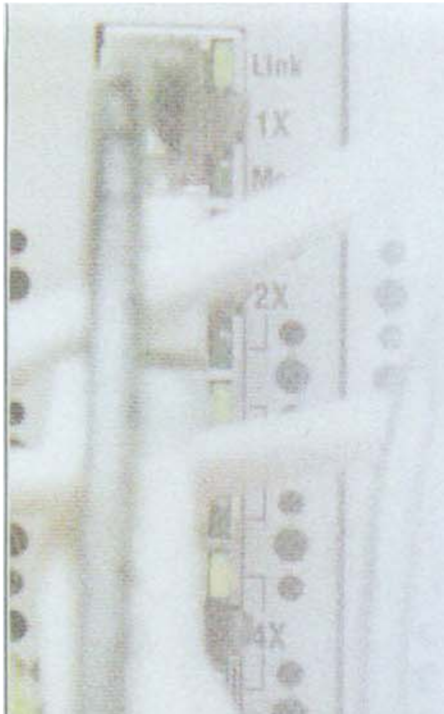

Efficiency Plus Reliability

By Christopher Kurkjian, P.E., Member ASHRAE; **Jack Glass, P.E.,** Member ASHRAE, and **Geoffrey Routsen, P.E.**

Enterprise data centers support computer operations for the majority of financial and corporate institutions. Normally, these data centers are required to operate 24/7. These centers typically are operating with computing power demands in the range of 50 W/ft^2 to 100 W/ft^2 (540 W/m^2 to 1080 W/m^2) where the area is defined to be the gross computer room floor area.

New enterprise facilities are generally being designed for power densities of 100 W/ft^2 to 200 W/ft^2 (1080 W/m^2 to 2160 W/m^2). Therefore, a 100,000 ft^2 (9290 m^2) center with a power density of 100 W/ft^2 (1080 W/m^2) would have a computer power demand of 10 MW. Even using high-efficiency, water-cooled centrifugal chillers to meet these large data center cooling needs, these centers can easily require 7 MW of power at full load to operate the cooling systems, support transformer and uninterruptible power supply (UPS) losses and other incidental loads necessary to support 10 MW of computing power.

Industry trends indicate continued growth of the data center market[1] and continued consolidation of financial institution and Internet business operations into larger data centers, many greater than 50,000 ft^2 (4645 m^2). As a result, many owners that are building new data centers are seeking to incorporate energy savings designs that exceed the energy-efficiency requirements of ANSI/ASHRAE/IESNA Standard 90.1. For the majority of financial institutions, energy saving features cannot come with the sacrifice of reliability, as the costs of downtime can far exceed the annual energy costs of operation. Mechanical designs for these enterprise data centers include energy saving features that do not reduce

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system reliability or pose additional operational risk.

This article reviews the major actions taken to reduce energy consumption in a recently constructed enterprise data center in the Northeast. This data center uses chilled water computer room air handlers (CRAHs) to cool the computers. Two water-cooled centrifugal chilled water plants, each equally capable of supporting the entire cooling load (2N redundancy configuration) supply chilled water to the CRAHs. The mechanical energy savings features of the design included:

- A series water-side economizer, designed into one of the two independent chilled water plants, to provide free cooling whenever ambient conditions permit;
- Variable speed drives applied to the motors of the centrifugal chillers, CRAH fans, secondary chilled water pumps and cooling tower fans, to provide for energy reductions during operation at partial load; and
- Heat recovery from the uninterruptible power supplies (UPS) to maintain the UPS batteries at the required 77°F (25°C) without the addition of supplemental heat.

Additionally, high-efficiency windings on the transformers were provided to reduce electric transformation losses, and a selective catalytic reduction (SCR) and soot trap system were fitted to the generator exhaust. Although not directly providing energy savings, the installation of the SCR reduces the environmental impacts of operating the generators by reducing significantly the carbon emissions to the atmosphere.

Chiller Plant Energy Reductions

Table 1 identifies the approximate breakdown of mechanical energy use in the subject center at the day 1 full build load (7500 kW of computer load). Chiller energy followed by fan energy were clearly the largest energy consumers of full-load mechanical equipment electric power and, were the focus for mechanical energy reduction. The day 1 build called for three 1000 ton chillers in each plant, with each plant capable of expansion to four 1,000 ton (3517 kW) chillers. The data center load was forecast to grow to the 3,000 ton (10 551 kW) cooling load over a period of five years. Highly efficient centrifugal (0.535 kW/ton at design) chillers were selected and fitted with variable speed drives to maximize efficiency at part load. The chillers were tested at the vendor's factory at full- and part-load conditions to verify energy performance. The ARI tests confirmed the full-load efficiency, and the partial-load test indicated 0.232 kW/ton efficiency at 400 tons with 60°F (15.5°C) condenser water. Condenser water reset, where condenser water supply temperature is reduced as the ambient wet-bulb temperature drops, was used to provide an estimated annualized efficiency of 0.37 kW/ ton. Variable speed drives were applied to the cooling tower fans to ensure accurate control of the condenser water temperature. The design chilled water temperature differential was selected to be 47°F to 59°F (8°C to 15°C), with the 0.535 kW/ton corresponding to the 47°F (8°C) supply water temperature.

To reduce centrifugal chiller plant energy a water-side economizer was implemented to provide reduced chiller operation whenever wet-bulb conditions permitted. The design team considered applying an air-side economizer, but decided in favor of the water-side economizer due to the reduced humidification and outside air introduction requirements associated with it. Design temperature and humidity conditions were set to be 68°F to 77°F (20°C to 25°C) and 40% to 55% RH.[2] Given the Northeastern location of the data center, maintaining 40% RH in the winter months would have resulted in a significant humidification load with the application of the air-side economizer. It was the design team preference to keep the humidification requirements to a minimum rather than installing the larger humidification system that would be required to implement an air side economizer.

The owner's team required that implementation of the economizer could not sacrifice reliability. The design team was tasked to exceed the design efficiencies required by Standard 90.1 and to provide maximum efficiencies at reduced load conditions. However, it was understood that computer outages would cost the owner significantly more than the annual energy savings the most efficient systems could provide. Special attention was given to all aspects of energy-savings features that added additional control sequences or devices. Three aspects of the water-side economizer were viewed as potential risks to continuous operation of the chilled water plant:

1. Low condenser water temperature. Excessively cold condenser water temperatures can keep centrifugal chillers from starting. This was of special concern on this project as large condenser water basins were constructed underneath the cooling towers to ensure continuous operation of the cooling towers even in the event of loss of domestic water makeup. The water in these large basins can take an unacceptably long period of time to warm up when mechanical cooling is enabled. Energy reductions with the water-side economizer are obtained by lowering the condenser water temperature below the range of the chilled water temperature. This colder condenser water temperature is then passed through a heat exchanger whereby the condenser water directly cools the chilled water. The cold start-up condition occurs during a changeover from full free cooling to mechanical cooling. When the changeover occurs, the cold condenser water is sent to the condenser bundle within the chiller and the chiller is started.

2. Changeover into and out of economizer cooling. To maximize energy savings, the heat exchangers in a water-side economizer often are enabled automatically whenever the wet-bulb conditions permit. The decision to automatically enable/disable economizer cooling generally is made by the building management system (BMS). This can result in the changeovers occurring without notification of the facility maintenance staff. Frequent and/or unscheduled changes of state are viewed as unacceptable risks to continuous data center availability.

3. Overly complicated controls. Many series-arranged, water-side economizers incorporate up to four motorized valves that control the water path through the heat exchanger and chiller, permitting each device to be operated separately or allowing them to operate together. These motorized valves are not required if economizer cooling is not provided and were identified as additional points that added system complexity and potential risk for failure.

Figure 1 is a simplified flow diagram for the water-side economizer. The plate-and-frame heat exchangers that provide free cooling were designed into only one of the two independent chilled water plants. This permits the other plant to be run at an elevated condenser water temperature, eliminating the risk of the chillers in that plant not starting due to condenser water that is too cold. Further, the two heat exchangers were limited to operation with only two of the cooling towers to permit warmer condenser water to always be maintained in two of the cooling towers in the chilled water plant using the free cooling. Variable speed drives and a condenser water bypass were applied to the cooling towers to ensure the operating basin condenser water temperature is maintained at the control point. In the event of a complete failure of the free cooling system at full load, the BMS is capable of automatically enabling full mechanical cooling in the other chilled water plant, where the condenser water temperature will always be maintained at 60°F (15.5°C) or warmer.

Mechanical Energy Consumption		
Category	Electrical Demand at Full Load (kW)	Percent of Full-Load Power Consumption
Water-Cooled Centrifugal Chillers	1,605	50%
Computer Room Fan Energy	713	22%
Condenser Water Pumps	260	8%
Chilled Water Pumps	257	8%
Humidification	144	4%
Cooling Tower Fan Energy	81	3%
Miscellaneous Fan Energy	157	5%

Table 1: Full-load mechanical energy use.

To eliminate the risks of unscheduled changeovers to economizer cooling, it was agreed that facility personnel would manually enable and disable free cooling by opening supply valves to the heat exchangers. The BMS is programmed to notify the facility operators when ambient conditions permitting

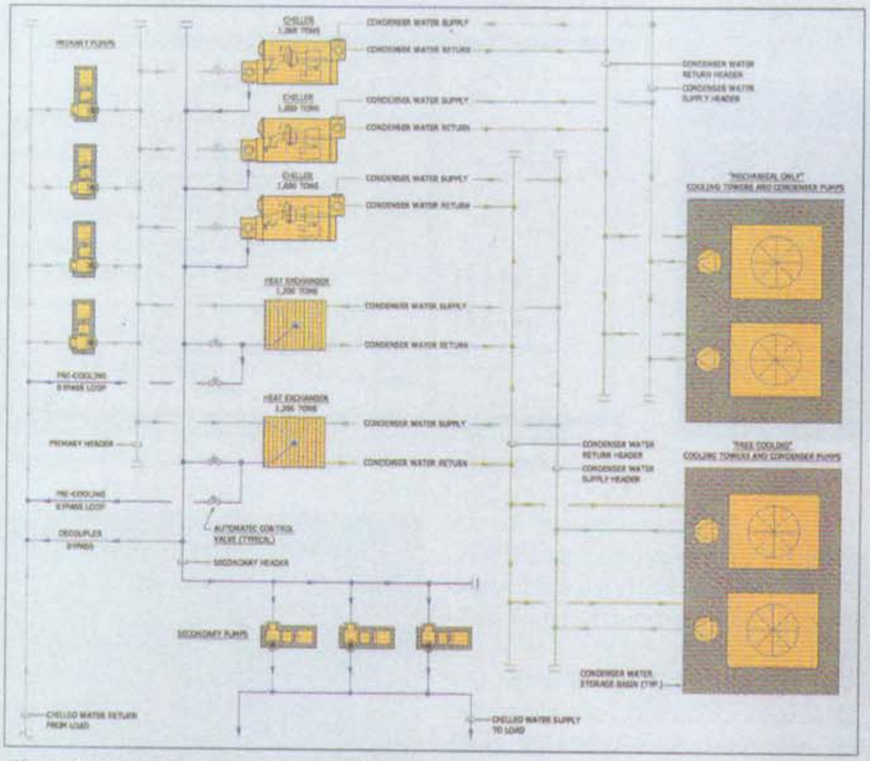

Figure 1: Water-side economizer cooling approach.

free cooling exist and when conditions requiring changeover back to mechanical cooling are imminent. When operators are required to enable/disable mechanical cooling this ensures that operators are present during the changeover and are available to revert back to full mechanical cooling if an unplanned event occurs during the changeover.

To reduce the automatic controls required to operate the economizer the heat exchanger was piped in parallel to the chillers and treated as if it were a chiller when enabled. A primary/secondary pumping arrangement was used, whereby the primary pumps were connected to one pipe header (*Figure 1*) permitting any of the primary pumps to be used with any of the chillers or heat exchangers. Two motorized valves were installed to permit either pre-cooling of the chilled water by the heat exchanger (i.e., an integrated approach using both the heat exchangers and chillers) or routing of the cooled water directly to the suction header of the secondary pumps (bypassing the chillers entirely). In the precooling mode the chilled water is sent directly from the heat exchanger to the suction side of the primary pumps. During free cooling operation, one of the chillers is unavailable for use. The design team agreed that this approach minimized the control work generally required to operate a series water-side economizer.

As indicated previously, the design chilled water temperatures were set to 47°F to 59°F (8°C to 15°C). The plate-and-frame heat exchangers were designed on a two-degree approach between condenser water and chilled water. At full load, this two-degree approach permits the operators to enable free cooling whenever condenser water of 55°F (13°C) or less can be generated. Due to size limitations imposed upon the cooling tower system, the design approach at the 55°F (13°C) condenser water supply temperature was set at 42°F (6°C) wet bulb. This permits approximately 3,800 hours of operation of the free cooling system (full- and part-load combined hours) with the ambient conditions corresponding to this facilities climate. An annual energy savings of \$110,000/yr is conservatively projected for system operation at the 3,000 ton (10 551 kW) load level, based on a \$0.10/kWh. *Photos 1* and *2* show

Photo 1: Concrete cooling towers.

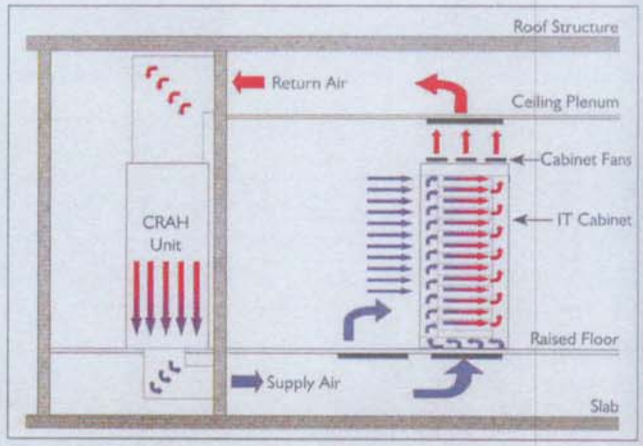

Figure 2: Raised floor air distribution.

the concrete cooling towers as well as the plate-and-frame heat exchangers used to accomplish the free cooling.

CRAH Unit Energy Reductions

To reduce fan energy, variable speed drives were applied to all of the computer room air handlers (CRAHs). Including the redundant units, 76 CRAH units were installed to meet the day 1 full build load of 7,500 kW. These units were selected to each deliver 18,000 cfm (8495 L/s), with 9kW of fan power consumption. The design made provisions for up to 102 CRAH units to support the day 2 installation of 10 MW. From a computer load standpoint, the subject data center is expected to grow to the day 1 installation capacity over a five-year period. This was viewed as an excellent opportunity to capitalize on the reduced airflow required by the computers during the growth period. In the subject data center, the compute servers are installed into cabinets that incorporate variable speed fans, which draw air from below the raised floor plenum as well as from the cold aisle. These fans operate to maintain a fixed (adjustable setpoint) temperature differential from the intake to the discharge of the cabinet. As the computing power varies within the cabinet (either as a result of load variations of the servers themselves or the installation or removals of servers) the cabinet fans vary the airflow through the cabinet to maintain the air temperature differential across the cabinet. The air outlet is at the top of the cabinet, where the air is discharged vertically and directly into a ceiling return plenum. Return air from the ceiling plenum is ducted back to the CRAH units where it is conditioned and then discharged into the raised floor plenum (*Figure 2*).

To match the airflow through the CRAHs as closely as possible to the airflow through the cabinets, pressure sensors were installed underneath the cabinets and set at the minimum air pressure necessary to support cabinet airflow at full load. The installation of multiple sensors and zoning of the sensors to specific CRAHs permits the operators to maintain different underfloor pressure conditions in different areas of the data center. Underfloor pressure monitoring is performed at the BMS, and the operators have the ability to adjust setpoints at the BMS workstations. These setpoints are then output to the CRAH units where control of the variable speed drive is accomplished through the individual CRAH unit controller. In the event that output signal from the BMS system either fails or goes out of the control range, the CRAH unit controller automatically operates the CRAHs fan at full speed. This ensures the reliability of this system. Based upon the proposed load growth profile to support the day 1 build, an average of $100,000/yr energy savings is expected during the five-year installation program of the IT equipment.

To further keep the computer room cooling system energy costs to a minimum, reheat and humidification were not installed in the air conditioners. Although this reduces the ability to control relative humidity, removal of the reheat and humidification from the CRAHs eliminates the possibility of having air conditioners that are serving a common computer room operating simultaneously in both heating and cooling modes. Construction costs also are reduced without the installation of the reheat and humidification. To control humidity and provide the minimum outside air required for ventilation and pressurization of the data center, a single air handler was installed with both heating and humidification capabilities. Dew-point sensors were installed in the computer room and used to enable humidification as required.

Photo 2: Plate-and-frame heat exchanger.

The CRAHs were designed to operate in a dry condition (all sensible cooling) to ensure that no unnecessary chiller energy is expended to perform latent cooling and no unnecessary humidification is required to maintain the dew-point condition in the room. The chilled water valve within the CRAH is controlled through the CRAH unit controller, and can be based upon either supply- or return-air control at the operator's discretion. Sequencing of the CRAH chilled water valve and the fan airflow is performed by the unit controller to maintain pressure and temperature inputs and ensure the cooling coil remains dry.

Heat Recovery for Room Temperature Control

A significant amount of heat is in the return air to the computer room air conditioners. Due to its relatively low temperature (ranging 76°F to 90°F [24°C to 32°C] depending upon the operating conditions), it can be difficult to find uses for this heat in the data center application, where little or no heat is required. One area where this low-grade heat can be used is in the battery rooms. For UPS applications, the battery manufacturers recommend their batteries be maintained at 77°F (25°C). This temperature generally is viewed as the optimum point to maximize battery life and discharge capability. Above this temperature battery life can be significantly reduced and, below this temperature, battery discharge capability is significantly reduced. A number of batteries often are installed with thermometers mounted to them to monitor temperature.

The subject data center used wet-cell batteries, which require continuous air ventilation to eliminate the chance of hydrogen gas buildup. Code requirements for this project were 1 cfm (0.5 L/s) per square foot or 2,500 cfm (1180 L/s) per battery room. Six battery rooms were required for the project. These rooms were all built-out for the day 1 installation due to the electrical configurations of the project. These systems required 15,000 cfm (7079 L/s) of exhaust air. Each battery room was served by two exhaust fans (one primary and one redundant).

Figure 3 is the airflow diagram showing the approach to recovering heat from the electric rooms. There are four air handlers (two primary and two redundant) sized at 7,200 cfm (3398 L/s) that provide conditioned air to the battery rooms. The additional 100 cfm (47 L/s) per battery room required by the exhaust system is infiltrated from the corridors to ensure the battery rooms remain at a slight negative pressure relative to the corridors. The two operating air handlers transfer 12,000 cfm (5663 L/s) of air from electric rooms containing transformers that operate continuously to support the computer room electrical systems. The electric rooms are cooled by separate utility area air handlers. The additional 1,200 cfm (5663 L/s) of air required by the battery room air handlers is taken directly from the outside. Based upon the winter design condition of 0°F (–18°C), this approach reduces the full load heating demand by 294 kW, reducing energy costs by approximately $50,000/yr.

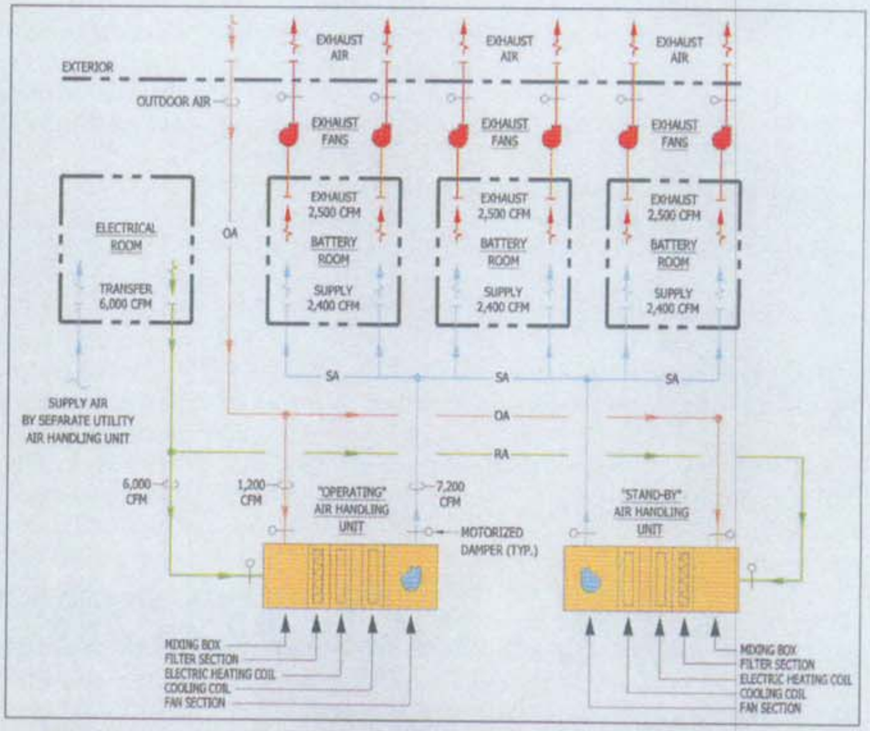

Figure 3: Battery room airflow diagram.

Conclusions

Although reliability is still the driver for the majority of enterprise data centers, significant reductions in energy can be attained without reducing system reliability through the incorporation of free cooling, variable speed drives, condenser water reset, operation of the chilled water system at temperatures greater than the standard 45°F (7°C) and the application of heat recovery systems whenever possible. These energy saving approaches were applied at the subject data center with careful consideration for minimizing additional equipment and control sequences, while ensuring appropriate fail-safe systems in the event the energy saving systems become inoperative. For the subject data center, the total annual energy savings for the facility have been estimated at approximately $260,000/yr. These estimates do not include demand charges or account for future increases in cost of power. It is expected that actual savings in future years will be greater.

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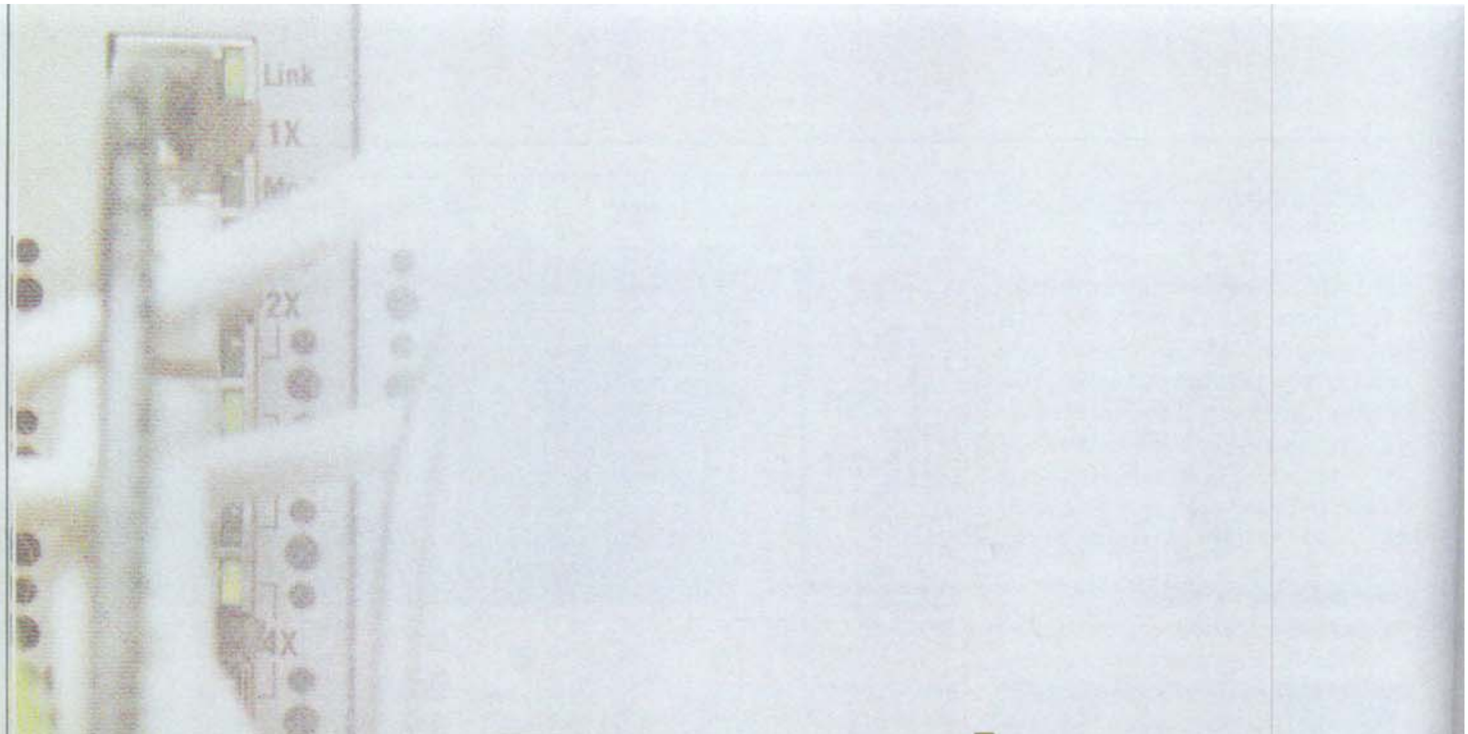

OA Economizers For Data Centers

By Vali Sorell, P.E.

Compared to most types of commercial buildings, data centers are energy hogs. A large data center, for example, can easily consume as much electrical power as a small city.

Consider an average size data center—say a facility with an average demand of 1 MW over the entire 8,760 hours of the year. The cost of driving that 1 MW computer load is \$700,000 per year (assuming a cost of electricity of \$0.08/kWh). The cooling load associated with the 1 MW load is 285 tons (1002 kW). At an average chiller efficiency of 0.5 kW/ton, the cost of running that chiller at full load year-round is approximately \$100,000 per year.

The chiller is easily the largest energy consumer of all the facility's HVAC equipment. Significant HVAC energy savings can be realized by reducing chiller energy. However, reducing the number of hours of chiller operation has a larger impact on lowering energy use in a facility than by selecting a more energy-efficient chiller.

The need to incorporate more sustainable and environmentally responsible design elements—especially in a facility that consumes staggering amounts of energy every single hour of the year—must accept a double-pronged approach toward a cooling system design. On the one hand, more energy-efficient equipment must be selected. On the other hand, a method of reducing the hours of operation of the equipment must be incorporated into the cooling system design. Bracketing this approach is the absolute necessity to ensure that overall system reliability is never compromised. If a design element reduces the overall reliability of the data center, it will not be implemented.

The two types of economizers that can reduce hours of chiller operation are water-side economizers, and air-side economizers. Water-side economizers are explained briefly below; the remainder of this article concentrates on issues relating to air-side economizers.

A water-side economizer uses the building's cooling towers to cool the chilled water by taking advantage of the hours of the year during which the outdoor wet-bulb temperature is sufficiently lower than the chilled water supply setpoint. Instead of running the chiller during

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those hours, the cooling tower water is bypassed around the chiller and diverted to a heat exchanger so that it can cool the chilled water directly. This type of economizer has advantages and disadvantages, but further discussion of a water-side economizer is beyond the scope of this article.

Air-Side Economizer—Brief Overview

An air-side economizer takes advantage of the hours of the year during which the outdoor enthalpy (energy content) is less than the return air enthalpy. Under certain outdoor conditions, using outdoor air reduces the load that is experienced at the air-handling unit when compared to using the return airflow.

When the outdoor enthalpy is less than the return air enthalpy *and* higher than the enthalpy of the supply air setpoint, some mechanical cooling is still called for to meet the requirements at the supply air setpoint. Under these circumstances, the chiller will be required to operate, although not at as high a load as would be required for a 100% return air system. This is a partial (or integrated) economizer.

When the outdoor enthalpy is less than the enthalpy of the required supply air setpoint, no chiller operation is required, and the actual supply conditions can be met by either mixing outdoor air with return air (if the outdoor air is below the supply air setpoint), or using 100% outside air (if the outdoor air is at the supply air setpoint). This is considered a full economizer. Significant energy savings can be realized whether a partial or full economizer is utilized. Figure 1 shows the basic flows involved with a 100% recirculating system and an economizer system.

Hours of Economizer Use in Various Cities

Exactly how many hours of the year are available for economizer use? The weather data for several representative cities was evaluated. These cities were Dallas, New York, San Francisco and London. For each of these cities, dry-bulb and dew-point conditions were examined and compared with an ideal facility using a supply air setpoint of 68°F (20°C) dry bulb and a dew point of 50°F (10°C). (For more discussion of why these conditions were selected, see the Temperature/Humidity Control section.) Using these weather data, three bins were established to collect and classify all the yearly data:

- The number of hours during which an air-side economizer is available to provide 100% of the facility's cooling needs;
- The number of hours during which an air-side economizer is available but cannot meet all the facility's cooling needs (partial economizer); and
- The number of hours during which an air-side economizer should not be used (i.e., the return air conditions are more favorable than the outdoor air conditions).

The results are summarized in *Table 1*.

Common sense would normally dictate that an outside air economizer in the hottest climates would not have a good payback. That logic may be applicable to a typical office building where there are approximately 2,500 hours of use in a year. However, a data center must run continuously, 24 hours a day, for a total of 8,760 hours per year. The number of hours of availability is, therefore, greatly increased. In Dallas, which is the warmest climate considered, the hours of availability amount to more than half of all the hours of the year. For cities such as San Francisco and London, where the annual hours of full economizer availability are higher than 8,000, using an economizer requires almost no complicated analysis and should be considered. (There may be other issues that come into play that could restrict the use of an air-side economizer, such as the lack of availability of sufficient building openings to the outdoors for air intake and exhaust. Under those circumstances, a water-side economizer should be considered.)

Using Economizers for Data Centers

Historically, the industry generally has avoided using outside air economizers when dealing with data centers. Even ASHRAE's Technical Committee 9.9, the technical committee which deals with issues of data center design and operation, has avoided making any recommendations about the application of outside air economizers until more research can be provided to either support or reject its use for data centers.

The main points to be considered when such large quantities of outside air are introduced into a data center are as follows:

- Introduction of outside air into a data center can be a source of dirt, dust, and other airborne particulate or gaseous contaminants that can place the investment of computer equipment in that center at risk; and
- Introduction of outside air into a data center can play havoc on the ability to control the space's relative humidity.

Both issues should be addressed during the design phase for any data center facility. Both issues have been addressed successfully in the past in other types of facilities that have critical needs *and* a requirement for large quantities of outside air, such as hospitals and laboratory facilities.

Yet, the data center design community will not accept the use of outside air economizers if doing so will result in lower facility reliability or reduced uptime. In recent case studies published by Lawrence Berkeley National Laboratory (LBNL),[1] it was determined that the challenges surrouding air contaminants and humidity control can be resolved. In short, those studies found slightly higher particulate concentrations (gaseous contaminants were not measured) in data centers with outdoor air economizer when compared to data centers with 100% recirculating systems. For both systems, the particulate concentrations within the spaces were significantly below the most conservative particle standards. In addition, the study states that the increase in particle concentration in systems with economizers can be negated with modest improvements in air filtration. This item is discussed in further detail below.

The issues relating to the use of outside air have been discussed within the industry and are receiving considerable attention. In response to the need to sort through the various issues relating to the use of outside air, TC 9.9 is working on a book addressing the issue relating to the use of outside air economizers and contamination within data centers. That book is expected to be published sometime in early 2008.

With those issues resolved, there should be a strong impetus to use outside air economizers based on improved energy efficiency. If a large data center that has been designed according to today's best practices uses as much energy as a small city, the savings that can be realized by implementing an economizer system can be equivalent to the energy used by a large community within that city.

	Yearly Use of Airside Economizer		
Representative Cities	Available Hours of Full Economizer	Available Hours of Partial Economizer	No Economizer Availability
San Francisco	8,563	197	–
New York	6,634	500	1,626
Dallas	4,470	500	3,790
London	8,120	300	340

Calculation of available hours based on a 68°F DB/50°F dew-point supply air.

Table 1: Hours of economizer use by city.

Outside Air Filtration

The first and easiest item to address is air contaminants. When introducing a large amount of outside air, it is necessary to increase the filtration at the air handlers. With 100% recirculating systems, filters with a MERV rating of 8 or 9 (ANSI/ASHRAE 52.2-1999; equivalent to 40% efficient based on the older "dust spot" efficiency rating of ANSI/ASHRAE Standard 52.1-1992) are typically used. These filters are intended to remove only the particulates that are generated by the activity within the space. When outside air is introduced, it is necessary to increase the MERV rating to 10 or 11 (equivalent to 85% efficient based on dust spot method) so that the filters can extract the increased loading of particulates (i.e., the smaller particulates) associated with construction, road and highway traffic, industrial processes, and other outdoor pollutants. This approach of using improved air filtration is consistent with the LBNL study.[1] The cleanliness of the outdoor air, in terms of particulate content, will be as good or better than the cleanliness of a recirculating system with the lower MERV rating. The higher MERV rating filter will create a higher operating pressure at the fan, and this is associated with an increase in energy use. However, this extra energy use is small in comparison to the savings associated with reduced operation of the chiller plant.

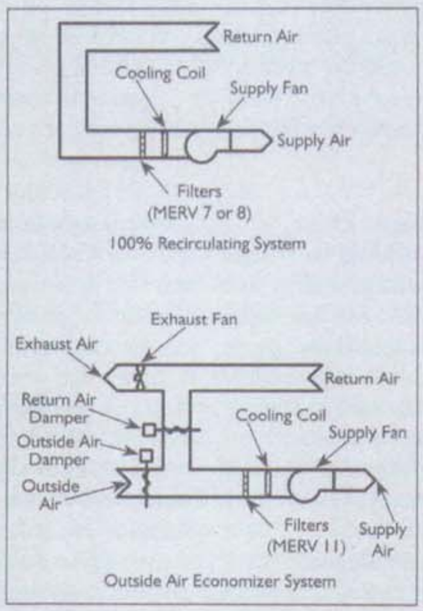

Figure 1: Comparison of 100% recirculating system and outside air economizer system.

Temperature/Humidity Control

With the publication of ASHRAE's 2004 book, *Thermal Guidelines for Data Processing Environments* (prepared by TC 9.9), the industry has come to a consensus about the optimal thermal environment for data processing equipment. The recommended environment is 68°F to 77°F (20°C to 25°C) DB and 40% to 55% relative humidity at the INLET of the equipment (*Figure 2*). The temperature at any other location within the data space is irrelevant. This is a critical point to understand, and it highlights the basic differences between cooling for comfort and cooling for critical computer equipment.

In a comfort cooling environment, the supply air is usually provided by an overhead system within a range of 55°F to 60°F (13°C to 16°C) DB. This air is then thoroughly mixed into the occupied space, and the temperature perceived by the occupants is represented by this mixed condition. The thermostat that controls the occupied space must be located in the occupied space to ensure optimal comfort.

In a data processing space with cooling provided from an underfloor plenum, the cold supply air is separated from the hot discharge air by arranging equipment cabinets in alternating hot and cold aisles. This separation of airstreams protects the equipment from overheating by preventing the hot discharge air from recirculating back to the equipment inlets. Therefore, when set up properly, the inlets to the equipment will be close to the supply air temperature. The thermostat should be placed in the supply airstream as close to the controlled environment as possible. The biggest mistake that designers make (and the computer room air-conditioning equipment manufacturers propagate this mistake) is to place the thermostat in the return airstream. The facility manager cannot properly control the equipment's thermal environment (i.e., at the inlet to the equipment) by sensing the return air temperature that is physically distant from the controlled environment.

Since the *Thermal Guidelines* promotes the warmer design temperatures, designing to a fixed supply air temperature opens up new possibilities. If the hot aisle/cold aisle concept is properly implemented, and sufficient air is supplied into the cold aisles to preclude hot air recirculation from the hot aisle back to the cold aisle, the cold aisle will remain uniformly at one temperature—the supply air temperature. There is no reason to cool supply air to 55°F or 60°F (13°C or 16°C) DB when

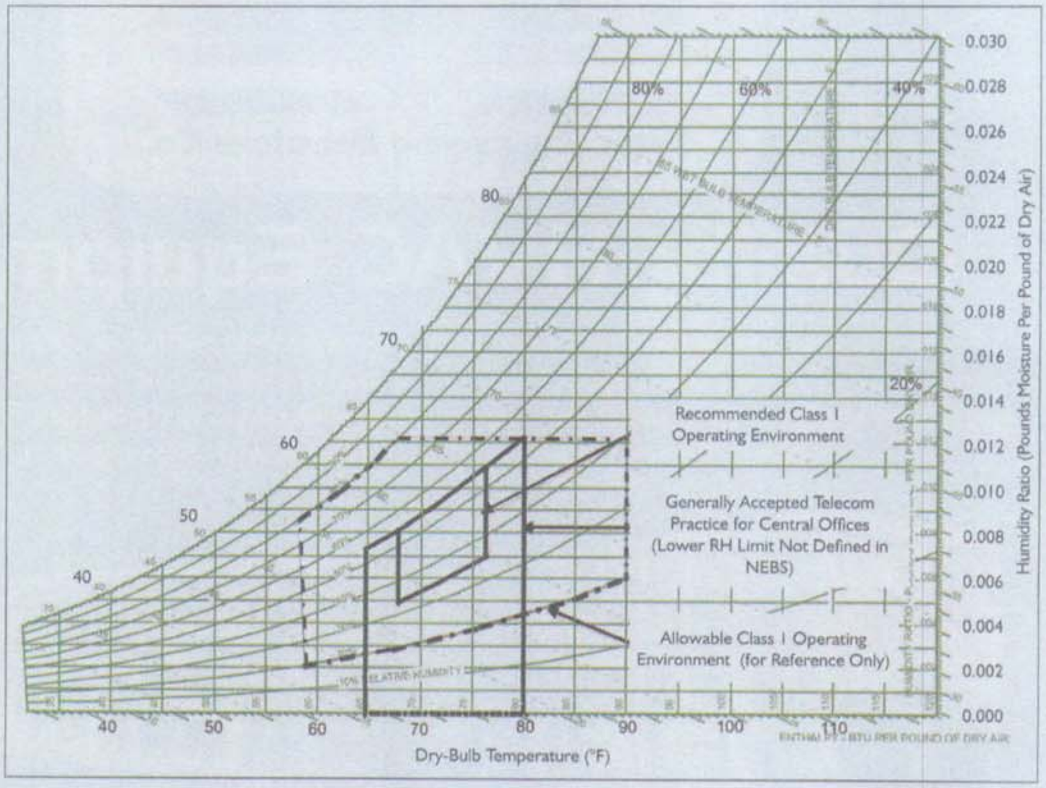

Figure 2: ASHRAE recommended Class 1 operating environment.

68°F (20°C) DB and higher is recommended for the equipment inlet temperature.

There are many benefits of designing around 68°F (20°C) DB as a supply air temperature:

- The thermal environment for the equipment will fall within the recommended envelope, in compliance with the *Thermal Guidelines*. The result will be optimized equipment reliability.
- There are many hours of the year during which the outdoor temperature falls within the range of 60°F and 68°F (16°C and 20°C). By designing around the higher supply air temperature, these hours become available for full outside air economizer use. The onset of chiller operation is delayed, and the total hours of chiller operation are reduced. (The analysis presented previously for various cities assumes a 68°F [16°C] supply air setpoint and captures the availability of these extra hours of economizer use.)
- Selecting a cooling coil for the higher supply air temperature also allows more sensible heat to be removed from the space for a given airflow. One reason for this is that the amount of latent heat removed unnecessarily by the coil is reduced or eliminated, as well as the unnecessary humidifier operation needed to return the lost humidity back to the space. As such, the air handler coil and the heat transfer process will operate more efficiently.
- With the higher supply air temperature, the chilled water temperature can be raised. As a result the chiller operates more efficiently for the fewer hours that it does operate.

This discussion has not yet addressed the issue of humidity control. *Thermal Guidelines* defines the top and bottom of the thermal environment envelope (as shown on a psychrometric chart) in units of relative humidity. This poses a design challenge since relative humidity varies depending on where in the space it's measured. The higher the local temperature, the lower the relative humidity. The result is that one can properly control the dehumidification or the humidification processes only if the relative humidity is measured where supply air temperature is measured (assuming that we are controlling by supply air temperature, as noted previously). This is not always practical.

There is a simple solution to this dilemma, and this involves using absolute humidity and/or dew-point sensors. Because there are no latent loads in data spaces, the measured absolute humidity or dew point will be uniform throughout the space—from the supply air, to the cold aisles, to the hot aisles, to the return air. This is not the case for relative humidity. If these sensors are placed in the return airstream, they are in the perfect location to measure the return air enthalpy. (Temperature sensors also must be included in the return airstream to help determine whether to use the air-side economizer or mechanical cooling. However, to avoid unstable temperature control and "fighting" between adjacent air-handling units, these temperature sensors must not be used to control the cooling coil. Supply air temperature sensors must be used for that purpose.) The dew-point sensors in the return airstream serve a dual purpose: they're used as the controlled point for space humidity control, and they're used as part of an enthalpy economizer.

Using an enthalpy economizer is the last component to consider in the use of outdoor air economizers. The issue of enthalpy economizer was introduced earlier, and was used only in the context that enthalpy relates to the energy content of an airstream. In essence, an enthalpy economizer looks at the temperature and humidity of the outdoor air and the return air, compares the condition of each, and determines which airstream (or which combination of airstreams) will use the least amount of mechanical cooling. With a full economizer, the chiller and its associated equipment is turned off. There is no mechanical cooling. With a partial economizer some mechanical cooling is needed, but cooling the outdoor air rather than the return air to the supply air setpoint leads to a net lower usage of energy. Humidity *is* considered under these conditions. In most major cities in the U.S. and Europe, an enthalpy economizer makes economic sense. For some cities in the southwestern parts of the United States, where the climate is dry when the outdoor temperatures are high, a simple dry-bulb economizer can work as well and cost less to install and implement.

With a full economizer in operation and the control components in place as described previously, humidity control becomes a straightforward process. If the resultant mixed air condition is too dry (i.e., the dew point is too low), a humidifier adds moisture into the airstream to increase the dew point. To reiterate what was noted above, this process must be controlled by dew point or absolute humidity sensors to ensure that cooling coils don't dehumidify while the humidifiers simultaneously add moisture to the space.

In addition, the dew point/absolute humidity sensors that contribute to the measurement of the moisture content or enthalpy of the space, the return air, and the outdoor air must be of the highest quality. They must be specified to be calibrated to tight tolerances, and a significant commissioning effort must be expended to ensure that all the sensors work in unison and actually measure the relative differences in moisture content of the various airstreams.

If the resultant space condition indicates too high a dew point, the chilled water supply temperature and supply air temperature should be bumped down in one degree increments over a several hour period to wring moisture out of the space through the cooling coil. This normally would occur whenever the outdoor conditions are too humid. However, during such conditions the economizer should turn off since the return air would provide the more favorable enthalpy. When the space conditions return to the dew-point setpoint, the chilled water supply temperature and the supply air temperature can be reset back to their normal conditions to maintain the dew-point setpoint. This condition of recirculating the return air should be maintained as long as the outdoor conditions are less favorable than the return airstream. This operating condition should be no different than any other system without an outdoor air economizer.

Conclusions

With the large energy use and costs associated with data centers, the incentive to find ways to reduce the staggering costs of operating these facilities is huge.

Outside air economizers provide one of the best ways of reducing these costs. Yet, there has been considerable resistance to these economizers in data centers. The main reasons for this resistance have been fear of outdoor contaminants entering the facility's critical equipment, the perception that humidity control can become complicated and unstable, and the difficulty of keeping humidity or enthalpy sensors calibrated. With the application of appropriate design principles and control strategies, the particulate contaminants can be addressed in a manner that ensures that the reliability of the facility is not compromised. However, gaseous contaminants and the ability to sense and control these still need to be addressed so that high reliability/availability can be achieved. Keeping humidity or enthalpy sensors calibrated can be addressed with the statement that there have been significant improvements in the last several years in the quality of sensors and the ability of DDC systems to use control strategies to maintain space conditions according to a project's design intent.

With reference to the ability of the DDC system to control to an absolute humidity or dew-point setpoint, the need for a thorough commissioning program at the inception of the project, coupled with a program of continuous retrocommissioning throughout the life of the facility, cannot be stressed enough. The commissioning effort is especially critical in guaranteeing that the outside air economizer does, in fact, perform using accurate information about the moisture content of the various air streams that flow through the air-handling systems. Any significant error or inconsistencies in those sensing devices could lead to wasteful consumption of energy for cooling and/or dehumidification or water for humidification.

References

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