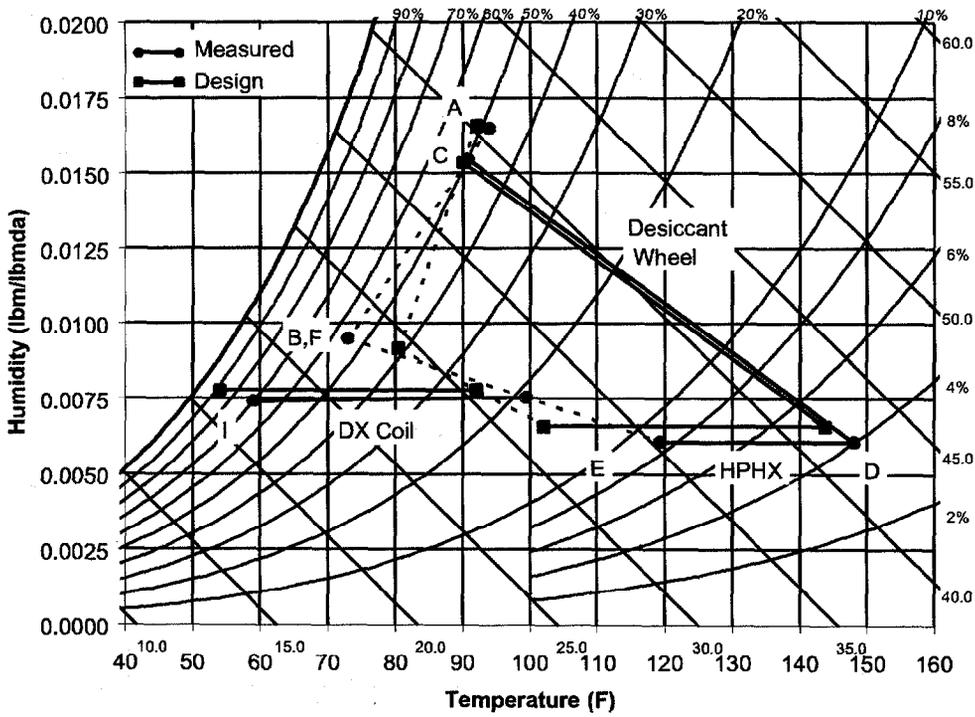


Figure 4 - Psychrometric chart comparing the measured performance of the DAAC to the design performance.

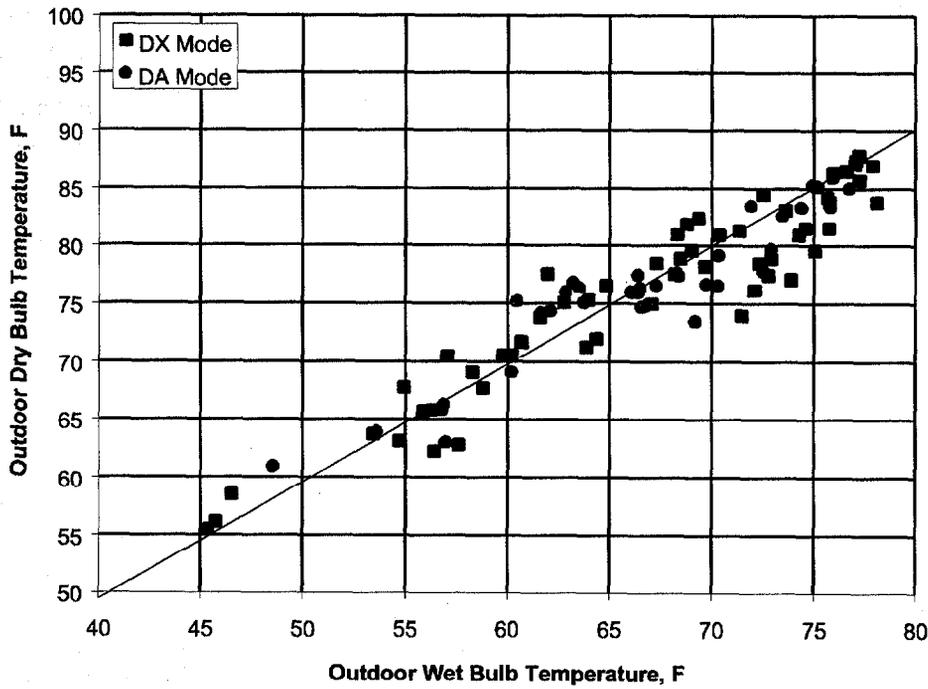


Figure 5 - Correlation of outdoor dry-bulb temperature and outdoor wet-bulb temperature.

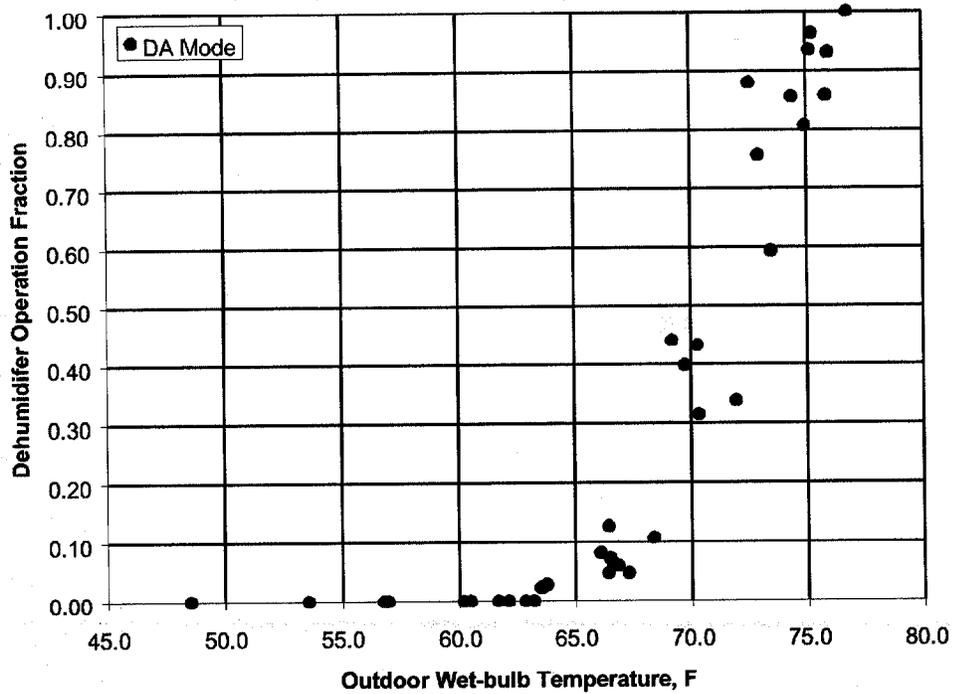


Figure 6 - Variation in the daily fraction of dehumidifier operation with outdoor wet-bulb temperature.

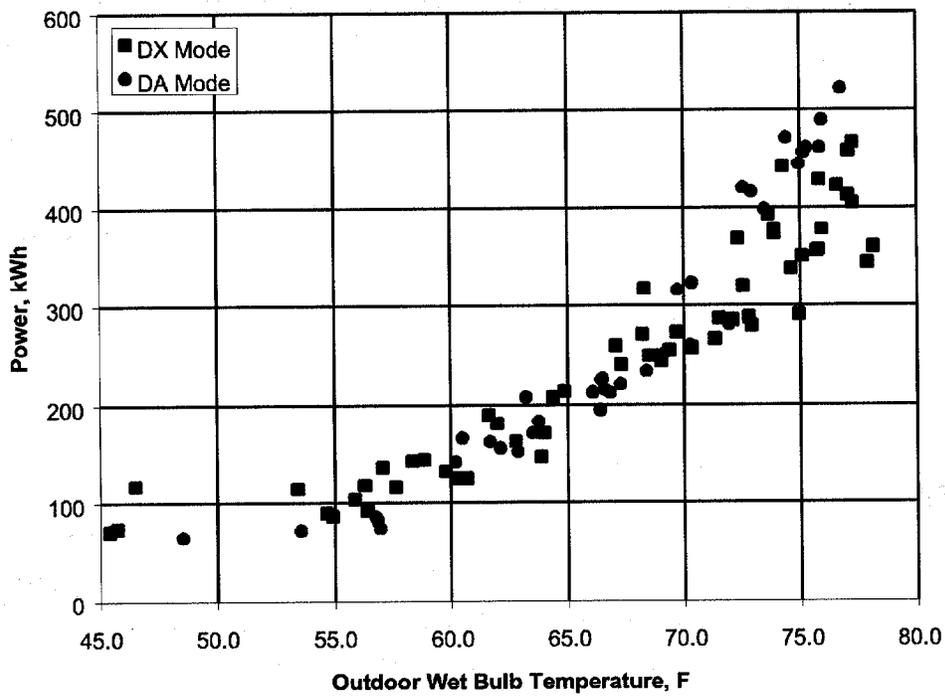


Figure 7 - Daily electric usage as a function of outdoor wet-bulb temperature.

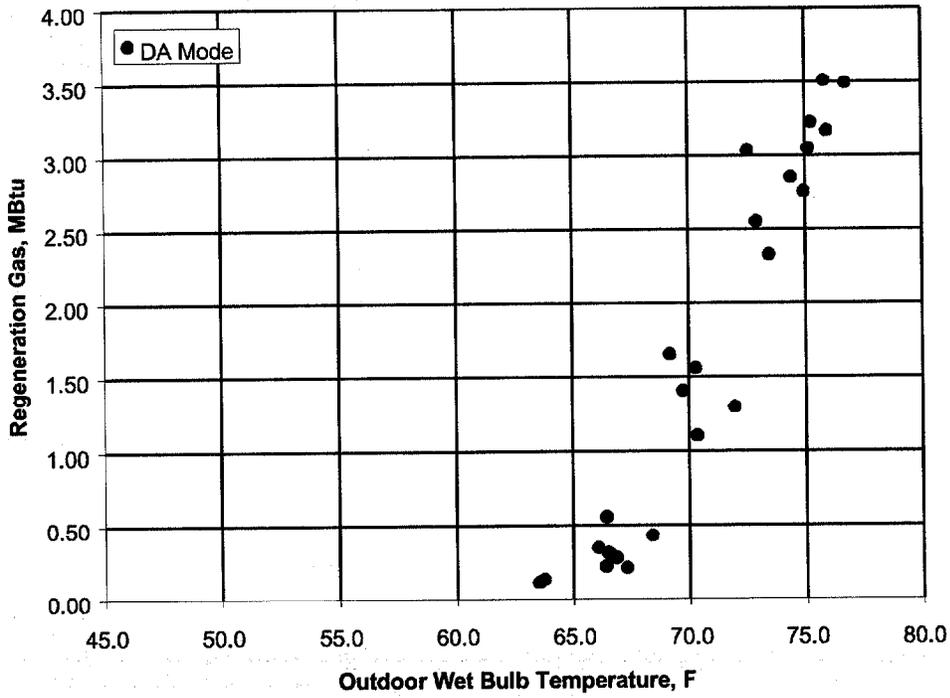


Figure 8 - Daily natural gas usage as a function of outdoor wet-bulb temperature.

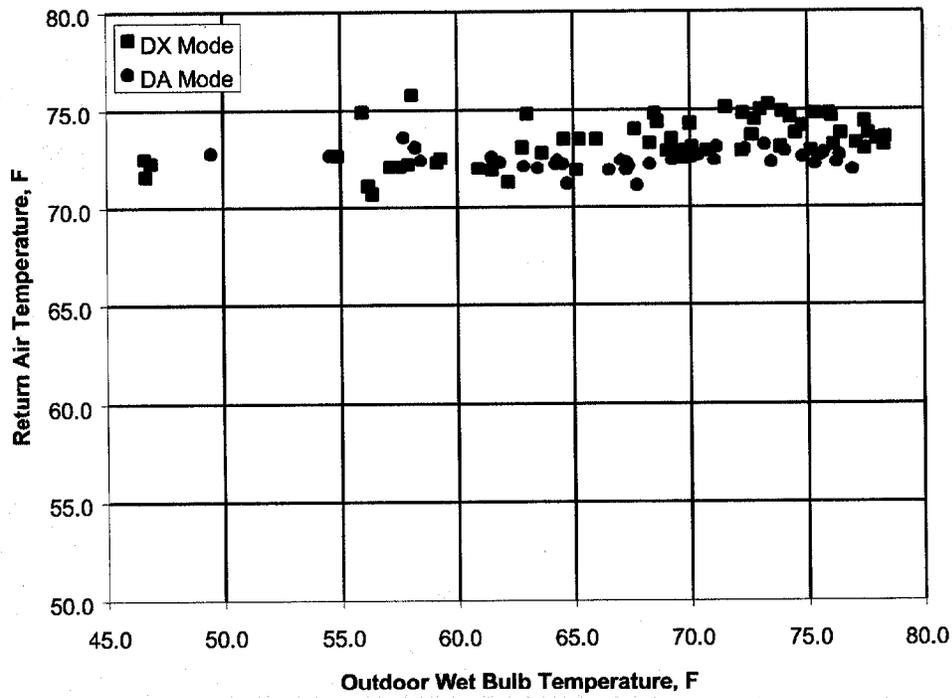


Figure 9 - Variation in the return temperature with the average outdoor wet-bulb temperature.

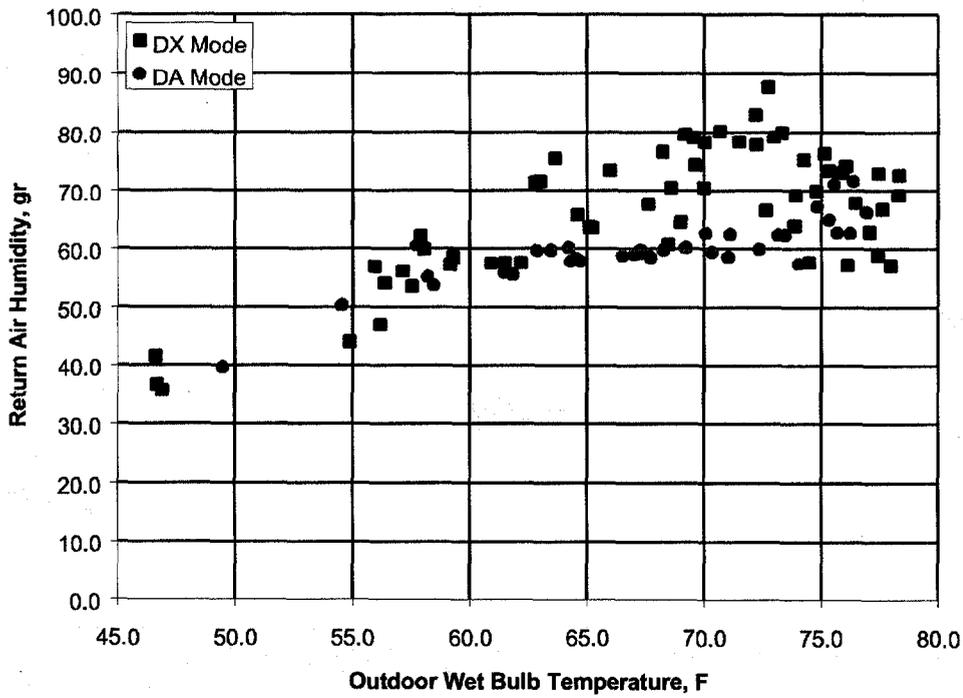


Figure 10 - Variation in the return humidity ratio with the outdoor wet-bulb temperature.

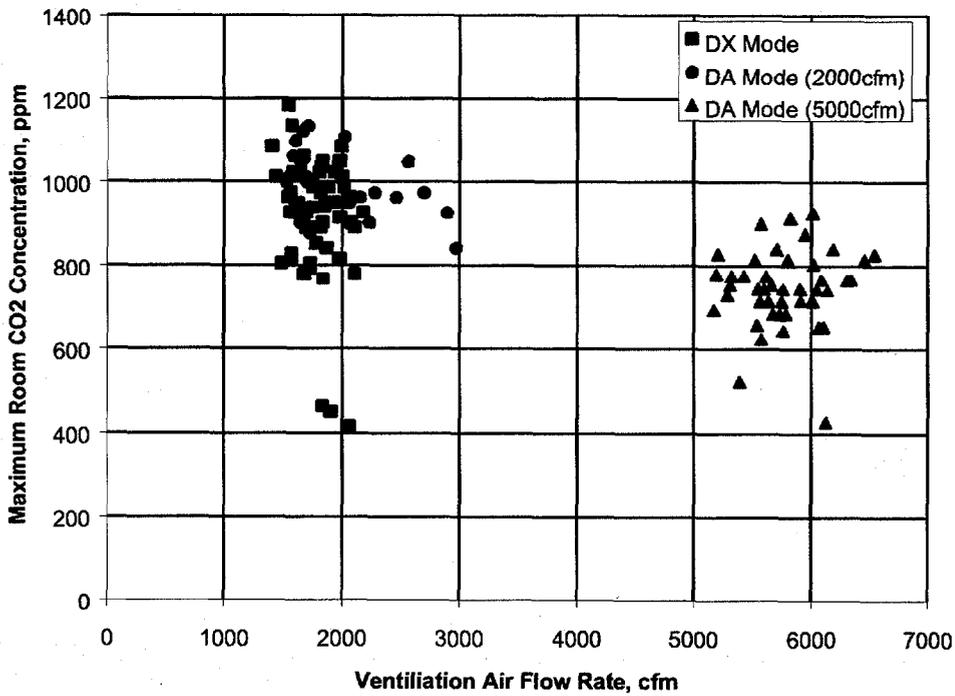


Figure 11 - Variation of maximum CO2 concentration in room number 1 with ventilation rate. (Note that the back ground concentration is approximately 425 ppm.)

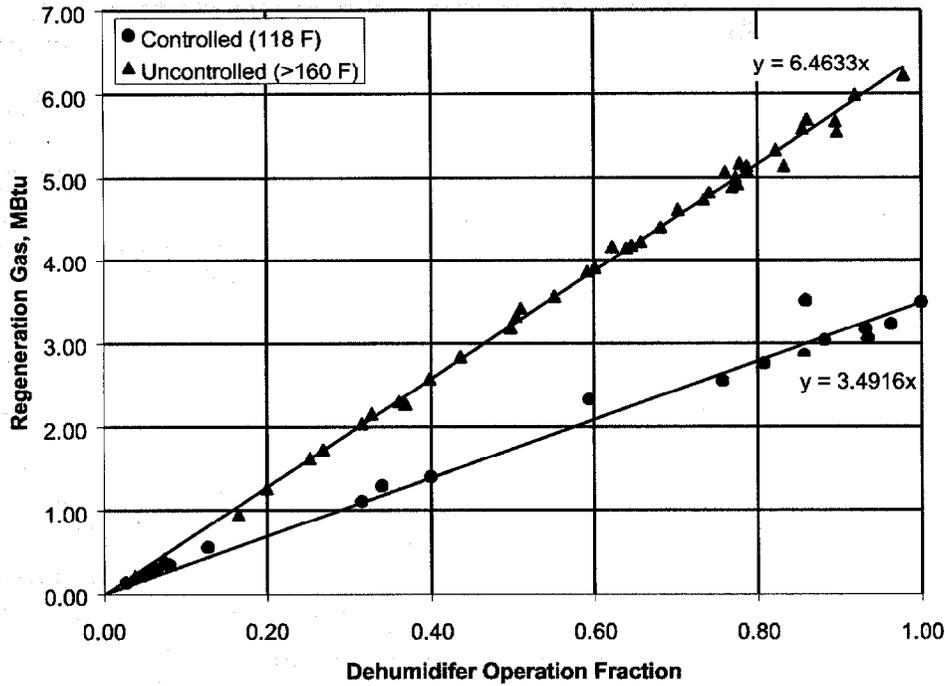


Figure 12 - Daily natural gas usage as a function of fraction of dehumidification.

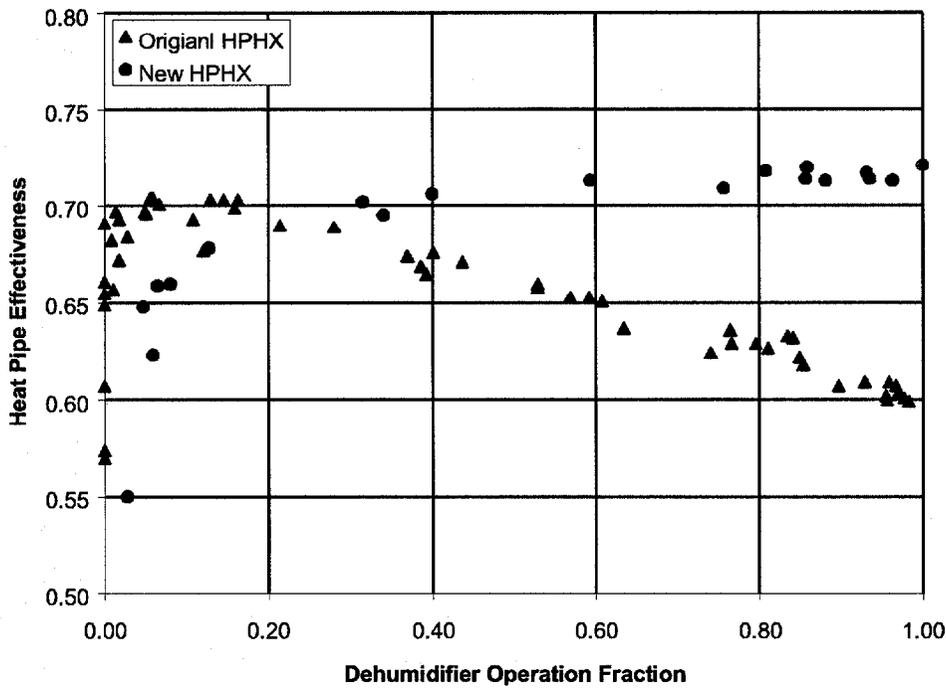


Figure 13 - Heat-pipe heat exchanger effectiveness as a function of the dehumidification load.