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Implications of Current Thermal Guidelines

For Data Center Energy Use

By **Mark Hydeman, P.E.**, Fellow ASHRAE

Most data centers are designed and operated according to existing industry guidelines for temperature and humidity control. These guidelines include ASHRAE Special Publication *Thermal Guidelines for Data Processing Environments*,¹ ANSI/TIA-942-1 2005 *Telecommunications Infrastructure Standard for Data Centers*² developed by the Telecommunication Industry Association (TIA), and the Network Equipment Building Systems (NEBS), which also is used by the telecommunication industry for telecom central office facilities. This article examines the impact of these guidelines on datacom facility efficiency and suggests ways to improve them.

Existing Thermal Guidelines

The current published recommendations for temperature and humidity control in data centers are shown in *Figure 1*. *Thermal Guidelines* defines *recommended* and *allowed* ranges for temperature and humidity as follows:

- **Recommended.** Facilities should be designed and operated to target the recommended range; and

- **Allowable.** Equipment should be designed to operate within the extremes of the allowable operating environment.

In addition, *Thermal Guidelines* addresses five classes of datacom facilities (*Table 1*). Class 1: Standard Data Center; Class 2: Information Technology (IT) Space, Office or Lab Environment; Class 3: Home or Home Office; Class 4:

Manufacturing Environment or Store; NEBS: Telecommunications Central Office Facility.

As shown in *Figure 1*, the recommended temperature and humidity envelope expanded in 2008. This change is significant as it increased the number of hours that a typical data center can be cooled using an economizer or evaporative (noncompressor) cooling. However, as discussed later, the upper and lower humidity limits severely handicap the operation of these efficient cooling technologies.

These specifications are defined at the inlet to the IT or data communications (datacom) equipment. A server typically has discharge temperatures 20°F to 60°F (11°C to 33°C) higher than the inlet.

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Thermal Guidelines' Impact on HVAC Energy Use

The 2008 changes to *Thermal Guidelines* significantly broadened the temperature range for recommended operation of a datacom facility. Therefore, what is the impact of raising the temperature of the air supplied to the datacom equipment in a datacom facility? The answer depends on the characteristics of the HVAC system. In general, HVAC system cooling capacity and efficiency increases. This is particularly true of systems that have air economizers, water economizers or evaporative cooling. Even without these technologies, a direct-expansion (DX) system uses less energy per unit cooling at a warmer supply air temperature setpoint due to a decrease in the compressor lift.

To simplify this discussion, I initially ignore the impact of elevated equipment inlet temperatures to the power draw of datacom equipment. As described later (in section "Effect of Server Inlet Temperature on Server Energy"), elevated inlet temperatures can increase the server fan speed (and energy) and also create leakage currents. The server energy and airflow do not increase significantly if we keep the server inlets at or below ~75°F (~23.9°C).

Figure 2 shows the impact of the outdoor air dry-bulb temperature (OAdb) on the cooling capacity and COP of an air-cooled DX unit.³ Although a decrease in outdoor air dry bulb is not precisely the same as the increase in supply air temperature, they are directly related as they both reduce compressor lift.* A decrease in outdoor air dry-bulb temperature reduces the operating condenser pressure and an increase in the supply air temperature increases the operating evaporator pressure. As shown in Figure 2, a 10°F (5.6°C) decrease in outdoor air dry bulb provides approximately a 5% increase in unit cooling capacity and an 11% increase in COP.

To emphasize the impact of supply temperature on energy efficiency of data centers, I analyzed bin data for four of the 16 ASHRAE Standard 90.1 climate zones: Washington, D.C., in Climate Zone 4A (mild and moist); Salt Lake City in Climate Zone 5B (cold and dry); Houston in Climate Zone 2A (hot and moist); and San Jose, Calif., in Climate Zone 3C (moderate and marine). According to the U.S. Department of Energy, these four ASHRAE climate zones represent approximately 91.4% of all commercial building new construction and renovations in the period 2003–2007.⁴

Table 2 presents the energy savings from air-side economizers on air-cooled direct-expansion CRAC units and chilled water CRAH units served by water-cooled chillers in the four climates

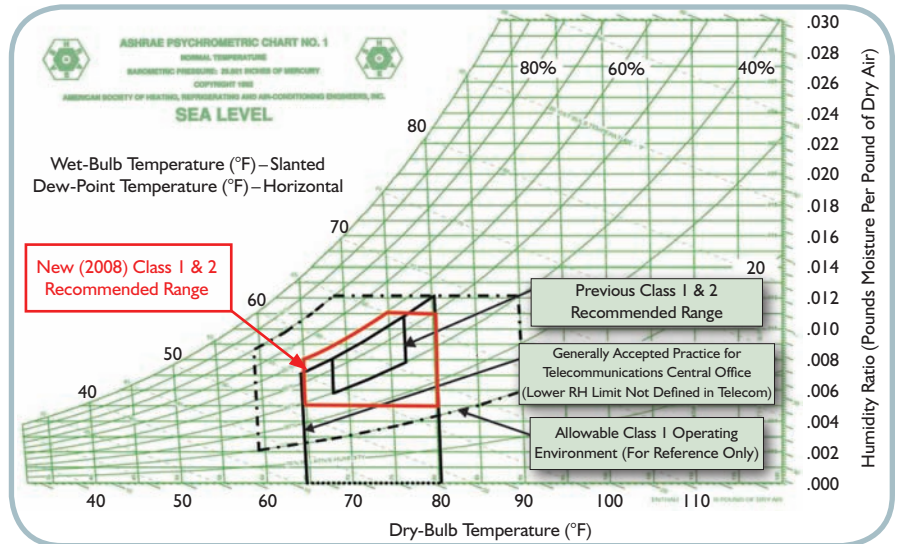


Figure 1: ASHRAE temperature & humidity guidelines for data centers and telecom facilities.

Class	Dry-Bulb Temperature		Percent Relative Humidity And Dew-Point Temperature	
	Allowable	Recommended	Allowable	Recommended
1	59°F–90°F	64°F–81°F	20%–80%	42°F DP–60% RH and 59°F DP
2	50°F–95°F			
3	41°F–95°F	N/A	8%–80%	N/A
4	41°F–104°F	N/A	8%–80%	N/A
NEBS	41°F–104°F	64°F–81°F	Maximum of 85%	Maximum of 55%

Table 1: Thermal guidelines for each of the five classes of datacom facilities.

for our study. This data came from computer simulations that were done for the U.S. Department of Energy's DC-PRO profiling tool for data centers (<http://tinyurl.com/34qavoz>). The simulations were calibrated to a manufacturer's CRAC and CRAH units and followed the baseline model assumptions from Chapter 11 of Standard 90.1 (Energy Cost Budget Method) for the chilled water plant equipment. The simulations presented in this table did not include humidification.

The data in the table shows the percent of the HVAC energy (including fans, pumps and compressors) that was saved through the addition of an air-side economizer. For the air-cooled CRAC units, the economizers saved between 19% and 63% of the HVAC energy. The largest savings were in Climate Zones 5B (cold and dry) and 3C (moderate and marine), and the smallest savings were in Climate Zone 2A (hot and moist). For the chilled water CRAH units served by a water-cooled chilled water plant, the air-side economizers saved between 12% and 37% of the HVAC energy. The greater savings for air-cooled DX is a reflection that this equipment is far less efficient than chilled water CRAH units with a water-cooled chilled water plant.

*Compressor lift is the difference between the evaporator and condenser refrigerant pressures.

Using bin analysis, *Table 3* shows the potential hours that an air-side economizer could be 100% effective in a data center as a function of temperature and humidity constraints. For each climate, *Table 3* gives the percentage of annual hours that the outdoor air dry-bulb temperature was less than or equal to both 70°F and 80°F (21°C and 27°C) as a function of four humidity constraints as described here and detailed in the left column of *Table 3*:

- No limits on either upper or lower humidity;
- The recommended lower humidity limit in *Thermal Guidelines*, but no constraint on the upper humidity limit;
- Both the recommended upper and lower humidity limits based on the dew-point temperatures in *Thermal Guidelines*, but not the 60% upper RH limit; and
- The full recommended upper and lower humidity limits based in *Thermal Guidelines*.

Figure 3 displays the dry-bulb bin data for each of these climates. Examination of *Table 3* shows the impact of the supply air temperature setpoint and the recommended humidity constraints on the performance of air-side economizers in each of the four climates. For example, looking at the climate of Washington, D.C., a supply air temperature of 70°F (21°C) can use free cooling approximately 76% of annual hours without any humidity constraints. However, this reduces to 3% if you impose the full recommended humidity constraints in *Thermal Guidelines*. If you increase the supply air temperature setpoint to 80°F (27°C), you can use free cooling approximately 91% of the time without any humidity constraints and 6% of the time with the full recommended humidity constraints. As you increase the design supply air temperature from 70°F to 80°F (21°C to 27°C) in a data center with an air-side economizer in Washington, D.C., that doesn't use the ASHRAE humidity limits, you get 15% more hours of free cooling without humidity constraints (91% to 76%). In any of the climates, regardless of the supply air temperature setpoint, imposition of the ASHRAE recommended humidity constraints greatly reduces the effectiveness of an air-side economizer.

This bin analysis is somewhat simplistic and does not fully account for the following issues: it does not present the full picture of the economizer's effectiveness as you get partial cooling benefit when the outdoor air is above the supply air temperature setpoint but below the return air temperature

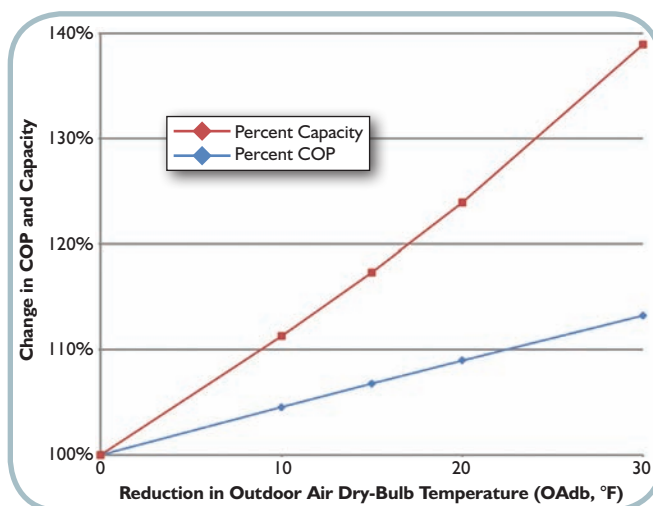


Figure 2: Effect of OA temperature on AC unit capacity and efficiency.

Climate	Percent of HVAC Energy Savings	
	Air-Cooled DX CRAC Units	Chilled Water CRAH Units With Water-Cooled Chillers
Washington, D.C. 4A	43%	25%
Salt Lake City 5B	54%	31%
Houston 2A	19%	12%
San Jose, Calif. 3C	63%	37%

Table 2: Analysis of air-side economizer savings.

(which is typically 10°F to 30°F [5.6°C to 17°C] above the inlet temperature to the servers in most data centers); in practice the supply air temperature at the CRAC or CRAH units must be lower than the rack inlet temperatures due to the mixing of air from the hot aisles to the cold aisles; and, the addition of the economizer dampers and filtration increase the fan energy. This exercise is illustrative and by examining this data we can draw three important conclusions:

- The potential for savings from air-side economizers is significant;⁵
- An increase in the supply air temperature increases the hours of “free” cooling if you have an outside air economizer; and,

	Washington, D.C. 4A		Salt Lake City 5B		Houston 2A		San Jose, Calif. 3C	
	$T_{db} \leq 70^\circ\text{F}$	$T_{db} \leq 80^\circ\text{F}$	$T_{db} \leq 70^\circ\text{F}$	$T_{db} \leq 80^\circ\text{F}$	$T_{db} \leq 70^\circ\text{F}$	$T_{db} \leq 80^\circ\text{F}$	$T_{db} \leq 70^\circ\text{F}$	$T_{db} \leq 80^\circ\text{F}$
No Upper or Lower Humidity Limits	76%	91%	79%	89%	47%	76%	88%	97%
Lower Humidity Limit: $T_{dp} > = 42^\circ\text{F}$	30%	45%	13%	20%	32%	60%	67%	76%
Both Upper and Lower Limits: $T_{dp} > = 42^\circ\text{F}$ and $T_{dp} < = 59^\circ\text{F}$	21%	25%	13%	20%	20%	24%	67%	75%
Both Upper and Lower Limits: $T_{dp} > = 42^\circ\text{F}$, $T_{dp} < = 59^\circ\text{F}$, and $\text{RH} < = 60\%$	3%	6%	7%	13%	2%	5%	10%	18%

Table 3: Dry-bulb bins analysis for the four ASHRAE climates.

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- The imposition of the upper or lower humidity limits dramatically decreases the number of hours of free cooling.

Adding an adiabatic humidifier (direct evaporative cooling or an ultrasonic humidifier) can significantly extend the hours of free cooling from the economizer while maintaining the lower humidity limit. Again, looking at the example of Washington, D.C., with a supply air temperature of 70°F (21°C), the system could have free cooling approximately 76% of annual hours without any humidity constraints or with an adiabatic humidifier and the lower humidity limits. The application of an adiabatic humidifier increases the site water use but greatly reduces cooling energy costs. The same is not true for nonadiabatic humidifiers (infrared or steam). They will increase the facility energy use if the economizer brings in cold dry air that is below the operating dew-point temperature of the humidifiers.

Table 4 and Figure 4 present a similar analysis of the wet-bulb temperature bins for these four climates. Table 4 presents the number of annual hours that the wet-bulb temperature is less than or equal to a threshold, which ranges from 80°F to 30°F (27°C to -1.1°C). Looking at Washington, D.C., the wet-bulb temperature is less than or equal to 80°F (27°C) 100% of the time. By contrast, the wet-bulb temperature for Washington, D.C., is less than or equal to 30°F (-1.1°C) only 15% of the time. This data is relevant to the operation of both evaporative cooling and water-side economizers. In laboratory testing, direct evaporative coolers have been shown to have effectiveness as high as 83% and indirect/direct evaporative coolers as high as 103%.⁶ At an effectiveness of 100%, an evaporative cooler supply temperature is equal to the wet-bulb temperature. Table 4 shows that an evaporative cooler with 100% effectiveness can provide 100% of the cooling in all four climates at 80°F (27°C), but this drops off to 62% for Houston and 89% for Washington, D.C., at a supply air temperature of 70°F (21°C). Benefits of evaporative cooling include air-washing of dust and gaseous contaminants and redundancy to the cooling systems.⁷

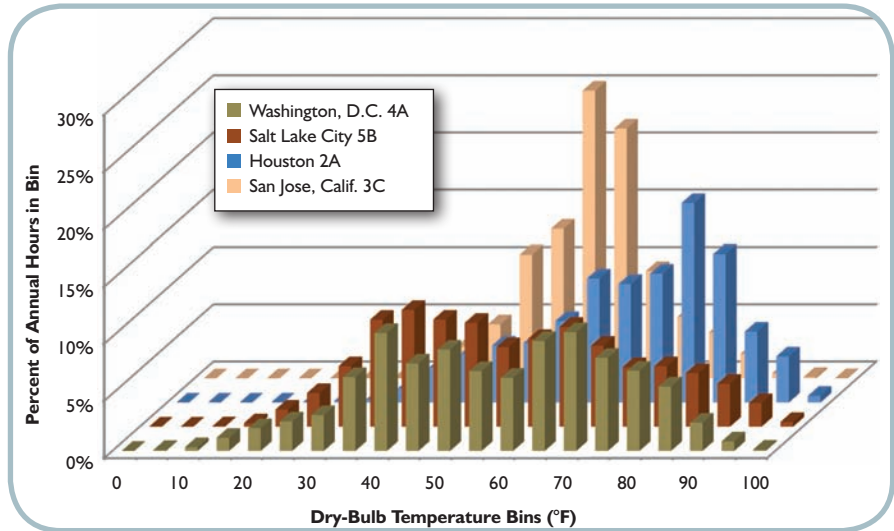


Figure 3: Dry-bulb bins for the four ASHRAE climates.

	Washington, D.C. 4A	Salt Lake City 5B	Houston 2A	San Jose, Calif. 3C
$T_{wb} \leq 80^\circ\text{F}$	100%	100%	100%	100%
$T_{wb} \leq 70^\circ\text{F}$	89%	100%	62%	100%
$T_{wb} \leq 60^\circ\text{F}$	66%	87%	36%	87%
$T_{wb} \leq 50^\circ\text{F}$	50%	32%	20%	32%
$T_{wb} \leq 40^\circ\text{F}$	33%	5%	7%	5%
$T_{wb} \leq 30^\circ\text{F}$	15%	0%	1%	0%

Table 4: Wet-bulb bin analysis for the four ASHRAE climates.

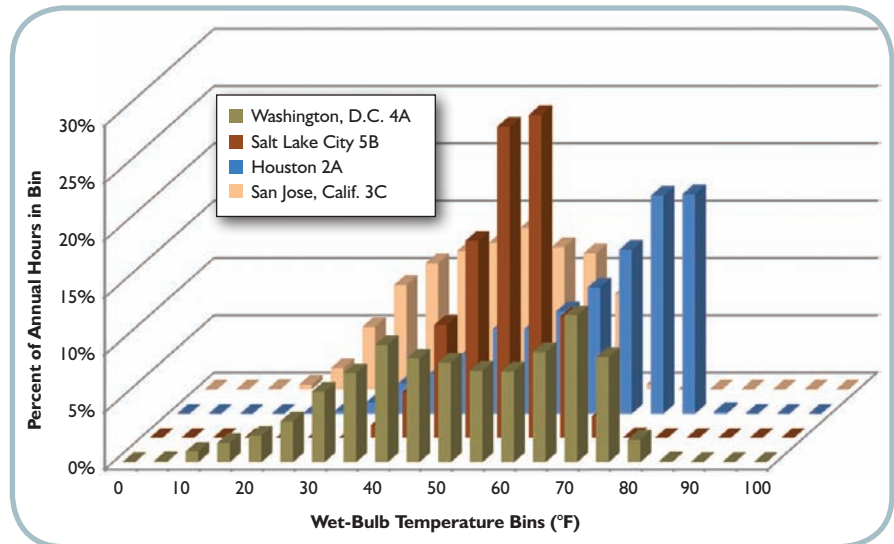


Figure 4: Wet-bulb bins for the four ASHRAE climates.

The analysis of water-side economizers is a bit trickier as their performance is a function of the outdoor wet-bulb temperature, the cooling tower design approach temperature, the heat exchanger design approach temperature and the return water temperature. It has already been established that the potential savings from use of water-side economizers in data centers is

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significant: A recent *ASHRAE Transactions* paper showed ~30% HVAC energy savings in El Paso, Texas, (Climate Zone 3B) and almost 50% in Helena, Mont., (Climate Zone 6B).⁸ The key issue is that a warmer supply air temperature increases the savings from a water-side economizer as it increases the chilled water return temperature from the CRAH units.

Using an NTU (number of transfer units)⁹ effectiveness coil model (or a manufacturer's coil selection program), you can establish the impact of supply air temperature on the chilled water flow and return temperature. The results using an NTU effectiveness model are shown in *Table 5*. The assumptions in this model include:

- 330,000 Btu/h (97 kW) chilled water coil;
- 44°F (6.7°C) chilled water supply; and
- 25°F (14°C) ΔT on the air.

Using the same coil for supply air temperatures of 70°F or 80°F (21°C or 27°C) results in leaving water conditions of around 60°F (16°C) for the 70°F (21°C) supply air temperature and around 65°F (19°C) for the 80°F (21°C) supply air temperature.

As discussed in the symposium paper,⁸ a warmer chilled water temperature yields more savings from the water-side economizer. In addition to the increased use of the water-side economizer, the pumping energy is reduced as the coils require less flow at the warmer supply air temperature (almost a 25% decrease in flow in the example in *Table 5*).

Potential Barriers to Energy Savings

I have established that raising the supply air temperature in a data center theoretically can lead to significant energy savings. However, to successfully raise the supply air temperature and realize the savings in a real data center there are

Dry-Side		
Airflow	12,000 cfm	12,000 cfm
RAT	95°F	105°F
SAT	70°F	80°F
Q_{tot}	330,000 Btu/h	330,000 Btu/h
Q_{sen}	330,000 Btu/h	330,000 Btu/h
Wet-Side		
Water Flow	41.25 gpm	30.75 gpm
CHWS	44°F	44°F
CHWR	60°F	65°F
Q_{tot}	330,000 Btu/h	330,000 Btu/h

Table 5: Coil analysis for impact on CHWR from change in supply air temperature.

several important issues to address. These include the need for air management; the impact of warmer temperatures on server energy use; the impact of the humidity limits on free cooling; and the application of demand-based controls.

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Air Management

Lawrence Berkeley National Laboratories (LBNL) performed benchmarking of dozens of data centers.¹⁰ One metric of performance it has tracked is the ratio of fan supply airflow to the airflow through the servers by measuring the ΔT across the servers and CRAC or CRAH units. In most of the data centers that were evaluated, the servers were running about 25°F (14°C) while the CRAC or CRAH units averaged around 10°F (5.6°C). This indicates a 2.5-to-1 ratio in airflow and an avoidable excess use of fan energy. The reason for this overflow is generally attributed to several factors: designers being overly conservative; the IT load in the criteria for the data center design not being realized in the field; and facility operators addressing hot spots by speeding up HVAC fans (if using variable speed fans) or turning on standby units.

These issues are effectively addressed through airflow management including: adding blanking plates to racks to prevent recirculation of air from the hot aisles to the cold aisles; plugging leaks in the floor and between racks; rebalancing floor tiles and adding hot or cold aisle containment.¹¹ In a demonstration by LBNL at one of its supercomputer facilities (*Figure 5*), LBNL was able to reduce fan power by 75% and reduce variation in rack inlet temperatures by as much as 26°F (14°C).¹²

In *Figure 5* you see the three rack inlet temperature sensors: the yellow sensor is near the top of the rack, the pink sensor is in the middle of the rack and the blue sensor is located near the bottom of the rack. Horizontally, the figure is divided into three regions: on the left is the baseline or as-found condition; in the middle is a period of adjustment as they set up the cold aisle containment; and to the right is the period of cold aisle containment test. In the baseline condition the temperatures range from a low of approximately 50°F (10°C) to a high of around 76°F (24°C). By contrast, the temperatures of the three sensors are much closer together during the containment demonstration: they range between 50°F and 60°F (10°C and 16°C). As shown in this graph, the baseline measurements

did not meet the ASHRAE recommended temperature limits. By contrast, the cold aisle containment test (Alternate 1) could have increased its supply temperature by approximately 20°F (11°C) and still have been within the 2008 ASHRAE recommended thermal guidelines. Cold

aisle containment decreased the spread in temperatures between the three rack inlet sensors.

In addition to getting better temperature control at the racks, the researchers also were able to reduce the operating speed and power of the CRAH units.

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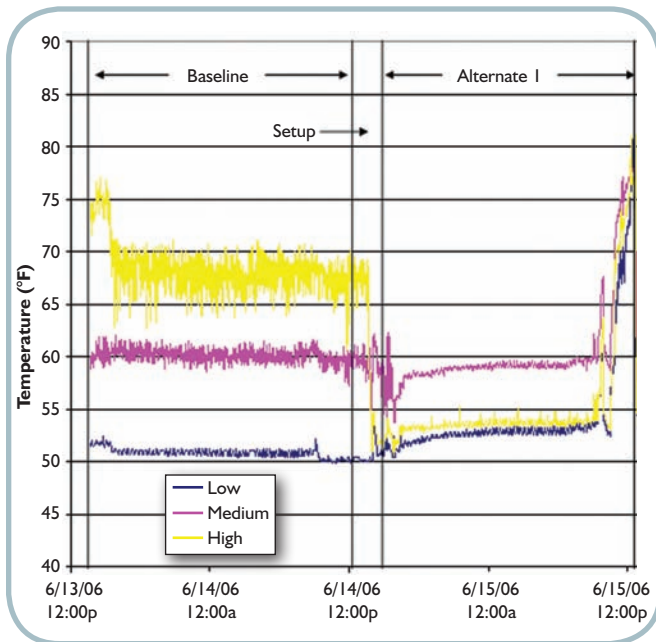


Figure 5: Rack inlet temperature distributions during containment tests.

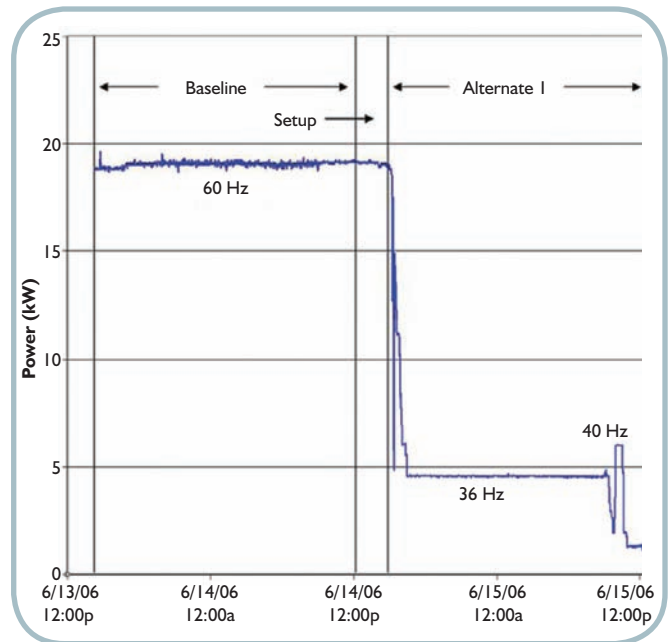


Figure 6: CRAH power and speed during containment tests.

Figure 6 shows that they were able to reduce the CRAH power by ~75% during the containment test.

Effect of Server Inlet Temperature to Server Energy

Another practical issue of warmer server inlet temperatures has to do with the way that servers are designed. Servers with either variable speed or two-speed fans demonstrate an increase in energy as the inlet temperatures increase. This increase in server energy is most pronounced as the server inlet temperatures go above approximately 77°F (25°C).¹³ In addition to the fans, “leakage currents” also can increase server power draw at elevated temperatures.¹³ In 2009 APC and Dell tested three types of servers in an environmental chamber to demonstrate the server power increase as a function of inlet temperature.¹³ Their test data corresponds to the data in Figure 7 from a field test by LBNL of server power as the server inlet temperature increases. As shown in Figure 7, the server energy increased nearly 14% as the inlet temperature increased from 70°F to 90°F (21°C to 32°C). Although this energy increase is significant, it is far less than the reduction in facility energy use from using air-side economizers, water-side economizers or evaporative cooling.

Server thermal designers can change the power response to inlet temperature through design. The fact that this equipment was traditionally designed for an inlet temperature of ~75°F doesn’t mean that it can’t be designed to safely operate at higher temperatures. Energy was relatively inexpensive when many of these server thermal designs were developed. Manufacturers of servers are now creating products that can operate in higher temperature environments. In the spring of last year, one manufacturer announced servers that are designed to operate in environments up to 104°F (40°C).¹⁴ At these temperatures it will be possible to cool most data centers completely with either unconditioned air or evaporative cooling.

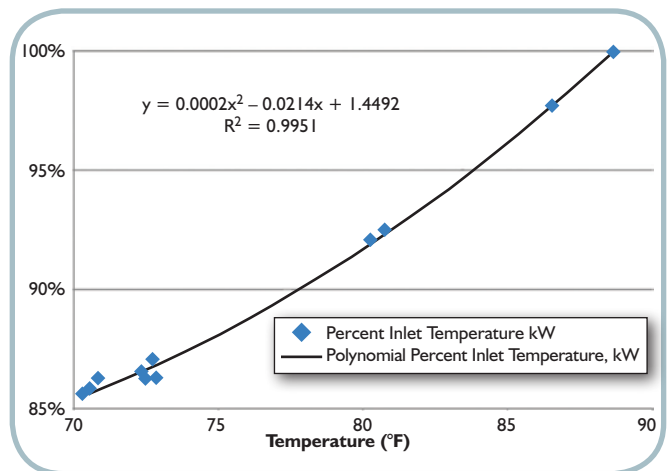


Figure 7: Field test of server energy as a function of inlet temperature from LBNL.

Server components are temperature sensitive and that thermal stress is a function both of absolute temperature and the rate of change. As we increase the temperatures in the data centers to save energy, we are cutting into the safety margins that the manufacturers have provided to protect their equipment.

Humidity Controls

As shown in Table 2, the imposition of the upper and lower humidity limits greatly reduces the effectiveness of air-side economizers in data centers. In the 2008 ASHRAE TC 9.9 publication on the expanded thermal guidelines, the authors defended the upper humidity limit on concerns of conductive anodic filament (CAF) growth, and the lower humidity on concerns about electrostatic discharge (ESD).¹⁵ Both of these issues are addressed in the March 2010 ASHRAE Journal article,

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*Humidity Controls for Data Centers: Are They Necessary?*¹⁶

As established in that article, the current humidity limits were set through industry consensus without the benefit of research or a formal review of failure analysis. Citing existing published research and industry standards, the article establishes that ESD is not effectively addressed by the use of humidity controls and that CAF formation does not occur below 80% RH, and only then if there are a number of coincidental environmental conditions present.

Variations in Server Loads and Airflow Requirements

A data center is a dynamic environment: the servers have variable loads and, in many cases, variable speed fans. To maintain inlet conditions and to maximize efficiency the HVAC cooling capacity and distribution have to change with the loads. The industry practice is moving from temperature sensors representing general conditions (often located in the CRAC or CRAH return) to distributed sensors representing rack inlet temperature (the criteria established in ASHRAE and EIA/TIA guidelines). These sensors are either external to the servers (often mounted on the doors at the rack inlet) or are internal to the servers and accessed through IT network protocols like SNMP.

Conclusions

There are considerable energy savings to be had from operating data centers at the upper end of the thermal envelope. For DX and chilled water systems, the warmer temperatures increase capacity and efficiency of the cooling systems. And, in chilled water systems, there are additional savings from the increase in coil ΔT that results in a reduction in pumping energy. For systems with air-side economizers, water-side economizers and evaporative cooling, the warmer temperatures increase the hours that these technologies can completely carry the load.

ASHRAE Technical Committee 9.9, Mission Critical Facilities, Technology Spaces and Electronic Equipment, took an important first step by expanding the upper end of the recommended thermal envelope from 77°F to 80.6°F (25°C to 27°C) in 2008. The next step should be the expansion or elimination of the humidity requirements. As established in this article, these greatly impair the potential savings from air-side economizers. We can achieve significant energy savings in data centers by operating closer to the upper recommended temperature limits but only if you are willing to violate the recommended humidity limits.

We would achieve even more savings if we could work together with the IT and data communications manufacturers to design their

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products and thermal management implementations for higher inlet temperatures. In an effort to realize these savings we need both research and cross industry collaboration. It is important to have a public forum for these broadly disparate industries to discuss and collaborate on these issues. It is the author's belief that the formation of a formal ANSI/ASHRAE Standard for these thermal guidelines would force this collaboration and would provide needed balance in the concerns of all of the affected parties including the IT equipment manufacturers, facility owners and operators, HVAC engineers, the research community, energy efficiency advocates, utility companies and HVAC manufacturers. Given the tremendous amount of energy used by datacom facilities worldwide this is an issue of concern to all of us.

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