Pacific Gas and Electric Company

Emerging Technologies Program

Application Assessment Report ________

Data Center Air Management Report

Lawrence Berkeley National Laboratory
National Energy Research Scientific Computing Center
Oakland, CA

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Executive Summary

The purpose of this study is to demonstrate to data center designers and operators the operational and energy benefits of thoughtful and efficient data center air management techniques. To do so, a baseline air distribution configuration that is representative of typical data centers today, was compared against two alternate configurations. In the baseline configuration, supply air enters the cold aisle through perforated tiles in a raised floor, passes through the server intakes, and is ejected to the hot aisle where, through stratification, it returns to the cooling unit. The 'Alternate 1' configuration is similar, except that the cold aisle is sealed at the top and ends of the racks to prevent hot air recirculating back into the cold aisle (Figure 1, below).

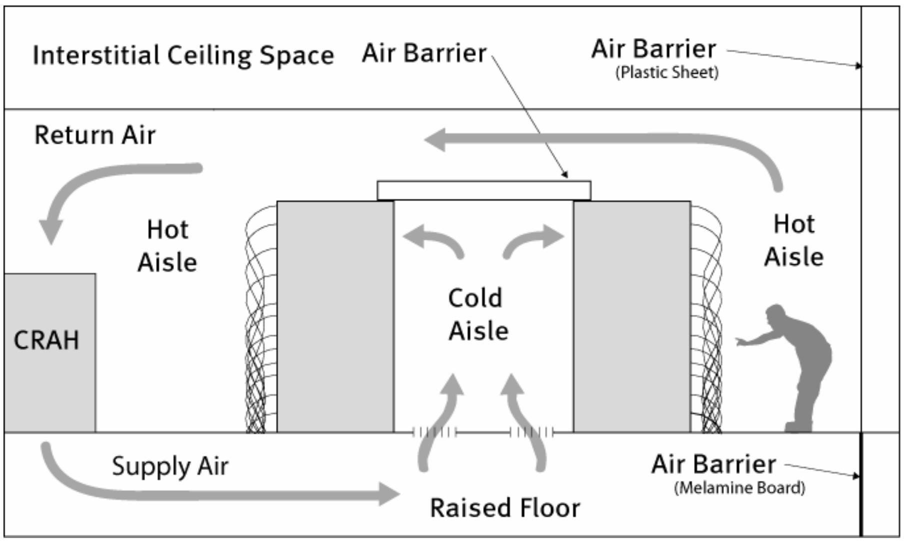


Figure 1: Alternate 1 - Cold Aisle Isolation

In the 'Alternate 2' configuration, plastic partitions sealed the hot aisles off from the cold aisle, and the interstitial ceiling space was used as a return air plenum.

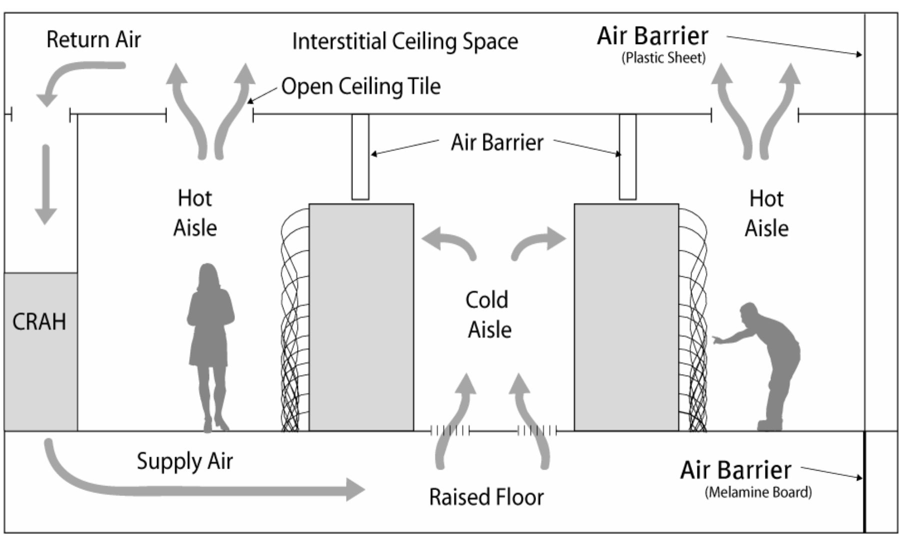


Figure 2: Alternate 2 - Hot Aisle Isolation

Each configuration was monitored for energy use, computer inlet and aisle temperatures, and air flow volume. These values have been compared to determine cooling effectiveness and efficiency. From this comparison, Alternate 1 was found to provide the most effective hot and cold air separation, and therefore the greatest cooling efficiency. Following is a summary of the findings of the study, which are discussed in detail in subsequent sections of this report:

Finding 1: Careful separation of hot and cold aisles allowed for better uniformity of the supply air temperature at the inlets to the servers. The variation in supply air temperature decreased from 26°F to 18.5°F between Baseline and Alternate 1. That is, the homogeneity of supply air temperatures improved as a result of better air management.

Finding 2: By minimizing air mixing, and allowing the temperature difference between the supply and return air to increase, fan power was able to be reduced by 75% without an adverse effect on the server environmental conditions.

Finding 3: Higher return air temperatures increased the capacity of the computer room air handling (CRAH) units by 30% - 49%

Finding 4: The temperature difference between the chilled water supply and return increased, which could allow for reduced chilled water flow, saving on pumping energy. Alternatively, a chilled water temperature differential of 20°F provides a 43% increase in piping capacity (but not chiller capacity) without modifying the chilled water system when compared to the as-found differential of 14°F.

Finding 5: When using variable speed server fans, power savings can be offset or decline to zero if the CRAH fan speed is reduced too low since the server fans speed up in response to the increased supply air temperature, which can increase overall data center power consumption.

Finding 6: By minimizing the mixing of supply and return air, the supply air and chilled water supply temperatures can be raised. Raising the chilled water supply temperature to 50°F can save 4% in chiller energy.

Finding 7: Raising the chilled water supply temperature to 50°F and utilizing integrated water-side economizing can save 21% in chiller energy, and result in increased chiller capacity.

For each of these findings, and where applicable, this report presents calculated energy savings and/or equipment capacity increases in the test area. Additionally, these values have been extrapolated to estimate the savings possible in applying the techniques presented here to the entire data center. This extrapolation was performed by multiplying the computer heat load density in the test area to the entire floor area of the data center. This load was then used to estimate cooling plant energy use. See Appendix II for detailed calculations and assumptions.

Summary of Potential Energy Savings for Entire Data Center

Finding #	*Annual Energy Savings (kWh)*	*Coincident Demand Reduction (kW)*	*Non-coincident Demand Reduction (kW)*	*% Reduction in Cooling Energy Use*	*% Reduction in Total Energy Use*
1	N/A				
2	740,000	84.2	84.2	12%	1.8%
3	N/A				
4	110,000	12.2	12.2	3.2%	0.3%
5	N/A				
6	210,000	24.5	24.5	3.3%	0.5%
7	1,140,000	24.5	24.5	18%	2.8%

Findings marked with "N/A" either increase capacity rather than save energy, or are qualitative observations.

It is instructive to note that the electrical load of the computing equipment in the data center far outweighs the electrical load of the cooling equipment. The ratio of cooling power to the total electrical power in a facility is a commonly-used metric in data centers. The test area, as found, was using approximately 63 kW of cooling power to cool approximately 270 kW of computing load. Thus, the ratio of cooling power to total power is approximately 20%. Therefore, dramatic reductions in cooling power required will not have as dramatic an effect on the overall facility's power use as a whole. This is illustrated by the last two columns in the table above.

Background & Objective

As part of Pacific Gas and Electric's (PG&E) Data Center Emerging Technologies Program, Rumsey Engineers was asked to conduct an air management demonstration and monitoring project in an operating data center, the goal of which is to demonstrate to data center designers and operators the operational and energy benefits of thoughtful and efficient data center air management techniques. Currently, data centers typically do not optimize air distribution when designing the floor layout resulting in increased energy use. Some of the methods presented in this study are currently in use in industry; however their application is inconsistent.

The demonstration took place in a small section of the National Energy Research Scientific Computing (NERSC) Center in downtown Oakland, California from June 12th – June 16th, 2006.

Demonstration Design & Procedure

An area containing two rows of server racks was selected because of its high heat density (approximately 175 W/sf) and ability to be isolated from the remainder of the facility. Cooling equipment serving the demonstration area consists of three 40-ton (nominal) downflow CRAH units with chilled water cooling coils. The chilled water is provided by a central chilled water plant, and the supply air is delivered via a 36" raised floor plenum. In order to maintain a constant chilled water flow for the purposes of the demonstration, the CRAH unit return air temperature setpoints were set at 55°F, a value that would ensure that the chilled water valves would remain wide open and not modulate the flow.

The rows of racks are separated into hot and cold aisles, and the return air is not ducted, but relies upon stratification to collect heat at ceiling level, which eventually makes its way back to the air intakes at the top of the CRAH units. The spaces between each of the computer racks are well sealed using Plexiglas-type material to prevent mixing of hot and cold air. These conditions served as the baseline configuration for this air management study, and are illustrated in Figure 3.

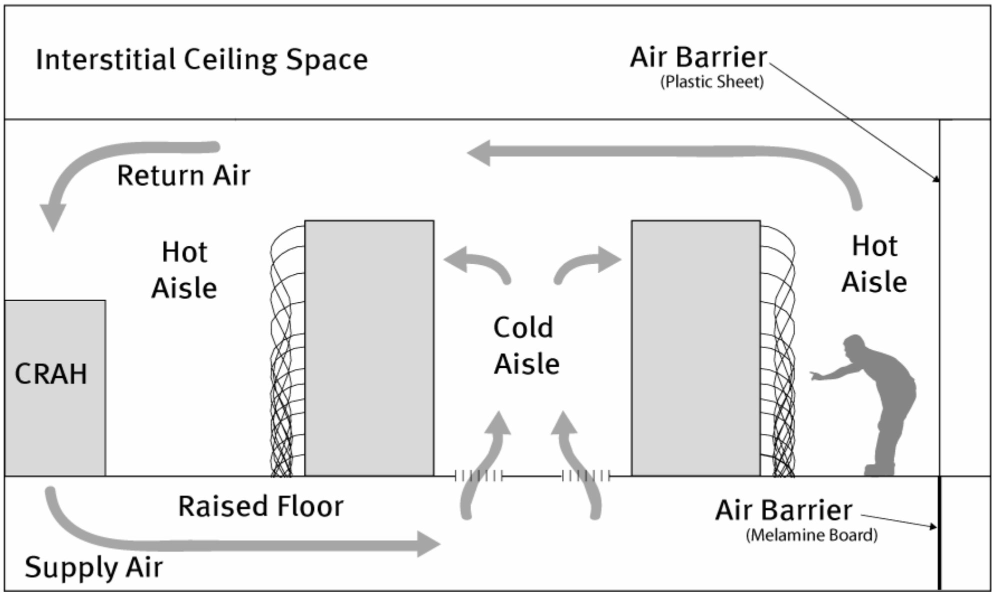


Figure 3: Baseline Air Flow Configuration

The demonstration and study looked at two improved air flow configurations to be compared to the baseline. The first, referred to here as "Alternate 1," involved sealing the cold aisle (top and sides) to minimize mixing of hot and cold air, and to more efficiently deliver the cooling air to the servers. The "cold aisle seal" was constructed of lightweight plywood and plastic sheeting, and is illustrated in Figure 1. Photographs are included in the Appendix.

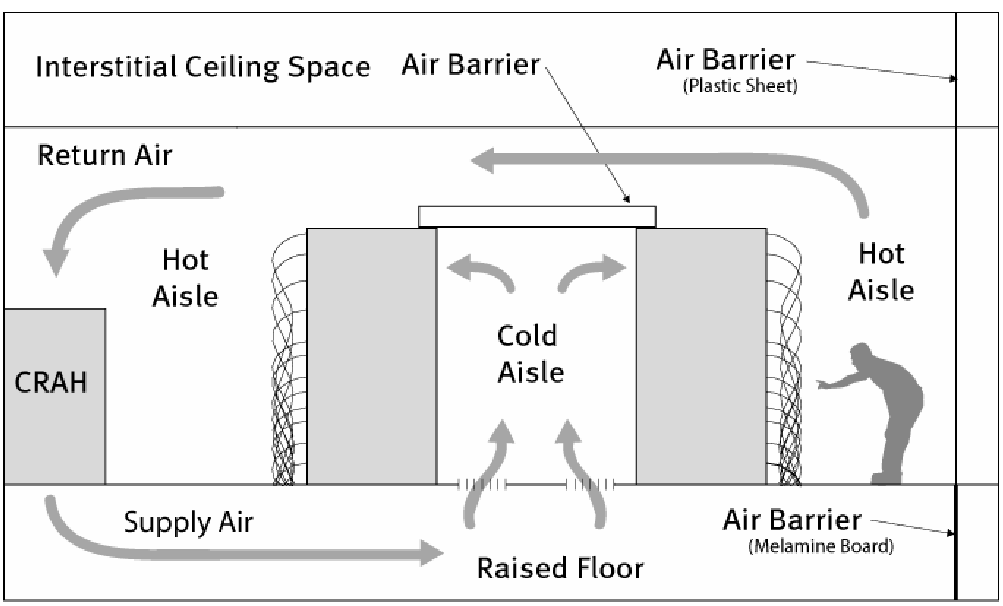


Figure 1: Alternate 1 - Cold Aisle Isolation
(Figure repeated from Page 3 for ease of comparison.)

The second revised air flow configuration, referred to as “Alternate 2,” provided a simulated plenum return coupled with rack top-to-ceiling air barriers to help minimize mixing of hot and cold air. The plenum return was accomplished by building intake hoods for the CRAH units, and utilizing the interstitial ceiling space as a return path, as illustrated in Figure 2.

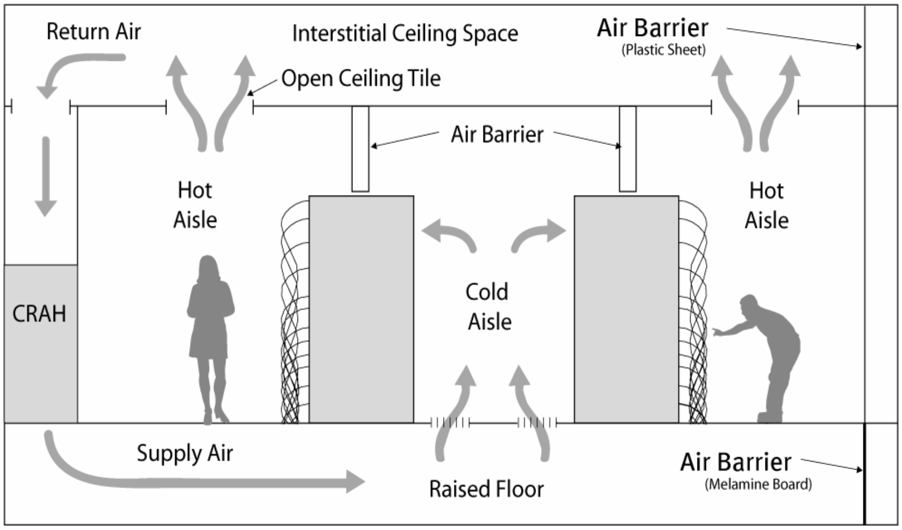


Figure 2: Alternate 2 - Hot Aisle Isolation
(Figure repeated from Page 3 for ease of comparison.)

Special note: After completing Alternate 1, it was noted that an experimental test rack in the project area began to overheat during the night-time portion of the experiment. The rack’s air inlet was located in the second hot aisle (the right side of Figure 3). To prevent further overheating, the rack’s inlet was covered with a tent that ducted air from the supply grille in front of the rack directly to the inlets. To avoid this issue reoccurring, the experiment run time was reduced during Alternate 2. Due to the large air change rate during the experiment, temperatures stabilized quickly during Alternate 2, allowing useful data to be collected.

The following measurements were taken during the baseline and subsequent two alternate configurations:

- IT equipment power
- Temperature distribution throughout the space
- CRAH unit(s) supply air temperature
- CRAH unit(s) return air temperature
- Chilled water supply temperature
- Chilled water return temperature
- CRAH fan energy
- Supply and return air volumetric flow rates

Findings

Finding 1: *Careful separation of hot and cold aisles allowed for increased uniformity of the supply air temperature at the inlets to the servers. The variation in supply air temperature decreased from 26°F to 18.5°F between Baseline and Alternate 1. That is, the homogeneity of supply air temperatures improved as a result of better air management.*

Savings Estimates

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh)	*N/A*	*N/A*
Coincident Demand Reduction (kW)	*N/A*	*N/A*
Non-coincident Demand Reduction (kW)	*N/A*	*N/A*
% Reduction in Total Energy Use	*N/A*	*N/A*

Discussion

In 2004, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) produced a document entitled, "Thermal Guidelines for Data Processing Environments." Included in this guideline are allowable and recommended temperature ranges for the inlets to server equipment similar to that seen in the air management demonstration facility. This publication cites an allowable range of 59°F to 90°F, and a recommended operating range of 68°F to 77°F server inlet temperature. Additionally, it presents a recommended range for relative humidity of 40% to 55% RH, and an allowable range of 20% to 80% RH.

While the layout of hot and cold aisles has gained widespread acceptance among data center owners, operators, and designers, there are still additional steps that can be taken to further prevent the undesirable mixing of the hot and cold air. A common occurrence in unsealed hot aisle/cold aisle configurations with under floor supply air (i.e. the baseline configuration) is that servers near the bottom of the racks tend to receive supply air at sufficiently low (or too low) temperature, while servers in the upper reaches of the same rack may receive supply air at a more elevated temperature. The servers at or near the top of the rack may have a shorter service life for that very reason (the supply air is too warm). The broad disparity of temperatures generally results from the mixing of hot and cold air at the tops of the racks, and at the ends of the rows.

By making an effort to segregate the hot and cold aisles as much as possible, this mixing can be significantly reduced, resulting in a decrease in the difference between supply air temperature at low and high level within the cold aisle. This is illustrated in the following set of temperature maps.

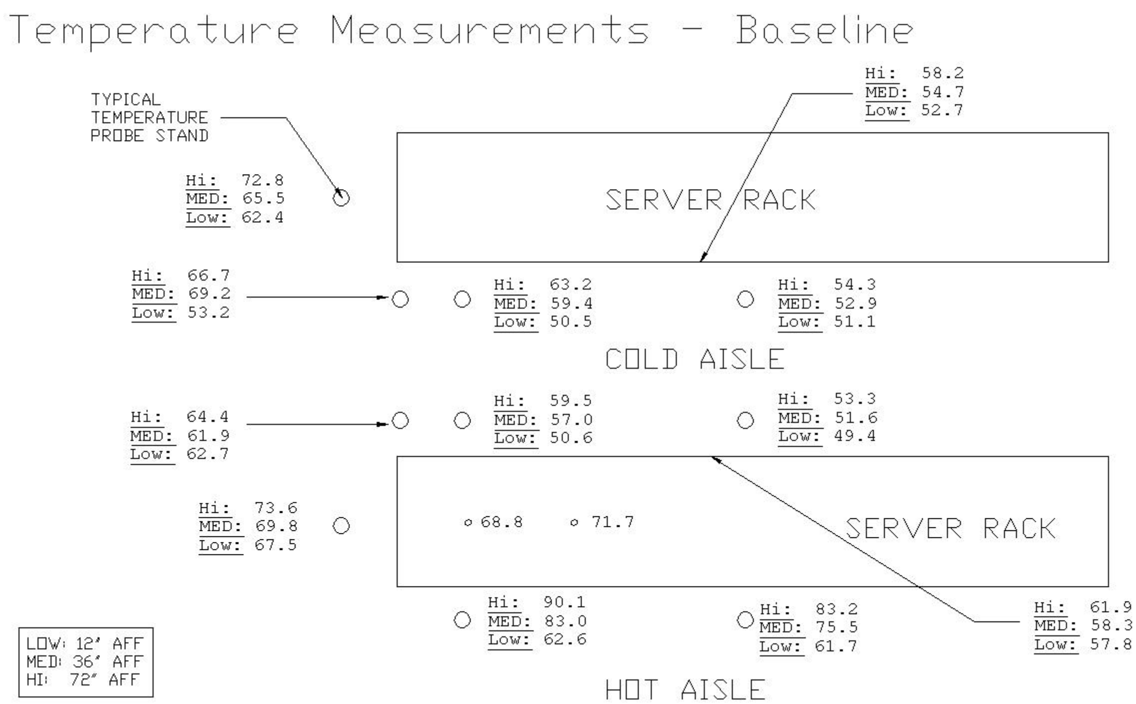


Figure 4: Average temperature measurements for the baseline configuration

The temperature map above (of the baseline conditions) shows widespread disparities in cold aisle temperatures. Note particularly the large differences in temperature between the lower level sensors and the higher level sensors. Also noteworthy is the difference in temperatures at the ends of the racks as opposed to the middle of the racks. At these locations, hot return air recirculates back into the cold aisle instead of returning to the CRAH units. It is also notable that the supply air temperatures to the servers are significantly below the ASHRAE recommended range <u>and in most cases, below the allowable range</u>. The supply air temperatures are likely lower than necessary in order to deal with the local hot spots in the cold aisle created by the mixing of hot and cold air.

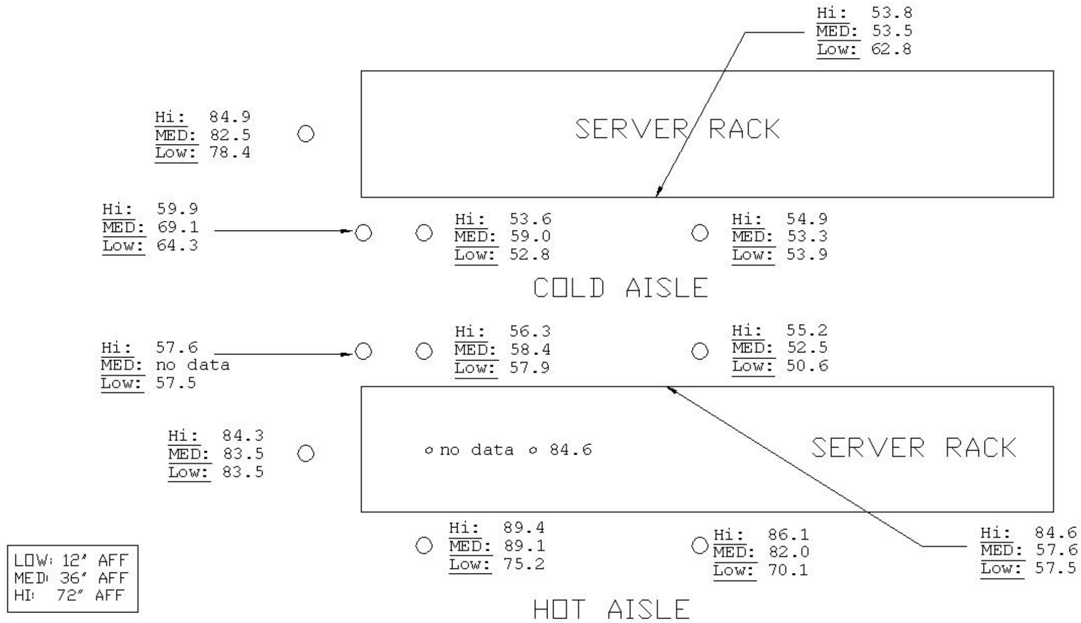


Figure 5: Average temperature measurements for the Alternate 1 configuration

Figure 5 shows the Alternate 1 case with the CRAH units set to a fan speed of 36 Hz using onboard variable frequency drives (VFDs), after the temperatures have reached steady-state. The temperatures in the cold aisle are more homogeneous, and the supply air temperatures near the end of the aisle are reduced.

In both the baseline and Alternate 1 cases, the supply inlet average temperatures are below the ASHRAE allowable level of 59°F. This is due to a very low under floor supply air temperature. Operational and energy savings can result from raising the overall supply air temperature to be more in line with the ASHRAE allowable range. But this is more difficult to achieve in the baseline configuration due to the wider disparity of supply air temperatures resulting from the mixing of hot and cold air. In other words, the supply air temperature needs to be kept lower than necessary in order to make up for the mixing that occurs at the tops and ends of the racks.

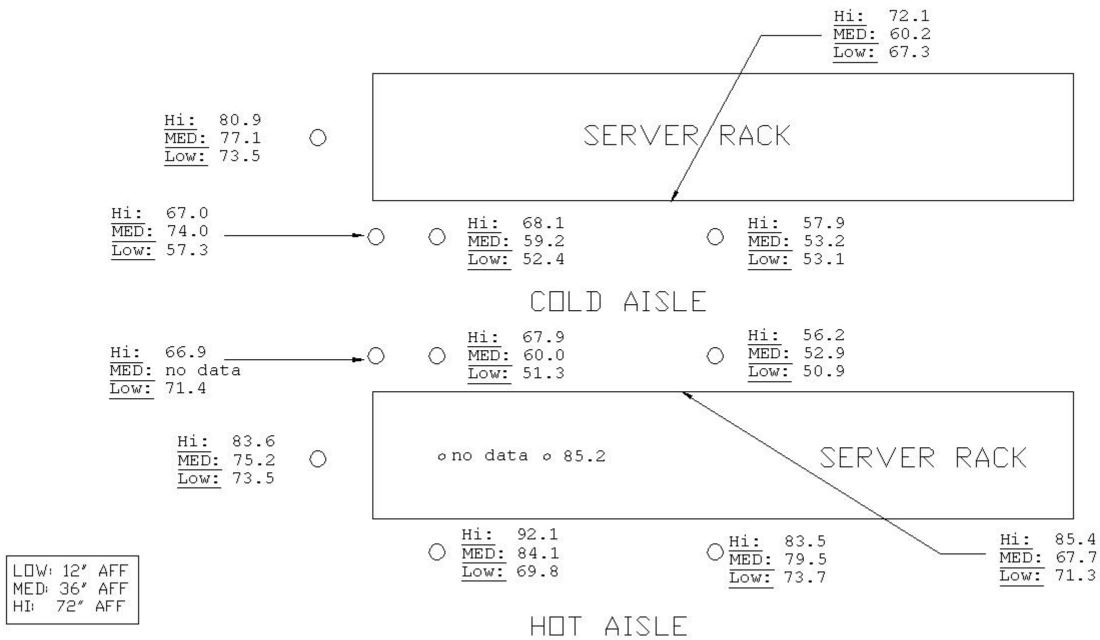


Figure 6: Average temperature measurements for the Alternate 2 configuration

As seen in Figure 6, Alternate 2 (with the CRAH VFDs set to 40 Hz and temperatures allowed to reach steady-state) displays more stratification than Alternate 1, and a nearly identical stratification as the baseline (for a statistical analysis of the supply air temperatures for all three configurations, see Appendix I). The more pronounced cold aisle air stratification resulted from the removal of the seal from the top and sides of the cold aisle. Additionally, the supply air temperatures are higher, and more in line with the ASHRAE recommendations. However, throughout both Alternatives 1 and 2, air infiltration through the test area isolation tent resulted in unexpected temperature anomalies, especially at the ends of the cold aisle.

Furthermore, it was noted that CRAH units 23 and 25 displayed readings of 17% and 83% RH, respectively, even though they are located within a few feet of one another. While humidity is not controlled in this particular facility, these readings present an excellent example of the potential for sensors to go out of calibration, especially humidity sensors. If humidity were to be actively controlled in this data center, these two units would fight each other, resulting in a tremendous amount of wasted energy.

Finding 2: *By minimizing air mixing, and allowing the temperature difference between the supply and return air to increase, fan power was able to be reduced by 75% without an adverse effect on the server environmental conditions.*

Savings Estimates

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	*117,000*	740,000
Coincident Demand Reduction (kW)	*13.3*	84.2
Non-coincident Demand Reduction (kW)	*13.3*	84.2
% Reduction in Total Energy Use	*N/A*	1.8%

Discussion

The efficient separation of hot and cold aisles results in an increase in the temperature difference ("Delta-T") between the supply and return air, which in turn allows one to serve the same cooling load with less air flow. This can be seen in equation 1,

$$Q = 1.08 \times CFM \times (RAT - SAT) \qquad \text{(eq. 1)}$$

where Q = cooling load in Btu/hr
CFM = volumetric flow rate of air in cubic feet per minute
SAT = supply air temperature in degrees Fahrenheit
RAT = return air temperature in degrees Fahrenheit
1.08 = constants and conversion factors for Standard Air (a reasonable approximation for the air temperatures and humidities encountered in this study).

Equation 1 shows that if you increase the Delta-T (RAT – SAT), and the load remains constant, then the required volumetric flow rate of air, or CFM, is reduced. This has even further energy saving implications when you consider the fan "affinity law," which describes a cubic relationship between fan air flow and the fan power required:

$$HP_2 = HP_1 \times (RPM_2 / RPM_1)^3 \qquad \text{(eq. 2)}$$

where HP_1 = power required in horsepower for situation 1
HP_2 = power required in horsepower for situation 2
RPM_1 = rotational speed of fan in RPM for situation 1
RPM_2 = rotational speed of fan in RPM for situation 2

An inspection of equation 2 shows, for example, that by cutting the rotational speed of the fan by 50%, the power required to run the fan is not reduced by half, but by more than 87%. Small reductions in fan speed can result in large reductions in fan power. The equation assumes a system of turbulent flow through ducts, but is a reasonable approximation in this case.

For the baseline configuration, the CRAH fan VFDs were left at their as-found frequency of 60Hz (full speed). For each of the Alternate configurations 1 and 2, the fans were then manually turned down at the VFD until temperatures in the cold aisle reached their upper limit. For the purposes of this study, it was discussed ahead of time with facility personnel that the upper limit cold aisle temperature would be 78°F, which is consistent with ASHRAE recommendations. For Alternate 1, the fans were able to be turned down to a minimum of 20 Hz, or one-third of their full speed. Alternate 2 allowed the fans to be turned down to a minimum of 40 Hz, or two-thirds of their full speed. Figure 7 shows the resulting power reduction realized at the CRAH units.

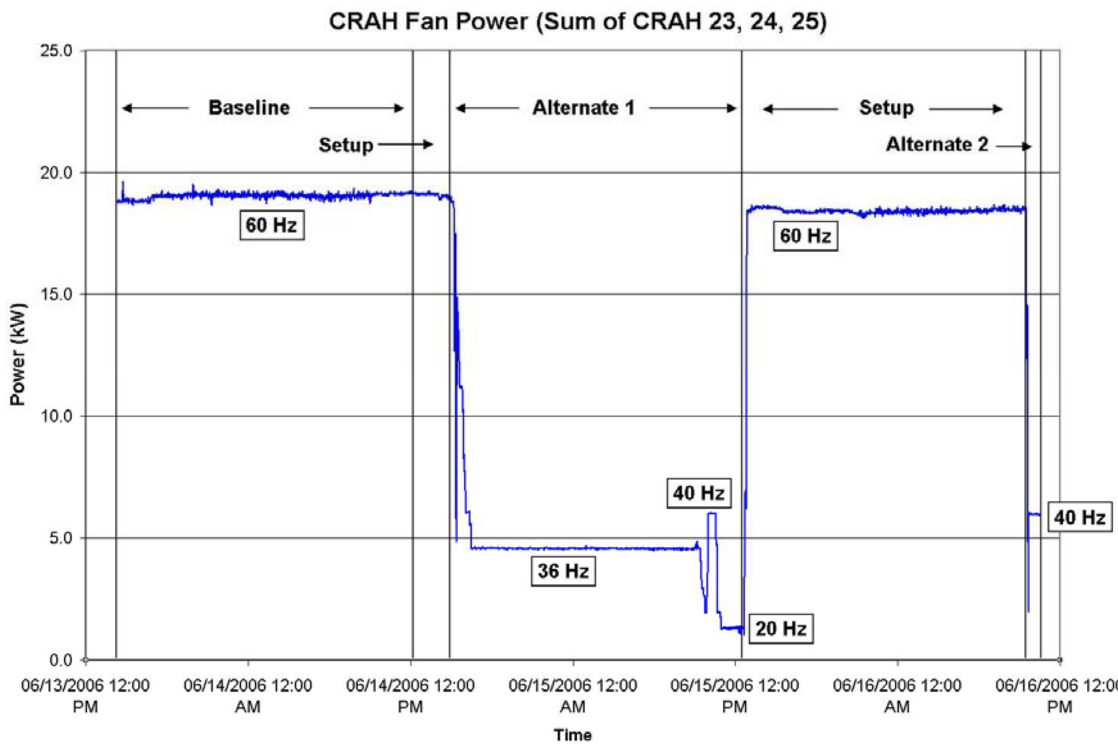


Figure 7: Computer Room Air Handler (CRAH) Fan Power

Figure 8 gives an example of the air temperature differences that allowed such power savings.

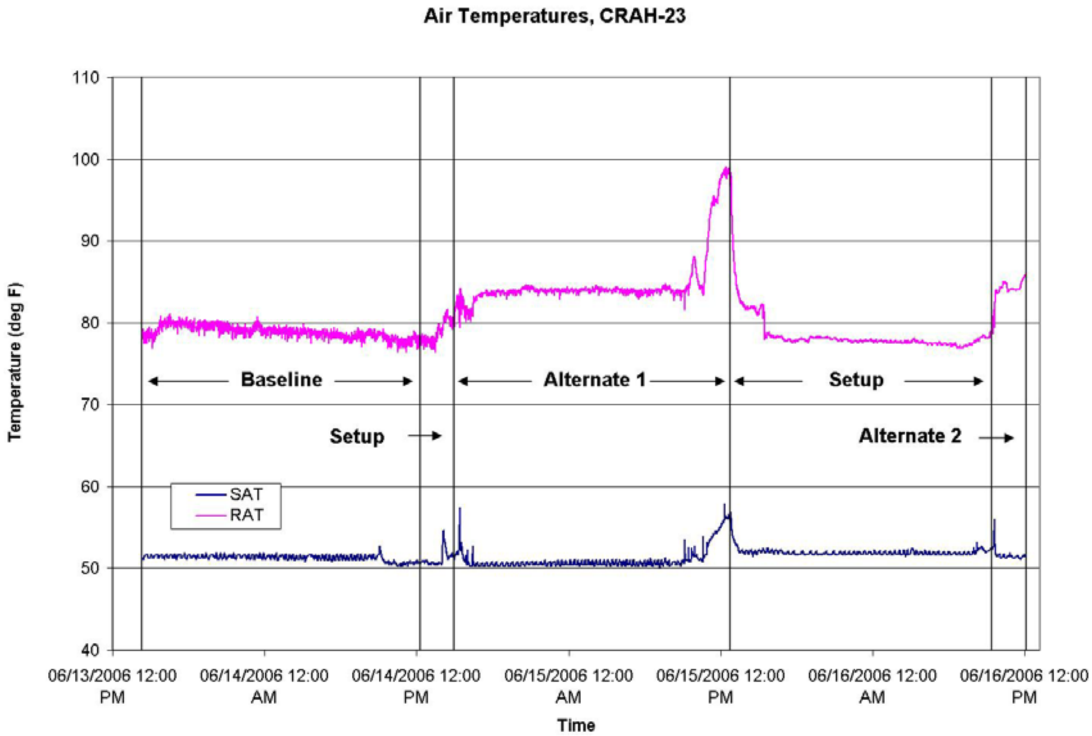


Figure 8: Supply and return temperatures for CRAH-23

It was observed during the Alternate 1 and 2 tests that the CRAH units were supplying a greater amount of air than was being returned. Because the supply fans were turned down, the cold aisle became negatively pressurized, which resulted in two unintended effects – infiltration from outside the test area, and recirculation. First, the negative pressure in the under floor supply plenum caused supply air from outside the test area to seep through the under floor barriers erected to isolate the test area. Secondly, exhausted air in the hot aisle recirculated back into the cold aisle through the plywood doors and floor tiles. However, recorded temperature data indicates that these effects did not greatly affect test results. For example, the difference between the supply and return air temperatures increased from an average of 20°F to 29°F between the Baseline and Alternate 2 cases. Assuming a constant load in the test area, the following equation applies:

$$\Delta T_{Baseline} * CFM_{Baseline} = \Delta T_{Alternate} * CFM_{Alternate} \quad \text{(eq. 3)}$$

In this case, the ΔT increased by 20/29 = 69%, which implies the air flow could be reduced by an equivalent amount. The fans in the Alternate 2 case were turned down to 40 Hz, or 67%. Therefore, the fans were turned down roughly 2% too far, with this small amount of extra cooling possible due to infiltration from outside the test area.

Finding 3: *Higher return air temperatures increased the capacity of the CRAH units by 30% - 49%*

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	*N/A*	*N/A*
Coincident Demand Reduction (kW)	*N/A*	*N/A*
Non-coincident Demand Reduction (kW)	*N/A*	*N/A*
% Reduction in Total Energy Use	*N/A*	*N/A*

Discussion

According to equation 1, by increasing the differential between the supply and return air temperatures, the CRAH fan speed can be reduced. Alternatively, increasing the temperature differential increases the cooling capacity of the same unit without increasing the fan speed. Table 1 below demonstrates the increase in capacity for a given temperature differential.

	Temperature Differential (°F)	Cooling Capacity (kBTUh)	% Increase over baseline
Baseline	19	410	0%
Alternative 1	32	606	49%
Alternative 2	27	529	30%

Table 1: Air temperature differentials and cooling capacities of CRAH units.

As can be seen, the cooling capacity of the CRAH units is increased the most in the Alternate 1 configuration. This is due to the superior isolation of the cold and hot aisles under this configuration.

Finding 4: *The temperature difference between the chilled water supply and return increased, which could allow for reduced chilled water flow, saving on pumping energy. Alternatively, a chilled water temperature differential of 20°F provides a 43% increase in piping capacity (but not chiller capacity) without modifying the chilled water system when compared to the as-found differential of 14°F.*

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	8,000	110,000
Coincident Demand Reduction (kW)	0.9	12.2
Non-coincident Demand Reduction (kW)	0.9	12.2
% Reduction in Total Energy Use	N/A	0.3%

Numbers above assume a chilled water temperature differential of 20°F

Discussion

The chilled water system in the data center facility is a central plant that serves all cooling in the data center, and not just the small section that was used for the air management demonstration. As such, the chilled water supply temperature could not be altered, and remained relatively constant throughout the demonstration period. But the chilled water return temperature from the CRAH units serving the demonstration area increased during Alternates 1 and 2, resulting in an increase in the overall temperature difference ("delta-T") between chilled water supply and return, as shown in Figure 9.

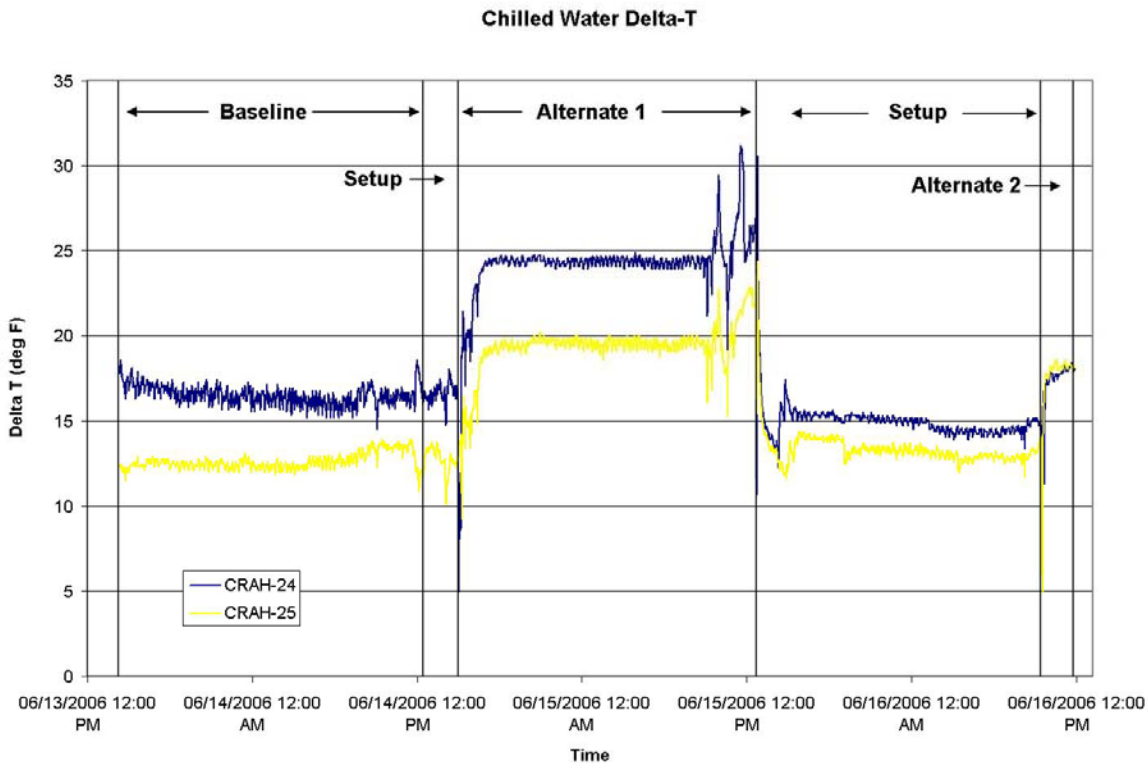


Figure 9: Chilled water temperature differentials for CRAH-24 and CRAH-25 (chilled water data for CRAH-23 was incomplete).

The increased chilled water delta-T has several heat transfer implications, and these can be seen by inspection of equation 3.

$$Q = 500 \times GPM \times (CHWRT - CHWST) \qquad \text{(eq. 4)}$$

where Q = cooling load in Btu/hr
GPM = volumetric flow rate of water in gallons per minute
CHWST = supply water temperature in degrees Fahrenheit
CHWRT = return water temperature in degrees Fahrenheit
500 = constants and conversion factors (8.33 lb/gal nominal water density * factor of 60 to account for gal/min and Btu/hr)
The density of water is directly proportional to temperature, but this effect is negligible within the range of temperatures presented here, hence the common value of 8.33 lb/gal (density water at 70 degF) is applied here.

The larger chilled water temperature difference at the coil in the CRAH allows for more heat transfer at a given chilled water flow rate. Therefore, for a given cooling load, the chilled water flow rate can be reduced, providing additional energy savings due to reduced pumping energy using a variable-flow system.

Alternatively, increasing the temperature differential between the chilled water supply and return temperatures increases the capacity of the cooling system without any modifications to piping or pumps. Again referencing equation 3, it can be seen that leaving the flow rate constant and increasing the chilled water temperature differential results in a linear increase in cooling capacity. Table 2 below presents estimates of the increase in cooling capacity at given chilled water temperature differentials.

Chilled Water Delta-T	% Pipe Capacity Increase	Test Area Capacity	Total Data Center Capacity
		tons	tons
15	7%	82	1,152
20	43%	110	1,535
25	79%	137	1,919

Table 2: Piping capacity increase at given temperature differentials.

Finding 5: *When using variable speed server fans, power savings could be offset or decline to zero if the CRAH fan speed is reduced too low since the server fans speed up in response to the increased supply air temperature.*

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	*N/A*	*N/A*
Coincident Demand Reduction (kW)	*N/A*	*N/A*
Non-coincident Demand Reduction (kW)	*N/A*	*N/A*
% Reduction in Total Energy use	*N/A*	*N/A*

Discussion

The server racks in the test area of this study are equipped with variable speed cooling fans built into the servers themselves. These fans speed up or down in response to changing environmental conditions. As seen in Figure 10, as the CRAH fans were slowed down; the supply air temperature began to rise.

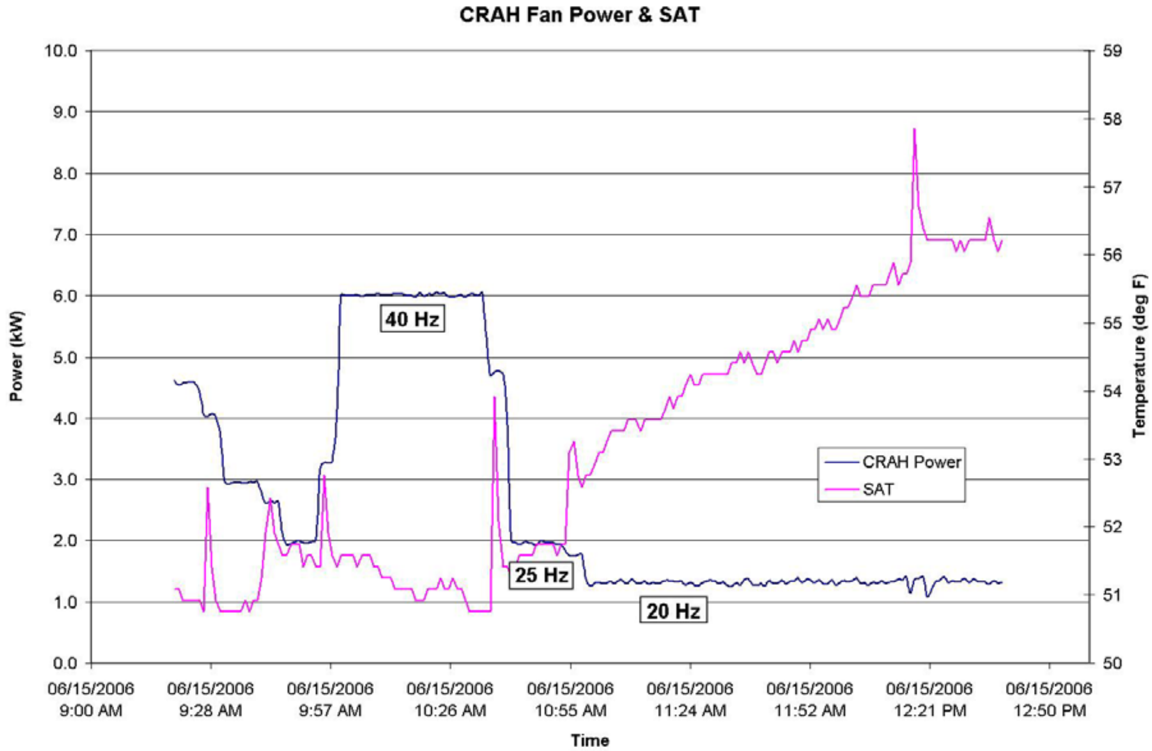


Figure 10: Supply air temperature increased due to the reduction in CRAH airflow.

In response, the onboard server fans sped up to compensate. Figure 11 shows that not only did the increase in the server fan power eliminate the CRAH fan power savings, the total energy use of the test area as a whole actually increased by 8%.

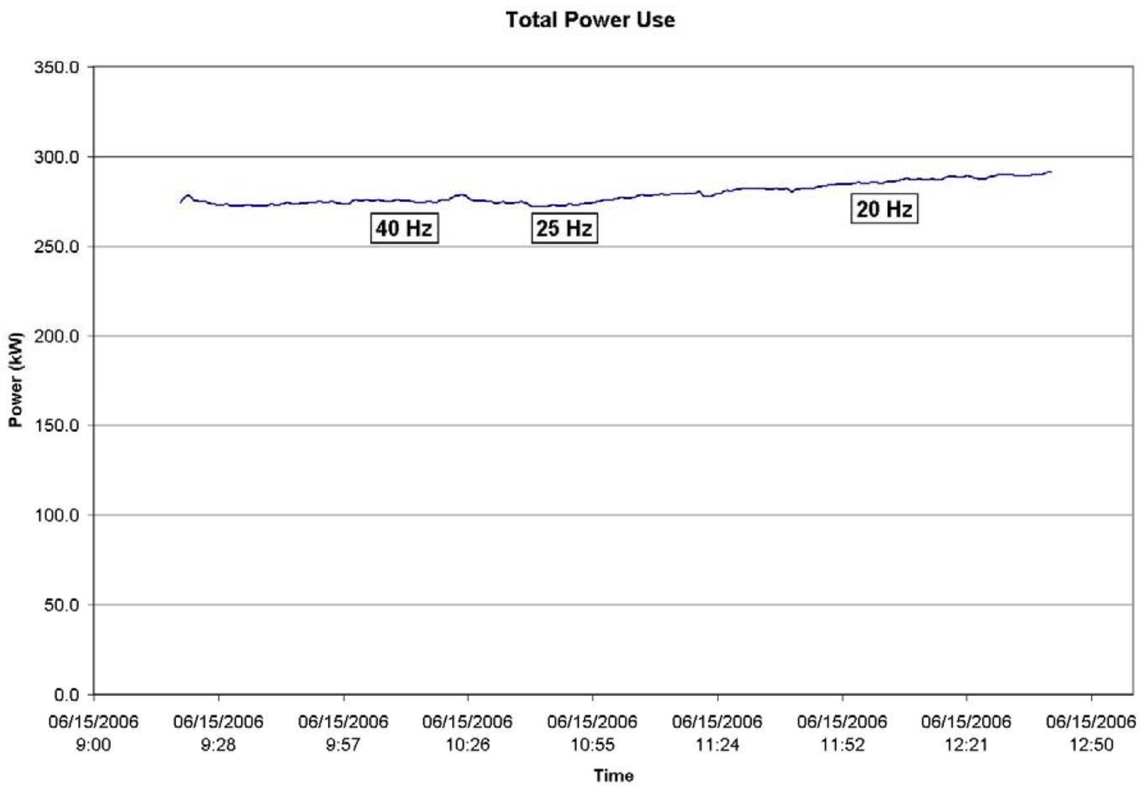


Figure 11: Total power use increases if the CRAH fan speed is too low

As can be seen above, it is important for data center operators to be aware of the airflow and temperature conditions that will cause internal server fans to speed up. Therefore, care must be taken to determine these conditions before slowing cooling fans down if energy savings are to be achieved.

Finding 6: *By preventing the mixing of supply and return air, the supply air and chilled water supply temperatures can be raised. Raising the chilled water supply temperature to 50°F can save 4% in chiller energy.*

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	15,000	210,000
Coincident Demand Reduction (kW)	1.8	24.5
Non-coincident Demand Reduction (kW)	1.8	24.5
% Reduction in Total Energy Use	N/A	0.5%

Discussion

As was discussed in Finding 1, properly isolating the supply and return aisles results in greater homogeneity in the supply air temperatures, especially at the ends of aisles. Because of this increase, it is no longer necessary to over-cool the supply air to insure that air at the top and ends of the server racks are at safe temperatures. This allows the supply air temperature to be safely raised, and therefore the chilled water supply temperature as well.

The higher the temperature of the chilled water, the more efficiently the chiller will operate. Typically, a chiller will operate 1% more efficiently for every 1°F increase in the chilled water supply temperature. However, care must be taken when increasing the supply temperature in existing facilities because of the cooling coils in the CRAH units. These coils are selected for a specific set of design parameters, including water temperature, water flow rate, air flow rate, and air temperature. Too great a change in any of these parameters will reduce the cooling capacity of the cooling coil, regardless of the operation of the chiller.

Additionally, to avoid extra pumping energy, the difference between the supply and return temperatures of the chilled water must be maintained. This is best achieved by maintaining a large temperature differential between the supply and return air temperatures, through such practices as described in Finding 3.

Finding 7: *Raising the chilled water supply temperature to 50°F and utilizing integrated water-side economizing can save 21% in chiller energy, and result in increased chiller capacity.*

	Test Area	Extrapolated to Entire Data Center
Annual Energy Savings (kWh/yr)	69,000	970,000
Coincident Demand Reduction (kW)	0	0
Non-coincident Demand Reduction (kW)	0	0
% Reduction in Total Energy Use	N/A	2.8%

Discussion

Utilizing water-side economizing, chilled water is produced by a cooling tower instead of a chiller, saving significant energy as a result of reduced compressor use. However, to utilize water-side economizing, several changes must be made to a typical data center (this facility included). First, the condenser water piping needs to be modified to interact with the chilled water. This can be done one of two ways, depending on the type of cooling tower installed. With an open loop cooling tower, the condenser water piping needs to be fitted with a heat exchanger because the contaminated condenser water cannot be used directly as chilled water (see figure 12, below). However, with a closed loop condenser water system, and a dedicated economizing tower, the condenser water piping can be fitted with three way valves to directly provide chilled water whenever possible. The water temperature produced by the cooling tower needs to be controlled using a reset method based on outside air conditions. This way the cooling tower is always producing water as cold as possible (as opposed to typical practice where the tower produces water at a constant temperature).

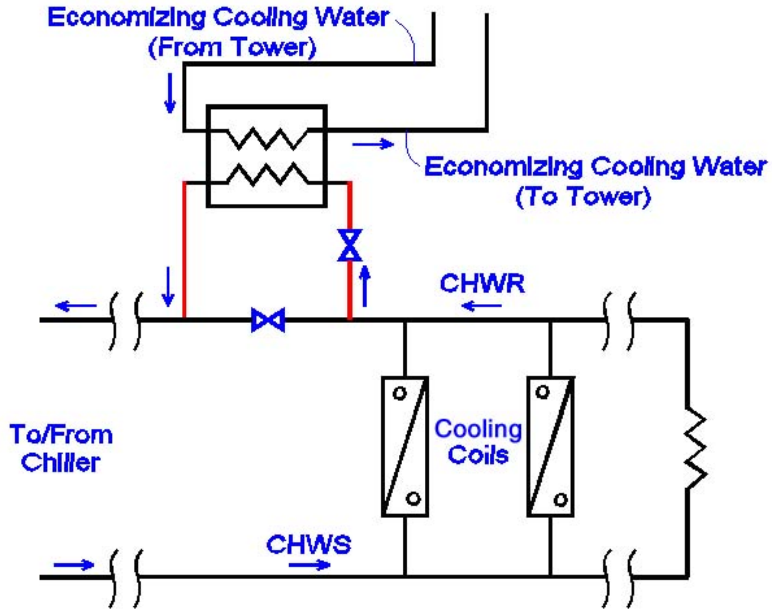


Figure 12: Typical water-side economizer with heat exchanger.

The term 'integrated' refers to a design where free cooling can be used to pre-cool the return chilled water prior to using a traditional chiller to provide additional cooling. This series piping arrangement, with the free cooling system first, can greatly extend the free cooling operating hours, simplify chiller staging, and increase overall system reliability. Lastly, the higher the chilled water temperature is raised, the more hours in a year water-side economizing can be used. By raising the chilled water supply temperature to 50°F, water-side economizing can be utilized (integrated or full-economizing) 3,800 hours in a typical year in the Oakland weather zone. The results in Table 2 below present savings based on several different chilled water supply temperatures.

	No Econo.	Chilled Water Supply Temp. (deg F)				
		46	48	50	52	54
Hours in a Year Available for Water Side Economizing	N/A	1,976	2,807	3,819	4,875	5,994
Estimated Yearly Energy Use (kWh)	5,400,000	4,900,000	4,600,000	4,200,000	3,800,000	3,200,000
% Energy Savings	0%	8%	14%	21%	30%	40%

Table 3: Estimated energy savings with water-side economizing at various chilled water temperatures. See Appendix for calculation details.

Additionally, raising the chilled water supply temperature during chiller operation, while holding the condensing temperature constant, results in an increase in chiller capacity as well as efficiency. This is because the compressor 'lift' is decreased as a result of the decrease in the difference between the chilled water supply temperature and the chiller's condensing temperature. As a general rule-of-thumb, one can expect approximately 1% increase in chiller capacity for every 1°F decrease in this temperature differential. Therefore, by holding all other parameters constant, raising the chilled water supply temperature from 44°F to

$50^{o}F$ could increase the capacity of an 800-ton centrifugal chiller to nearly 850 tons. This was verified using chiller selection software.

Conclusion

Properly isolating supply and return air streams achieves several goals for the data center operator. First, this minimizes temperature stratification in the cold aisle, thereby potentially increasing the life of computer equipment placed in the upper reaches of a rack. Second, increasing the differential temperature between the supply and return air flows increases the heat transfer capacities of CRAH units. By reducing mixing of hot and cold air, the CRAH fan speed can be reduced to deliver the same amount of cooling previously available, or the operator can realize a capacity increase without purchasing additional units. Similarly, the chilled water pumps can be slowed down, yet deliver the same amount of cooling or deliver greater cooling with the same piping system. Third, in variable speed server fan applications, it is important to be aware of the server fan speed when implementing these changes, as too dramatic a decrease in CRAH fan speed can negate savings because the server fans speed up in response. Fourth, raising the chilled water temperature can increase the efficiency of the chiller. Fifth, implementing water-side economizing in a data center, while simultaneously raising the chilled water temperature, can save a large proportion of chiller energy use in a given year. Lastly, the proper isolation of hot and cold air allows for higher supply air temperatures due to the decreased mixing and stratification within the cold aisle, enabling a more homogeneous cold aisle temperature distribution that can be within the ASHRAE recommended range. Often this could result in raising the temperature of the air delivered by the CRAH units.

Recommendations for Future Work

These results demonstrate the energy savings and/or cooling capacity increase possible through proper air flow management techniques. However, it is recommended that further studies be conducted to investigate and quantify additional gains.

First, it is important that this type of study be conducted on an entire data center. This will allow a more thorough analysis of energy savings on the cooling plant and fan units, instead of performing an extrapolation estimate. Additionally, further work needs to be conducted to quantify savings from completely sealing floor openings and gaps between the server racks, because cold supply air slips out of these openings.

Finally, further education is needed regarding appropriate supply and return air temperatures. Many IT and facility managers prefer to supply air below ASHRAE recommended levels; they need to be made aware of the energy and capacity losses when using over-cooled air. Also, these managers should be educated on the benefits of using a high air-side temperature differential, and the strategies to achieve it.

Appendix

Appendix I – Collected Data

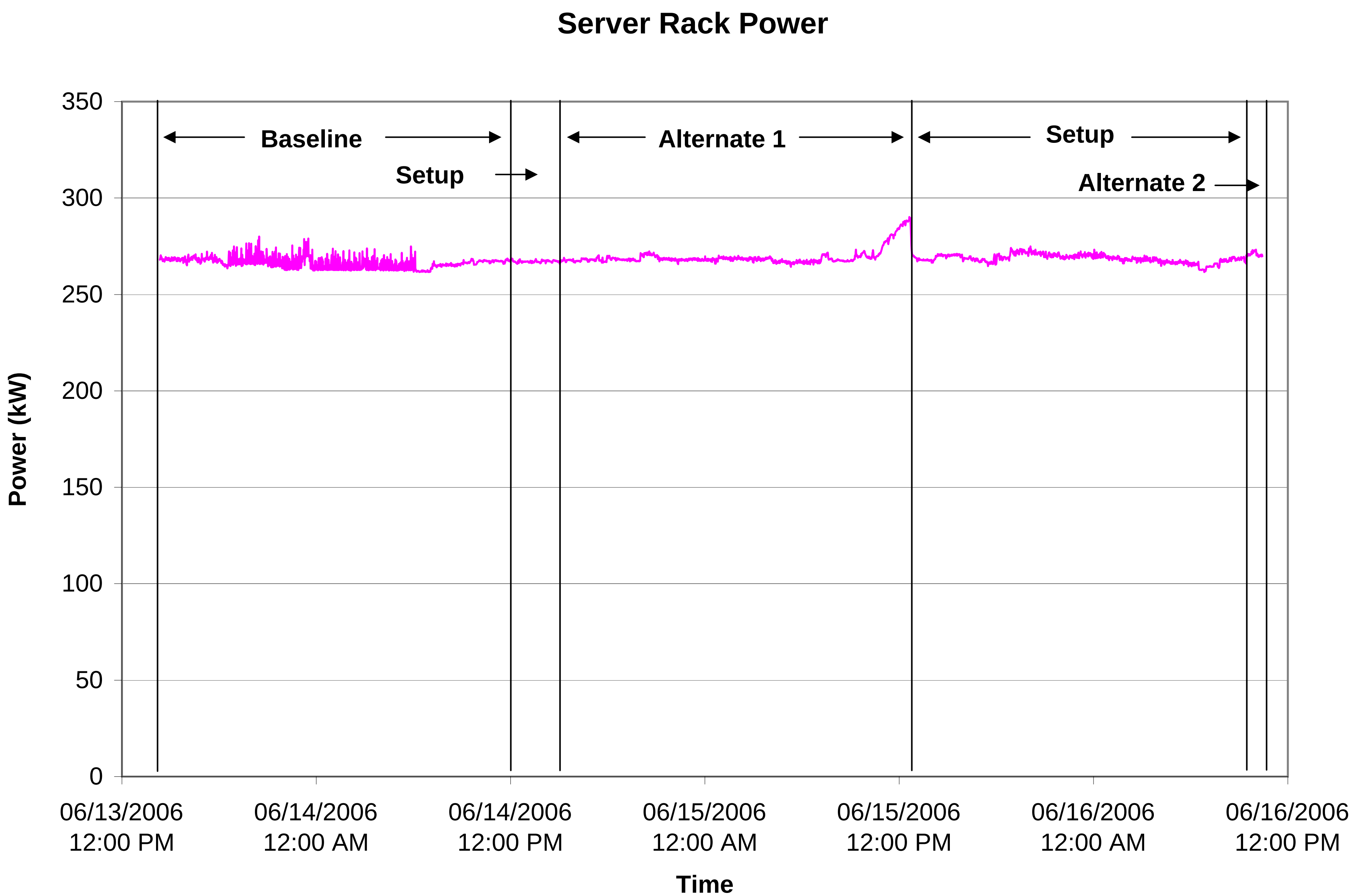
Appendix II – Detailed Calculations

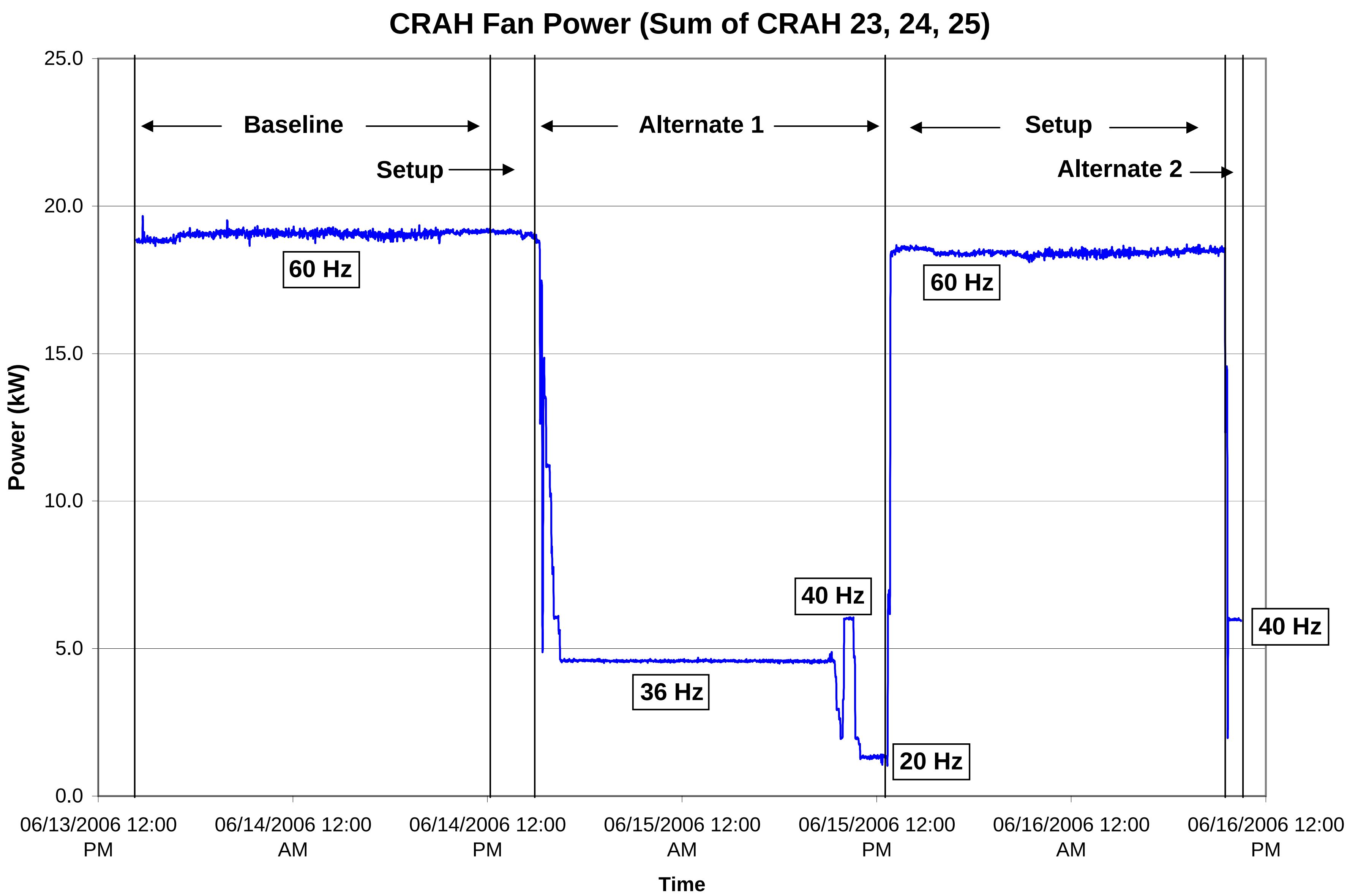
Appendix III – Photographs

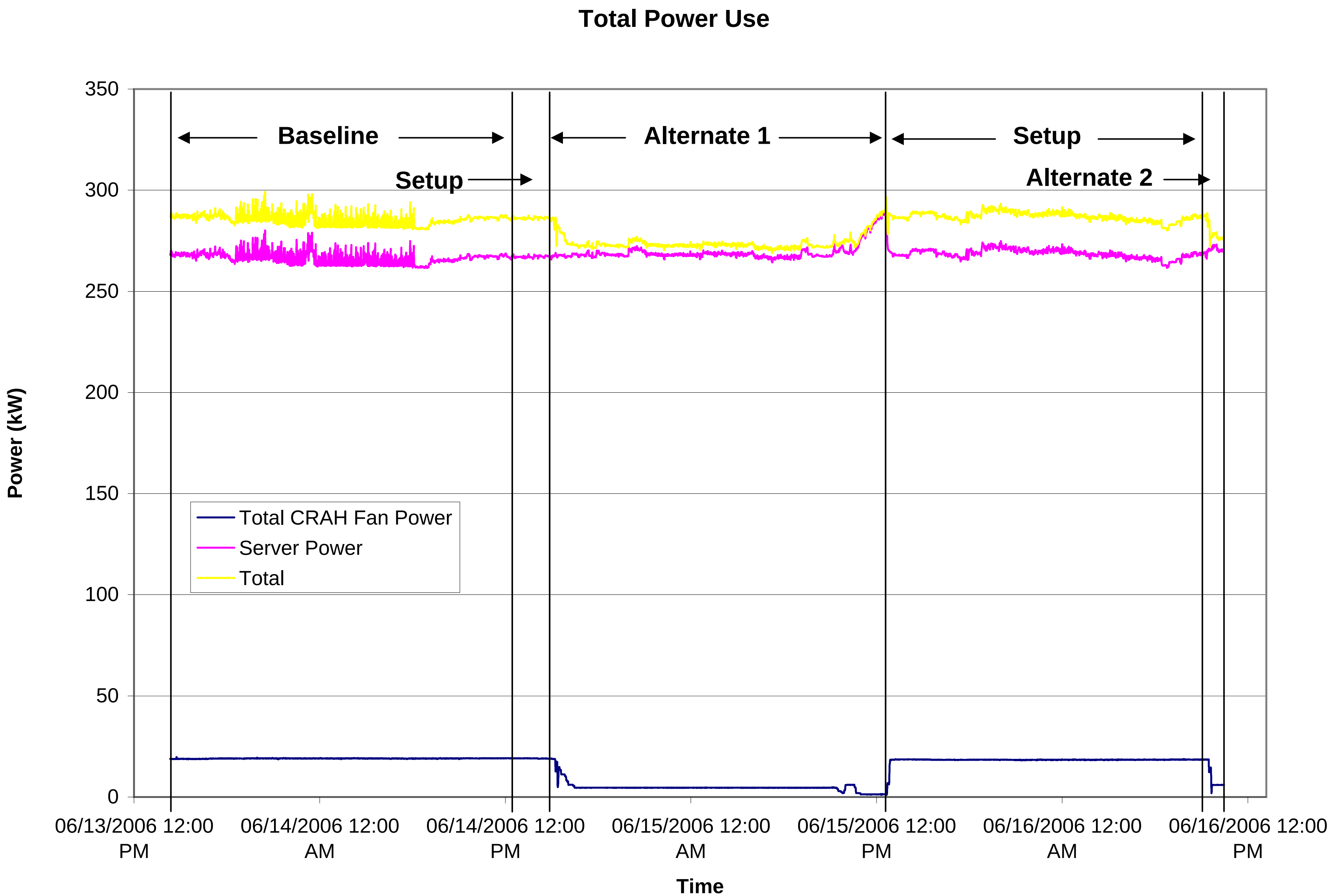
Appendix IV – Sensor Locations

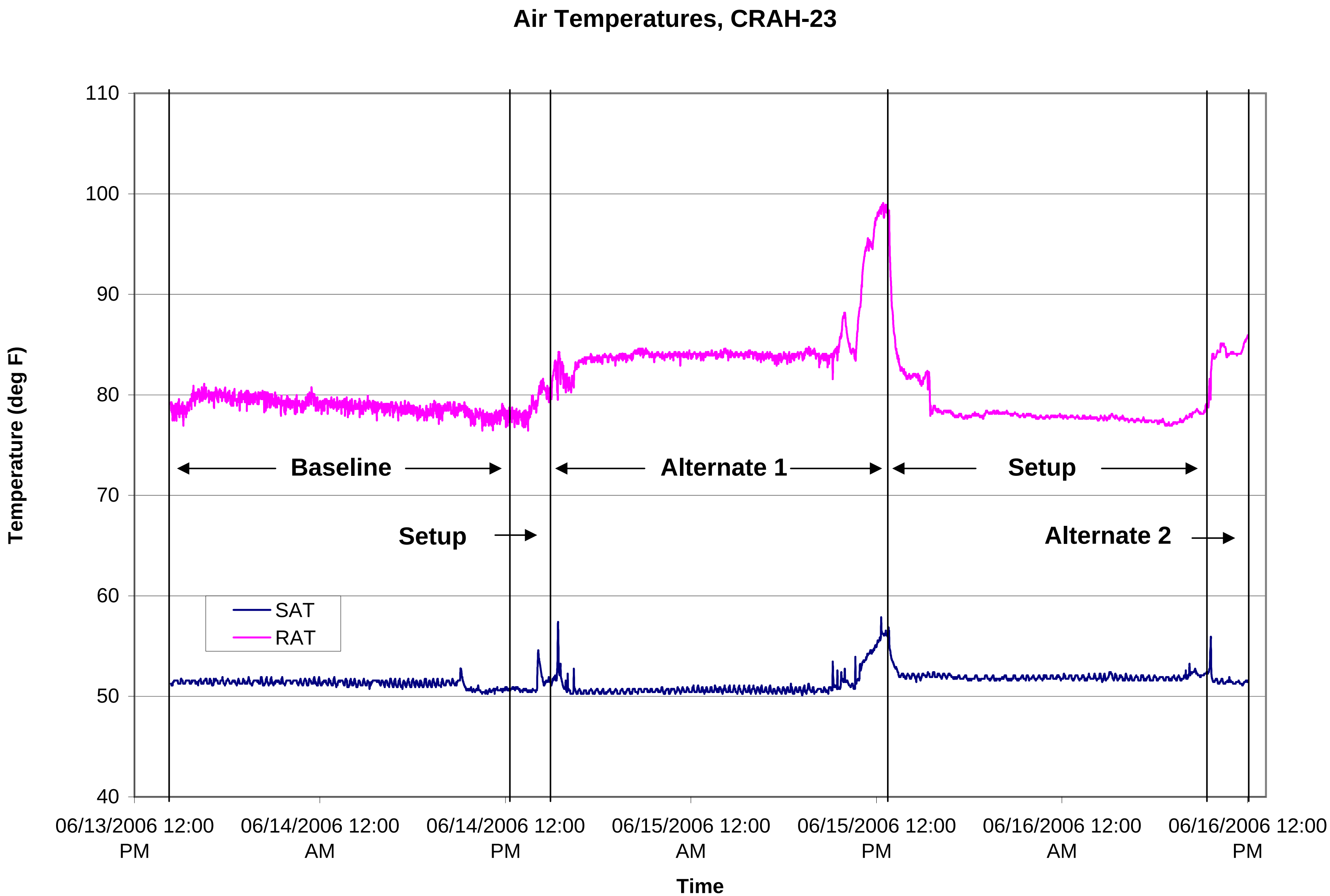
Appendix V – Revised Monitoring Plan

Appendix I – Collected Data

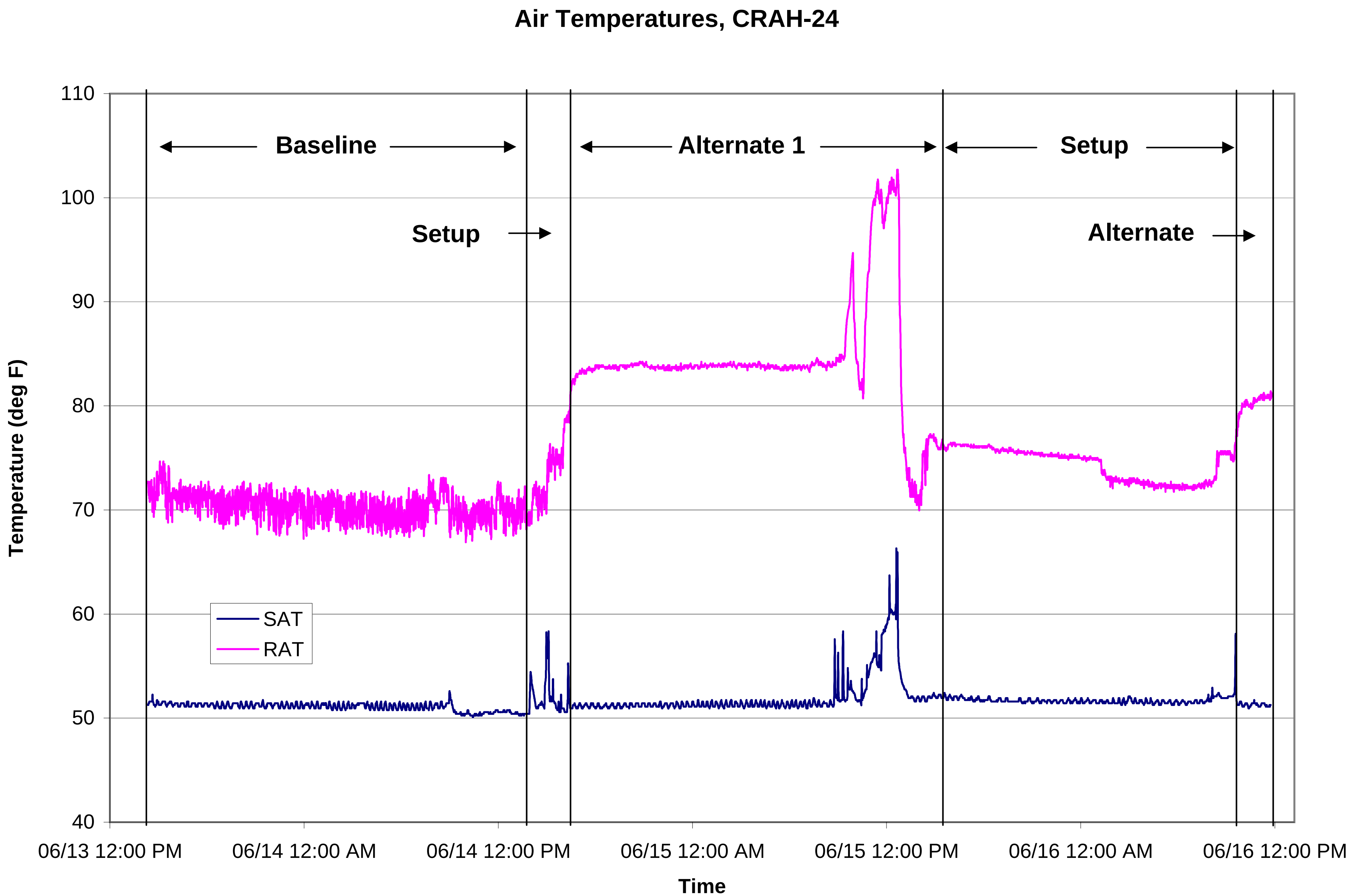

Server Rack Power
350
300
250
200
150
100
50
0
Power (kW)
Baseline
Setup
Alternate 1
Setup
Alternate 2
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

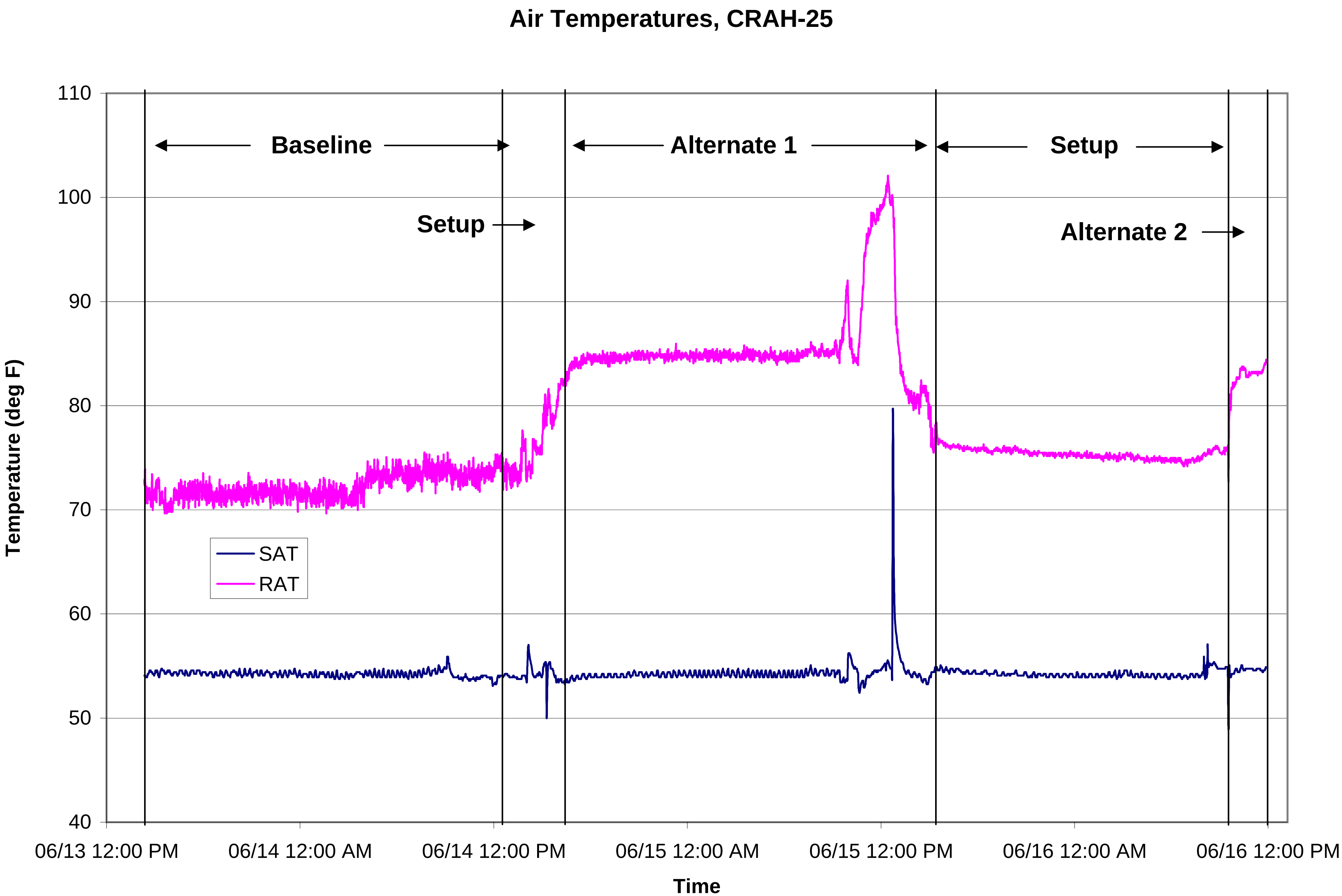

CRAH Fan Power (Sum of CRAH 23, 24, 25)
Baseline
Setup
Alternate 1
Setup
Alternate 2
60 Hz
60 Hz
36 Hz
40 Hz
20 Hz
40 Hz
Power (kW)
25.0
20.0
15.0
10.0
5.0
0.0
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time


Total Power Use
350
300
250
200
150
100
50
0
Power (kW)
Baseline
Setup
Alternate 1
Setup
Alternate 2
Total CRAH Fan Power
Server Power
Total
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

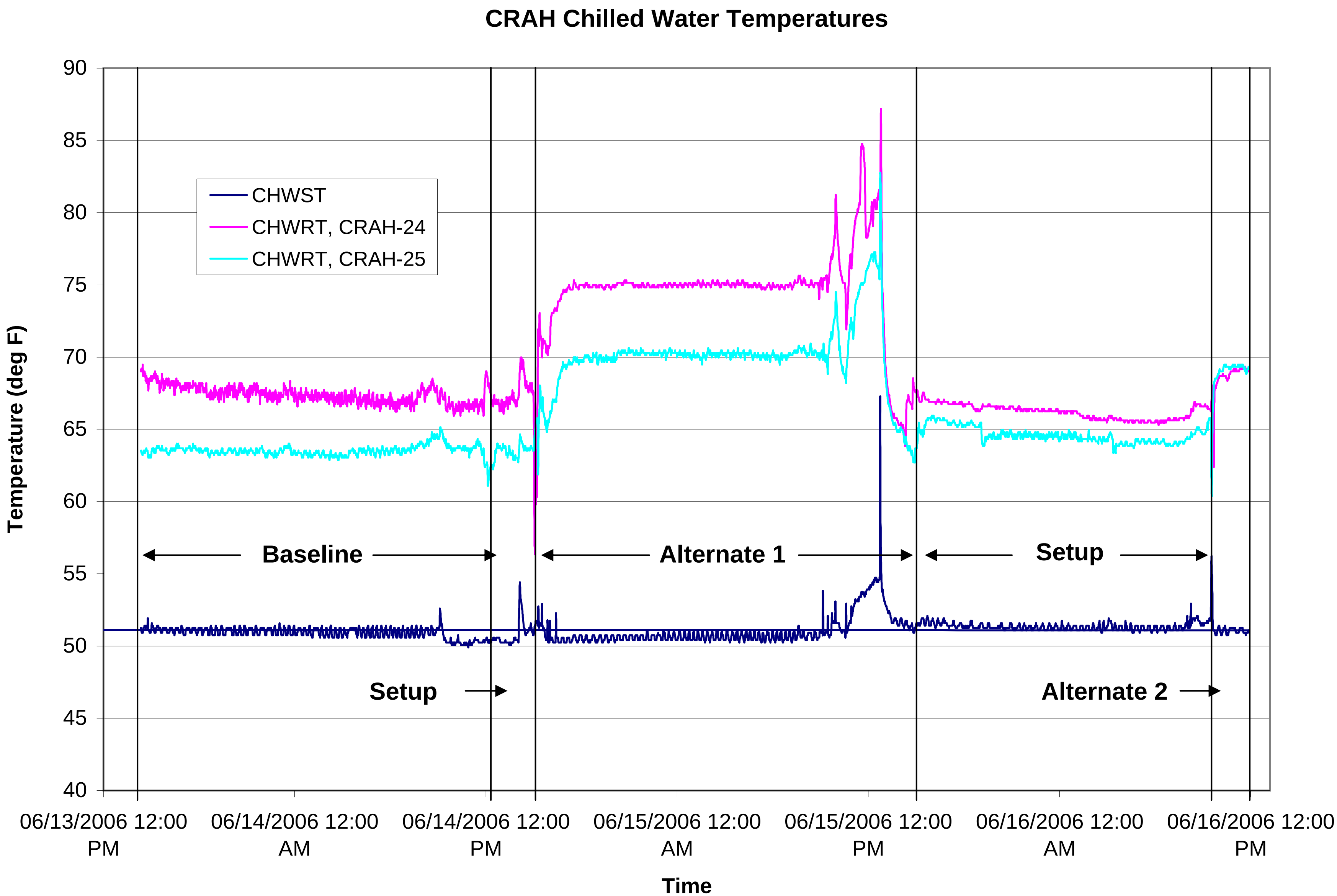


Air Temperatures, CRAH-23
Temperature (deg F)
110
100
90
80
70
60
50
40
Baseline
Setup
Alternate 1
Setup
Alternate 2
SAT
RAT
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

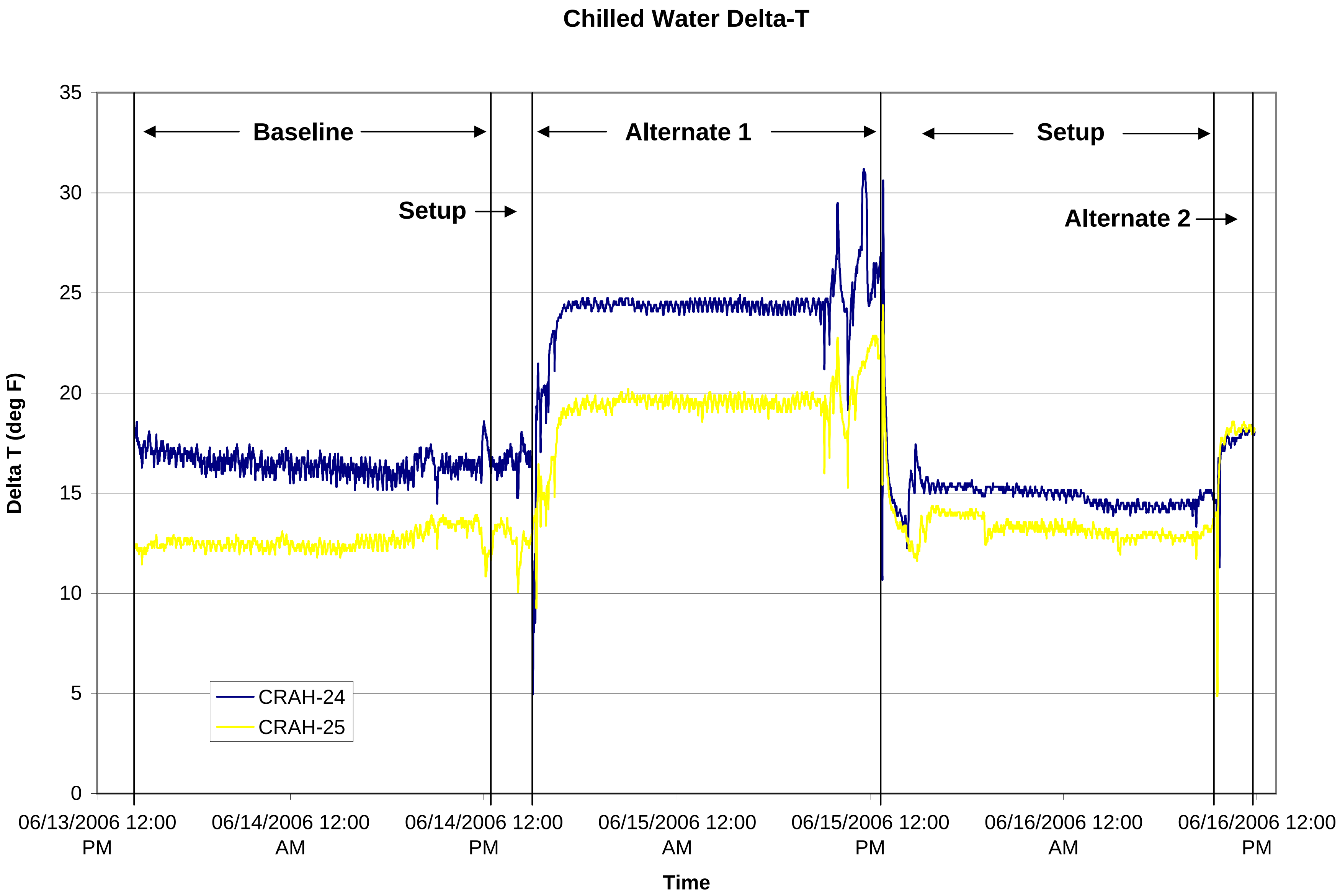

Air Temperatures, CRAH-24
110
100
90
80
70
60
50
40
Temperature (deg F)
Baseline
Setup
Alternate 1
Setup
Alternate
SAT
RAT
06/13 12:00 PM
06/14 12:00 AM
06/14 12:00 PM
06/15 12:00 AM
06/15 12:00 PM
06/16 12:00 AM
06/16 12:00 PM
Time

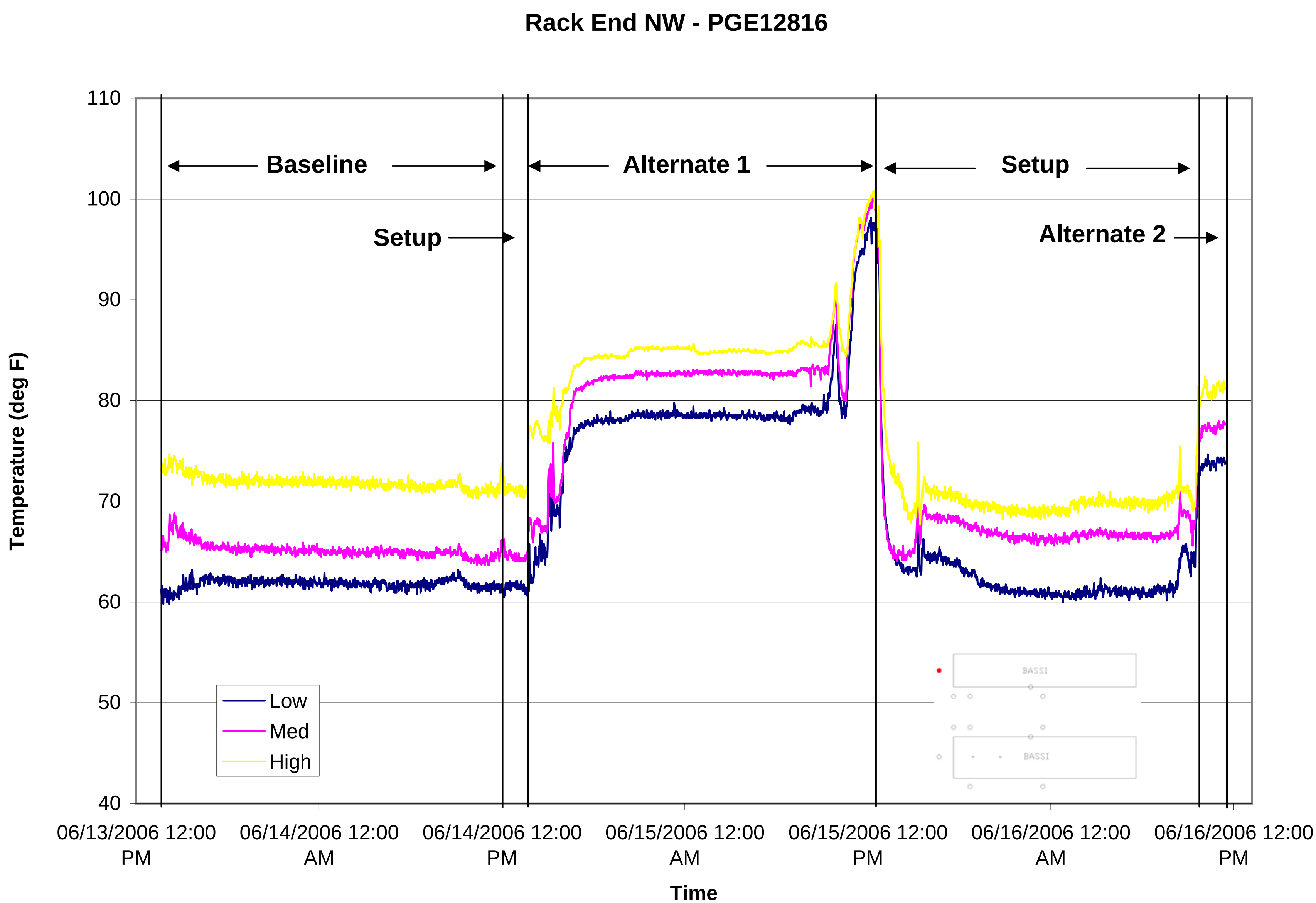


Air Temperatures, CRAH-25
110
100
90
80
70
60
50
40
Temperature (deg F)
Baseline
Setup
Alternate 1
Setup
Alternate 2
SAT
RAT
06/13 12:00 PM
06/14 12:00 AM
06/14 12:00 PM
06/15 12:00 AM
06/15 12:00 PM
06/16 12:00 AM
06/16 12:00 PM
Time

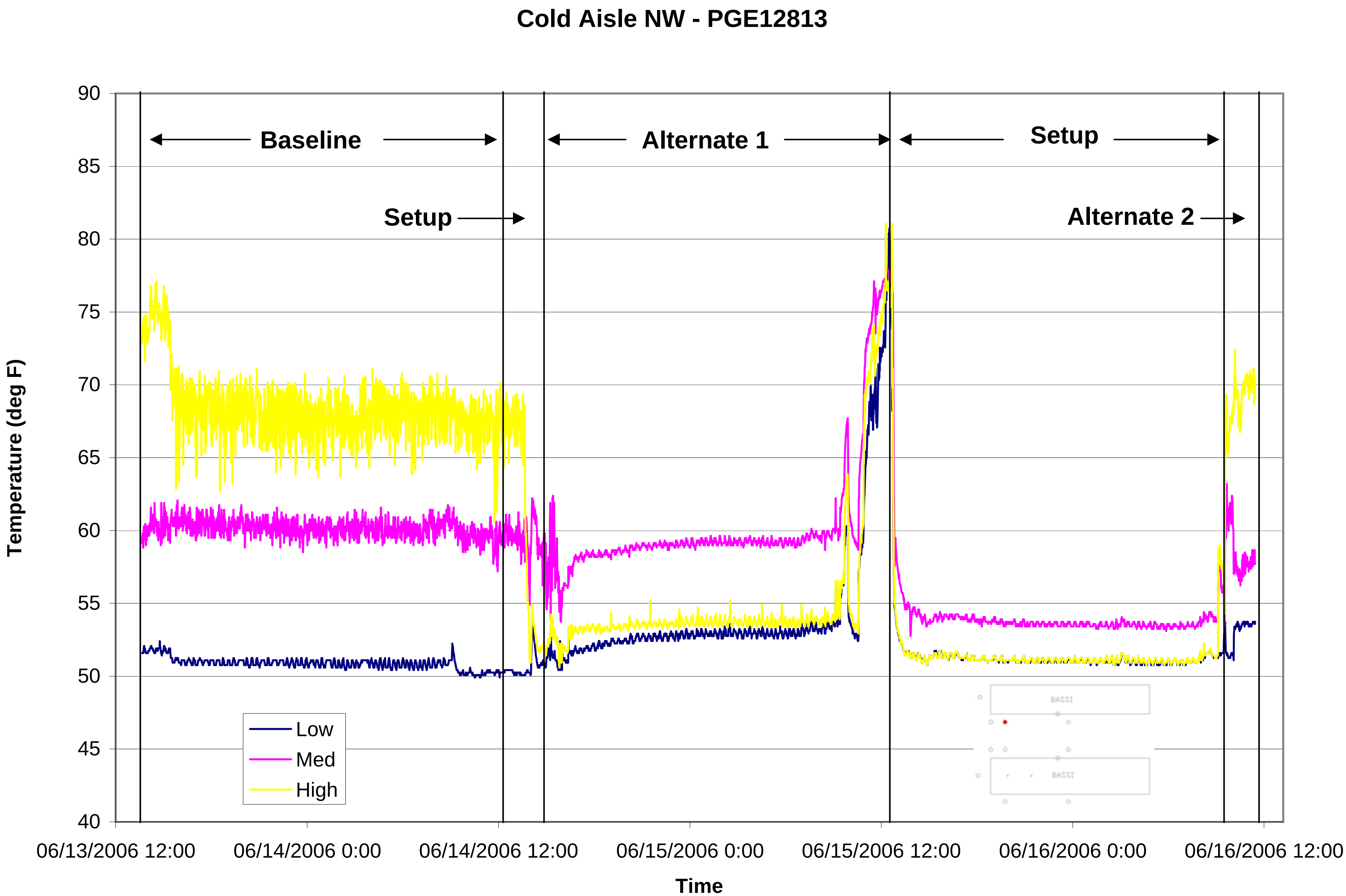


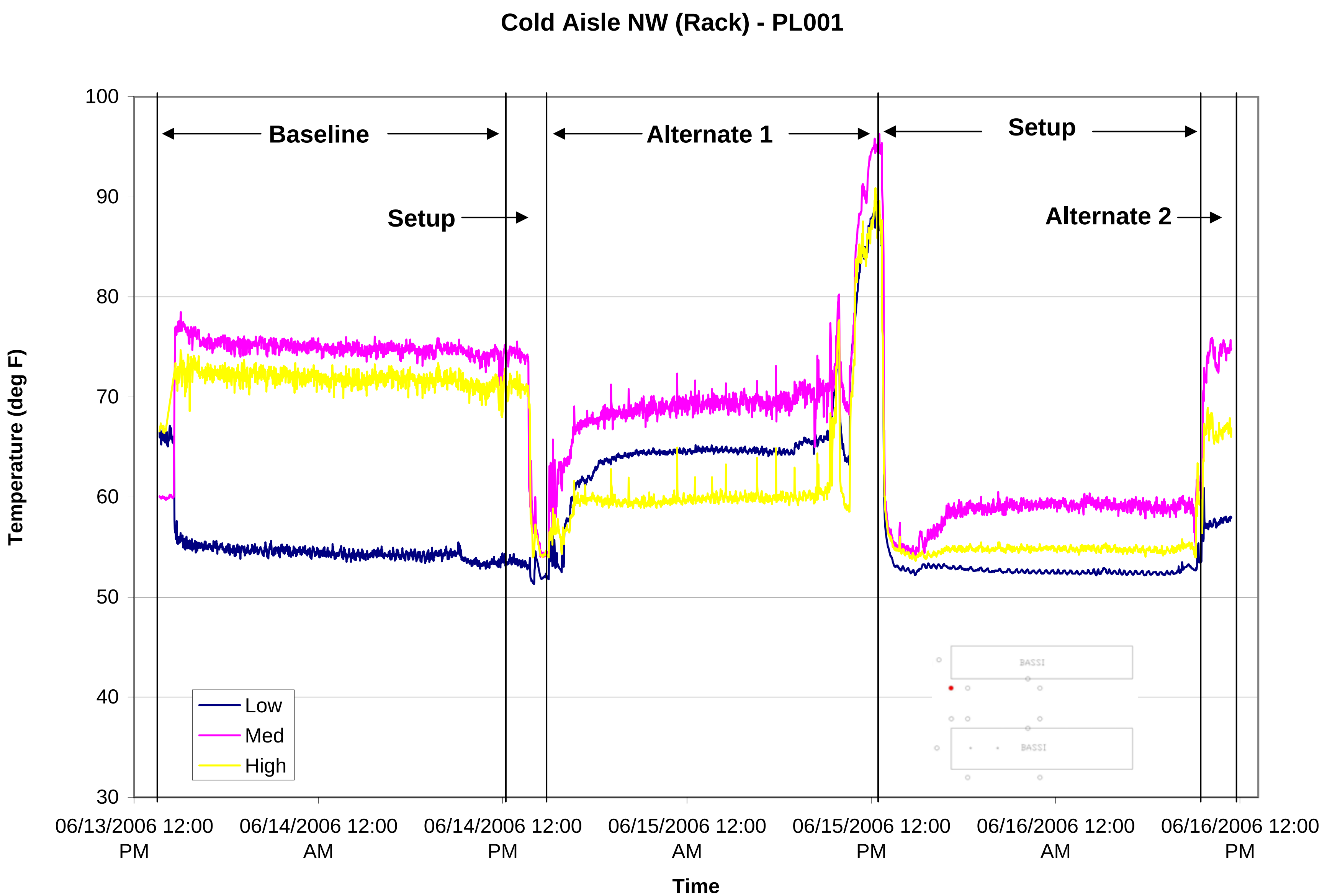
CRAH Chilled Water Temperatures
Temperature (deg F)
90
85
80
75
70
65
60
55
50
45
40
CHWST
CHWRT, CRAH-24
CHWRT, CRAH-25
Baseline
Alternate 1
Setup
Setup
Alternate 2
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

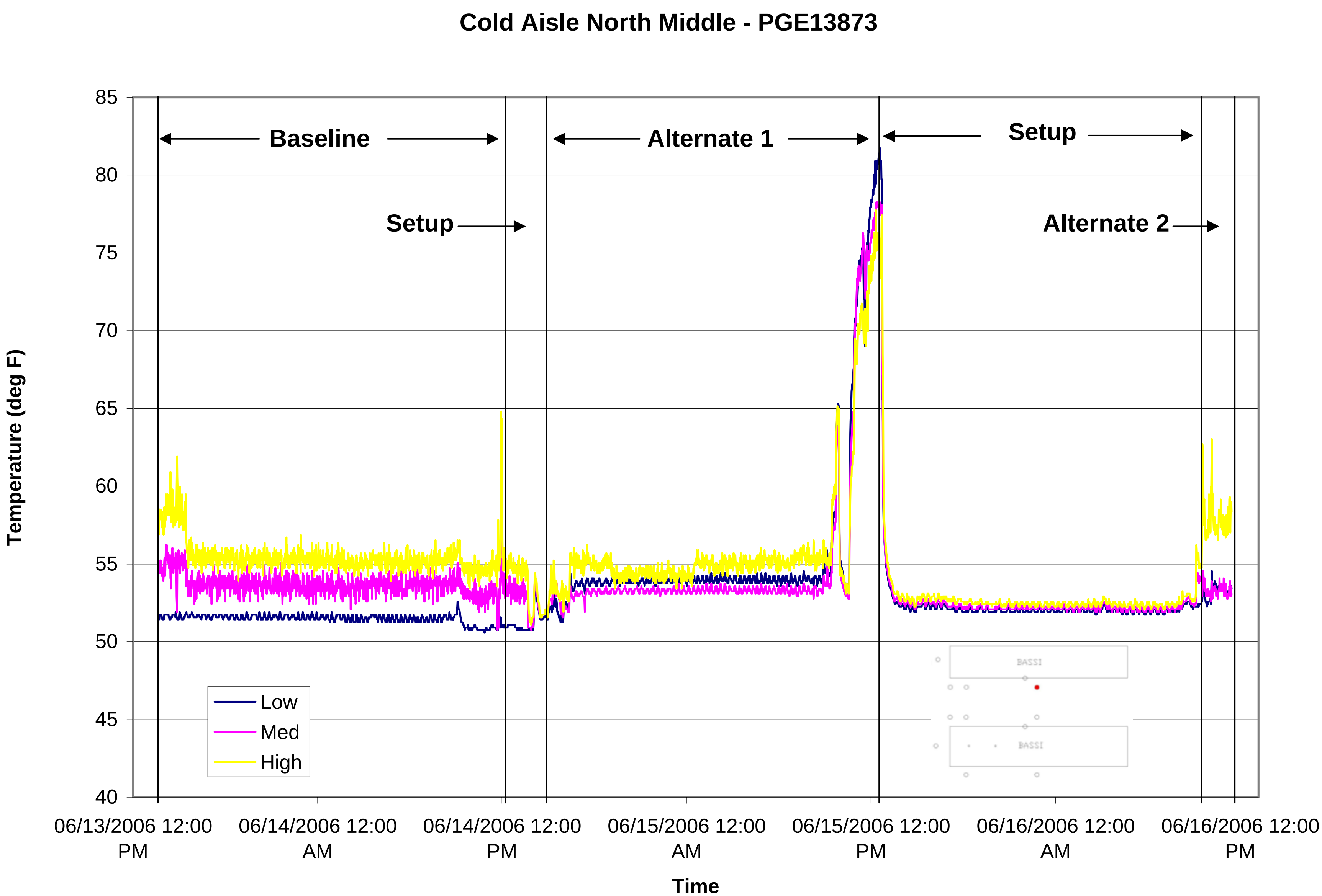

Chilled Water Delta-T
Baseline
Setup
Alternate 1
Setup
Alternate 2
Delta T (deg F)
35
30
25
20
15
10
5
0
CRAH-24
CRAH-25
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time



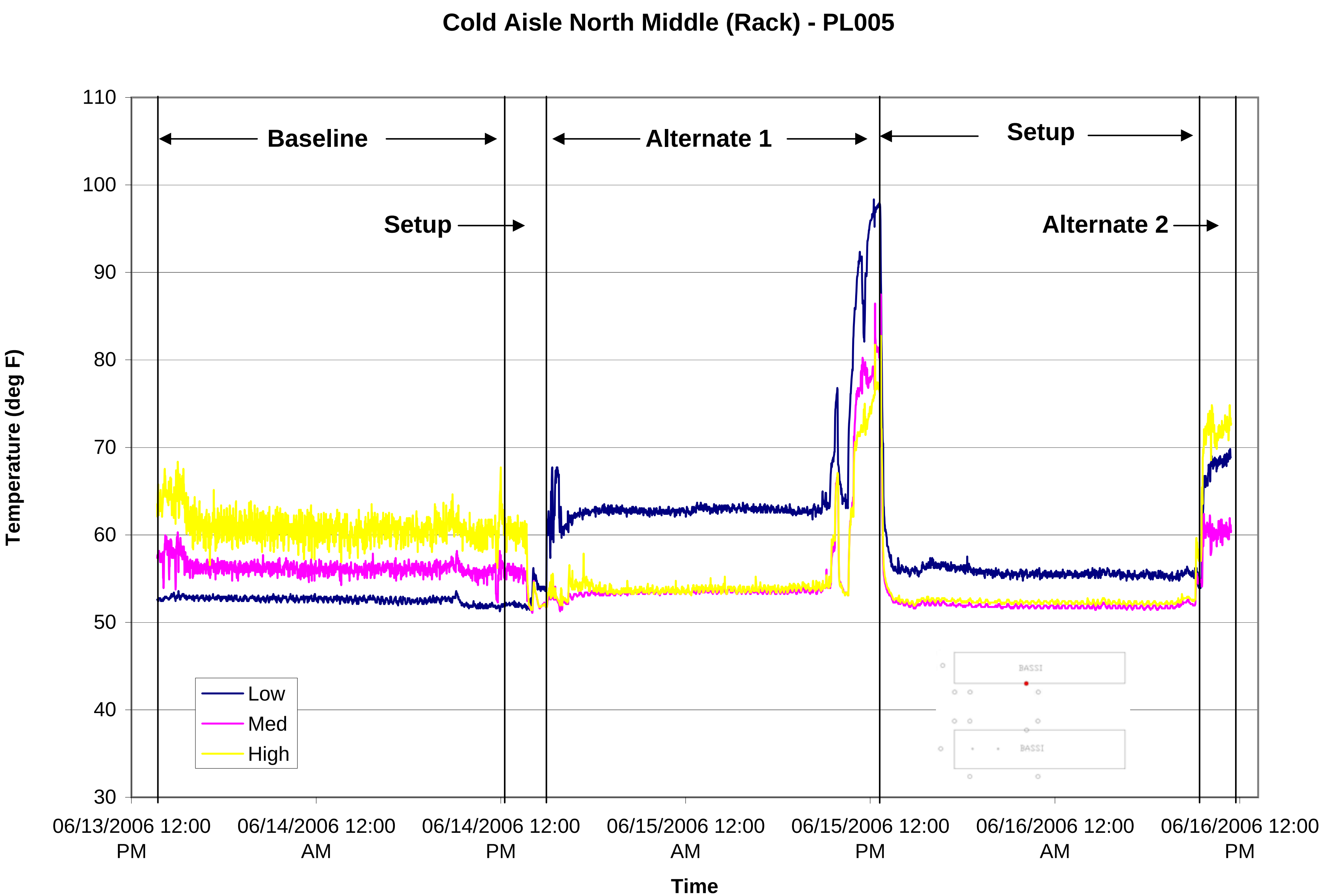
Rack End NW - PGE12816
Baseline
Setup
Alternate 1
Setup
Alternate 2
Temperature (deg F)
110
100
90
80
70
60
50
40
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time
Low
Med
High

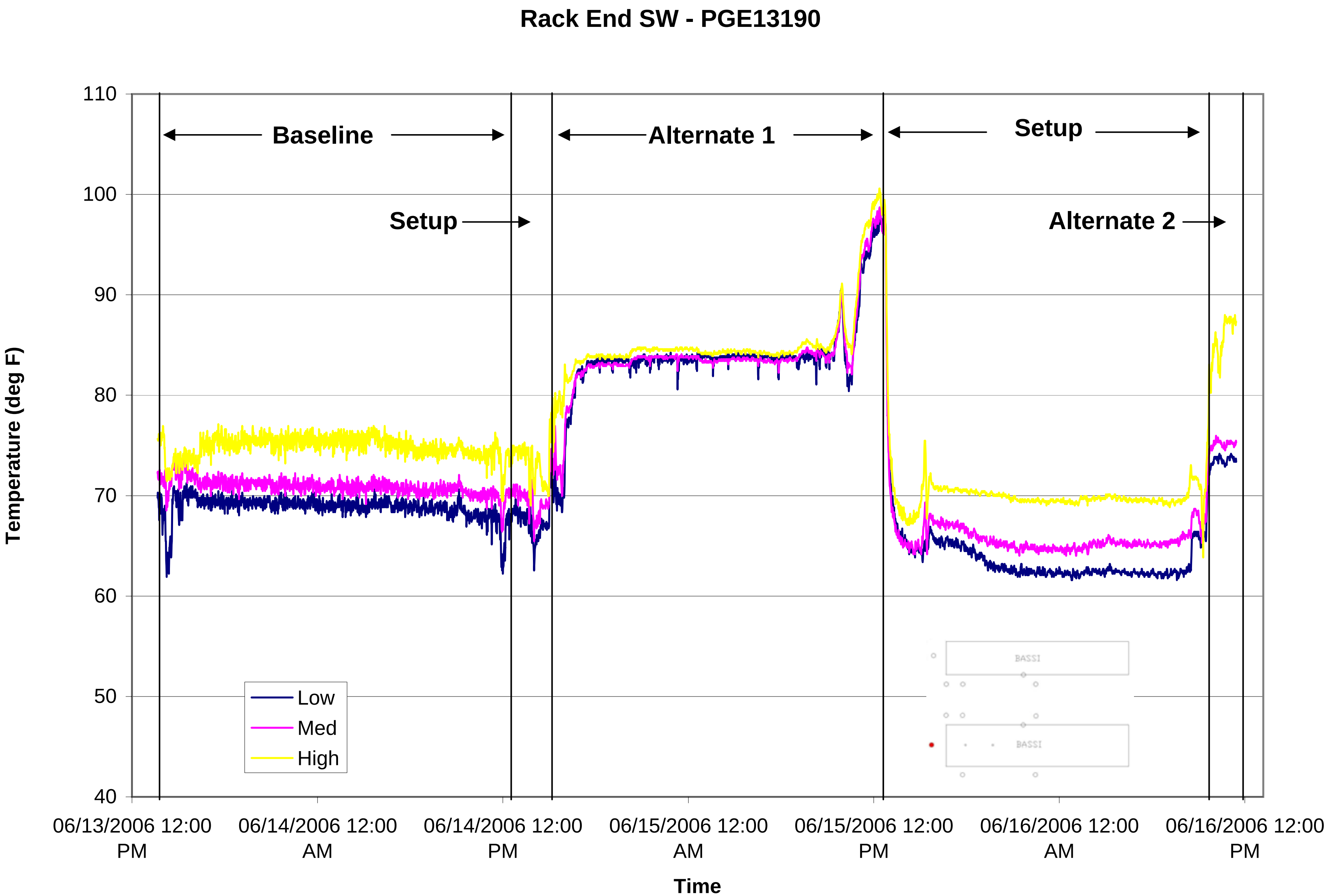

Cold Aisle NW - PGE12813
Baseline
Setup
Alternate 1
Setup
Alternate 2
Low
Med
High
Temperature (deg F)
90
85
80
75
70
65
60
55
50
45
40
06/13/2006 12:00
06/14/2006 0:00
06/14/2006 12:00
06/15/2006 0:00
06/15/2006 12:00
06/16/2006 0:00
06/16/2006 12:00
Time


Cold Aisle NW (Rack) - PL001
Baseline
Alternate 1
Setup
Setup
Alternate 2
Temperature (deg F)
100
90
80
70
60
50
40
30
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time
BASSI
BASSI

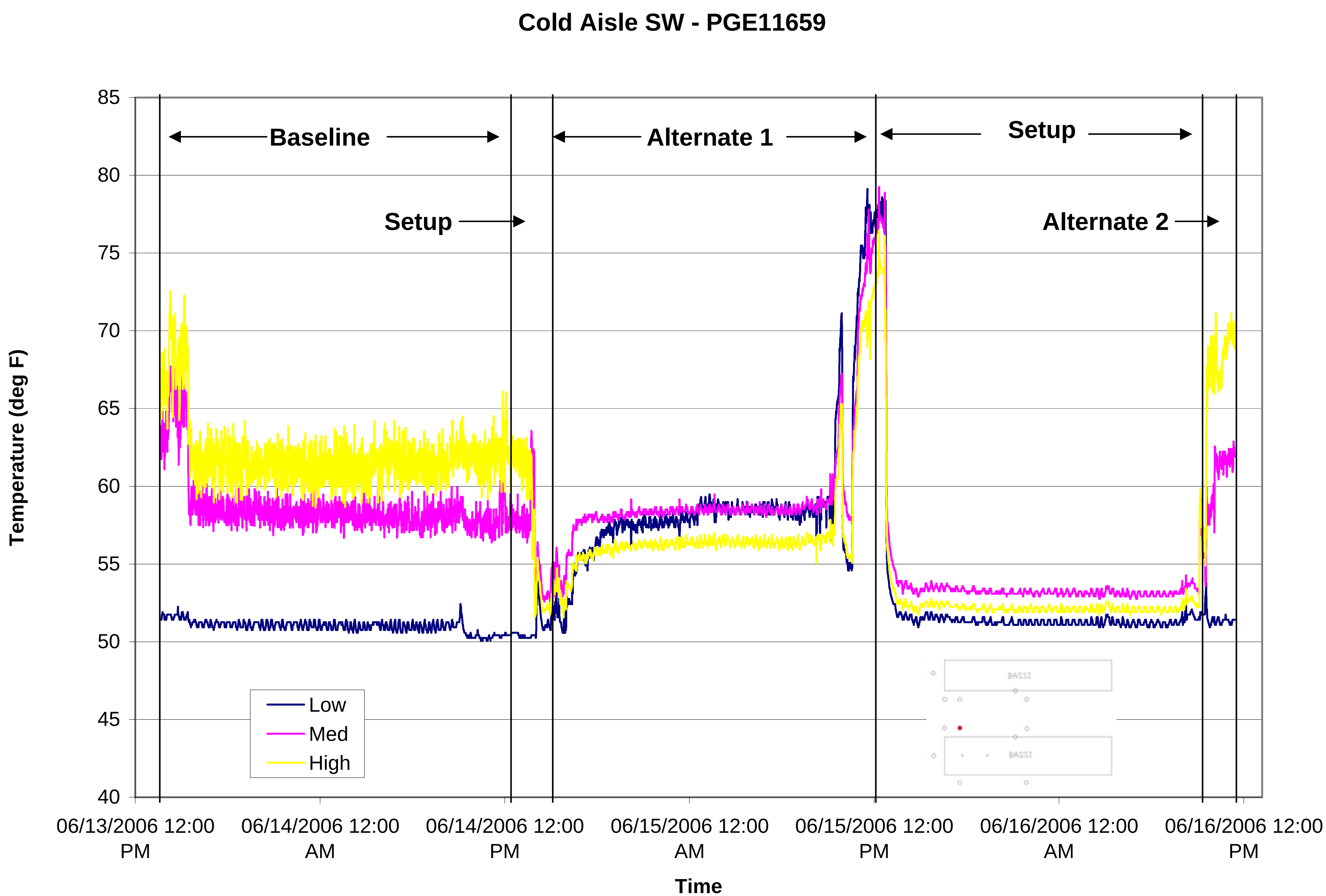


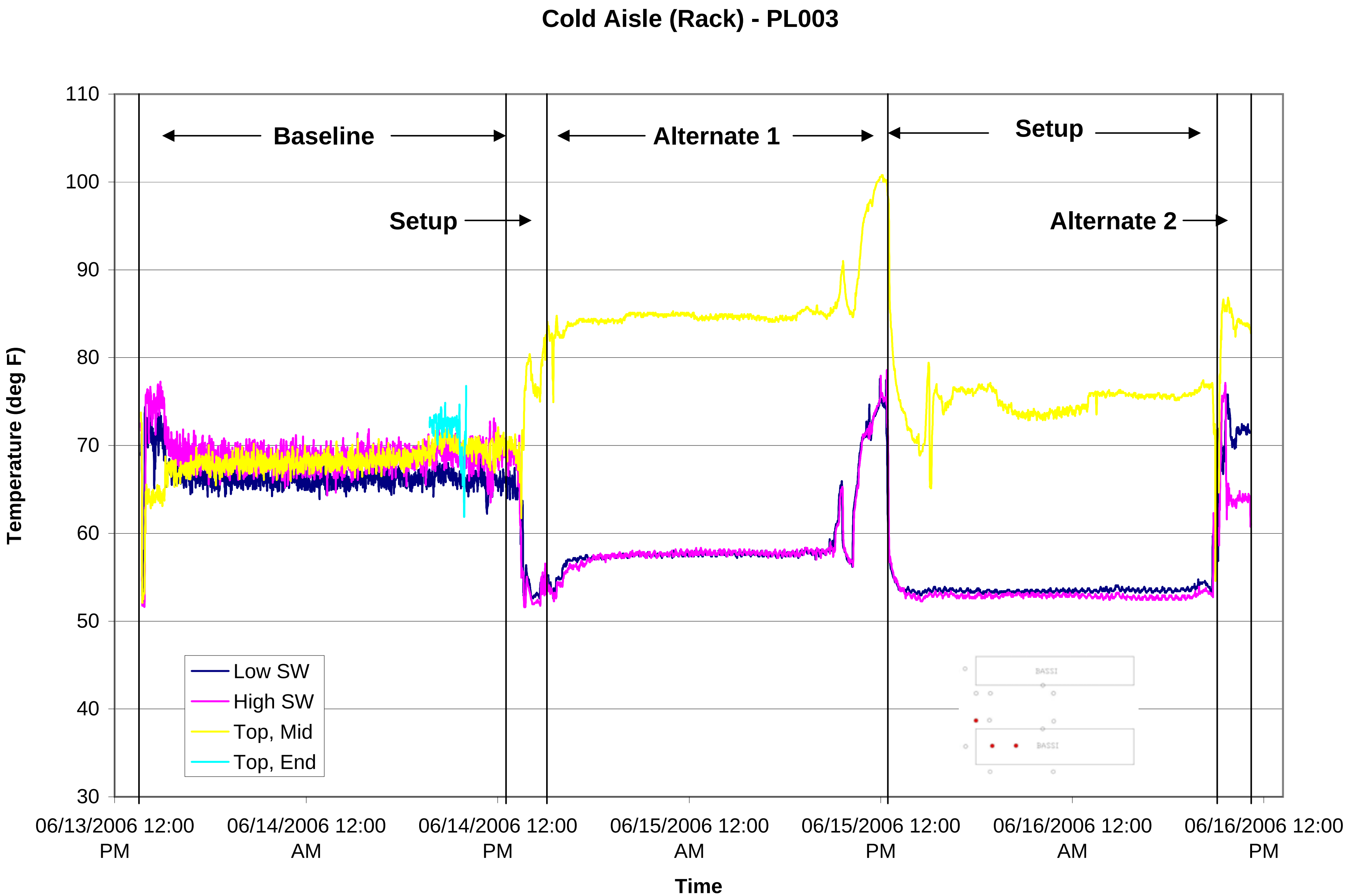
Cold Aisle North Middle - PGE13873
Baseline
Setup
Alternate 1
Setup
Alternate 2
Temperature (deg F)
85
80
75
70
65
60
55
50
45
40
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time
Low
Med
High

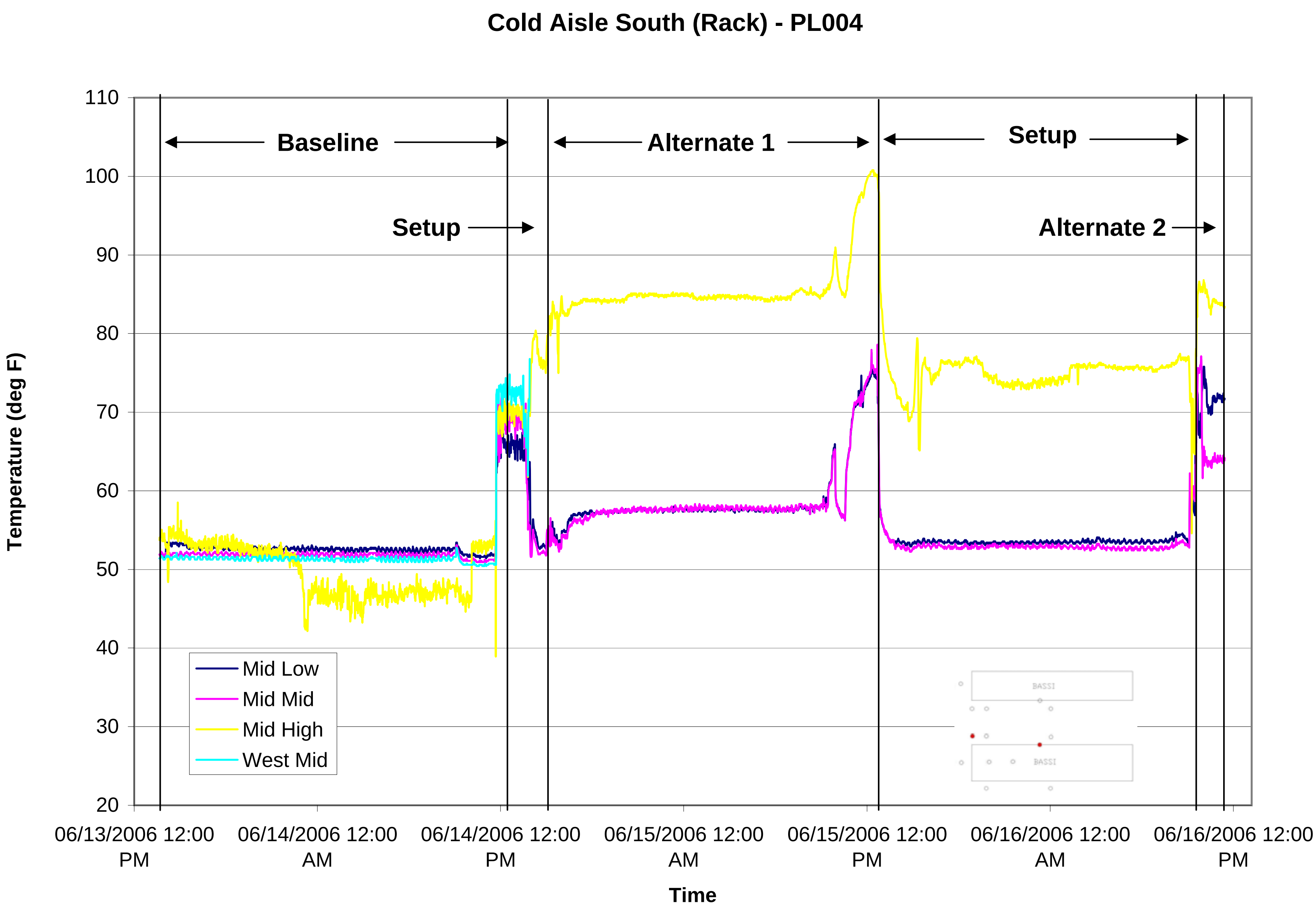

Cold Aisle North Middle (Rack) - PL005
Temperature (deg F)
110
100
90
80
70
60
50
40
30
Baseline
Alternate 1
Setup
Setup
Alternate 2
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

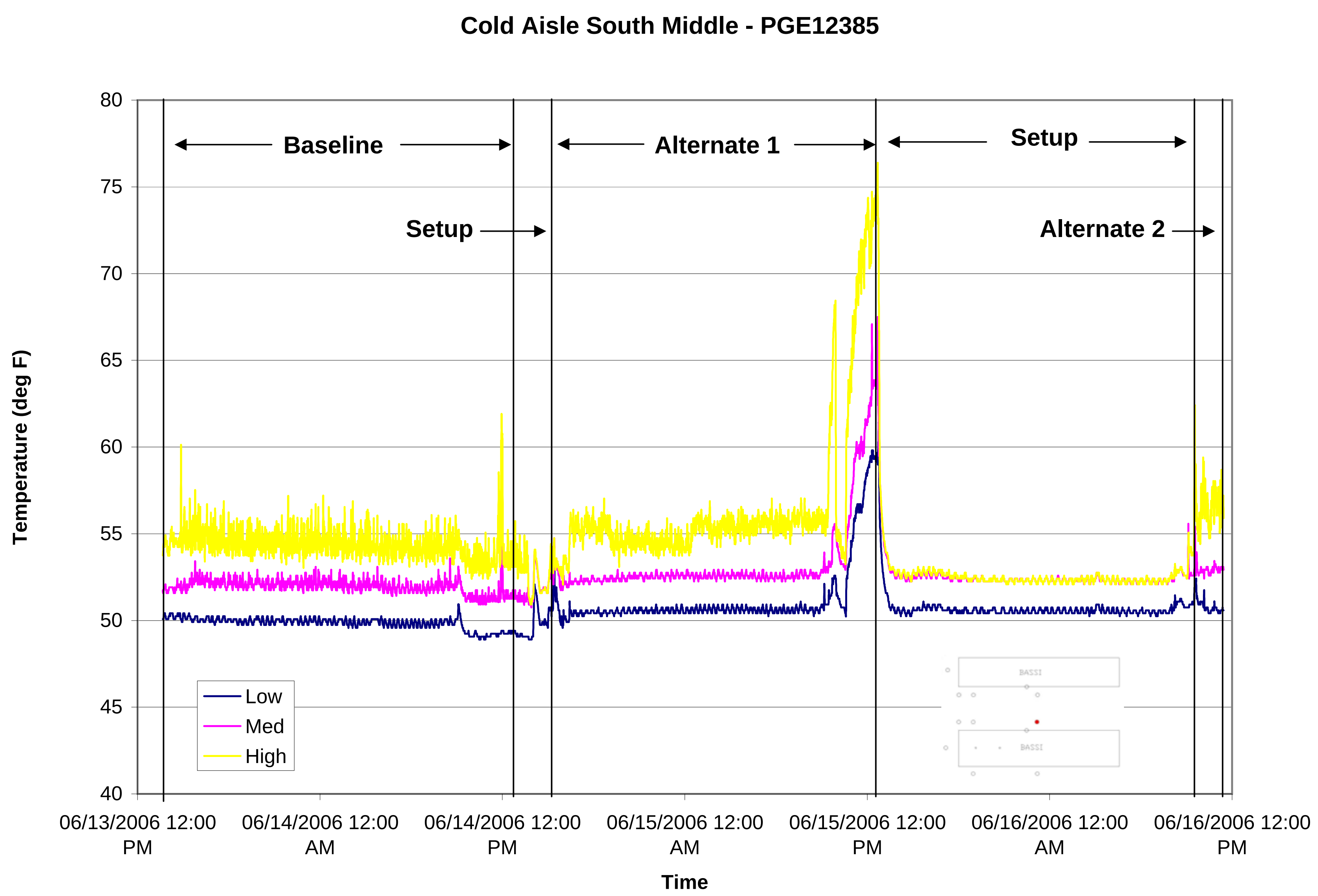


Rack End SW - PGE13190
Baseline
Setup
Alternate 1
Setup
Alternate 2
Temperature (deg F)
110
100
90
80
70
60
50
40
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

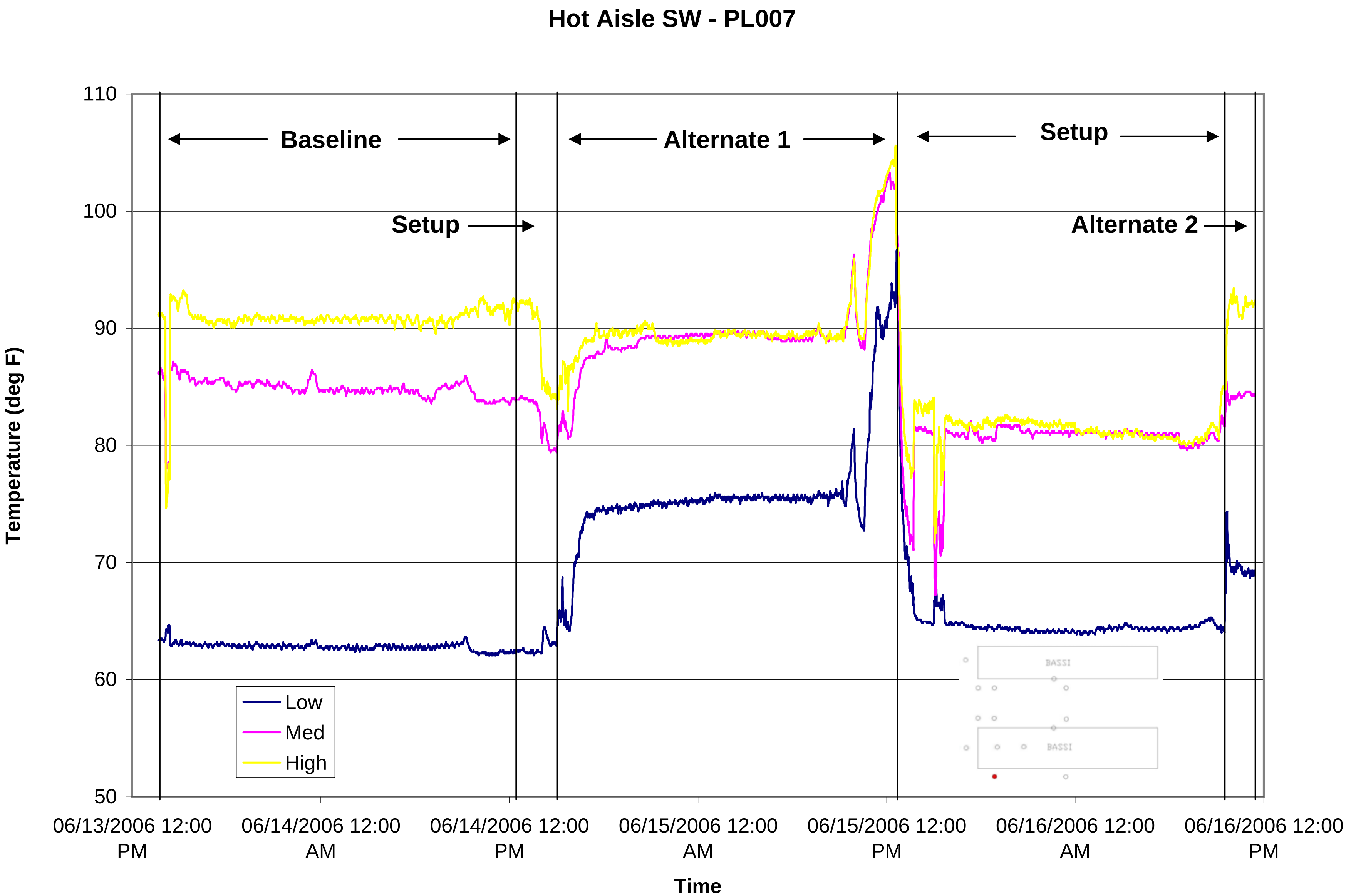

Cold Aisle SW - PGE11659
Baseline
Setup
Alternate 1
Setup
Alternate 2
Temperature (deg F)
85
80
75
70
65
60
55
50
45
40
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time


Cold Aisle (Rack) - PL003
Baseline
Setup
Alternate 1
Setup
Alternate 2
110
100
90
80
70
60
50
40
30
Temperature (deg F)
Low SW
High SW
Top, Mid
Top, End
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

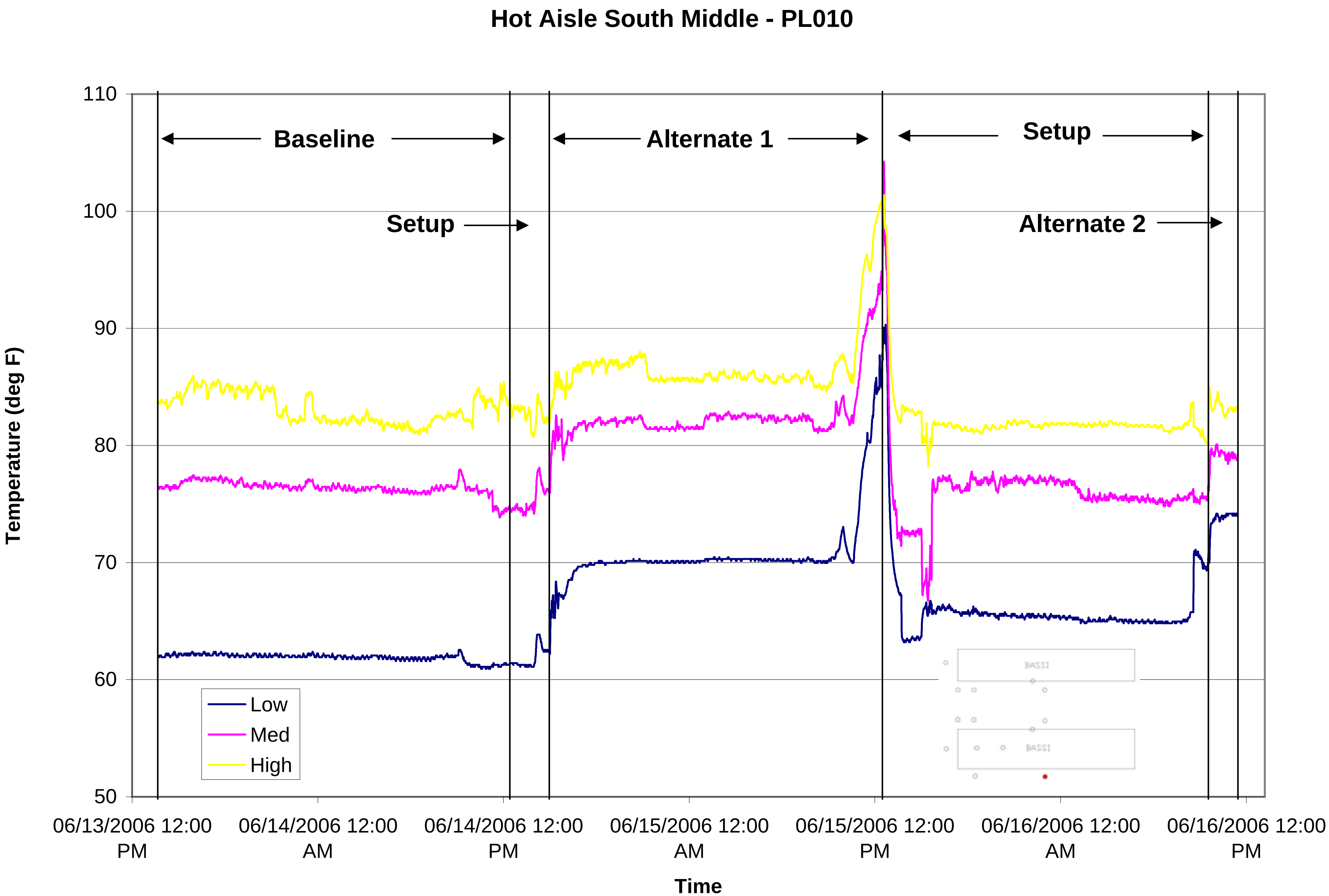

Cold Aisle South (Rack) - PL004
Baseline
Setup
Alternate 1
Setup
Alternate 2
110
100
90
80
70
60
50
40
30
20
Temperature (deg F)
Mid Low
Mid Mid
Mid High
West Mid
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

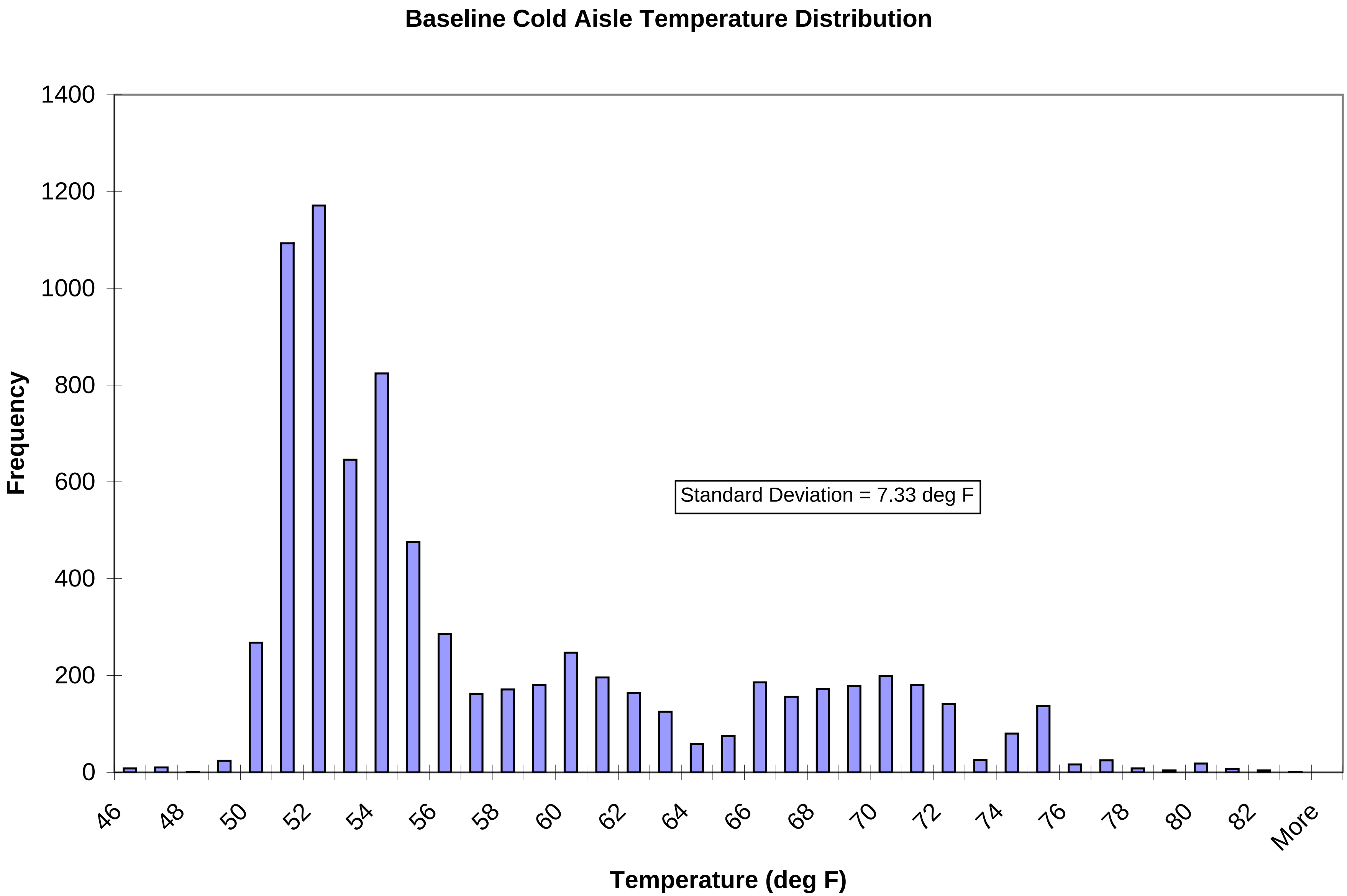


Cold Aisle South Middle - PGE12385
Temperature (deg F)
80
75
70
65
60
55
50
45
40
Baseline
Setup
Alternate 1
Setup
Alternate 2
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

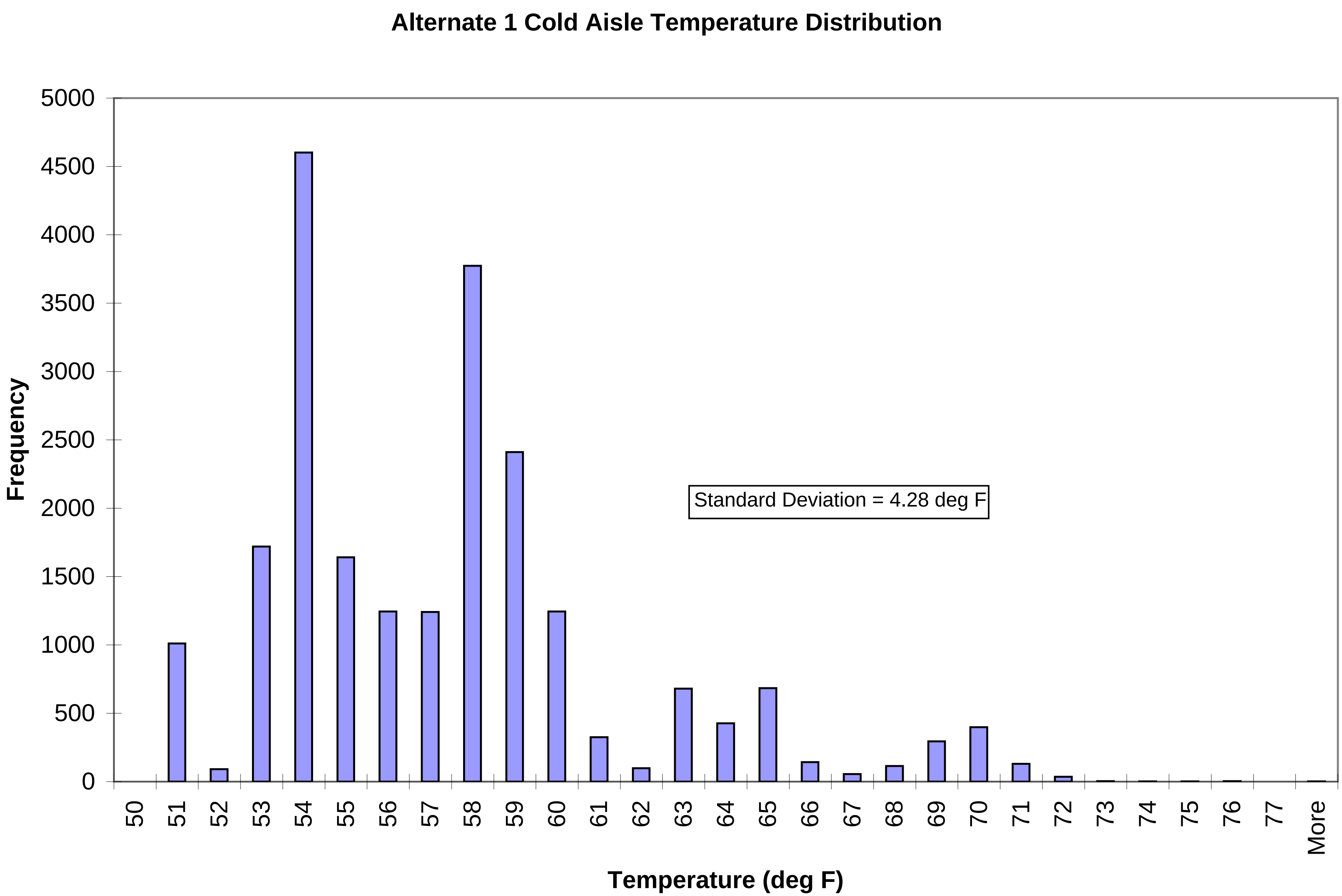


Hot Aisle SW - PL007
Baseline
Setup
Alternate 1
Setup
Alternate 2
Temperature (deg F)
110
100
90
80
70
60
50
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time

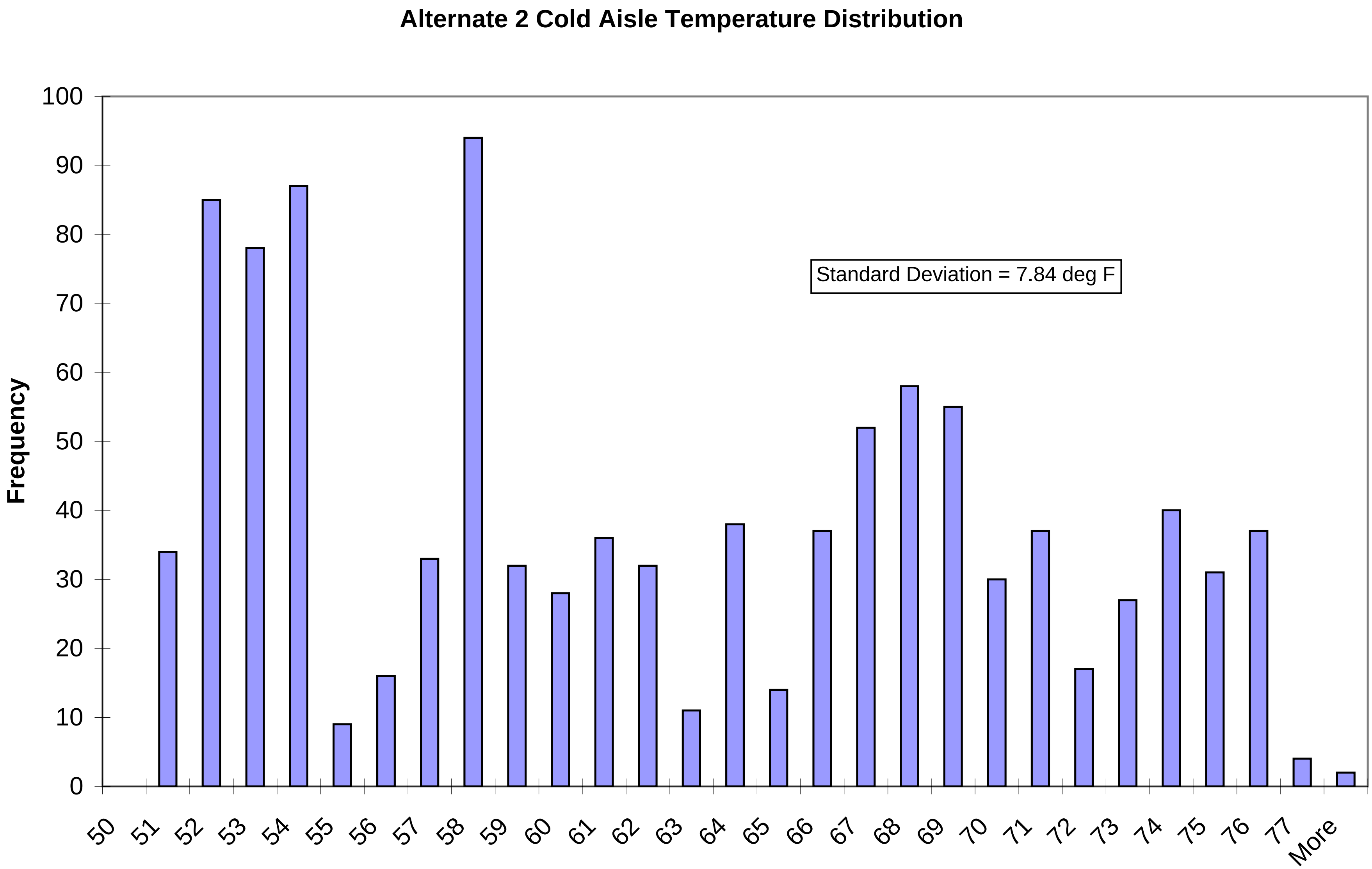

Hot Aisle South Middle - PL010
Temperature (deg F)
110
100
90
80
70
60
50
Baseline
Setup
Alternate 1
Setup
Alternate 2
Low
Med
High
06/13/2006 12:00 PM
06/14/2006 12:00 AM
06/14/2006 12:00 PM
06/15/2006 12:00 AM
06/15/2006 12:00 PM
06/16/2006 12:00 AM
06/16/2006 12:00 PM
Time



Baseline Cold Aisle Temperature Distribution
Frequency
1400
1200
1000
800
600
400
200
0
Standard Deviation = 7.33 deg F
46
48
50
52
54
56
58
60
62
64
66
68
70
72
74
76
78
80
82
More
Temperature (deg F)



Alternate 1 Cold Aisle Temperature Distribution
Frequency
5000
4500
4000
3500
3000
2500
2000
1500
1000
500
0
Standard Deviation = 4.28 deg F
50
51
52
53
54
55
56
57
58
59
60
61
62
63
64
65
66
67
68
69
70
71
72
73
74
75
76
77
More
Temperature (deg F)


Alternate 2 Cold Aisle Temperature Distribution
Standard Deviation = 7.84 deg F
Frequency
0
10
20
30
40
50
60
70
80
90
100
50
51
52
53
54
55
56
57
58
59
60
61
62
63
64
65
66
67
68
69
70
71
72
73
74
75
76
77
More
Temperature (deg F)

LBNL/PG&E Emerging Technologies Datacenter Airflow Study
Air Flow Data (all velocities in FPM)

N
W ^ E
S

Baseline Configuration
All three fans at 60 Hz

Underfloor Supply

Number of floor grates	28	
Assumed area factor	0.9	
Total area per grate	4	sq.ft.
Average CFM per grate	1,588	cfm
Total CFM for all	44,464	

418	485	524	469	489	426	453	435	376		
341	482	432	429	405	498	507	456	469	382	
429	434	400	403	432			449	466	442	420

AHUs Return Air

Free Area for Return 25 sq.ft.

AHU-23	663	656	600
AHU-24	712	682	655
AHU-25	847	737	753

	AHU-23	AHU-24	AHU-25
Average Velocity	640 fpm	683 fpm	779 fpm
Total cfm	16,050 cfm	17,137 cfm	19,546 cfm

TOTAL AHU (all) RA	52,733	cfm

LBNL/PG&E Emerging Technologies Datacenter Airflow Study
Air Flow Data (all velocities in FPM)

N
W ^ E
S

Alternate 1 Configuration
All three fans at 20 Hz

Underfloor Supply

Number of floor grates	28	
Assumed area factor	0.9	
Total area per grate	4	sq.ft.
Average CFM per grate	1,564	cfm
Total CFM for all	43,790	

415	478	498	495	443	445	465	399	425		
176	477	497	507	494	462	469	475	437	430	
82	460	500	473	453			489	485	411	324

AHUs Return Air

Free Area for Return 25 sq.ft.

AHU-23	240	222	219
AHU-24	188	232	227
AHU-25	250	240	244

	AHU-23	AHU-24	AHU-25
Average Velocity	227 fpm	216 fpm	245 fpm
Total cfm	5,696 cfm	5,411 cfm	6,139 cfm

TOTAL AHU (all) RA	17,246	cfm

LBNL/PG&E Emerging Technologies Datacenter Airflow Study
Air Flow Data (all velocities in FPM)

N
W ^ E
S

Alternate 2 Configuration
All three fans at 40 Hz

Underfloor Supply

Number of floor grates	28	
Assumed area factor	0.9	
Total area per grate	4	sq.ft.
Average CFM per grate	1,357	cfm
Total CFM for all	37,991	

378	423	409	367	377	384	365	366	330		
352	395	386	391	394	395	401	379	371	347	
321	394	351	359	388			384	386	386	374

AHUs Return Air

Free Area for Return 25 sq.ft.

AHU24, 25 not accessible - assumed same airflows as AHU23

AHU-23	210	234	189
AHU-24	210	234	189
AHU-25	210	234	189

	AHU-23	AHU-24	AHU-25
Average Velocity	211 fpm	211 fpm	211 fpm
Total cfm	5,294 cfm	5,294 cfm	5,294 cfm

TOTAL AHU (all) RA	15,883	cfm

Appendix II – Detailed Calculations

Datacenter Air Management Demonstration

Summary of Energy Efficiency Measures

Measures	CRAH Capacity Increase	Piping Capacity Increase	Test Area		Extrapolated to Entire Data Center		
			Peak Power Reduction	Energy Reduction	Peak Power Reduction	Energy Reduction	% Reduction in Total Energy Use
			kW	kWh	kW	kWh	
Increase Air Side Delta-T to 30F	49%	N/A	13.3	116,508	84.2	737,884	1.8%
Increase Chilled Water Delta-T to 25F	N/A	43%	0.9	7,609	12.2	106,525	0.3%
Increase Chilled Water Supply Temperature to 50F	N/A	N/A	1.8	15,333	24.5	214,664	0.5%
Water Side Economizing with 50F Chilled Water Supply Temperature	N/A	N/A	0.0	81,673	0.0	1,143,420	2.8%

Facility Wide Power Calculations

Parameters

Average Test Area Server Power	270	kW
Average Test Area Fan Power	18.5	kW
Average Chiller Power, entire data center	613	kW
Annual Hours of Operation	8760	hrs

Calculations

Power use, extrapolated to entire data center	4,652	kW
Annual Energy use	40,748	MWh

Increase Air Side Delta-T

Parameters	Value	Units	Comments
Current Average Supply Temp	55	Deg F	Average from data collected during observation phase
Current Average Return Temp	72	Deg F	Average from data collected during observation phase
Current Average Cooling Delta-T	19	Deg F	
Number of CRAH Units (test area)	3		From CAD data center layout
Number of CRAH Units (entire data center)	19		From CAD data center layout

Deluxe System 3 Chilled Water CRAC Capacities

Entering DB Temp	Leaving DB Temp	Cooling Delta-T	Cooling Capacity
Deg F	Deg F	Deg F	kBTUh
75	55.8	19.2	410.5
80	56.7	23.3	471.6
85	57.7	27.3	534.1

Data from Mark Daley of Liebert

Calculated Values

Linear line fit from provided data	15.258x + 117.07	
Current Cooling Capacity	407.0	kBTUh

Cooling Capacity Calculations

Cooling Delta-T	% Capacity Increase over Current Operation
20	4%
27	30%
32	49%
35	60%

Energy Calculations	Test Area	Entire Data Center	Units	Comments
Current Power Use	18.9	119.7	kW	Assumes 19 CRAHs operating. Power use per CRAH based on data collected from CRAH-23 during normal operation
Power use with 30F Delta-T	5.6	35.5	kW	Assumes 19 CRAHs operating. Power use per CRAH based on data collected from CRAH-23 when the measured cooling delta-T was approximately 30F.
Peak Power Reduction	13.3	84.2	kW	Current power use minus power use with 30F Delta-T
Yearly Energy Reduction	116,508	737,884	kWh	Assumes 24/7 operation.

Increase Chilled Water Delta-T

Parameters	Value	Units	Comments
Chilled Water Supply Temperature	51	deg F	Measured
Chilled Water Return Temperature	65	deg F	Measured
Chilled Water Delta-T	14	deg F	CHWRT-CHWST
% of Total Area Occupied by Test Area	7%		From drawings provided by LBNL
Test Area Rack Power Use	270	kW	Measured
Total Internal Load	1,075	tons	This value was obtained by calculating a power density for the test area (rack power input divided by floor area), and then applying that same density to the entire data center floor area
Chilled Water Pumping Power	22	w/gpm	ASHRAE 90.1, Table 11.4.3A, note 5.
Yearly Hours of Operation	8,760	hrs	24/7 operation

Pipe Capacity Calculations

Chilled Water Delta-T	% Pipe Capacity Increase	Test Area Capacity	Total Data Center Capacity
		tons	tons
15	7%	82	1,152
20	43%	110	1,535
25	79%	137	1,919

Energy Calculations

Chilled Water Delta-T	Flow Rate	Chilled Water Pumping Power	Yearly Energy Use
	gpm	kW	kWh
14	1,842	40.5	355,083
15	1,720	37.8	331,411
20	1,290	28.4	248,558
25	1,032	22.7	198,846

Increase Chilled Water Supply Temperature

Parameters

Average Chiller Efficiency	0.57	kW/ton
Chiller Efficiency Increase for every 1 deg F increase in CHWST	1%	
Heat Exchanger Approach Temperature	2	deg F
Chilled Water Delta-T	12	deg F
Proposed Chilled Water Supply Temperature	50	deg F
Internal Load	1,075	tons
Chilled Water Flow Rate	2,150	gpm
Annual Hours of Operation	8,760	hrs

Analysis

		Chilled Water Supply Temperature (deg F)				
	Units	46	48	50	52	54
Chilled Water Flow Rate	GPM	2,150	2,243	2,336	2,430	2,523
Average Chiller Efficiency	kW/ton	0.57	0.56	0.55	0.54	0.52
Average Chiller Power	kW	613	600	588	576	564
Estimated Annual Energy Use	kWh	5,366,591	5,259,260	5,151,928	5,044,596	4,937,264
% Piping Capacity Decrease		0%	-4%	-9%	-13%	-17%
% Energy Savings		0%	2%	4%	6%	8%

Utilize Water Side Economizing with Increased Chilled Water Supply Temperature

Parameters

Average Chiller Efficiency Efficiency	0.57	kW/ton
Chiller Efficiency Increase for every 1 deg F increase in CHWST	1%	
Heat Exchanger Approach Temperature	2	deg F
Chilled Water Delta-T	12	deg F
Proposed Chilled Water Supply Temperature	50	deg F
Internal Load	1,075	tons
Chilled Water Flow Rate	2,150	gpm

Results	No Econo.	Chilled Water Supply Temp. (deg F) 46	48	50	52	54
Hours Available for Water Side Economizing	N/A	1,976	2,807	3,819	4,875	5,994
Estimated Yearly Energy Use (kWh)	5,366,591	4,940,436	4,615,996	4,223,171	3,759,362	3,230,948
% Energy Savings	0%	8%	14%	21%	30%	40%

Hourly Analysis

Oakland weather data

Date	Hour	Drybulb Temp	Dewpoint Temp	Wetbulb Temp	Water Temp from CT	Economizing Mode: 1=Full, 2=Integrated, 3=None — Chilled Water Supply Temperature 46	48	50	52	54	Additional Cooling Required from Chiller — Chilled Water Supply Temperature 46	48	50	52	54	Chiller Power — Chilled Water Supply Temperature 46	48	50	52	54	No Economizing
		°F	°F	°F	°F	°F	°F	°F	°F	°F	tons	tons	tons	tons	tons	°F	°F	°F	°F	°F	
1-Jan	1	42.8	42	42.2	52.2	2	2	2	2	1	554	375	196	16	0	316	209	107	9	0	613
1-Jan	2	41.5	40	40.7	50.7	2	2	2	1	1	420	240	61	0	0	239	134	34	0	0	613
1-Jan	3	39.9	38	39.3	49.3	2	2	1	1	1	291	112	0	0	0	166	63	0	0	0	613
1-Jan	4	39.0	38	38.5	48.5	2	2	1	1	1	226	47	0	0	0	129	26	0	0	0	613
1-Jan	5	39.6	37	38.3	48.3	2	2	1	1	1	202	23	0	0	0	115	13	0	0	0	613
1-Jan	6	42.1	38	40.1	50.1	2	2	2	1	1	365	186	7	0	0	208	104	4	0	0	613
1-Jan	7	43.5	37	40.7	50.7	2	2	2	1	1	418	239	60	0	0	238	134	33	0	0	613
1-Jan	8	45.1	37	41.4	51.4	2	2	2	1	1	486	307	128	0	0	277	172	70	0	0	613
1-Jan	9	46.9	37	42.2	52.2	2	2	2	2	1	555	376	197	18	0	317	210	108	10	0	613
1-Jan	10	48.6	37	42.9	52.9	2	2	2	2	1	622	443	264	85	0	355	247	144	45	0	613
1-Jan	11	52.0	36	44.4	54.4	2	2	2	2	2	749	569	390	211	32	427	318	214	113	17	613
1-Jan	12	55.6	35	45.8	55.8	2	2	2	2	2	874	695	516	337	158	498	388	282	181	83	613
1-Jan	13	59.0	34	47.0	57.0	2	2	2	2	2	985	806	627	448	269	562	450	343	240	141	613
1-Jan	14	57.7	33	45.9	55.9	2	2	2	2	2	889	709	530	351	172	506	396	290	188	90	613
1-Jan	15	56.5	31	44.8	54.8	2	2	2	2	2	789	610	431	252	72	450	341	236	135	38	613
1-Jan	16	55.2	29	43.6	53.6	2	2	2	2	1	677	498	319	139	0	386	278	174	75	0	613
1-Jan	17	53.1	27	41.7	51.7	2	2	2	1	1	510	331	152	0	0	291	185	83	0	0	613
1-Jan	18	50.7	24	39.6	49.6	2	2	1	1	1	327	147	0	0	0	186	82	0	0	0	613
1-Jan	19	48.6	22	37.7	47.7	2	1	1	1	1	154	0	0	0	0	88	0	0	0	0	613
1-Jan	20	47.3	21	36.8	46.8	2	1	1	1	1	75	0	0	0	0	43	0	0	0	0	613
1-Jan	21	46.4	20	36.2	46.2	2	1	1	1	1	17	0	0	0	0	10	0	0	0	0	613
1-Jan	22	[illegible]	[illegible]	[illegible]	[illegible]	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
1-Jan	23	[illegible]	[illegible]	[illegible]	[illegible]	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
1-Jan	24	[illegible]	[illegible]	[illegible]	[illegible]	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	1	[illegible]	[illegible]	[illegible]	[illegible]	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	2	[illegible]	[illegible]	[illegible]	[illegible]	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	3	37.6	28	33.7	43.7	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	4	35.1	26	31.4	41.4	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	5	36.3	28	32.9	42.9	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	6	37.6	30	34.3	44.3	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	7	38.8	32	35.9	45.9	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0	613
31-Dec	8	45.1	35	40.3	50.3	2	2	2	1	1	389	209	30	0	0	221	117	17	0	0	613
31-Dec	9	51.8	37	44.8	54.8	2	2	2	2	2	791	612	433	254	75	451	342	237	136	39	613
31-Dec	10	58.3	40	49.0	59.0	3	2	2	2	2	1,075	988	809	630	450	613	552	443	337	236	613
31-Dec	11	59.5	42	50.2	60.2	3	3	2	2	2	1,075	1,075	913	734	554	613	600	499	393	291	613
31-Dec	12	60.4	43	51.2	61.2	3	3	2	2	2	1,075	1,075	1,003	824	645	613	600	549	441	338	613

All Hours of the year are included in this analsys, but most have been omitted from this printout for brevity.

31-Dec	13	61.7	45	52.4	62.4	3	3	3	2	2	1,075	1,075	1,075	929	750	613	600	588	498	393	613
31-Dec	14	60.4	45	51.9	61.9	3	3	2	2	2	1,075	1,075	1,067	888	709	613	600	584	476	372	613
31-Dec	15	59.5	45	51.6	61.6	3	3	2	2	2	1,075	1,075	1,040	861	682	613	600	569	461	358	613
31-Dec	16	58.3	45	51.2	61.2	3	3	2	2	2	1,075	1,075	999	820	641	613	600	547	439	336	613
31-Dec	17	56.3	43	49.5	59.5	3	2	2	2	2	1,075	1,026	847	668	489	613	573	463	358	256	613
31-Dec	18	54.7	42	47.9	57.9	2	2	2	2	2	1,068	889	710	531	351	609	496	388	284	184	613
31-Dec	19	53.4	40	46.9	56.9	2	2	2	2	2	974	795	615	436	257	555	444	337	234	135	613
31-Dec	20	50.9	40	45.3	55.3	2	2	2	2	2	832	653	474	295	116	474	365	259	158	61	613
31-Dec	21	48.7	39	44.0	54.0	2	2	2	2	2	717	538	359	180	1	409	301	196	96	0	613
31-Dec	22	46.4	38	42.5	52.5	2	2	2	2	1	585	406	227	48	0	333	227	124	26	0	613
31-Dec	23	44.8	36	40.7	50.7	2	2	2	1	1	423	244	65	0	0	241	136	35	0	0	613
31-Dec	24	43.7	34	39.2	49.2	2	2	1	1	1	289	110	0	0	0	165	62	0	0	0	613
	Sum					24,206	23,302	22,135	20,879	19,425						4,940,436	4,615,996	4,223,171	3,759,362	3,230,948	5,366,591
	Ave.	57.4	49.0	52.7	62.7	2.8	2.7	2.5	2.4	2.2	989	943	881	801	703	564	527	482	429	369	613
	Max.	90.9	68.4	72.7	82.7	3.0	3.0	3.0	3.0	3.0	1,075	1,075	1,075	1,075	1,075	613	600	588	576	564	613
	Min.	34.2	11.1	29.8	39.8	1.0	1.0	1.0	1.0	1.0	0	0	0	0	0	0	0	0	0	0	613
	Econo. Hrs					1,976	2,807	3,819	4,875	5,994											

Appendix III – Test Area Photographs

Baseline Photographs

Image 1 – Entrance to the test area. The entire area was sealed from floor to ceiling, as well as in the under floor supply plenum.

Image 2 – CRAH-23 on the left, and CRAH-25 on the right. CRAH-23 is adjacent to the hot aisle of the test area.

Image 3 – Temperature sensors in the cold aisle.

Image 4 – Temperature sensors in the hot aisle.

Alternate 1 Photographs

Image 1 – Temperature sensors in the cold aisle. The cold aisle is sealed with plywood above the racks and covering the ends of the aisle.

Image 2 – Plywood roof sealing off the top of the cold aisle.

Image 3 – Sealed off cold aisle with temperature sensors at the end of each of the server racks.

Alternate 2 Photographs

Image 1 – Temperature sensors in the cold aisle. The cold aisle is sealed from the top of the server racks to the ceiling.

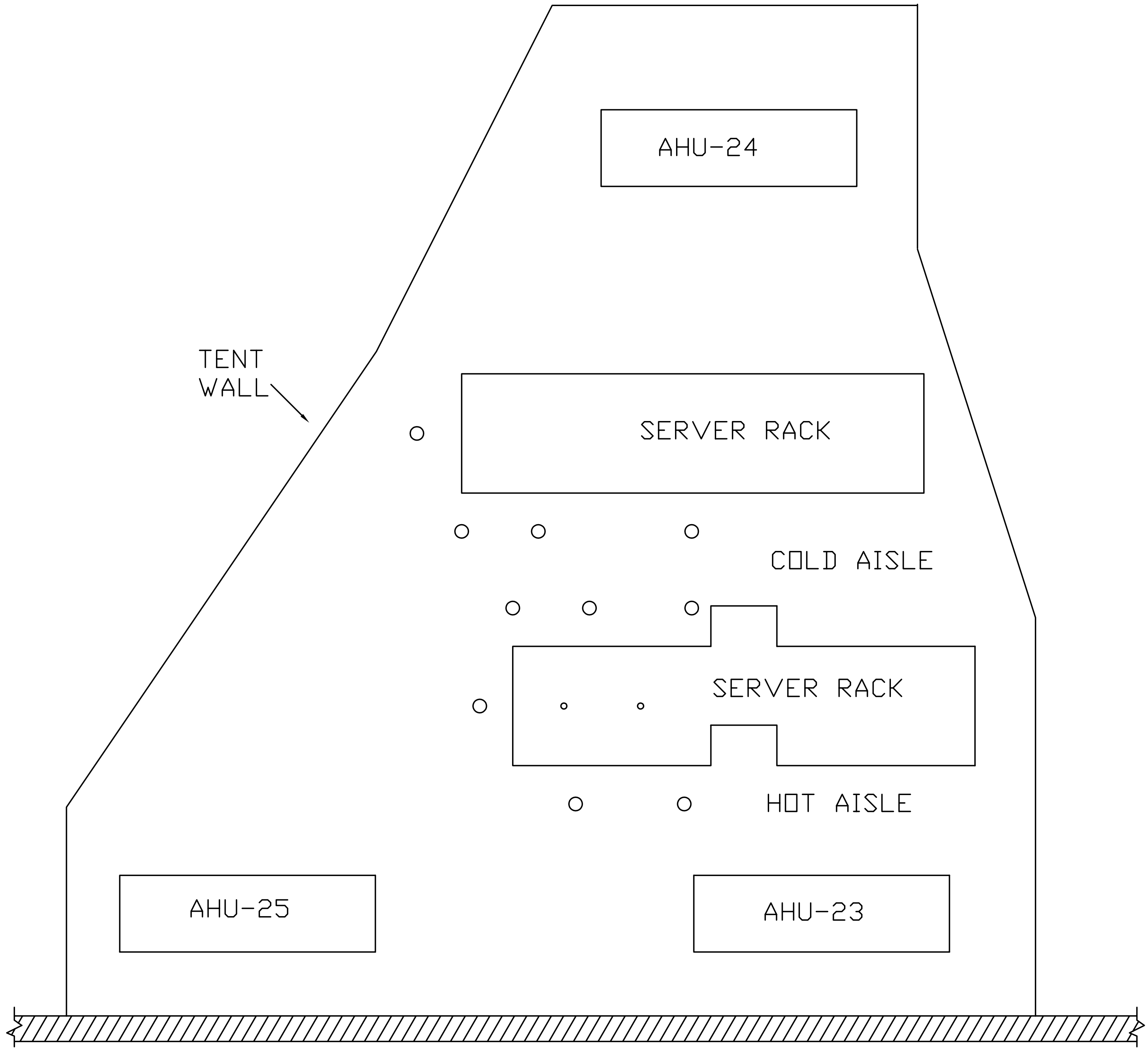
Image 2 – Sensors in the hot aisle and at the end of the server rack.

Image 3 – Open ceiling tiles used to route air to CRAH units. The interstitial space above the T-bar ceiling was used as a return air plenum.

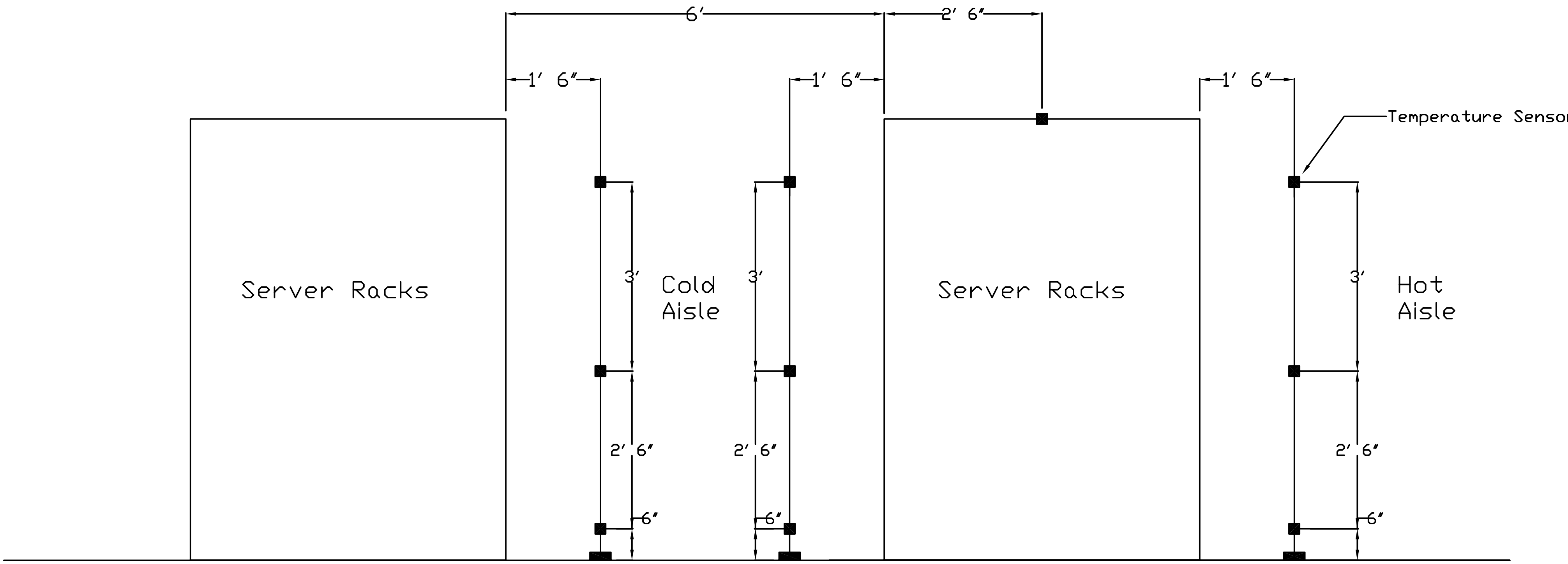
Image 4 – Temporary return air ducting for CRAH-25.

Appendix IV – Sensor Locations

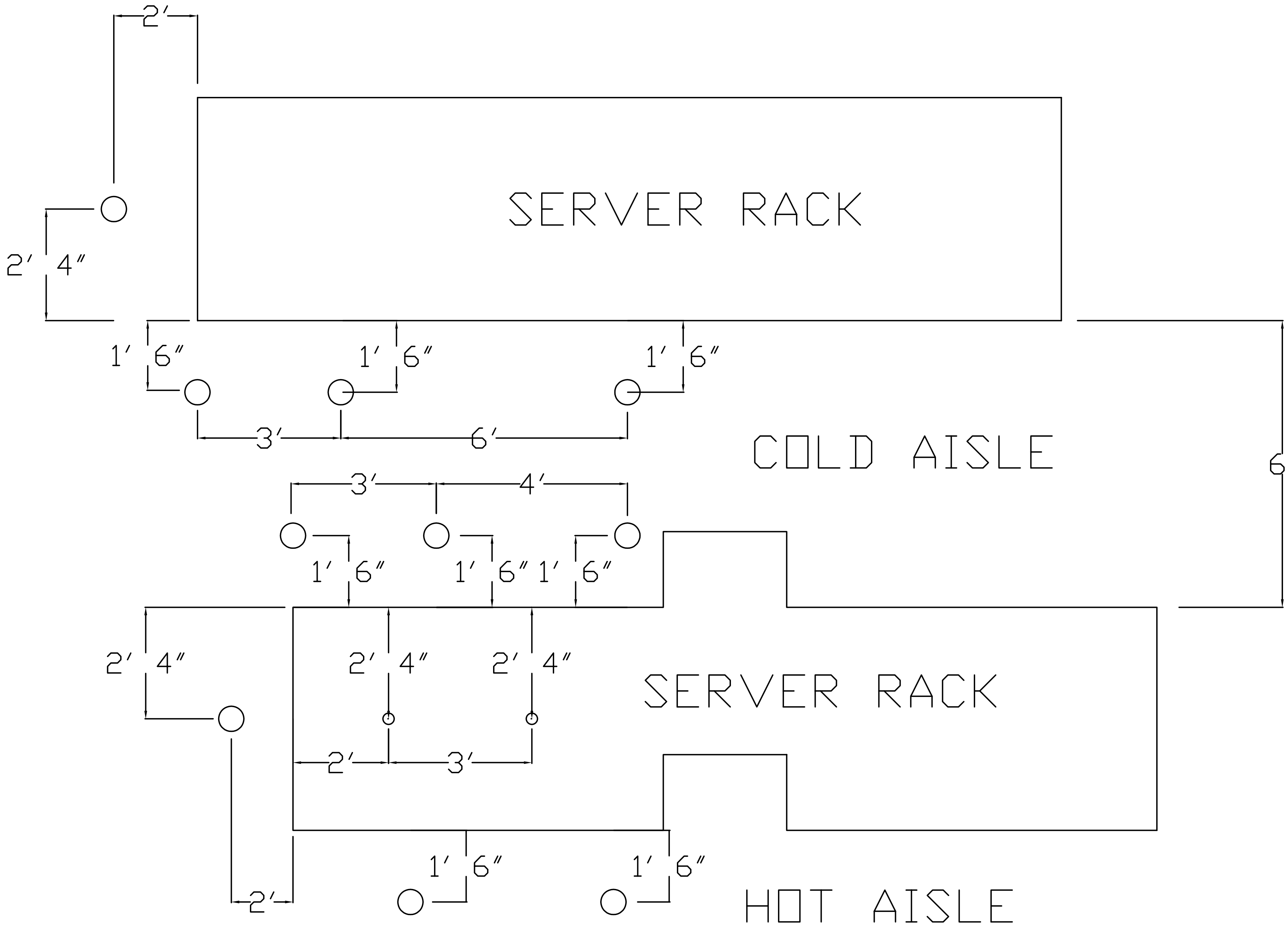
EXPERIMENT CONFIGURATION



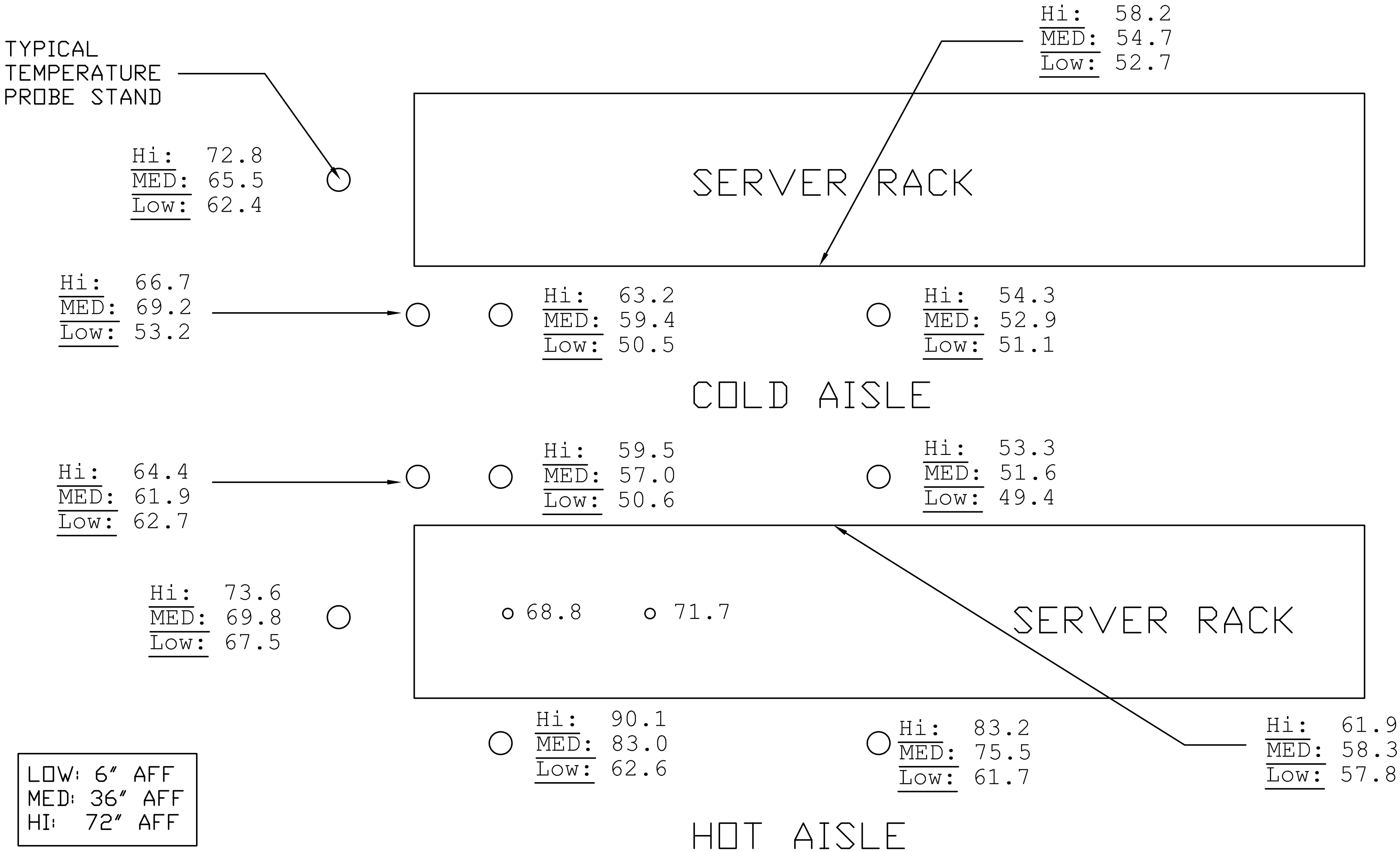
SENSOR LOCATIONS - ELEVATION VIEW



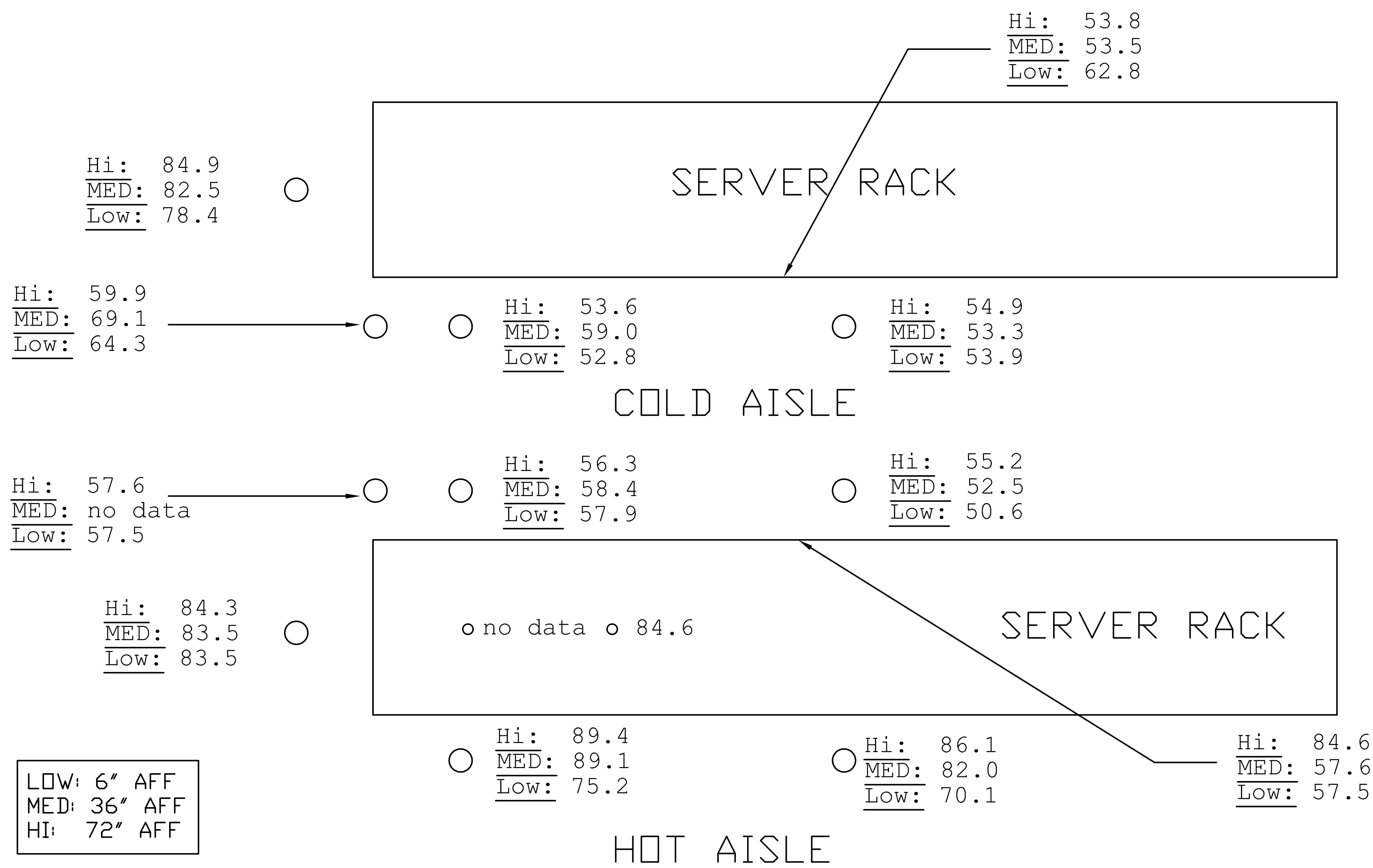
SENSOR LOCATIONS - PLAN VIEW



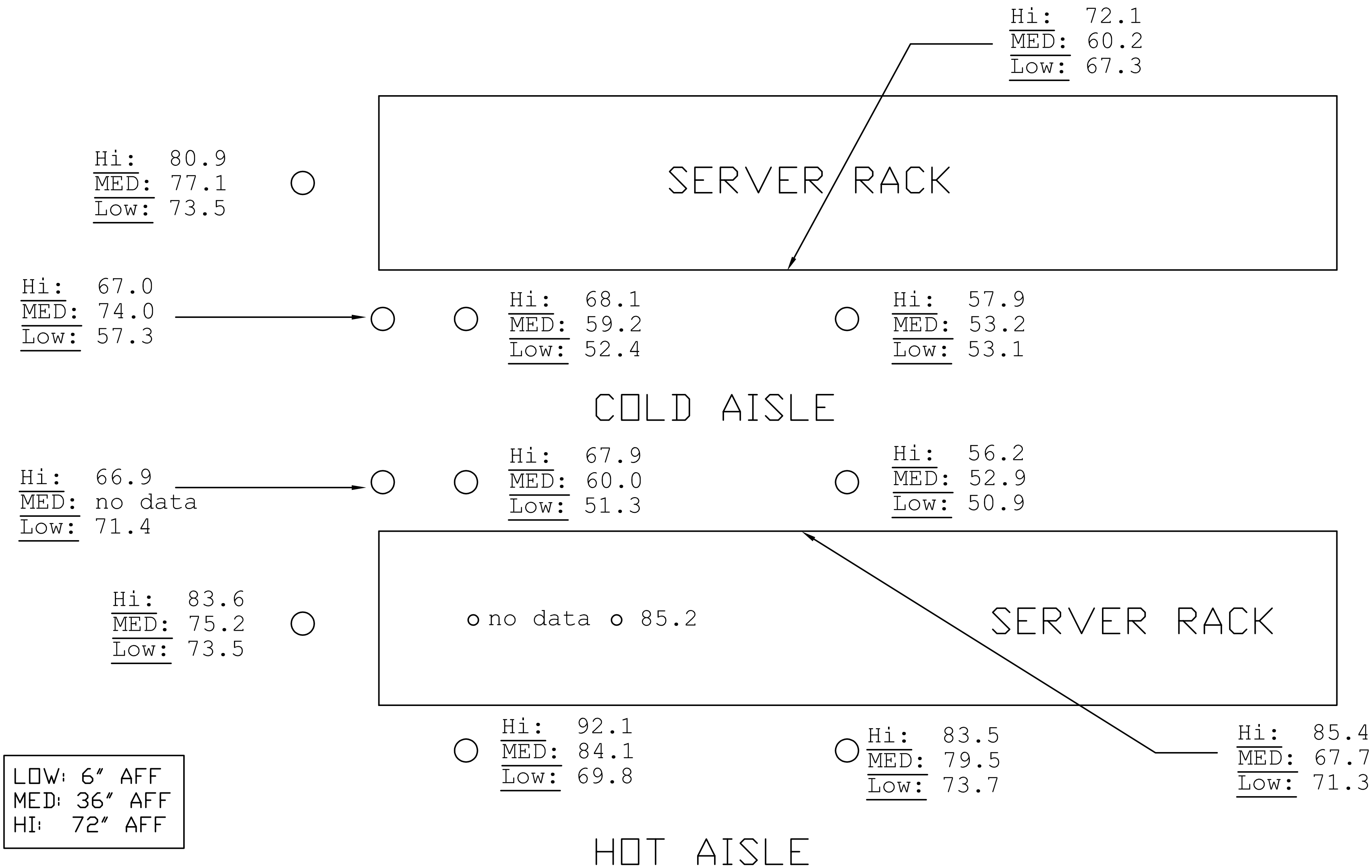
Temperature Measurements - Baseline



Temperature Measurements - Alternate 1



Temperature Measurements - Alternate 2



Appendix V – Revised Monitoring Plan

DATA CENTER AIR MANAGEMENT DEMONSTRATION

MEASUREMENT AND MONITORING PLAN

REVISED JULY 13, 2006

SPONSORED BY:

PACIFIC GAS & ELECTRIC

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Background

Due predominantly to their characteristically high internal cooling loads, data centers account for a significant electrical load in the Pacific Gas and Electric (PG&E) service territory. As such, PG&E has sponsored the "Data Center Emerging Technologies" program, which is targeted toward nurturing technologies that have the potential to reduce the energy consumption of these facilities, but have not yet gained universal acceptance among data center designers and operators. Participants in this program have identified several technologies that would be well-served by demonstration and monitoring projects, in order to show designers and operators the benefits of incorporating the technologies into their work.

Surveys conducted under the auspices of the Emerging Technologies program showed that the majority of data center professionals expressed interest in a demonstration project showing the benefits of thoughtful and efficient air flow management techniques in an operating data center. Lawrence Berkeley National Laboratory has asked Rumsey Engineers to assist in the demonstration, monitoring, and presentation of the results. The goal of this demonstration project will be to show designers, operators, and owners of data center facilities the potential increase in energy efficiency and/or cooling capacity that can be gained from improved air management techniques.

A demonstration site has been selected, and is to be located in a small section of an existing, large data center in downtown Oakland, California. According to the facility's web site, that subsection of the facility that will serve as the demonstration site contains the following computing equipment:

- 122 8-processor nodes (with 32GB memory each)
- 1.9 GHz POWER 5 processors
- 111 compute nodes (888 processors)
- 3.5 TB aggregate memory on compute nodes
- 7.6 GFlops/sec peak processor speed
- 6.7 TFlops theoretical peak system performance
- 100 TB of usable disk space in GPFS
- 2 login nodes
- 6 nodes supporting the General Parallel Filesystem (GPFS)
- Nodes configured to use 16 GB "Large Page" memory

Cooling equipment serving the demonstration area consists of two 40-ton Liebert downflow computer room air handling (CRAH) units with chilled water cooling coils. The chilled water is provided by a central chiller plant, and the supply air is delivered via a 36" raised floor plenum. The racks are separated into hot and cold aisles, and the return air is not ducted, but relies upon stratification to make its way back to the intake of the CRAH units, as shown in Figure 1.

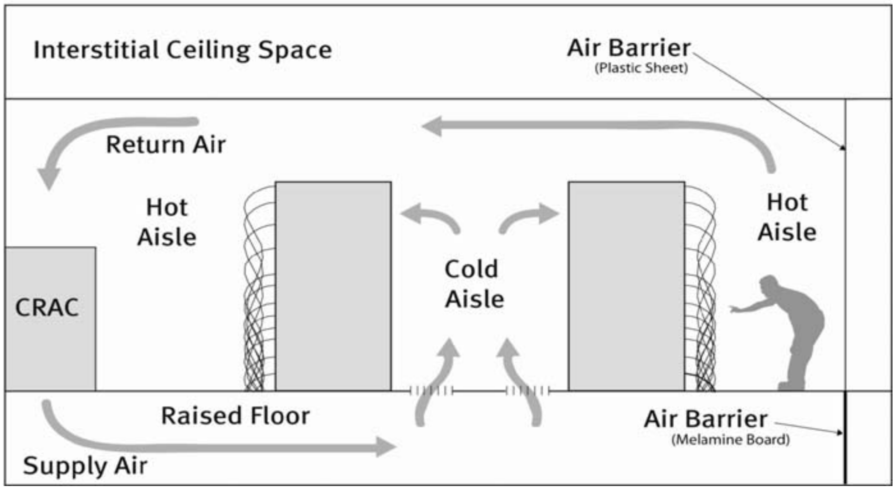

Figure 1: Baseline Air Flow Configuration

Preliminary Preparations

Construction of Air Barriers

The demonstration area is not currently separated physically from the remainder of the large data center in which it is contained. One of the first steps will be to construct an air barrier around the entire demonstration area, separating it from the rest of the data center. This barrier will be from floor to ceiling, as well as underfloor (but not in the interstitial ceiling space), and will completely enclose the demonstration area and the two CRAH units. It is recommended that the underfloor barrier be easily removable such that in the event of a sudden loss of air from the dedicated CRAH units, the barrier can be quickly removed to allow for access to the remainder of the underfloor cooling air from the rest of the data center. Facilities personnel are working with a contractor to construct this and subsequent air barrier systems for the demonstration. Care will be taken in the construction of the barriers so that barrier materials do not restrict airflow through the racks.

User-Operability of CRAH Fan VFDs

The Liebert Units are equipped with variable frequency drives (VFDs) on the fan motors, but the controls are internal to the unit and factory-wired. This means the user can't manually adjust the fan speed using the VFD as they are currently configured. Facilities personnel are working on re-wiring the VFD controls to ensure user-operability. It is imperative for the demonstration that fan motors be able to be manually adjusted.

Determination of Cooling Load

Facility personnel measured the rack load on May 30, 2006, and the result was approximately 230 kW. This translates into a cooling load of approximately 65 tons, meaning more than one of the 40 ton CRAH units will be needed. Spot checks of the measured load will be made throughout the demonstration in case the load changes appreciably. Regardless of the measured cooling load and determination of equipment needed, **there will be no computing down time required for any aspect of this demonstration project.**

Determination of Temperature Limit

The demonstration facility staff and users have informed Rumsey Engineers that high temperature limits are forthcoming from the computer manufacturer. In other words, the manufacturer will be providing the upper limit of safe operating temperatures for the space and equipment. Once determined, this limit will guide the demonstration such that any changes in operational setpoints will ensure that the upper limit shall not be reached. Additionally, the manufacturer may be able to provide information on relative humidity tolerances if necessary, and these too will be incorporated into the constraints.

Baseline Configuration

The baseline configuration to which all measurements will be compared is illustrated in Figure 1, and includes the underfloor supply air from the CRAH unit(s), the stratified unducted return air back to the CRAH unit(s), and the air barriers enclosing the entire demonstration area. The baseline configuration will not entail any added separation of hot and cold aisles in comparison to the existing conditions. With this configuration in place, the following system parameters will be measured in a single day:

- IT equipment power
- Temperature distribution throughout the space
- CRAH unit(s) supply air temperature
- CRAH unit(s) return air temperature
- Chilled water supply temperature
- Chilled water return temperature
- CRAH fan energy
- Supply and return air volumetric flow rates

Measurements/Monitoring

Power will be measured using a power meter suitable for 480V/3ph/60Hz electrical power. For overall electrical load measurements at the racks (to determine cooling load), a power meter will be connected to the electrical panel that serves the rack area. For CRAH fan energy measurement, the meter will measure the total power draw by the unit, assuming that fan power dominates and the remainder of the electrical load from the unit is negligible (i.e. LCD readout, chilled water valve actuator, etc.)

Temperatures will be measured using portable data loggers and temperature sensors (thermistors), Chilled water supply and return temperatures will be monitored and trended by placing thermistors directly upon the outer pipe surface and covered with insulation. Air temperatures will be measured by placing temperature sensors underfloor for supply air, at the CRAH intake for return air, and scattered throughout the hot and cold aisles to measure temperature distribution throughout the space. Supply and return air flow rates will be measured using a Shortridge AirData Multimeter.

The results of the measured baseline parameters will then be used as a benchmark to compare the same parameters throughout subsequent configurations of the air flow demonstration.

Alternate 1 Configuration

The Alternate 1 air management test will involve making a concerted effort to "seal off" the cold aisle, which should serve to decrease the degree of mixing between hot and cold aisles, and hence increase the temperature difference between the supply and return air. This larger "Delta-T" enables the data center operators to serve the same cooling load while moving less air to do so, saving on fan energy. The sealing of the cold aisle will be accomplished by placing an air barrier at the top (and sides) of the racks as shown in Figure 2. The spaces between racks have already been blocked-off, so that with the added barriers on top and sides of the cold aisle, the path of least resistance for the cooling air will be through the computer equipment as intended.

The increased air Delta-T can be accomplished by raising the return air temperature setpoint to maintain the original underfloor supply air temperature found in the baseline configuration. Once this is achieved, the delta-T should be higher than before, but it will likely be able to be increased further and still stay within reasonable bounds of safe operating temperatures (to be provided by the manufacturer). Maximizing this temperature difference minimizes the necessary fan energy, and the optimal point will be one in which the delta-T is the highest possible, but still within the operational limitations of the computer equipment.

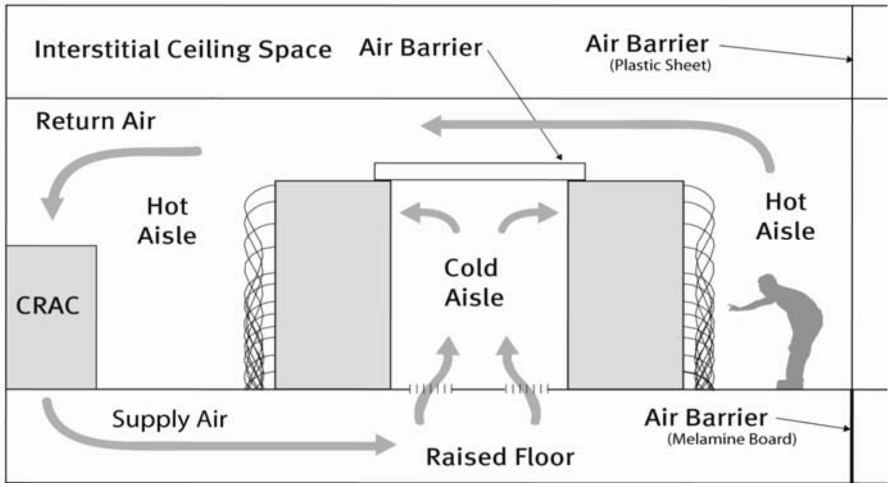

Figure 2: Alternate 1 Air Flow Configuration

Once the Alternate 1 configuration has been installed by the barrier contractor, the CRAH fans will be slowed down to a minimum level that does not affect server performance. Once the fan speed has been reduced, another series of measurements will be undertaken (same parameters as in the baseline case) and the results compared to the baseline case. The measurements will be taken in a single day.

Alternate 2 Configuration

The Alternate 2 air management test will attempt to "encourage" the hot return air back to the CRAH unit(s) by utilizing the interstitial ceiling space as a return path. Ceiling tiles above the hot aisles and CRAH unit(s) will be removed, and a "hood" will be fitted onto the CRAH air intake, as illustrated in Figure 3.

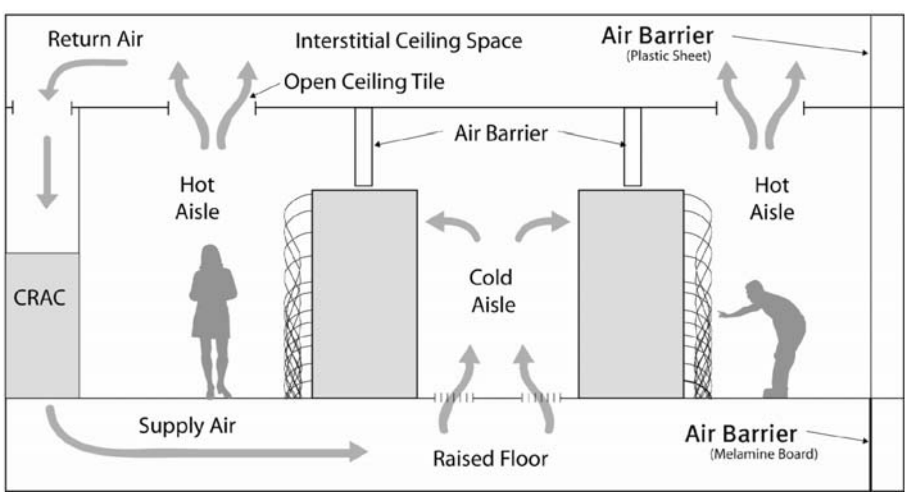

Figure 3: Alternate 2 Air Flow Configuration

As in Alternate 1, the CRAH fan speeds will then be reduced as far as safely possible. A third and final round of measurements will then be taken. After all

Alternates are complete, the results will be analyzed, and compared to the baseline configuration. All results will then be compiled and presented to PG&E.

Schedule

Week of May 22 – May 26, 2006

- Rumsey Engineers personnel visit demonstration site (by 5/23)
- Draft Monitoring Plan reviewed and commented upon (by 5/26)

Week of May 29 – June 3, 2006

- Measurement of IT equipment load at PDUs (by 5/30)
- Monitoring Plan finalized (by 5/30)
- Barrier constructed around demonstration area (by 6/3)
- CRAH fan VFD user-operability achieved (by 6/3)

June 12 – June 16, 2006

- Measurement/monitoring of baseline configuration (6/12)
- Construction of Alternate 1 configuration (6/13)
- Measurement/monitoring of Alternate I configuration (6/14)
- Construction of Alternate 2 configuration (6/15)
- Measurement/monitoring of Alternate 2 configuration (6/16)

June 19 – July 13, 2006

- Review and analysis of data
- Preparation of summary report in support of PG&E workshop
- Preparation of Powerpoint presentation in support of PG&E workshop (optional)