Efficiency plus reliability

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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.sup.2] to 100 W/[ft.sup.2] (540 W/[m.sup.2] to 1080 W/[m.sup.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.sup.2] to 200 W/[ft.sup.2] (1080 W/ [m.sup.2] to 2160 W/[m.sup.2]). Therefore, a 100,000 [ft.sup.2] (9290 [m.sup.2]) center with a power density of 100 W/[ft.sup.2] (1080 W/[m.sup.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.sup.2] (4645 [m.sup.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 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[degrees]F (25[degrees]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

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.

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[degreees]F (15.5[degrees]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[degrees]F to 59[degrees]F (8[degrees]C to 15[degrees]C), with the 0.535 kW/ton corresponding to the 47[degrees]F (8[degrees]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[degrees]F to 77[degrees]F (20 [degrees]C to 25[degrees]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

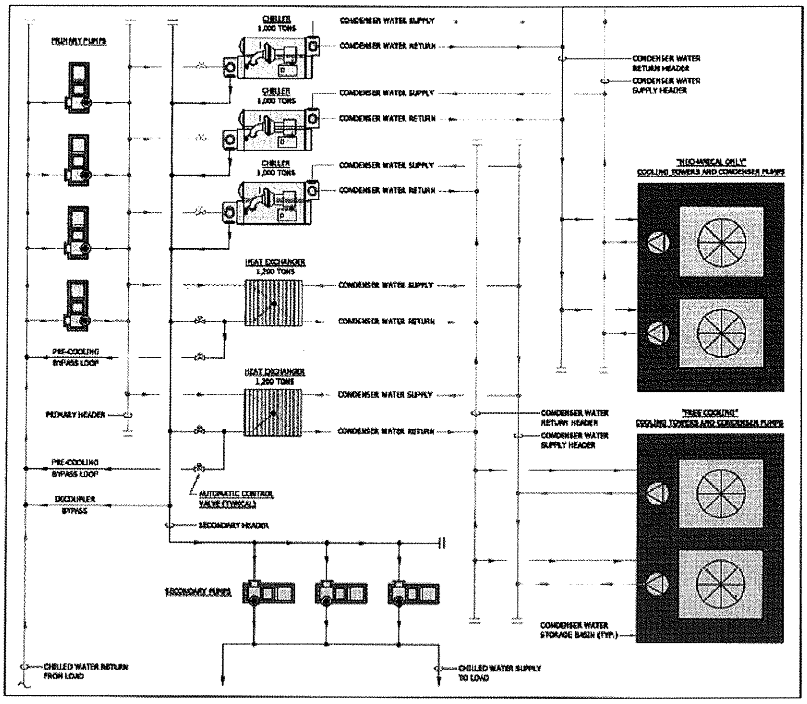

Figure 1: Water-side economizer cooling approach.

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[degrees]F (15.5[degrees]C) or warmer.

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 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[degrees]F to 59[degrees]F (8[degrees]C to 15[degrees]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[degrees]F (13[degrees]C) or less can be generated. Due to size limitations imposed upon the cooling tower system, the design approach at the 55[degrees]F (13[degrees]C) condenser water supply temperature was set at 42 [degrees]F (6[degrees]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 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

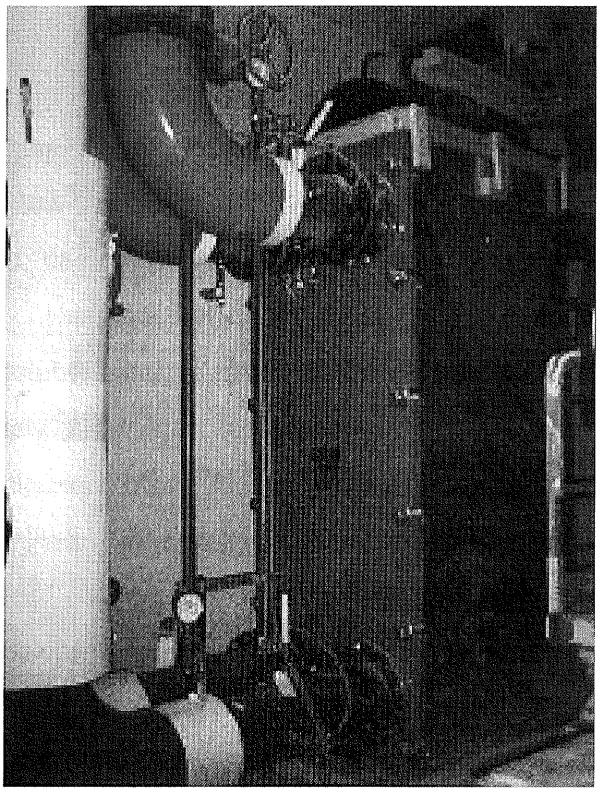

Photo 2: Plate-and-frame heat exchanger.

the cabinet. The air outlet is at the top of the cabinet, where the air is discharged vertically and indirectly 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.

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 operators 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.

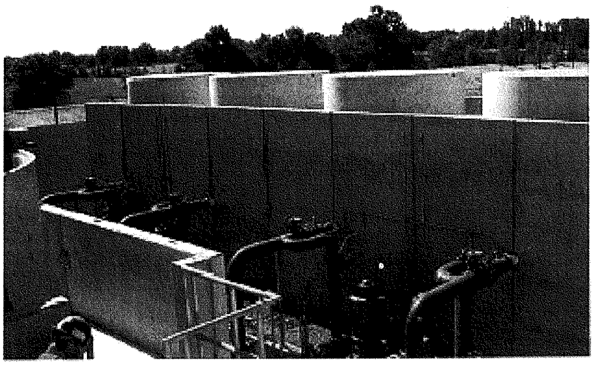

Photo 1: Concrete cooling towers.

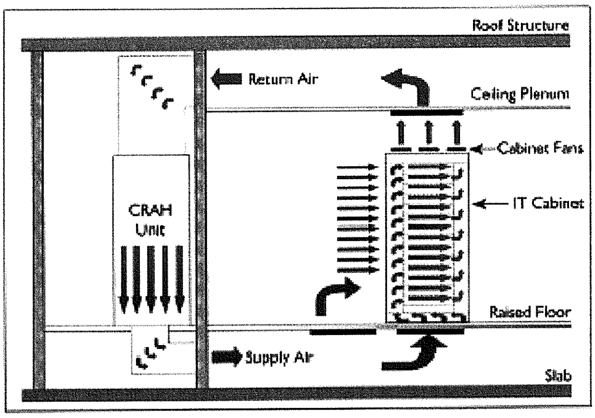

Figure 2: Raised floor air distribution.

Heat Recovery for Room Temperature Control

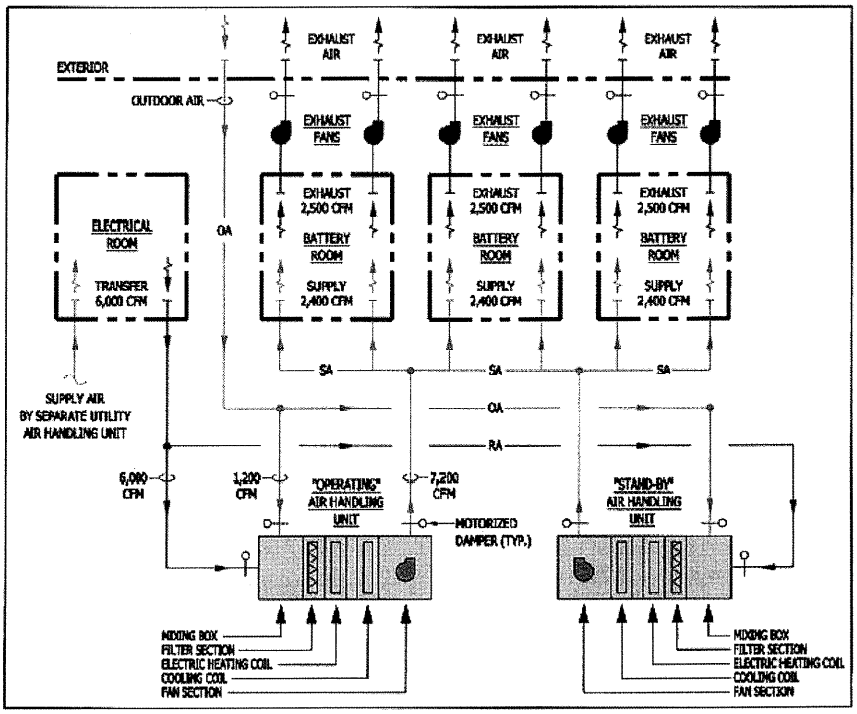

Figure 3: Battery room airflow diagram.

A significant amount of heat is in the return air to the computer room air conditioners. Due to its relatively low temperature (ranging 76[degrees]F to 90[degrees]F [24[degrees]C to 32[degrees]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[degrees]F (25[degrees]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[degrees]F (.18[degrees]C), this approach reduces the full load heating demand by 294 kW, reducing energy costs by approximately $50,000/yr.

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[degrees]F (7[degrees]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.

References

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