CONTROL OF HUMIDITY WITH SINGLE - DUCT, SINGLE - ZONE, CONSTANT AIR VOLUME SYSTEM - A CASE STUDY

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ABSTRACT

The lecture hall of the Richardson Petroleum Building at Texas A&M University is a large lecture hall, with a total floor area of approximately 2500 ft^2. The lecture hall was served by a constant air volume (CAV) air handling unit (AHU) which had no reheat coil. This resulted in high room humidity levels although the room temperature was satisfied for part load conditions, especially when there was very little sensible load from the room. This paper presents Continuous Commissioning efforts (CC), which turned this inefficient, humid lecture hall into a comfortable learning environment. This case study also explores other possibilities to solve the humidity control problem with single-duct, single-zone constant air volume systems.

INTRODUCTION

The lecture hall in the Richardson Petroleum Building is a large computer lab (laptop lab). During the summer of 1999, the room had a very strong musty smell, mold was growing, and all the computers were moved out due to the high humidity. The Utilities Energy office was called to help and the CC group of the Energy Systems Lab (ESL) performed a field survey for the disfunctional lecture hall. The primary purpose of this study was to improve the high humidity conditions of the lecture hall with an emphasis on improved indoor air quality. The CC group performed extensive field tests and analyses on the single-duct constant air volume equipment and its Energy Management Control System (EMCS) algorithm controls. Several problems were observed and most were related to the very high humidity of the local climate (e.g., 98% design conditions are 94/75 DB/WB (°F /°F) for College Station, Texas). Deficiencies (in design and/or operating conditions) which caused the computer lab to fail to control humidity levels were identified. Some technologies such as variable speed drive, heat pipes, run-around coils, dual-path air-conditioning and return air face and bypass may be used to reduce space relative humidity (Gatley, D., P., 1992). Most use less energy and have less electric demand. However, building owner does not like to take high cost investment to obtain benefits, so rejected these technologies. This paper presents a long-term solution, which involved installing a reheat coil, and a short-term approach, which reduced supply airflow with colder discharge temperature, in order to reduce cooling season relative humidity in the lecture hall. The Continuous Commissioning results identified that installing a reheat coil for the existing AHU would completely solve the humidity problem. However, until a reheat coil could be installed, reducing total supply airflow with cold deck temperature has proposed as a short-term solution. The reduction of total supply airflow significantly improves comfort, minimizes or eliminates mold formation in the lecture hall. The

Continuous Commissioning process has turned this inefficient, humid lecture hall into a comfortable learning environment.

HVAC SYSTEM

This large computer lab of the Richardson Petroleum Building is an interior zone with a total floor area of approximately 2500 ft^2. The existing air handling unit (AHU) is a single-zone, single duct, constant air volume AHU serving only the lecture hall. The air handling unit has a pre-heat coil, a cooling coil, one supply air fan and a return air fan. The existing AHU schematic diagram is shown in Figure 1.

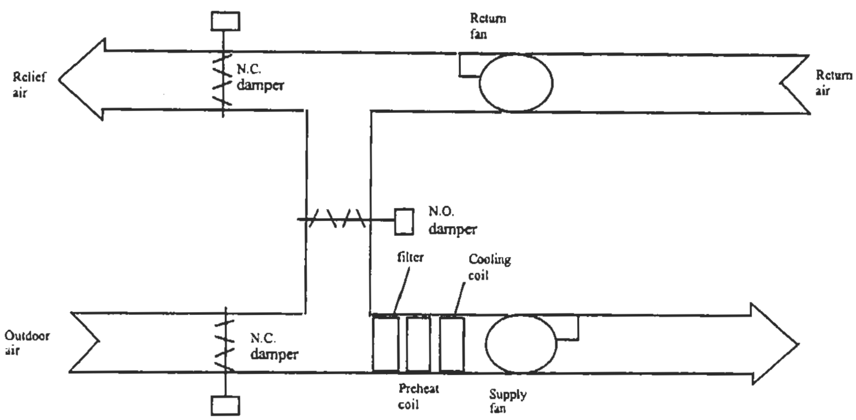

Figure 1 The Existing Single-duct, Single Zone AHU

The existing air handling unit is controlled by a direct digital control (DDC) system. A PID loop controls the chilled water valve and return air damper to maintain an adjustable room temperature. There is one temperature sensor on the wall in the room. The 5 horsepower supply fan, in the first floor, was designed to provide cooling airflow to the lecture hall. The total supply airflow was designed to be 3750 cfm and return fan airflow was designed to be 2650 cfm. The outside air intake was designed to range from 33 % (1,125 cfm) to 100% (3,750 cfm) of total airflow rate depending upon outdoor air temperature. The cooling coil capacity and the original design information is shown in Appendix A, Tables A.1- A.3.

AS-FOUND HVAC SYSTEM PERFORMANCE

The existing single-duct, single zone AHU and the lab conditions were inspected. The lab space conditions are shown in Table 1. The airside measurements and waterside measurements for the AHU are presented separately in Tables 2 and 3.

Different control combinations were used to investigate possible solutions to the problems. For example, the room temperature set point was changed several times and different signals were given to the return air fan, return air damper, and outside air damper. For each combination, airflow, temperature, humidity, and static pressure were measured. Tables 1- 3 below show the measured results of these different control combinations. The findings for the lab condition and the AHU are as follows:

1. Space Conditions

There was visible mold on the ceiling around the diffusers, as well as visible mold on the walls, tables and chairs. High humidity levels often resulted in mold, mildew, poor comfort and odor problems (Gatley, D., P., 1992; Shakun, W., 1992). The interior relative humidity affects the well-being and health of the occupants. Health complication from bacteria, viruses etc. increase with high humidity (Sterling, et al. 1985). People were upset above the space conditions, so all the computers and equipment were moved out although the classes in the lecture hall were getting ready to start.

The lab condition (see Table 1) was measured close to noon and in the early morning on July 9, 1999 and July 13, 1999. During the 11:52 a.m. measurements the ambient temperature was 85.7 °F and the ambient relative humidity was 57.5%, the room temperature was 72.2 °F and the room relative humidity level was 75%. At 8:55 a.m. the ambient temperature was 74.6 °F and the ambient relative humidity was 89.5%, the room temperature was 69.6 °F and the room relative humidity level was 80.4%.

Although the lab temperature varied from 69.6 °F to 72.2 °F it was close to design condition during both measurement periods. However, the lab relative humidity levels varied from 75% to 80.4%.

2. Discharge Temperature and Chilled Water Conditions of the existing AHU

The measured data from Table 1 and Table 2 showed the cold deck temperature was 67.6 °F and the measured air condition leaving the diffusers was 70.5 °F and 76.5% of the relative humidity. Since there was almost no sensible load on the system, the discharge air temperature is nearly equal to the room temperature set point. Most of the time the chilled water valve was barely open. Table 3 shows that the chilled water supply temperature was 43.9 °F and the chilled water return temperature was 71.3 °F. Table 3 also shows that there was little difference between the supply and return chilled water pressure.

3. Supply Airflow and Outside Airflow Rate and Conditions

The measured total supply airflow rate was 4490 cfm, which was 33% higher than the design value of 3,375 cfm for the lecture hall shown in Table 2. The measured outside air intake was 2,490 cfm, which 121% higher than the designed value of 1,125 cfm for that specific outside air condition at 10:25 A. M. on July 9, 1999.

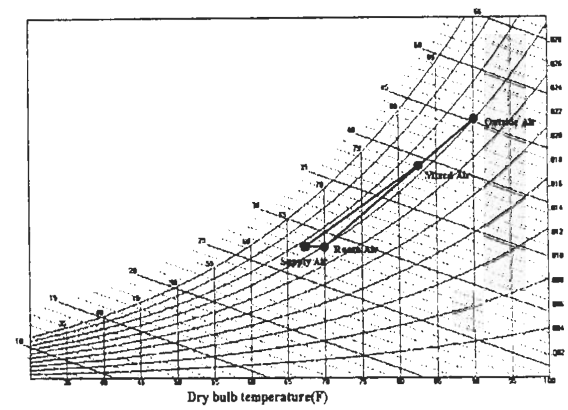

Figure 2 Psychrometric Process for Existing Supply Airflow Rate, Outdoor Airflow Rate and Air Condition

Table 1 Measured Air Condition Data for the Lecture Hall

Date, Time	7-9-99 11:52 AM	7-13-99 8:55 AM
Room Temperature (°F)	72.2	69.6
Room Humidity (% RH)	75.0	80.4
Room Dew Point (°F)	64.4	63.4
Supply Temperature from Diffuser (°F)	70.5	N/A
Supply Humidity (% RH)	76.5	N/A
Supply Dew Point (°F)	63.6	N/A
Outside Temperature (°F)	85.7	74.6
Outside Humidity (% RH)	57.5	89.5
Outside Dew Point (°F)	69.4	71.0

Table 2 Measured Airside Data for the AHU

Return Fan: on; Return Damper: fully open; Outside Air Damper: fully open (signal)				7-9-99 10:25 AM
	Supply	Mix Air	Return Air	Outside Air
Dimensions	24" x 24"	20" x 36"	14" x 34"	N/A
Air Flow (CFM)	4450	4530	2000	2490
Velocity (FPM)	1180	1091	750	N/A
Humidity (% RH)	83.0	72.3	70.0	57.5
Temperature (°F)	67.6	73.0	72.1	85.7
Dew Point (°F)	62.9	64.0	62.0	69.4
Static Pressure	+0.75	-0.19	-0.19	-0.10

Consider above field measurement shown in Tables 1-3 that before reducing the supply air and outside air flow rates, the conditions for the psychrometric process analysis shown in Figure 2 are as follows: (1) the supply air temperature: 67 °F; (2) outside air condition: 90 °F and 70% RH; (3) outside airflow fraction: 55% (outside air intake: 2490 cfm; total supply air: 4490 cfm); (4) return air: 70 °F and 70% RH. The measured data from Table 4 and Table 5 showed that there was not much heat gain load from the lecture hall, the discharge air temperature is almost equal to the room temperature set point and the cold deck temperature (around 65 °F) is far above the dew point (58 °F) for the mixed air. This is the reason why the chilled water valve was barely open for most of the time, and the mixed air moisture content from outside air intake and return airflow could not be dehumidified by the cooling coil. This operation could not control humidity levels in the lecture hall. The Figure 2 shows the psychrometric process for existing supply airflow rate, outdoor airflow rate and air condition.

Table 3 Measured Chilled Water Data for the AHU

Chilled Water Return Control Valve (spring range: 1-13 psi, normally open): 8.2 psi			
	Morning Pressure (psi)	Afternoon Pressure (psi)	Temperature (°F)
CW Supply to Coil	73.5	75.0	43.9
CW Return from Coil	74.5	75.0	71.3

4. Existing AHU

The single zone central air handling system in this lecture hall contains the supply and return fans, pre-heating/cooling coils, and the filter section, along with the exhaust relief damper and outside air intake dampers. The fan is located downstream of the coil. Dampers are used to facilitate mixing of return air and outside intake air. Figure 1 shows the single-zone air handling draw-through unit configuration. Of the all air handling systems, the single-zone air handling system is perhaps the simplest and most common type of system. The unit is controlled to respond to the room conditions as indicated by the room thermostat (see Figure 1). No reheat coil was designed for this system, so dehumidified primary air or recirculated room air could not be reheat if the cooling coil was opened for humidity control.

The original design data of the AHU (shown in Appendix A, Tables A.1- A.2) and the measured air condition data (shown in Table 1-3) both proves the original mechanical engineering design data about the cooling capacity and supply airflow volume is adequate for actual sensible load, but it not for latent load. This single-zone AHU system without reheat coil may be good in a hot dry climate, but is not adequate in a hot and humid climate because the AHU system can not handle space humidity level, particularly, a space does not have sufficient heat gain. Different geographic location has different ratio of latent load to sensible load and different AHU type has different advantage in different geographic location. The ratio of latent load to sensible load should be considered as a criterion to design the HVAC system and select equipment for different climates (Harriman, Plager and Kosar, 1999).

5. Operation and Control Sequence

The original engineers HVAC sequence of operations were obtained in order to compare current measurements and operation with the original design intentions. The existing single-duct, single zone AHU started and stopped on a time of day basis. When the system was energized, the supply and return fans was energized and the minimum outside air damper would open. When the chilled water system is disabled, space sensor would modulate the maximum outside air, return air, exhaust air and pre-heat control valve in sequence to maintain the lecture hall conditions. When the chilled water system was enabled, the maximum outside, return and exhaust air dampers would remain in their normal positions and space sensor would modulate, in sequence, the chilled water and pre-heat coil control valves to maintain lecture hall conditions. There was no humidity control sequence in the sequence of operation for the lecture hall.

6. Various Control Combination Investigation

As mention above, varying control sequences for AHU was tested to collect the problems. Tables 4-6 summary the measured results.

The outside air damper was leaking badly even when it was fully shut. Table 4 shows whether or not the outside air damper was fully open, the outside airflow intakes almost kept the same values. When the outside air damper was fully closed, shown in

Table 5, the mix air temperature (84 °F) was 2 degrees F less than the outside air temperature (86 °F). The badly leaking outside air dampers were sticky and in near full open positions. No matter what signal the EMCS commanded to the outside air dampers, the outside air dampers could not respond, resulting in lots of outside airflow to the lecture hall. Table 6 also concurs this finding.

Table 4 Measured Air Flow for Different Conditions of the AHU

						7-13-99 1:40 PM
Return Fan	Signal to Return Damper (% open)	Signal to Outside Air Damper (% open)	Mix air flow (CFM)	Mix air velocity (FPM)	Return air flow (CFM)	Return air velocity (FPM)
Off	100	100	4420	1040	2160	810
Off	100	100	4300	N/A	2050	N/A
Off	100	0	4400	N/A	2080	N/A
Off	50	0	4030	N/A	N/A	N/A
Off	15	0	2860	N/A	N/A	N/A
Off	0	0	2780	N/A	N/A	N/A

Table 5 Measured Air Conditions for the AHU

Return Fan: off; Return Damper: fully closed; Outside Air Damper: fully closed		7-13-99 1:40 PM
AHU	Mix Air	Outside Air
Temperature (°F)	84.0	86.0
Humidity (% RH)	67.0	61.0
Dew Point (°F)	70.5	71.5

LONG-TERM AND SHORT-TERM APPROACHES

From the site visit and comparing the design information and the measured data, we come the following conclusions and commissioning activities:

1. Conditions After Reducing Supply Airflow Rate

The actual total supply airflow for the entire computer lab, 4,490 CFM, was 33% higher than the design airflow value of 3,375 cfm and actual outside air intake was 2,490 cfm, which 121% higher than the designed value of 1,125 cfm. In order to dehumidify supply airflow moisture content level and then make the space relative humidity around 55%, the two things could be done: one is to increase room sensible heat gain from lighting and equipment. The other is to reduce total airflow and outside air to the lecture hall so supply temperature would change. The original total airflow rate could not match the actual total supply airflow rate from Appendix A, Table A.2. The reduced total airflow and outside air rates will significantly improve the room relative humidity conditions [Liu, et.al., 1996]. If the total airflow rate is reduced from 4,490 cfm to 3,750 cfm or lower

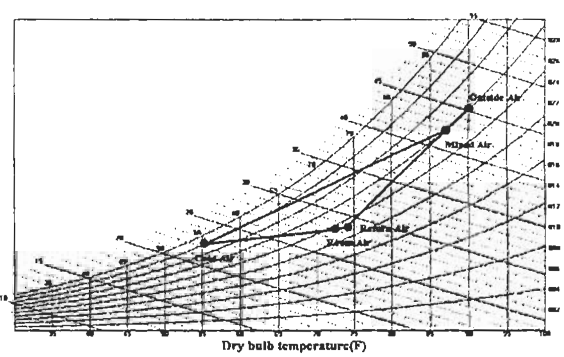

Figure 3 Psychrometric Process After Reducing Supply Airflow Within Typical College Station Weather

(2,150 cfm), the calculations show that the lab supply air temperature would be reduced and the room relative humidity level would fall to below to 58% RH. The impacts of the reduced airflow rates may be seen in the Figure 3. We temporarily turned off the return air fan, and adjusted the return air damper and OA damper to reduce supply airflow shown in Table 7, the measured airside data for AHU after adjusted total airflow conditions. The measured data from Table 7 shows the moisture content in the lecture hall was decreased to 56% RH.

Of course, this total airflow reduction for AHU is a temporary or short-term approach to abate the humidity level in the lecture hall. In the long run, installing a reheat coil will completely solve the high humidity problem.

If the total airflow is reduce to above airflow rates and the cold deck temperature is controlled at a temperature around 55 °F, the lab relative humidity should be maintained around 55% RH. The moisture content in the lecture hall mainly depends on the amount of total airflow, it can receives at 55 °F and still maintains 72 °F room temperature, so excessive airflow can cause a high relative humidity level. Reducing the total airflow will improve the lab relative humidity level by causing the cooling valve to open in order to maintain room temperature setpoint at 72 °F [Liu, et.al., 1999]. This reduced total airflow psychrometric process is shown in Figure 3 within typical College Station weather.

Table 6 Measured Air Side Data for the AHU

Return Fan: off; Return Damper: closed; Outside Air Damper: closed (signal)					7-14-99 9:44 AM
AHU	Supply Air	Return Air to AHU	Outside Air	Total Returning and Release Air	Mix Air
Dimensions	24" x 24"	14" x 34"	N/A	23" x 19"	20" x 36"
Total Air Flow (CFM)	N/A	620	N/A	250	3490
Average Velocity (FPM)	N/A	N/A	N/A	N/A	820
Temperature (°F)	65.5	73.3	79.2	69.6	79.1
Humidity (% RH)	85.0	68.7	92.3	70.0	72.0
Dew Point (°F)	61.2	63.2	76.2	59.6	69.4
Static Pressure	+0.36	-0.9	N/A	0.0	-0.9

Table 7 Measured Air Side Data for the AHU after Adjusted Conditions

Return Fan: off; Return Damper: fully closed; Outside Air Damper: shut (signal)			7-15-99 11:55 AM
AHU	Mix Air	Return Air	Outside Air
Dimensions	20" x 36"	14" x 34"	N/A
Air Flow (CFM)	2,150	530	1,620
Humidity (% RH)	74 %	56 %	76 %
Temperature (°F)	83.0	72.0	87.5
Dew Point (°F)	66.0	64.4	69.8
Static Pressure	-1.44	-1.43	N/A

2. Installing a Reheat Coil

There was no reheat coil in the single-duct, single-zone, and constant air volume system. Therefore the room humidity level could not be controlled although the room temperature was at 70 – 72 °F (see Table 1), especially when there was little sensible load from the room. A reheat coil will raise the supply air temperature from 55 °F to 70 °F or around 70 °F and will maintain good humidity levels under all loads, should be installed or inserted in the duct system for dehumidification purposes [Liu, et. al., 1996]. To cool the air below its dew point and remove latent load in the form of condensed moisture, as well as sensible heat, the mean surface temperature of the cooling coil must be below the dew point temperature of the air. This type of system provides a constant quantity of supply air, delivering it warmer or cooler depending on whether the lecture hall is calling for heating or cooling [Liu and Claridge, 1994]. The thermostat in the lab controls the reheat or heating system when the temperature falls below thermostats setting. Figure 4 shows the psychrometric processes after installing reheat system.

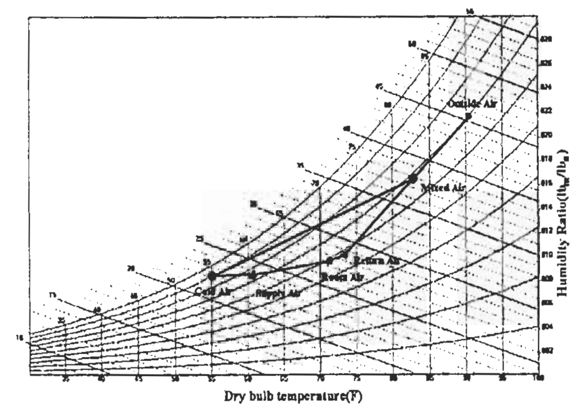

Figure 4 Psychrometric Processes After Installing Reheat System

3. Operation Sequence for Load Control

The HVAC system was supposed to be delivery air at a variable cold air temperature and at variable outside airflow rates designed to offset the maximum load (cooling and heating) and control humidity in the lab. A return air relative humidity sensor should be installed to measure the lecture hall humidity level. Variable low cooling coil temperature and variable outside airflow rates are used to reduce the space humidity to low values because space humidity is largely a function of coil temperature and outside airflow rates. On a call for heating in the lecture hall after the reheat coil was installed, the thermostat reduces the control line pressure allowing the normally open (N.O.) reheat valve to open and increase the supply air temperature. On a call for cooling, the thermostat increases its control line pressure, first modulating heating valve to the closed position and then opening the normally closed (N.C.) cooling valve to increase the cooling coil output. At part-load, total airflow remains at a constant low temperature of 55 °F, the heating valve is varied to maintain the desired room temperature, supplying only the heating, or cooling energy needed.

4. Monitoring Space Conditions

Stick-on temperature and humidity sensors were used to monitor the room condition during airflow rate adjustment for the lecture hall. Monitors (Chart) for CO_2, temperature, and humidity were placed in the room to record room conditions on July 15, 1999 and July 16, 1999. The CO_2 meter was used to monitor the lecture hall CO_2 level to keep the indoor concentration from exceeding a guideline level for concentrations of air contaminants (ASHRAE standards 62-1989).

5. Reheat Coil Calculation and Design

Reheat coil cooling capacity was calculated and the reheat coil was designed and ordered. The reheat coil was installed in late August 1999. Preliminary design data for the reheat coil is shown in Appendix B, Table B.1. The malfunctioned OA damper and an actuator of return damper were fixed before August 1999.

COMMISSIONING RESULTS

In the long run, installing a properly sized reheat coil will completely solve the humidity problem. For the short term, until a reheat coil could be installed, reducing supply airflow to produce colder discharge temperatures was sufficient to reduce humidity levels and maintain comfortable temperature.

We repeated the measurements shown in Table 8 on July 15, 1999, July 16, 1999 and September 9, 1999 and all the office workers said that they were satisfied with the lab conditions. The measured room conditions before September 9, 1999 is from the short-term approach, reducing supply airflow with colder discharge temperature, and the measured room data on September 9, 1999 is from the long-term approach, installing a reheat coil for control of space temperature. Table 8 presents the situation for the lecture hall.

Table 8 Measured Data for the Lecture Hall

Lecture Hall	7-15-99 2:52 PM	7-16-99 9:32 AM	9-9-99 11:00 AM
Room Temperature (°F)	74.0	68.0	72.0
Room Humidity (% RH)	51.0	58.0	56.0

CONCLUSIONS

This paper describes Continuous Commissioning measures for the humidity control of a large lecture hall, the Richardson Petroleum Building. In the case study building, the main reasons identified for the room high humidity in the lecture hall were no reheat coil in the existing AHU as well as the excessive airflow. A reheat coil or heating coil in the single-duct, single-zone, constant air volume system is important in the relative humidity control, especially in the hot and humid regions of U.S. where the latent cooling load is a large percentage of the total load.

Installing a reheat coil completely solved the humidity problem in the lecture hall. However, before a reheat coil could be installed, reducing supply airflow with colder discharge temperature also proved to be sufficient. The airflow reduction and resetting room control program decreased the room relative humidity from 70% to 58%. Above long-term and short-term approaches had been applied to the existing AHU to turn this inefficient humid lecture hall to a comfortable learning environment.

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APPENDIX A

Table A.1 AHU Design Data for the AHU

AHU	Fan (rpm)	Air Flow (CFM)	Total Static Pressure (W.G.)	Outside Air (CFM)	Max Outside Air (CFM)	Fan Size (HP)
Unit Data	1,087	3,750	2.5”	1,125	3,750	5

Supply Fan Motor: 1,745 rpm

Table A.2 Cooling Coil Design Data for the AHU

	Air Flow (CFM)	Capacity		DP (W. G.)	E.A.T		L.A.T	
		S.H	T.H		D.B.	W.B.	D.B.	W.B.
Cooling Coil Data	3,750	121,500	166,300	2.2”	82.7	67.0	52.7	52.3

Table A.3 Return Fan Design Data for AHU

	Air Flow (CFM)	Static Pressure (W.G.)	R.P.M.	DP (W.G.)	Motor Size (HP)	Motor R.P.M.	Interlock with
Return Fan	2,625	1 ¼”	1,460	1.9”	1	1,750	AHU

APPENDIX B

Table B.1 Reheat Coil Design Data for Lecture Hall

Air Flow (CFM)	Capacity (Btu/Hr)	Enter Air (°F)		Exit Air (°F)	Enter. Water (°F)	Exit Water (°F)	Air DP (W. G.)	WTR. DP (W. G.)	WTR. Flow (GP.M)	Air Duct Size to the Coil
		D. B.	W B.							
4,400	142,560	55	52	85	150	130	0.03”	0.1”	14	24”x24”