OA Economizers for Data Centers

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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 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.

facility using a supply air setpoint of 68? F(20[degrees]C) dry bulb and a dew point of 50[degrees]F (10[degrees]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

	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.

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

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.

Outside Air Filtration

The first and easiest item to address

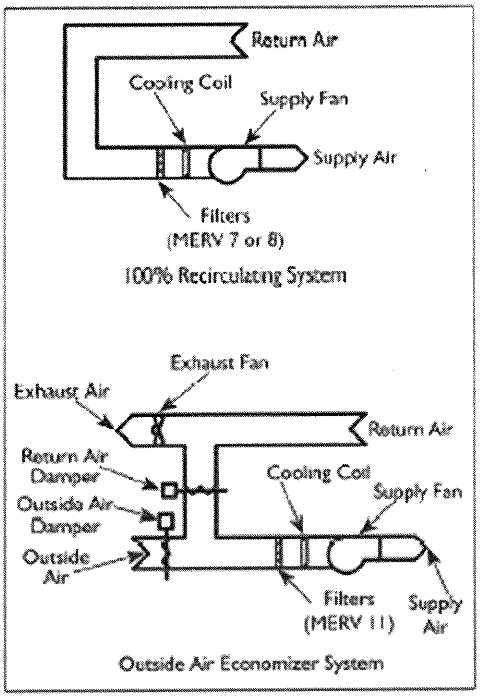


Figure 1: Comparison of 100% recirculating system and outside air economizer system.

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 52.11992) 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.

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[degrees]C to 25[degrees]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[degrees]C to 16[degrees]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.

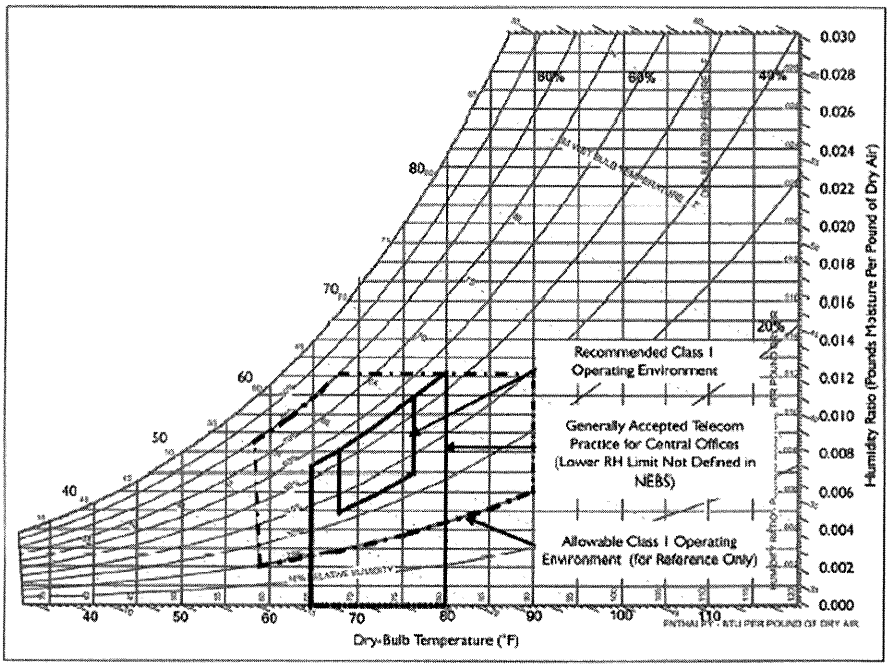
In a data processing space with cooling provided from an underfloor plenum, the cold supply air is separated from the hot discharge air by arranging equipment cabinets in alternating hot and cold aisles. This separation of airstreams protects the equipment from overheating by preventing the hot discharge 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 temperaturethe supply air temperature. There is no reason to cool supply air to 55[degrees]F or 60[degrees]F (13[degrees]C or 16[degrees]C) DB when 68[degrees]F (20[degrees]C) DB and higher is recommended for the equipment inlet 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[degrees]F and 68[degrees]F (16[degrees]C and 20[degrees]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[degrees]F [16[degrees]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



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

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 straight-forward 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

[1.] Tschudi, W. 2007. "Data Center Economizer Contamination and Humidity Study." LBNL/ Pacific Gas & Electric. http://tinyurl. com/ 3y4es7 (or http://hightech.lbl.gov/documents/ DATA_CENTERS/EconomizerDemoReport March13.pdf).

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