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Exploring Maximum Humidity Control and Energy Conservation Opportunities with Single Duct Single Zone Air-Handling Units

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Abstract: Humidity control for single-duct single-zone (SDSZ) constant volume air handling units is known to be a challenge. The operation of these systems is governed by space temperature only. Under mild weather conditions, discharge air temperature can get much higher than the space dew point and the dehumidification capability of the system is diminished. Buildings served by this type of air handler often experience exceptionally high humidity levels under humid weather conditions. Many potential solutions and improvements exist. However, these solutions require system modifications or upgrades and therefore are less attractive to some facility owners.

A Critical Humidity Control Program (CHCP) was developed to change the normal control sequence of the air-handling units during high humidity periods to help improve the moisture removal capability of the system. The program was not designed to solve the problem completely, but the overall humidity levels can be lowered and controlled within a reasonably low range (58% - 65%) for a significant part of the high humidity seasons. This approach is relatively easy to implement and does not require any hardware changes.

This paper also summarizes various potential solutions to improve humidity control for SDSZ units. The advantages and disadvantages for each solution are compared.

1. INTRODUCTION

A typical single-duct single zone unit consists of a preheating coil, a cooling coil and a constant-speed supply fan (Figure 1). This type of units is popular in many facilities due to its simplicity in both operation and maintenance, and low initial costs.

The preheating valve and the cooling valve are directly controlled by the space temperature. A 2-4ºF offset between the heating and the cooling setpoints is typical to prevent simultaneous heating and cooling.

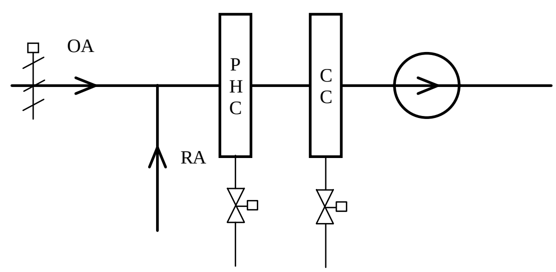

Fig.1. Typical single-duct single-zone unit

The supply air temperature, as called for by space temperature control sequence, depends solely on the heating/cooling loads of the conditioned space. Under low cooling load conditions, the supply air temperature leaving the cooling coil is relatively high, and the amount of moisture being removed is therefore quite limited. If the discharge air temperature is lowed in order to control the space relative humidity, then the space temperature will drop below the cooling setpoint. Therefore, in a cool and humid day, the humid air passes through the coils with little moisture being removed, resulting in a high humidity level (over 65% and as high as 80%) in the conditioned space.

2. POTENTIAL SOLUTIONS

A few potential solutions exist and are commonly used in practice for better humidity control, such as reheat, pre-treating outside air, variable volume control, or desiccant dehumidifier.

2.1 Reheat coil

One of the most often-seen solutions is to add a reheat coil to the existing unit, as show in Fig.2. In a unit with a reheat coil, the cooling coil temperature can be controlled to low enough to remove moisture from the mixed air, therefore controlling the space humidity level. The air is then reheated to maintain the space temperature. This solution provides good humidity control, but involves major equipment modifications (adding the reheat coil and all associated piping). The energy cost is also raised significantly due to simultaneous cooling and heating.

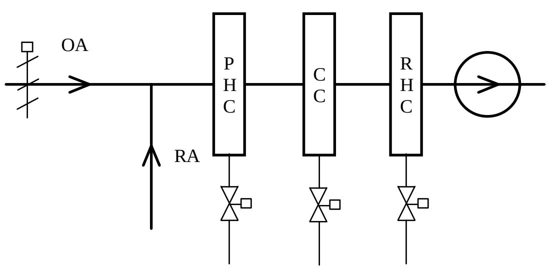

Fig.2. Rooftop units with added reheat coil

2.2 Dedicated outside air handling units

Another alternative is to pre-treat the outside air for all the single-zone units in each building, with a dedicated outside air handling unit. This solution provides good humidity control if the major source of the moisture comes from the outside air. However, it also involves major ductwork changes.

2.3 Variable speed control of supply airflow

A variable speed drive (VFD) saves fan power by slowing down the fan speed and delivering less supply air to the conditioned space during partial load conditions. It also helps improving space humidity control. For a unit with VFD, the cooling coil temperature can be controlled to low enough (i.e., 55 °F) to remove moisture from mixed air. The space temperature is controlled by adjusting the supply fan speed. However, if the space cooling load drops below the point where the supply airflow reaches the minimum flow required for proper ventilation, the supply air temperature has to be raised to keep the space temperature from dropping below the setpoint. Therefore, humidity control under low cooling load conditions is still limited with this solution. Besides, the installation of VFD device is also relatively expensive.

2.4 Variable speed control with reheat

The limitation of the variable speed control mentioned above can be removed if a reheat coil can be installed after the cooling coil. The cooling coil valve will be controlled to maintain the temperature after the cooling coil at certain low temperature (e.g. 55ºF). The space temperature is controlled by the fan speed. When the fan speed approaches a predefined minimum speed, the reheat coil valve will be modulated to control the space temperature. This solution provides the best humidity control and the best energy performance. However, the initial cost for system modification is also the highest.

2.5 Supply air bypass control

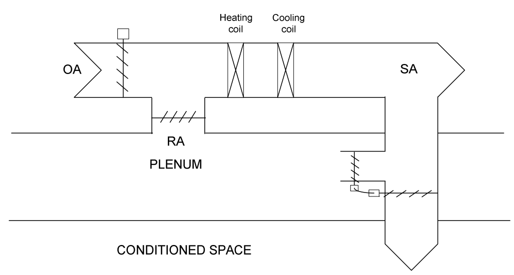

Fig. 3. Unit with supply/bypass damper

This solution is an alternative to the variable speed drive. The proposed retrofit only involves the addition of an automated supply / bypass damper, as shown in Fig. 3. The bypass damper and the supply air damper should be linked together so that one opens when the other closes. During the cooling mode, the discharge air temperature should be maintained at a low temperature setpoint (e.g. 55ºF). As the space cooling load decreases, the bypass damper should be gradually open (and the supply damper gradually close) to introduce less primary air into the conditioned space. This approach will not save fan power, however it should provide much better humidity control. This solution has the same limitation as the variable speed drive at relatively small costs.

2.6 Desiccant dehumidification

Both liquid and solid dehumidification equipment are commercially available. However, the feasibility of this solution is limited by the complication of the operation and maintenance.

2.7 Complementary portable dehumidifier.

Commercial portable dehumidifiers are readily available in different capacities. The main draw back of this solution is the added maintenance tasks for the maintenance staff.

All potential solutions discussed above involve the addition of hardware or system modifications, and are therefore less attractive to some facility owners due to the associated initial costs or other physical limitations. In an effort to seek a means to mitigate the humidity problems without significant system modifications, a "Single Zone Unit Critical Humidity Control Program"(CHCP) was developed.

3. CRITICAL HUMIDITY CONTROL PROGRAM

First of all, CHCP was not developed to solve the poor humidity control of the single zone units – a prototype single zone unit is simply not capable of effective humidity control. Rather, CHCP is a modified control sequence designed to relieve the high humidity problems suffered by many facilities during humid weather conditions. The basic concept behind the program is "reheat" in nature - lower the cooling coil temperature to remove moisture and then reheat the air to retain the temperature control. However, this is done without a reheat coil. CHCP is only enabled upon sensing a high humidity level in the conditioned space.

Three relative humidity levels are defined to indicate the severity of the space's humidity problems. When the space humidity is below 55%, the humidity level (HL) is defined as "Normal"; it is defined as "High" when the space humidity rises above 65%; when the relative humidity is above 75%, HL is designated as "Severe".

When the humidity level is "Normal", a single-zone unit is operated under the normal control sequence, i.e., the cooling valve and the heating valve are sequenced to maintain the space temperature setpoint.

When the humidity level is "High", a single-zone unit is operated under the First Stage Critical Humidity Control mode. The operation of the unit is divided into a cooling cycle and a neutral cycle. Assume the space temperature is at the cooling setpoint (e.g., 74°F) and rising.

Step 1: Once the space temperature reaches the cooling setpoint, a cooling cycle is started. Chilled water valve is fully open no matter what cooling load is in the conditioned zone; the preheat coil valve is fully closed. Since the chilled water valve is fully open, the supply air temperature after the cooling coil may well drop below 55°F, which effectively removes moisture from air.

Step 2: The space temperature shall start to drop, as well as the relative humidity. The chilled water valve will maintain the fully open position until the space temperature reaches the heating setpoint (e.g., 71°F).

Step 3: Once the space temperature drops below the heating setpoint, a neutral cycle is started. Both the chilled water and the hot water valves are fully closed, and the outside air damper is also closed. Untreated re-circulating air is supplied to the conditioned space. No moisture is being removed during a neutral cycle.

Step 4: Depending on the space's cooling load, the space temperature shall rises back to the cooling setpoint slowly or quickly. Once the space temperature rises above the cooling setpoint again, a new cooling cycle (step 1) will start again.

The reason the CHCP sequence can help lower the relative humidity is because it has at least a fraction of time that is involved in moisture removal. Under normal operation scenario, when the space's cooling load is relatively low, the zone temperature control may call for a 65°F supply air temperature. If this temperature is higher than the dew point of the mixed air, the cooling coil does not remove any moisture at all. The CHCP control sequence splits the process into periodic cooling and neutral cycles, whereas the cooling cycle supplies low temperature (e.g., 55°F) air to remove moisture and the neutral cycle supplies medium temperature (e.g., 72°F) air to "reheat" the space.

Unfortunately, this control logic does not work well for the cool and humid weather conditions. The problem is that if the space's cooling load is really low, the neutral cycle will dominate the periodic operation sequence and the moisture-removal cooling cycle is diminished. In fact, if the space has no cooling load at all, a 72°F supply air during the neutral cycle would never bring the space temperature back to the cooling setpoint, while the space humidity level keeps rising. That's where a Second Stage CHCP comes into play.

When the First Stage CHCP fails to lower the space's relative humidity and the humid level eventually reaches the "SEVERE" status, the Second Stage CHCP will be engaged. The Second Stage CHCP is similar to the First Stage CHCP, except for the neutral cycle. Under the Second Stage CHCP, the neutral cycle is replaced by a heating cycle. During the heating cycle, the hot water valve will be open to heat the space back to the cooling setpoint quickly. This will shorten the heating cycle so that the moisture-removal cooling cycle will be longer. Unlike the First Stage CHCP, the Second Stage CHCP control sequence effectively adds heating energy to the space to force the unit switching back to the cooling cycle quickly. With the Second Stage CHCP, the space humidity level is lowered, and the human comfort is improved, but at the cost of increased energy consumption.

4. CASE STUDY

A state health facility consists of more than 30 buildings with a combined gross area of over 360,000 square feet. The majority of the buildings are served by single-duct single zone units described in this paper. The exceptionally high humidity in many residential and office buildings has long been a notable problem and has received many complaints. The relative humidity levels in some conditioned spaces constantly rose above 65% during the summer high humidity seasons and sometimes reached as high as 80%.

4.1 Implementation of CHCP

The CHCP control sequence had been implemented for all single-zone constant volume rooftop units in several buildings on the campus. With the help of the new control sequence, the overall humidity levels within the buildings can be lowered and controlled within reasonably low range (55% -65%) for most of the warm seasons when the cooling loads are relatively high. However, during cool and high humidity weather conditions, or if the space doesn't demand cooling at all, the relative humidity in some spaces may still reach above 70% for short periods of time.

In order to demonstrate the effectiveness of the CHCP control sequence, operational data were trended and collected for a few units before and after the (First Stage) CHCP was engaged. shows the modulation of the chilled water valve and the control of the humidity for an example unit. Before the CHCP was engaged, the chilled water valve opening varies from 40% to 90% daily, in response to the daily cooling load variation. As a result, the space relative humidity was cycling between 60% and around 75% daily. After the CHCP control mode was engaged at around 10 a.m., Aug. 15, the space humidity was effectively controlled within the range between 55% and 65%. Meanwhile, the space temperature swings between the heating setpoint and the cooling setpoint, as shown in Fig.5.

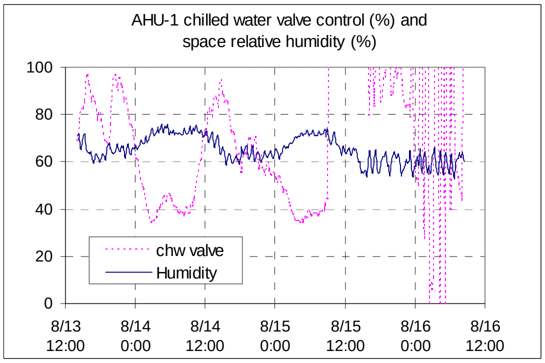

Fig.4. AHU-1 chilled water valve control and space relative humidity

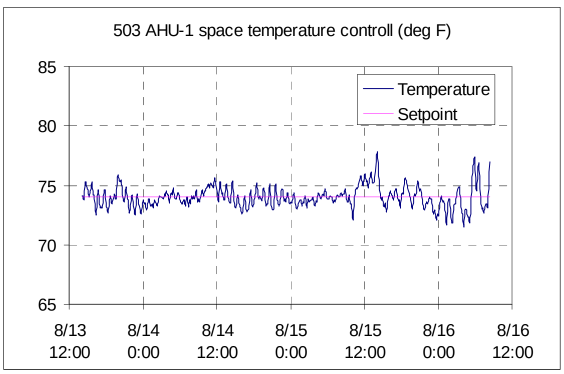

Fig.5. AHU-1 space temperature control

4.2 Supply air reduction

It was noticed during many Continuous Commissioning® project that, many single-zone constant volume units are oversized for their current functionality. The supply fan of a single zone unit is sized to provide enough airflow for peak cooling conditions. If the fan is oversized, the supply air flow is higher than necessary, resulting higher supply air temperature, which in turns causing higher space humidity.

In the same health facility mentioned above, trend data indicated that supply air temperatures of those units were all above 60°F (above 65ºF for some units), even during the peak cooling load conditions when outside air temperature was above 100°F, while the space temperatures setpoints are maintained. For the same peak cooling load conditions, the space cooling demand could also have been satisfied with less supply airflow, but at lower supply air temperatures. The reduction of the supply airflows will save fan powers and the lowered supply temperatures will improve humidity control. Ideally, the supply airflow should be reduced to the amount so that the corresponding supply air temperature is around 55°F (or even lower) for the peak cooling load weather conditions.

On each single-zone rooftop unit on the facility, a return-duct balancing damper can be adjusted to control the supply airflow. Fig.6 shows the change of supply air temperature on RTU-3, as a result of supply airflow reduction at around 1 p.m., Aug. 17. Notice that the time period shown in Fig.6 was the hottest days in the year according to the National Weather Service data. The supply airflow was reduced to around 75% of the original, and as a result, the lowest supply temperature was lowered from around 60°F before the airflow adjustment, to around 55°F after the airflow adjustment.

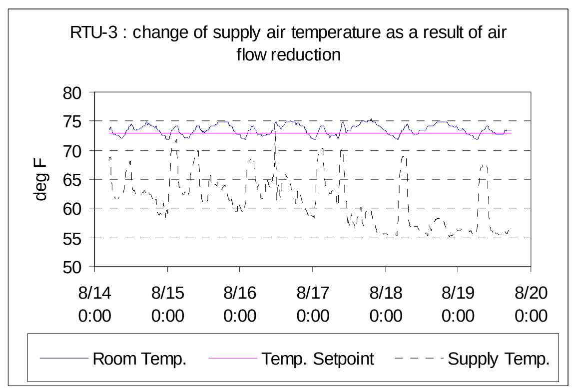

Fig.6. Change of supply air temperature as a result of airflow reduction

From the trended data for each unit during the peak cooling load period, the lowest supply air temperature for each unit can be determined. The target airflow can also be calculated. For example, if the lowest supply air temperature was 60ºF for the peak cooling load condition and it is desirable to lower it to 55ºF. The maximum temperature drop across the cooling coil will be increased from 15ºF (assume a 75ºF mixed air temperature) to 20ºF. In order to provide the same amount of cooling to the conditioned space, the supply airflow can be reduced to 75% of the original.

A more energy efficient alternative to reduce the supply airflow of an AHU is to reduce the fan speed through downsizing the motor pulleys. The fan speed (and the supply airflow) is proportional to the motor pulley size. The new motor pulley sizes can be determined from the airflow ratio (new airflow over the original airflow).

Table 1 shows the comparison of the measured fan power for three different scenarios. The first scenario is the base scenario before any change; under the second scenario, the return damper was partially closed to reduce the total air flow to 75% of the base scenario; under the third scenario, the total air flow was reduced to the same amount by changing the motor pulley to a smaller size while keeping the return damper fully open. The fan power of scenario # 3 was only 1/3 of the base scenario.

With the significant savings in fan power, retrofitting the fan motor pulley will typically have a very quick payback (1~2 months).

5. CONCLUSIONS

The humidity control capabilities of single-zone units are by design very limited. Many potential solutions exist but most involves expensive system modifications (equipment, ductwork, or piping). With the help of the Critical Humidity Control Program, the space humidity problems can be mitigated to a certain level. Supply airflow reduction by slowing down fan speeds will not only improve the humidity control capability of a single zones unit, but also save the fan power significantly.

Table 1. Motor pulley downsizing power measurements

	Scenario #1			Scenario #2			Scenario #3		
Phase	Red	Blue	Black	Red	Blue	Black	Red	Blue	Black
Volts	118.8	119.6	118.3	119.0	119.8	118.1	119.8	120.8	119.3
Amps	12.2	12.5	12.3	10.0	10.3	10	8.32	8.55	8.24
KW	1.01	1.1	1.08	0.7	0.78	0.75	0.36	0.3	0.41
KVA	1.45	1.52	1.45	1.2	1.23	1.18	0.99	1.04	0.98
KVAR	1.0	1.04	0.96	0.94	0.97	0.9	0.93	0.96	0.89
PF	0.71	0.72	0.75	0.6	0.6	0.63	0.36	0.4	0.42
Total kW	3.19			2.23			1.07		
Fan DP (inch)				1.49			0.26		
Air velocity (ft/min)				479			473		