ABSTRACT

SIMON, MICHAEL JAMES. A Case Study and Analysis of Cleanroom Energy Use. (Under the direction of Dr. Stephen Terry and Dr. Herbert Eckerlin).

A plant has a cleanroom HVAC system which is known to be a large energy user. This study found that the current cleanroom operation is indeed wasteful, and stands to benefit from improved controls and operation protocols. Proper data collection is instrumental in understanding how the system is operating. After analyzing data, the real cleanroom operation was found to be different than the assumed cleanroom operation, a significant conclusion.

Data on cleanroom conditions, power draw for both dedicated cleanroom chillers, and power use of all five air handling units are collected. It is revealed that cleanroom chiller and reheat power cost is approximately $103,000/yr, much higher than necessary, but also much less than the original estimate of $385,000/yr. During the data collection process, changes to the cleanroom operation were made, and some of them are seen to have desirable results, but there are also very negative unintended consequences, which are seen in the data collected. The cleanroom reheats are now seen to be sensitive to times of high humidity, using an average of 200 kW of electric strip reheat when the dew point is above 50°F.

These consequences stem from an inadequate control system. Whether it is the case that each of the five air handling units have no idea about what the other units are doing, or whether the control over incoming and outgoing air is not as complete as it was originally

thought is unknown. The data collected from this project forces the plant to take a second look into how completeness and accuracy of its control setup.

Also notable is the finding that chiller power draw is heavily influenced (>100 kW) by daily fluctuations in outside temperatures, even though there is very little internal cooling load placed on the chillers due to outside air infiltration (<14 kW), and no roof, wall or floor conduction load. This suggests that the condenser of the air-cooled chillers operate at high pressure when temperatures outside are hot, decreasing efficiency dramatically. This helps set maximum summer demand, making its cost disproportionate to its energy waste. Also of note, the large chiller helical-screw compressors do not unload very well.

A Case Study and Analysis of Cleanroom Energy Use

by
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DEDICATION

This thesis is dedicated to my committee co-chairs, Dr. Herbert Eckerlin, and Dr. Stephen Terry, who took it upon themselves to help me in my education, and work. Of course, I also thank my father and mother, Terry and Vicki Simon, who provided a stable home for me to grow up in.

BIOGRAPHY

Michael Simon was born at Touro Infirmary in New Orleans, Louisiana on August 7th, 1984. His parents soon moved to Greensboro, North Carolina, and, as a 5 year old, he had no choice in the matter. Greensboro was kind to young Michael, and he graduated with honors from Walter Hines Page High School in 2003. Heading to North Carolina State University to study Engineering, he was soon attracted to Mechanical Engineering, where he took a particular liking to the thermal sciences. After graduating that program in 2007, the Industrial Assessment Center at NCSU offered a chance to work with them while pursing towards a Masters Degree. Michael will shortly reside in Odessa, TX, where he has accepted a job post–graduation.

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Chapter 1: Introduction

The objective of this thesis is to provide a case study and analysis of the HVAC energy use of one particular cleanroom. This study stems from a North Carolina State University Industrial Assessment Center audit done at this facility, when it was estimated that the cleanroom uses 338 tons of cooling, and 750 kW of electric reheat, for a total cleanroom HVAC energy cost of about $385,000/yr. The reheat was thought to be needed because there is not enough of a heat load in the room, which is closely related to the problem of overcooling. Doing some quick calculations led us to believe that the facility could save around $315,000/yr of this number, the vast majority of which is assumed to be reheat savings. This study was commissioned to measure cleanroom parameters, determine the current cleanroom operation, identify the savings possible, and suggest improvements on the current operation.

1.1 Cost of Energy

Because of confidentiality issues, specific details of plant operation, product, size, and location cannot be disclosed. Fortunately, for the purposes of this study, that information is largely irrelevant anyway.

Initially, the North Carolina State University Industrial Assessment Center was contacted to do an energy survey of this interested facility. Duke Power provides power to this facility, and the Rate schedule used to determine power cost is OPT-I. This information is freely available on Duke Power's website [1]. Duke identifies four summer months (June-September) where demand cost is about $10.03/kW, and about $5.52 kW during the remaining eight winter months. Note in Table 1.1 that Duke has a tiered rate schedule based on maximum demand. Doing any cost savings analysis one would use the incremental value. For example, if a cost savings recommendation will reduce plant demand from 6,000 kW/month to 5,800 kW/month, one would use the Over 5,000 kW demand line.

The middle tier will be used in this analysis in order to not disclose plant size. Duke also differentiates between on-peak power at about $0.0453/kWh and off-peak power at about $0.0273/kWh. On peak during the summer is identified as the hours between 1 p.m. and 9 p.m. M-F, and during the winter as 6 a.m. to 1 p.m. M-F. All other times, and some various holidays are designated as off-peak. Also note that demand is only set during times that are designated as on-peak, although there is a nominal charge for excess off-peak demand, termed economy demand, as shown in the Table below.

Table 1.1: Energy Cost from Rate Schedule

	June-September ($/kW)	October-May ($/kW)
First 2,000 kW demand	$10.9490	$6.4446
Next 3,000 kW demand	$10.0296	$5.5167
Over 5,000 kW demand	$9.1017	$4.5804
(Economy Demand)	$0.8688	$0.8688

By including demand cost with on-peak power usage, one can quickly see that the cost of electricity on-peak is more than twice what it is off-peak. This is very difficult to quantify exactly, because just a single 30 minute interval of high power use will negate an entire month of demand reduction efforts. A good example of this is the extra cost in demand of that air conditioning system on a hot, muggy southern sunny summer day. Although this is clearly a regional rate, this is representative of how power companies bill on a national basis.

Why does it benefit electric companies to charge for demand?

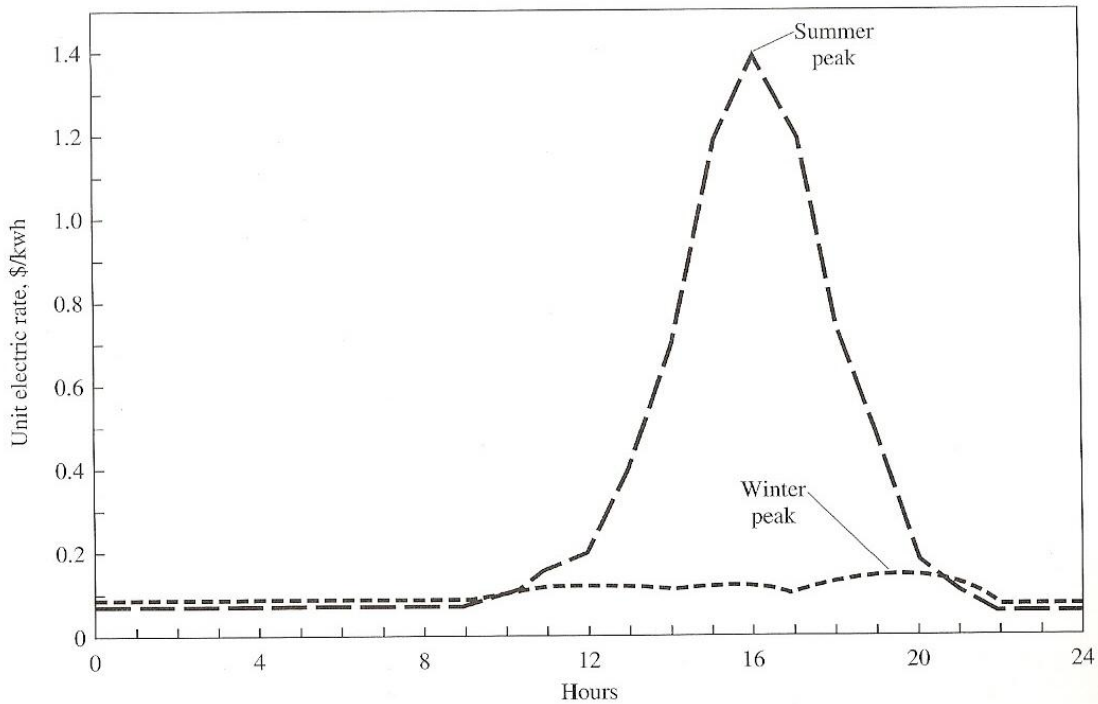

Figure 1.1: Daily Real-time Pricing Unit Electric Rates at Summer & Winter Peaks [2]

Figure 1.1 from Shan K. Wang [2] is dated by about 10 years, but it perfectly demonstrated why companies desperately want to avoid having to provide that extra kW. It hurts their profit margins.

1.2 Cleanroom Description

The facility cleanroom itself is completely enclosed within the facility, not only on every wall, but also around the ceiling of the cleanroom, so only the floor is an external surface, as roughly illustrated by Figure 1.2.

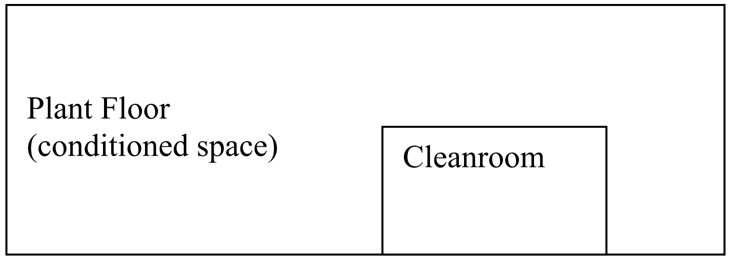

Figure 1.2: Rough Plant Sketch

The facility itself is air conditioned, and this will be key in understanding the cleanroom operation. Below, in Figure 1.3, plant and outside conditions are shown on the same graph over a 15 day period. This graph is a summer month (June), and the relationship of outside temperature to plant temperature is there but severely mitigated, and the effect of dew point swings is mitigated as well, although to a lesser degree. This is important as it pertains to clean room infiltration, and heat gain or loss through the walls and roof.

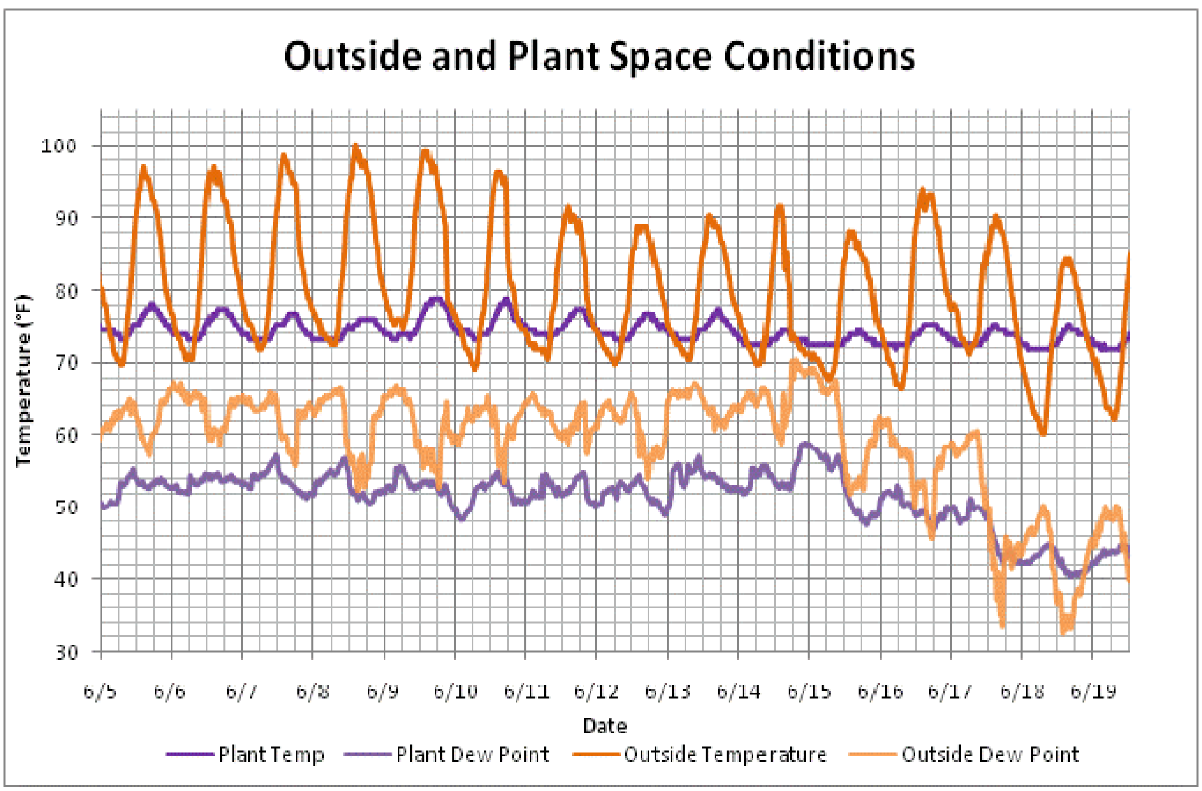

Figure 1.3: Outside and Plant Space Conditions

The cleanroom as currently set up is not pressurized, although the plant is in the process of changing the current operating protocols and systems, which will be discussed in detail in Chapter 5 later in the report. The cleanroom operates 5 zones: Zones 1-4 are in the cleanroom proper, and zone 5 is in the cleanroom entrance area. All of the zones are similar in size, but due to process changes since the cleanroom was constructed, zones 1, 3 and 5 serve areas with little heat load. Zone 5, however, does not have the precise temperature and humidity controls placed on it that zones 1-4 do, and it is served by a much smaller AHU and electric reheat.

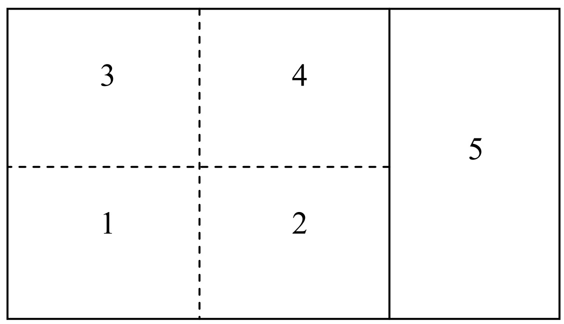

Figure 1.4: Cleanroom Zone Layout

Zones 1-4 are served by AHU's 1-4, each of which is a separate air handling unit. Notice the dashed lines in Figure 1.4. The AHU zones 1-4 are not separated in the cleanroom itself by any kind of wall or shield. So people, goods, and air can move freely between the 4 zones. Only zone 5 has a wall between it and zones 2 and 4, but even it has open doorways to zones 2 and 4. These AHU's are provided cooling through two dedicated 240 ton air-cooled chillers, and reheat is provided by electric strips. Operating as a class 10,000 cleanroom (explained in Chapter 2), 180,000 cfm of air enters the return ducts goes through the fans. After this, it goes through the cooling coils, which cool the air to approximately 55°F, due to the control system. After going through the cooling coils, the air then goes through the reheat coils, and the air is brought back up to near room temperature, typically 65-70°F. Next, the air goes through the humidifiers, where air is humidified if necessary.
After this, it goes through the HEPA filters on the way back to the room.

This is how the cleanroom system was thought to operate before this project was started. Chapter 7 will present some postulated variations to the system as described above, and Chapter 6 will present the new operation, as conceived and after 2 months of data recording.

At the beginning of the study a total of 5,000 cfm of extra air was brought in from the outside as makeup air. This air is but a small trickle flow through each of the 4 main AHU's. The cleanroom needs this makeup air because there is an exhaust flow of about 6,000 cfm of air from process equipment outside. This 6,000 cfm is hot, contaminated air. Basic mass conservation dictates that about 1,000 cfm is not accounted for. This explains the small amount of plant infiltration into the cleanroom, an issue the plant is aware of, and is working on correcting.

Over the course of the study, significant modifications were made, both physically installed and in the controls. These changes will be presented in Chapter 6.

1.3 Chiller Conditions

In Chapter 4, the condition of the cleanroom will be discussed as it pertains to internal temperature, humidity, and reheat required. Of course, any discussion of cleanroom energy consumption would be incomplete without taking a hard look at the chilled water system, which is completely dedicated to cleanroom cooling. To give an overall quantification of the percentage of plant demand used by these chillers, it is typically between 5-10% of total plant demand.

The cleanroom chilled water is provided by two identical 240 ton dedicated air-cooled package chillers. Each chiller has three helical-screw compressors and 17 condenser fans. Of the three compressors, there is one larger compressor sized at 100 tons, and two 70 ton compressors. The system is controlled such that the two smaller compressors are on one circuit, and the larger compressor is on the other circuit. Interestingly, data logging suggests that the chillers only operate in one of four modes: off, 100 tons, 140 tons, or 240 tons of compressor capacity switched on. This limits the ability of the system to load match for maximum efficiency operation.

The chillers are assumed to operate at an Energy Efficiency Rating (EER) of 9.6, which is standard efficiency of a 240 ton Series R Chiller Model RTAC according to Trane [3]. The unitless Coefficient of Performance (COP) is as follows:

$$COP = EER/3.414 \tag{1.1}$$
$$COP = 2.81$$

Throughout the report, a chiller kW/ton value will be referred to. This simply means the number of kW of power that needs to be provided to the chiller to get one ton of cooling. This value is as shown below:

$$\text{kW/ton value} = 12 / EER \tag{1.2}$$
$$\text{kW/ton value} = 1.25 \text{ kW/ton}$$

Please note that this is the value for a brand new unit operating efficiently. This number is suspected to decrease in efficiency as the chiller is unloaded and as the outside temperature rises, but this value is still used, making <u>all</u> calculated chiller power cost numbers conservative.

In addition, there are three 15 hp pumps which pump chilled water to and from the cleanroom. These pumps do turn on and off as necessary, but were not data logged.

Unfortunately, without running controlled experiments at different conditions, it is difficult to determine exactly how much power the compressors use for a given condition.
One might suggest doing a thermodynamic energy balance on the system from estimated

(and, in some cases, measured) cleanroom cooling information, but this does not provide an accurate picture of chiller energy usage, as we shall see below. The closest this thesis can come to running controlled experiments is running natural experiments, by collecting information as the chillers run in normal operation. Obviously, problems occur because of the lack of control over external variables, but it is possible to come up with the most likely explanation for the issues raised by the data collection.

1.4 Controls Description

The controls in the cleanroom are set up to try and always keep 50% relative humidity, and 70°F temperature. However, a simple glance at the psychrometric chart will illustrate that the dew point at 70°F temperature is not 55°F, as previously stated as the approximate air temperature leaving the cooling coil, but between 49-50°F. Thus, the humidity control ends up running the chillers hard during the summer, and the reheats are not quite able to keep up in certain areas of the cleanroom, since some areas of the cleanroom have little to no heat generation. The end result is wild swings in relative humidity, when what is desired is a relatively constant relative humidity, as illustrated below in Figure 1.6. To explain the figure legend, the cleanroom is demarcated into the arbitrary labels "hot side" and "cold side". The hot side roughly corresponds to zones 2 and 4, from Figure 1.4, and the cold side corresponds to zones 1 and 3. Zones 1 and 3 are labeled as the cold side because there is very little production activity occurring in these zones, and so have little internal heat load. The cold side swings in relative humidity are thus slightly smaller, but have not been included for clarity's sake.

The cleanroom hot side temperatures in Figure 1.6 track well against the outside temperatures well, on the surface. Days such as July 28, and August 1-3 help reinforce the viewpoint that cleanroom internal conditions are almost exclusively affected by outside conditions. But then there are days like July 30th, where indoor conditions are almost the opposite of outdoor conditions over the same time span, which suggest that other conditions are in play.

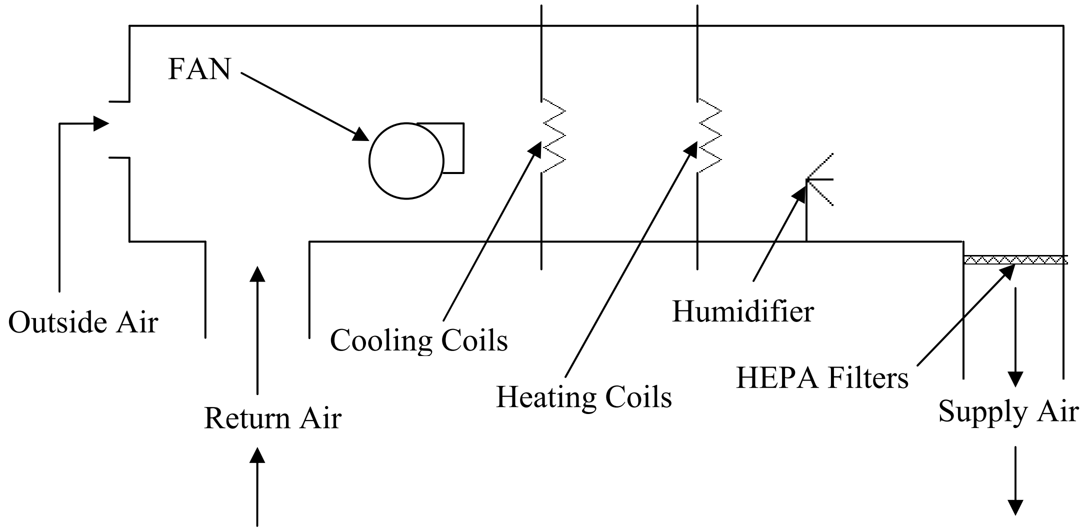

Figure 1.5: Sample AHU Setup

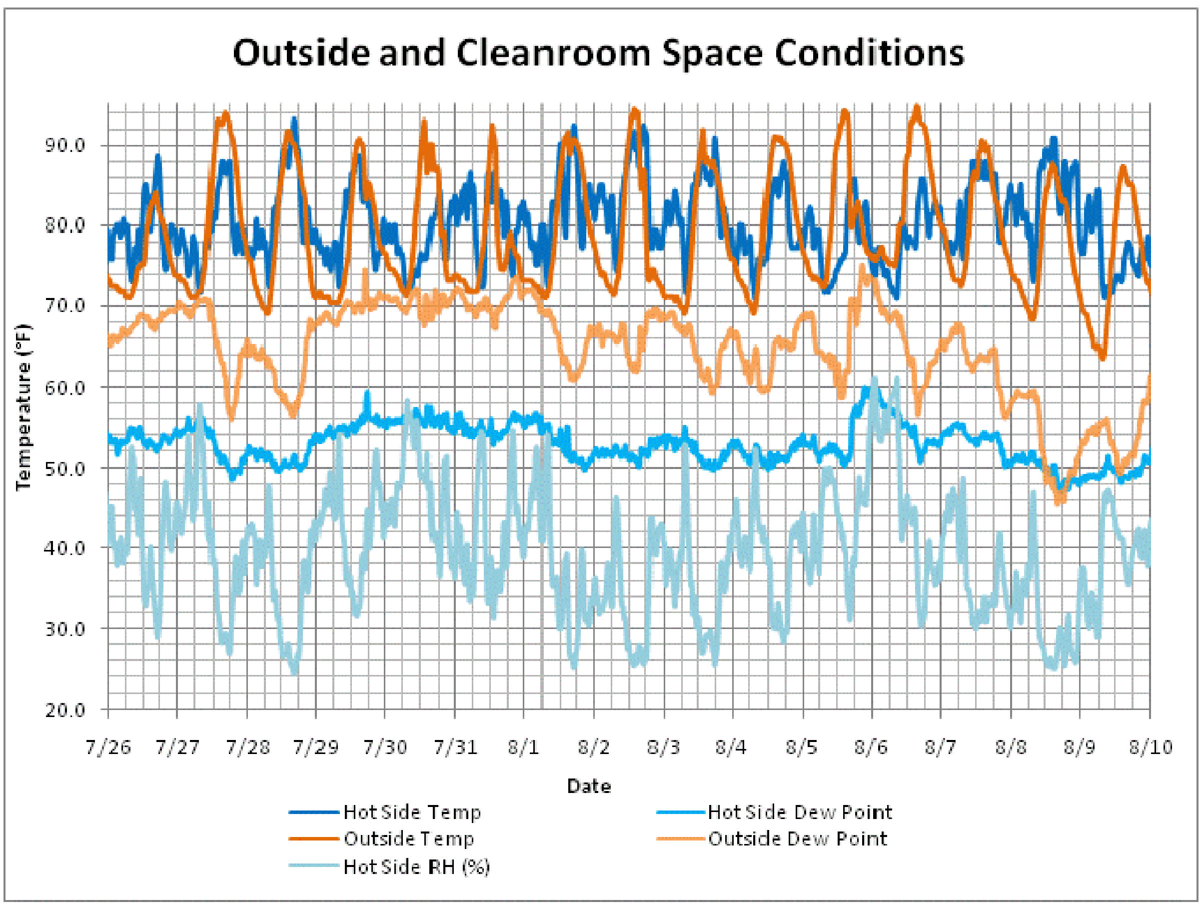

Figure 1.6: Outside and Cleanroom Space Conditions

1.5 Discussion of Inefficiencies

There are several energy inefficiencies inherent in this system. First, all of the air which needs to be filtered must also go through the rest of the cooling and reheat cycle, even if it does not need to be cooled or dehumidified. To state this more clearly, the cold air flow needs of the space are much less than the filtration flow. Therefore, air is always overcooled, then reheated to the space temperature, with the exception of some winter days, where the wet bulb temperature is below 45°F. This places an extra load on the chillers and, most importantly, the reheat coils. This issue will be discussed at length later in the thesis.

A small inefficiency exists where the fans are placed before the cooling coils. This means that the coils have to cool the fan energy, whereas the fan energy, if placed after the coils, could be used to help reheat. This cannot be changed without extensively redesigning the system, and so will not be considered in this analysis. However, it may be useful to consider designing a system which has multiple fans; a fan before the filters and coils, and one which comes after the filters and the coils, to utilize some of the sensible heat provided by the fan. The best system, however, is one that requires little to no reheat.

Initially, the room was not pressurized, and so infiltration from the plant floor is inevitable (now it is pressurized). From the other end of the system, the chillers are air cooled, which is not a particularly efficient way to operate. The chillers themselves are also oversized, and therefore do not match loads very efficiently.

Chapter 2: Background

This thesis deals with a case study of the associated costs of heating and cooling a cleanroom. A cleanroom can be precisely defined as follows, per Federal Standard 209E:

‘A room in which the concentration of airborne particles is controlled and which contains one or more clean zones’ [4]

The classification system of a cleanroom is set by the desired concentration of airborne particles. Technically, this means that there are separate classifications of cleanliness, as defined by the particulate count in the air. Practically speaking, this is done through enclosure of the space, and the dedication of a separate HVAC system to meet the filtration, pressurization, heating, cooling, and humidification requirements of the space.

The air change requirements for cleanroom systems vary greatly by manufacturing process. The scale of classification, as defined most simply by Federal Standard 209D, is measured in particles/ft^3, where a particle is ≥ 0.5 μm [4]. It ranges from 1 particle/ft^3 to 100,000 particles/ft^3, where each increase in classification is one order of magnitude greater. This, of course, is a massive simplification of the varying international systems, and indeed is an outdated standard, but is still in wide use throughout these United States today.

Each Class of cleanroom has a corresponding requirement for the number of air changes required/hour, and even the direction of flow these air changes need to take. The cleanroom presented in the study below is currently operating as a Class 10,000 cleanroom, meaning 10,000 particles/ft^3.

If one is serious about reducing the energy costs associated with operating the cleanroom in their facility, it is suggested as a first step to know exactly how clean the process needs to be. In many facilities today, the cleanroom operator has approached the issue from a production viewpoint, and this is completely understandable. After all, the company would quickly go out of business if it was not good at producing widgets. However, if the cleanroom is run at, for example, Class 1,000 when only the one process at the back needs this high quality air, why subject the whole room to this high degree of filtration? This may seem obvious, but is especially applicable if the cleanroom is not operating as the designer intended, and so energy may be being unknowingly wasted.

The next step, one that may be done simultaneously as the first step, is to figure out what difference does it make. Literally, and to the bottom line. Actually identifying the energy use by the various HVAC systems associated with the cleanroom will give the company an idea of how much it should care about the energy used by the cleanroom. This could be a simple process in any given facility, but each individual case will present its own surprises. It would also be useful discover the relationship between outside conditions, cleanroom temperatures, and energy usage.

Once it is known where the energy is being used, intelligent, informed decisions can be made about where to focus efforts on energy reduction. As an added benefit, this thesis has discovered that in performing this kind of analysis, plant HVAC infrastructure capacity is quantified for use in future production plans. This will make management more inclined to approve such projects, once the connection between HVAC and production is quantified.

Chapter 3: Data Collection

3.1 Data Collection Overview

Before discussing the conditions inside the cleanroom, it is important to describe the methodology of data collection. Certain issues are unavoidable by using the instruments available. Mitigating them, however, is important to the applicability of the conclusions. Potential problems with data collection always manifest themselves in one of two ways: reliability of data and completeness of data.

Reliability issues are not too big of a deal with these data loggers, with two notable exceptions. Data was recorded over 8 adjacent periods of time, stretching from June to April. During this period of time there were loggers which ceased functioning partway through, and loggers which were unable to have data downloaded for some unknown electronic failure. However, the reliability of the loggers that are functioning is assumed to not be an issue, and there is little reason to suspect that there is, in most cases. Two notable exceptions are chilled water temperatures and dew point data. This will be discussed in detail later in the thesis, in Section 4.1.

Completeness is a much bigger issue, and would add that it typically is in real world systems. How do you capture all of the data necessary and not be overwhelmed by the sheer amount collected? During this data collection sequence, six data loggers were added throughout the course of the study, from fourteen loggers to twenty loggers. Several other things we would have liked to log, but were simply unable to do so. This will be discussed at length throughout the paper.

3.2 Data Loggers

The interior temperature and humidity loggers recorded the temperature and humidity at its location every 24 minutes. This was done by necessity to allow time to pass between site visits. Each logger has a finite amount of memory, and since conditions do not change all that frequently, it was thought that this would be sufficient for our purposes. The exterior temperature and humidity logger recorded temperature data every 2 minutes, since it was a higher capacity logger. The loggers on the AHU’s and chillers are set to record data every 10 minutes. The other limitation on data collection is Microsoft Excel. All of the data was converted into Excel spreadsheets, and excel will not graph anything with more than 32,000 data points, and quite honestly begins to operate unacceptably slowly before then.

The loggers on the reheats of all five AHU’s work by clamping a Current Transducer (CT) around each leg of the three phase power. These CT’s come in various sizes for various expected maximum amperages. The smaller CT’s have the ability to record finer data, because each CT has a discrete number of separate amperages it can read. These CT’s are then plugged in to a small data recording unit, called a HOBO. The HOBO’s are programmable to function in a variety of data storage applications, and will record up to 32,000 data points.

These loggers have an 8 bit analog to digital converter. Since $2^8 = 256$, the logger only has 256 bins to assign data. So a 100 amp CT has a resolution of 0.39 amps (i.e., 100A/256=0.39A), and a 50 amp CT has a resolution of 0.2 amps. Of much greater consequence to this paper is the knowledge that temperature data has a resolution of 0.69°F. This limits the precision of energy balance calculations done later in the report, but it will be shown that these calculations are still useful. These loggers are the older variety, and the new line of HOBO’s uses a 12 bit analog to digital converter, and thus has 4,096 bins to assign data. Future work should therefore use newer data logging equipment. This method of data collection is desirable for our purposes because power does *not* need to be disrupted to the equipment to record the required data. In addition, data can easily be exported into Microsoft Excel in order to work with the data in any way the end user wishes.

All three legs of the reheat must be recorded, to ensure accurate power readings. This is because each leg is essentially a one phase to ground system that is independently controlled.

Thus, one leg may be using power, while the other two are not used. The total power is taken by recording one leg of total power, and then to assume that the other legs are somewhat equal. This assumption appears to be flawed. Since the total power includes the strip reheat, humidifier, and fan load, if any one of the three draws different power loads off each leg, then the equal leg assumption is not correct. This will be discussed in Section 4.1.4 and 4.4. The details of each logger, the connected instrumentation, and recording interval are provided in Table 3.1 below.

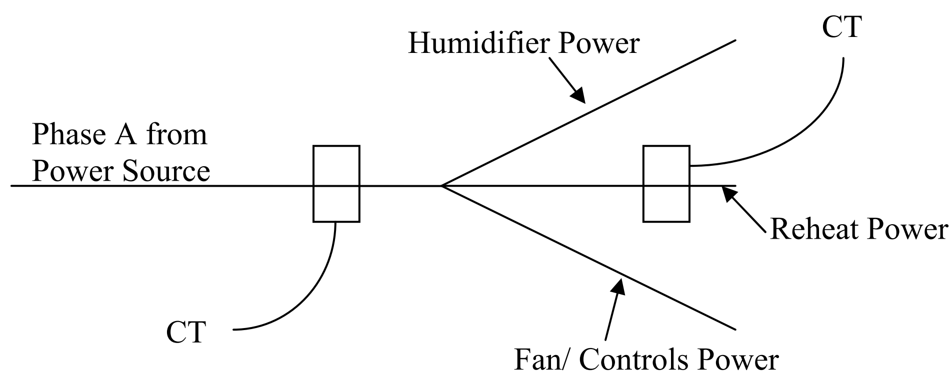

Figure 3.1: Diagram of AHU Data Recording

Table 3.1: Logger Locations and Descriptions

Location	Collection Method	Time Interval	Month of Installation	Description
AHU 1	1 - 200 Amp CT	10 minutes	June	One leg of the total AHU 1 Power. Includes reheat and fans
	3 - 100 Amp CT's			Each leg of AHU 1 Reheats
AHU 2	1 - 200 Amp CT	10 minutes	June	One leg of the total AHU 2 Power. Includes reheat and fans
	3 - 100 Amp CT's			Each leg of AHU 2 Reheats
AHU 2 Humidifer	1 - 50 Amp CT	10 minutes	December	One leg of the total humidifier load
AHU 3	1 - 200 Amp CT	10 minutes	June	One leg of the total AHU 3 Power. Includes reheat and fans
	3 - 100 Amp CT's			Each leg of AHU 3 Reheats
AHU 4	1 - 200 Amp CT	10 minutes	June	One leg of the total AHU 4 Power. Includes reheat and fans
	3 - 100 Amp CT's			Each leg of AHU 4 Reheats
AHU 5	3 - 50 Amp CT's	10 minutes	September	Each leg of AHU 5 Reheats
Outside	Temperature Sensor	2 minutes	June	Outside Temperature
	Humidity Sensor			Outside Dew Point and Relative Humidity
Plant	Temperature Sensor	24 minutes	June	Plant Temperature
	Humidity Sensor			Plant Dew Point and Relative Humidity
Hot Side 2	Temperature Sensor	24 minutes	June	Hot Side 2 Outside Temperature
	Humidity Sensor			Hot Side 2 Dew Point and Relative Humidity
Cold Side	Temperature Sensor	24 minutes	June	Cold Side Temperature
	Humidity Sensor			Cold Side Dew Point and Relative Humidity
Hot Supply	Temperature Sensor	24 minutes	June	Hot Supply Temperature
	Humidity Sensor		July	Hot Supply Dew Point and Relative Humidity
Cold Supply	Temperature Sensor	24 minutes	June	Cold Supply Temperature
	Humidity Sensor		July	Cold Supply Dew Point and Relative Humidity

Table 3.1 continued..

Zone 5 Return	Temperature Sensor	24 minutes	September	Zone 5 Return Temperature
	Humidity Sensor			Zone 5 Return Dew Point and Relative Humidity
Hot Side 1	Temperature Sensor	24 minutes	October	Hot Side 1 Temperature
	Humidity Sensor			Hot Side 1 Dew Point and Relative Humidity
Hot Side 3	Temperature Sensor	24 minutes	October	Hot Side 3 Temperature
	Humidity Sensor			Hot Side 3 Dew Point and Relative Humidity
Hot Side 4	Temperature Sensor	24 minutes	October	Hot Side 4 Temperature
	Humidity Sensor			Hot Side 4 Dew Point and Relative Humidity
Chiller 7	1 - 200 Amp CT	10 minutes	June	One leg of Chiller 7 Power. Includes 1 compressor and 7 fans
	1 - 200 Amp CT			One leg of Chiller 7 Power. Includes 2 compressors and 10 fans
Chiller 8	1 - 200 Amp CT	10 minutes	June	One leg of Chiller 8 Power. Includes 1 compressor and 7 fans
	1 - 200 Amp CT			One leg of Chiller 8 Power. Includes 2 compressors and 10 fans
Chiller 1-5	1 - 600 Amp CT	10 minutes	June	One leg of Chiller 1-5 Power. Includes all compressors and fans
Chilled Water Temperatures	Thermocouple	10 minutes	June	Cleanroom chilled water return temperature
	Thermocouple			Cleanroom chilled water supply temperature

The table above shows that insufficiencies in data collection were continually identified, and attempted to be corrected through more complete data collection. Areas to include in any future attempts should consider adding cleanroom chilled water pumps, temperature, humidity and air flow sensors for the supply and return air for *each* air handling unit, internal cleanroom power consumption, temperature, humidity, and flow rate of any exhaust air, pressure readings of the cleanroom and plant (to better understand infiltration load), and cleanroom humidifiers. While this list of additional logging is suggested as ways to ensure the issues with incomplete and ambiguous data are solved.

Chapter 4: Results

4.1 Data Collection Issues

There are a few deficiencies in the raw data. Some of these are obvious, such as the data taken from AHU 1, where the total amperage is always recorded as being lower than the reheat amperage, and the even more egregious examples of data recorders failing to work. Some of these are much less obvious, such as the tendency in AHU 4 for each of the reheats to give a different amperage value, thus skewing the "total AHU 4 amperage" data.

4.1.1 AHU 1

The total power for AHU 1 is typically recorded as being about $1/5^{th}$ the reheat power, which is clearly incorrect. Figure 4.1 below illustrates the problem very well. How can total current be less than reheat current? Significant to add here is that the reheat amperage on this particular unit are recorded as being very close to the same at all points in time. The average difference is within ±5%, with a smaller standard deviation, so no matter which leg of power is being recorded, the total is reflected as accurate. In Figure 4.1 below, this small difference between the reheat current is shown where all of the three reheat lines are pretty well on top of each other.

Also of note on the AHU 1 graph is the seemingly weekly pattern of total power increases. This power increase is thought to be the fan, although other instruments like the humidifier are hooked up to this power source. This actually is not a weekly occurrence, and the peaks occur on the weekends and weekdays, both during the day and night.

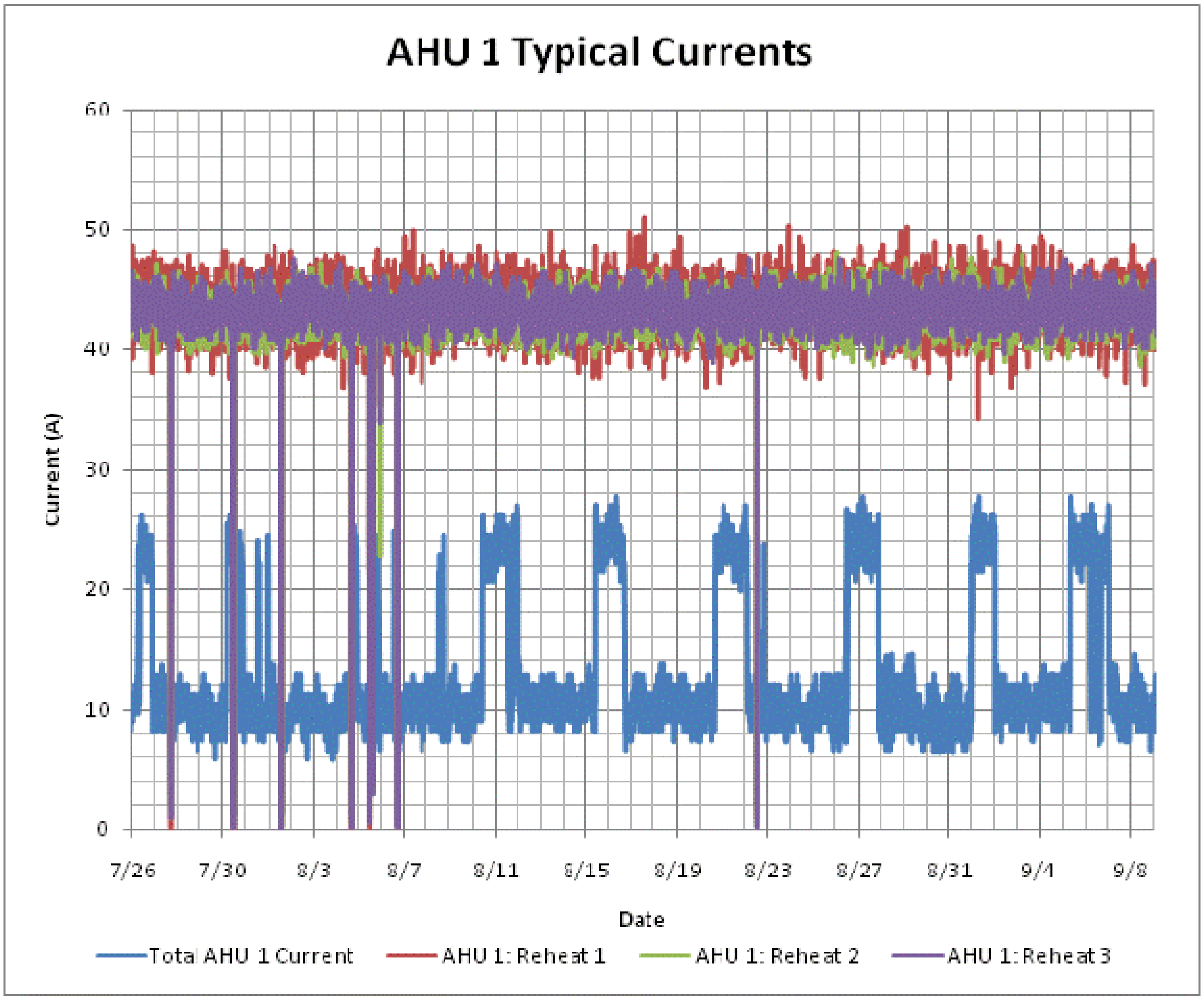

Figure 4.1: AHU 1 Typical Currents

Why is it that total current can be less than reheat current? First, data loggers were switched out, and the batteries changed, but the results stayed consistant. In fact, AHU 1 effectively ran exactly as shown above (with varying days of assumed fan usage) from the beginning of data collection in early June 2008 until abruptly changing modes at 9/29/08 at 1:30 PM (9/29 will be discussed later, in Section 4.3.1). Proper installation was also checked, and this seemed to be the problem. For the CT's to work correctly, the split core magnet has to be connected properly, i.e, they cannot be clamped backwards. On the smaller CT's, this is not a problem, but on the 200 amp CT's, the clamping end can actually be easily detached (typically on older, well used CT's), and then easily (and accidentally) reattached in the reverse, since the correct direction is *not* marked by the manufacturer. This seems to be what happened. After discovering this possible failure mode, it was tested independently, and amperage was indeed underrecorded. This is part of the learning curve for using the equipment.

The next graph illustrates the attempt to correct for the obvious total power error. The other AHU's seem to operate in a very similar overall pattern, where the total power seems to only have two other sources of power draw: a low set level about 4-5 kW greater than the reheat load, and a higher ~21 kW step load. One thought is that this is the fan load at low and high power. However, the fans are equipped with VFD's, so this would be unusual. For AHU's 3 and 4, the typical low set level power draw is 9% of the reheat power, and the step load is a fairly constant 21 kW, giving a total of about 25 kW. Therefore, to estimate total AHU 1 total power, it was assumed that AHU 1 follows in the same pattern as the other AHU's. Clearly, this has limitations, but it provides a much better estimate of total power than the data set as recorded. Throughout the rest of this thesis, total AHU 1 power will be this estimated figure, in order to try and more accurately reflect the total.

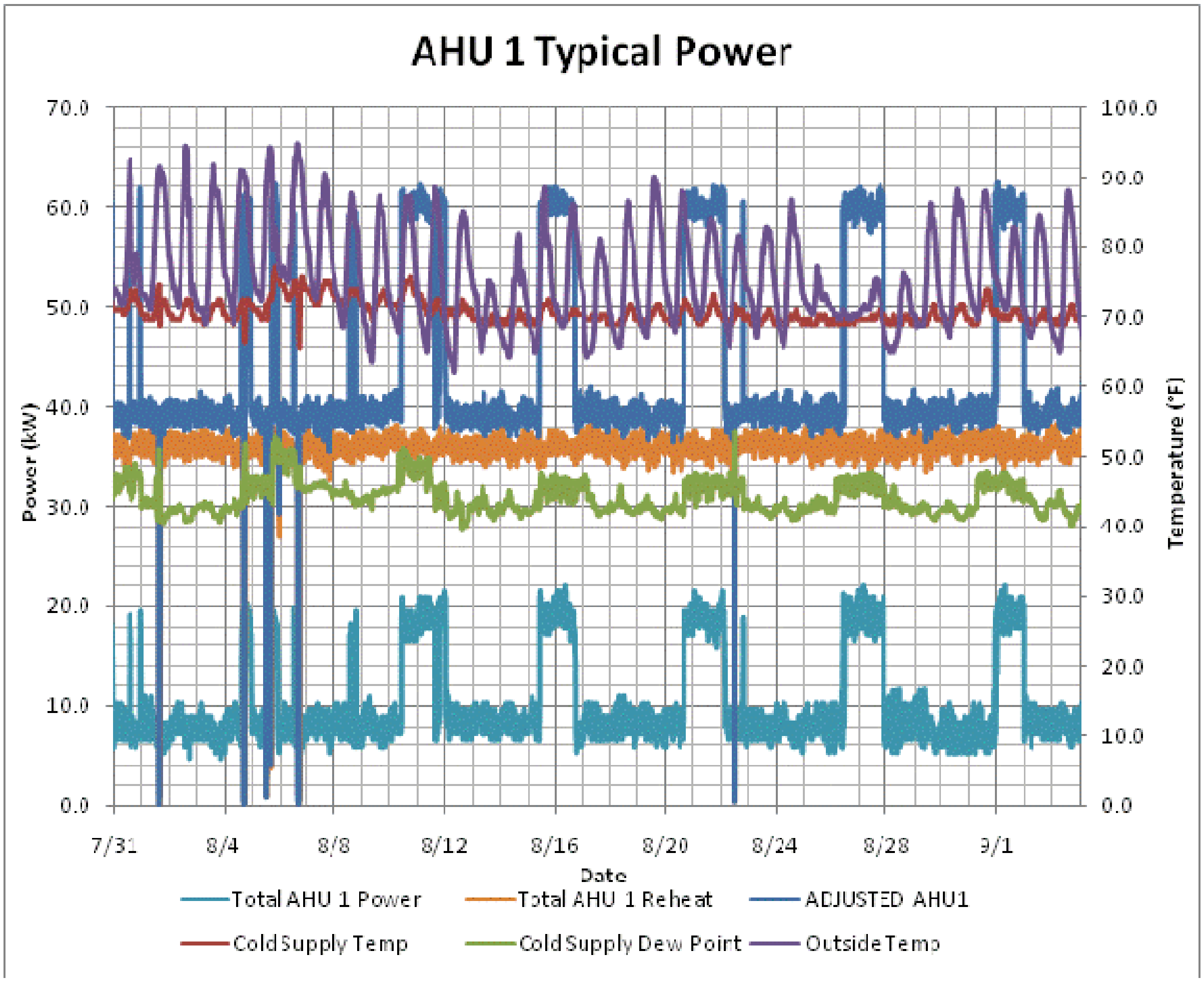

Figure 4.2: AHU 1 Typical Power

In the figure above, the anomaly of the cold supply dew point being higher than dew points on surrounding days looks to be related to the increase in power draw by the AHU. This increase in dew point of the supply air in the cold zone is about 3.5 °F, a fairly substantial increase. However, examining 8/31 shows that the relation may be secondary, and not primary.

4.1.2 Outdoor Dew Points

The next egregious error in data recording occurs in the outdoor temperature and dew point data. During July, the HOBO decided to record dew points as large negative numbers, which just happen to look like a scaled down, negative version of the temperature data. The data for months before and afterwards have no issues with accuracy, since this data can be spot checked for accuracy against local weather data, and has proven to be reliable in other months, as well as for the temperature data for July. This dew point data was thrown out.

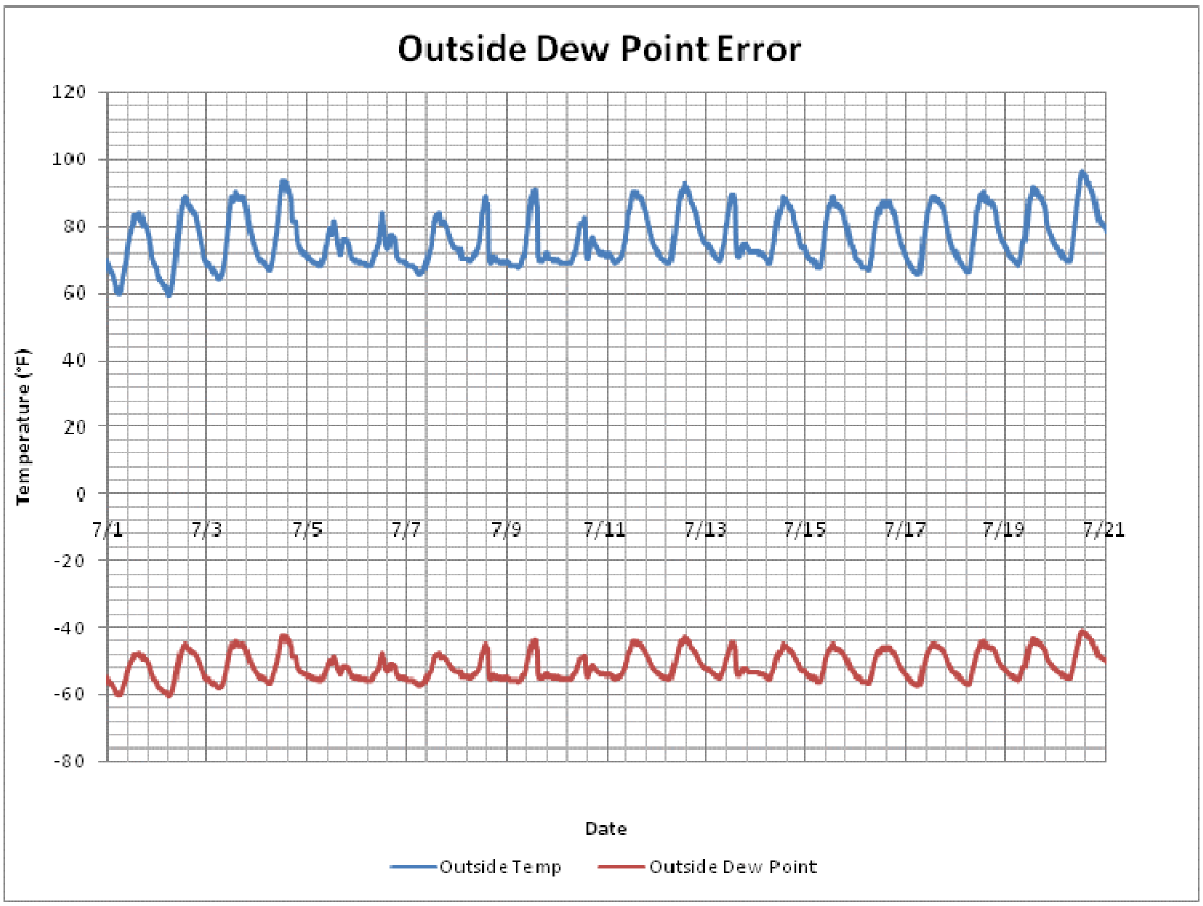

Figure 4.3: Outside Dew Point Error

4.1.3 Chiller Temperatures

Next, there are issues with the cleanroom chilled water supply and return temperatures. This data was collected in a different method than the other temperature data, as described in Table 3.1. Instead of using the HOBO's internal temperature recorder, external thermocouples were used to read temperature. This brings up possibilities for error in the collection methodology not present in the other temperature recordings. First, the thermocouples could be bad. However, since the temperatures given are in the neighborhood of that expected (± 5-10 °F), it is assumed that this cannot be the whole reason, with one notable exception, presented below.

Instead, the focus lies on how the thermocouples are utilized. The supply and return lines for the cleanroom chilled water system are insulated, and there are no easy places to measure water temperature through direct contact. Instead, the insulation does not cover the chilled water loop's drain pipe, which split off in the compressor room. From here, thermocouples were inserted in between the insulation and the metal pipe of the drain lines, as illustrated in Figure 4.4 below.

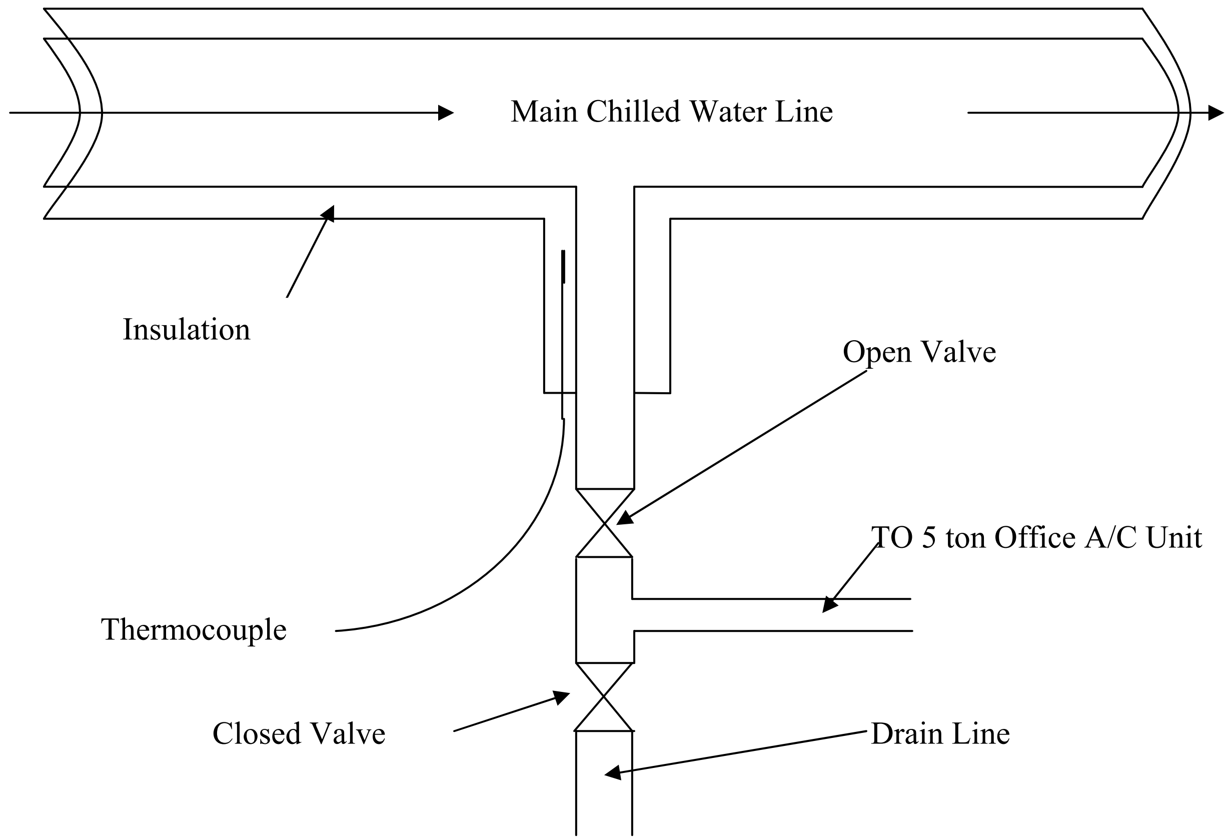

Figure 4.4: Cleanroom Chiller Temperature Collection Diagram

There are several difficulties associated with this placement. First, the chilled water lines are more than 10 feet off the floor, which makes thermocouple placement difficult. This makes it hard to tell how far the thermocouple has gone into the insulation. The end of the thermocouple is metallic, but the rest of the line leading to the HOBO is flexible cable. Gravity also plays a role, and the thermocouples fell out several times while downloading data.

The author of this paper remains unconvinced that the thermocouple is taking a direct reading of the cleanroom chilled water temperatures. The offshoot line is likely filled with stagnant water in cold times, and may only receive (or reject) heat from the main chilled water line mainly through conduction during these times. The Figures 4.5 and 4.6 below help illustrate the problems associated with the chilled water temperatures.

It is also very odd that the dedicated cleanroom chilled water system has a small A/C unit on the system. It appears as if the drain line was there first, and then the office unit was added on later as an afterthought because it is so close to the cleanroom chilled water line.

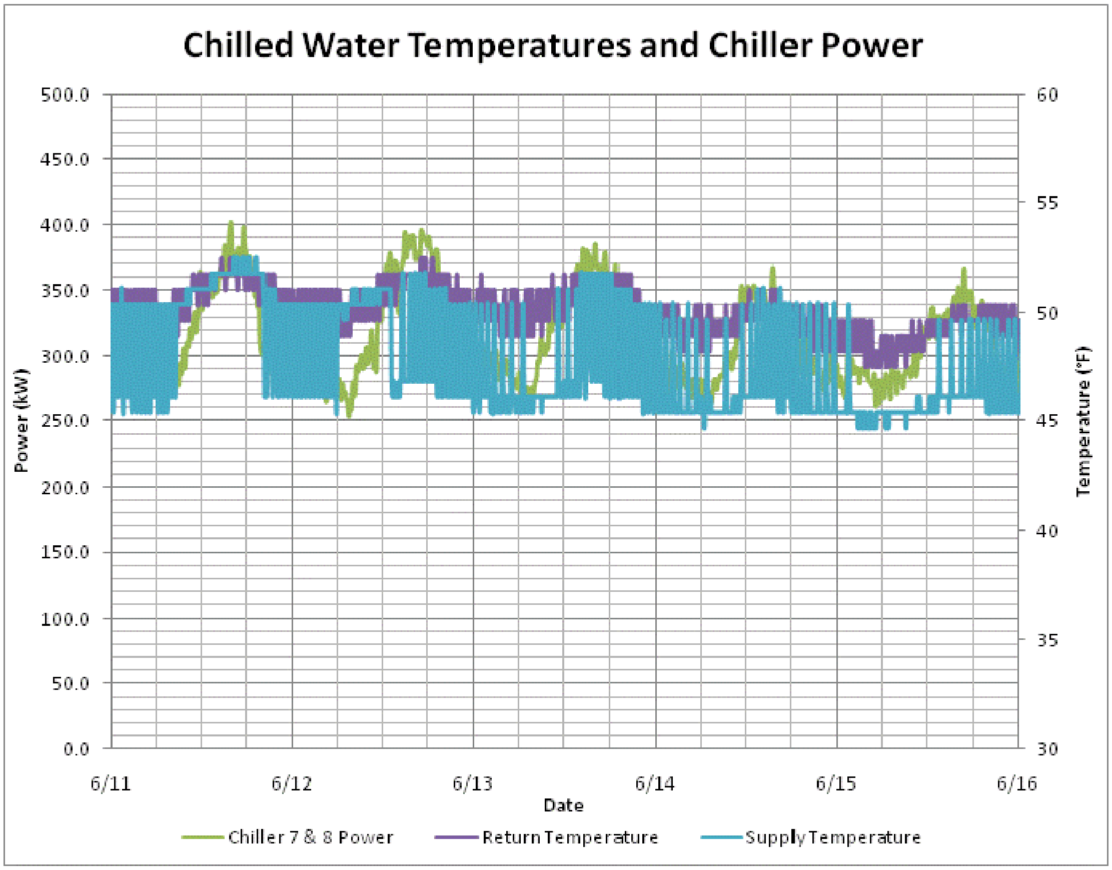

Figure 4.5: Chilled Water Temperatures and Chiller Power, June

In the figure above, notice the data for June 11th in particular. The chillers have loaded up during the day, but, if the chilled water temperature data is to be believed, to no avail. Just to be clear, the date is marked at the beginning of the day, i.e. midnight. So the chilled water temperature does not begin to get colder until it suddenly drops 5°F at 8:30 PM on June 11th. In the previous paragraph, I alluded to the possibility of thermal lag as a possible reason for the large delay, but if this were the case then the temperature would not suddenly be able to drop 5°F in 5 minutes, which is the time between data points. On this date, a normal summer day, it would seem that the office A/C unit would be running, but apparently it is not, until it runs all night long. In addition, June 12th does not seem to have this problem. This graph is representative of all data taken from the beginning of data collection in June until July 11th. The temperature data simply cannot be taken as reliable in this timeframe.

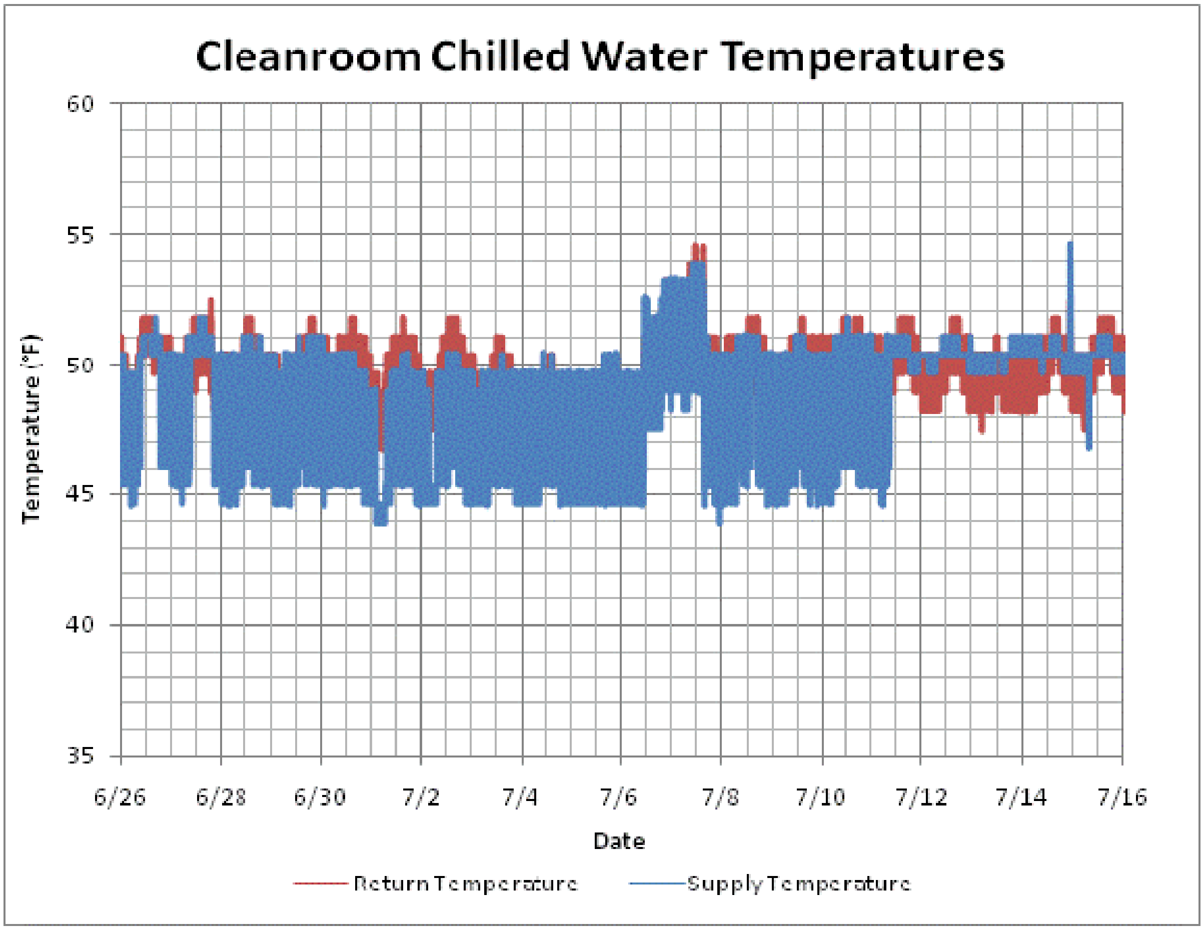

Figure 4.6: Cleanroom Chilled Water Temperatures

In Figure 4.6 above, a typical three week period is shown. Note the ~5°F variation in supply temperature which suddenly ends on 7/11. Just as the graph previous to 7/11 is representative of all the data recorded since June, and the period is after 7/11 is typical of all the data recorded afterwards until September. The supply temperature after 7/11 seems to be recorded as warmer than the return temperature. This actually leads to a slight bit of confusion in August as to which temperature is supply and which is return, since both temperatures average out to be within 0.5°F of each other. In either case, it does not satisfy the first law of thermodynamics, as energy would have to be both created and destroyed. (The other option would be that the cleanroom HVAC system is in some way massively broken. However, we know the system is operational since the cleanroom is receiving cold air from the AHU's.) It will be assumed that the data between July and September is also of little or no practical significance to the problem at hand. A possible exception is presented in Section 4.5.

What about the rest of the chilled water data?

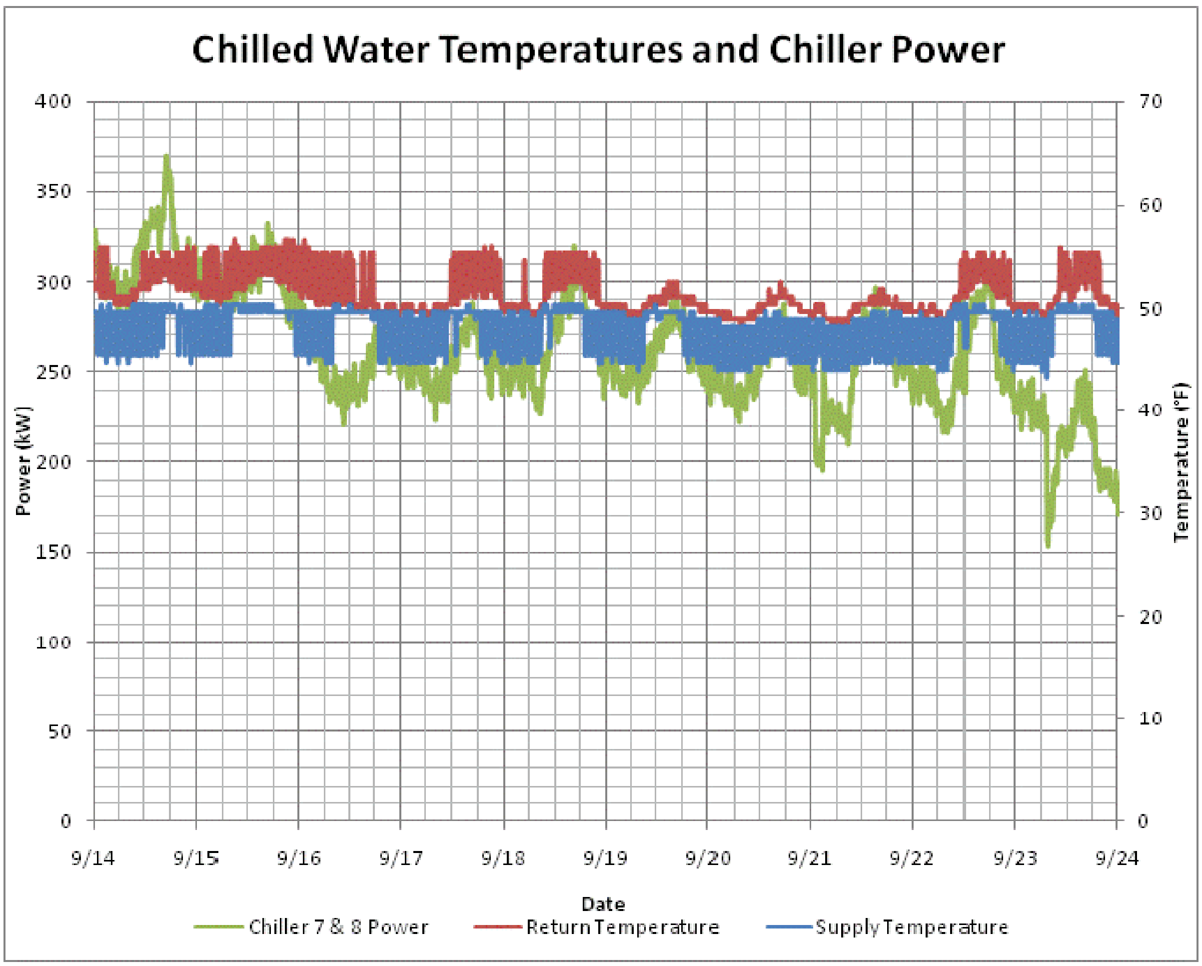

Figure 4.7: Chilled Water Temperatures and Chiller Power, September

As shown in Figure 4.7 above, the chilled water temperature difference does look more reasonable in places, but not substantionally so. There really is no reason why the chilled water temperatures would increase and decrease so rapidly. If the small A/C unit is the chief driver on the measured supply and return temperatures, it is thought that both the supply and return temperatures would be affected. In any case, the chillers themselves are in modulation mode, and thus the chilled water temperatures should have a fairly steady profile, not random swings of 5°F in both supply and return. Also note that we do not see any affects from the extreme change in cleanroom conditions that occur on 9/22. (see Section 4.3.1)

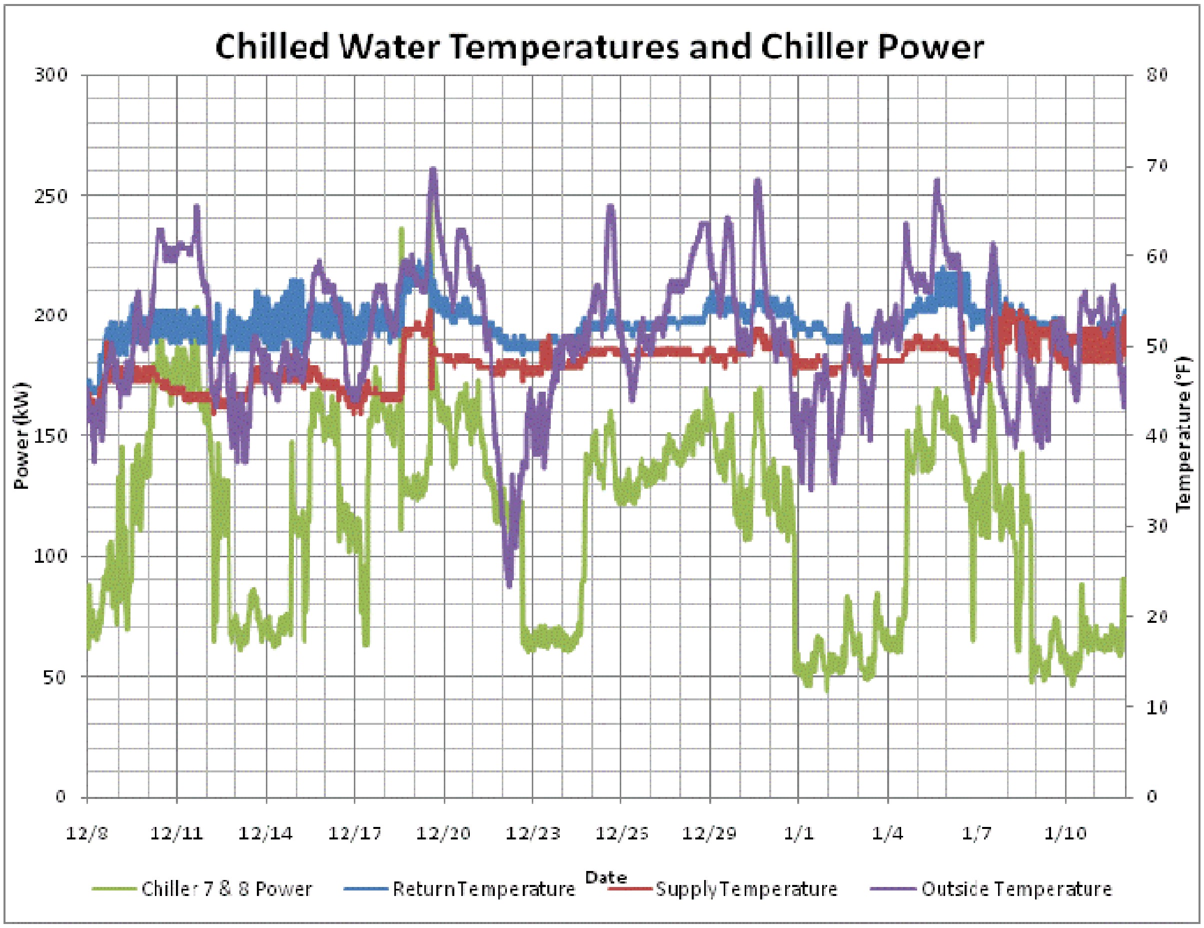

Figure 4.8: Chilled Water Temperatures and Chiller Power, Winter

In the graph above, it is key to note that much of the variation in chiller power is explained by winter production demands, and is not driven by the outside temperature. Still, the encouraging aspect is that a much smoother and more reasonable supply and return temperature profile is seen.

The three degree temperature difference between supply and return the week of Christmas is entirely reasonable, since the chillers are operating at a fraction of the full load power.

4.1.4 AHU 4

With AHU 4, the trouble is not in the data recording, but rather in the circumstances surrounding collection. Like all of the AHU's, AHU 4 has 3 legs of reheat recorded, but only one leg of the total power. A quick glance at the figure below will show the problem, as illustrated on July 5th, for example. How can the current for one leg of reheat be greater than the current for total? The answer is somewhat obvious: The leg which recorded the total power clearly is not the same leg that reheat 1 is recorded on.

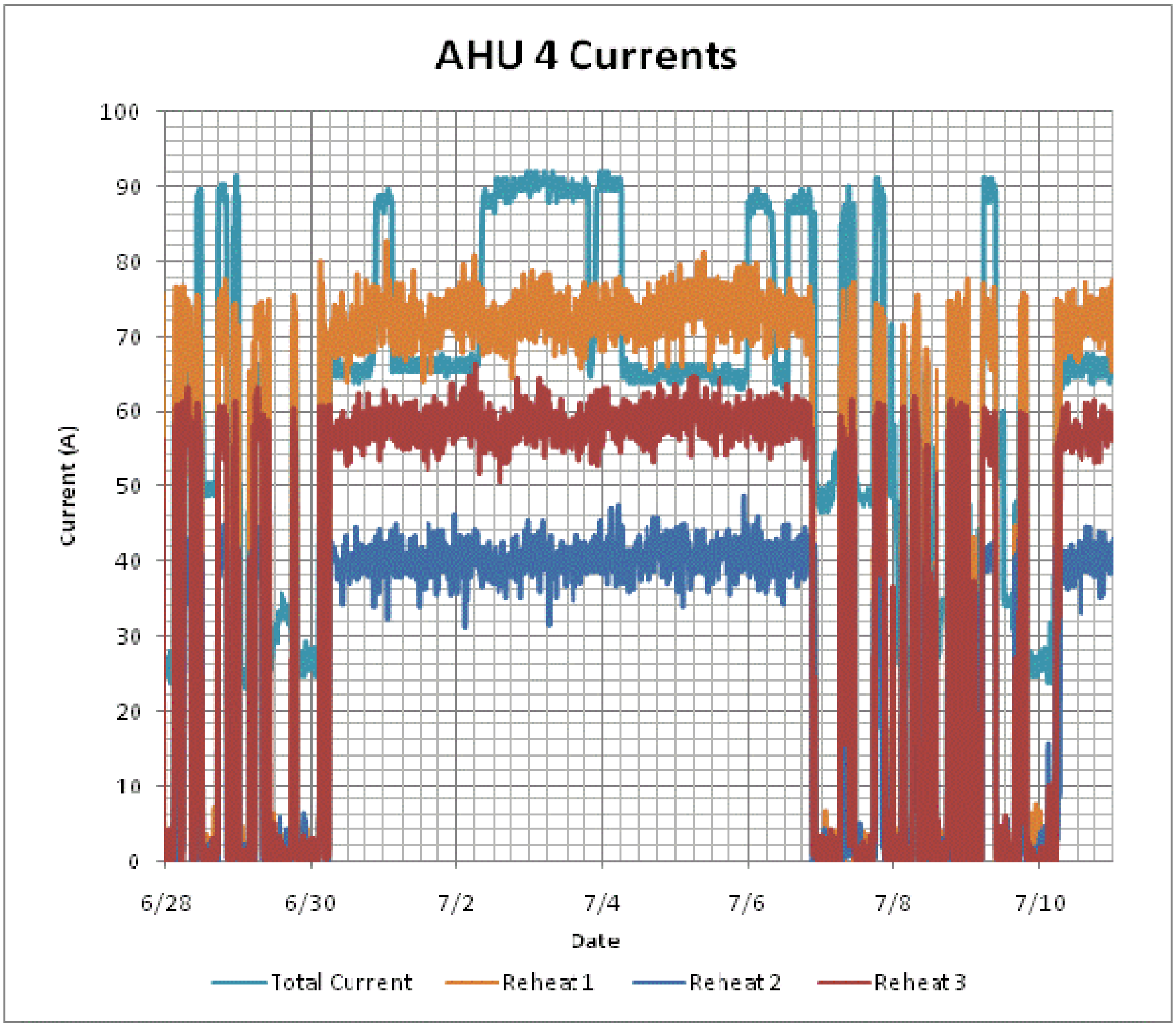

Figure 4.9: AHU 4 Currents

So what should be done? Assuming that the low fan power is around 5 kW, as it is on the other units, allows us to confirm that the leg that records the total power is the same leg that holds reheat 3. Since reheat 3 is typically the middle value, this ends up making a very small difference in the final results, ≥ 1 kW. However, it is important to mention, since this is a relatively easy mistake that can easily skew the results of an otherwise well thought out experiment.

4.1.5 Cleanroom Dew Points

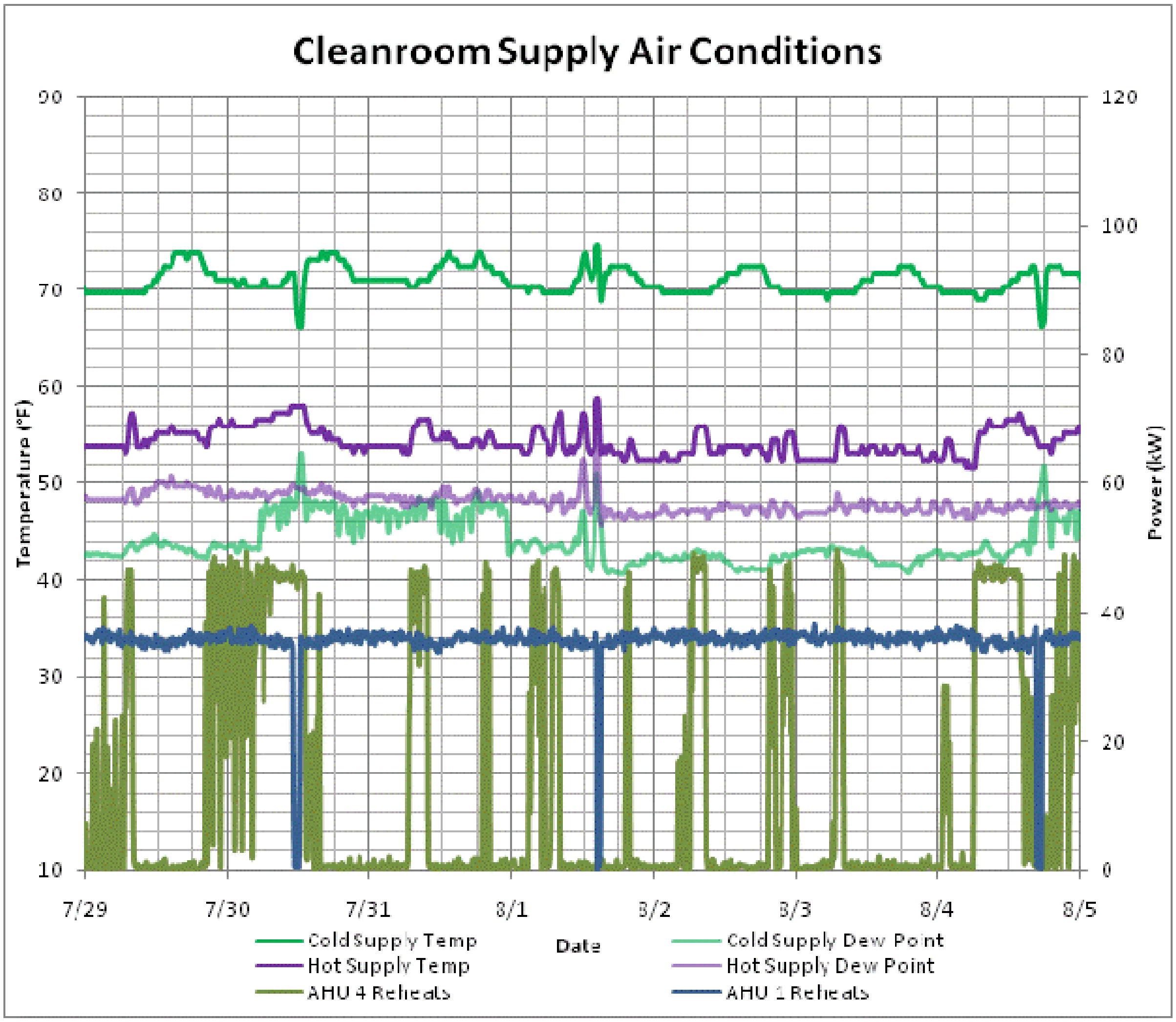

Figure 4.10: Cleanroom Supply Air Conditions

Consider Figure 4.10 above. The temperatures and dew points of all of the air above is all supply air, so the air coming out is either at saturation, or has been reheated. Recall that the logger labeled "cold supply" is in zone one, which is fed by AHU 1, and the logger labeled "hot supply" is in zone four, which is fed by AHU 4. The response of both temperatures to the reheat power is pretty clear for AHU 1, and the same for AHU 4, although a little less obvious (this will be explained later in Chapter 4 in detail). In any case, the problem with the dew point reading for the cold supply air is obvious: it is way too cold, bottoming out at 41°F. None of the other dew point readings ever get that low in the summertime. It would also require the chiller supply temperature to be in the upper 30's, which is simply not the case.

4.2 Cleanroom Results

The results of the cleanroom study are very interesting. Using cleanroom temperature data, AHU 4 reheat can be verified thermodynamically. There are several examples of this on Figure 4.10 above. Take the AHU 4 reheat spike on the morning of 8/2 from Figure 4.10 (also Appendix A, Figure A.1). Since this is all sensible heating, a standard Cp of air = 0.24 Btu/(°F lb) can be used. The temperature changes from 52.5°F to 56°F and back to 52.5°F again, while the reheat goes from 0 to 47 kW to 0 again. We can compute the flow rate of air through the AHU, with all other conditions being constant.

$$47\ kW \times \frac{3{,}413\ BTU}{kW\ hr} = \dot{m}_{AHU\ 4} \times \frac{0.24\ BTU}{°\text{F}\ lb}(56°\text{F} - 52.5°\text{F}) \qquad (4.1)$$

$$\dot{m}_{AHU\ 4} = \left(\frac{160{,}411\ {}^{BTU}/_{hr}}{0.84\ {}^{BTU}/_{lb}}\right) \Bigg/ \left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right)$$

$$\dot{m}_{AHU\ 4} = 42{,}440\ {}^{ft^3}/_{min}$$

This results in a calculated flow rate through AHU 4 which is very close to the 45,000 cfm the AHU is supposed to have. Since the temperature loggers have a 0.69°F resolution, this is essentially dead on. It also is representative of the rest of the summer.

If the reheat explains small rises in temperature, then what exactly explains the 6-8°F difference between the hot supply dew point and the hot supply temperature?

It is well known that humidistats are fickle in nature. So, four HOBO's identical in type *and* age to those in use in the field were tested in the lab to determine their reliability, accuracy and precision. As a control, two different types of handheld instruments were tested, as well as two brand new late-model HOBO's. The four older HOBO's tested had dew points between 2 and 8°F below that of the control, while the control had readings very similar to each other, within a degree. While not definitive proof of inaccuracy, it does provide a very plausible explanation for the difference between the hot supply dew point and the hot supply temperature. Also of note, all of the HOBO dew point data follows the same exact trend, just shifted in magnitude. Therefore dew point trends seemingly can be trusted more than dew point magnitudes.

However, temperature readings were used to help build the case against dew point readings. If the dew points are off, can the temperature readings be off, too? Well, at the same time, all of the temperatures on the same loggers were tested. All of the temperatures registered within a degree or closer to each other. Again, while this is not definitive proof of logger temperature accuracy, it lends credence to the assumption that temperature readings will be much more accurate than dew point readings.

The temperatures of the hot side as recorded can peak over 90°F, in the location of the data logger. Figure 4.11 below illustrates how the cleanroom temperature typically rises in the daytime and falls at night. However, exceptions can easily be found. This stands in contrast to the chiller power vs. outside temperature relationship discussed later in the chapter. What in the controls helps to cause these temperature swings? Could this be production related?

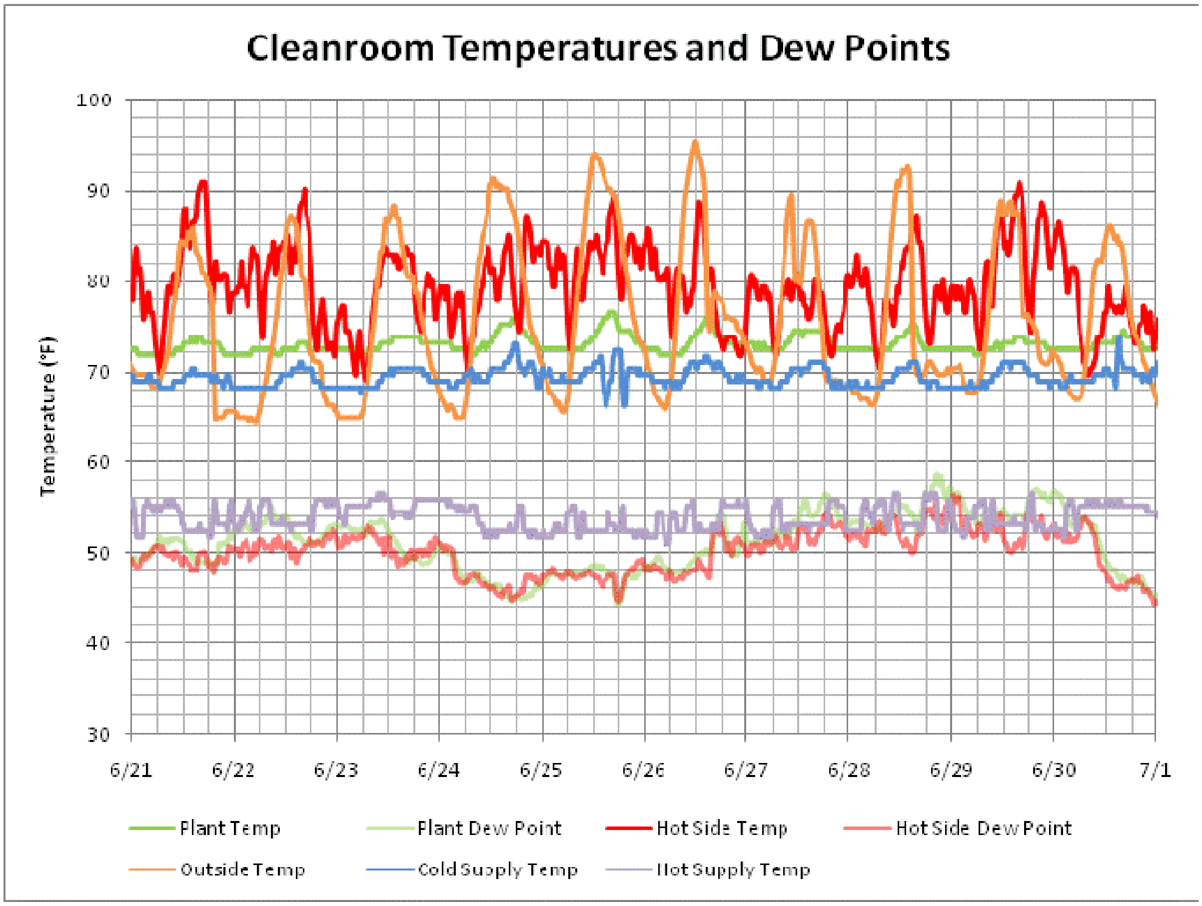

Figure 4.11: Cleanroom Temperatures and Dew Points, June

Consider Figure 4.12 below, where cleanroom temperatures have been graphed vs. AHU power use for a single week. The cleanroom temperature is the coolest where the reheat is the MOST. This completely defies logic, on the surface at least. However, upon reflection, this is exactly what the initial hypothesis predicted. Remember that the initial hypothesis stated that there was not enough of a heat load in the room to effectively reheat all of the air to the desired final temperature. One of two things could be happening. Either the cooling load greatly increases, in which case this would be reflected in the when the chillers increase their load, or the production load is decreased, necessitating more reheat. Figure 4.13 illustrates that it is not caused by an increase in cooling. Therefore it is most likely production related. A cause effect time delay relationship cannot be established because of the 24 minute time elapse between data points.

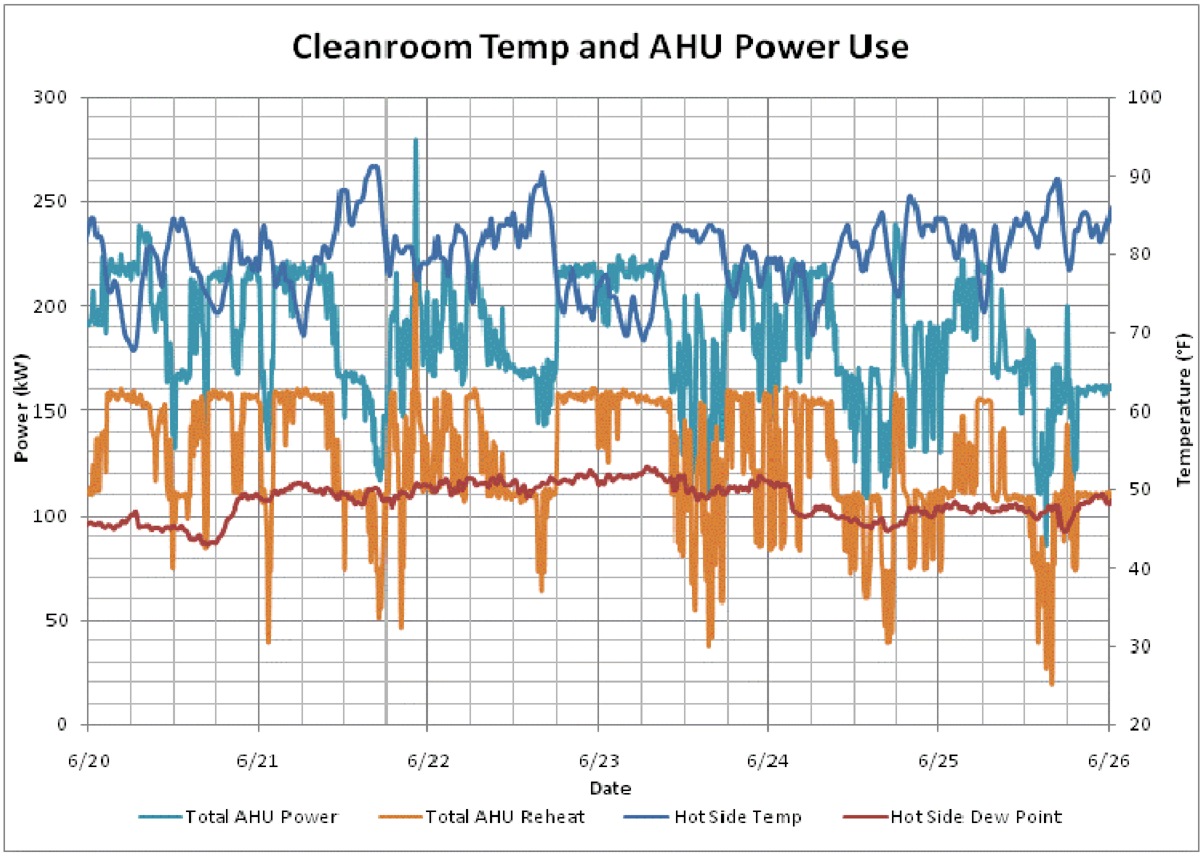

Figure 4.12: Cleanroom Temperatures and AHU Power Usage

Notice also that for this particular week in time, the total power is in large part just shifted up from the total reheat. At first, the times of heat and cold seem to be quite random, but during on peak times the AHU Power during the timeframe from 6/20 to 6/30 is 8 % LOWER, which is equivalent to 14 kW. Remember that the on-peak timeframe is an arbitrary 8 hour limit set by Duke Energy, and extending past sunset to 9 p.m. This reinforces the theory that reheat power is tied to production. Production may simply be down slightly at night.

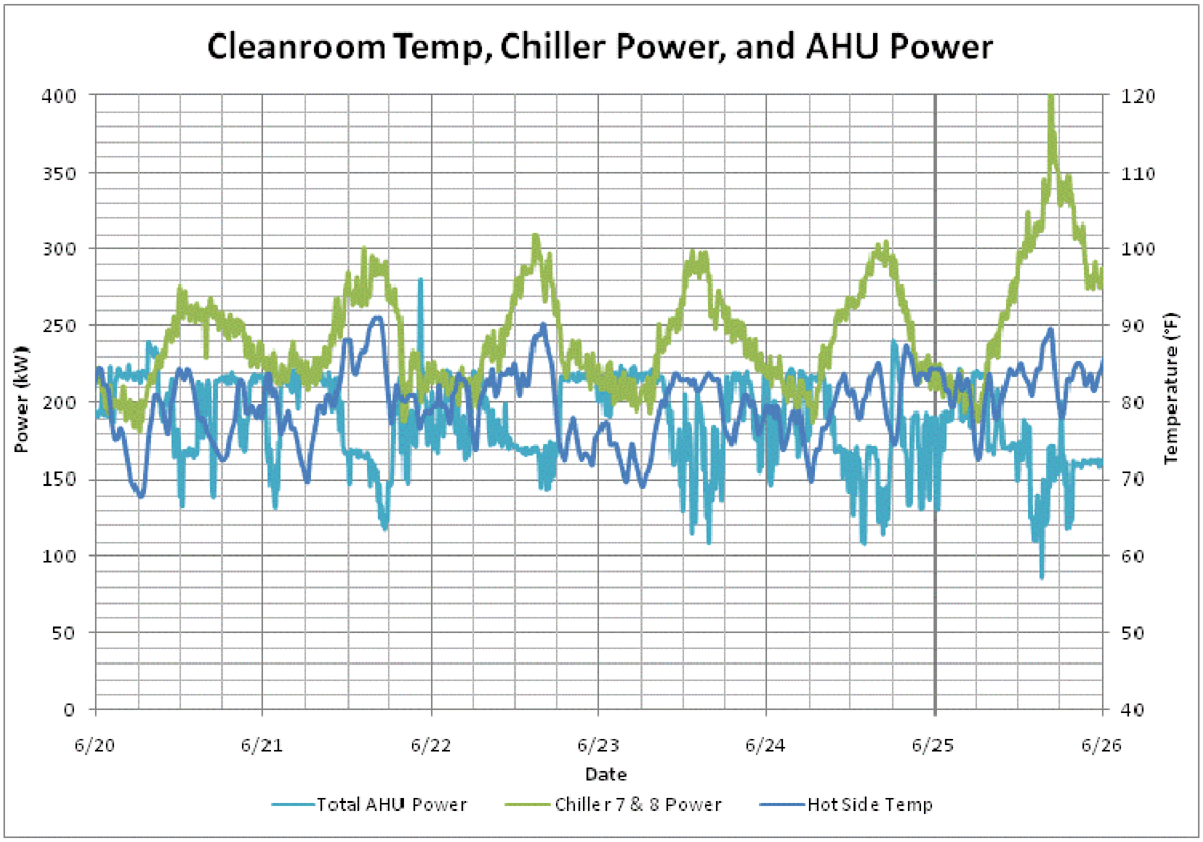

Figure 4.13: Cleanroom Temperatures, Chiller Power, and AHU Power

Figure 4.13 above shows the temperature and reheat correlation against the chiller power. Chiller power looks like it correlates somewhat, but the magnitude of reheat increase and decrease is much less than the magnitude of reheat increase and decrease this will be discussed in detail in Section 4.5.

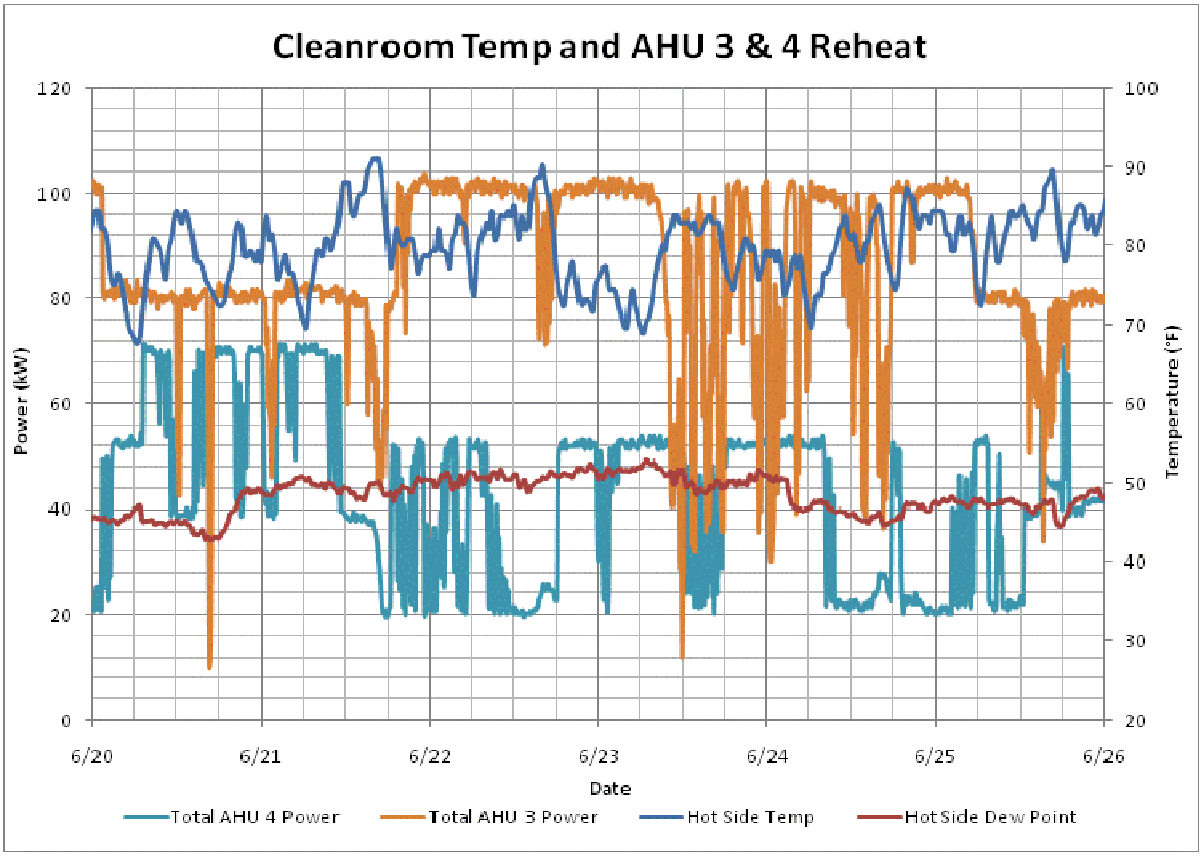

Figure 4.14: Cleanroom Temp and AHU 3 & 4 Reheat

From the plant layout in Chapter 1, it is shown that the hot side of the room is located in zone 4. Since the reheats for 3 and 4 are by far the most active during this time period, these two are graphed to see if AHU 4 reflects quicker on temperature fluctuations. It does need to be said that because of the 24 minute time interval between temperature readings, this kind of conclusion will be hard to reach. That being said, the cleanroom zone 4 (hot side) temperature seems to be a function of all of the AHU reheats (as shown in Figure 4.12), not just AHU 4. On 6/23 and 6/24 the AHU 4 power stays constant for several hours at a time, while other AHU currents fluctuate, and this seems to produce some temperature changes. This result isn't conclusive, but it does help explain why zones 3 and 4 seem to have most of the used reheat capacity, even though zones 1 and 3 need more reheat.

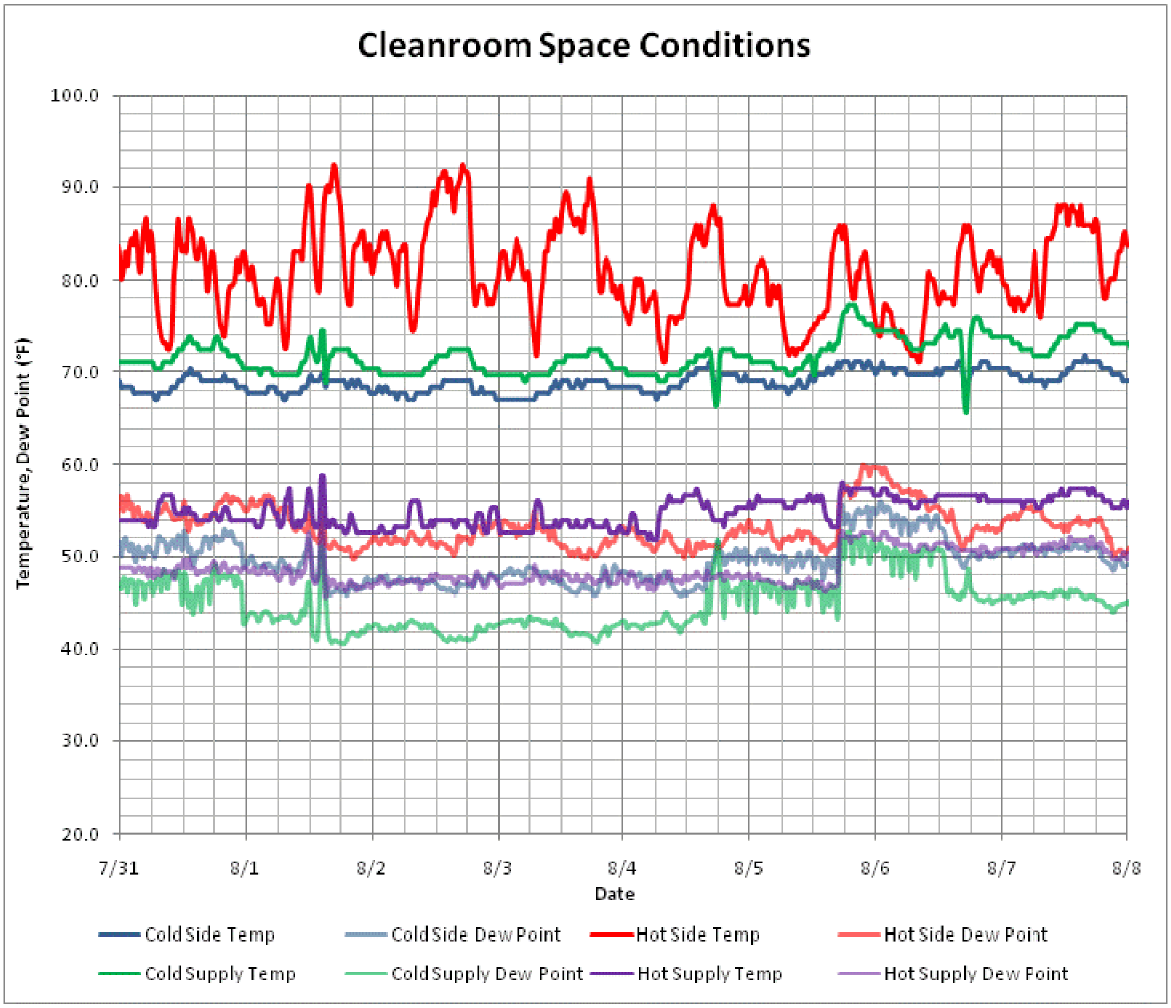

Figure 4.15: Cleanroom Space Conditions, August

The above graph shows cleanroom conditions in two zones for a week in early August. At four separate locations, temperature and dew point data was logged. The cold and hot side data reflect room temperatures in zones 1 and 4, respectively, while the cold and hot supply data reflect the supply air conditions into those areas. It is assumed that the other two zones are close to the conditions presented above, depending on whether or not they are in the cold or hot zone. Notice the cold supply temperature and dew point. The increase in cold supply dew point for small contiguous segments of time was noted in Section 4.1.1, and is mirrored by an increase in the cold side dew point. Also note the jump in dew points on 8/5. This will be of interest in the chiller results section.

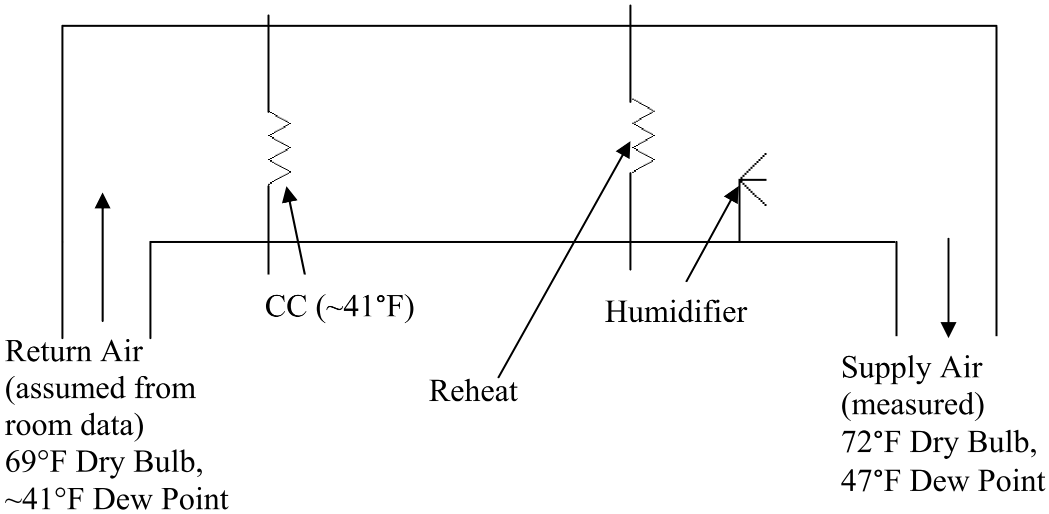

Figure 4.16: AHU 1 Energy Setup

Recall the Sample AHU Setup, Figure 1.5. This AHU looks physically like that one, but the assumed and measured temperatures have been added in Figure 4.16.

The real interest here is the difference between the cold supply dew point and the cold supply temperature. On the afternoon of 8/2, for example, the cold supply temperature is 72°F, while the cold supply dew point is recorded at 41°F. In Zone 1, therefore, there should be an astonishing 31°F of sensible reheat on approximately 45,000 cfm of air flow, with no heat recovery at all. If dehumidification is occurring as assumed, then the air leaving the cooling coil is saturated and 41°F.

The low dew point problem is addressed somewhat in Section 4.1.5. Even further though, when the actual loggers used were tested for dew point accuracy, the one logger which reads the lowest dew point by several degrees is the cold supply logger. It is worth noting that the temperature on this unit still reads accurately. Even if a dew point of 45-50°F is assumed there still needs to be a minimum of 21°F of sensible reheat on that large flow rate of air, or about 300 kW worth. The problem here is that AHU 1 records an almost constant reheat of 36 kW, not 300.

Recall how the cleanroom HVAC system is setup, from an energy standpoint. First, the air goes through the fan, and gets the fan energy added. Next, it goes through the cooling coils, and is cooled uniformly to the desired dew point, such as in Figure 4.16. This also takes out all of the added fan energy. Then, it goes through the reheat coils, and, if necessary, the humidifier. Therefore, the only source of heat for the cold supply in zone one is the cleanroom is the reheat on AHU 1. At a reheat of 36 kW, and for an assumed dew point/ leaving coil temperature of 45°F, the flow rate for zone one would be as follows:

$$36\ kW \times \frac{3{,}413\ BTU}{kW\ hr} = \dot{m}_{AHU\ 1} \times \frac{0.24\ BTU}{°\mathrm{F}\ lb}(72°\mathrm{F} - 45°\mathrm{F}) \tag{4.2}$$

$$\dot{m}_{AHU\ 1} = \left(\frac{122{,}868\ ^{BTU}/_{hr}}{6.48\ ^{BTU}/_{lb}}\right) \Big/ \left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right)$$

$$\dot{m}_{AHU\ 1} = {}^{4{,}210\ ft^3}/_{min}$$

This is a whole order of magnitude less than the design 45,000 cfm, so assuming a dew point of 50°F really does not change the point. Suppose as a second possibility, for a moment, that there is little or no cooling in AHU 1. If this is the case, then the ~4 kW fan energy would be included in the reheat, increasing the reheat energy to about 40 kW, on average. It would also explain why the cold supply temperature is recorded as being consistently about 3°F greater than the cold side temperature (see Figure 4.17 below), if it is assumed that the cold side temperature is close to the cold return temperature. On 8/2, for example, the cold supply temperature in the room is 72°F, and the cold side temperature is 69°F.

$$40\ kW \times \frac{3{,}413\ BTU}{kW\ hr} = \dot{m}_{AHU\ 1} \times \frac{0.24\ BTU}{°\mathrm{F}\ lb}(72°\mathrm{F} - 69°\mathrm{F}) \tag{4.3}$$

$$\dot{m}_{AHU\ 1} = \left(\frac{136{,}520\ {}^{BTU}/_{hr}}{0.72\ {}^{BTU}/_{lb}}\right) \Big/ \left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right)$$

$$\dot{m}_{AHU\ 1} = {}^{42{,}140\ ft^3}/_{min}$$

This answer is almost right on the 45,000 cfm, considering the possible error in the temperature data. However, this second situation comes with its own problems. First, supply temperature would then be warmer than existing room temperature. This would only be a plausible situation if the cold air from the hot supply area was making its way over to the cold side of the room. The only heat gains in zone are minimal (from the light and fan/reheat load), which is really the only way this could even be a plausible explanation. Below it is shown that in the adjacent hot zone, zone 2, AHU 2 has essentially no reheat. This may be unlikely, since the exhausted air is leaving from the hot side of the room.

In order to do these calculations, several assumptions have been made about the recorded data, which should be listed explicitly. First, it is assumed that the temperature data is correct. Although it was shown in Section 4.1.5 that there is a basis for doubting the dew point data, it has not been shown that there is any reason to doubt the temperature data. Secondly, it is assumed that the cold side room temperature is the same as the cold side return temperature. Third, it assumes that the fan load is around 4 kW, and does not take into consideration the possibility that fan load may not be measured, explained in Section 4.4. Finally, it also assumes that there is either cooling to the dew point (much cooling), or no cooling at all. There is no allowance for a small (but appreciable) amount of cooling. These may or may not be correct assumptions.

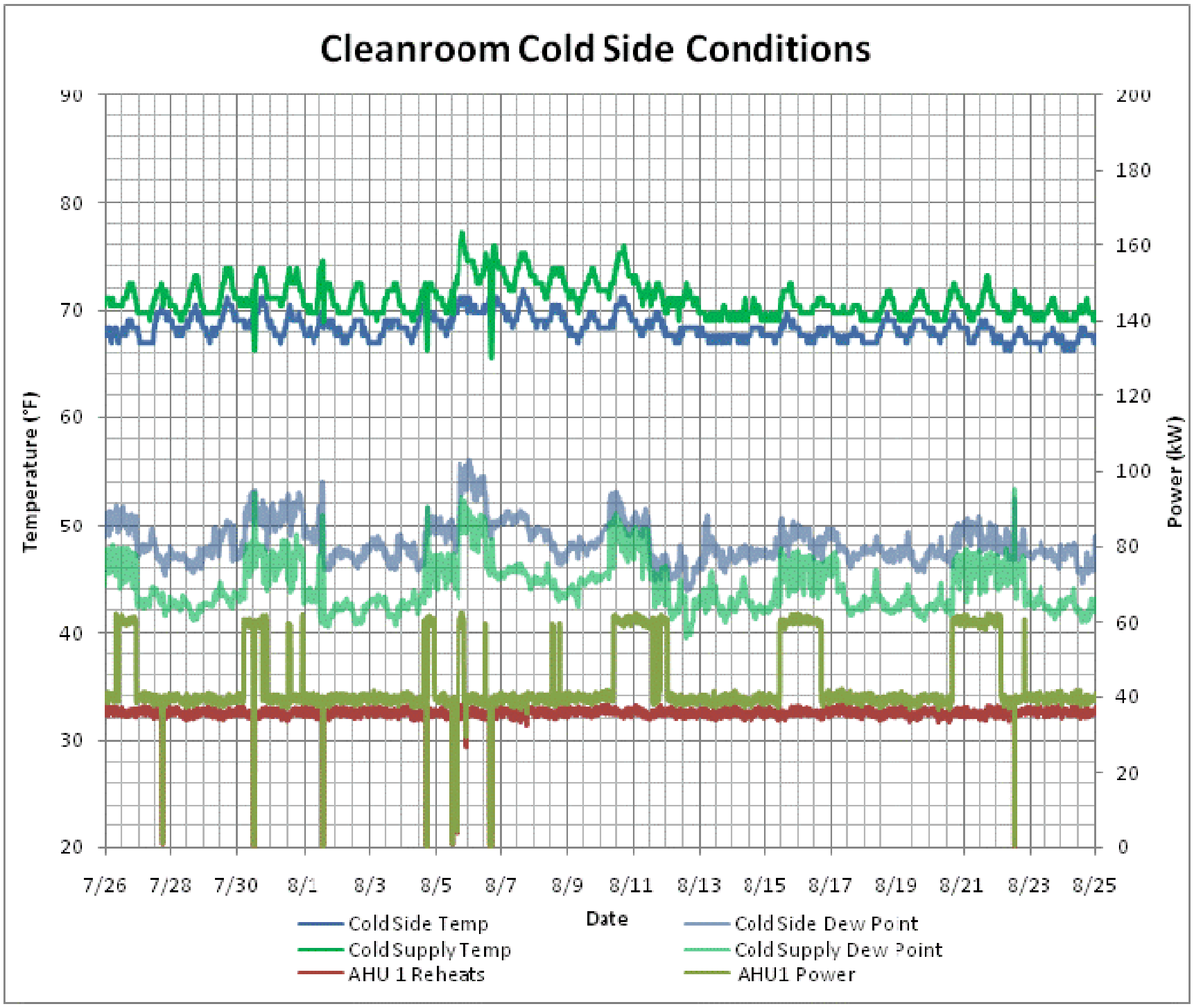

Figure 4.17: Cleanroom Cold Side Conditions, August

The figure above illustrates the position that the temperature for the cold side always stays between 2-4° F warmer than the supply. It is also shown that the supply temperature rises slightly more than that during the period between 8/5 and 8/11, which will be addressed later in Section 4.5. It also points to the possibility that there may be some small residual cooling, decreased by the higher chilled water temperatures. Another obvious curiosity is the continuation of the seemingly random spikes in dew point. It seems like the spikes in the AHU 1 total power are correlated to the spikes in supply dew point, but not perfectly. This is very puzzling. If the spike can be attributed to humidification, then it should correlate perfectly with raised supply dew point. One possibility is that this is functioning similarly to something mentioned in Section 4.1.4 with the AHU 4 reheats. Recall that since each reheat was operating at a different load, the total power (which is only measured on one leg) had the potential to be greatly misstated. The humidifiers are electric, and it is very possible that they could be staged by leg, instead of incrementally, and so therefore staged in a way hidden from the data logging equipment. However, without further proof, this is merely speculation.

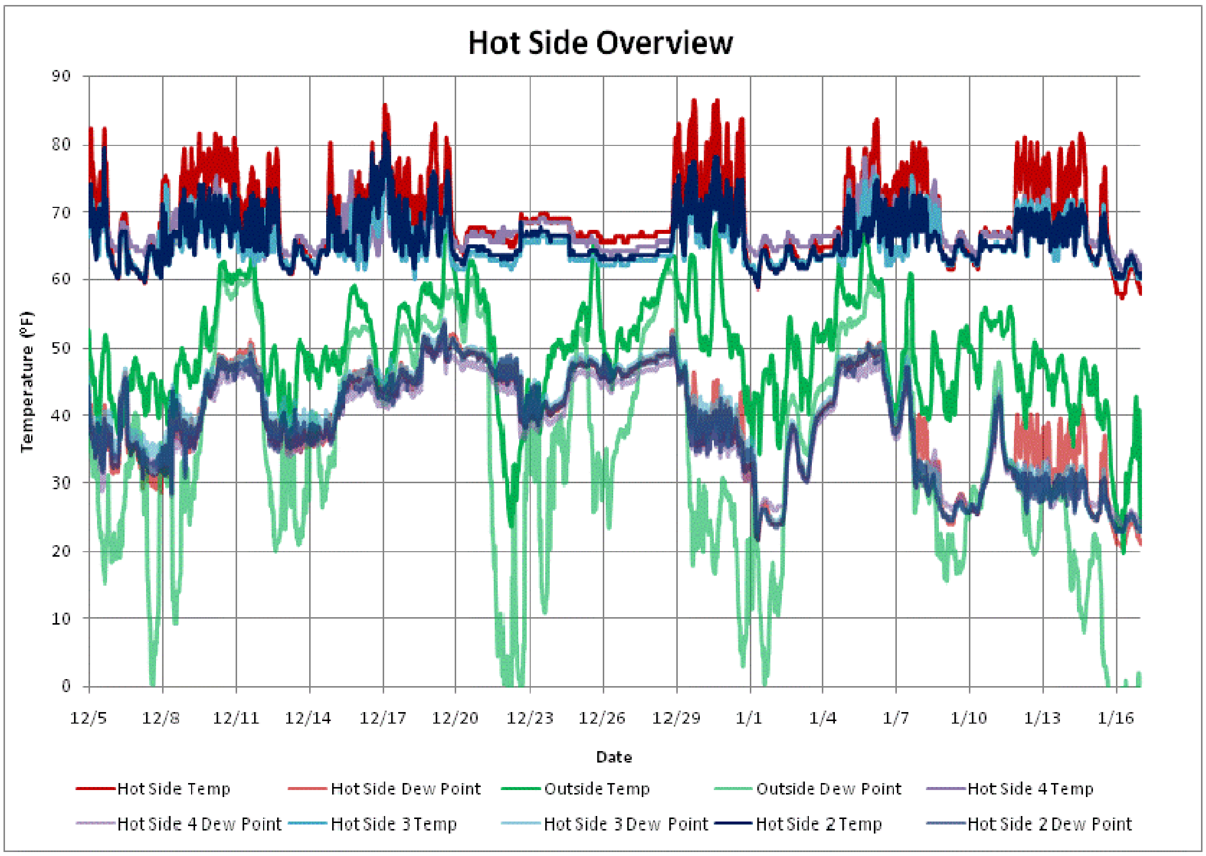

Figure 4.18: Hot Side Overview, Winter

Figure 4.18 above shows some wintertime cleanroom temperatures. Note how the temperatures and dew points for the four different loggers in the cleanroom are so similar. The hot side 1 temperatures are higher than the other three temperatures, but this is to be expected, as the other three loggers are placed in front of machines in areas with slightly more airflow. From this graph, days where the cleanroom are not in operation are visible as smooth ~65°F lines, such as Christmas week.

4.3 AHU Results

The different AHU's all seem to have typical operating conditions. Summer conditions are easy to identify, since production is fairly steady during these summer months. Figure 4.19 below shows a typical month. AHU 1 runs at a constant 35-36 kW of reheat from June until September 29th. AHU 2 is effectively at zero reheat. AHU 3 consistently runs highest. The reheats are rated at 70 kW, so AHU 3 is typically operating at the maximum its reheats can provide, until 9/22. During the time period below, for example, it operated with a load factor of 87%.   Anecdotal data suggests that this zone (zone 3) is always cold, according to plant employees. AHU 4 oscillated between a seemingly set maximum value and zero. This apparent maximum was at 47 kW before 8/14, and at 28 kW between 8/14 and 9/22.

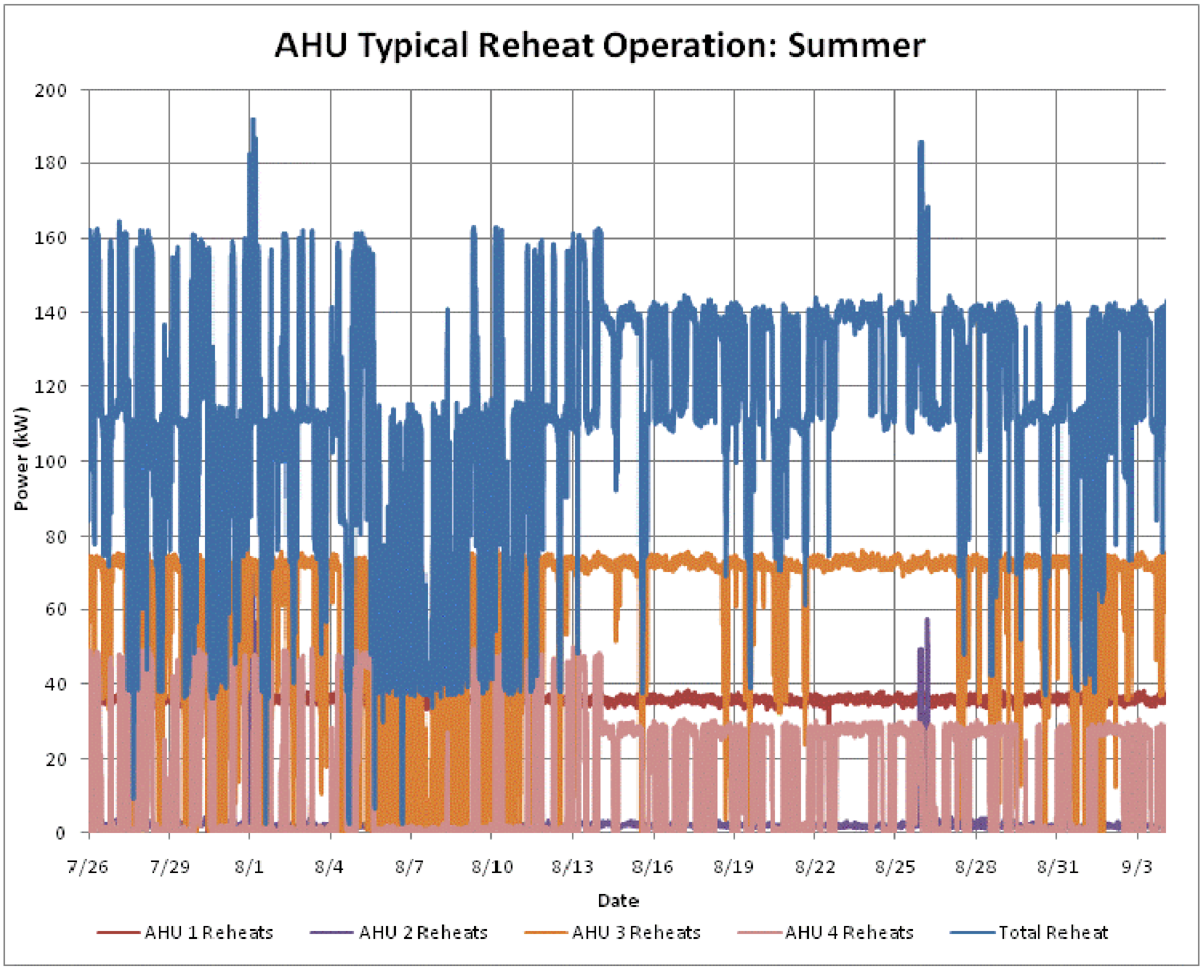

Figure 4.19: AHU Typical Reheat Operation, Summer

The total reheat is seen to operate on a fairly consistently variable basis. The data between 8/5 and 8/11 is much lower, and this is likely related to the changes in chiller operation mentioned in Section 4.5.

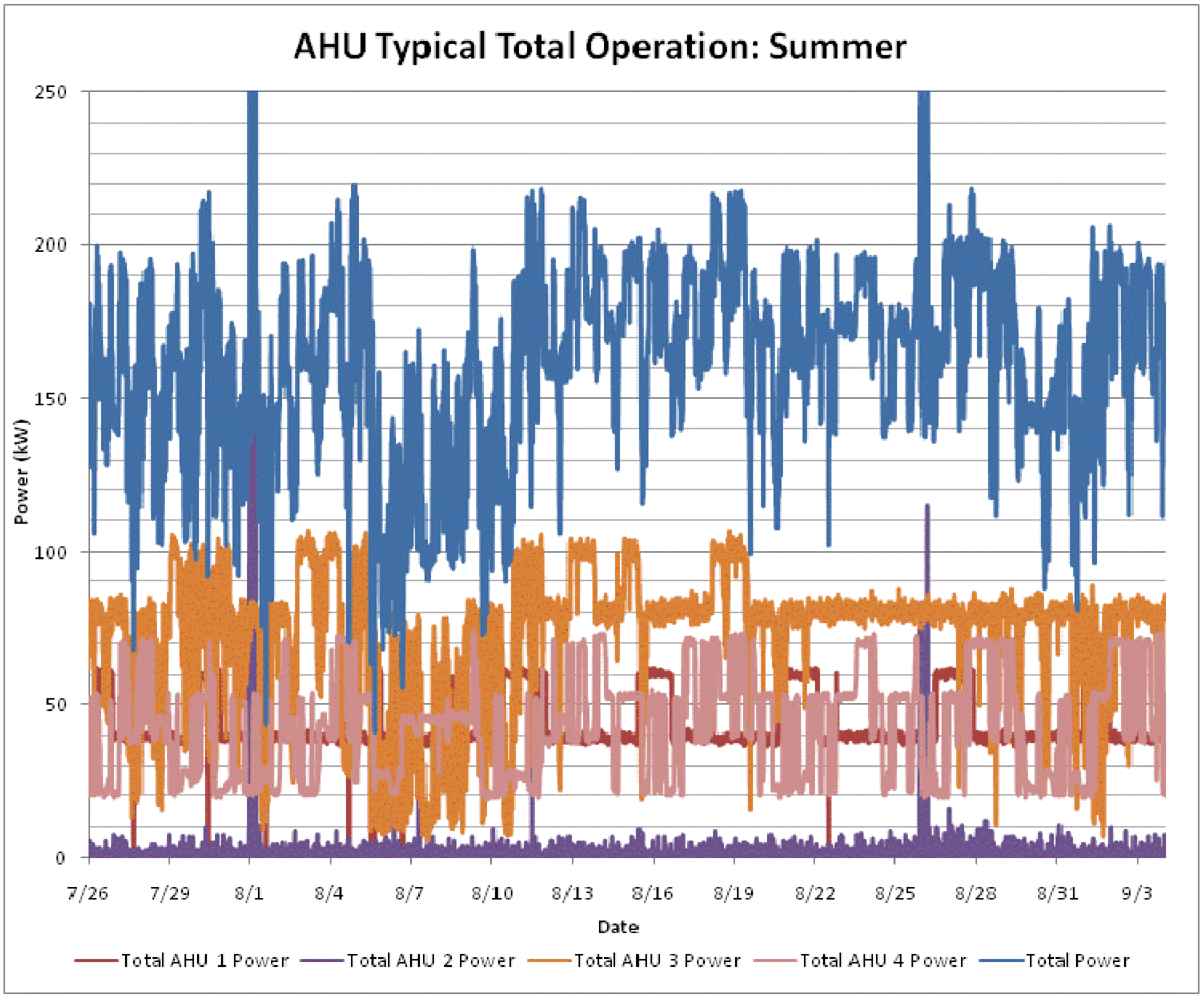

Figure 4.20: AHU Typical Total Operation, Summer

The total AHU power includes reheat, and thus is a strong function of reheat. There is less predictability among the total power load, unlike the reheat load. Why is this? Fan speed should be fairly constant, and humidification should not be needed in the summer. Reheat is the load where the most variability is expected. This is not the case in reality.

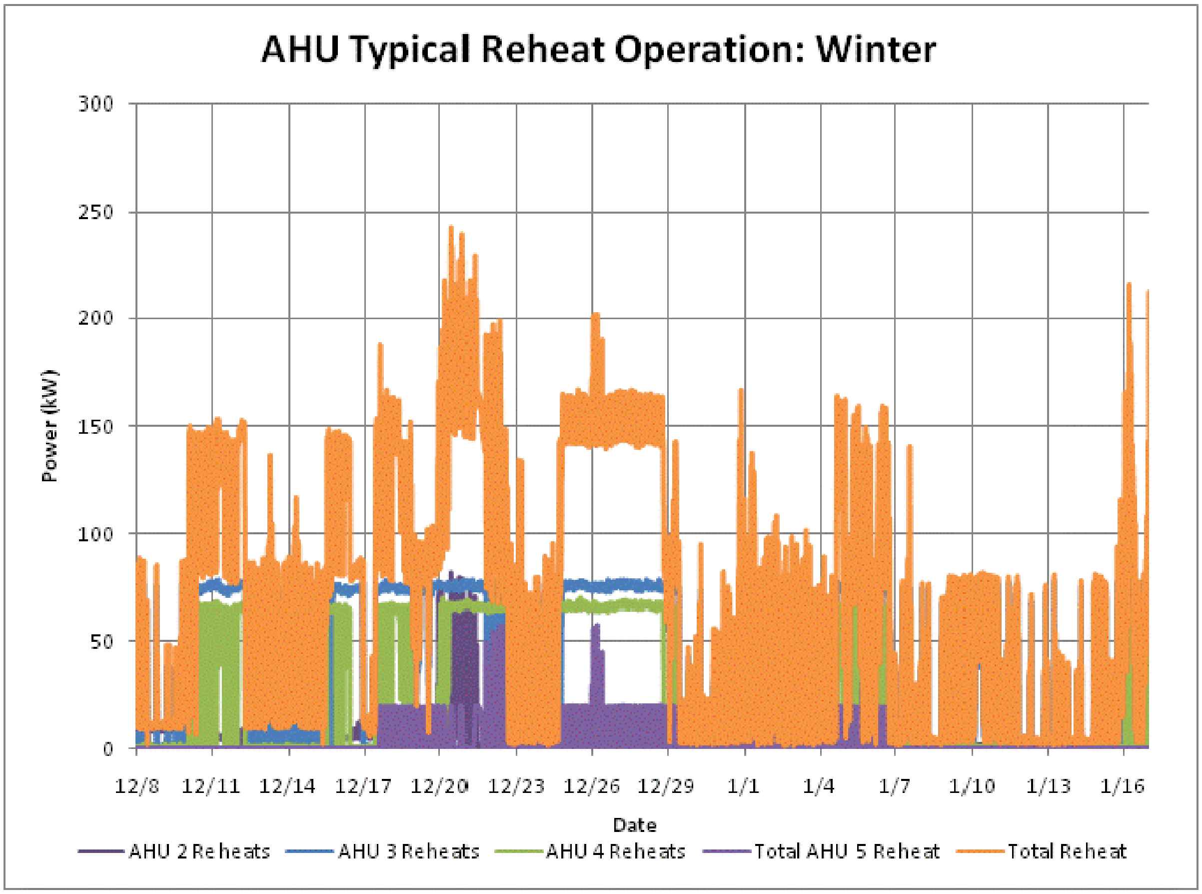

Figure 4.21: AHU Typical Reheat Operation, Winter

The winter operation is different. AHU 1 is not represented on the chart because it has been taken out of service for retrofitting, which will be described later in Chapter 5. Production in the winter is less than it was during the summertime, and this lack of production accounts for much of the reheat and chiller power consumption in the cleanroom.

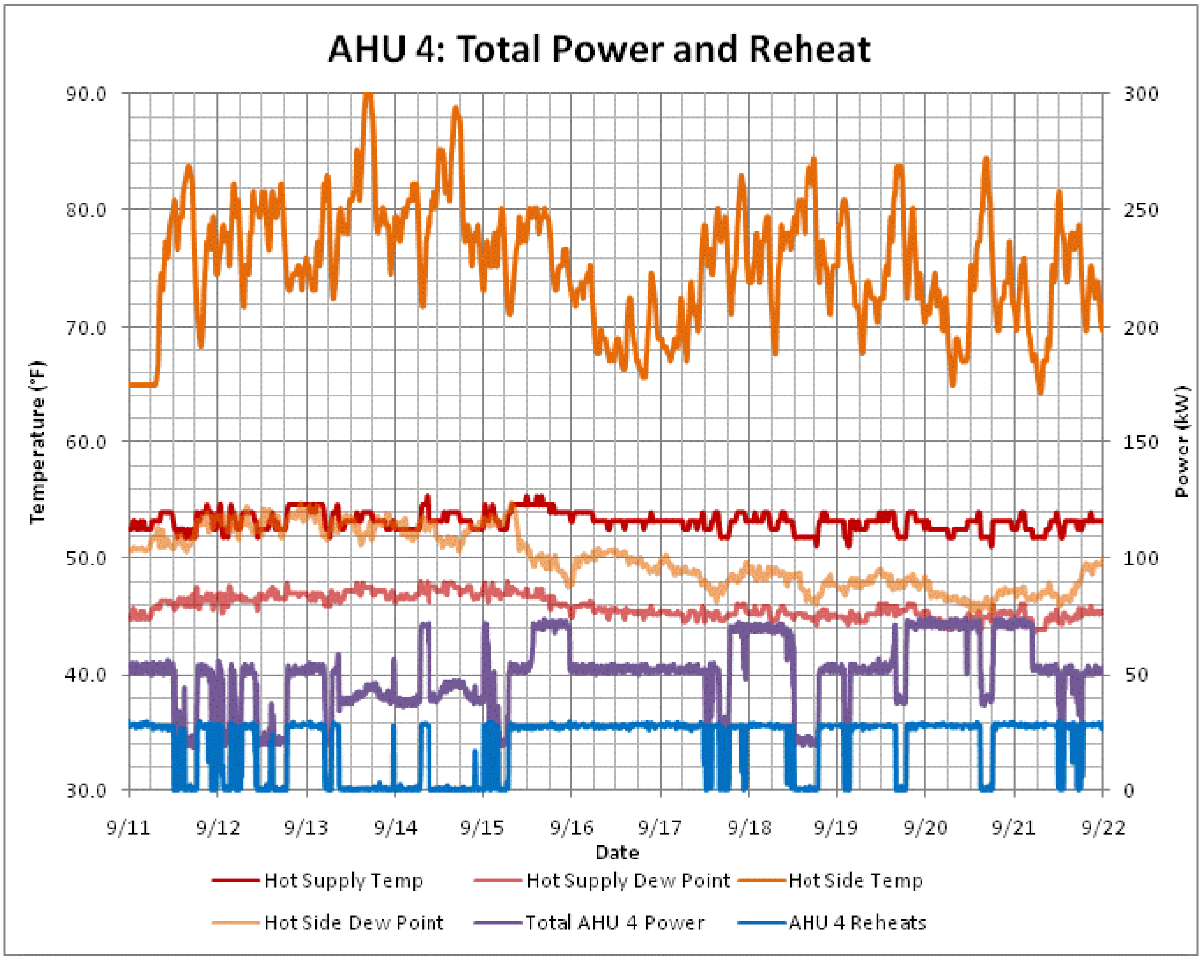

Figure 4.22: AHU 4: Total Power and Reheat

Consider Figure 4.22 above. It was stated previously there was a strong possibility that the sudden increases in power use (such as on the afternoon of 9/15) on the AHU's was due to an increased fan load. However, if the fan load is increased as hypothesized, then any reheat on the supply air will be proportionally diluted, resulting in a lower supply temperature. This does not seem to be the case on AHU 4, according to 9/15, 9/18, and 9/20-21 on the figure above. Another alternative presented is that the increase is really due to the humidifier. Again, this does not seem to be the case for AHU 4, since the supply dew point does not change at all. Then what accounts for the sudden increase in load?

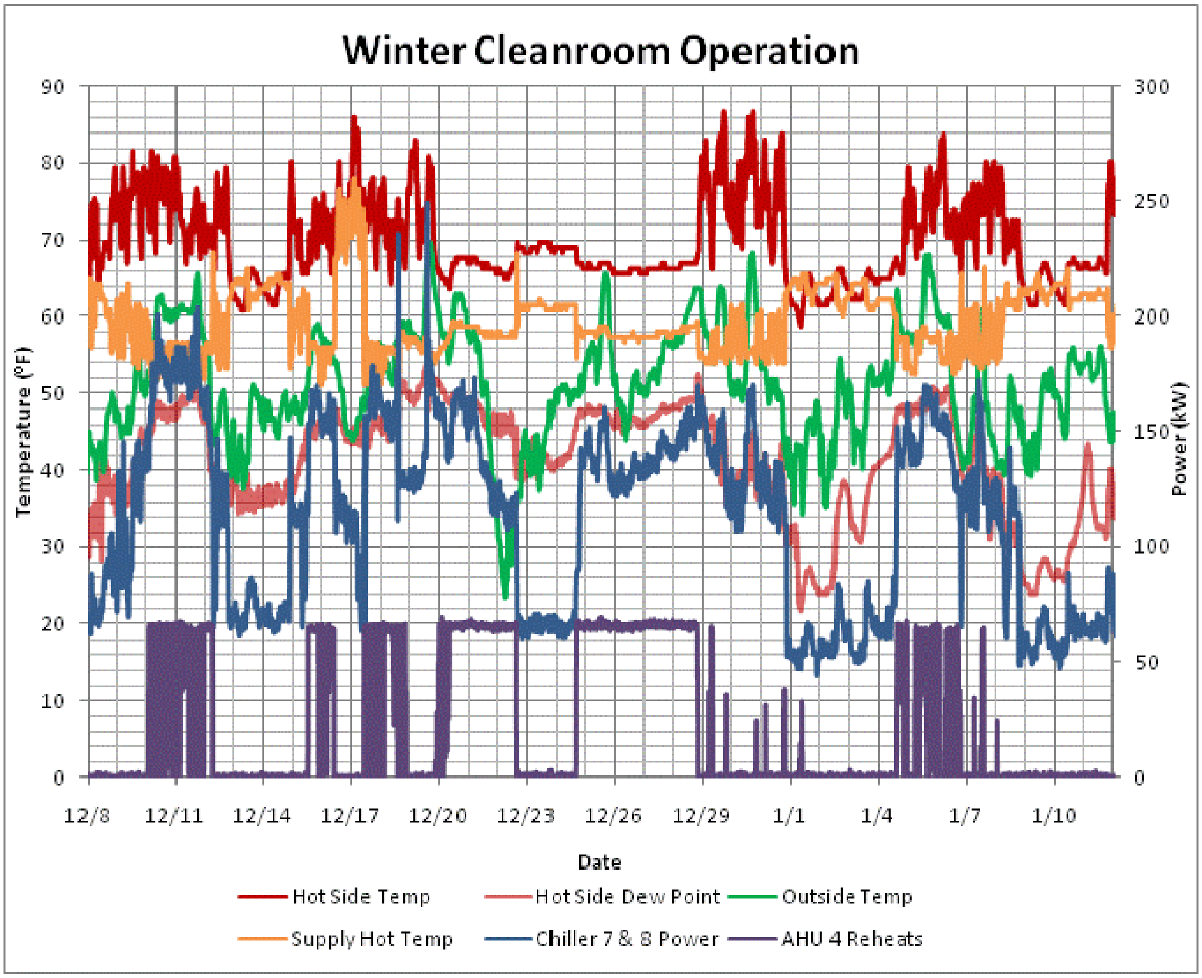

Figure 4.23: Winter Cleanroom Operation

The winter cleanroom operation is clarified somewhat by showing the reheat, chiller, and cleanroom temperatures in Figure 4.23. It appears as though there is an "economizer mode" that happens when the outside temperature goes below 45-50°F. However, that mode comes with its own problems. On 12/17, for example, the supply temperature is almost the same as the return, apparently due to the decrease in chiller operation. On 12/18, Chiller 7 shuts off, and this allows Chiller 8 to run much more efficiently, providing the same cooling with 30 kW less. There is also more opportunity to run the cleanroom more efficiently during shutdowns, like over Christmas.

4.3.1 The curious case of 9/22-9/29

On the morning of September 22^{nd}, something happened to the operation of the cleanroom. This is illustrated perfectly by Figure 4.24 below. At around 10:15 AM on 9/22, AHU 3 reheat goes from an average of 73 kW to zero. Then, an hour and a half later, AHU 4 reheat goes from an average of 28 kW to zero. Suddenly, 101 kW of reheat just stops. At 7:15 AM the next day, the 100 ton compressor on Chiller 7 switches off. The outside temperatures during this timeframe are nothing unusual; they vary from the mid 70's as a daytime high, to the upper 50's as a nighttime low. The dew point, as shown on Figure 4.25, dips below 50°F on the afternoon of 9/22, but this does not seem unusual.

Then, even more suddenly, on 9/29 between 1:15 and 1:30 PM, AHU 1 reheat changes from being a constant 35 kW to oscillating between 0 and 70 kW. AHU 3 returns to the same pattern of operation it had before 9/22, and AHU 4 now operates between 65 kW and 0, oscillating. Before 9/22, the reheats averaged 135 kW. Afterwards, from 9/29 to 10/21, they averaged 148 kW. From 9/22 to 9/29, they averaged only 45 kW.

The only logical explanation seems to be that somebody changed the setpoints.

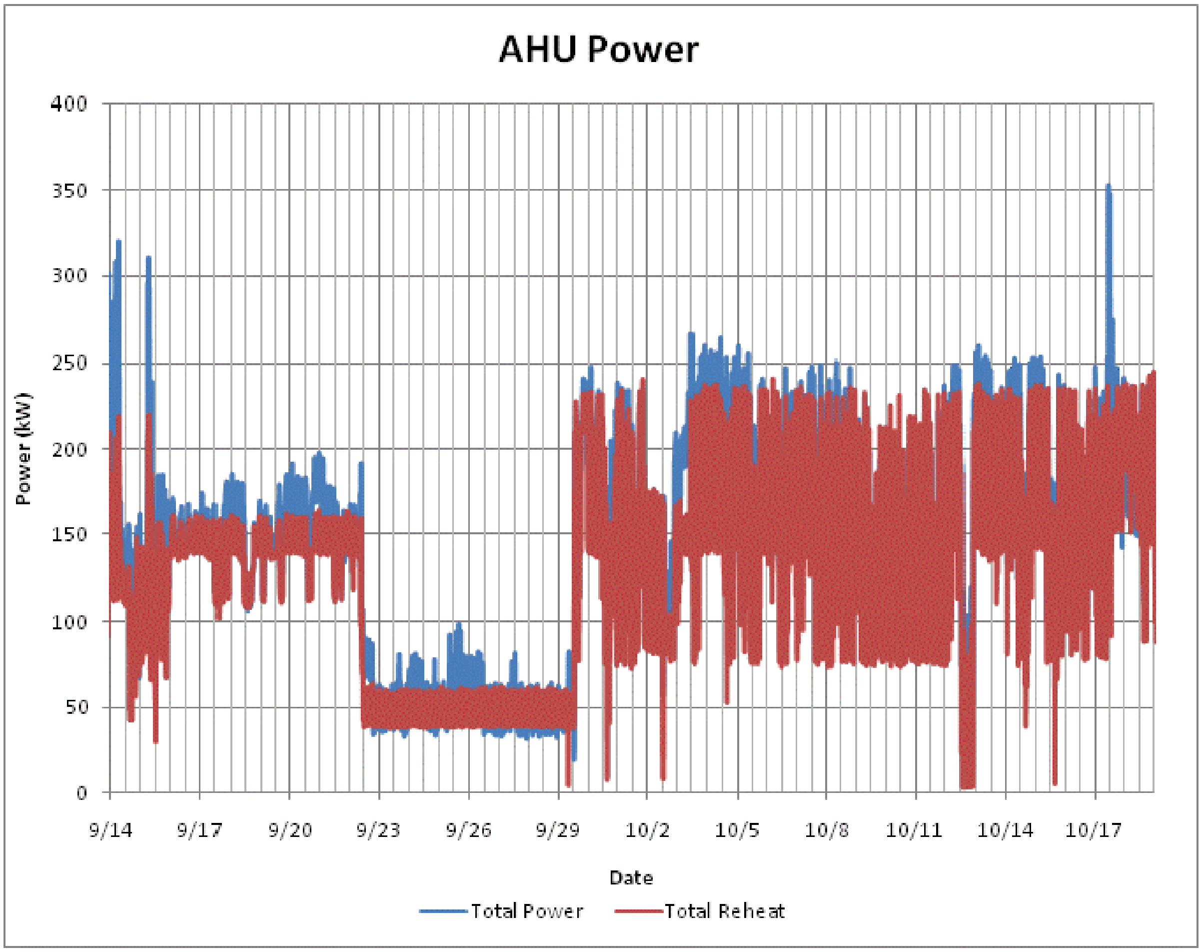

Figure 4.24: AHU Power, 9/22-9/29

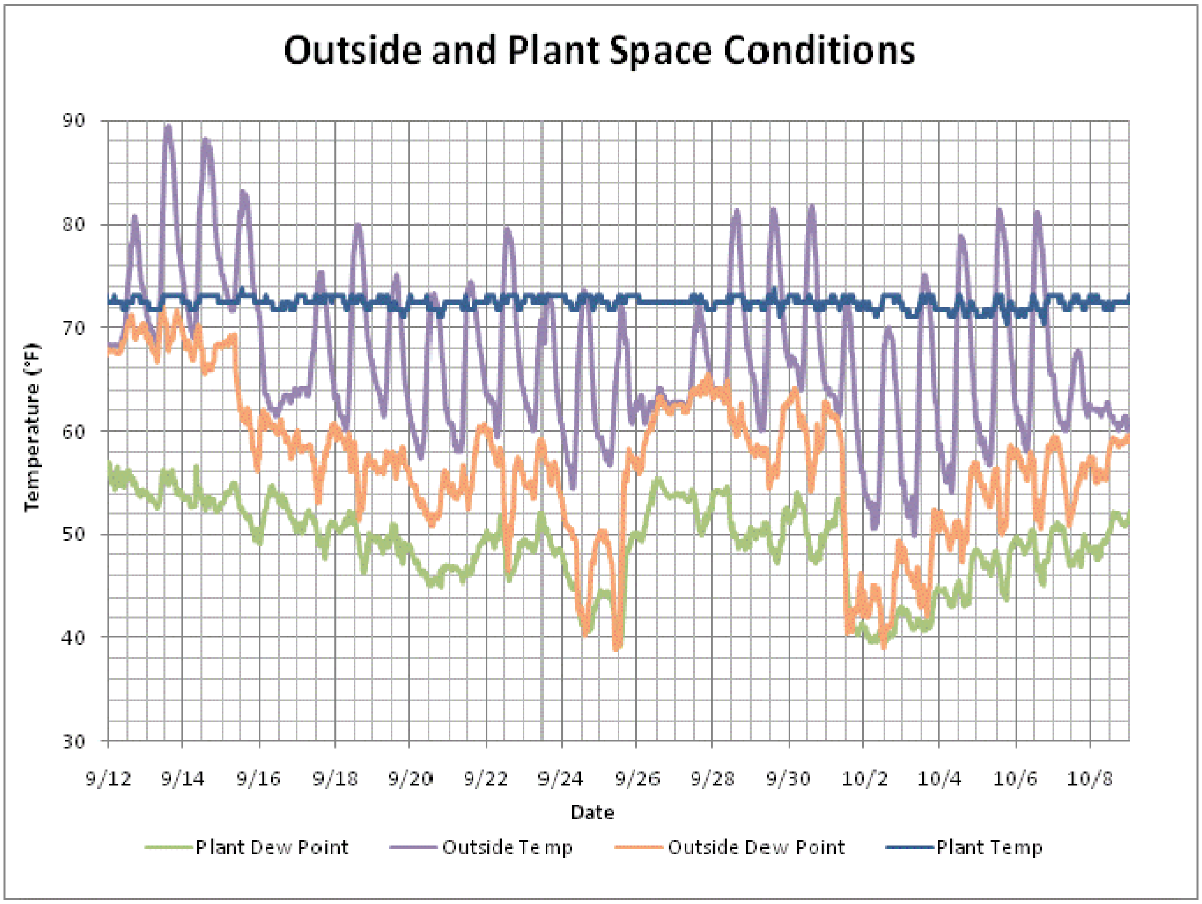

Figure 4.25: Outside and Plant Space Conditions, 9/22-9/29

The outside conditions during this timeframe do not cause this drastic change.

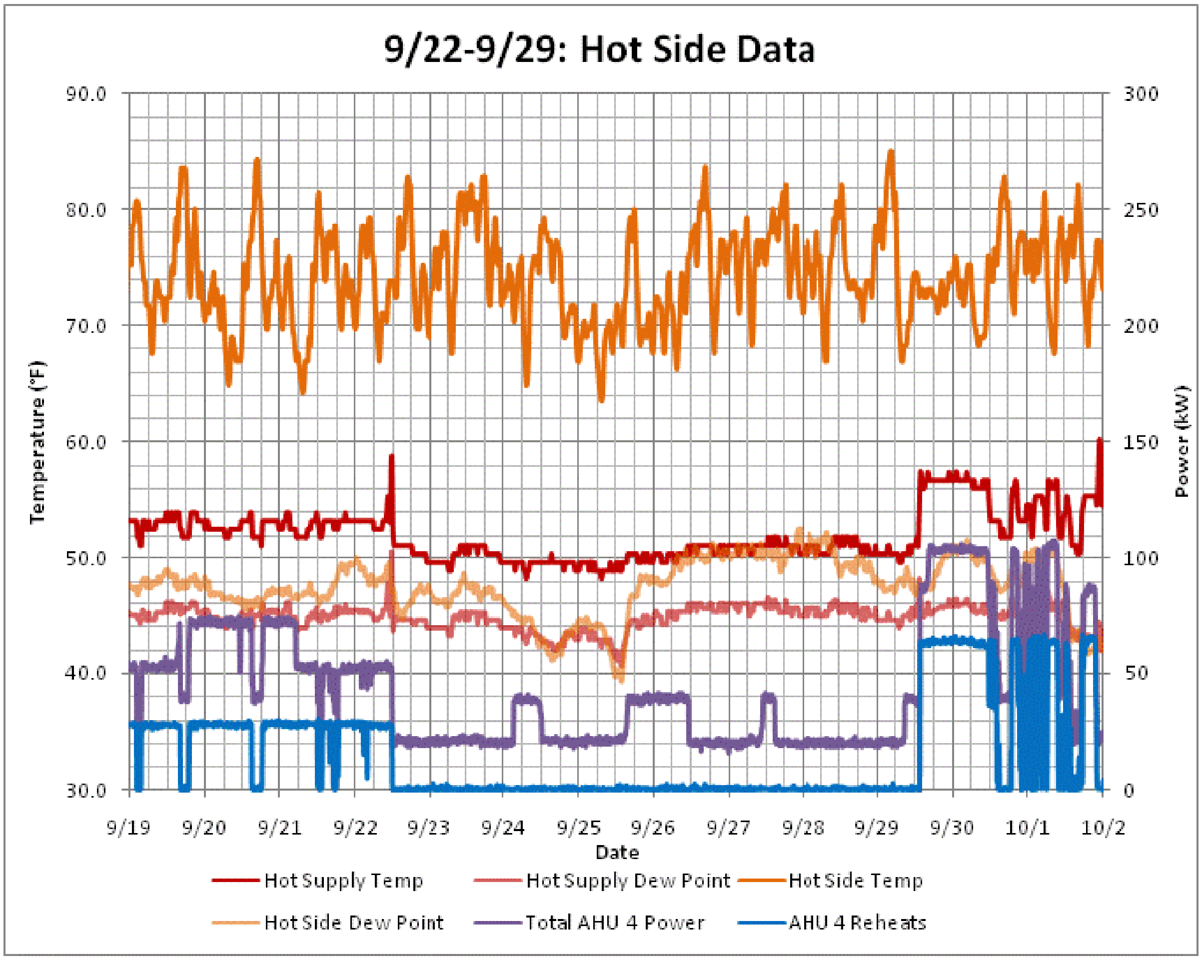

Figure 4.26: 9/22-9/29: Hot Side Data

One interesting side effect of this weird AHU shutdown is the ability to do another energy balance. Looking closely at the time on Figure 4.26 before 9/22, the temperature is a fairly constant 53°F, with a constant reheat of 28 kW. Between 9/22 and 9/29, the temperature drops from 53°F to ~50°F, with no reheat. Then, between 9/29 to 9/30, reheat increases to 64 kW, and the temperature increases to 56°F.

$$28\ kW \times \frac{3{,}413\ BTU}{kW\ hr} = \dot{m}_{AHU\ 4} \times \frac{0.24\ BTU}{°\mathrm{F}\ lb}(53°\mathrm{F} - 50°\mathrm{F}) \tag{4.4}$$

$$\dot{m}_{AHU\ 4} = \left(\frac{95{,}564\ {}^{BTU}/_{hr}}{0.72\ {}^{BTU}/_{lb}}\right) \Big/ \left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right)$$

$$\dot{m}_{AHU\ 4} = {}^{29{,}500\ ft^3}/_{min}$$

While this number may seem to be lower than 45,000 cfm, keep in mind that it flow rates during this period may also have been disrupted.

$$64\ kW \times \frac{3{,}413\ BTU}{kW\ hr} = \dot{m}_{AHU\ 4} \times \frac{0.24\ BTU}{°\mathrm{F}\ lb}(56°\mathrm{F} - 50°\mathrm{F}) \tag{4.5}$$

$$\dot{m}_{AHU\ 4} = \left(\frac{218{,}432\ {}^{BTU}/_{hr}}{1.44\ {}^{BTU}/_{lb}}\right) \Big/ \left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right)$$

$$\dot{m}_{AHU\ 4} = {}^{33{,}710\ ft^3}/_{min}$$

This number is very similar, again, to what is expected. Reinforcing this concept also undermines the thought that the spikes in the total energy graph is caused by a fan load.

4.3.2 AHU 2 Humidifier

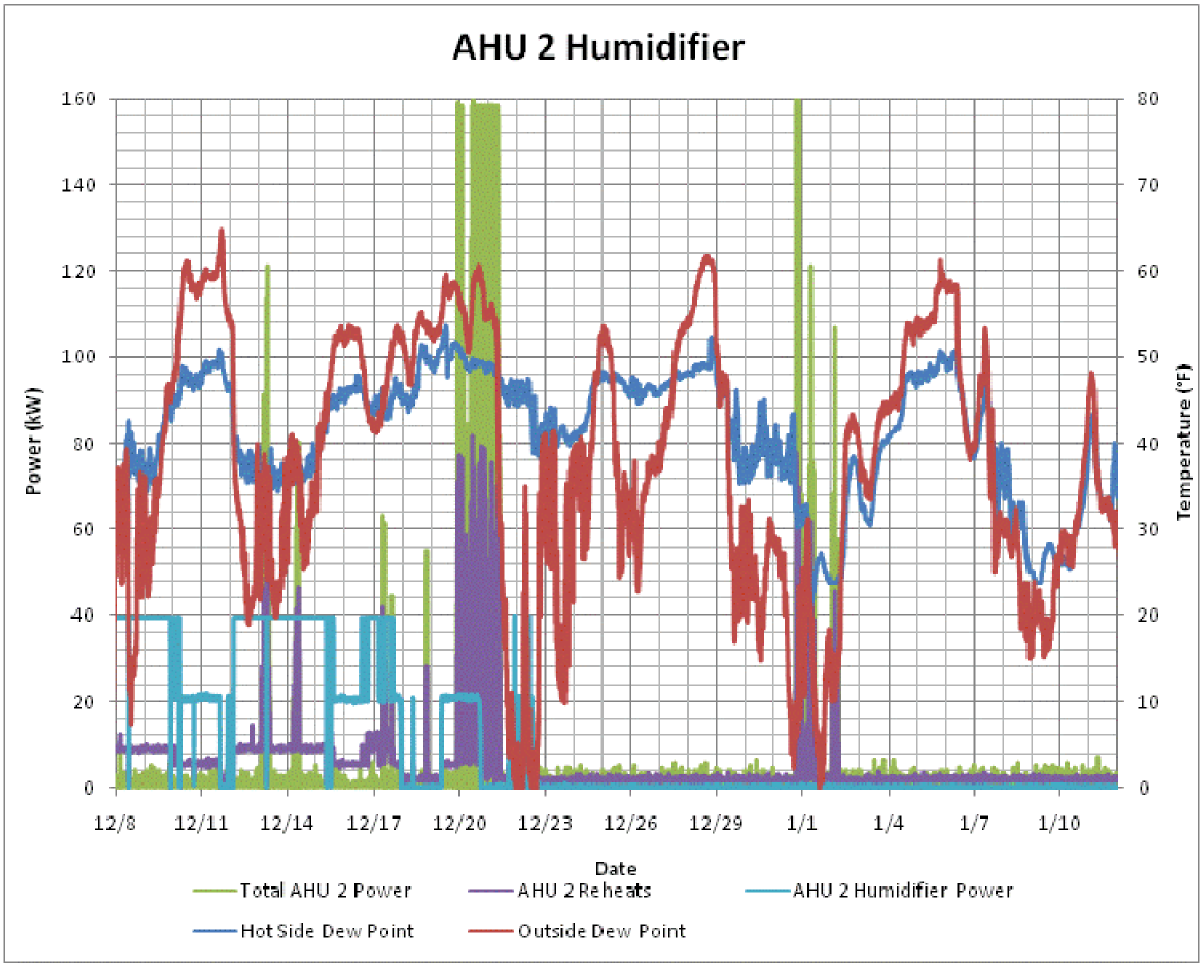

Figure 4.27: AHU 2 Humidifier

Notice in Figure 4.27 above the relationship between the humidifier power and a low amount of reheat. There is always one leg (and exactly one leg) of reheat on whenever the humidifier is on. This one leg of reheat also steps up in power just as the humidification steps up in power. This relationship holds through the new operation as well (see Figure 6.7). This is cannot be accidental, the relationship is exact almost down to the data point. Why is there a power draw on one leg of reheat when there is a power draw on the humidifier, and only for this AHU? The reheat relationship only shows up in one leg of the reheat, and only one leg of humidification is recorded. The humidification power, as listed above, is shown as if each of the three legs of power is drawn at the same amount. But it is very possible that this is not the case. This is why the total AHU 2 power can be zero while the reheat and humidifier show appreciable power draw. Also, the humidifier is apparently drawing power, but this does not seem to have too much of a humidification affect on the cleanroom (and is also puzzling).

Also of interest is the simple observation that total power draw is zero over nearly the whole time period, and really since data recording began in June. The data logger and CT were swapped out multiple times, to ensure this is not merely equipment failure, but the results always have the same character. Since it is thought that the fan has to be running to keep 45,000 cfm of air moving through the filters, where is the fan drawing its' power on AHU 2? Since all of the AHU's are identical in make, size, and age, this also begs the following question: If AHU 2's 100 hp fan receives power from a different box, do the others as well?

4.4 Fan Results

Up until this point in time it has been assumed that the small ~5 kW in AHU 1, 3 and 4 was due to the fan load. This may not, in fact, be the case. The air handling units each have 100 hp fans (as we read from the nameplate) running on Variable Frequency Drives (VFD's). At full load, these fans would draw 75 kW. The reheat on each unit totals a rated 70 kW, and the humidifiers are rated at 60 kW on each unit. In the electrical box for each AHU, there is a main three phase power, where each phase is split into three components. The largest three wires head to the fan, and the other wires go to the reheat and the humidification. These wires were traced to ensure accuracy of data logging.

Clearly, 5 kW is a small fraction of the maximum power used by the fans. This is such a low load that if accurate, the fans are probably not running on an efficient area of the fan curve. What is known is the relationship between fan speed, pressure and power. The flow rate through the AHU is directly proportional to fan speed (N). The pressure rise through the fan is a function of the speed squared. Finally, the power used by the fan is a function of the speed cubed.

$$\frac{Flow\ Rate_1}{Flow\ Rate_2} = \frac{N_1}{N_2} \tag{4.6}$$

$$\frac{Pressure_1}{Pressure_2} = \left(\frac{N_1}{N_2}\right)^2 \tag{4.7}$$

$$\frac{Power_1}{Power_2} = \left(\frac{N_1}{N_2}\right)^3 \tag{4.8}$$

If the power being utilized is $1/15^{th}$ (6%) the motor power, the fan speed (and therefore flow rate) will be 41% of the maximum flow rate, and the pressure rise will be 16% of the maximum pressure rise.

So what does all this mean? This means that if our readings are accurate (and correctly interpreted), the fans are grossly oversized for their current usage, and the design pressure rise is much larger than currently experienced. It also means that AHU 2 rarely runs a fan (as stated in Section 4.3.2), since it almost never has a total load.

Is there a case for the fans being oversized? Is there a reason it could have been designed that way? It is possible. The room may have been designed for larger flow rates for a higher cleanroom classification. After all, the chillers are oversized as well. Also, it may have been designed for a higher pressure drop across the HEPA filters, or a sizable cleanroom pressurization level. This last reason seems most plausible.

Is there a case for the readings not being accurate (or correctly interpreted)? Unfortunately, there is. Undersized fans and pumps risk running in a very unfavorable area of their performance curves, causing some problems.   In addition, it is unusual for VFD equipped motors to receive power from a split power line. This is to protect the VFD against power oscillations from the other equipment (the VFD is expensive) and to protect the other equipment from the harmonics from the VFD. Thus, it is possible that the 5 kW recorded as being the fan load is really some small control or humidifier load, and that the fan power is therefore supplied by another set of cables not measured.

4.5 Chiller Results

In the graphs that follow, the chiller operation is detailed with respect to many parameters.

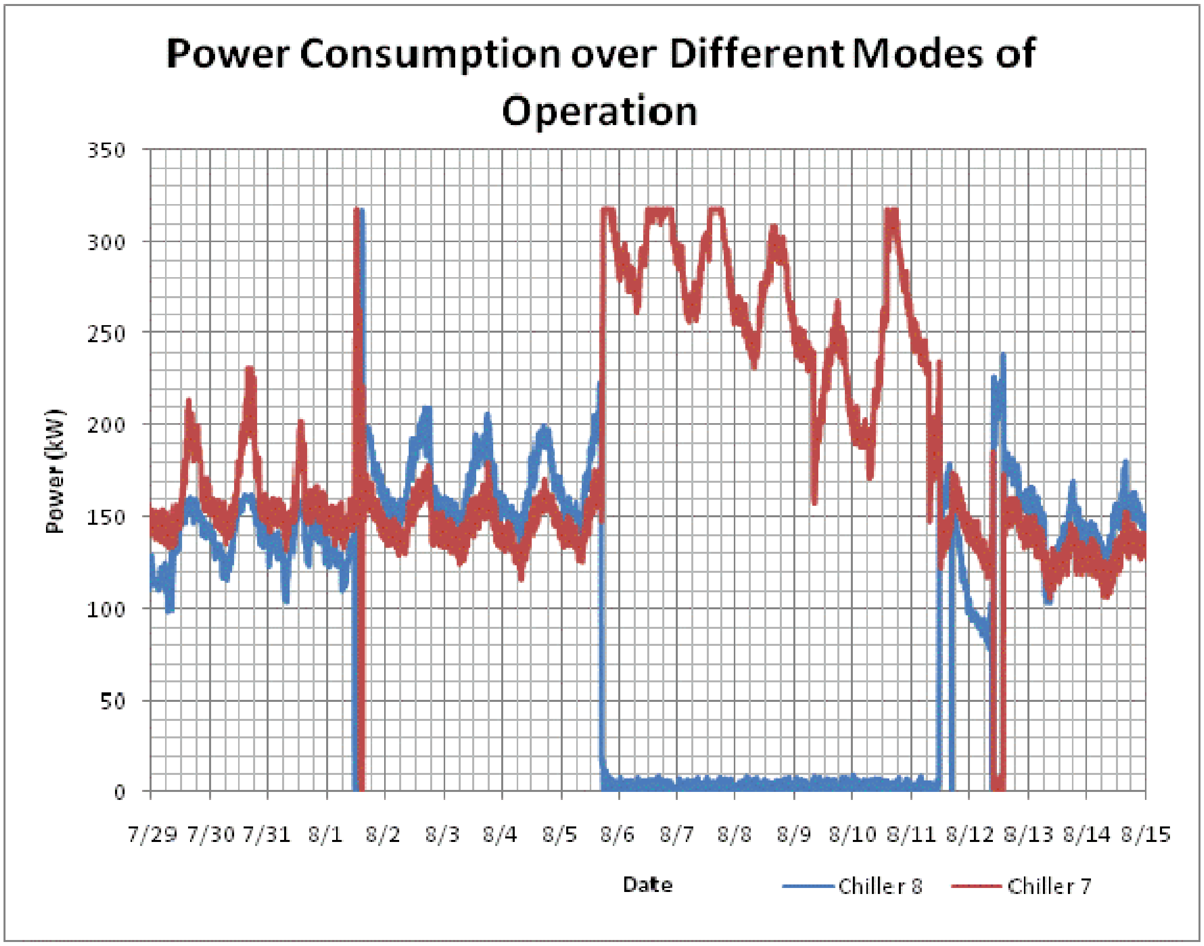

Figure 4.28: Chiller Power Consumption over Different Modes of Operation

The Figure 4.28 above shows one of the most interesting findings of this whole data collection process. This graph shows three different operating conditions of the cleanroom chillers (explained in Section 1.3), designated chillers 7 and 8. From 8/6 to midday 8/11, chiller 8 does not operate at all. From midday 8/1 to midday 8/5 and from roughly 8/12 to

8/15, the chillers have all 6 compressors running (all four circuits), although not completely loaded. The beginning of the graph (and the small period around 8/12) shows only 5 compressors operating (three circuits). Here, the 100 ton compressor on Chiller 8 is not operating.

Initially, it is important to remember that temperature, production, and cleanroom conditions all affect chiller power consumption. But for the moment, consider the changes taking place on the graph as if all of the other conditions are equal. Daily chiller power seems to ramp up very well, but then does NOT seem to ramp down very well at night. It is remarkable how the minimum power (except from 8/6 to 8/11) seems to find the same value night after night, even as the daily maximum varies wildly. Possible reasons for this include the following: temperature, humidity, production, cleanroom set points, and even chiller compressor limits. Considering the period where all 6 compressors are on between 8/1 and 8/5, the average chiller power use is 314 kW. The average for the period before this (7/29-8/1) is 293 kW, resulting in a net gain of 21 kW. Consider the period between 8/6 and 8/11. Here, the average use is 266 kW, a 48 kW decrease over the previous period, and the average power use for time after 8/12 is about 288 kW.

On the week of 8/5-8/11 (figure 4.29), the chiller temperatures jump substantially (~5°F). It is debatable as to whether or not one chiller can handle the load over this summer time period. On 12/18, in Figures 4.8 and 4.23, the chilled water temperature is again seen to take a jump when one chiller goes completely offline. However, in late December, whether or not one chiller can handle the load is not an issue. Clearly, one chiller can handle the load with ease. So the question becomes even more mysterious: Why does the chiller temperature suddenly hike?

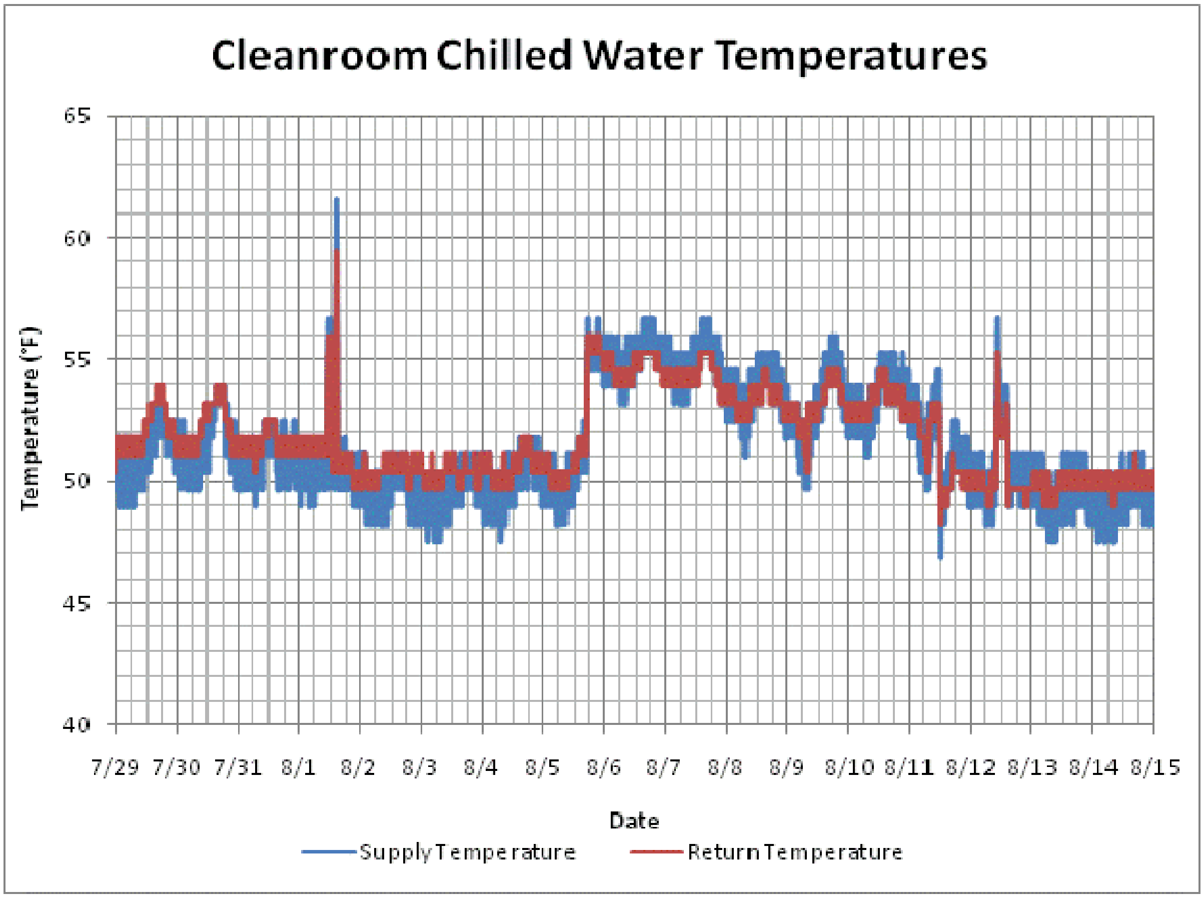

Figure 4.29: Cleanroom Chilled Water Temperatures, August

The supposed deficiencies of the cleanroom chilled water temperature measurements have been well documented in the previous chapter, but it is worth noting that a spike in chilled water temperatures is seen on days corresponding to the days which chiller 8 was offline. This really does not make much sense from a cause-effect point of view. Chiller 7 is operating nowhere near capacity from 8/8 through 8/12. The chillers are set up in parallel, not in series. This may be indicative of some kind of "economizer mode", an independent experiment run by plant personnel, or the programmed response to one chiller going offline.

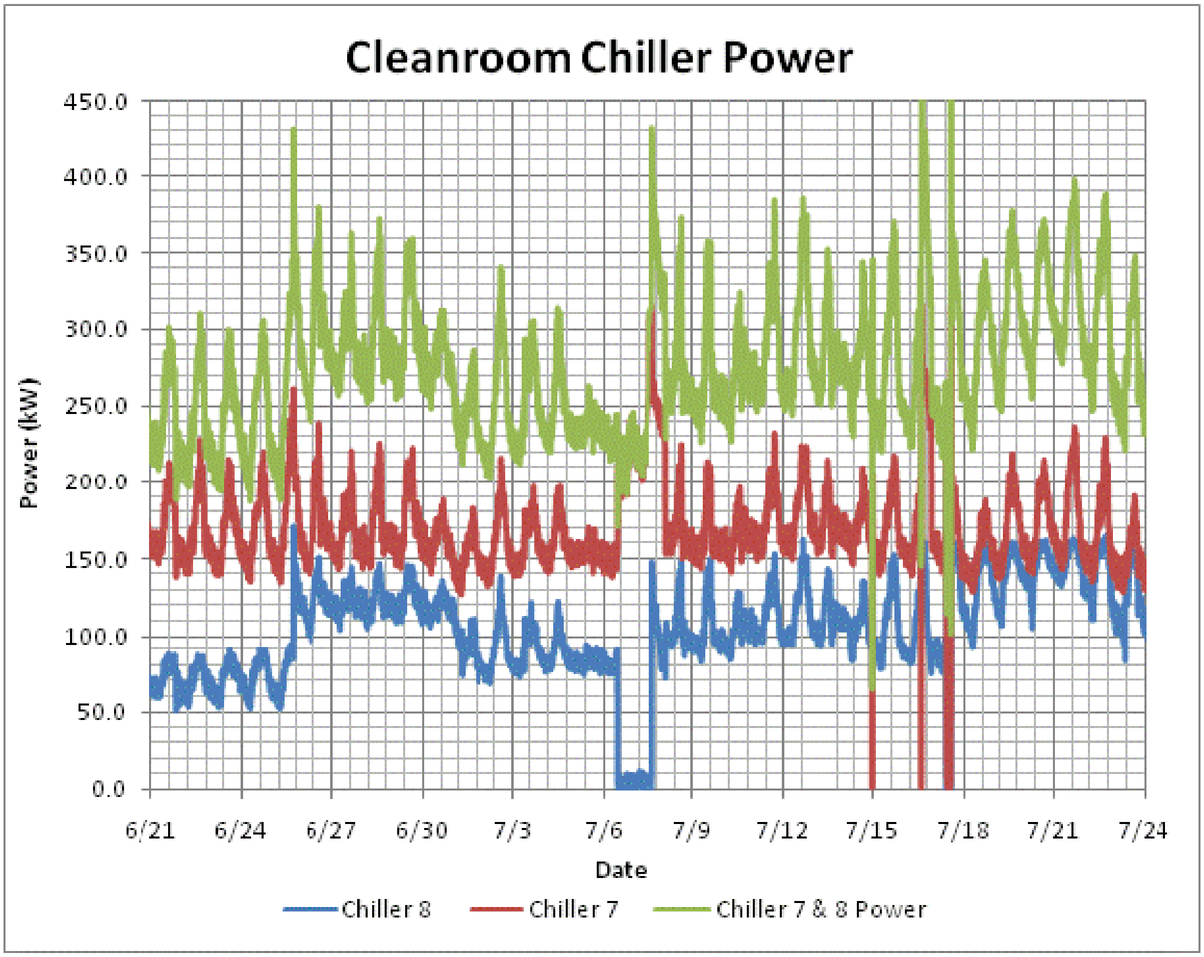

Figure 4.30: Cleanroom Chiller Power, July

Figure 4.30 is another graph illustrating the same principle. For chiller 8, notice the sudden increase in power usage on 6/25, and then the drop in power usage on 7/1. Up until 6/25, Chiller 8 is operating with only one circuit on, the twin 70 ton compressors. After 7/1, Chiller 8 is only operating with the other circuit on, the single 100 ton compressor. The difference in overall power draw is noticeable. The following table gives the values of the power averaged between certain points in time. Between 6/19 and 6/25 there was a weekend, just as between 6/25 to 7/1. The difference, as seen in Table 4.1, comes entirely from chiller 8, and is about 50 kW.

Table 4.1: Chiller Power Totals

Timeframe of Average	# of Averaged Data Points	Chiller 8 (kW)	Chiller 7 (kW)	Total (kW)
6-19 to 6-25	874	71.1	169.1	240.3
6-25 to 7-1	808	121.6	168.7	290.3
7-1 to 7-6	743	87.9	158.2	246.2
7-6 pm to 7-7 pm	149	3.2	216.9	220.1
7-7 pm	21	2.3	307.5	309.7
7-7 pm to 7-8 am	66	102.2	247.8	350.0
7-8 to end	2,506	120.4	166.5	286.9
Total/ Average	5,167	103.5	169.2	272.6

However, as noted by the last week in Figure 4.30 above, the single compressor on Chiller 8 really loads well towards the end of July. This is interesting because Chiller 7 seems to be unaffected by how loaded Chiller 8 is, especially with respect to minimum load. At 240 tons and an assumed 1.25 kW/ton (from Section 1.3), this gives an estimated max load of 300 kW. Looking at Chiller 7 alone, it seems very reluctant to go very far under 150 kW at night, bottoming out at about 130 kW, or 45% of maximum power. Given that it is known that rotary screw compressors operate very inefficiently when unloaded, it is very possible that the compressors are not able to use less power than 45% of max load while on.

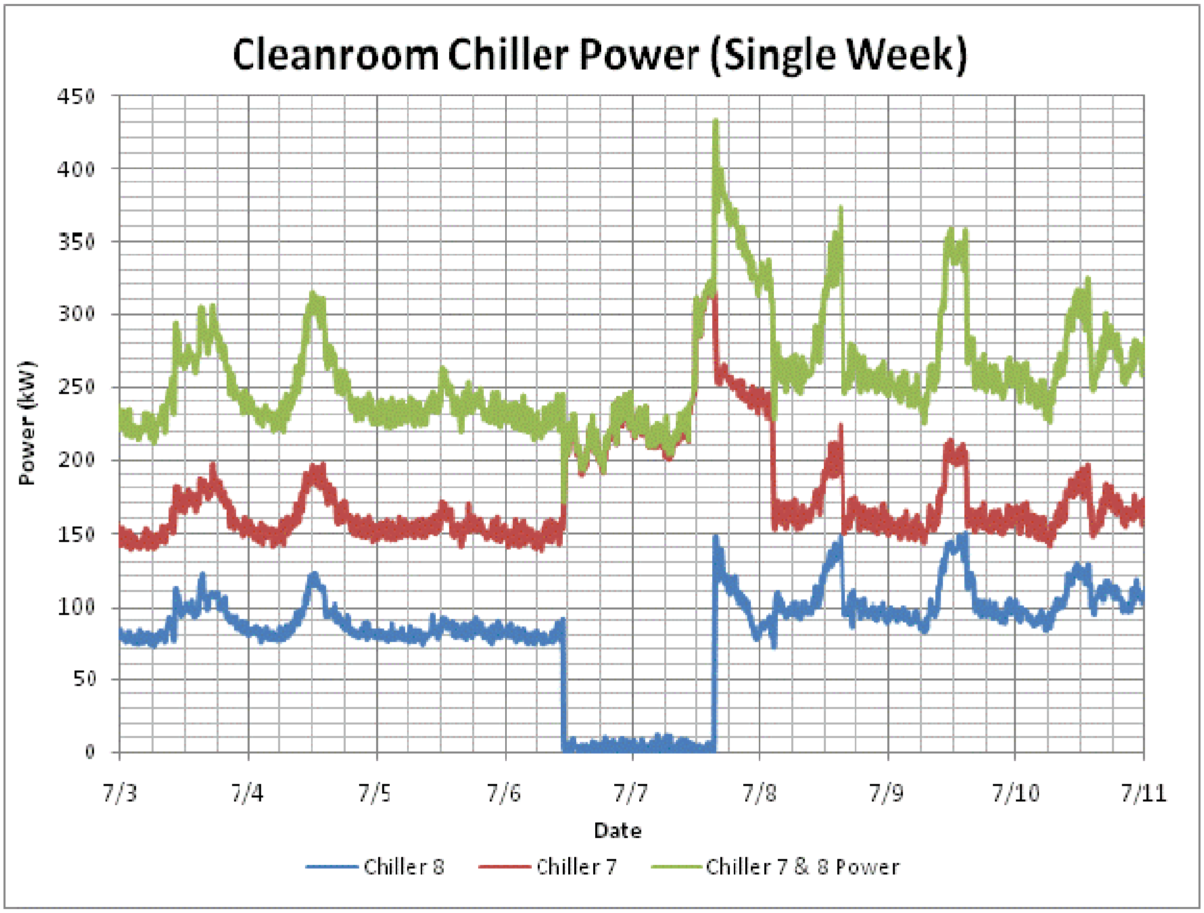

Figure 4.31: Cleanroom Chiller Power (Single Week)

This single week chart is provided to illustrate the strange chiller behavior on the afternoon of July 7th. For all of the data before and after July, sustained chiller power spikes on either chiller are a function of any one of the four chiller circuits coming on or off, and also typically tied to the actions of the other chiller. For example, on July 6th, the 100 ton compressor on Chiller 8 turns off, and so Chiller 7 increases its power usage, but on both circuits. However, on July 7th, the power draw on the 2 compressor circuit jumps suddenly on Chiller 7, with no increase in power draw on the 1 compressor circuit. Several hours later the 100 ton compressor on Chiller 8 turns on, and the 2 compressor circuit drops suddenly while the power draw on the 1 compressor circuit stays constant. This is the only data which suggests that the 2 parallel compressors have the ability to switch on and off independently.

However, this possibility is tempered by the next jump in data, in the early hours on July 8th, in which the single compressor circuit on Chiller 7 suddenly drops 40 kW, with a small reaction from Chiller 8. Obviously, the single compressor cannot shut halfway off, so this change was probably done with controls, since it is unlikely that the chiller is being tweaked at 2:30 am.

So what was happening with respect to outside operating conditions?

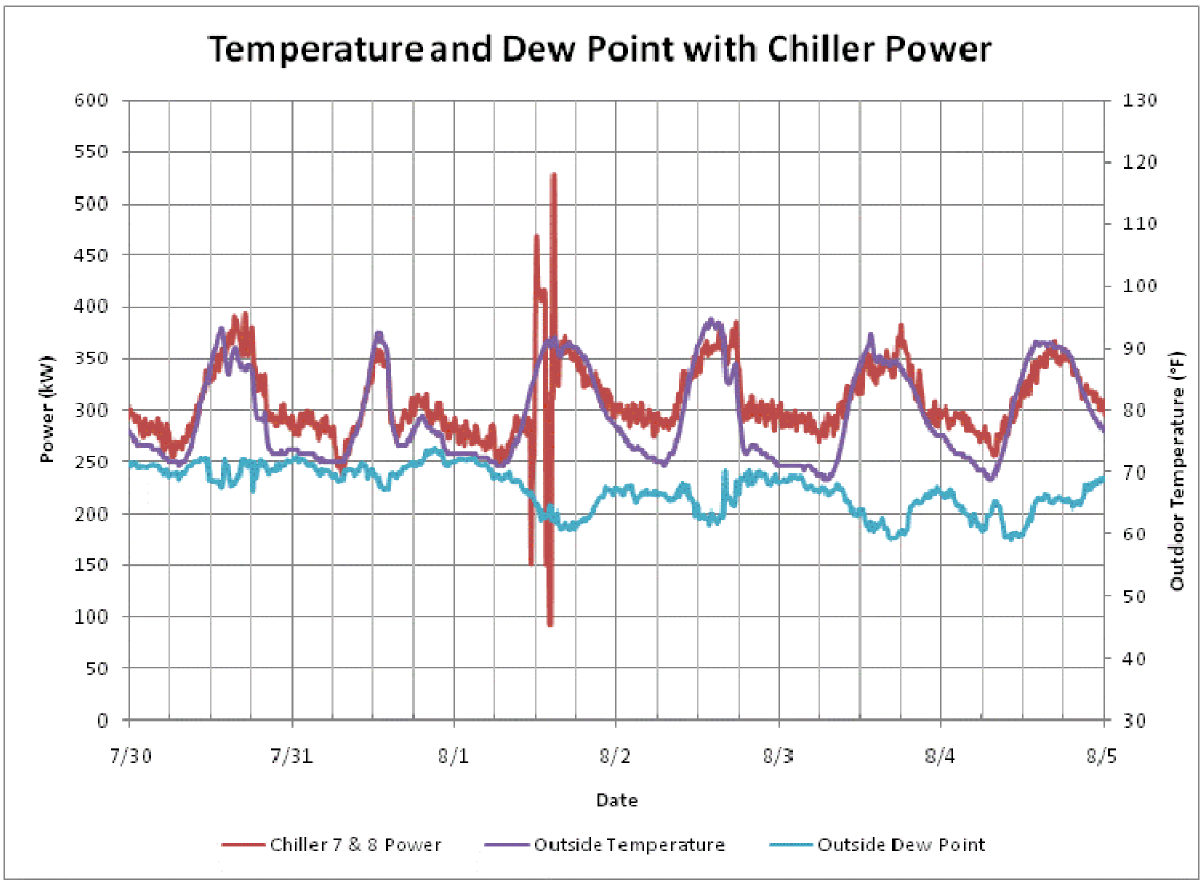

Figure 4.32: Temperature and Dew Point with Chiller Power (One Week Graph)

As shown in Figure 4.32 above, outdoor temperature and power correlate very well. Obviously, the scale has been changed to enhance the visual effect. Do note how the nighttime power, especially after 8/2, tends to not unload as well as might be expected. Contrast this with the expanded view provided in Figure 4.33 below. It is obvious that total power use is below that which would be expected, given the performance data for the week previous. Yes, the humidity is down overall, but the humidity is comparable in the early hours of 8/10 to the early hours of 8/4, and power use is still way down.

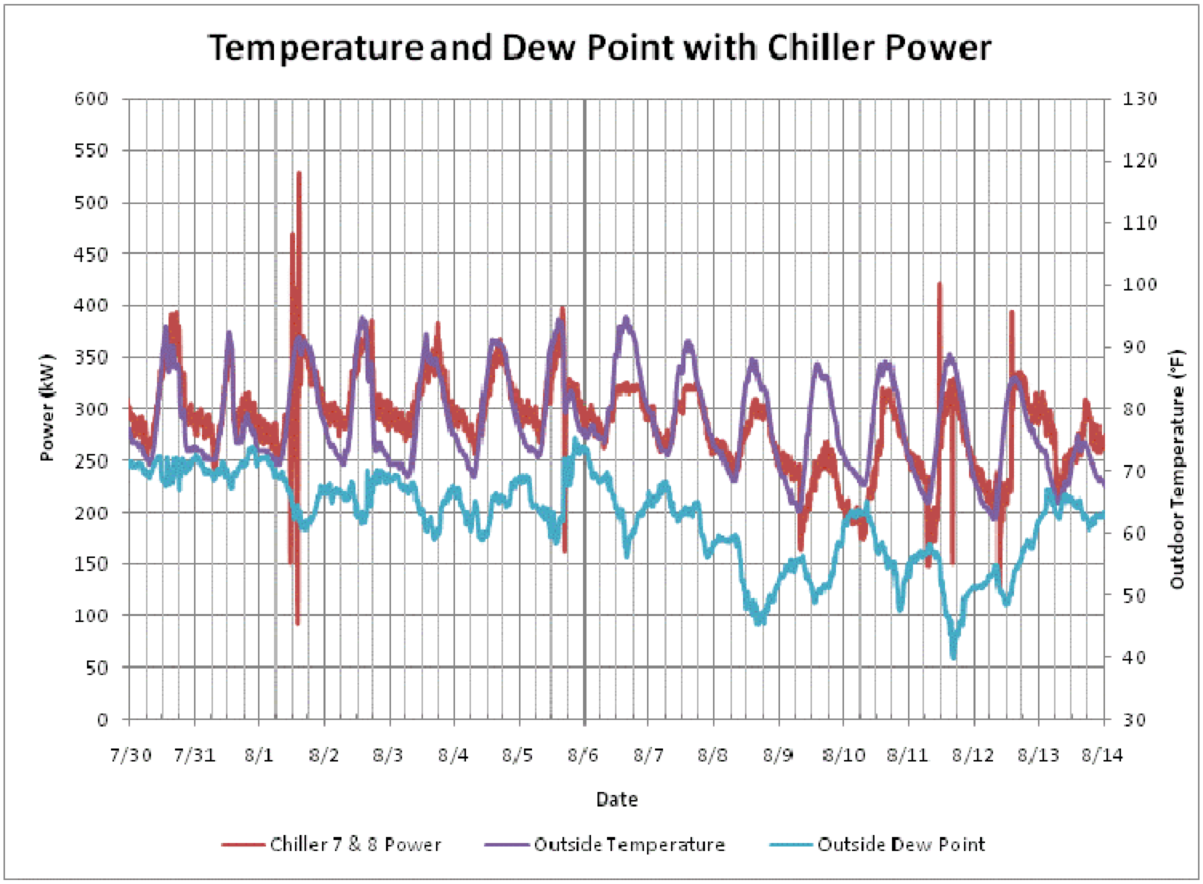

Figure 4.33: Temperature and Dew Point with Chiller Power (Two Week Graph)

It seems as though the chillers do not follow the trend on August 6th and 7th, and essentially has the daily spike removed. Remember that those days the data recorders max out at 158.4 kW on both circuits on chiller 7, giving 316.8 kW. Since the data stays above this for several hours, it is probably safe to assume that it is not at maximum capacity. At 316.8 kW and 1.25 kW/ ton, the chiller would be providing 253 tons of cooling, more than the 240 nominal tonnage.

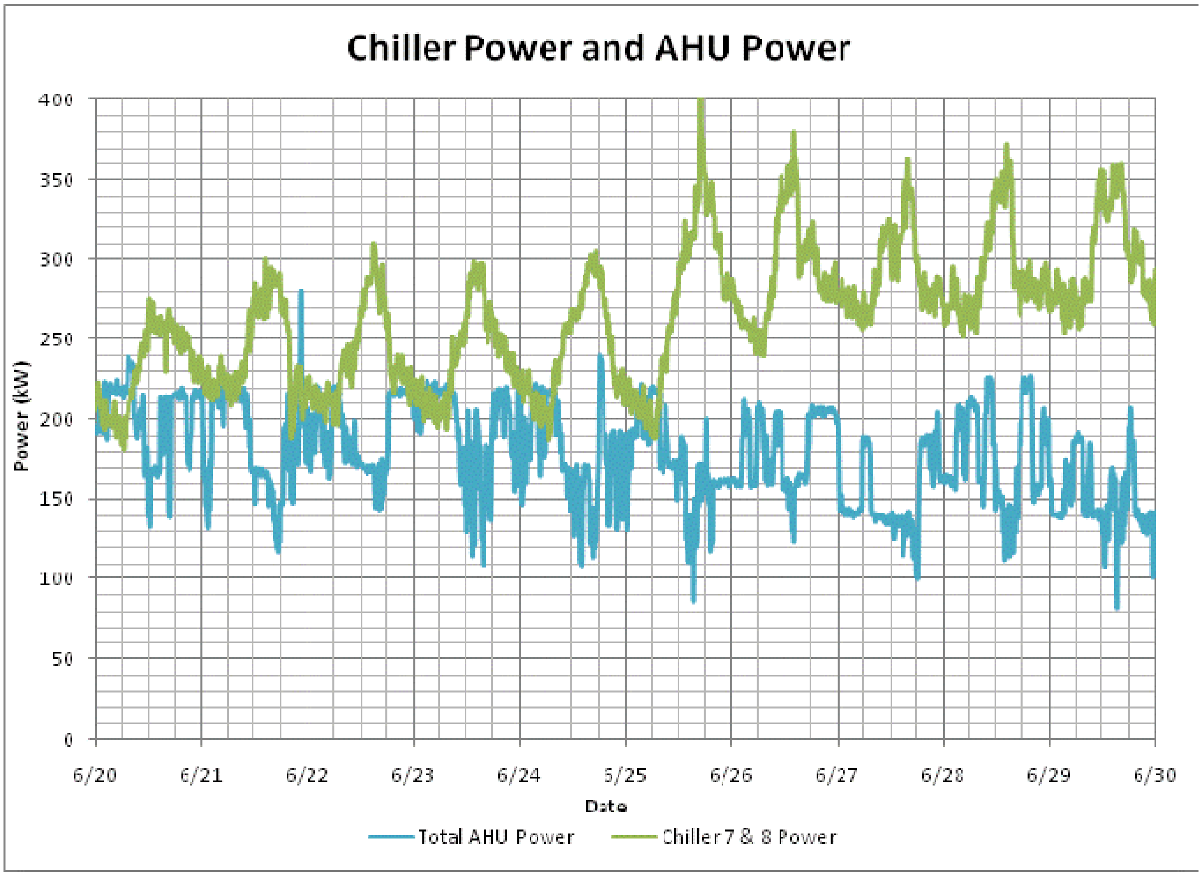

Figure 4.34: Chiller Power and AHU Power

In Section 4.2, it was mentioned that decreased production could be the cause for slightly higher reheat usage at night. If this is the case, then could it be possible that the daily increase in chiller capacity is related to the cleanroom production? On 6/22, for example, the daily change in chiller power is 100 kW. The daily change in reheat over that period is 70 kW. However, the cooling effect the chiller is having is greater.

$$\frac{100\ kW}{1.25\ kW/_{ton}} \times 12{,}000\ BTU/_{ton} \times \frac{1\ kW}{3{,}413\ BTU} = 281\ kW \qquad (4.9)$$

This is much greater than 70 kW. Also, the cleanroom production would not vary with outdoor temperature, as the chiller power clearly does very closely. Also, not all days have a large swing, and on some days, the swing is nearly random. The averaged on peak increase in chiller power over that period is 52 kW, while the average increase in on peak reheat is 14 kW. That is only 9.5% of the energy, so there exists the possibility that this relationship may be lost in the data, so to speak.

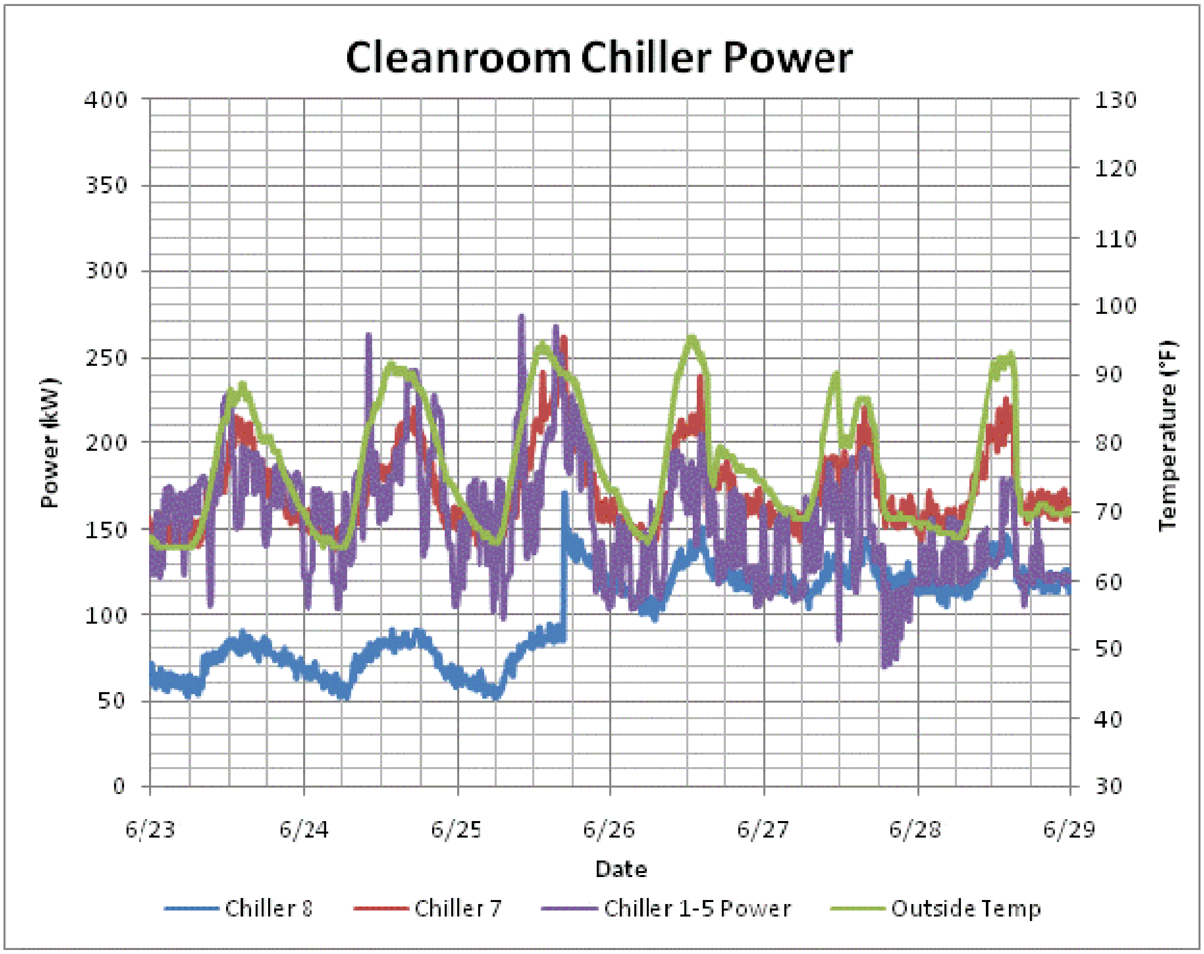

Figure 4.35: Cleanroom Chiller Power (6/23-6/29)

The chiller power of units 1-5 are almost completely production based. There are large users of chilled water which may demand chilled water at anytime, which helps explain the variability. These users are almost entirely production based. What it does not explain is the dependence of the chiller power use on outside temperature. The plant is air conditioned by rooftop units, so any daytime load should act *independent* of outside temperature, and rather as a function of increased daytime work activity. But as the graph shows, it clearly appears to be a function of outside temperature.

Why could this be the case? It could be the case that there are several small A/C systems which actually do draw from the chilled water loop. However, to cause 50-100 kW changes in demand, these must be 40-80 tons of A/C capacity. This does not seem too likely. The other possibility is that the increase in daytime drybulb temperature is lowering the condenser heat transfer so much that the chiller is forced to raise the refrigerant temperature (and thus the pressure), which lowers the efficiency of the units. This is much more likely. Remember that all of the chillers at the facility are package air-cooled units, where space is at a premium. Simple thermodynamics dictates that a good condenser is a big condenser, (i.e., more surface area over which heat transfer can occur). There are only 17 fans on each 240 ton chiller, each of which draw 1.5 hp (1.2 kW), which equates to one fan for every 14 tons of chiller capacity.

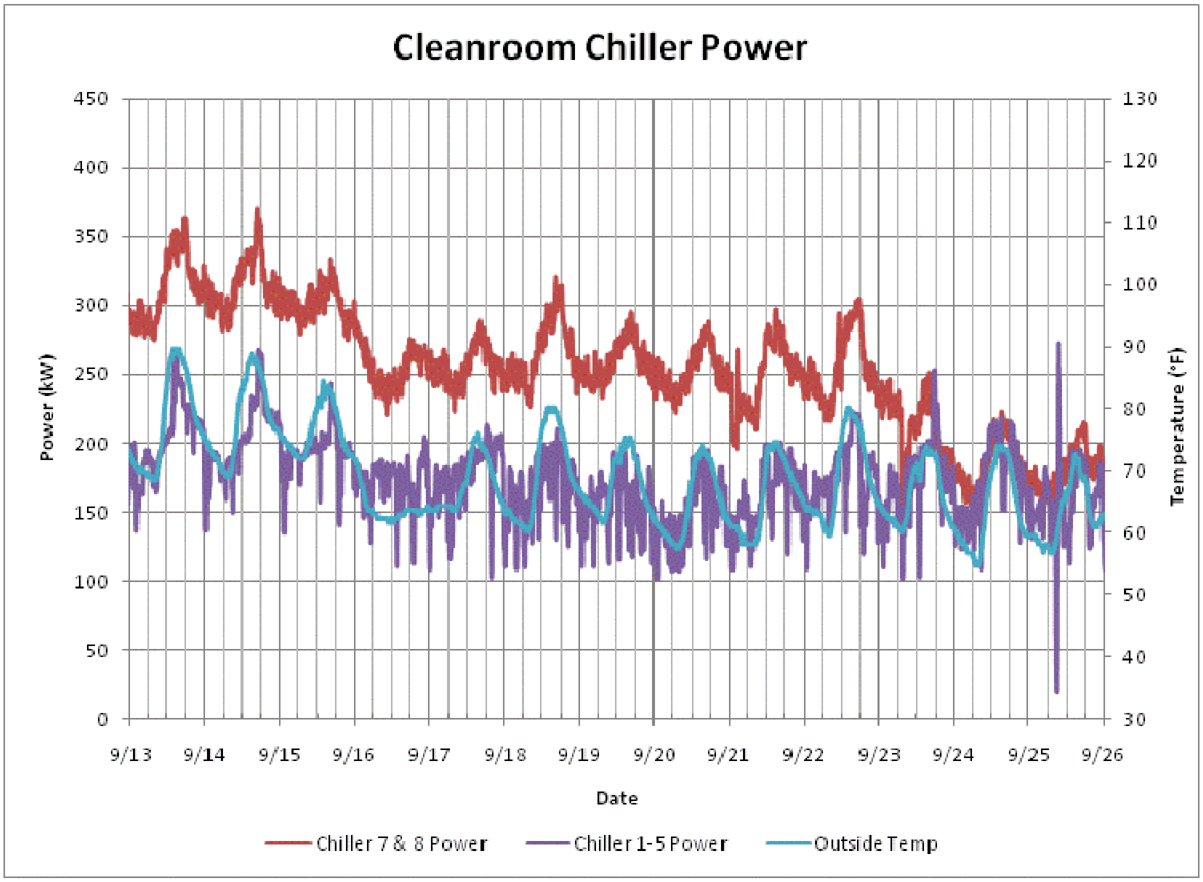

Figure 4.36: Cleanroom Chiller Power (9/13-9/26)

In the above graph, 9/26 is the last day that the data recorder on chiller 8 recorded, before it went offline. However, 9/22 was included in the graph above on purpose. Chiller 1-5 has the same operation before and after 9/22, further pointing to the happenings that week being isolated to the cleanroom. Again, the relationship between outside temperature and chiller 1-5 power is clearly shown *even when the outside temperature would rule out air conditioning.* This is proving to be an argument for the adoption of a water cooled chiller system, since not only is the plant losing here for overall increase in power usage, it is also losing in increased demand.

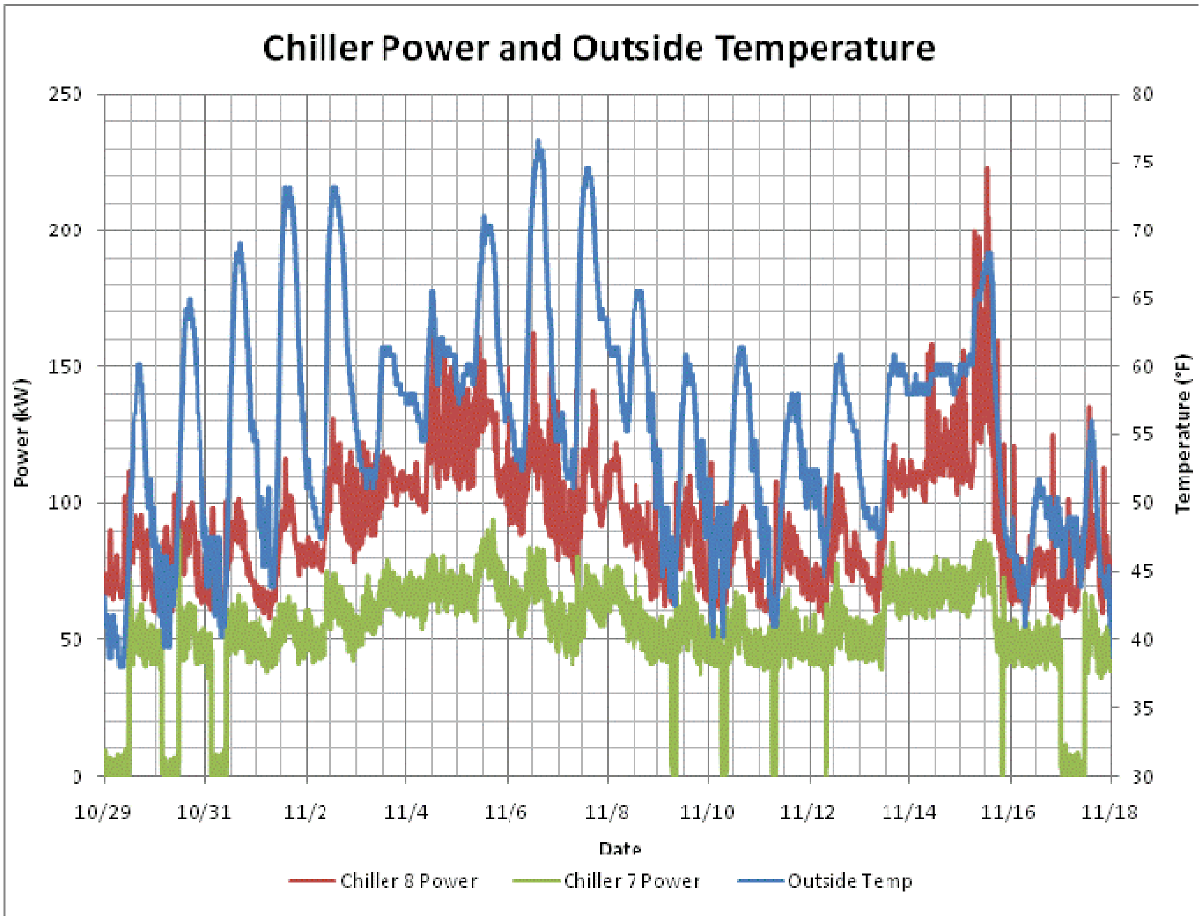

Figure 4.37: Chiller Power and Outside Temperature, November

As Figure 4.37 demonstrates, the chiller power use is still highly dependent on the outside temperature up until Thanksgiving, when production becomes more sporadic for the winter.

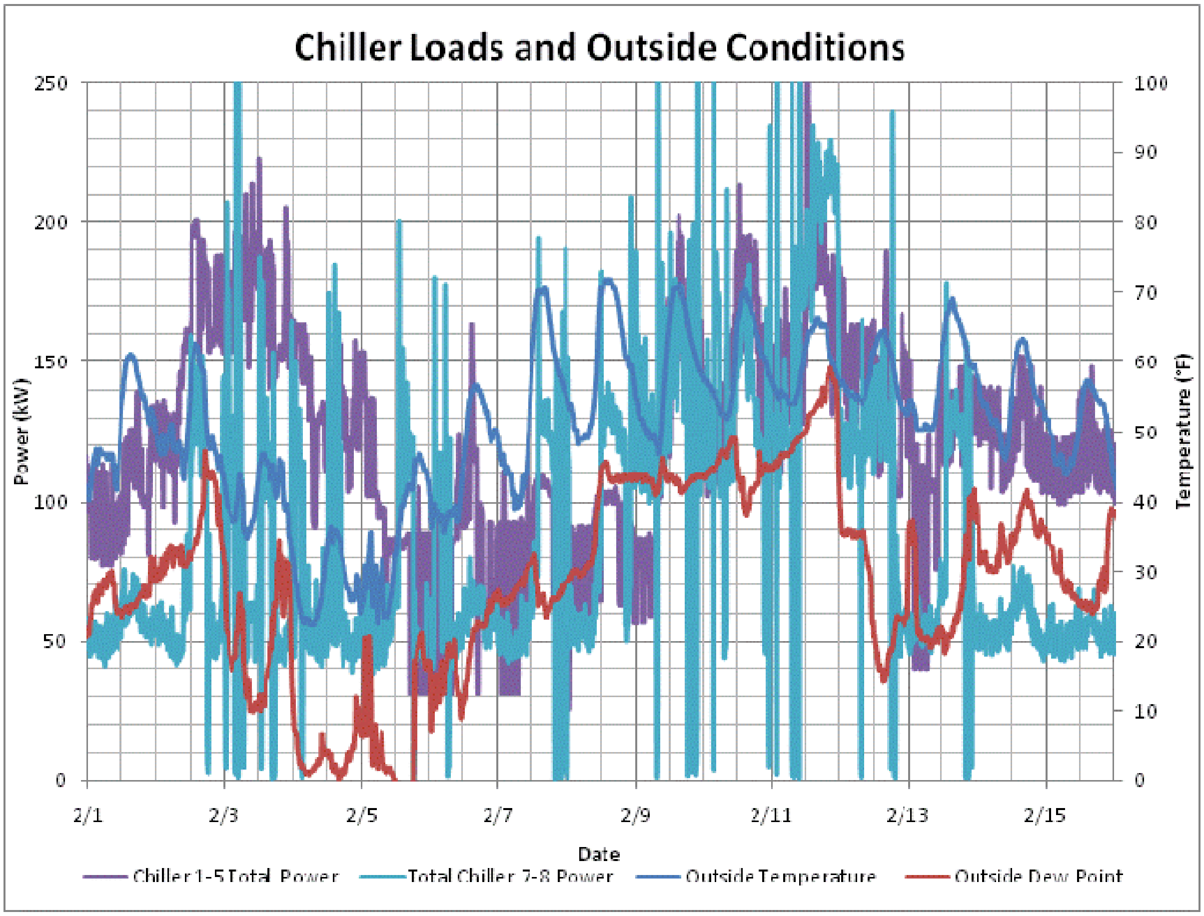

Figure 4.38: Chiller Loads and Outside Conditions

In the graph above, it is clear that winter chiller load fluctuates often. Chillers 1-5 show how the power fluctuates with plant load and with outside temperature. The peak of chillers 1-5 always occurs in when the outside temperature goes to maximum, even in the wintertime. This difference between daily maximum and minimum is typically well under 50 kW, unlike in the summertime, but is still present. This may signal that afternoon production is always slightly more demanding than all other times, or that the chillers are becoming more and less efficient as a function of outside temperature. Note how the chiller 7 and 8 power shifts in large increments. This is because lots of changes are occurring in cleanroom operation. Note how at the end of the graph, 2/14-2/16, the chiller power is so low when chillers 1-5 indicate the plant is operating.

Chapter 5: Analysis

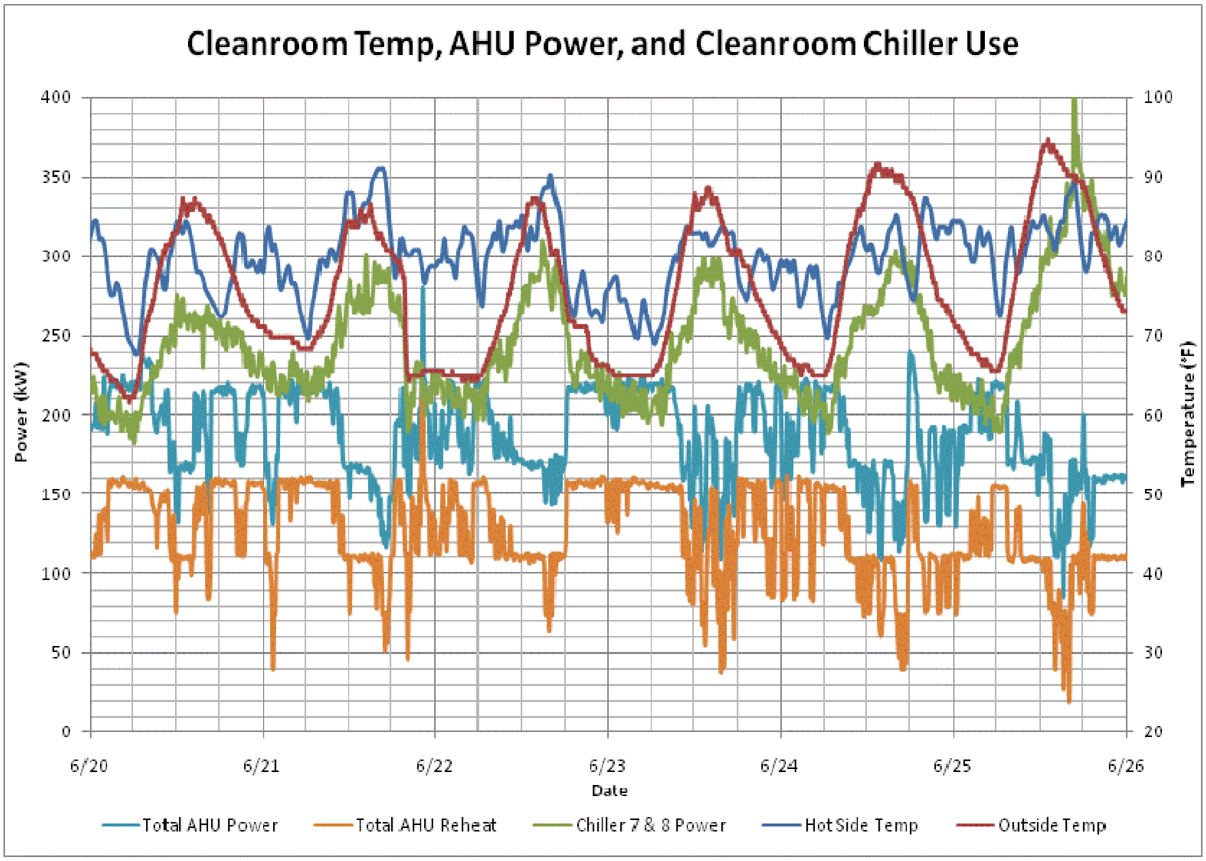

Figure 5.1: Cleanroom Temp, AHU Power, and Cleanroom Chiller Use

It was explained above that the cleanroom has about 5,000 cfm of outside air entering the AHU's. As explained in Chapter 6, this amount has been increased to a measured 15,000 cfm, in order to pressurize the cleanroom to about 0.5 inches Hg, which has now been achieved. (Although the outside air flow has been increased by threefold, it does *not* follow that the energy required to condition it also increases threefold. See Chapter 6 for explanation.) Therefore, 5,000 cfm can be seen as a reasonably accurate figure. On a typical summer day, for example, the *maximum* outside daytime enthalpy, according to ASHRAE 1% cooling data for Charlotte, NC [5], is 37.4 BTU/lb (91°F dry bulb, 74°F wet bulb). The hot side supply temperature on the day is a fairly constant 53°F, and the dew point is around

50°F. The cold side supply temperature is a balmy 70°F, and has a supposed dew point of 42°F. Since this is going to be a conservative calculation for the *maximum possible* infiltration load, it will be assumed that the air is all cooled to the cold side dew point (42°F is too low, so 48°F will be assumed), and then reheated. Thus, an enthalpy of saturated air at the cold supply dew point of 19.2 BTU/lb will be assumed.

$$\dot{Q}_{infmax} = \frac{5{,}000\ ft^3}{min} \times \frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\left(\frac{37.4\ BTU}{lb} - \frac{19.2\ BTU}{lb}\right) \tag{5.1}$$

$$\dot{Q}_{infmax} = \frac{409{,}500\ {}^{BTU}/_{hr}}{12{,}000\ {}^{BTU}/_{ton}} \times 1.25\ kW/ton$$

$$\dot{Q}_{infmax} = 42.7\ kW$$

So, there is a maximum possible daytime load from infiltration on a hot summer day of about 43 kW. From Figure 5.1, the chiller power can be seen to start at 197.3 kW in the morning, and peak at 309.9 kW in the afternoon. The daytime loading on the chillers is over 100 kW! Consider for a moment all the other sources of heat, to make sure that none of them could explain the extra load. The cleanroom is completely surrounded by the plant, which is air conditioned. In fact, the plant peaks at 73°F that day, and the dew point for the plant is within a degree of the dew point in the cleanroom. The other sources of heat are process related. Process is supposed to be 24/7, but it was shown in Section 4.5 that the process likely averages 14 kW more during the day. However, the chiller load profile follows what we would expect if the load were to slowly ramp up during the day, peak when outside conditions peak, and then slowly ramp down at night. Process loading does not do this, but outside conditions do. Also, the chiller load typically tracks outside temperature almost perfectly, even on rainy days, so process load being solely responsible is highly unlikely, although not impossible. Thus, some non-negligible amount of cooling load remains unaccounted for by infiltration load and process load. The exact number is subject to variation, but 50-60 kW of load seems more than reasonable.

Is this a one day phenomenon? No. This happens day after day, month after month. This points to one problem in particular: condenser heat transfer. It could be that this is merely a consequence of design, and as the outside air temperature increases, there simply is not enough heat transfer space to compensate for without increasing the temperature difference. This would be done by increasing the pressure and temperature of the refrigerant, and as a consequence lowering the COP of the chillers.

There could be a much more mundane cause, however. The plant already has a scheduled program to clean the outside heat transfer surfaces on all of their chillers. Are the heat transfer surfaces inside the chillers periodically checked for sludge buildup?

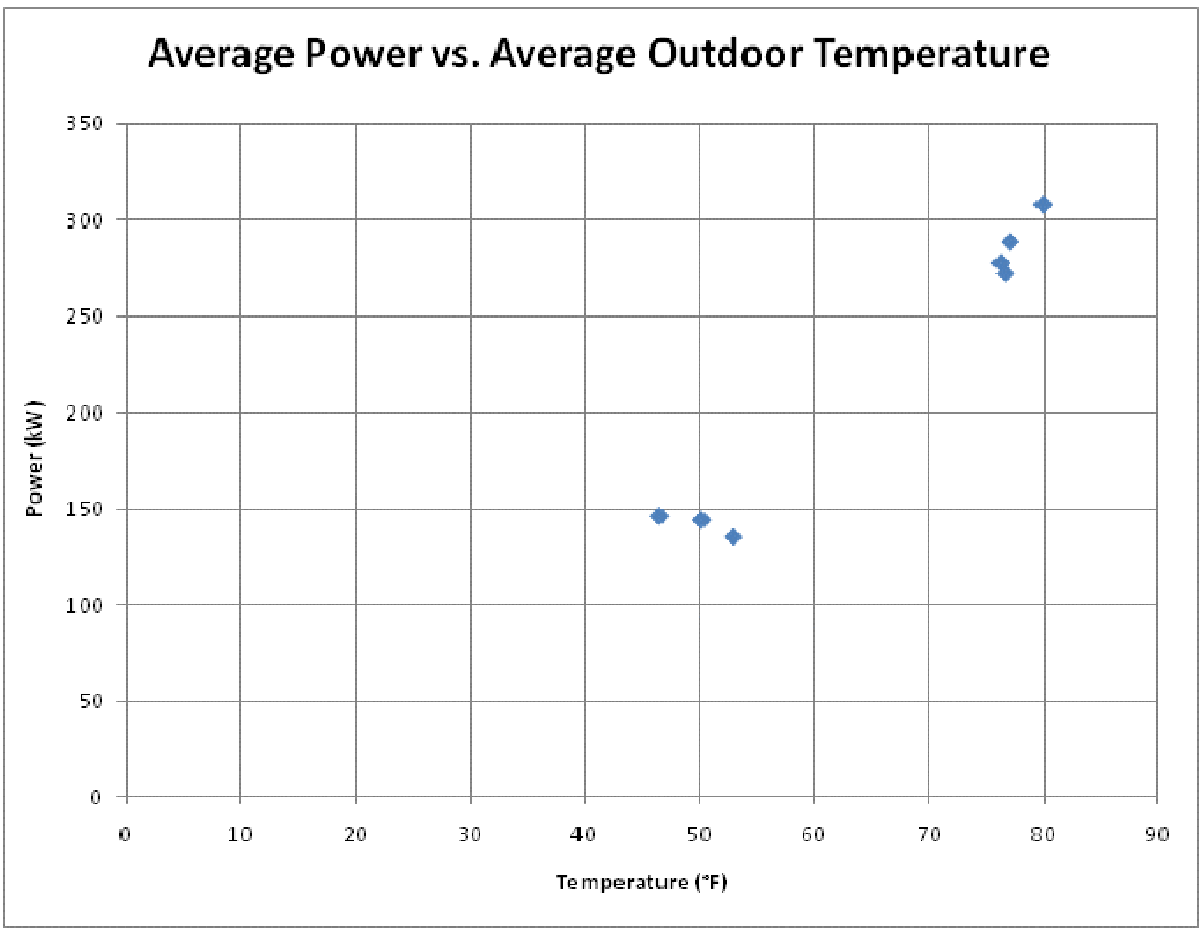

Figure 5.2: Average Power vs. Average Outdoor Temperature

Figure 5.2 shows a graph of chiller power vs. outside temperature. The 7 data points on the graph each average at least a week of data, to try and eliminate short term variables which may confuse the relationship. The minimum chiller load appears to be 120-130 kW. This relationship will be used to try and estimate a total cost of running the chiller in a typical year.

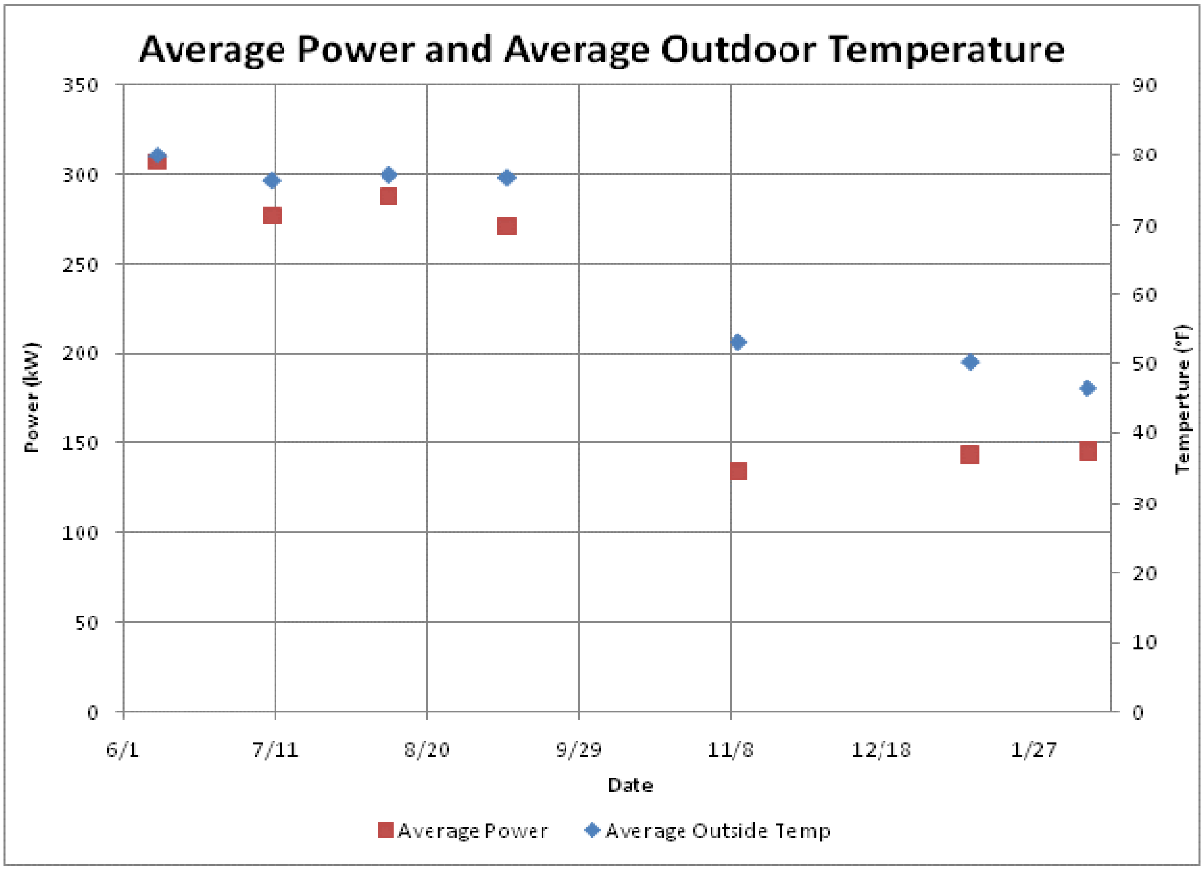

Figure 5.3: Average Power and Average Outdoor Temperature

Figure 5.3 shows chiller power and outside temperature. This figure has the time graphed as well. The time at which the data points are graphed is the middle of the period around which the temperatures and powers were averaged. It is clear that there appeared to be a minimum chiller power use in the winter. This may be helped with some kind of effective economizer setting, where outside dampers can be opened up, and return dumped into the plant.

The actual cleanroom data from July is shown in Table 5.1 below.

Table 5.1: July Chiller Power Cost

	Daily Data Averages			Sum Monthly Average			Percent Cost Understatement
	Energy / Power		Cost ($)	Energy / Power		Cost ($)	(%)
Offpeak	148,360	kWh	$4,052	200,758	kWh	$6,344	
Onpeak	52,398	kWh	$2,375				
Energy Subtotal	200,758	kWh	$6,427	200,758	kWh	$6,344	1.3%
Demand	372	kW	$3,732	279	kW	$2,797	25.1%
Max Demand	467	kW	$4,688				40.3%
Total Cost			$10,159			$9,141	10.0%
Max Total Cost			$11,115				17.8%

In Table 5.1, it is shown that the average kW draw in July was 279 kW. However, the maximum daily on peak average was 372 kW, and the maximum 30 minute average demand was 467 kW! It is likely that that hot summer day set the peak at the facility. To be conservative, assume that the demand is only 372 kW. This demand figure is 25% more costly than the average demand would cost, and 10% more costly overall. This is typical of all the summer months, and this is actually more extreme in the winter, since daily maximum chiller power use can peak at over 200 kW.

From historical weather data, average monthly temperatures can be found for the site. An average temperature in the upper 70's corresponds to 270-280 kW average power draw. An average temperature of 45-50°F corresponds to an average power draw of 150 kW, extrapolating from these figures allows us to come up with some numbers for a typical year.

Table 5.2: Extrapolated Chiller Energy Cost

	Energy (kWh)	Energy Cost ($)	Demand (kW)	Demand Cost ($)
Summer (4 mo.)	784,800	$24,800	1,090	$13,500
Winter (8 mo.)	892,800	$27,700	1,240	$8,500
Total	1,677,600	$52,500	2,330	$22,000

In the table above, the total cost comes to $74,500/yr. Note that a 25% demand surcharge on all the extrapolated demand figures.

So what should the chiller system need to provide? The heat load in the cleanroom now would need to be roughly estimated. If this load is ~400 kW, and the typical daily maximum summertime infiltration load is 42.7 kW (from earlier in this section) the maximum amount of energy the chiller should have to remove is around 440 kW (1,500,000 BTU/hr), which is roughly equivalent to 150 kW of chiller power at 1.25 kW/ton.

The possible potential for chiller power savings is up to 150 kW of summer load shedding, depending on the internal heat generation of the cleanroom. Unfortunately, since it has been shown that the chillers do not efficiently handle high outdoor temperatures or low loading, the easiest part to concentrate on is eliminating reheat.

The reheat is seen to average about 120 kW throughout the whole summer all the way through late September, and averaged 50 kW during December and January (not including Christmas break, where it averaged 140 kW). The reheat does not match the outside temperature (like chiller load is shown to do in Section 4.5), but it is clear that the cold months use less reheat. In order to come up with an extrapolated reheat energy cost some more assumptions need to be made. In order to be conservative, it will be assumed that there are five months where the reheat operates at 120 kW, and seven months where it operates at 50 kW.

Table 5.3: Extrapolated Reheat Energy Cost

	Energy (kWh)	Energy Cost ($)	Demand (kW)	Demand Cost ($)
Summer (4 mo.)	345,600	$10,900	480	$4,800
Winter (8 mo.)	338,400	$10,500	470	$2,400
Total	684,000	$21,400	950	$7,200

This is seen to total $28,600/yr from the table above. In direct chiller savings, at 1.25 kW/ton, it would save 43 kW of load from the chiller in a summer month, and 18 kW of load in a winter month.

The total cleanroom reheat and chiller cost is therefore seen to average about $103,100/yr, in a “normal” year. It was noted that fan cost may not be included, and peripherals such as chilled water pumping cost and HVAC maintenance were not included in this figure.

Chapter 6: New Cleanroom Operation

6.1 New Cleanroom Operation Overview

The company is hitting this problem from multiple angles. In Chapter 4, it was discussed that several things can be done to help the cleanroom operate more efficiently, largely with the same equipment. This chapter will explore the things the company is currently doing to discover the potential of the measures. Recall from Chapter 1 that the cleanroom is currently taking in 5,000 cfm of outside air, equally spread across AHU's 1-4. Now, the plant wants to change how the AHU's operate. The plant wants to utilize AHU's 2-4 as filters and temperature control alone, hoping to completely eliminate reheat on these three units.

AHU 1 will now take 15,000 cfm of outside air to pressurize the cleanroom, and not condition any return air. Note that this is an additional 10,000 cfm from the previous operation. This AHU will be utilized to control humidity, and therefore will potentially operate at a different temperature from the other units, depending on cleanroom conditions. Humidification by AHU's 2-4 still may be necessary during cold or dry periods.

Should the latent cooling load be too much for AHU 1 to handle, it is recommended that one other AHU (say, AHU 4) be used for dehumidification, in a periodic manner. Should this need to happen, it is also suggested that sensible cooling be lowered dramatically on AHU's 2 and 3 to minimize the amount of reheat required. This should be rare, however. Since the new cleanroom operation eliminates all possible sources of outside infiltration (other than AHU 1), the only possible way this should be required is by an increase in internal humidity generation.

In addition, cleanroom operating conditions have been changed from the previous rigid 50% humidity. Now, the summertime maximum is as high as 58% humidity, while the winter minimum is as low as 42% humidity. This in and of itself will go a long way towards fixing the control issues the cleanroom was having. It is no secret to plant employees that the cleanroom HVAC system was having trouble keeping the room at 72°F, but it was also struggling to keep the cleanroom at 50% RH, as shown throughout Chapter 4. It is thought that the AHU's will not attempt to fight each other locally now, like they were before. Whether or not this has been accomplished yet will be discussed below.

6.2 AHU 1 Modifications

The plant currently has about 6,000 cfm of exhaust as hot contaminated air leaving the cleanroom through roof vents. An air to air heat exchanger has been installed in this air stream to preheat the incoming outside air in the winter. The design HX effectiveness is 63.4%, and it is estimated to be is use when the outside wetbulb temperature drops below 57°F. It has the ability to increase the temperature of the incoming 15,000 cfm of air by 25°F, illustrated by example air temperatures in Figure 5.1 below. In addition, if the outside air is too cold, AHU 1 has the capability to draw plant air as makeup as well, in order to avoid electric strip heat. The plant has a gas heating system, and with natural gas running between $5-$10/MMBTU, it is preferable to utilize the plant heating system. Since the cleanroom will now be pressurized, it provides conditioned air to the plant through infiltration, thus potentially using it as a source of makeup air is not likely to cause the plant HVAC engineer any ulcers.

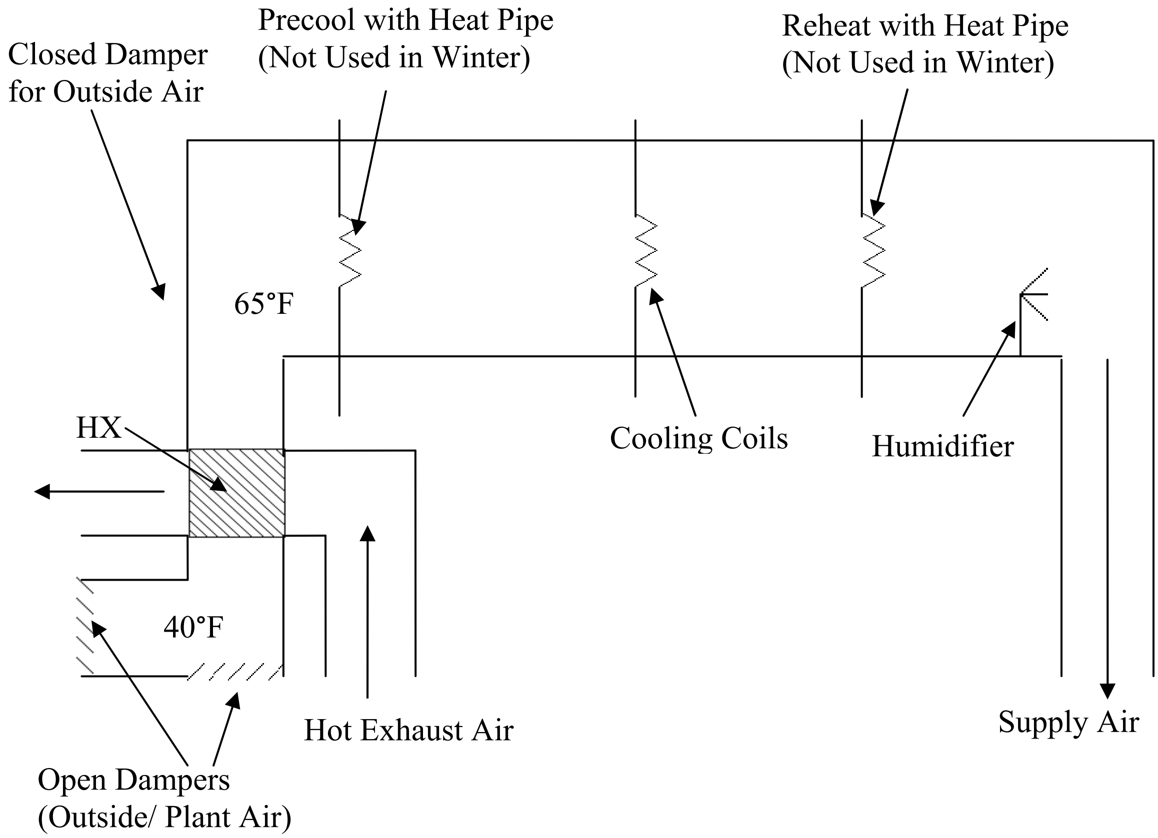

Figure 6.1: New AHU 1 Winter Setup

In the summertime, hot, humid air is cooled in order to remove latent heat. Remember, AHU 1 now needs to do the latent cooling for the whole cleanroom, acting like a Faison bypass. A Faison bypass allows the HVAC system to decide how much air should be cooled to produce the desired dehumidification, essentially, it tries to decouple the sensible and latent cooling. The control it will utilize is the ability to cool incoming air to a range of temperatures, from 48-55°F. The previous operation only utilized 5,000 cfm of outside air. To avoid loading up the cooling coil on hot summer days with 15,000 cfm of hot air, a heat pipe has been installed, which is capable of essentially shifting the heat around the cooling coil. Figure 6.2 includes an example of hot air moving through AHU 1. Air at 90°F is cooled to 80-81°F by the heat pipe, and then the cooling coil cools this air to 55°F, for example. Then, the other side of the heat pipe cleverly reheats the air to 65°F. The COP for moving heat from hot to cold is infinity, since no work is done to move the heat in this system.

Please note that the heat pipe is only useful if there is a supply of heat (hot incoming air) and a source for that heat (reheat on the cold side). There is no ability to control temperature coming off the heat pipe. Thermodynamics dictates that what energy is pulled out of the air on one side has to put back in on the other. Thus sizing is <u>very</u> important for any heat pipe application.

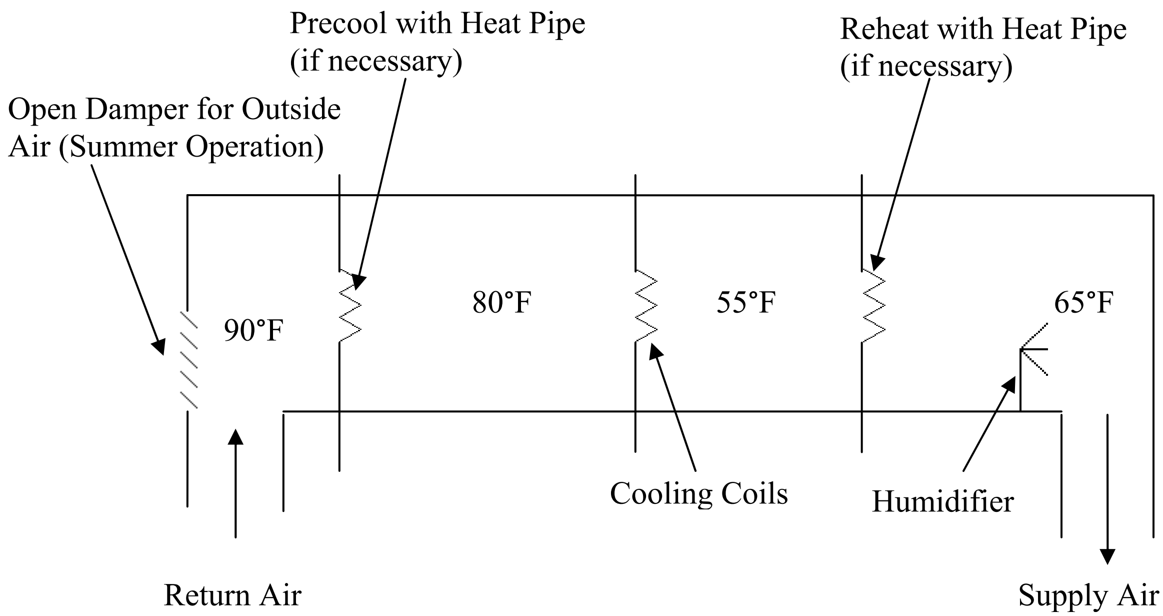

Figure 6.2: New AHU 1 Summer Setup

What is the energy transferred by the heat pipe? This can be done by a simple thermodynamics calculation, as done in equation 6.1 below.

$$\dot{Q}_{Heat\ Pipe} = 15{,}000\ cfm \times \frac{0.24\ BTU}{°\mathrm{F}\ lb}(90°\mathrm{F} - 80°\mathrm{F})\left(\frac{0.075\ lb}{ft^3} \times \frac{60\ min}{hr}\right) \tag{6.1}$$

$$\dot{Q}_{Heat\ Pipe} = 162{,}000\ BTU/hr$$

What does this mean in terms of money saved? The plant actually saves twice, on chiller cost and reheat cost. The chiller savings will be less than the reheat savings, since the chiller has an associated COP.

$$\dot{Q}_{Chiller\ Savings} = \frac{162{,}000\ {}^{BTU}/_{hr}}{12{,}000\ {}^{BTU}/_{ton}} \times 1.25\frac{kW}{ton} \tag{6.2}$$

$$\dot{Q}_{Chiller\ Savings} = 16.9\ kW$$

So, when the heat pipe is working, it saves approximately 17 kW of chiller power. The heat pipe will operate whenever the outside dry bulb temperature is over 65°F. In North Carolina, this actually is about 40% of the year. The actual power savings may be more or less. The chiller has been shown to operate less efficiently at higher temperatures, thus possibly increasing the actual power savings as the temperature climbs. The chillers have also shown a reluctance to decrease load efficiently, possibly lowering savings. Actual monetary savings will undoubtedly be greater than the average kWh and kW values, since higher temperatures correspond strongly with summer, on-peak time periods. The reheat savings are computed to be:

$$\dot{Q}_{Reheat\ Savings} = \frac{162{,}000\ {}^{BTU}\!/_{hr}}{3{,}413\ BTU/kW} \tag{6.3}$$

$$\dot{Q}_{Reheat\ Savings} = 47.5\ kW$$

The total savings provided by the heat pipe is therefore 64.4 kW, and will be available for 40% of the year. Assuming the temperature differences across the reheat coils are fairly constant and that demand will only be saved for the four summer months and 4 winter months, and that the energy savings will be averaged equally over on and off peak hours, total heat pipe savings is estimated at $10,950/yr.

6.3 New Operation Analysis

The last data set taken shows the new cleanroom operation. However, comparing Figure 6.3 and Figure 4.21 does not show much of an improvement, on the surface. The reheat still oscillates between 0 and 85 kW, just as in the winter operation in December. Even worse, the reheat spikes to an incredible 230 kW in late March.

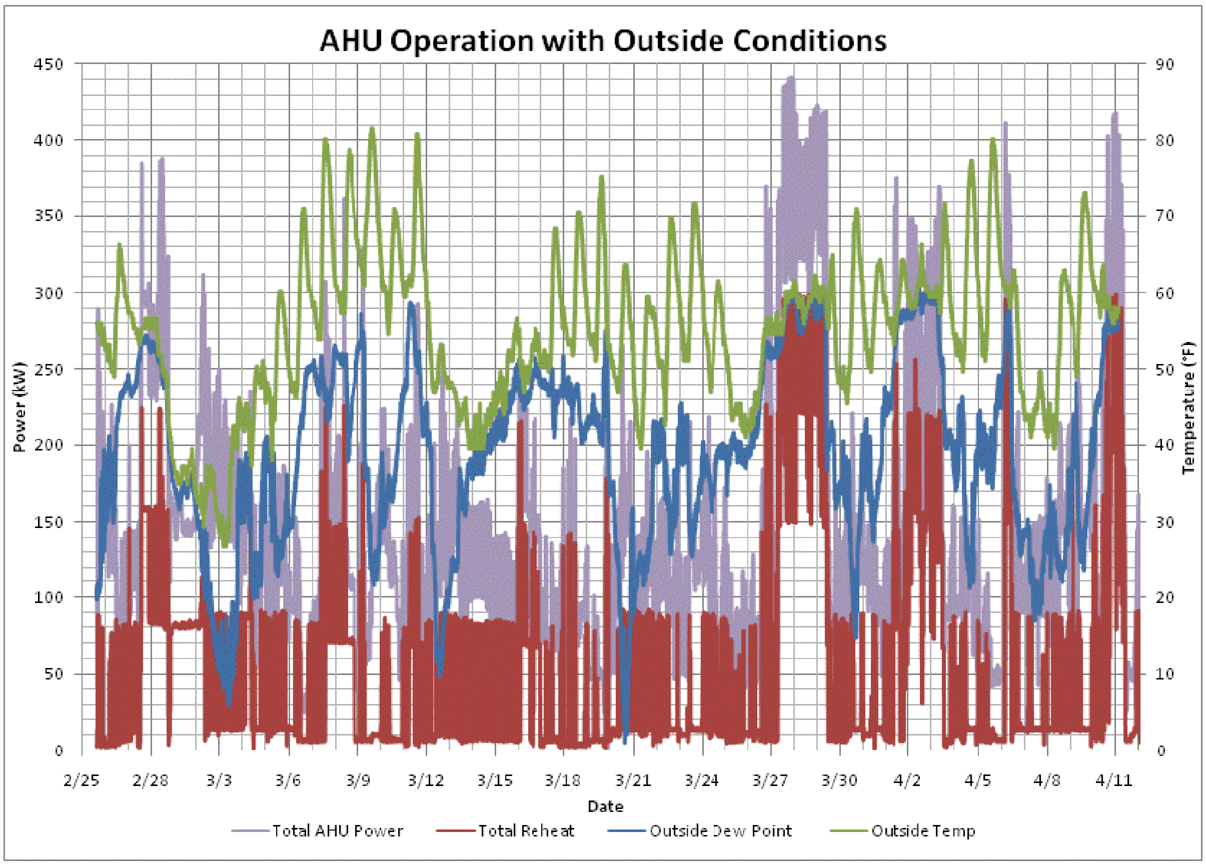

Figure 6.3: New AHU Operation with Outside Conditions

As Figure 6.3 shows, there is still a healthy amount of reheat. The average reheat value for the whole time period is 54.4 kW. Even excluding 3/27-3/29, the average is still 47.3 kW. But this is just an overall breakdown. The cleanroom also changed operation in specific ways. There still is a large amount of reheat being used during operating periods. For example, Thursday, April 2nd was a day where reheat spiked. The outdoor temperature range was typical for an April rainy day: 66°F high, 58°F low. Reheat averaged 167 kW on this day. The magic number for this type of reheat spike seems to be a dew point of around 50°F. Later dates in April show the same trend, and suggest that this has to do with outdoor humidity, not indoor cleanroom conditions. Recall that winter reheat spikes seem to be centered on periods of low cleanroom activity, namely weekends, and contrast this with these spikes, which center on wet periods.

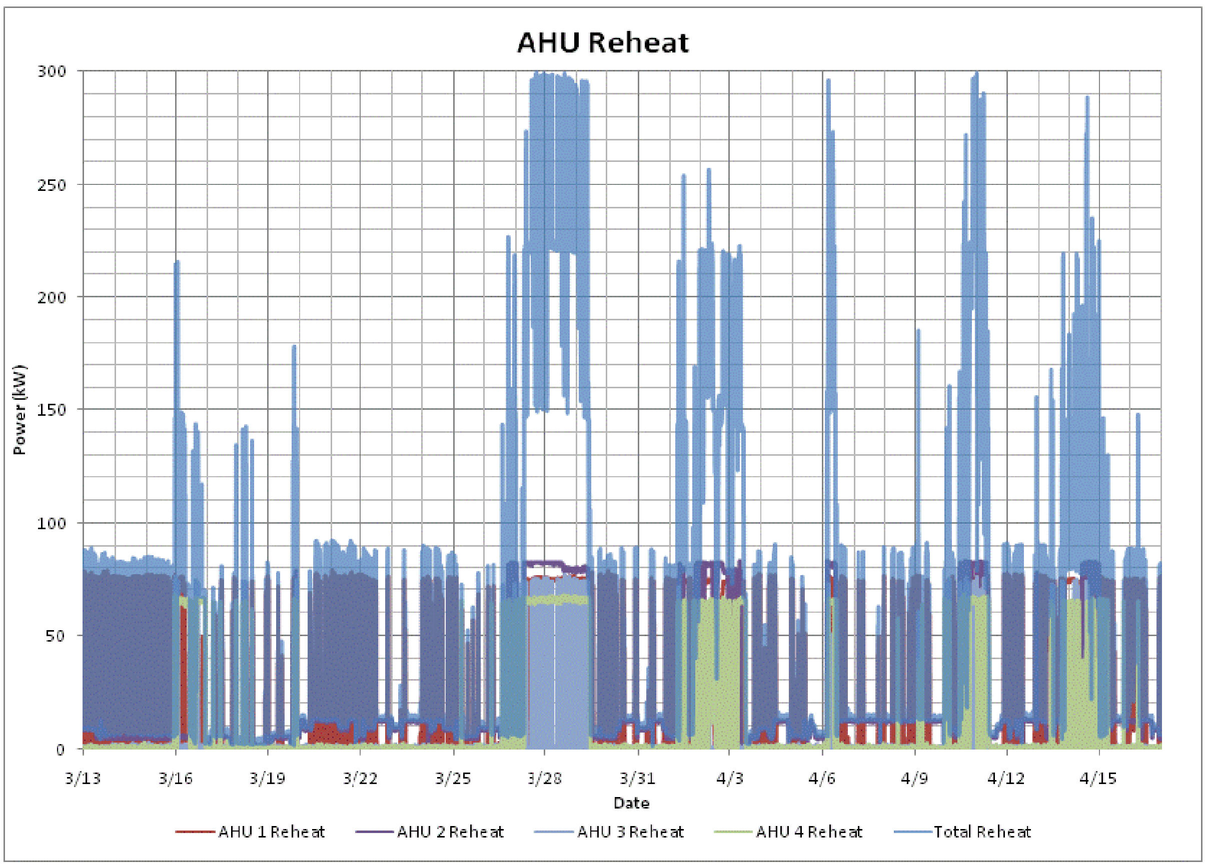

Figure 6.4: New AHU Reheat

On the above figure, it is shown that all reheats get involved at times. The reheat most responsible for increasing average power draw is AHU 1 reheat. However, during periods of high humidity, AHU 2 and AHU 4 are more to blame than AHU 1.

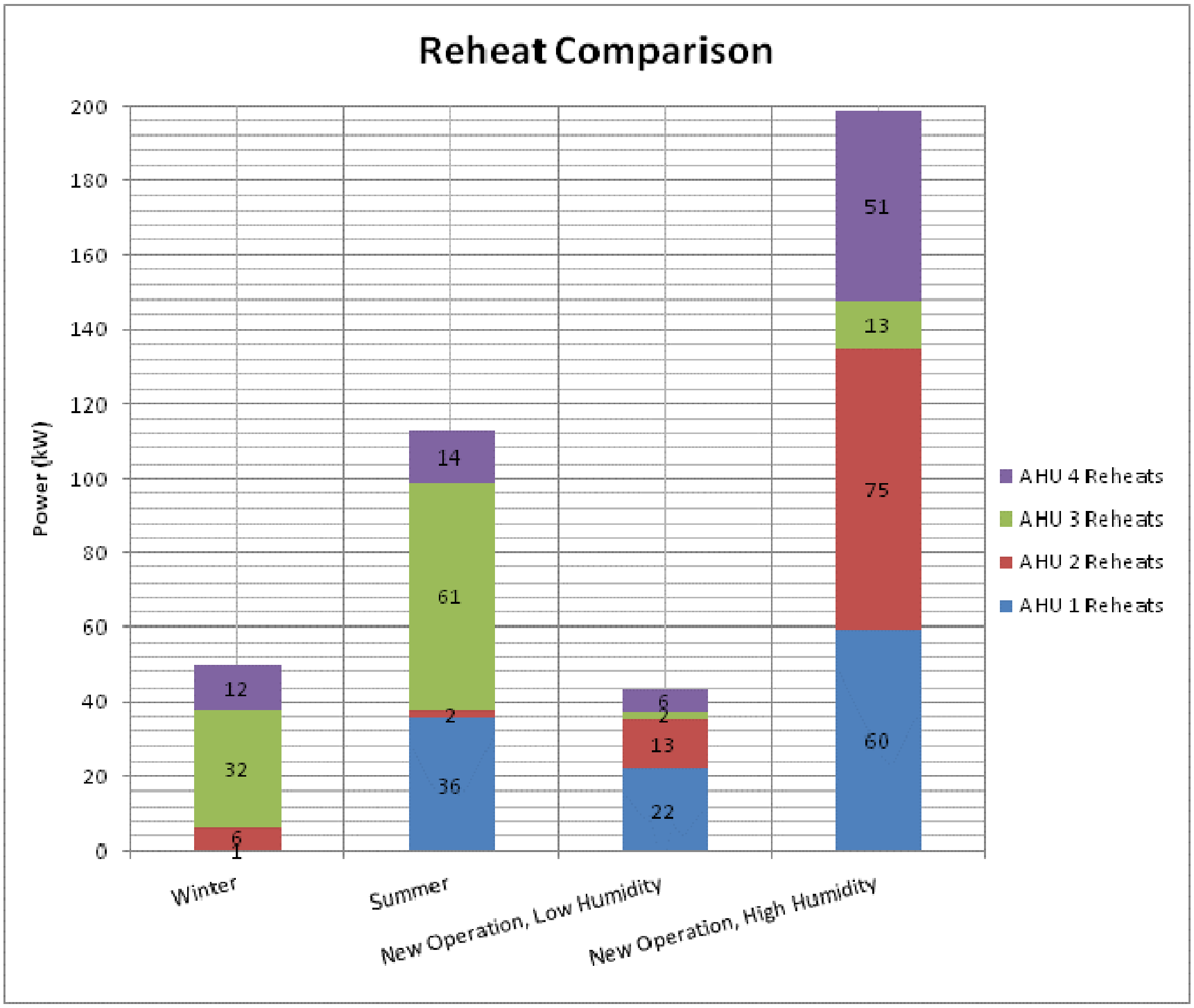

Figure 6.5: Reheat Comparison

Figure 6.5 shows the average reheat for a season and which units use it. Clearly the reheat on AHU 3 was a large component of the former cleanroom operation. In the winter operation column, December-January data was used, with the three days after Christmas removed due to very unusual operating behavior. For the summer column, August data was used, and the month is typical of all summer ``data. For the new operation, several periods of low humidity are lumped together, and notable periods of high humidity are lumped together, such as the weekend of 3/27-3/30 and 4/2-4/3. With the new setup AHU 1 is a large part of the cleanroom reheat operation, especially during times of low humidity (51% of the total during low humidity). This is exactly what was expected. However, how (and to some extent why)

AHU 2 maxes out at *over* 83 kW of reheat when on is a mystery. What is clear is that during times of high humidity the cleanroom reheats really kick on. This does *not* bode well for summer operation. Summer in the south is hot and sticky. The reheat on AHU 3 is largely unused now, with the notable exceptions being spot use during times of high humidity.

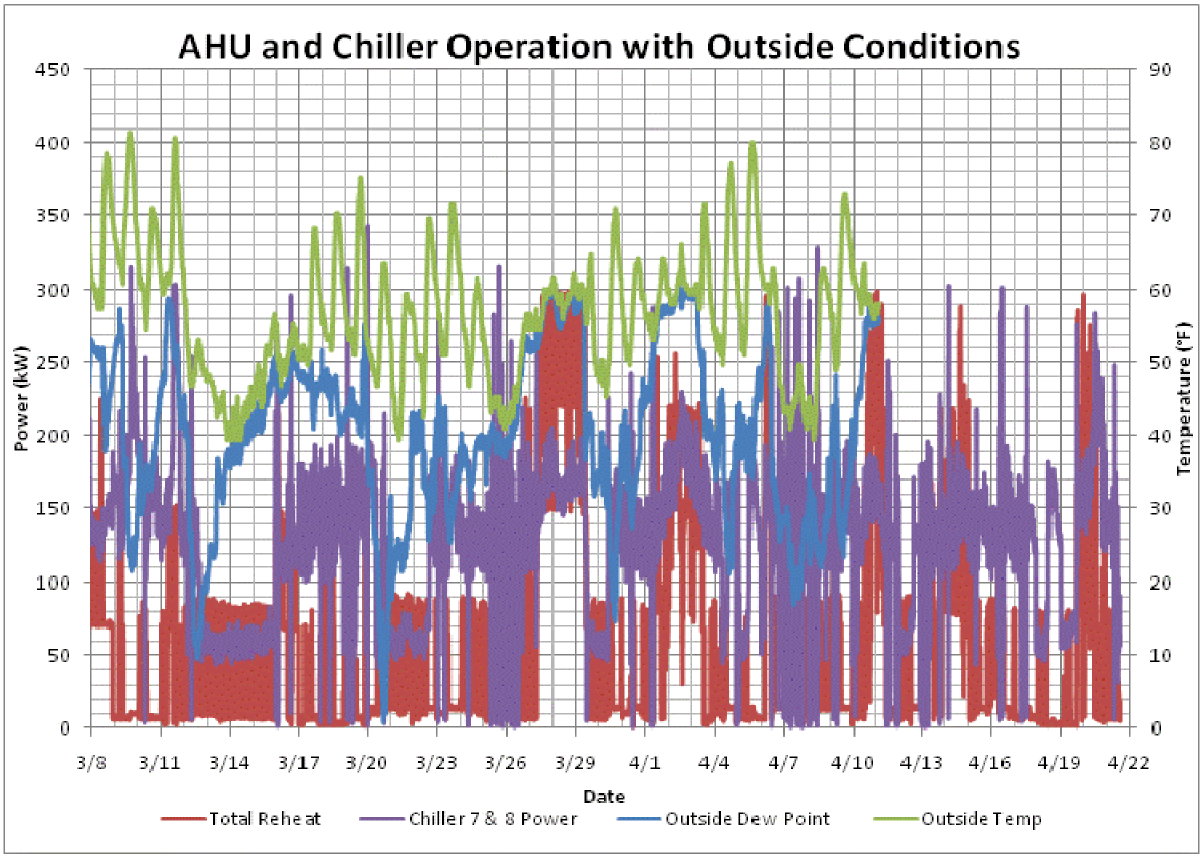

Figure 6.6: AHU and Chiller Operation with Outside Conditions

Notice Figure 6.6 above. During periods of high reheat, the average reheat is 199 kW and during periods of low reheat, the average is 44 kW. A small amount of dehumidification is taking place, thus a massive amount of overcooling on the order of 530,000 BTU/hr (the 155 kW of reheat) should be visible. This would be equivalent to 44 tons of cooling, or 55 kW of chiller power at 1.25 kW/ton. When the reheat is high, the chiller averages 175 kW, and when the reheat is low, the chiller averages 115 kW of cooling during this two month span. That is 60 kW of cooling, and assuming ~5 kW for the increased outdoor infiltration due to higher temperature and dew point, this is amazingly accurate.

6.4 Operation Examples

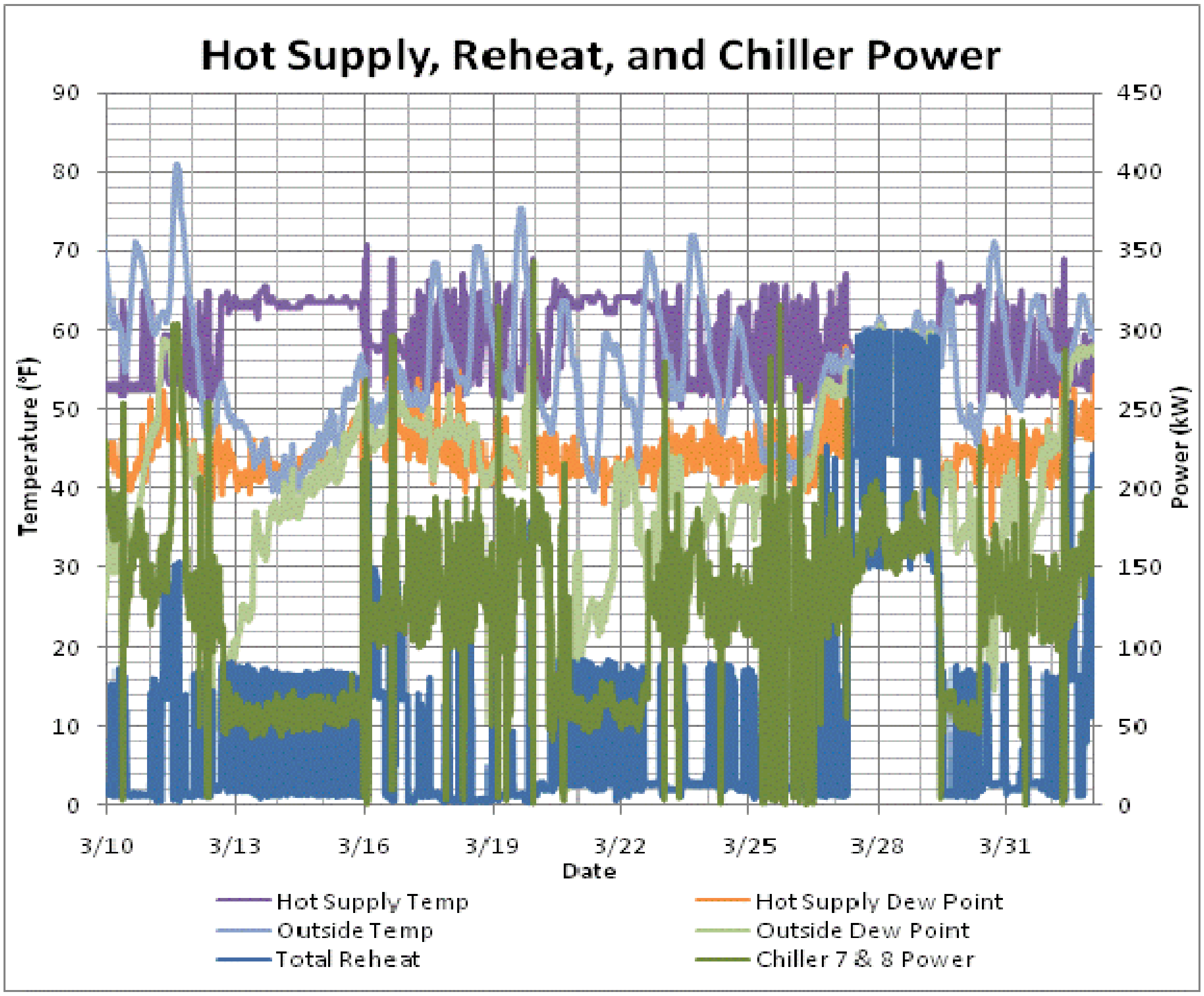

Figure 6.7: New Operation Hot Supply, Reheat, and Chiller Power

Figure 6.7 illustrates some of the improvements with the new cleanroom operation. Notice in particular the difference between 3/13 and 3/28. On 3/13, the cleanroom appears to be in a shutdown mode, in winter operation, and thus there does not need to be any cooling or reheat. This is definitely an improvement over previous periods of winter shutdown operation, as shown in Figure 4.21 and 4.23 over Christmas week. Now compare the three day period around 3/28 to the week after Christmas. The reheat averages 233.4 kW, which is almost double what the reheat operation ran on average the previous summer. Actually, this average is even a little low, since AHU 2 maxes out the 100 amp reheat CT's and the 200 amp total power CT.

So why is this the case? From chiller 1-5 data, it is known that between 12:40 PM on 3/27 and 7:40 AM 3/30 (Friday afternoon to Monday morning) the plant is operating at a low level. Figure 5.3 backs this up for the cleanroom by showing a constant temperature between these two periods of time. However, on 3/29 at about 10:50 AM, this temperature changes from 57.6°F to 63.4°F, the chillers decrease from 169 kW to 58 kW (essentially no cooling. The 140 ton compressors on chiller 7 seem to be merely on), and the reheats suddenly go from 230 kW to 30 kW. The change itself can be explained by the outside wet bulb temp, which drops from 57.9°F to 46.8°F inside of an hour and a half on 3/29. However, this type of reheat heavy HVAC operation was supposed to have been changed. There is not supposed to be massive over cooling anymore. This clearly is not the case for this time period.

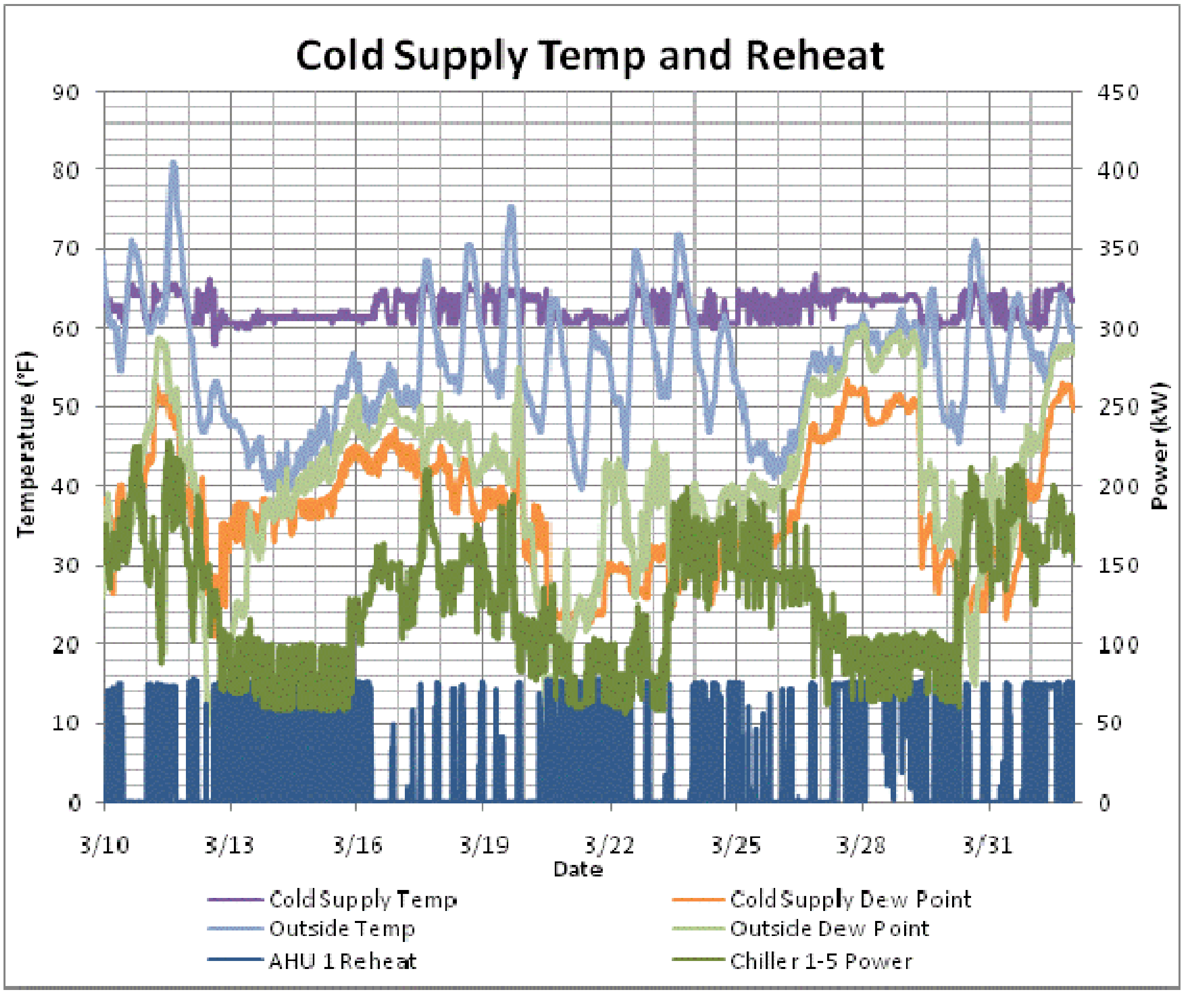

Figure 6.8: Cold Supply Temp and Reheat

AHU 1 now handles 15,000 cfm of outside air. Figure 6.8 shows the cold supply temp and reheat over this period. When the plant is in low operation as chiller 1-5 power is reduced. These periods of low operation are all weekends, as expected. Incoming cold supply air is typically hotter when the plant is in operation, probably signifying that the heat exchanger shown in Figure 5.2 is in operation, and the damper controls are apparently working, since a nice supply temperature between 60-65°F is being maintained. Since the cleanroom is not in operation on the weekend of 3/28, it will be assumed that there is no hot exhaust during the period between 3/27 and 3/30, so the whole of the cold supply incoming air now needs to be heated by the reheat.

The operation makes it clear that 67 kW of reheat on AHU 1 are needed during the period on 3/28-3/29 that the AHU is dehumidifying. It is also clear that as soon as the outside humidity drops, the reheat demand drops to 10 kW that day. That night, the temperature drops some, and the reheat responds by slowly ramping up to 20 kW. This sensible load is almost irrelevant when compared to the load the dehumidification reheat puts on the system. Why is this the case?

6.5 Humidity Controls Discussion

Some of the possibilities for the issues shown above will be discussed below. These possibilities may not be all inclusive.

First, it is useful to try and eliminate some basic causes of the massive reheating during times of high humidity. The cleanroom system as it is now set up should not care about outside humidity, with the sole exception of AHU 1.

Let us examine a hypothetical situation with the cleanroom system as it is thought to operate now. It is 110°F outside, with a dew point of 90°F. An extreme example, to be sure, but it will help serve the point. The plant itself is air-conditioned, and for the purposes of the discussion, stays under 80°F, with a dew point of 60°F. The cleanroom is pressurized, making infiltration from the plant into the cleanroom difficult. AHU's 2-5 only recirculate cleanroom air, and only cool the cleanroom created sensible load, which requires no reheat. To them, the outdoor conditions should be irrelevant. AHU 1 uses a heat pipe to cool the outdoor temperature to 100°F, and the chillers to cool the air from 100°F to 50°F. The heat pipe then turns around and heats the air coming off the backside of the cooling coils to 60°F, making reheat a minimal consideration on AHU 1, if necessary at all.

In summary, reheat is unnecessary, with the possible exception of AHU 1. Yet the cleanroom is apparently sensitive to dew points in the upper 50's! Clearly, something is not as it seems. The most obvious way for the cleanroom to be sensitive to outside conditions would be infiltration from the outside. Are the dampers on AHU 2-5 really closed? There are also machines that receive outside air for cooling, then add some indoor air, and exhaust this air to the outside. Is some of this air making its way into the cleanroom? If it is, is it filtered?

However none of the previous answers account for the fact that controls should now be set up differently, not allowing those units to attempt dehumidification except under extreme circumstances. From looking at the dew points in Figure 6.9, even considering the possible range of inaccuracy in the dew point measurements, it can hardly be said that the hot side dew point constitutes an extreme situation. It only seems different because a small natural variance in humidity is now being allowed.

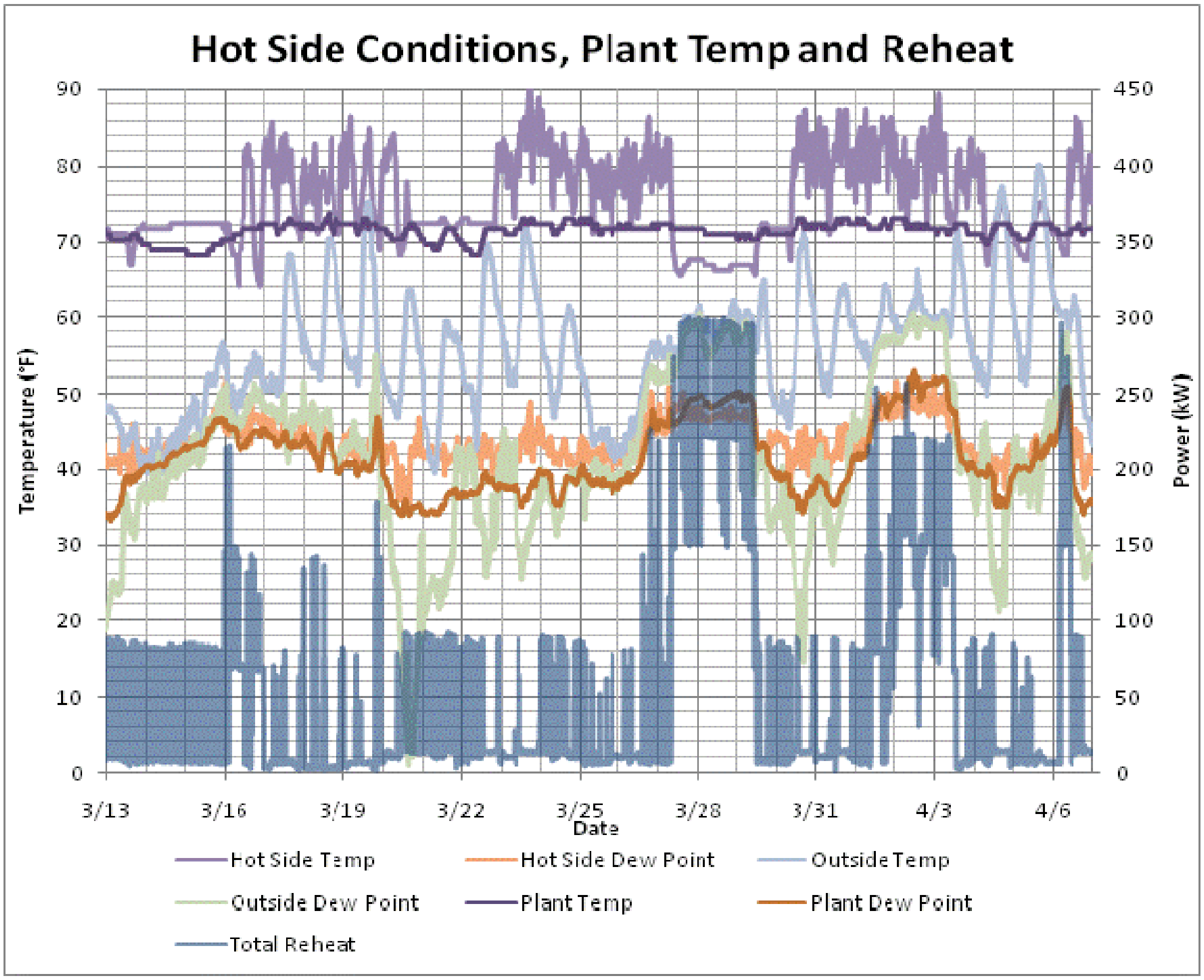

Figure 6.9: Hot Side Conditions, Plant Temperatures and Reheat

So the next in the logical sequence is controls. How does AHU 1 know what temperature to cool the air to? Does it know what the rest of the AHU's are doing, or does it only have a picture of the space immediately around it. Does AHU 1 measure humidity before the air enters the space, or after? Where do the other AHU's measure their air for temperature and humidity?

The next thing to look at would be the chilled water controls. How is chilled water flow managed? Is it controlled that way in reality? What about valving? Is there a bypass system, or are there three-way valves? Could some valves be stuck?

The humidity question is a critical one to answer. Assuming that all 199 kW of reheat is unneeded in the summer, and therefore that much overcooling, the cost of reheat is as follows:

$$Reheat\ Cost = \left(199\ kW\ \times\ 24\frac{hrs}{day} \times 30\frac{days}{mo} \times 4\frac{mo}{summer} \times \frac{\$0.0316}{kWh}\right) \quad (6.1)$$

$$+ \left(199kW \times \frac{\$10.03}{kW} \times \frac{4mo}{summer}\right)$$

$$Reheat\ Cost = \$18{,}110 + \$7{,}990$$

$$Reheat\ Cost = \$26{,}100$$

The power it takes to overcool the air enough so that it has to be reheated is:

$$Overcool\ Power = 199\ kW \times \frac{3{,}413\frac{BTU}{kW}}{12{,}000\frac{tons}{BTU}} \times 1.25\frac{kW}{ton} \quad (6.2)$$

$$Overcool\ Power = 70\ kW$$

The total cost on the chiller for 70 kW of average demand is below. Keep in mind that the total cost to run a demand load on the chiller was found to be 25% more than the demand charge, as shown in Chapter 5. This has been included below:

$$Overcool\ Cost = \left(70\ kW\ \times\ 24\frac{hrs}{day} \times 30\frac{days}{mo} \times 4\frac{mo}{summer} \times \frac{\$0.0316}{kWh}\right) \quad (6.3)$$

$$+ \left(70\ kW \times \frac{\$10.03}{kW} \times \frac{4mo}{summer} \times 1.25\ chiller\ demand\ penalty\right)$$

$$Overcool\ Cost = \$6{,}380 + \$3{,}520$$

$$Overcool\ Cost = \$9{,}900$$

$$Total\ Cost = \$36{,}000 \quad (6.4)$$

The total cost of not fixing this situation could run $36,000 this summer. That is a substantial cost, and it does not include May and September, which are also likely to have many humid days. This situation bears looking into.

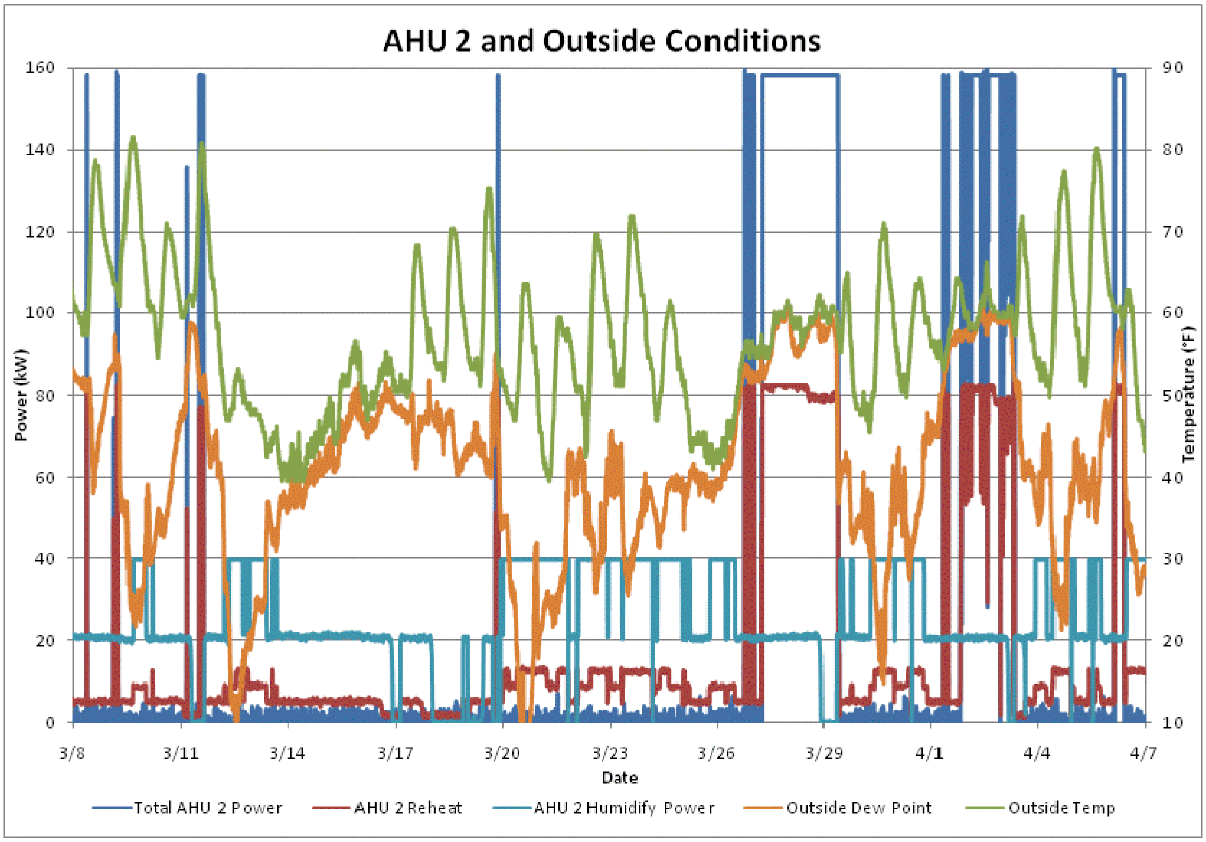

Figure 6.10: AHU 2 and Outside Conditions

By looking at the figure above, it is clear that the humidification on AHU 2 comes on during periods of low outside humidity. Also, it is clear that the CT's max out on AHU 2 in almost all the locations reheat is on. In addition, it is clear that reheat is on when the humidity is high. All of these changes mean that the cleanroom should use less energy to heat and cool 15,000 cfm of outside air than it did for 5,000 cfm of outside air, but kinks in the system do need to be worked out.

Chapter 7: Alternative Old Cleanroom Operation

The description necessarily has holes in it, due to data collection issues, but an attempt to explain some of the data readings not explained by the current setup will be listed below. Since the author is not on site and assumes the accuracy of the data logging equipment where the data logging equipment has proven to be reliable in the past, there exists the real possibility that much of Chapter 7 could be incorrect.

The air handling units have fans which take power from a different circuit than the total power circuit which was data logged in this study. This would explain the rather low power readings for the fan on AHU 1, 3, and 4, as well as the lack of a fan reading on AHU 2, as described in Section 4.4. The fans will be assumed here to run at 50% power, which would make the pressure rise 70% of design, and the fan speed and flow rate 79% of maximum.

Fan energy = 4 x 400 hp x 0.5 = 150 kW

Two chiller pumps are assumed to run continuously at 15 hp each, with the third acting as backup.

Chiller Pumps = 2 x 15 hp = 22 kW

Summer Operation (outside temperature above ~45°F)

AHU 1 does not dehumidify. This explains the temperature readings. It sounds plausible because there is no humidity load in zone one, so this control may not know the rest of the room is fighting humidity so hard. It would also provide a great explanation of how AHU 1 could be humidifying every few days in the summertime. Energy balances presented in Section 4.2 now actually balance. Sharp readers point out that 50 hp of fan sensible heat would now need to be accounted for in the energy balance. While this does increase the flow rate required to balance this energy equation to the upper edge of the range of error, it is useful to remember that there may indeed still be a small residual cooling load, if the valves do not shut all of the way, and 50 hp of fan energy is essentially a blind guess.

AHU 4 works as advertised. There is not enough reheat during times of low production. This makes the assumption that AHU 1 does not dehumidify even more reasonable, since supply temperatures are shown in zone 4 to be a direct function of the reheat capacity when dehumidification is in place.

Daytime production is slightly more than nighttime production.

Chillers 7 and 8 have inefficiencies associated with maximum outside temperature, and the ability to run at part load. The chillers have 6 compressors total, but only 4 can run independently.

Winter Operation (outside temperature below ~45°F)

Humidification runs on AHU 2 when necessary. Fortunately, reheat does not seem to run when it is sufficiently cold outside. However, there are many winter days where the high is in the 50's, and so reheat does come on for a time during the winter. When the plant is shutdown for holidays, the chillers and reheats tend to run pretty hard.

Chapter 8: Conclusions

There are several important conclusions to be made from this system, and the data presented above.

First, the importance of complete and accurate data cannot be understated. This can really help fill in the gaps between what is thought about how the system is operating and how the system is actually operating. It also allows for an accurate understanding of that is at stake. In the initial hypothesis, it was thought that the system had 742 kW of reheat operating, and 338 tons of chiller power, for a cleanroom base reheat and chiller cost of $385,000/yr.

From the extensive data logging, it was found that for the wintertime operation (dew point under 45°F) the cleanroom reheat is only 50 kW, but is a more substantial 120 kW in the summertime operation. This is, at the maximum, 16% of the reheat that was originally suspected. Also surprising is the summer cooling load of 232 tons (290 kW, assuming 1.25 kW/ton), which is only 69% of the suspected cooling load. Remember that much of the data showed a large probability that chiller efficiency decreases with outside temperature, so real chiller load is probably less than 200 tons.

All of the numbers in the initial hypothesis are based on cooling 180,000 cfm of air to the dewpoint, and then reheating this air back up to temperature. An assumption was made regarding the internal heat generation of the cleanroom, ~400 kW (~1,350,000 BTU/hr). This assumption may or may not be true, but for the purposes of the former cleanroom chiller operation, this should be almost irrelevant since 180,000 cfm of air cooled from 75°F (and 50% RH) to 55°F is about 4,000,000 BTU/hr (338 tons, 422.5 kW). Since the real chiller use does not even approach this value, the cleanroom cannot possibly be operating in this manner.

What about reheat? Even assuming that the low reheat is due to a much larger internal heat generation of the cleanroom, this still cannot account for the lack of overall cooling. Some of this low reheat is due to the fact that there is insufficient reheat, and the cleanroom therefore gets cold in certain segments.

Having proved that the cleanroom does not cool 180,000 cfm to the dew point as was originally intended, is it possible that 180,000 cfm of air is not being circulated? Plant personnel already know that the cleanroom HVAC controls fight each other. Some areas are cold, and some are hot. To take it a step further, it is probable that this fight extends to air flow. The cleanroom may not be seeing 180,000 cfm of filtered air. Without supply and return data from AHU's 2, 3 and 5, it is impossible to tell from the data to what extent this is the case, which is a place future work can focus on. What can be said is that some energy balance equations in Chapter 4 showed that it simply is not possible for the data to be correct and 45,000 cfm to be flowing through AHU 1 into the cleanroom. Since the data is assumed to be correct, the flow rate is assumed to be wrong.

Another conclusion is to try and do a better job managing load on the chillers. It was seen in Section 4.5 that this may be causing an extra 30-50 kW of load. This may become even more important after switching over and perfecting the new setup, where only one chiller may be needed year round.

Chiller power is a function of outside air temperature. The degree to which this is the case is very surprising, since outside conditions should not affect indoor cleanroom load. It has been found that this is due to the decrease in efficiency of the chiller itself, since the condenser temperature and pressure will increase.

It is necessary to fix the situation where outside dew point over 50°F causes reheat to average about 200 kW. Why this is the case was shown to be unsure in Chapter 6, and this is an area of further research. It is also necessary to have the proper monitoring equipment to ensure that system changes have the desired affects, and do not have unintended side effects. For example, installing better controls, or at least monitoring equipment, on the reheats and chillers to alert plant personnel that reheat is on, or that the chillers are operating inefficiently would be a good step. This is especially important during non-routine operation of the plant, such as holidays and slow production.

Bibliography

[1] Duke Energy. Schedule OPT-I Optional Power Service. 1 Jan. 2009. 15 Apr. 2009 <http://www.duke-energy.com/pdfs/NCScheduleOPTI.pdf>.

[2] Wang, Shan K. Handbook of Air Conditioning and Refrigeration. New York, NY: McGraw-Hill, 2001.

[3] Trane. Air-Cooled Series R Rotary Liquid Chiller. Nov. 2006. 15 Apr. 2009 <http://www.trane.com/Commercial/Uploads/Pdf/1056/rlc-prc006-en_1106.pdf>.

[4] Whyte, W. Cleanroom Design. West Sussex, England: John Wiley & Sons Ltd, 1991.

[5] McQuiston, Faye C., Jerald D. Parker, and Jeffery D. Soitler. Heating, Ventilating, and Air Conditioning: Analysis and Design. Hoboken, NJ: John Wliey & Sons, Inc, 2005.

Appendix

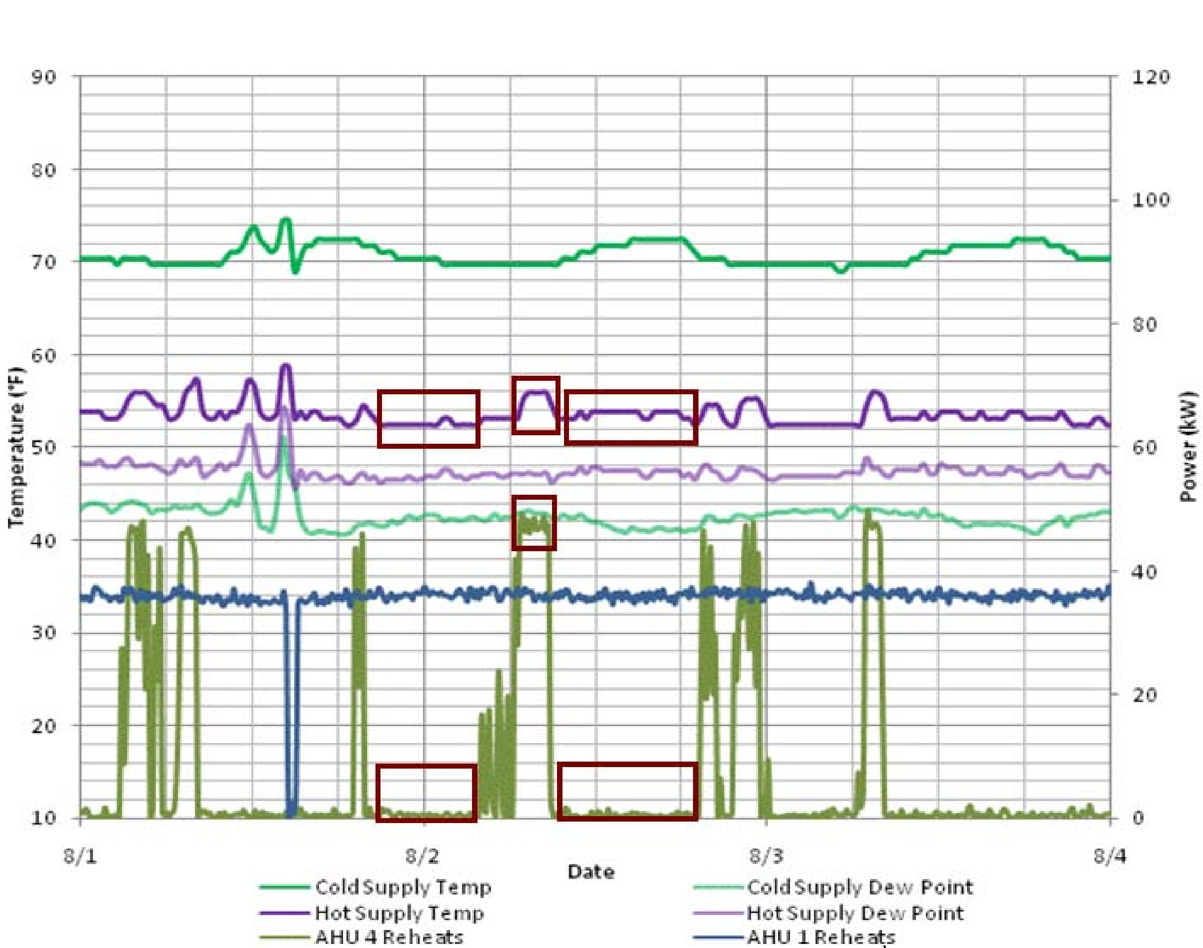

Figure A.1: Energy Balance on August 2nd

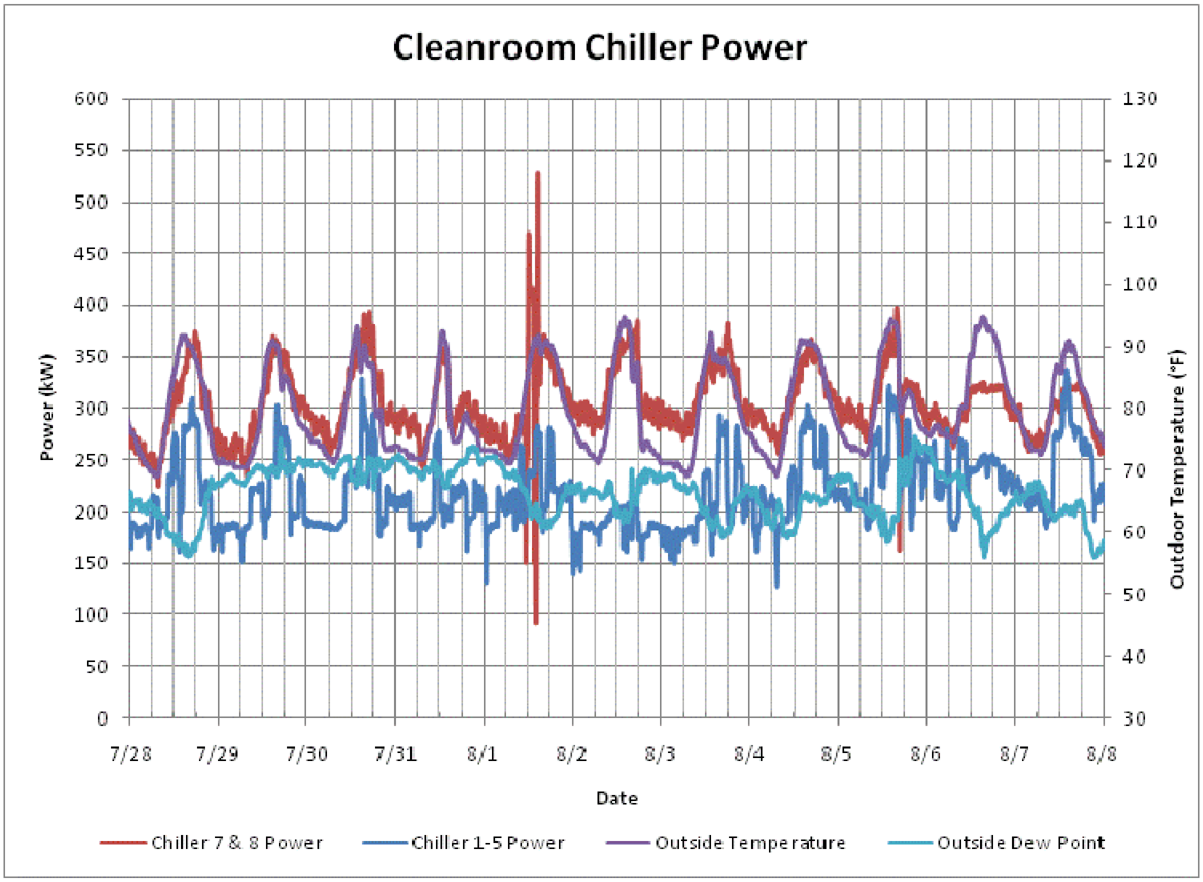

Figure A.2: Two Week Cleanroom Chiller Power, August

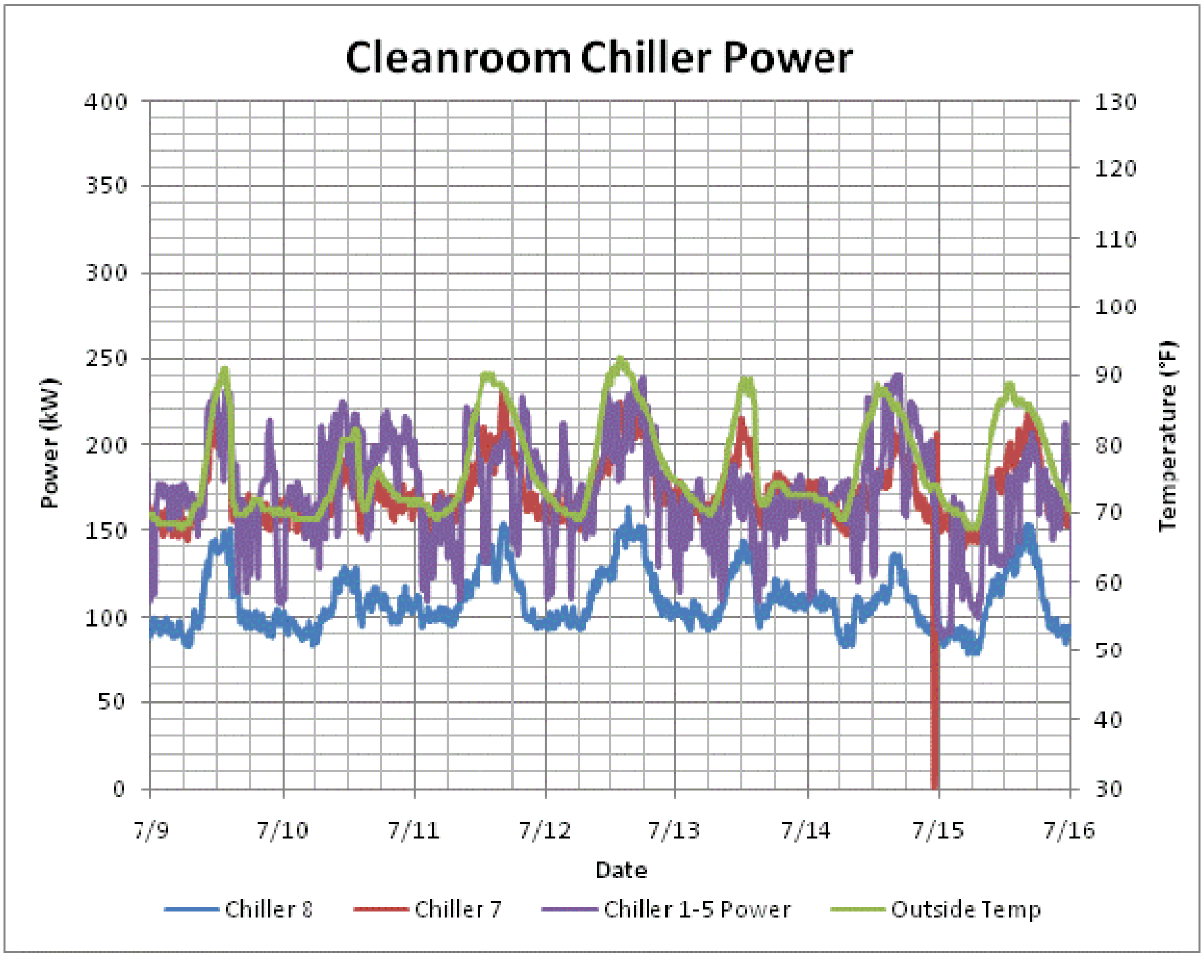

Figure A.3: Cleanroom Chiller Power, July 9-16

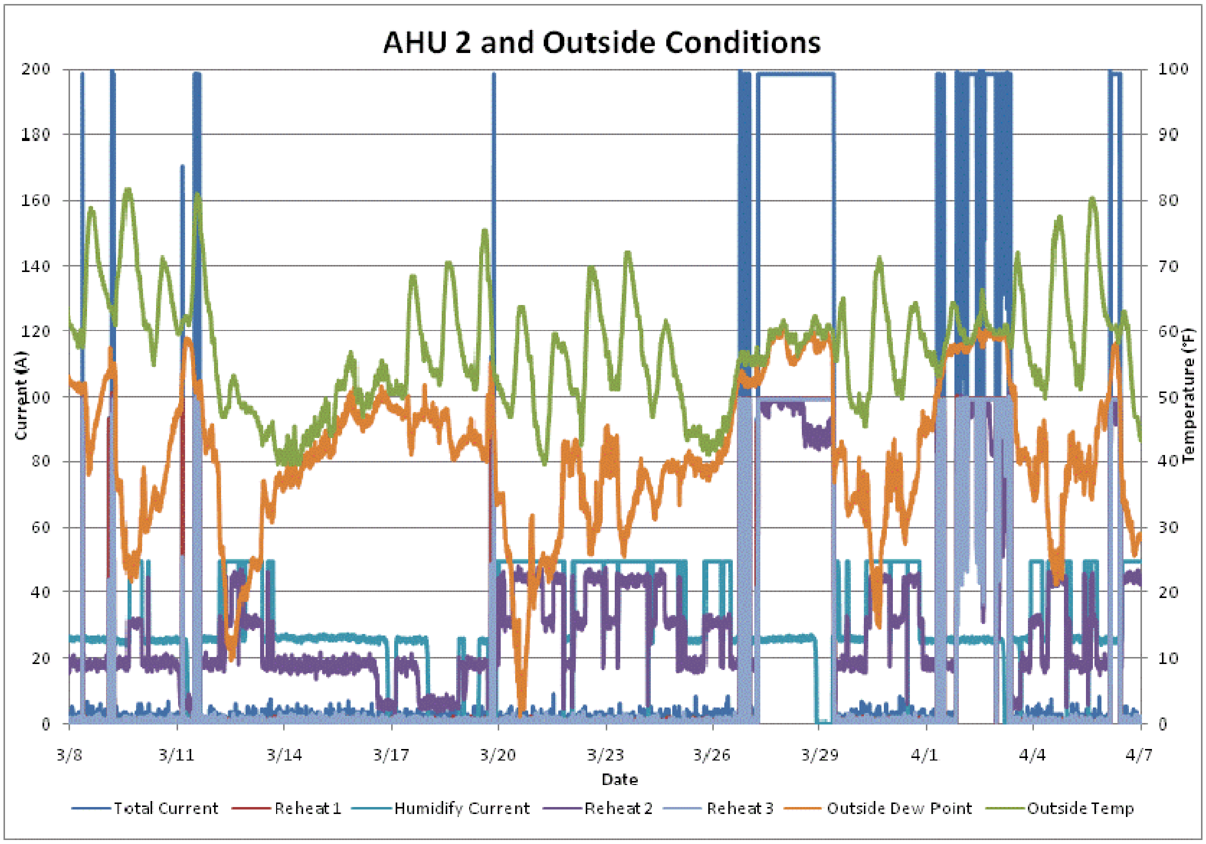

Figure A.4: AHU 2 Current and Outside Conditions

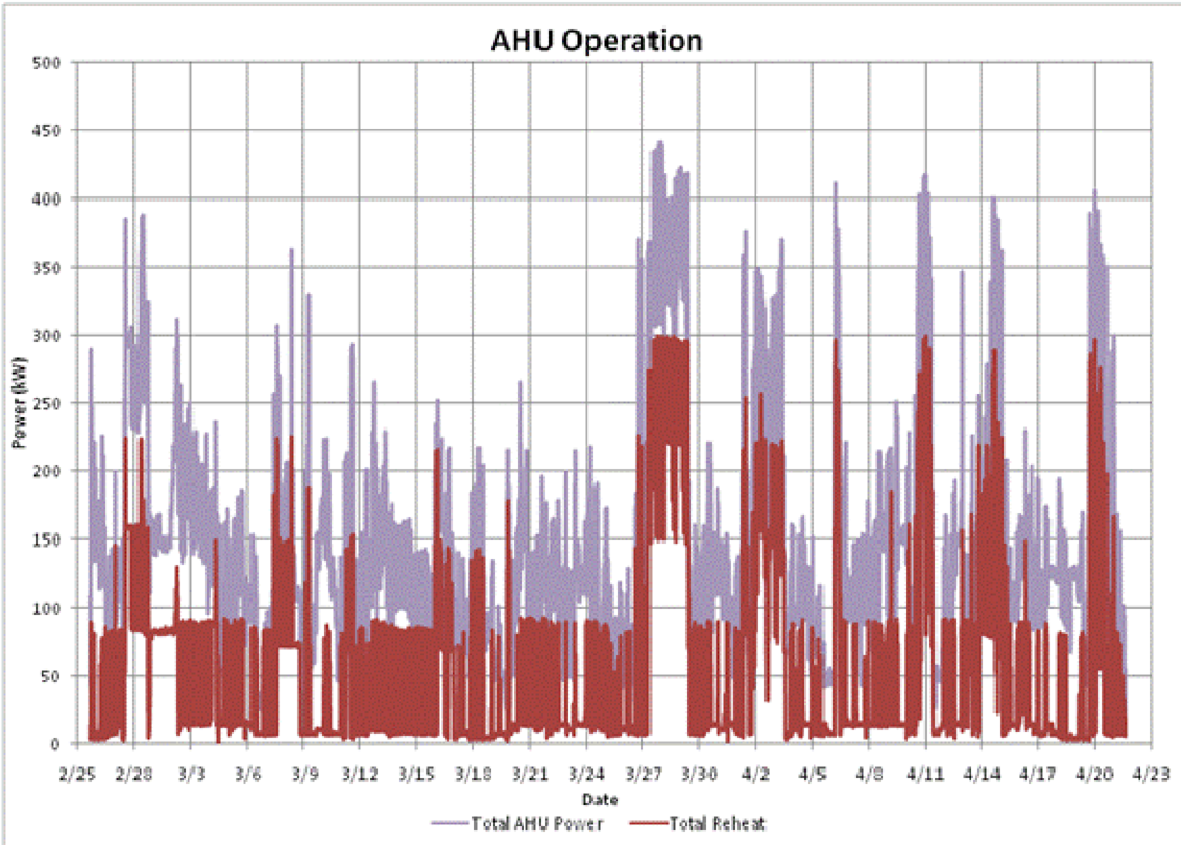

Figure A.5: AHU Operation 2/25-4/23