

Challenges of data center thermal management

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The need for more performance from computer equipment in data centers has driven the power consumed to levels that are straining thermal management in the centers. When the computer industry switched from bipolar to CMOS transistors in the early 1990s, low-power CMOS technology was expected to resolve all problems associated with power and heat. However, equipment power consumption with CMOS has been rising at a rapid rate during the past 10 years and has surpassed power consumption from equipment installed with the bipolar technologies 10 to 15 years ago. Data centers are being designed with 15–20-year life spans, and customers must know how to plan for the power and cooling within these data centers. This paper provides an overview of some of the ongoing work to operate within the thermal environment of a data center. Some of the factors that affect the environmental conditions of data-communication (datacom) equipment within a data center are described. Since high-density racks clustered within a data center are of most concern, measurements are presented along with the conditions necessary to meet the datacom equipment environmental requirements. A number of numerical modeling experiments have been performed in order to describe the governing thermo-fluid mechanisms, and an attempt is made to quantify these processes through performance metrics.

Introduction

Because of technology compaction, the information technology (IT) industry has experienced a large decrease in the floor space required to achieve a constant quantity of computing and storage capability. However, power density and heat dissipation within the footprint of computer and telecommunications hardware have increased significantly. The heat dissipated in these systems is exhausted to the room, which must be maintained at acceptable temperatures for reliable operation of the equipment. Data center equipment may comprise several hundred to several thousand microprocessors. The cooling of computer and telecommunications equipment rooms has thus become a major challenge.

Background

Considering the trends of increasing heat loads and heat fluxes, the focus for customers is in providing adequate airflow to the equipment at a temperature that meets the

manufacturers' requirements. This is a very complex problem considering the dynamics of a data center, and it is just beginning to be addressed [1–8]. There are many opportunities for improving the thermal environment of data centers and for improving the efficiency of the cooling techniques applied to data centers [8–10]. Airflow direction in the room has a profound effect on the cooling of computer rooms, where a major requirement is control of the air temperature at the computer inlets. Cooling concepts with a few different airflow directions are discussed in [11]. A number of papers have focused on airflow distribution related to whether the air should be delivered overhead or from under a raised floor [12–14], ceiling height requirements to eliminate “heat traps” or hot air stratification [12, 15], raised-floor heights [16], and proper distribution of the computer equipment in the data center [13, 17] such that hot spots or high temperatures would not exist. Different air distribution configurations, including those described in subsequent sections, have

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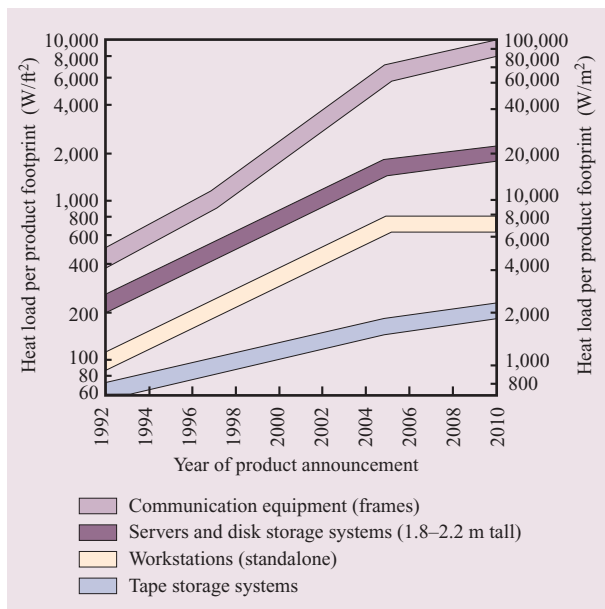


Figure 1

Power trends in data center equipment. Reprinted from [20], with permission.

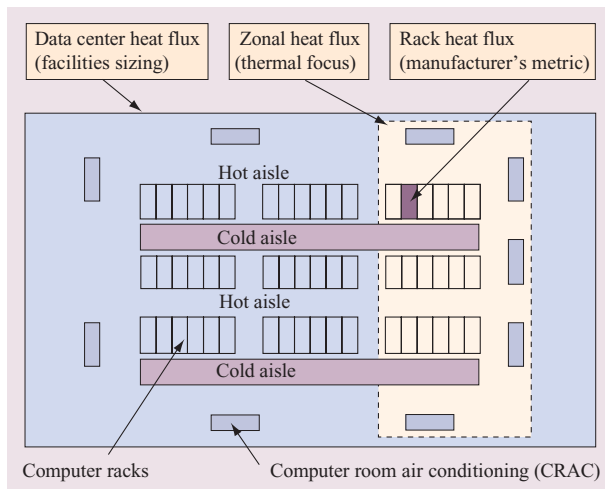


Figure 2

Organization of computer racks in a hot aisle–cold aisle layout.

been compared with respect to cooling effectiveness in [14, 16, 18, 19].

Trends—Thermal management consortium for data centers and telecommunications rooms

Since many of the data center thermal management issues span the industry, a number of equipment manufacturers

formed a consortium¹ in 1998 to address common issues related to thermal management of data centers and telecommunications rooms. Since power used by datacom equipment was increasing rapidly, the group's first priority was to create the trend chart shown in **Figure 1** on power density of the industry's datacom equipment to aid in planning data centers for the future. This chart, which has been widely referenced, was published in collaboration with the Uptime Institute [20]. Since publication of the chart, rack powers have exceeded 28,000 W, leading to heat flux based on a rack footprint in excess of 20,000 W/m².

Recent activities of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE)

In January 2002 ASHRAE was approached to create an independent committee to specifically address high-density electronic heat loads. ASHRAE accepted the proposal and eventually formed the technical committee known as TC9.9 Mission Critical Facilities, Technology Spaces, and Electronic Equipment. The first priority of TC9.9 was to create a Thermal Guidelines document that would help to standardize the designs of equipment manufacturers and help data center facility designers to create efficient, fault-tolerant operation within the data center. The resulting document, "Thermal Guidelines for Data Processing Environments," was published in January 2004 [21]. More recently, another publication entitled "Datacom Equipment Power Trends and Application," published in February 2005 [22], provided an update to the power trends documented in [20] and shown here in Figure 1. The heat load of some datacom equipment was found to be increasing at an even faster rate than that documented in the original chart published by The Uptime Institute [20]. Other ASHRAE publications will follow on data center thermal management, with one planned for the end of 2005 that will provide information on air and liquid cooling of data centers, energy conservation, contamination, acoustics, etc.

Physical design of data center systems

Layout of computer rack equipment

Data centers are typically arranged into hot and cold aisles, as shown in **Figure 2**. This arrangement accommodates most rack designs, which typically employ front-to-back cooling, and partially separates the cold air exiting the perforated tiles (for raised-floor designs) and overhead chilled airflow (for non-raised-floor designs) from the hot air exhausting from the rear of the racks.

¹The following companies formed the Thermal Management Consortium for Data Centers and Telecommunication Rooms: Amdahl, Cisco Systems, Compaq, Cray, Inc., Dell Computer, EMC, HP, IBM, Intel, Lucent Technologies, Motorola, Nokia, Nortel Networks, Sun Microsystems, and Unisys.

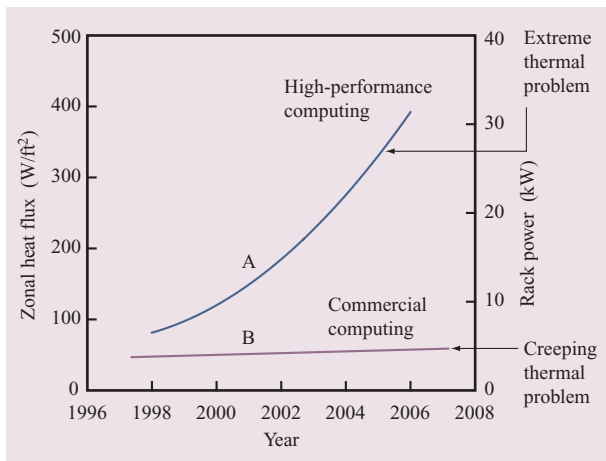


Figure 3

Trends for zonal data center heat flux.

The racks are positioned so that the fronts of the racks face the cold aisle. Similarly, the rear sides of the racks face each other and provide a hot-air exhaust region. This layout allows the chilled air to wash over the front sides of the data processing (DP) equipment, while the hot air from the racks exits into the hot-air aisle as it returns to the inlets of the air conditioning (A/C) units.

With the arrangement of DP equipment in rows within a data center, zones may exist in which all of the equipment contained in a zone dissipates very high heat loads. This arrangement of equipment may be required in order to achieve the desired performance; however, such high-performance zones (**Figure 3**) may create significant challenges to maintaining an environment within the manufacturers' specifications. This figure shows the trend (high-performance computing) for these high-heat-flux zones based on the equipment power trends shown in Figure 1. In contrast, the trend toward data centers containing a mix of computer equipment that employs lower-power racks (commercial computing) is also shown in Figure 3. Many data centers currently follow this lower trend line.

Air distribution configurations

Airflow distribution within a data center has a major impact on the thermal environment of the data processing equipment located in these rooms. A key requirement of manufacturers is that the inlet temperature and humidity to the electronic equipment be maintained within their specifications. To provide an environment of controlled temperature and humidity, two types of air distribution configurations are commonly utilized for such equipment: raised-floor and non-raised-floor layouts. These and other

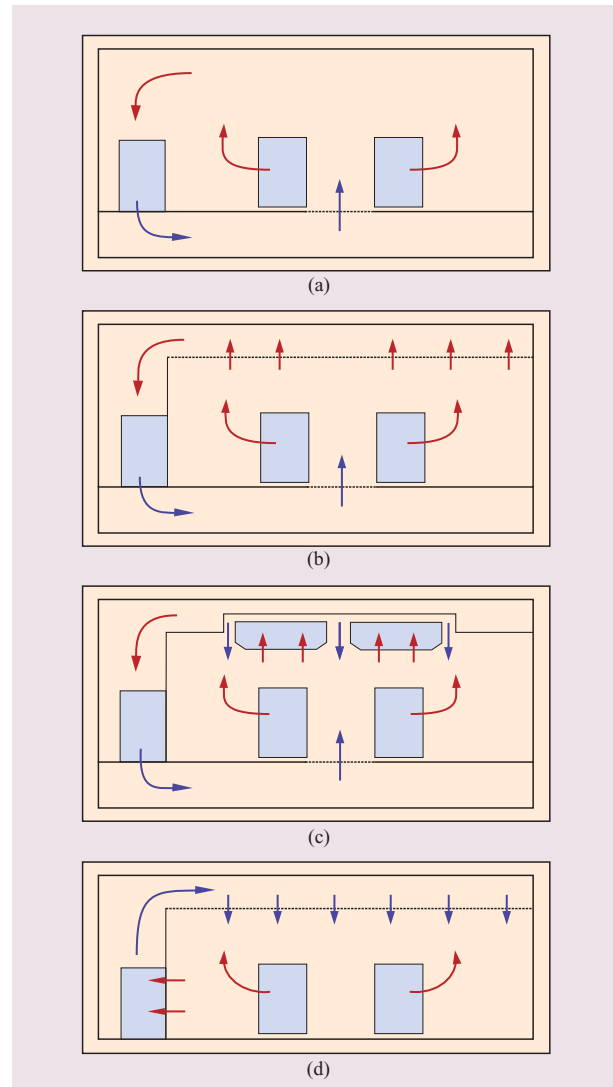


Figure 4

Some data center airflow configurations: (a) raised-floor supply; (b) raised-floor supply/ceiling return; (c) raised-floor supply/ceiling supply; (d) non-raised floor/ceiling supply.

types of air distribution configurations are shown in **Figure 4**.

Raised-floor room cooling

Figure 4(a) shows a raised-floor arrangement in which chilled air enters the room via perforated tiles in the floor and exits the room into air conditioning units known as CRAC (computer room air conditioning) units. Computers typically have numerous cables connecting the components within a rack and between racks. To maintain an organized layout, a raised floor (also known as a false floor or double floor) is used, with other

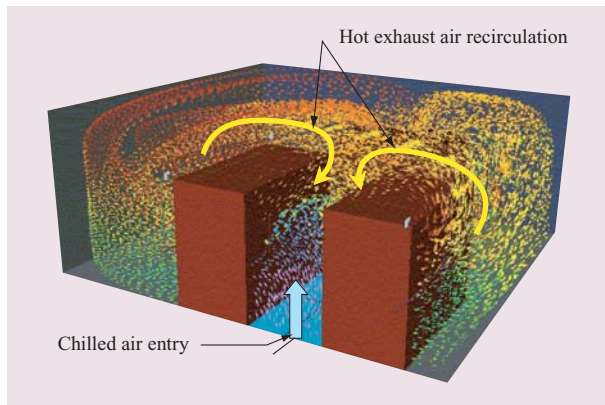


Figure 5

Computer-based simulation of hot-air recirculation in a raised-floor data center.

interconnect cabling connected beneath the raised floor. The space under the raised floor is also used as an air-supply plenum, with perforated tiles exhausting chilled air. Similarly, it is possible for the room to have a false ceiling (also called a dropped ceiling), with the space above the false ceiling used as the air supply or the return plenum, as shown respectively in Figures 4(b) and 4(c).

Non-raised-floor room cooling

Figure 4(d) shows another configuration, in which the chilled air enters the room from diffusers in the ceiling and exits the room via vents that may be placed at different locations. Cooling air can be supplied from the ceiling in the center of the room, where computers are located, with exhausts located near the walls. Short partitions are installed around the supply openings to minimize short-circuiting of supply air to returns (short-circuiting occurs when air takes the path of least resistance). Similarly, cool air from outlets distributed across a ceiling can be supplied, with the exhaust located around the perimeter or a return in the floor, as shown in Figure 4(d). Alternatively, a design employed by the telecommunications industry and more recently adopted by the computer industry utilizes heat exchangers located above the racks near the ceiling. The racks are arranged using the hot and cold aisle concept in which hot air is collected in the heat exchanger and, once cooled, is forced down into the cold aisles, using the fans mounted at the bottom of the heat exchanger.

Factors influencing rack air inlet temperatures

Temperature and humidity specifications

As previously stated, the primary focus in thermal management for data centers is to meet the temperature

and humidity requirements for the electronic equipment housed within the data center. For example, one large computer manufacturer has a 42U rack² configured for front-to-back air cooling and requires that the temperature of the air supplied to the front of the rack be maintained between 10°C and 32°C for elevations up to 1,287 m (4,250 feet). Higher elevations require lowering of the maximum dry bulb temperature 1°C for every 218 m (720 feet) above an elevation of 1,287 m (4,250 feet) up to a maximum elevation of 3,028 m (10,000 feet). These temperature requirements are to be maintained over the entire front of the 2-m height of the rack where air is drawn into the system.

Hot rack exhaust air recirculation—A major cooling problem

In such air-cooled racks, with airflow nominally front-to-back, the chilled-air supply, whether from a raised-floor tile [Figure 4(a)] or via diffusers from the ceiling [Figure 4(d)], is typically only a fraction of the rack airflow rate. This is due to the limitation of tile or diffuser flow rate. The remaining fraction of the supply-side air is made up by ambient room air through recirculation. This recirculating flow is often extremely complex in nature and can lead to rack inlet temperatures that are significantly higher than expected. **Figure 5** shows the complex air circulation patterns resulting from a computer-based simulation of a raised-floor design such as the one depicted in Figure 4(a). As may be seen in Figure 5, there is significant recirculation of hot exhaust air, which can be detrimental to the performance and reliability of computer equipment. Elimination of this recirculation with barriers above the datacom racks is typically not a viable alternative because of fire code restrictions. Data processing equipment is typically designed to operate for rack air inlet temperatures in the 10°C to 35°C range. Because of recirculation, there could be a wide range of inlet air temperatures across the rack. For a raised-floor layout [Figure 4(a)], the inlet air temperature can range from 10°C–15°C at the bottom of the rack close to the chilled air supply to as much as 30°C–40°C at the top end of the rack, where the hot air can form a self-heating recirculation loop. Since the rack heat load will be limited by the rack inlet air temperature at the “hot” part, this temperature distribution correlates to inefficient utilization of available chilled air. Also, since data center equipment almost always represents a large capital investment, it is of paramount importance from a product reliability, performance, and customer satisfaction standpoint that the temperature of the inlet air to a rack be within the desirable range. The efficient cooling of such computer systems and the amelioration of

²U is equal to 1.75 in.

localized recirculation currents of hot air returned to the rack constitute a focus item of the present paper.

Impact of high financial investment in data centers

In addition to a significant increase in the power density of the equipment in the data center, other factors affect data center thermal management. Information technology equipment must be deployed quickly to obtain maximum use of a large financial asset; this may mean that minimal time can be spent on site preparation, which can result in thermal issues after the equipment is installed. The construction cost of a data center exceeds \$15,000/m² in some metropolitan areas, with an annual operating cost of \$500 to \$1,500/m². For these reasons, the aim is to obtain the most from data center space and maximize the utilization of the infrastructure. Unfortunately, the current situation in many data centers does not allow for this optimization. The equipment installed in a data center may have been obtained from many different manufacturers, each having a different environmental specification. With these varying requirements, the data center must be overcooled to compensate for the equipment with the tightest requirements.

Thermal profiling of high-power-density data centers

To aid in the advancement of data center thermal management, Schmidt published the first paper that presented a complete thermal profile of a high-power-density data center [23]. The paper provided basic information on the thermal/flow data collected from such a data center, and it also provided a methodology for others to follow in collecting thermal and airflow data from data centers so that the information could be assimilated for comparison. This database could then provide the basis for future data center air cooling design and aid in the understanding of deploying racks with higher heat loads in the future.

The data center profiled was the National Center for Environmental Prediction (NCEP), located in Bethesda, Maryland. All of the equipment was located on a raised floor in an enclosed area of 22.6 m × 25.6 m (74 ft × 84 ft). A plan view of the data center, indicating the location of the electronic equipment, power distribution units (PDUs), CRAC units, and perforated floor tiles, is shown in **Figure 6**. Most of the servers (51 racks) are IBM Model 7040 (pSeries® 690). The other systems were a mix of switching, communications, and storage equipment.

The key parameters measured within the data center in order to calculate the total airflow rate into the room were flow rates from perforated tile openings, cable openings, and other openings. The total power entering the room was also measured using PDUs and selected

Table 1 Key characteristics of the NCEP data center.

Parameter	Measurement
Perforated tile flow rate	21.4 m ³ /s (61% of total flow)
Cable opening flow rate	9.5 m ³ /s (27% of total flow)
Other opening flow rate	2.7 m ³ /s (8% of total flow)
Tile leakage flow rate	1.5 m ³ /s (4% of total flow)
Total power dissipated	534.4 kW
Heat flux in region of p690	1,830 W/m ² (170 W/ft ²)
Average perforated tile flow rate	0.17 m ³ /s (363 ft ³ /min)
Average temperature rise to top portion of rack	10.5°C

racks. Finally, the inlet air temperatures for the racks at a height of about 1.8 m (6 ft) were measured. Preliminary temperature measurements showed that the location near the top of the rack exhibited one of the highest inlet air temperatures. Measurements of power and airflow rate were also obtained from each CRAC unit located around the perimeter of the room; these measurements were deemed critical to thermal profiling of the data center. Some of the parameters of this data center are given in **Table 1**.

The data indicated that the total airflow rate, including airflow through the perforated tiles adjacent to the racks and the cable openings associated with a rack, did not equal the rack flow rate. For the p690 racks, the airflow through the perforated tiles adjacent to the rack was approximately one fourth to one half of the airflow through the rack in order to maintain system inlet air temperatures within the specifications. The perforated-tile airflow rate plus the cable cutout flow rate was about one half to two thirds of the total airflow through the rack. Even though the chilled airflow adjacent to the server racks did not appear adequate, the total room flow rate was adequate to handle the data center heat load. The convection currents that occurred at the room level were sufficient to bring the local air temperatures for the high-powered racks within the temperature specifications.

More recent data center measurements were conducted on a cluster of 134 server racks in the IBM Poughkeepsie Data Center, as shown in **Figure 7**. The machines were mainly IBM p690 and p655 servers. The area of the data center studied was 688 m² (7,400 ft²), and the average heat flux was approximately 1,615 W/m² (150 W/ft²). However, there existed a high heat flux region spanning 33.8 m² (364 ft²) and encompassing two rows of servers that reached a heat flux level of 6,290 W/m² (585 W/ft²)

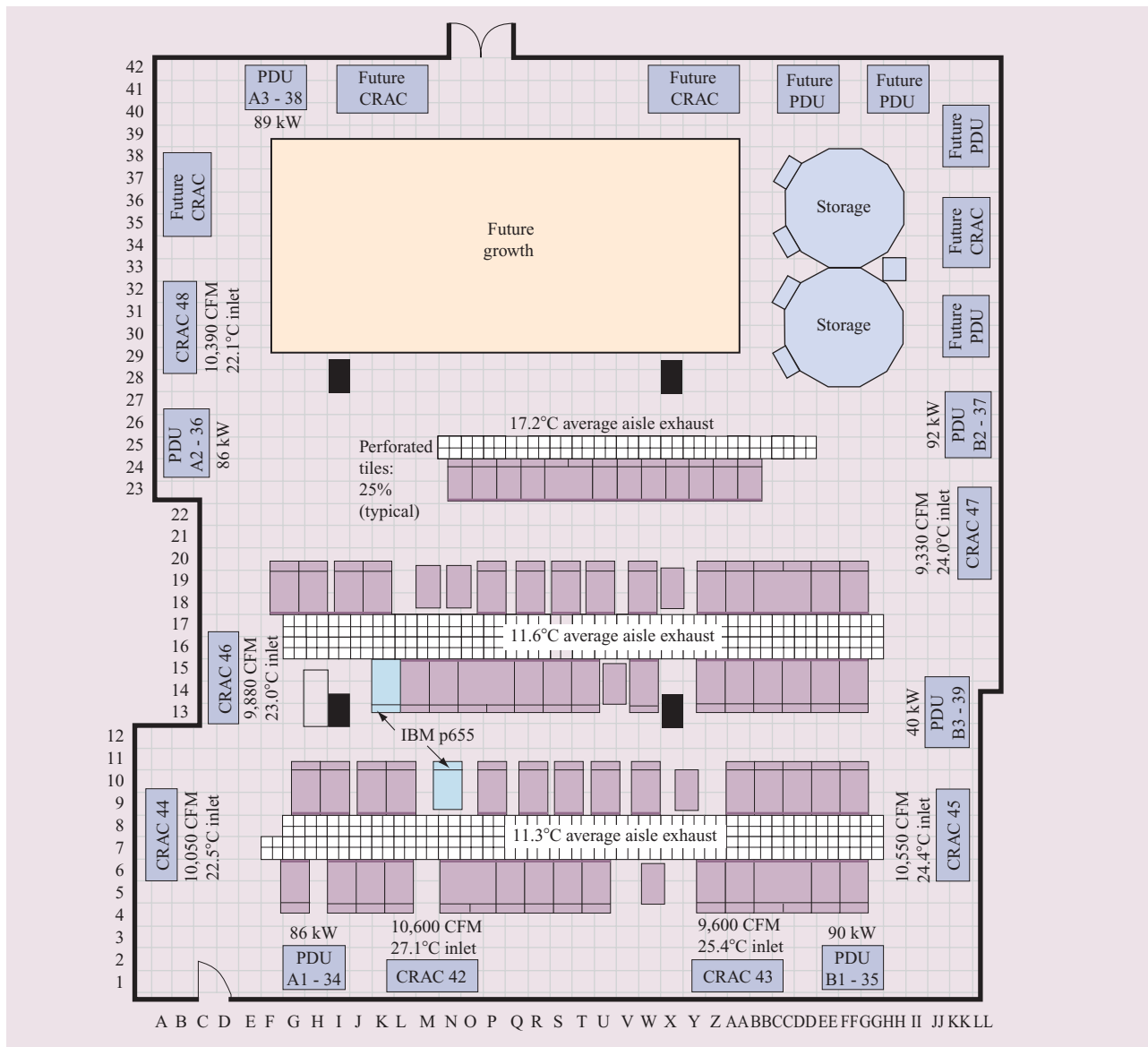


Figure 6

Detailed layout for thermal profiling of high-power-density data center (NCEP.) (CFM = cubic feet per minute.) From [23], reprinted with permission.

(see the area in Figure 7 outlined by the dashed box). Results obtained for this data center were similar to those for the one described above. More significantly, the inlet rack air temperatures measured at a height of 1.8 m were within the equipment specifications even with the very high power densities experienced.

Metrics for data center characterization

Two kinds of metrics are commonly employed in describing a data center—those that help describe the

system (inputs), and those that reveal system performance (outputs).

Description metrics

The most frequently used description metric is the heat flux of the data center (W/m^2), which is defined as the ratio of the total heat load to the footprint area of the layout under consideration. By extending the heat flux to include the volume of the facility, a volumetric heat load (W/m^3) can also be used.

Norota et al. [24] utilized a statistical metric known as the Mahalanobis generalized distance, D^2 , in an innovative manner to characterize the nonuniformity in rack heat load throughout a data center facility. In the data center context, D^2 can be computed as

$$D^2 = (X - x_{ave} \ Y - y_{ave}) S^{-1} \begin{pmatrix} X - x_{ave} \\ Y - y_{ave} \end{pmatrix}, \quad (1)$$

where X and Y are the central coordinates of the CRAC units, x_{ave} and y_{ave} are the central coordinates of the heat-generating racks, and S^{-1} is the inverse of the variance-covariance matrix S . In the rack heat loads across the entire layout, a large D^2 value suggests significant nonuniformity, and a small D^2 value suggests uniformity. Norota et al. conducted experiments for several different rack layouts that yielded a range of D^2 values (3–9) showing a clear correlation between the energy efficiency of the particular layout and its corresponding Mahalanobis Generalized Distance [24].

Performance metrics

The temperature differential between the rack inlet temperature at a specific vertical height and the chilled air entering the data center environment, either through a perforated tile (raised floor) or through an overhead diffuser (non-raised floor), has been used by Schmidt [1] and by Schmidt and Cruz [2–6] to identify localized hot spots.

Dimensionless metrics that characterize data center global cooling efficiency can be found in [25]. These metrics, namely the supply heat index (SHI) and the return heat index (RHI), help evaluate the design of hot and cold aisles. The SHI for a data center is expressed as

$$SHI = \frac{\Delta h_{c-h}}{\Delta h_{total}}, \quad (2)$$

where Δh_{c-h} is the air enthalpy rise due to the heat transfer between the cold and the hot aisles, and Δh_{total} is the total enthalpy rise in the air that passes through the racks.

The RHI for a data center is calculated using

$$RHI = \frac{\Delta h_{crac}}{\Delta h_{total}}, \quad (3)$$

where Δh_{crac} is the enthalpy removed from the hot exhaust air that passes through the chilled air conditioning units (CRACs).

The sum of the SHI and the RHI is equal to unity. A higher value of SHI indicates poor data center design due to significant superheating of the chilled air in the cold aisle, caused by mixing of the cold- and hot-aisle air streams. This results in low data center cooling efficiency. If the hot and cold aisles are perfectly insulated from each other, the value of SHI is 0, and the data center is

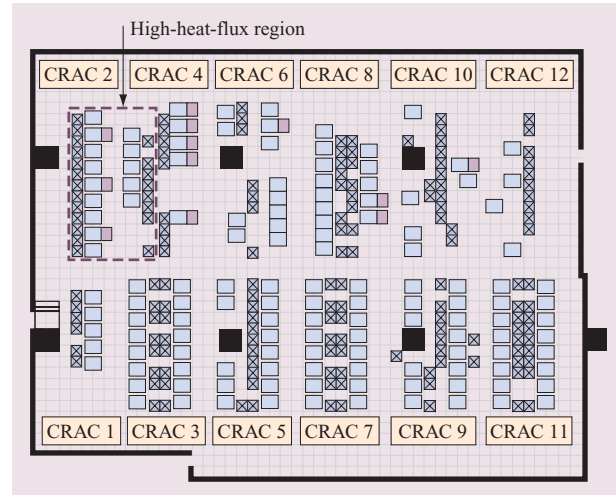


Figure 7

High-heat-flux data center.

perfectly efficient with respect to its flow configuration. Hot-air recirculation in the data center also adversely affects the cooling efficiency by increasing the value of SHI. However, the SHI and RHI are largely “global” metrics, and do not characterize the localized nature of data center hot spots. A data center could display very favorable values for SHI (low) and RHI (high), while possessing significant local hot spots which would result in device failure.

This local inefficiency can be captured by using a ratio of the temperature differentials, β , as given by

$$\beta = \frac{\Delta T_{inlet}}{\Delta T_{rack}}, \quad (4)$$

where ΔT_{inlet} is the temperature differential between the rack inlet air and the chilled-air entry, and ΔT_{rack} is the temperature rise in the air as it passes through the racks.

The common range for β is between 0 and 1. The numerator for Equation (4) that defines β represents a local value, while the denominator is an average rack value. Therefore, the value of β is different for different locations in front of the rack (as well as for different racks). Some parts of the rack may show a β value of 1 or possibly greater than 1, but the averaged value will always be between 0 and 1. A value of 0 for β indicates no impact of hot-air recirculation at that location, and a value of 1 indicates that the local inlet temperature is the same as the average rack exit temperature. A value for β greater than 1 means that there exists a local self-heating loop that raises the air temperature locally more than if it had passed just once through the rack.

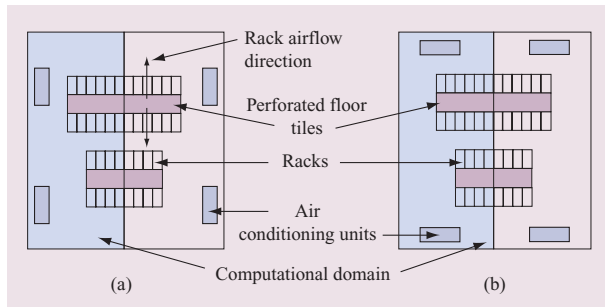


Figure 8

CFD model data center layout: (a) A/C units perpendicular to rows; (b) A/C units parallel to rows. From [2], reprinted with permission.

Norota et al. [24] defined a heat-source energy consumption coefficient, ε_{COM} , that is calculated using

$$\varepsilon_{\text{COM}} = \frac{\sum_n (q_{\text{rack}} + W_{\text{blower}})}{\sum_n q_{\text{rack}} + \sum_m W_{\text{crac}}}, \quad (5)$$

where q_{rack} is the rack heat load, n is the number of racks, W_{blower} and W_{crac} are the work performed respectively by the blower and the air conditioners, and m is the number of computer room air conditioning (CRAC) units.

Metrics similar to ε_{COM} are commonly used to evaluate the energy efficiency of the data center infrastructure.

Data center numerical modeling

Raised-floor configuration

Using numerical modeling techniques, Schmidt [1] and Schmidt and Cruz [2–6] studied the effects of various parameters on the inlet temperature of a raised-floor data center. The numerical models were run using Flotherm 4.2 [26], a commercially available finite control volume computer code. The computational fluid dynamics (CFD) code used a $k-\varepsilon$ turbulence model³ in the flow solver. The data center consisted of 40 racks of data processing (DP) equipment, with front-to-back cooling, which were placed in rows in an alternating hot aisle/cold aisle arrangement located in the center of the floor with four CRAC units at the perimeter of the 12.1-m-wide by 13.4-m-long room. The chilled cooling air provided by the CRAC units was delivered through perforated tile openings via the air plenum under the floor. Only the above-floor (raised-floor) flow and temperature distributions were analyzed in order to reduce the computation domain and time. By taking advantage of the symmetry in the model, only half of the data center had to be modeled. The results of these

papers are presented in order to provide some guidance on the design and layout of a data center.

A plan view of the two data center layouts is shown in **Figure 8**. Two symmetrical arrangements of the CRAC units were investigated, thereby requiring modeling of only half of the data center. Figure 8(a) shows the CRAC units perpendicular to the rows of racks, while Figure 8(b) shows the CRAC units in the parallel configuration. Each of the four CRAC units was sized to provide the necessary heat removal capability for one quarter of the heat dissipated by the electronic equipment inside the DP units installed on the raised floor. The physical size of the CRAC units (890 mm wide by 2,440 mm long by 1,800 mm high) was consistent with those offered by manufacturers in the industry. The DP equipment, which used front-to-back cooling, was arranged in rows so that the inlets faced each other, creating the cold aisle, and the backs faced each other, creating the hot aisle. This type of layout allows the chilled air to exhaust from the cold-aisle perforated tile openings and wash the fronts of the DP racks while the hot air exhausting from the DP racks exits into the hot aisle before returning to the CRAC unit inlets. All of the DP racks measured 610 mm wide by 1,220 mm deep by 2,000 mm high. The spacing between the rows of DP racks was 1,220 mm, so that two 610-mm by 610-mm square raised-floor tiles separated the rows.

Owing to the flow rates and sizes involved in a data center model, the flow was turbulent and was modeled using the two-equation $k-\varepsilon$ turbulence model. The $k-\varepsilon$ model is the most appropriate, given the large scales and open spaces of the data center. In the numerical model, the entire volume that is being modeled is discretized into a large number of cells. The $k-\varepsilon$ model is used to calculate the fluid (air) viscosity parameter for each cell for the turbulent flow regime. The steady-state form of the governing equations is standard and can be found in [28]. In each of the papers [1–6], the same nonuniform 135,000-cell grid was used. An appropriate grid sensitivity study on the nonuniform grid can be found in [1].

The typical temperature difference in DP equipment and CRAC units is 10°C. In order to maintain the 10°C air temperature difference as the power dissipation of the DP racks increases, the flow rate through the rack must increase in direct proportion. This is not always the case in practice, since many DP racks with higher heat loads are now experiencing temperature differences as high as 15°C to 20°C. For a rack heat load of 4 kW, the resulting flow rate was 20 m³/min to maintain the 10°C temperature difference across the DP rack. In this closed system, the mass flow rate under the floor had to match the CRAC floor rate. For the case in which the floor-tile flow rate matched the rack flow rate, the CRAC units delivered 200 m³/min each.

³In the $k-\varepsilon$ model, k is the turbulence kinetic energy and ε is the turbulence dissipation rate [27]; this is the most popular numerical model used in commercial solvers.

Three temperature monitor points were added to the inlets of each of the DP racks. One point was placed at each of three levels above the raised floor: 1,000 mm, 1,500 mm, and 1,750 mm. The papers [1–6] presented data on the rise of the air temperature at the inlets of the DP racks over the perforated-tile air temperature. This metric allowed for comparison of the effects of different changes to the data center and its DP equipment on the inlet temperature of the DP racks. DP equipment manufacturers require that certain inlet air temperature specifications be met in order for the DP equipment to run reliably. Considering the amount of capital invested in the DP equipment and the critical business applications that are run on it, keeping it running reliably is a major concern for IT and facilities personnel.

In [1], Schmidt investigated two DP rack heat loads (4 kW and 8 kW), two CRAC unit locations (perpendicular and parallel to the rows of DP racks), three ceiling heights (2.44 m, 2.74 m, and 3.05 m), and three variations of chilled airflow rates exhausting through the perforated tiles. The boundary conditions in the numerical model for the three variations in chilled airflow rates from the perforated tiles were achieved by adjusting these flow values in the model to be equal to three fractions (80%, 100%, and 120%) of the airflow rate through the rack that results in a 10°C increase in air temperature through the DP equipment rack. For example, in the 4-kW DP rack heat-load cases, each of the DP racks moved 20 m³/min of air to maintain a 10°C temperature difference across the DP rack, and the perforated tile directly in front of each DP rack produced 16, 20, and 24 m³/min of chilled air for the 80%, 100%, and 120% cases, respectively. The flow through each of the DP racks was left unchanged for this study. It was discovered that hot-air recirculation created by two different means was the main cause of increased inlet temperatures to the DP racks. The first case was recirculation of hot exhaust air over the top of the DP rack back into the front of the same rack, as may be seen in Figure 6. In the second case, hot exhaust air recirculated around the sides of the end DP racks. These recirculation cells were present in each case, but the recirculation over the tops of the DP racks was substantially more pronounced with increased ceiling height and in the perpendicular CRAC unit layout. DP rack inlet temperatures increased as much as 9°C when the ceiling height was increased from 2.44 m to 3.05 m. The second means of hot-air recirculation was created from the hot exhaust of neighboring rows of DP racks. Decreasing the under-floor airflow rate also increased the inlet temperatures by increasing the amount of air that each DP rack pulled from the room. Increases in DP rack inlet temperatures were in the range of 10°C to 15°C in both the 4-kW and 8-kW cases. The primary result of the

study was that once the chilled air from the perforated tiles was drawn in by the lower portions of the rack, satisfying conservation of mass, the upper region of the rack required that air be drawn in from other portions of the room, primarily heated exhaust air from the rear of the same rack.

Schmidt and Cruz [2] examined the effects of shifting some of the chilled air from the cold aisle to the hot aisle. This was done in order to examine the effects of cable cutouts at the backs of the DP equipment and also to determine whether the injection of cool air into the hot aisle helped reduce the recirculated air temperature at the inlets of the DP racks. The variables altered in this paper consisted of three rack powers (4 kW, 8 kW, and 12 kW), two CRAC unit locations (perpendicular and parallel to the rows of DP racks), and nine variations of chilled airflow rates exhausting through the perforated tiles. A fixed ceiling height of 2.74 m was chosen for all simulation runs. As in [1], the total chilled air provided to the data center by the CRAC units was varied to 80%, 100%, and 120% of the DP rack flow rate, but it was distributed in three configurations to both the cold and hot aisles. In the first case, 100% of the chilled air exited in the cold aisle, while the second case delivered 75% of the flow to the cold aisle, with the remaining 25% exiting in the hot aisle. In the third case, the chilled air was split evenly between the cold and hot aisles so that 50% went to the cold aisle and 50% exited in the hot aisle. Exhausting the chilled air into the hot aisle did not help reduce the DP rack inlet temperatures, and, in all cases, it was better to have all of the available chilled air exit in the cold aisle.

Schmidt and Cruz [5] studied the effect of situating higher-powered (12-kW) DP racks among lower-powered (4-kW) DP racks. Twelve high-powered DP racks were arranged with half of them having a single high-powered DP rack and the other half having two high-powered DP racks. A fixed ceiling height of 2.74 m and a fixed total perforated-tile flow of 80% of the total DP rack flow rate were chosen for all simulation runs. Two different perforated-tile flow arrangements were studied; the first had an unrealistic increase in the perforated-tile flow in only the tile directly in front of the high-powered DP racks, while the second arrangement distributed the additional flow across the cold aisle with the high-powered DP racks. In the first case, the inlet temperatures of the high-powered DP racks did not exceed 4°C above the under-floor air temperature. However, compared with the baseline case of a fully populated low-powered DP data center, the temperatures of the low-powered DP racks increased significantly at an inlet height of 1,500 mm, with most increases in the range of 5°C to 10°C and the maximum being 21°C. At a height of 1,750 mm, the low-powered DP racks showed minimal

Table 2 Effect of removed racks on inlet air temperatures into high-powered racks (at height of 1,750 mm)—temperature reductions compared to baseline case (°C) [4].

No. of adjacent removed racks	One high-powered rack		Two high-powered racks	
	Average ΔT inlet (°C)	Standard deviation in ΔT inlet (°C)	Average ΔT inlet (°C)	Standard deviation in ΔT inlet (°C)
1	8.9	2.8	7.1	3.1
2	12.0	0.8	12.8	1.9
3	14.8	0.5	13.8	1.9
4			16.5	1.5

Table 3 Perforated tile flow rates (m³/min) to maintain an average inlet air temperature of 10°C [3].

Rack heat load (kW)	Perforated-tile flow rate (m ³ /min)		Increase in flow rate from tile (%)
	For $\Delta T = 20.8^\circ\text{C}$ across rack	For $\Delta T = 10^\circ\text{C}$ across rack	
4	12.5	18	44
8	18	20.5	14
12	20	25.5	28

changes compared with the baseline case. The inlet temperatures in the second case, where the additional chilled airflow was distributed along the cold aisle, showed a high dependency on the location of the high-powered DP racks. When the high-powered DP racks were placed near the symmetry line, the inlet temperatures at 1,750 mm increased as much as 24°C over the under-floor air temperature. However, when the high-powered DP racks were near the ends of the row, the temperature rise was on average only 13°C above the under-floor air temperature. At the height of 1,750 mm, the temperature of the high-powered DP racks ranged from −3°C to 5°C above that of the low-powered DP racks at the same height. This is a minimal difference considering a factor of 3 increase in airflow and power dissipated. Low-powered DP racks located near the high-powered DP racks showed an increase in inlet temperature at the 1,500-mm height, while those farther away from the high-powered DP racks showed a decrease. When the high-powered DP racks were near the ends of the rows, the inlet temperatures of the low-powered DP racks decreased at a height of 1,750 mm, in some cases by as much as 16°C. On the basis of these results, the best position for the high-powered DP racks is near the end of the row.

Schmidt and Cruz [4] found that removing DP racks adjacent to a high-powered DP rack reduced the air inlet

temperatures of not only the high-powered DP rack, but also all of the neighboring low-powered racks. The effects of removing DP racks on air inlet temperature into the high-powered DP rack can be seen in **Table 2**. These results show that, as more and more DP racks are removed close to the high-powered DP rack, the inlet air temperatures of the high-powered DP rack decrease. The greatest improvement in inlet air temperatures to the high-powered DP racks occurred when only one DP rack was removed adjacent to the high-powered DP rack. The low-powered DP racks situated on the same cold aisle as the removed DP racks experienced reduced inlet air temperatures, with the DP racks closest to the removed DP racks experiencing the greatest reduction in inlet air temperature.

Schmidt and Cruz [3] also examined the effects of reducing the airflow rate through the DP rack for a higher temperature rise across the DP rack. Although the airflow rate exiting the floor tiles matched the DP rack airflow rate, decreasing DP rack airflow rates [or increasing the change in temperature (ΔT) through the DP rack] increased DP rack air inlet temperatures. This result was unexpected, since recent studies predict that the air temperatures into the DP racks would have not varied much if the chilled airflow rate exiting the perforated tiles matched the DP rack flow rate. However, for the same perforated-tile flow rate, decreasing the DP rack flow rate resulted in decreased DP rack air inlet temperatures. For example, for clusters of uniformly powered 8-kW DP racks, the average inlet temperature of all DP racks at a height of 1,750 mm decreased by 12°C when the DP rack flow rate was cut in half. To maintain a given inlet temperature, the chilled airflow rate exhausting from the perforated tiles should increase with increased rack flow rate (see **Table 3**).

Schmidt and Cruz [6] explored the effects of under-floor air distribution on the above-floor DP equipment air inlet temperatures. Earlier papers had focused only on the above-floor air and assumed a uniform perforated-tile air distribution. The space underneath the raised floor was modeled using a commercially available CFD code called

Table 4 Average air inlet temperature rise (°C) for DP rack powers, different perforated-tile open areas, and raised-floor heights [6].

<i>Rack power = 4 kW</i>		<i>Average air inlet temperature for different perforated-tile free areas and different raised-floor plenum heights (°C)</i>					
<i>Variation in rack air inlet height (mm)</i>	<i>Uniform</i>	<i>25% free area in perforated tile</i>			<i>60% free area in perforated tile</i>		
		<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>	<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>
1,750	16.1	15.3	15.8	16.0	9.8	13.4	14.2
1,500	6.6	6.1	6.5	6.5	5.9	6.3	6.2
1,000	0.0	0.0	0.0	0.0	1.0	0.2	0.0

<i>Rack power = 8 kW</i>		<i>Average air inlet temperature for different perforated-tile free areas and different raised-floor plenum heights (°C)</i>					
<i>Variation in rack air inlet height (mm)</i>	<i>Uniform</i>	<i>25% free area in perforated tile</i>			<i>60% free area in perforated tile</i>		
		<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>	<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>
1,750	7.7	3.2	3.5	3.5	2.3	3.0	3.2
1,500	3.5	1.9	2.1	2.1	1.9	2.1	2.0
1,000	1.2	1.2	1.0	1.0	1.8	1.3	1.0

<i>Rack power = 12 kW</i>		<i>Average air inlet temperature for different perforated-tile free areas and different raised-floor plenum heights (°C)</i>					
<i>Variation in rack air inlet height (mm)</i>	<i>Uniform</i>	<i>25% free area in perforated tile</i>			<i>60% free area in perforated tile</i>		
		<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>	<i>For height of 0.15 m</i>	<i>For height of 0.30 m</i>	<i>For height of 0.60 m</i>
1,750	6.9	2.3	2.3	2.3	1.5	1.9	2.3
1,500	4.2	1.7	1.7	1.7	1.4	1.8	1.9
1,000	1.8	1.2	1.2	1.2	1.7	1.4	1.2

Tileflow**. The CFD calculation requires the solution of the three-dimensional turbulent flow in the under-floor space. As in the above-floor CFD model, the turbulence is characterized by the $k-\varepsilon$ turbulence model. The inflow to the calculation domain is created by the CRAC units, while the outflow is governed by the local pressure under the perforated tiles and the resistance of the perforated tiles. The computation takes into account the flow resistance due to under-floor obstructions (such as cables and pipes) and the resistance of the support structure for the raised floor. Three DP rack powers (4 kW, 8 kW, and 12 kW), two perforated-tile percentage-open areas (25% and 60% open), and three raised-floor heights (0.15 m, 0.30 m, and 0.60 m) were examined. For the same data center total airflow rate, the smaller the plenum height and the greater the perforated-tile percentage open, the more poorly distributed was the airflow exiting the tiles.

For the 4-kW case, the more poorly distributed the airflow, the lower were the average DP rack inlet temperatures (see **Table 4**). Slight variations in poorly distributed tile airflows (less than 8%) resulted in significant inlet temperature reductions of up to 8°C for the 8-kW case and 10°C for the 12-kW case.

Non-raised-floor configuration

A representative non-raised-floor data center system, depicted in Figure 4(d), has been modeled using the commercial software tool Flotherm. The rack layout for this analysis is identical to that for the raised-floor studies as conducted [1–6] and illustrated in Figure 8. The non-raised-floor model, comprising a half-symmetry section of a 40-rack data center arranged in a cold-aisle/hot-aisle fashion, was organized as two rows of six racks and two rows of four racks, respectively. Overhead ducts that feed

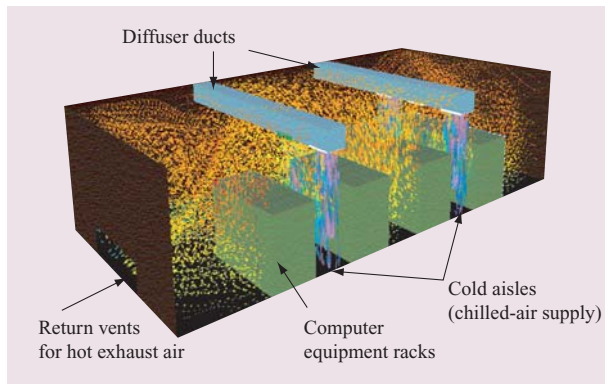


Figure 9

Schematic of non-raised-floor model for computer-based simulations.

chilled air to the diffusers were constructed as part of the model. Return air vents were placed at either the bottom or the top of the bounding walls of the model and were located on the wall perpendicular to the hot and cold aisles, or on the walls parallel to the aisles.

Figure 9 illustrates the results of a simulation using the non-raised-floor CFD model developed for this study. Symmetry boundary conditions were enforced on the center wall that is in contact with the racks. The fluid flow was numerically solved using the $k-\epsilon$ turbulence model. The size of the model ranged from 150,000 to 200,000 elements. Satisfactory convergence for temperature and continuity was typically achieved within 1,000 iterations, and required approximately two to four hours on a

high-end PC. Temperature data was recorded for the numerical simulations at five specific locations in front of the rack, namely at vertical heights of 330, 660, 1,000, 1,333, and 1,666 mm. Average temperatures for five volumetric regions at the inlet to the rack were also ascertained.

A parametric study is currently being conducted to characterize the effect on the temperature of the inlet air to the rack. **Table 5** provides detailed information regarding variables that were considered for the study. As seen in the table, the input variables to the design of experiments are the rack heat load ($\times 2$), rack flow rate, air temperature rise through the rack ($\times 1$), diffuser flow rate ($\times 6$), diffuser location (height) ($\times 3$), diffuser pitch and arrangement ($\times 4$), angle of air exiting the diffuser ($\times 5$), room ceiling height ($\times 7$), and return vent location for the exhausting air ($\times 8$). In addition to these variables, special configurations that characterize zero power racks ($\times 9 = \text{alternate zeros}$) as well as nonuniform rack heat load ($\times 9 = \text{alternate halves}$) are incorporated into the study. Statistical techniques are to be utilized to analyze the data, and the results are to be used to formulate quantitative and qualitative design guidelines.

Data center energy requirements

The amount of electric power drawn by computer hardware can be very large, and the power required for the cooling can add an additional 30%–50%. For example, the NCEP data center described above and depicted in Figure 6 requires approximately 0.5 MW of power for the computer equipment installed within the

Table 5 Parametric variables for non-raised-floor study.

Variable description			Variation				
Name	Description	Units	1	2	3	4	5
$\times 1$	Rack temperature rise	$^{\circ}\text{C}$	10	15	20	NA	NA
$\times 2$	Rack heat load	kW	4	12	20	28	36
$\times 3$	Diffuser height	m	3.05	At ceiling	NA	NA	NA
$\times 4$	Diffuser location	—	Alternate 1	Alternate 2	NA	NA	NA
$\times 5$	Diffuser angle	degrees	0	30	NA	NA	NA
$\times 6$	Total diffuser flow rate	% of rack flow rate	100	80	60	NA	NA
$\times 7$	Ceiling height	m	3.66	5.18	6.71	NA	NA
$\times 8$	Return air vent location	—	Bottom of perpendicular wall	Top of perpendicular wall	Bottom of parallel wall	Top of parallel wall	NA
$\times 9$	Rack heat load nonuniformity	—	Uniform	Alternate zeros	Alternate halves	NA	NA

NA: Less than five variations considered.

data center. Although a detailed study was not performed on the energy required for cooling, it is estimated that this energy was in the range of 150 kW to 250 kW. On an annual basis, at a rate of 10 cents per kilowatt-hour, this results in an annual energy expense of \$130,000 to \$220,000 per year just for cooling. This is a large expense devoted to cooling the equipment and, considering the number of data centers in the world, requires that a significant effort be put into optimizing the use of this energy. Such an effort requires a detailed examination of the various inefficiencies that occur in a data center in order to determine how to decrease the amount of energy spent.

The design and layout of data centers and their cooling infrastructure are inherently wasteful of energy. Bash et al. [10] outlined three key factors in lowering energy usage and thereby maximizing performance:

1. Minimize the infiltration of hot air into the rack inlets.
2. Minimize the mixing of hot return air with cold-air streams prior to return to the CRAC units.
3. Minimize the short-circuiting of cold air to the CRAC inlet.

They postulated that research in the area of airflow optimization in the data center space was lacking. They maintained that in the absence of airflow research data, air-handling infrastructure in data centers has suffered from overdesign and extravagant redundancy. A methodology for understanding this was outlined and tested on a simple data center model in [29]. From this analysis, Bash et al. [10] focused on the following areas:

1. Three-dimensional numerical modeling of fluid mechanics and heat transfer within data centers, with emphasis on optimization and real-time solutions.
2. Distributed sensing and control of cooling resources within data centers.
3. Development of metrics and correlations to guide data center design and predict performance.

Although much work remains to be done in this area, their view of future progress was outlined in [9]. A cooling system comprising variable-capacity computer room air conditioning units, variable air-moving devices, adjustable vents, etc., which are used to dynamically allocate air conditioning resources where and when needed could be installed. A distributed metrology layer could be used to sense environment variables such as temperature, pressure, and power. The data center energy manager would then redistribute the computer workloads

on the basis of the most energy-efficient availability of cooling resources.

In addition to the optimization that should occur within the data center, solutions should also be pursued to reduce the energy that is supplied to the facilities. Felver et al. [30] showed that a new data center under design for Sandia National Laboratories in Livermore, California, will use variable-air-volume air-handling units that provide backup evaporative cooling. This proposed two-stage evaporative cooling design offers a major benefit to both utilities and data centers by shifting the peak electrical demand of the HVAC system to an off-peak period. This minimizes the burden placed on them by the data centers while allowing electrical power to be purchased by the data center at a reasonable rate during periods of peak demand. The use of this technique, through the elimination of CRAC units, allows for a recapturing of valuable computer-center floor space and a reduced risk of water damage in computer rooms.

Future data centers

Each data center is unique and has its own limitations; for example, data center power density limits vary greatly. To resolve the environmental issues, manufacturers of HVAC equipment have begun to offer liquid cooling solutions to aid in data center thermal management. The objective of these new approaches is to move the liquid cooling closer to the electronic equipment that is producing the heat. Placing cooling near the source of heat shortens the distance that air must be moved and minimizes the required static pressure. This increases the capacity, flexibility, efficiency, and scalability of the cooling solutions. Several viable options based on this strategy have been developed: 1) rear-mounted fin and tube heat exchangers; 2) internal fin and tube heat exchangers, either at the bottom of a rack of electronic equipment or mounted on the side of a rack; and 3) overhead fin and tube heat exchangers. Although each solution involves liquid cooling adjacent to the air-cooled rack, the liquid can be either water-based or refrigerant-based. New approaches such as these and others will continue to be promoted as heat loads of data communication equipment continue to increase.

Summary

With the increased performance requirements of computer equipment and the resulting heat dissipated by this equipment, a significant strain is placed on the data center and its environment. Air cooling for some data centers has reached its limit; racks of equipment with high-density loads must be distributed among other racks with less dense equipment loads in order to spread out the heat and make the data center "coolable." The value proposition of packing the racks as close as possible

for maximum performance is significantly diminished because the air conditioning capacity within a data center is not sufficient to permit such an arrangement. These cases require closer examination and a better understanding of the flow and temperature dynamics within each data center. Because of the extreme complexity of this problem and the uniqueness of every data center, this effort is only just beginning. Much more thermal data from data centers must be collected, examined, correlated, and understood, along with detailed thermal and fluid models. Non-dimensional metrics must be identified to provide guidance for data center thermal management. As more is understood of data center thermal behavior, energy optimization and savings may follow. Since data centers typically require an additional 30% to 50% of the data processing energy for cooling, it may be possible to save a large amount of energy in the thousands of data centers around the world. Significant savings can be obtained if an improvement of as little as 5% is achieved in energy-related expenditures. As equipment power densities increase along with expenditures for energy to power this equipment, the HVAC industry as well as the server manufacturers may have to embark on liquid cooling solutions in order to resolve some of the future temperature problems that will occur within the server racks and the environment in which they reside.

Note added in proofs [31]

A water-cooled heat exchanger attached to the rear door of the rack reduces the effect of the hot-air recirculation. Extraction of heat at the point of generation is much more efficient than with CRACs positioned around the perimeter of a data center. A total cost-of-ownership analysis was performed which showed a significant advantage in both first-time cost and annual cost when this attached heat exchanger approach is used.

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