REPORT

ON THE IMPACT OF

EVAPORATIVE COOLING

VIA HUMIDIFIER

WATER SPRAYS

REGARDING

ENERGY USE

A P S
ANALYSTS OF PNEUMATIC
SYSTEMS LIMITED
PHONE (905) 640-2333
FAX (905) 640-2444
http://www.apscontrols.ca

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EXERCISING COMMON SENSE

Before a person can improve a system's energy performance, or even calibrate an existing system, it is necessary that he or she acquire a full understanding of the mechanical and control components' relationships. **This means that person completely understands a logical plan that they would employ if they were manually operating the mechanical system.**

THE PLAN

1) Look at the fan system drawing on the next page and consider operating it manually.

2) Relating to this case study you must have access to all relevant values for temperatures and humidity. If you do not know the temperature or the relative humidity values, you can not determine the action required correcting an unacceptable condition.

3) You must be able to gradually open or close the dampers for fresh air, return air and exhaust air.

4) You must be able to gradually open or close the chilled water valve and the heating valve.

5) You must be able to start and stop the humidity spray pump

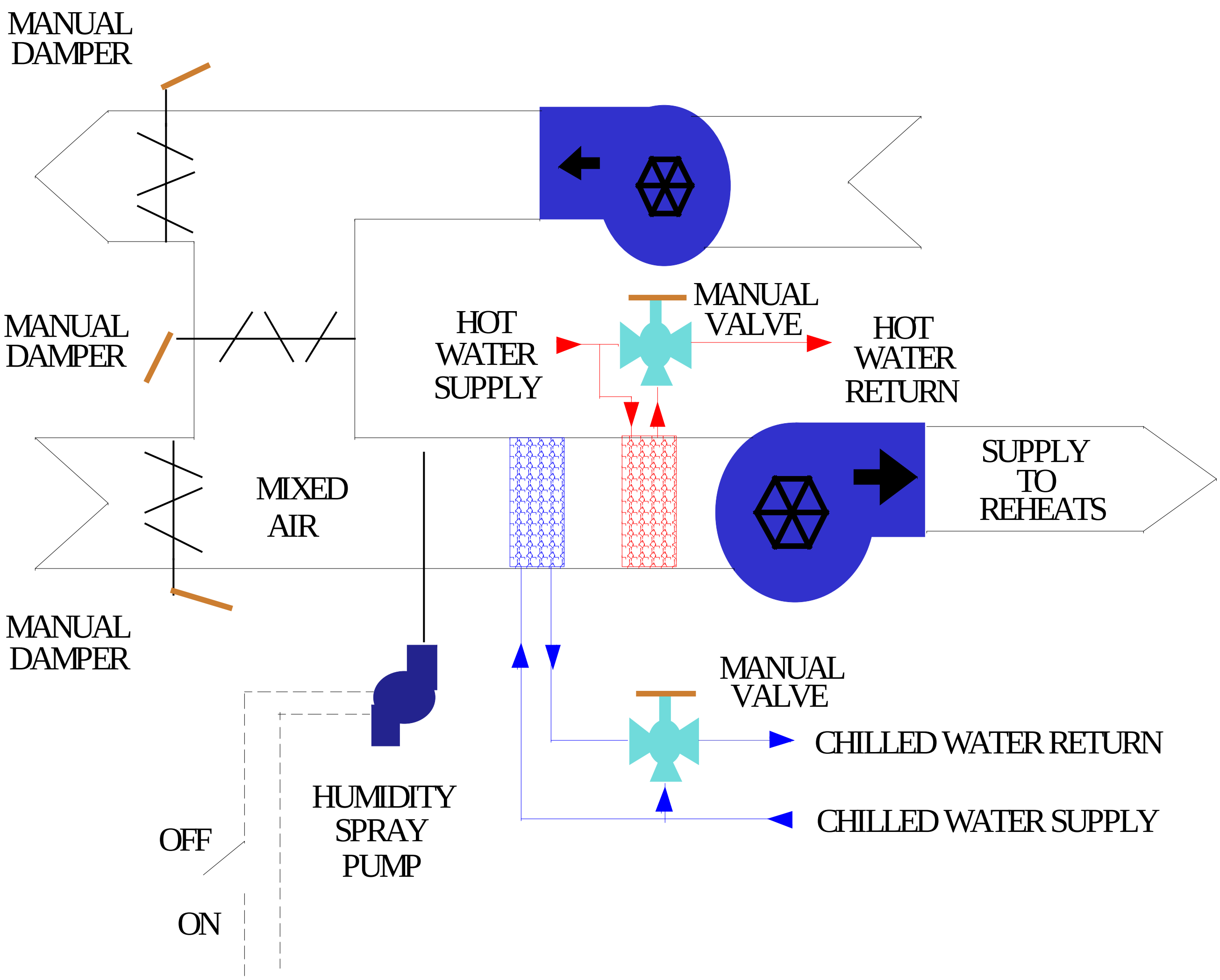
6) You must be able to start and stop the fan if it is not required at unoccupied times.

7) You must understand the impact that each function exerts overall on the system when employed to varying degrees.

When you understand all the interrelationships between the mechanical system and the building served, you are prepared to assess the control system.

Remember if the control system does not make the same mechanical adjustments as an intelligent human, given the same input values, there is something wrong with the control system.

MANUAL OPERATION OF FAN COMPONENTS



(1) How would one adjust the manual dampers to address minimum ventilation and free cooling requirements? What would be a typical target mixed air temperature?

(2) How would one control the manual cooling and heating valves?
What would be a typical target supply air temperature?
What condition would cause one to allow hot water to flow through the coil?

(3) How would one control the manual humidity spray pump?
What would be a typical target range normally be for return air relative humidity?

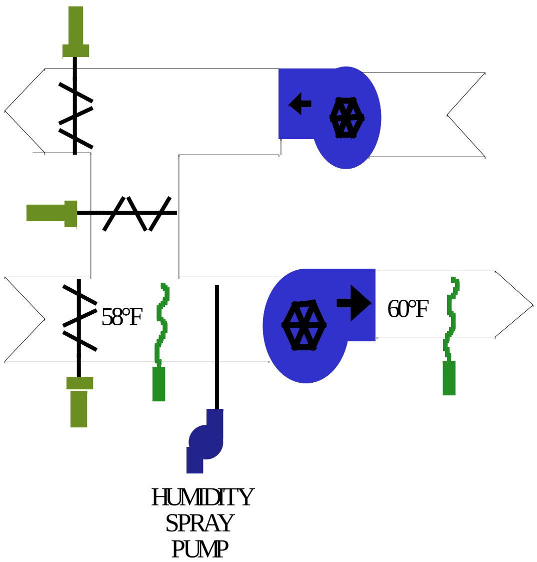
(4) If the mixed air is at the target value (example 58°F), and the heating valve is closed to the coil, what would the supply temperature do when the humidity sprays are activated?

(5) What action would typically be taken to maintain the target supply air temperature?

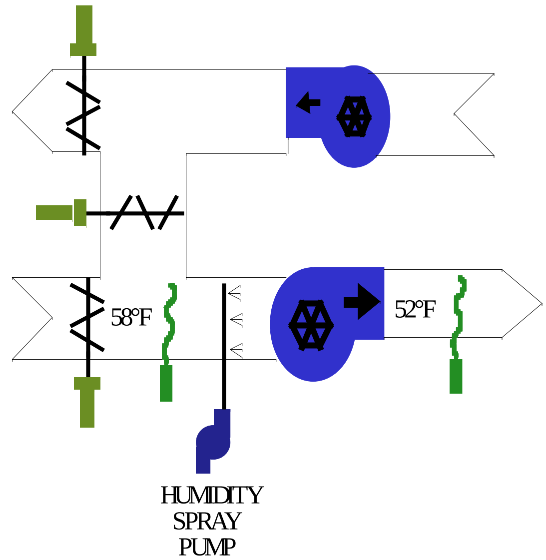
2.2

BEFORE CONTROL CHANGES

HUMIDITY SPRAYS OFF

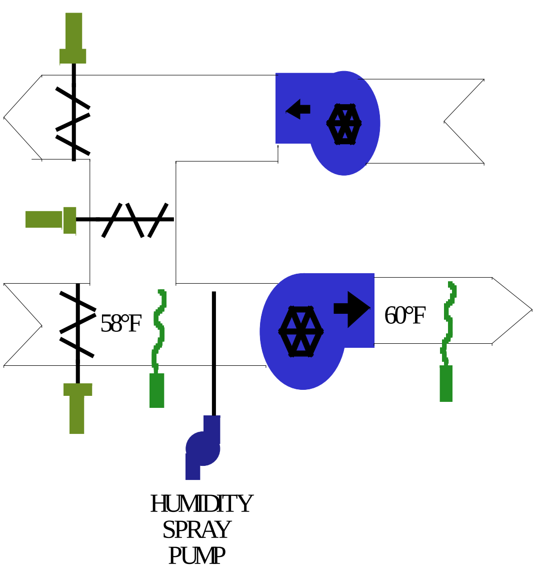


HUMIDITY SPRAYS ON

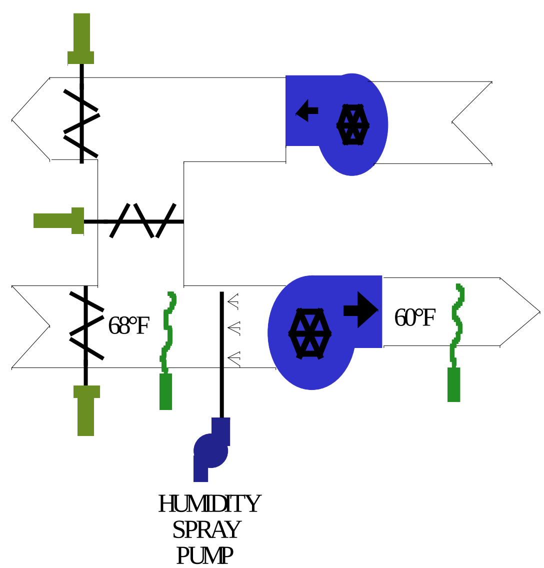


AFTER CONTROL CHANGES

HUMIDITY SPRAYS OFF



HUMIDITY SPRAYS ON



EFFECT OF EVAPORATIVE COOLING CAUSED BY HUMIDITY SPRAYS ON ENERGY USE

Many HVAC systems use water sprays to satisfy the humidity demands of the building based on the return air relative humidity level. Improper control reference points can result in large losses in energy to achieve the desired humidity levels. This report graphically illustrates how the losses occur, the magnitude of the losses and a control solution to the problem.

THE PROBLEM

Many systems control the mixed air to fifty-eight degrees Fahrenheit via averaging sensors in the mixed air plenum. The mixed air is drawn through the fan and forced to the occupied space.

If the humidity water sprays are not active, the normal discharge air temperature from the fan will be in the range of sixty degrees Fahrenheit. (Allowing for some heat gain from the fan.)

If the humidity sprays are active the supply air will drop to near dew point. This will cause the heating equipment to use more heat than while the sprays are not active. The heat, which has already been paid for, is being forced out the exhaust dampers via the mixed air control loop.

This report demonstrates the loss in one such system and clearly illustrates the savings created by causing the control logic to use heat contained in the return air for purposes of evaporating the moisture required for humidification. The system automatically raises the mixed air temperature to compensate for the cooling effect of the humidity sprays. This maintains a relatively constant supply air temperature while maximizing the use of energy already paid for in the building.

GRAPH #1

This graph uses the black line to illustrate the mixed air temperature, which was averaging about fifty-nine degrees Fahrenheit. The mixed air temperature was not affected by the status of humidity sprays; therefore, it remained relatively constant.

The blue line illustrates the discharge air temperature.

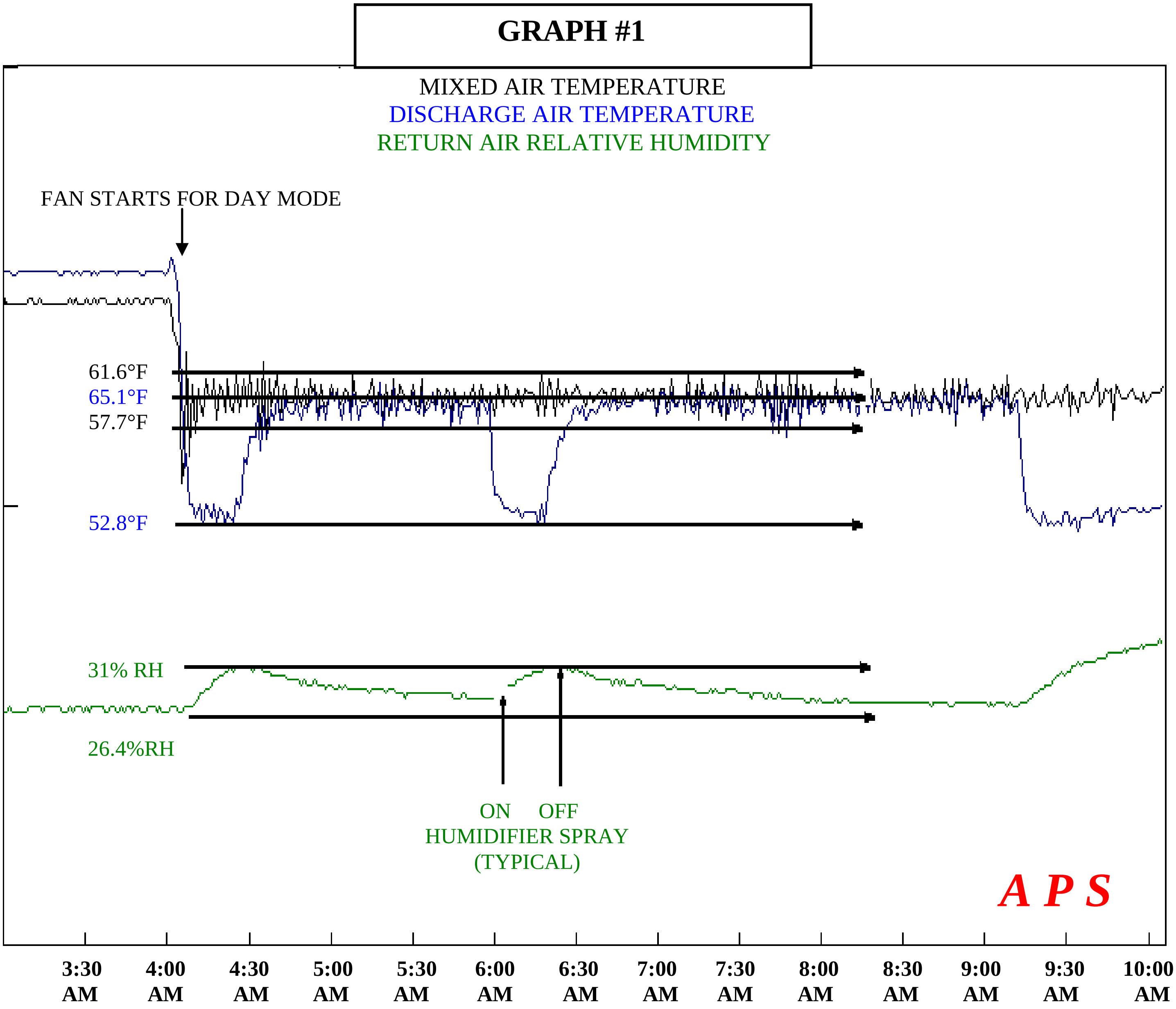
When the humidity sprays were not active the discharge air temperature averaged about sixty-three degrees Fahrenheit.

When the humidity sprays were active the discharge air temperature averaged about fifty-three degrees Fahrenheit.

The green line illustrates the return air relative humidity. This line indicates the run times for the humidity spray pump.

The energy loss is demonstrated by the dips in the discharge air temperature (blue line). At these times the heating system has to contend with a heating load approximately ten Fahrenheit degrees cooler than if the building did not have a humidification capability.

A control strategy was developed to allow the discharge air temperature controller to over-ride the mixed air controller to raise the mixed air temperature automatically when the humidity sprays were active. This was intended to keep the discharge air temperature relatively stable and remove the large dips in the discharge air temperature created by the spray activation. This would save energy and reduce the complaints of cold drafts by the occupants. The discharge air controller also retained its control over the fan system's chilled water valve for mechanical cooling.


GRAPH #1
MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY
FAN STARTS FOR DAY MODE
61.6°F
65.1°F
57.7°F
52.8°F
31% RH
26.4%RH
ON OFF
HUMIDIFIER SPRAY
(TYPICAL)
A P S
3:30 AM
4:00 AM
4:30 AM
5:00 AM
5:30 AM
6:00 AM
6:30 AM
7:00 AM
7:30 AM
8:00 AM
8:30 AM
9:00 AM
9:30 AM
10:00 AM

GRAPH #2

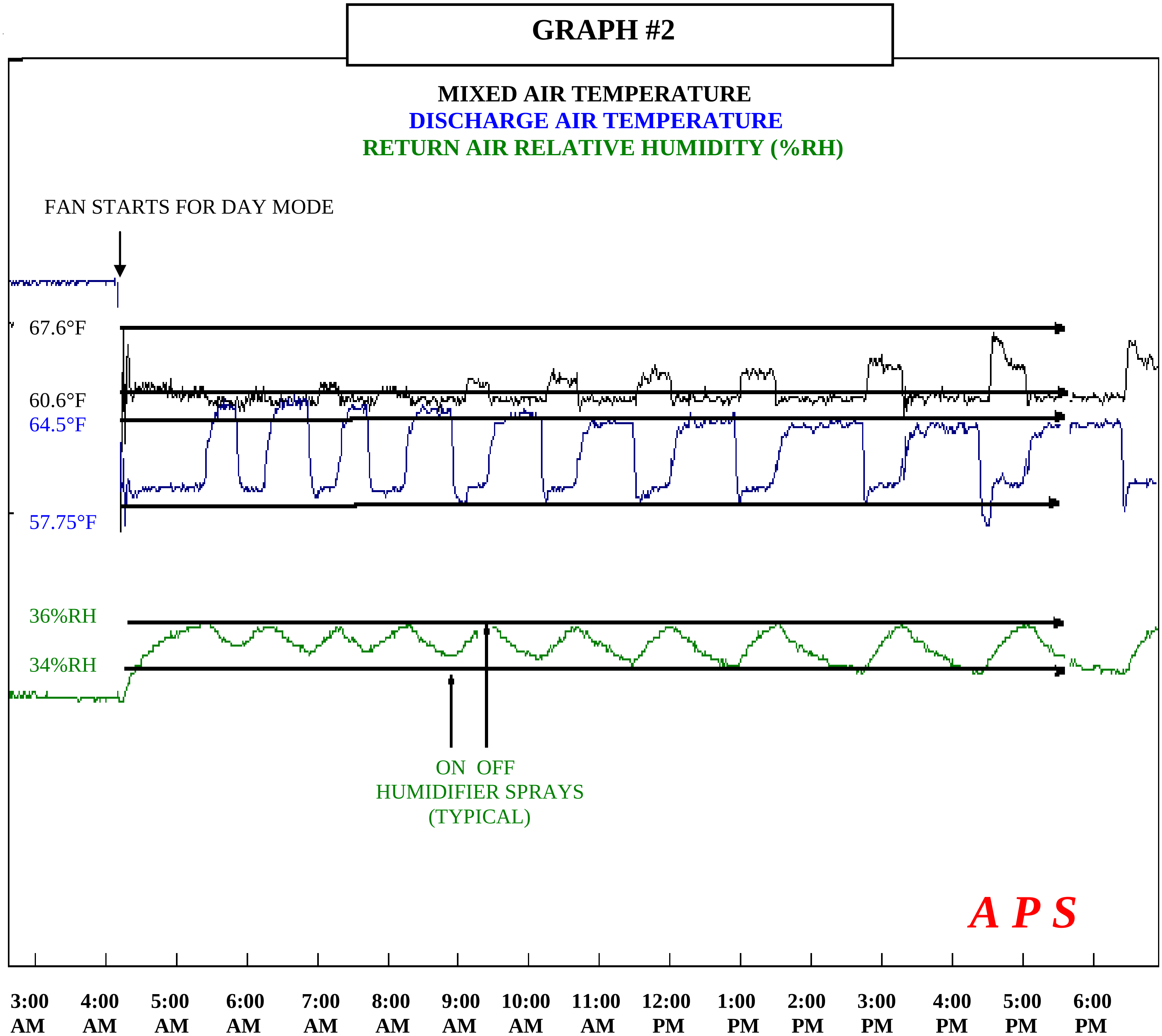
A pneumatic low selector was added to the circuit to allow the discharge temperature controller to automatically over-ride the mixed air controller; as well as control the cooling valve as per its original intent. The chiller is off during free cooling mode; therefore, the discharge controller will only have an effect on the mixed air dampers during the winter

The black line illustrates the mixed air temperature, the blue line illustrates the discharge air temperature and the green line illustrates the return air relative humidity.

The graph demonstrates that the change tended to correct the problem. The mixed air temperature tended to rise when the humidity sprays were active; however, the discharge air temperature still dropped an unacceptable amount when the sprays were active.

We had to increase the sensitivity (lower the proportional band) of the discharge controller to allow that controller to respond with a reduced control point shift. (Lessen the drop in the discharge air temperature when the sprays were active.) The problem was that the existing sensitivity was set up to eliminate the cycling of the loop when controlling the cooling valve as per the original intent.

We altered the circuit via a turn down ratio relay to effectively allow the discharge air controller to have two different sensitivities: one when controlling the cooling valve and the other when it was controlling the mixing dampers. We first used a simulator to establish the exact sensitivity with which the controller was historically running. We increased the sensitivity to the highest level to minimize the drop in discharge air temperature when the sprays were active. We allowed the discharge controller to impact on the mixed air temperature based on evaporative cooling by sending its signal directly to a low selector, which also received a signal from the mixed air controller. The discharge controller also sent a signal to the turn down ratio relay, which allowed the cooling valve loop to operate with the same sensitivity as it had before we entered the job. The turn down ratio relay sent its signal to the chilled water valve.


GRAPH #2
MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY (%RH)
FAN STARTS FOR DAY MODE
67.6°F
60.6°F
64.5°F
57.75°F
36%RH
34%RH
ON OFF
HUMIDIFIER SPRAYS
(TYPICAL)
A P S
3:00 AM
4:00 AM
5:00 AM
6:00 AM
7:00 AM
8:00 AM
9:00 AM
10:00 AM
11:00 AM
12:00 PM
1:00 PM
2:00 PM
3:00 PM
4:00 PM
5:00 PM
6:00 PM

GRAPH #3

The black line illustrates the mixed air temperature, the blue line illustrates the discharge air temperature and the green line illustrates the return air relative humidity.

The graph three shows almost an inverse situation from graph one. On graph three the mixed air temperature rises significantly while the discharge experiences only a small drop in temperature when the humidity sprays are active.

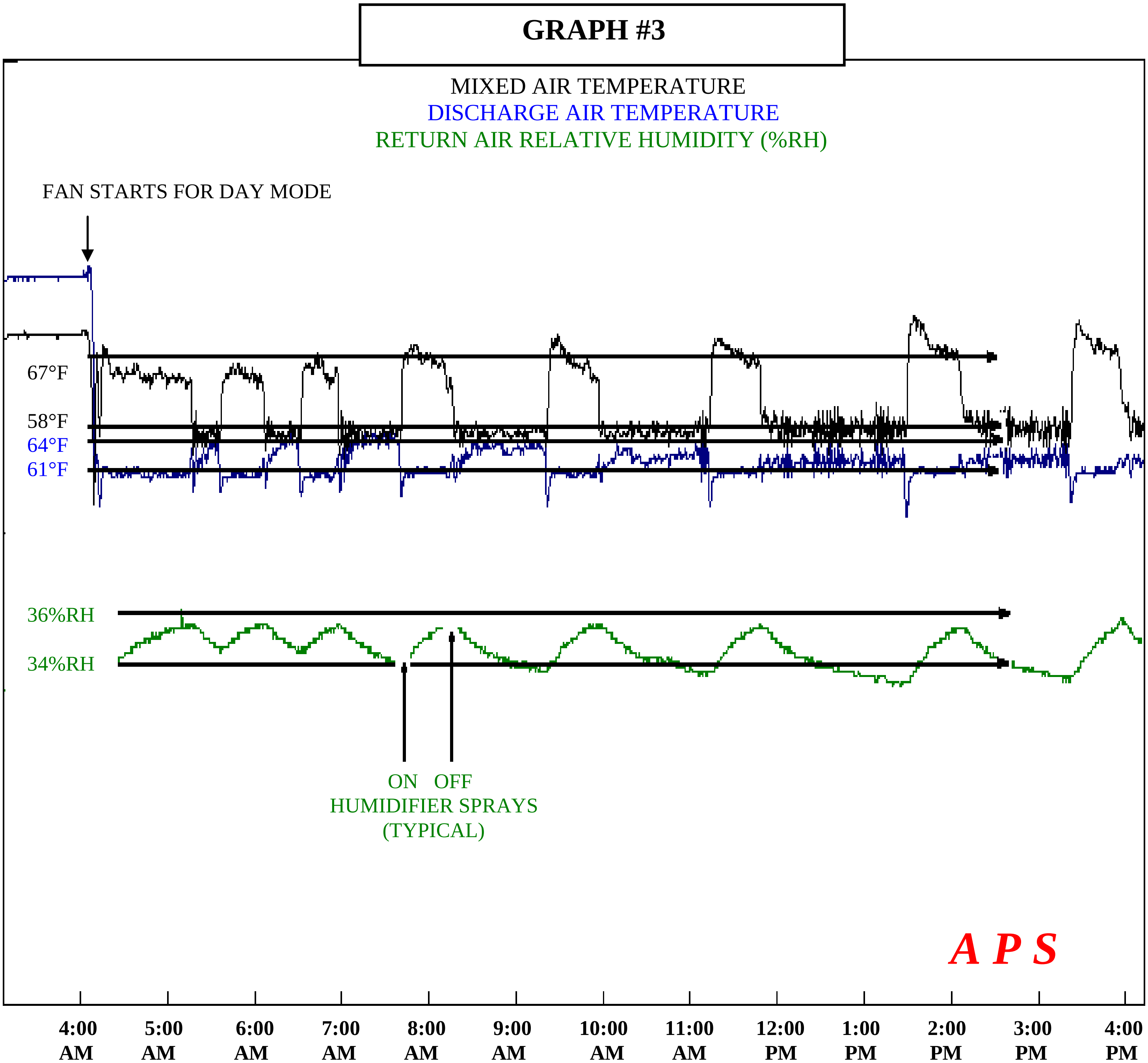
This graph illustrates the more effective use of energy where the system automatically uses heat contained in the return air to provide the energy required to evaporate the moisture for humidification demands. Prior to the control change the humidity circuit forced the heating equipment in the building to provide the heat to compensate for the evaporization.

GENERAL

The evaporative cooling has to be compensated for with heat or the space will become too cold. The heat can come from:

1. the mechanical heating system as illustrated in graph one at great expense and pollution generation from extra combustion, or
2. the return air where the heat is already paid for, which saves money and reduces pollution by reducing combustion.

The discharge air temperature could have the drop in temperature eliminated completely by automatically resetting the discharge controllers set point as required, but the range of control experienced with the sprays active and inactive was well within the acceptable limits.


GRAPH #3
MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY (%RH)
FAN STARTS FOR DAY MODE
67°F
58°F
64°F
61°F
36%RH
34%RH
ON OFF
HUMIDIFIER SPRAYS
(TYPICAL)
A P S
4:00 AM
5:00 AM
6:00 AM
7:00 AM
8:00 AM
9:00 AM
10:00 AM
11:00 AM
12:00 PM
1:00 PM
2:00 PM
3:00 PM
4:00 PM

ACTUAL GAS CONSUMPTION AMOUNTS FOR THE ETOBICOKE EDUCATION CENTRE ILLUSTRATING THE IMPACT OF EVAPORATIVE COOLING AND LOAD ANALYZING TECHNIQUES

READING DATE	GAS CONSUMPTION (X 100)
AUG. 29/75	1,324.8
SEPT. 30/75	2,069.8
OCT. 30/75	1,965.1
DEC. 01/75	1,776.8
DEC. 30/75	1,930.1
JAN. 30/76	2,806.9
FEB. 27/76	2,308.1
MAR. 30/76	2,624.8
APRIL 29/76	1,749.6
MAY 28/76	1,676.1
JUNE 29/76	1,053.2
JULY 28/76	924.6
JAN. 26/77	9,168.4
FEB. 21/77	1,276.8
MAR. 22/77	1,192.7
APRIL 21/77	613.6
MAY 20/77	522.5
JUNE 21/77	609.2
JULY 21/77	683.7
AUG. 22/77	1,066.6
SEPT. 22/77	1,156.2
OCT. 21/77	1,328.5
NOV. 22/77	1,224.2
DEC. 21/77	815.1
JAN. 20/78	927.5
FEB. 20/78	1,078.8
MAR. 22/78	920.5
APRIL 21/78	771.2
MAY 19/78	526.8

WHEN COMPARING THE GAS CONSUMPTION FOR THESE TWO SETS OF MONTHS, ONE FINDS JUNE AND JULY REQUIRED 13.5% MORE GAS Before THE CHANGES, THAN DECEMBER AND JANUARY REQUIRED AFTER THE CHANGES.

FEB. 21/77 – JULY 21/77: INITIAL CONSERVATION DESIGN EMPLOYING EVAPORATIVE COOLING EFFECT OF THE HUMIDITY WATER SPRAYS. OTHER TECHNIQUES WERE IMPLEMENTED SIMULTANEOUSLY.

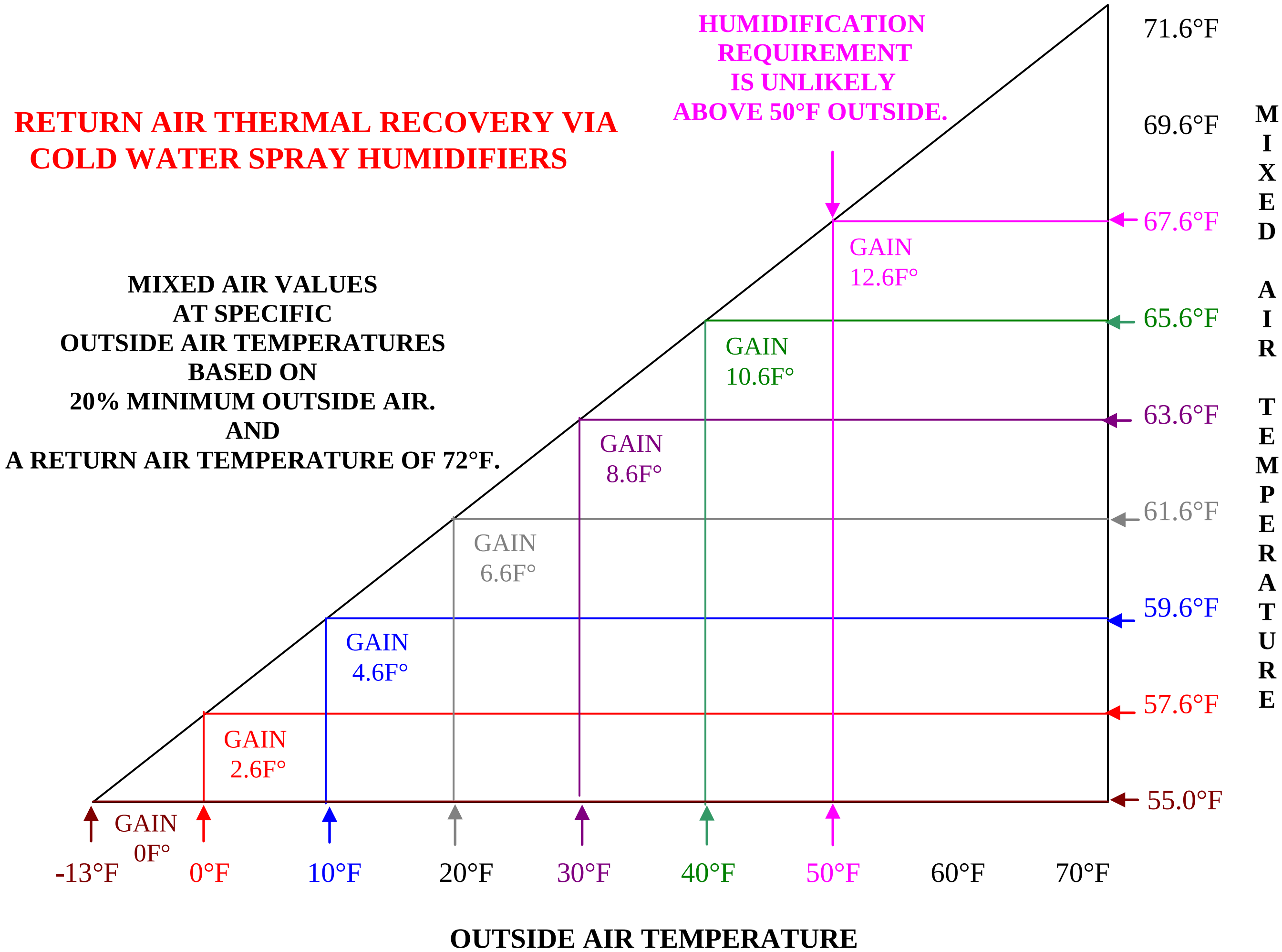
AUG. 22/77 – NOV. 22/77: THE WATER SPRAYS WERE REPLACED WITH STEAM HUMIDIFIERS CAUSING THE COOLING EFFECT TO BE LOST.

DEC. 21/77 – MAY 19/78: AN ALTERNATE CONSERVATION CONTROL TECHNIQUE WAS DESIGNED AND INSTALLED. LOAD ANALYSING FROM THE WARMEST SAMPLE THERMOSTAT RE-ESTABLISHED THE CONSUMPTION REDUCTION.

52.6% REDUCTION WITH EVAPORATIVE COOLING TECHNIQUE.

60% REDUCTION WITH ALTERNATE TECHNIQUE.

RETURN AIR THERMAL RECOVERY VIA COLD WATER SPRAY HUMIDIFIERS



NOTES:

-1- The mixed air set point is normally 55°F. Proportional controllers will droop below 55°F in cold weather conditions increasing the potential heat recovery.

-2- If the outside air is -13°F, the return air is 72°F and the required quantity of minimum fresh air is 20%, the mixed air temperature will be 55°F. At this outside air temperature, or below, there is no savings available, via sequencing logic with the humidity sprays as the system is on minimum ventilation only.

-3-The energy savings available, when the sprays are active, at outside air temperatures above -13°F, gradually increases as the outside air temperature rises.

Example:

At 20°F outside the control logic will automatically raise the mixed air from its normally controlled value of 55°F to 61.6°F. The humidification system will use the energy contained in the 6.6F° gain to evaporate the moisture in the sprays.

-4- The heat recovered, during spray activation, from the return air per 1000 CFM =
(Actual mixed air temperature - 55°F) x 1.08 x 1000 = BTU/HR/1000 CFM

-5- When the fan system is on 100% outside air at 55°F and 100%RH the air will drop to 35%RH when the temperature rises to 72°F, if no other moisture is introduced into the air.