MEE 488
April 18, 2006

Redesign of Bennett Hall HVAC System

Greg Andreasen
Michael Chicoine
Florent Hohxa
Jason Jacobe
Mechanical Engineering, University of Maine, Orono ME 04473, USA

ABSTRACT

Our task was to redesign the current Heating Ventilation and Air Conditioning (HVAC) system in the basement of Bennett Hall. An attempt was made to alter the current system to meet humidification needs while maintaining a comfortable working environment for those who conduct research in the Bennett basement.

INTRODUCTION

The current HVAC system in Bennett Hall was installed in 2000. Originally, the system was designed to maintain comfortable working conditions while complying with OSHA requirements. Post installation, professors working in the basement found that the relative humidity of the air in the basement is too high.

Experiments conducted in the basement cause very low equipment temperature. This, in turn, causes the air closely surrounding the equipment to drop below its dew point. Condensation and sometimes frost will form on equipment. This is not acceptable for the professors working in the area. Condensation forms on a piece of equipment that creates a very high voltage and makes testing impossible. Frost has also been forming on a hydrogen dewer. This freezing makes slight equipment adjustments difficult, and the professor claims, “This effectively creates a hydrogen bomb in the Bennett Basement.” In addition to experimental time being sacrificed by the unacceptably high relative humidity, the shop equipment has been suffering as well. The high humidity level has been allowing rust to form on shop equipment, sacrificing its quality. Pictures of experimental and equipment problems are shown below.

Figure 1 – Professor Hess and Smith’s Equipment and Rusty Shop Tools

Through the experiences of those working in the basement, it has been found that humidity is primarily only a problem in summer months. Unfortunately, this is the time that professors have the most time to conduct their experiments.

A redesign of the current HVAC system is needed so that humidly levels are considered as well as comfort and OSHA regulations.

BACKGROUND

The current system brings in 4565 cubic feet off air each minute. This air is brought in from the outside by the main air-handler fan, where it is first treated. In winter months the air is heated by being passed over a heating coil. The heated coil is supplied with steam to bring up the temperature of the air. In summer months the air is passed over a cooling coil. The cooling coil is filled with a 40% Propylene Glycol mixture. This chilled liquid is run through the cooling coils throughout the whole system. The liquid is chilled in a 25 ton chiller made by Trane. The chiller keeps the liquid cold through the use of a vapor compression refrigeration cycle, sending liquid into the system at 45 F. The liquid is pumped into the system and sent through insulated pipes to either the main coil in the air handler or smaller individual room unit coolers. A schematic is drawn below.

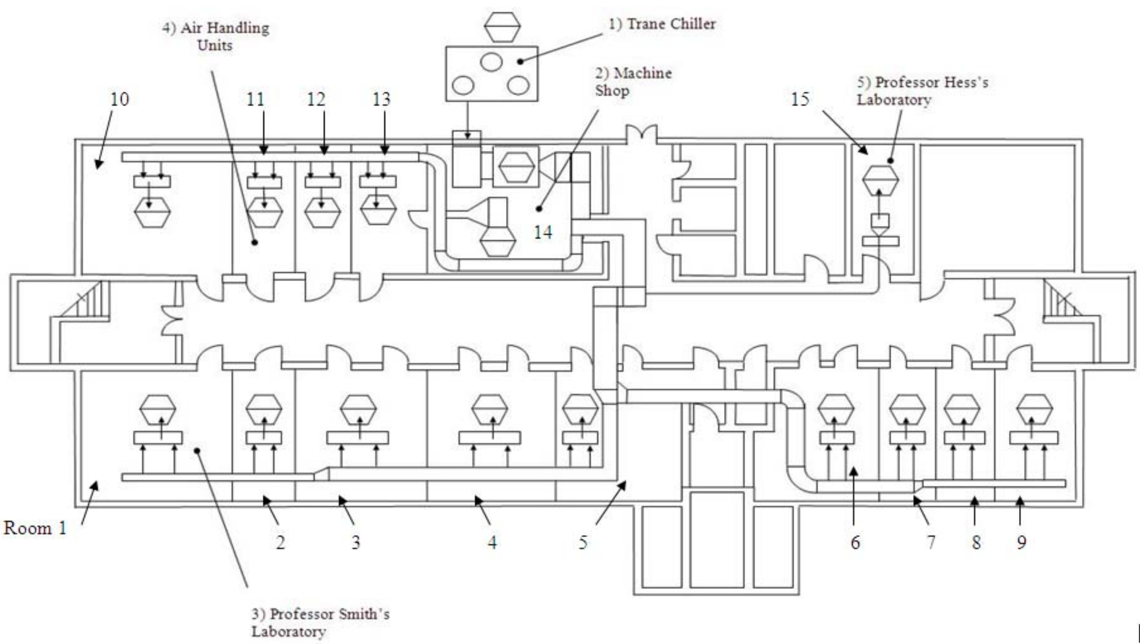

Figure 2 – Schematic of Basement with Room Numbers

The air is passed over the coils bringing its temperature down past its dew point, thus moisture condenses from the air onto the coil and is drained from the system. Several system components including the cooling coil are pictured below.

Figure 3 – System Components Clockwise from top left: Chiller, Main Air Handler, Main Fan, Cooling Coil, and Individual Room Cooler

The air is then blown out of the main air handler and through a series of ducts to the individual rooms. Here the air is mixed with room air and then cooled again by being blown over another cooling coil with the same 45 F chilled liquid running through it. The air is discharged into the room for ventilation and removal of the room heat load and the chilled liquid is sent back to the chiller to go through the refrigeration cycle again.

DESIGN CONSTRAINTS

CURRENT SYSTEM – The redesign had to be considered using existing system components. After a meeting with Associate Executive Director of Facilities Management for Maintenance and Service, Stewart Harvey, it was found that the University was looking to solve the problem without major hardware changes to the system. Thus, the redesign was considered with the following components from the original system:

Chiller – Trane, 25 ton, capacity of roughly 300,000 Btu/hr

Pump – 70 gallon per minute capacity.

Air Schedule – Airflows in the original design accounted for the 6 air changes per hour regulated by OSHA. Also, these flows pressurized rooms and accounted for needed exhaust. For initial Air Schedule see Appendix 1.

TEMPERATURE

Comfort - Comfortable air temperature is important in creating an environment suitable for those in the area. The ideal temperature is 75 degrees Fahrenheit in the summer. This condition is considered flexible by plus or minus 3 degrees Fahrenheit.

Outside, Dehumidification – For calculations involving dehumidification it was found that to meet the dehumidification goal 99.6% of the time for Bangor, Maine the design outside air temperatures would be set at 78.4 F dry bulb and 70.1 F dew point.[1]

Outside, Heat Load – To account for the heat load that is taken on by the basement by the outside air a temperature range was considered. The scenario where the building was under the most heat load was when outside temperature was set at 93.8 F dry bulb and 77.5 F wet bulb. 93.8 F is ASHRAE's extreme annual outside dry bulb temperature[2] and 77.5 F is the corresponding wet bulb at 50% humidity, however this wet bulb is not factored into load calculations. The lower load was considered with an outside temperature at 75 F dry bulb and 62 F wet bulb. One could use the 78.4 F dry bulb dehumidification design temperature but we assumed 75 F to have no heat transfer, the difference in the load is small enough to be considered negligible, also the 75 F ensures that the room will be comfortable even when temperatures drop below the design temperature.

DESIGN APPROACH

The team first considered two design possibilities. Dehumidification be done either by cooling coils or by desiccant dehumidification. After the meeting with Mr. Harvey, it was found that the University was looking to correct the problem with no major overhaul, thus the team eliminated the desiccant design. The team then considered two dehumidification methods using coils. The first scenario partially cools and dehumidifies the outside ventilation air in the central air handler and then splits it up sending the required ventilation air flow rate to each room. In a unit cooler in each room its ventilation air is mixed with recirculated room air and the mixed air is further cooled and dehumidified to remove the room heat load and supply air at the proper state to maintain comfort and the desired humidity level in the room. The second scenario cools and completely dehumidifies the outside ventilation air in the central air handler and then sends it on to the room unit coolers where it would be mixed with room air. The cooling to reduce humidity level to that desired for the room would likely produce temperatures too low for comfort unless the ventilation air were partially reheated before being split up and sent to the room unit coolers. Each room unit cooler would then have to do essentially only sensible cooling to remove the room heat load. These will be discussed in detail later in the report.

Both design approaches must satisfy the constraints of the comfortable working temperature of 75 F dry bulb and a corresponding wet bulb that will result in no higher than 50% relative humidity. Selecting these characteristics it was found that the wet bulb temperature must not exceed 62.5 F. A Psychometric chart was used to find enthalpies throughout the design.

The heat load placed on the building is used in design of each scenario. The building was measured and the area of the walls, ceiling, windows, and doors were found. Then heat transfer analysis was performed to calculate the heat transfer through the walls, ceiling, windows, and doors. To calculate this load the outside temperatures were set 93.4 F and 75 F and the inside temperature was at the 75 F. The heat transfer equations for elements are as follows, and are completely worked out in a MathCad sheet in Appendix 2.

$$Q_{wall} := A_{wall} \cdot \frac{1}{R_{wall} \cdot \tau_{wall}} \cdot (T_o - T_i)$$

Equation 1

Where A is the wall area of the room exposed to the outside temperatures. R is the wall resistance which was chosen to be .09 $\frac{(ft^2 \cdot F \cdot hr)}{Btu \cdot in}$ because this is the average R value for concrete.[3] τ is the wall thickness and this was found to be 16 inches. To is the outside temperature and Ti is the inside temperature.

$$Q_{wind} := A_{wind} \cdot U_{wind} \cdot (T_o - T_i)$$

Equation 2

[1] 2005 ASHRAE Handbook - Fundamentals
[2] 2005 ASHRAE Handbook - Fundamentals
[3] 2005 ASHRAE Handbook – Fundamentals Chapter 25 Section 8 Table 4

Where A is the window area exposed. U is the window loss coefficient which has been chosen to be $.81 \frac{Btu}{hr \cdot ft^2 F}$ because this is the loss coefficient for double glazed windows with ½ inch air space with aluminum frames and no thermal break.[4] As before To and Ti are the outside and inside temperatures respectively.

$$Q_{ceiling} := \frac{A_{ceiling}}{R_{ceiling} \cdot \tau_{ceiling}} \cdot (T_1 - T_i)$$

Equation 3

Where A is the ceiling area of the room. R is the resistance of the concrete assumed to be of the same resistance of the walls. τ is the ceiling thickness and this was found to be 6 inches. T1 is the first floor temp set at 83 F the average first floor temperature. Ti is once again the basement temperature. This is done for each room and the loads are incorporated into the calculations needed for coil selection. The Results for both load cases are as follows:

Heat Loads on Each Room BTU/Hr		
Room	Qroom(93.4)	Qroom(75)
1	10710	0
2	3660	0
3	7320	0
4	7320	0
5	7049	0
6	5659	0
7	3660	0
8	3660	0
9	6472	0
10	10710	0
11	3660	0
12	3660	0
13	3660	0
14	14460	0
15	3297	0

Table 1 – Outside Loads on Each Room for Maximum and Minimum Heat Loads

The next step in the design process was to conduct an analysis of the air as it would pass through each system. This was done several times over, each time cooling the air to different temperature levels in the main air handler and cooling in the unit coolers to get the desired room temperature.

Scenario 1 - In scenario 1 we chose to cool the air in the main air handler and cool it further in the room. A schematic is drawn below. The Psychometric analysis is accompanied in Appendix 3.

[4] 2005 ASHRAE Handbook – Fundamentals Chapter 31 Section 8 Table 4

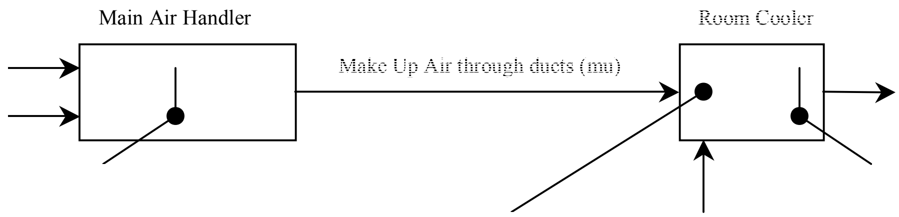

Figure 4 – Schematic of Scenario 1 with Terms (States)

The outside air is brought into the main air handler and passed over the cooling coil. The cooled air is referred to as makeup air, and is split up and sent to the rooms. The required load the cooling coil must provide is

$$q_{ah} := m_{ah} \cdot \left(h_{oa} - h_{mu}\right)$$

Equation 4

The mass of the air flowing over the coil was found by converting the volumetric flow rate 4565 cubic feet per minute to pounds per hour by multiplying it by 60 minutes in one hour and dividing by 13.29 cubic feet needed for one pound of air. This conversion has been done for all the volumetric flow rates given in the design specifications for the heat transfer analysis. The outside air enthalpy is set by the ASRAE Dehumidification constraint. The makeup air temperature, and thus enthalpy, is variable and several different temperatures were used through the iteration of the design. This makeup air is then sent to the rooms through the ducts. Here, it is mixed, in the room coolers, with recirculated room air. Next the mixed air temperature must be found.

$$T_{mix} := \frac{v_{mu}}{v_{sup}} \cdot T_{mu} + \frac{v_{recirc}}{v_{sup}} T_{room}$$

Equation 5

This is needed as an input into the coil selection program. The volumetric flows rates come from current system designs. Tmu is the make up air temperature and Troom is set at 75 F for comfort. Next the enthalpy of the supply air, that is discharged in the rooms, is needed. This is found by applying the cooling load equation used in equation 4 and manipulating to solve for h supply.

$$h_{sup} := \left(\frac{-q_{room}}{m_{sup}}\right) + h_{room}$$

Equation 6

This finds the needed enthalpy of the supply air that will be able to cool the room and handle the load of the outside heat load. Next the enthalpy of the mixed air must be found. This is done similar to equation 5.

$$h_{mixed} := \frac{m_{mu}}{m_{sup}} h_{mu} + \frac{m_{recirc}}{m_{sup}} h_{hoom}$$

Equation 7

Now the load the room cooling coil must perform may be found.

$$q_{coil} := m_{sup}\left(h_{mixed} - h_{sup}\right)$$

Equation 8

This analysis was performed on every room with corresponding outside heat loads and air flows. The analysis was run for both outside heat load considerations (93 F and 75 F). Iterations were done using different make up air conditions until a suitable scenario was found. The total load placed on the coil of the main air handler and all the room units may not exceed the 300,000 Btu/hr the chiller can supply. The final worked out solution of Scenario 1 maybe accessed in Appendix 4. The final air states can be found in the results section.

Scenario 2 - In scenario 2 we chose to cool the air in the main air handler then reheat the air and split it up and send it out to the rooms and cool it further in the room coolers. A schematic is drawn below.

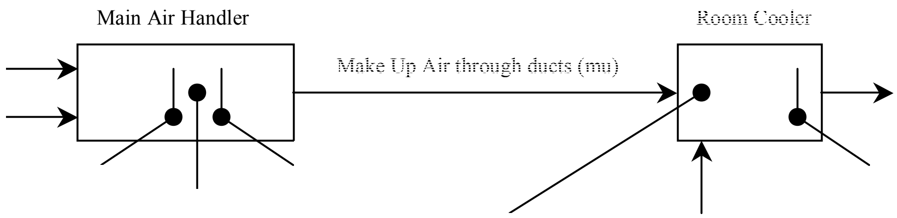

Figure 5 – Schematic of Scenario 1 with Terms (States)

Similar to the first scenario the outside air is brought into the main air handler and passed over the cooling coil. The required load the cooling coil must provide is

$$q_{ah} := m_{ah} \cdot \left(h_{oa} - h_{ca}\right) \quad \textbf{Equation 9}$$

This air is then reheated with a heating coil. The enthalpy after the air has been heated is found by taking the point of the cooled air state on you psychometric chart and moving horizontally right, maintaining constant weight of the water in the air, until one reaches their desired dry bulb temperature of the heated air. The selection of the cooled air and makeup air is variable, and several different combinations were used to find a suitable design. When selecting a reheat coil it is important to consider the size of the coil (ie length and height). For our heating coil selection, design inputs were selected, but the dry bulb temperature output was not able to be specified. We had to try numerous coil sizes that produced varying dry bulb temperatures. For example our first coil selected heated the air to 87 F dry bulb even though we specified 70 F. To account for this we downsized the physical coil size, thus heating a smaller amount of the air. A simple diagram is shown below.

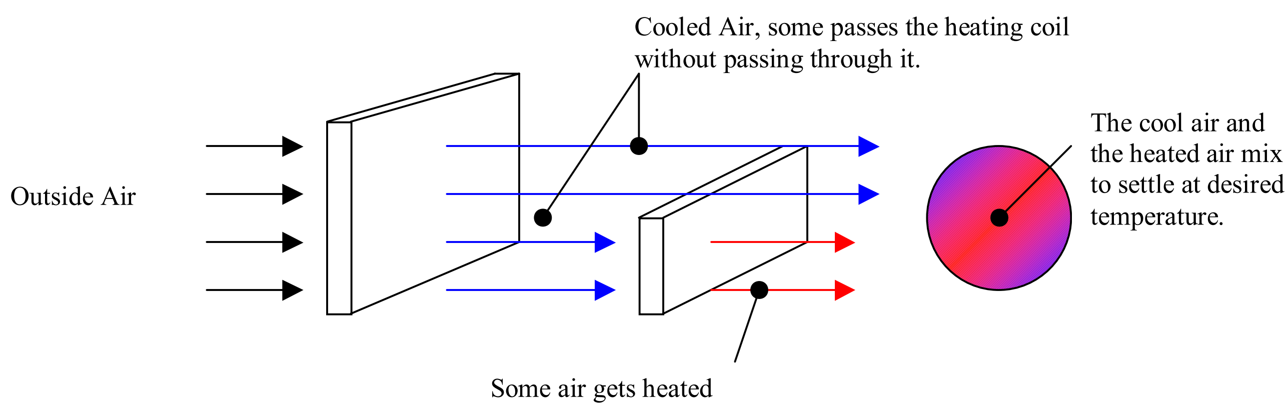

Figure 6 – Sketch of how smaller heating coil allows for some cool air to pass by without being heated

All the air is cooled but since the heating coil only fills half the space thus only cools about half the air. The air cool air and heated air mix just past the heating coil, this air is split up and sent to the rooms. The process from here on is very much the same as Scenario 1. This heated mixed will be called makeup air it is mixed with recirculated room air. Next the mixed air temperature in the room coolers must be found.

$$T_{mix} := \frac{v_{mu}}{v_{sup}} \cdot T_{mu} + \frac{v_{recirc}}{v_{sup}} T_{room} \quad \textbf{Equation 10}$$

This is needed as an input into the coil selection program. Next the enthalpy of the supply air is needed. This is found by applying the cooling load equation and manipulating to solve for h supply.

$$h_{sup} := \left(\frac{-q_{room}}{m_{sup}} \right) + h_{room}$$

Equation 11

This finds the needed enthalpy of the supply air that will be able to cool the room and handle the load of the outside heat load. Next the enthalpy of the mixed air must be found. This is done similar to equation 7.

$$h_{mixed} := \frac{m_{mu}}{m_{sup}} h_{mu} + \frac{m_{recirc}}{m_{sup}} h_{hoom}$$

Equation 12

Now the load the cooling coil must perform may be found.

$$q_{coil} := m_{sup}\left(h_{mixed} - h_{sup}\right)$$

Equation 13

This analysis was performed on every room with corresponding outside heat loads and airflows. Again, this was performed for both the high and the low outside heat loads. Iterations were done using different make up air conditions until a suitable scenario was found. We tried combinations of cooling in the main air handler to 55 F db and 54 F wb, or 56 F db and 55 F wb and several other combinations before the reheat. Many of these combinations created an overall load higher than the 300.000 Btu/hr our chiller can handle. Thus they would require the purchase of a new chiller, which was not an option. The first suitable condition we found was 60 F db and 59 F wb. The final worked out solution of Scenario 2 maybe accessed in Appendix 5, a table of all air sates is in the results section.

Finally before coil selection, the number of rows of coils in each room unit must be found. The team went over to Bennett Hall basement and opened up each room cooler. It was found that there were 4 rows to each coil in the room unit coolers.

RESULTS

Several combinations of temperatures were run through this analysis. These all achieved the 50% relativity. Designs were eliminated if they totaled a heat load over the 300,000 Btu/hr. In all designs there was a slight reheat needed when the outside heat load at its lowest. This is because the outside and inside air are at the same temperatures, therefore there was no load. Thus, those designs from each scenario that had the lowest reheat were selected. The required loads on each room coil maybe found in Appendix's 4 and 5. The air states for the settled on for final solution are in the following tables. The tables are for room one, but those for the other rooms are very similar and temperatures are with 5 F.

Scenario 1 for Room One					
Air State Point	Location/Process	Dry Bulb F	Wet Bulb F	Relative Humidity	Enthalpy Btu/lb
Outside Air		78.4	72.5	75.7	36.2
Make Up Air	Cooled in Main AH	63	62	94.7	27.8
Room Air	Set by Design	75	62	48.1	27.8
Mixed Air	Mix of Room and MU	74	62	50.8	27.8
Supply Air	Cooled in Room Cooler	54.7	54.1	96.4	22.7
Final	After Picks up Room Load	75	61.8	47.4	27.6

Table 2 – Sates of Air for Scenario 1

Scenario 2 for Room One					
Air State Point	Location/Process	Dry Bulb F	Wet Bulb F	Relative Humidity	Enthalpy Btu/lb
Outside Air		78.4	72.5	75.7	36.2
Cooled Air	Cooled in Main AH	60	59	94.4	25.8
Heated MU Air	Some Air Heated by Coil	70	62.6	66.6	28.2
Room Air	Set by Design	75	62	48.1	27.8
Mixed Air	Mix of Room and MU	74.6	62	49.4	27.8
Supply Air	Cooled in Room Cooler	55.9	55.2	96.3	23.4
Final	After Picks up Room Load	75	62.4	49.5	28

Table 3 - States of Air for Scenario 2

The total load on the chiller for the whole system is shown in table 4. The BTU's needed per hour for each situation are as follows

	Scenario 1 (75)	Scenario 1 (93.8)	Scenario 2 (75)	Scenario 2 (93.8)
Q total (BTU/Hr)	168,300	256,700	208,200	299,900

Table 4 – Calculated Loads in Each System

It should be noted that the addition of the steam coil creates a load 107,000 BTU/Hr on the current heating system. The coils were searched using the criteria found in design in TOPPS Coils Selection program made available by Trane. Nick Vecchione, a salesman from Trane, was contacted and he said that there is only one four row cooling coil for each size room cooler. Therefore those selected from the TOPPS program are already in the units and just the flow rates had to be changed. The selected coils may be found in Appendix 6. The current system's pump is pumping 61.1 gallons per minute and neither case exceeds this, shown in table 5, therefore no new pump purchase is required.

Pump Schedule			
Scenario 1		Scenario 2	
Room	GPM	Room	GPM
Main AH	27.97	Main AH	23.61
1	6.41	1	4.52
2	0.4	2	0.39
3	1.07	3	0.83
4	1.07	4	0.83
5	0.99	5	0.78
6	0.68	6	0.65
7	0.44	7	0.36
8	0.4	8	0.39
9	7.27	9	0.98
10	6.41	10	6.34
11	0.44	11	0.36
12	0.54	12	0.5
13	0.54	13	0.5
14	3.36	14	3
15	0.51	15	0.42
Total	58.5	Total	44.46

Table 5– Pump Schedule for Each System

ECONOMICS

One of our objectives for redesign was to maintain lowest cost possible. A detailed analysis was performed in the system to see how much it cost to run each of the existing units and the over all cost of running the HVAC system during the summer months. Because of the OSHA requirements some of the units run all the time and for analyzing them the total amount of KWH they use for the summer months was taken and multiplied with the cost in dollars of a KWH, see Table 6. Figure 7 shows a graphical presentation of the power usage.

Individual Units Power usage	**HP**	**KWH per day**
FC-1	0.125	2.237103
FC-2	0.0625	1.118552
FC-3	0.125	2.237103
FC-4	0.125	2.237103
FC-5	0.125	2.237103
FC-6	0.125	2.237103
FC-7	0.125	2.237103
FC-8	0.0625	1.118552
FC-9	0.125	2.237103
FC-10	0.125	2.237103
FC-11	0.125	2.237103
FC-12	0.05	0.894841
FC-13	0.05	0.894841
FC-14	0.125	2.237103
FC-15	0.05	0.894841
Pump Power usage	1.73	30.96151
Air handling unit supply fan Power usage	2.2	39.37301

Table 4 – Power usage for Individual Units

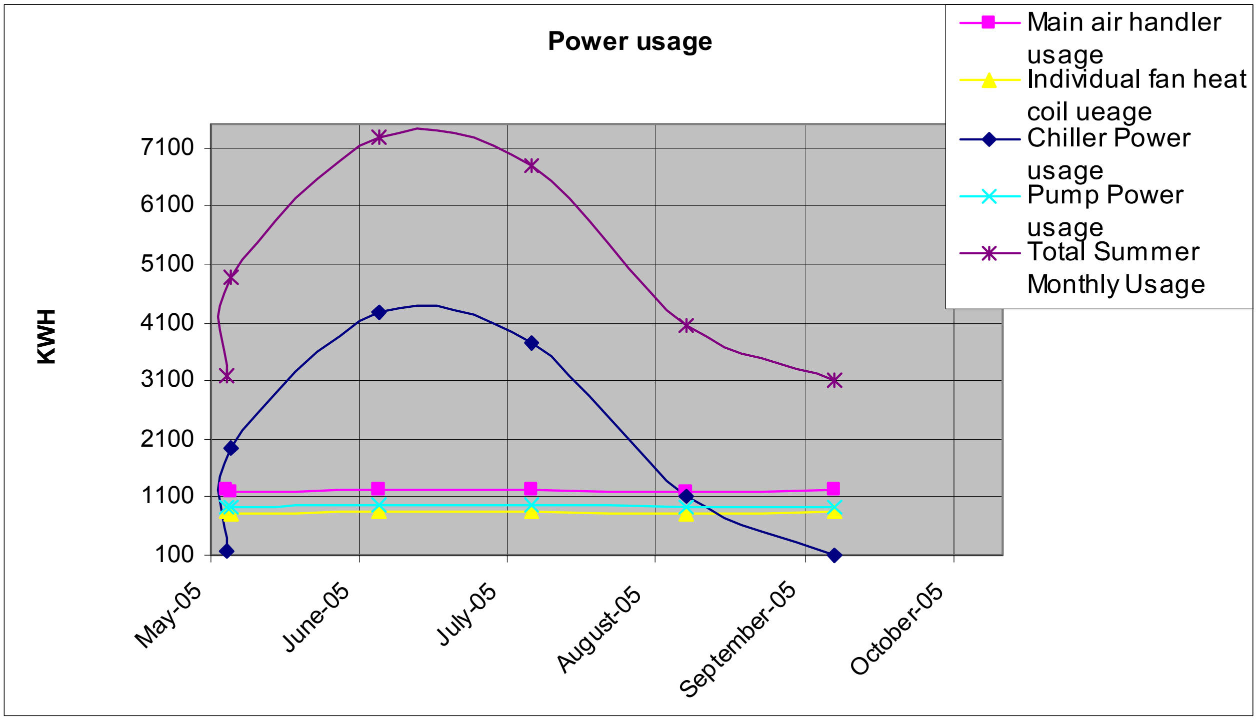

Figure 7 – Graph of total Power Used

The more challenging task was to figure how the chiller worked because the chiller operates based on the outside temperature. The hours of specific outside temperatures were counted and added for summer months and then multiplied with the KWH at that temp to figure total usage, see Table 7.

Outside Temperature(F)	75	80	85	90	
System KWH at above temp.	23.6	24.615	25.8	27.285	
					KWH per month
May hours	4	4	0	0	192.86
June hours	56	17	6	2	1949.425
July hours	86	60	25	4	4260.64
August hours	94	50	12	0	3758.75
September hours	22	9	12	3	1132.19
October hours	5	0	0	0	118

Table 5 – Chiller Power Used

All these KWH were taken and multiplied with the dollar cost per KWH to figure the total power usage. The prices for appropriate months and the totals for each month are shown below in Table 8 and Figure 8 numerically and graphically.

Months	May	June	July	August	September	October	Total
Average Price $/ KWh	.09395	.09482	.09556	.09707	.0959	.09507	
Total monthly Cost for Summer 2005 ($)	212.28	374.49	604.63	565.47	300.37	207.69	2057.24

Table 8 – Total Costs of Running System

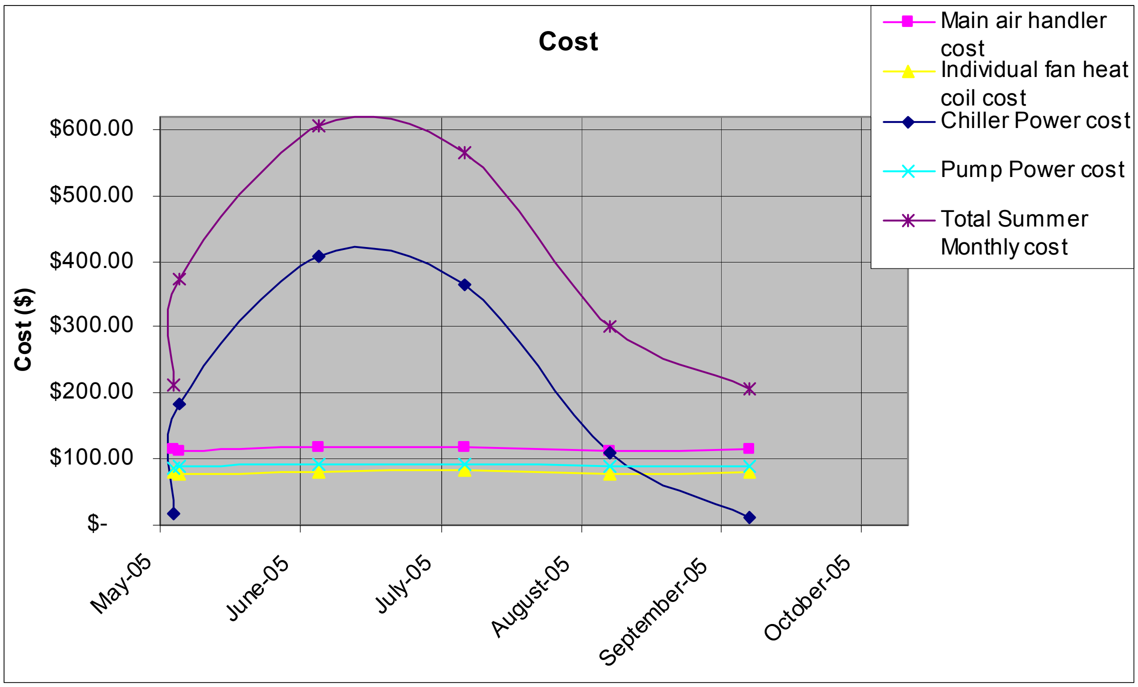

Figure 8 – Total Costs Graph

We were successful in achieving our objective because we designed so we can use the same components that are already designed so the only cost will be the new coil and the installation for it. We are currently waiting on the price of the new coil. Nick Vecchoine, a Trane salesman, is currently generating a quote.

Conclusion

In conclusion, the team is in favor of installing Scenario 1. This involves the purchase of just one cooling coil to go in the main Air Handler. This solution satisfies the relative humidity condition as well as cools for comfort. From Appendix's 4 and 5 it can be seen the reheat needed in the low out side load condition is close to the reheated needed in Scenario 2 under the same low load. However, these loads are relatively small and maybe offset by equipment and people in the room. We have found that the purchase of one coil and the adjustments of the chilled liquid flow rates should correct the current systems problems. We feel that this redesign is suitable for facilities to compare to their analysis and be considered in solving the humidity problems in Bennett Hall Basement.