Effect of Rack Server Population on Temperatures in Data Centers

Rajat Ghosh, Vikneshan Sundaralingam, Yogendra Joshi
G.W. Woodruff School of Mechanical Engineering
Georgia Institute of Technology,
Atlanta, GA 30332-0405
Phone: (404) 385-2810
Fax: (404) 894-8496
Email: yogendra.joshi@me.gatech.edu

ABSTRACT

To reduce energy wastage during equipment upgrades that require changing the number of servers inside a rack, this paper investigates the effect of the server population of a rack on air temperatures in a data center facility. A rack-level containment system was implemented and air temperatures in cold and hot aisles with different numbers of servers were measured. A computational fluid dynamics (CFD)-based numerical model is used to study the air flow field. In addition, this paper includes the investigation of the effect of void locations on CPU temperatures. The study reveals that the server population inside a rack has a significant impact on air temperatures.

Keywords: Data Centers, Air Temperature, Energy Efficiency, Containment.

NOMENCLATURE

N	Number of servers in the test rack.
IT	Information Technology.
CFD	Computational Fluid Dynamics.

INTRODUCTION

Typically, data centers employ forced air cooled multi-scale thermal management systems [1]. This requires external power for computer room air conditioning (CRAC) blower units, heat exchangers in the CRAC, and server fans. Benchmarking studies reveal that the cooling infrastructure in a typical data center consumes as much as 40% of the total facility power, and the life-cycle cost of cooling is becoming comparable to that of IT equipment [2]. Nevertheless, a rather conservative design approach [3], prevalent in current practice, wastes a substantial amount of cooling resources [4].Preventing this requires identification of potential opportunities and a detailed characterization of energy usage. For example, streamlining energy usage during an equipment upgrade, which involves changing server population, may result in operational energy savings. Such upgrades cause substantial changes in air temperatures [5]. A detailed characterization of the effect could potentially facilitate a better thermal management strategy for data centers.

In this paper, we investigate how the number of servers in a rack influences surrounding air temperatures. The test rack was isolated through an airflow containment system. The number of servers inside the rack is varied, and air temperatures in the cold and hot aisles are measured. The effect of air flow field on air temperature is also studied via a CFD-based numerical simulation. In addition, we also monitor the effect of void locations inside the test rack on CPU temperatures and server fan speeds.

EXPERIMEANTAL SETUP and MEASUREMENT

Fig. 1 shows details of the test cell. The facility employs a raised floor plenum inflow and ceiling return airflow. The height of raised floor plenum is 914 mm (3') and that of the drop ceiling 1,524 mm (5'). The test section has 10 racks of height 2,000 mm (6'6") arranged in a 2x5 alternating cold/hot aisle arrangement. During the experiment, CRAC-1 operates at 60% capacity and supplies cooling air at a volumetric flowrate of 4.38 m^3/s (9281 CFM) into the floor plenum. The remaining two CRAC units are kept inactive. On average, the porosity of perforated tiles in the cold aisle is 36%.

Fig. 2 shows details of the test rack (Rack 8). It consists of 42 1-U (where 1-U=1.75 inches or 44.45 mm) Dell servers with two dual-core processors and a head node. The heat loads of different servers are measured via OSIsoft PI system [6]. For the test rack, we developed a rack-level containment system, shown in Fig. 3. The containment system simulates an airflow isolation scheme: cooling air from the perforated tile in front of the test rack in the cold aisle is entirely guided into the test rack and hot exhaust air is driven out via an exhaust plane at rack height 2000 mm (6'4"). Airflow rate into the confinement system is measured as 0.3 m^3/s (639 CFM). The containment system prevents intermixing between air flowing through the tile in front of the test rack and other parts of the room. A controlled experiment is designed with the server number as the independent variable.

For temperature measurements in hot and cold aisles, two grid-based [7] thermocouple networks were deployed. Fig. 4 shows a thermocouple grid unit which consists of 21 thermocouples. The thermocouples are arranged in a square. The 590 mm x 590 mm square cross-section of the measurement grid is consistent with the perforated tile dimension and the size of the confinement. The thermocouples used are T-type copper-constantan thermocouples made from 28 gauge thermocouple wires (0.907 mm diameter). These provide a response time of around 20 ms, suitable for rack-level air temperature (time scale~1s) measurements. The temperature data obtained are processed by digital data acquisition system and subsequently transmitted to I/O terminal by a router. In addition, Table 1 shows uncertainties in the measurements.

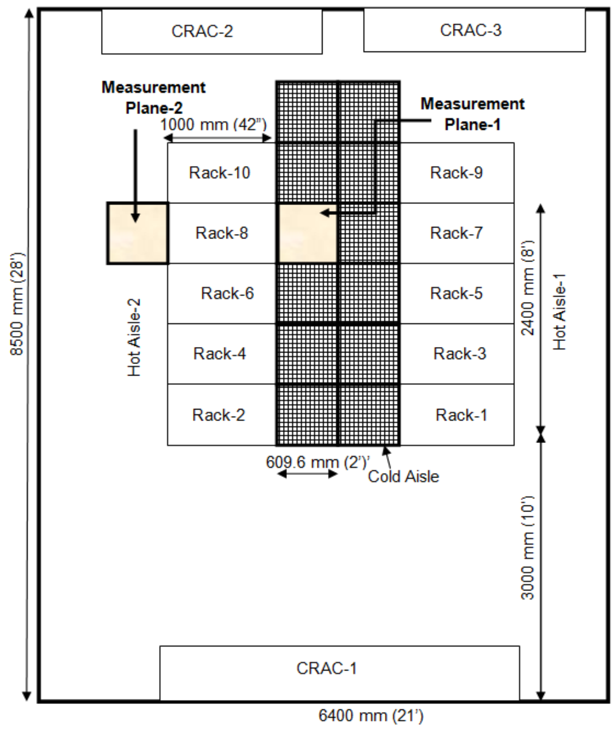

Fig. 1 Plan view of the experimental test cell. The corresponding room height is 2.74 m (9'). Heights of measurement planes are equal to the test rack height of 1.83 m (6').

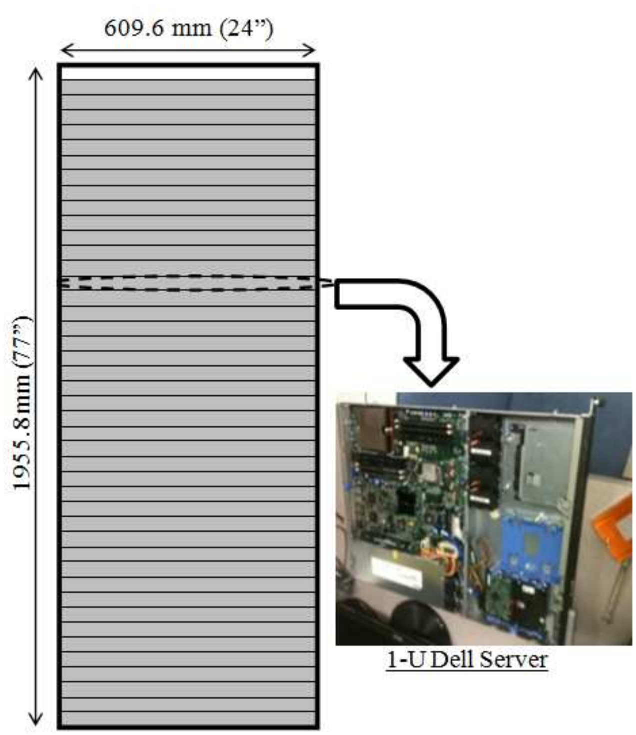

Fig. 2 Front view of the test rack. It includes 42 1-U Dell servers and a headnode. Each Dell server consists of two dual-core processors.

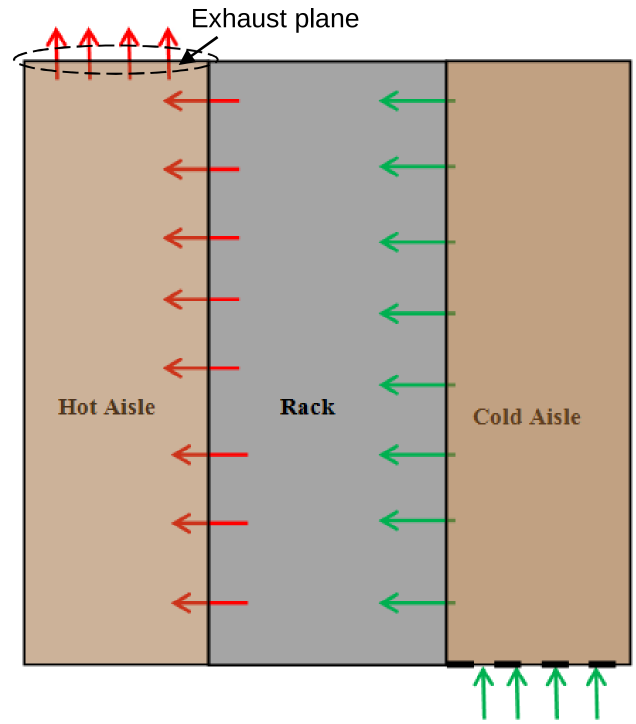

Fig. 3 Rack-level containment system to isolate airflow into the test rack.

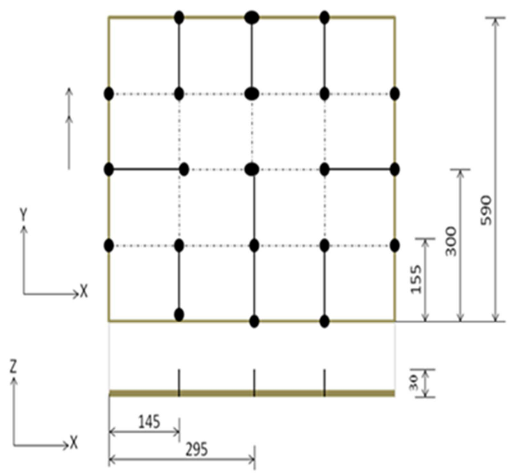

Fig. 4 Thermocouple-based temperature measurement grid. All units are in mm. Z-axis indicates the direction parallel to the rack height.

Table 1. Measurement Uncertainties

Variable	Uncertainty
Temperature	$\pm 0.2^0 C$
Flowrate	0.00944 m3/s (20 CFM)
Length	1 mm

In addition to the experimental measurements, a CFD analysis was also conducted to investigate the details of the flow field in the vicinity of the rack. Fig. 5 shows the geometry for the CFD analysis.

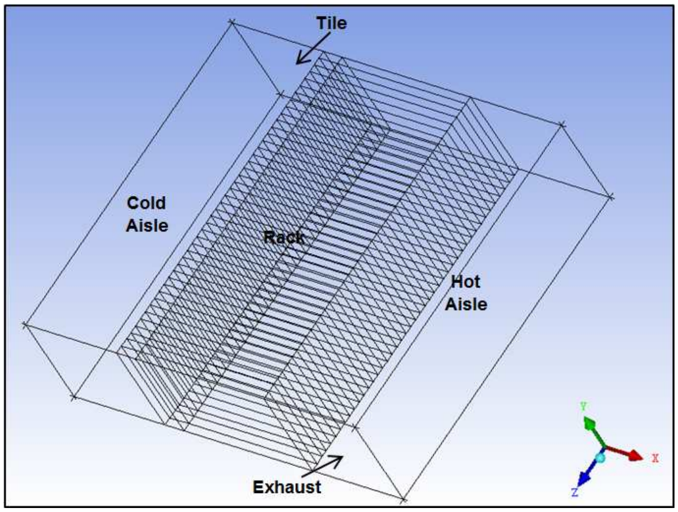

Fig. 5 Geometry for the CFD analysis of rack-level airflow. Cooling air is supplied from the tile and then drawn into servers by server fans. Hot air leaves through the exhaust.

The CFD analysis includes 42 servers and one headnode simulating the rack shown in Fig. 2. Table. 2 includes the details of the CFD analysis. 1.4 million grids are used for the reported solution. In addition, the CFD solution was computed with 200,000, 600,000, and 1 million grids. The deviation between CFD predictions with 1 million and 1.4 million grids is significantly smaller than that of between 200,000 and 600,000 or 600,000 and 1 million. Going beyond 1.4 million grids was computationally prohibitive.

Table 2. Details of CFD Analysis

Parameters	Description
x-dimension	2.46*m*
y-dimension	0.6*m*
z-dimension	1.95*m*
1-U server	Box of 1mx0.6mx0.04m
Server Fan	$p(v) = 0.187v^3 - 6.76v^2 + 38.48v + 315.4$
Tile	Velocity inlet with 0.8 m/s to match 639 CFM (0.3 m^3/s) supply
Exhaust	Pressure Outlet
Server inlet and outlet	Porous Jump
# of nodes	1.4 million nodes provide grid-independent solution

RESULTS

Cold and hot aisles temperatures were measured for N=42, 32, 22, and 12, where N is the number of servers inside the rack. Based on the capacity of the rack, N=42 is equivalent to loading the rack to its full capacity. The server power along the test rack was measured, as shown in Fig. 6, which indicates server powers are non-uniform.

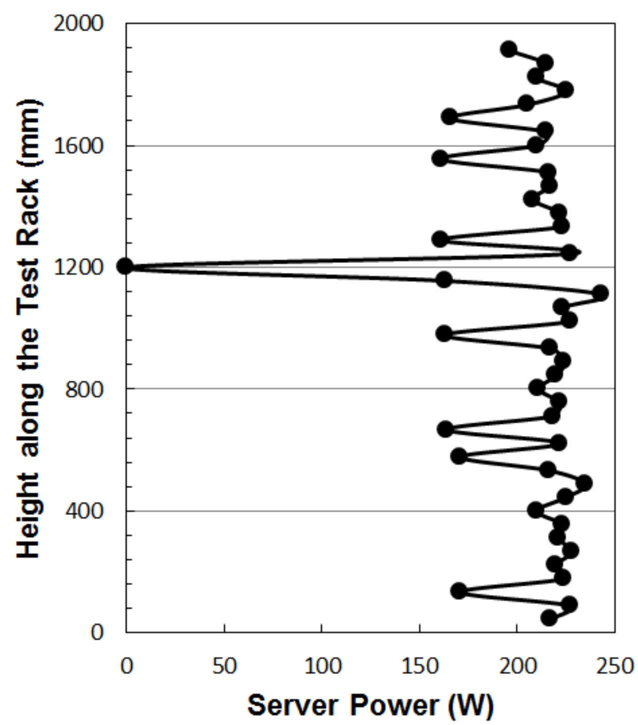

Fig. 6 Server power variation with the height along the test rack. The solid dots indicate corresponding server power. .

During the experiment, the variation in the supply air temperature from the CRAC was monitored. As shown in Fig.7, the maximum variation is nearly 3 ^{o}C.

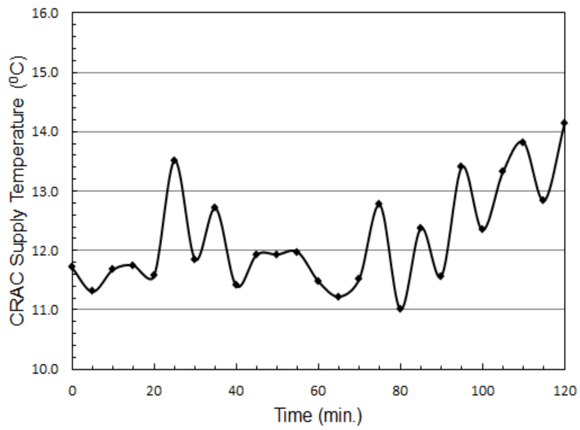

Fig. 7 Temporal variations of supply air temperature from the CRAC unit.

Supply air from the CRAC unit is driven into an under-floor plenum from where it is drawn into the cold aisle confinement via perforated tiles. Thereafter, it is driven into the servers by fans and the momentum of air decreases with height in the cold aisle. As a result, convective air temperature field changes considerably with height. To monitor the variation of the height, air temperatures at six different heights (h) along the test rack: h= 304.8 mm (1'), 610 mm (2'), 914 mm (3'), 1,219 mm (4'), 1,524 mm (5'), and 1,829 mm (6') were measured.

Fig. 8 shows the temporal variation of average air temperatures in the cold aisle at different heights, and it indicates that the average temperature is highest at h=1,829 mm (6'). The hottest zone at the top of the cold aisle can be explained by gradually diminishing cooling air momentum with rack height. Nevertheless, the average cold aisle temperatures do not decrease uniformly with height. In fact, the next hottest zones are at heights h=305 mm (1'), 1,524 mm (5'), 609 mm (2'), 1,219 mm (4'), and 914 mm (3') respectively. Such a non-uniform trend in average temperatures arises from non-uniform server heat loads, and suggests complex airflow patterns exist inside the confinement.

Fig. 9 shows the variation of average air temperatures in the hot aisle, and it indicates the hottest zone is at h=304.8 mm (1') (nearest to the floor). Moving up, the cold room air intermixing decreases average air temperature gradually. The trend is uniform, with decreasing air temperatures for heights h=610 mm (2'), 914 mm (3'), 1,219 mm (4'), and 1,524 mm (5'). However, we observe a sudden rise in average temperature at h= 1,829 mm (6').

The air temperature differences across server inlets and outlets indicate thermal advection at the rack length-scale. Also, since airflow varies with the height of the rack, the temperature difference varies with height along the rack. Fig. 10 shows temperature differences across the test rack at various heights along the test rack with N=42. Fig. 10 suggests that the temperature difference increases as we move up along the rack, and depends on the local airflow structure. The momentum of the cooling air reduces with rack height; therefore, it undergoes a larger change in temperature in comparison to airflow in the lower region.

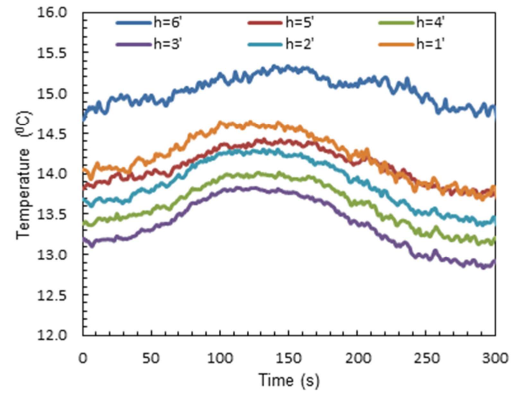

Fig. 8 Temporal variations of average air temperature in the cold aisle at different heights.

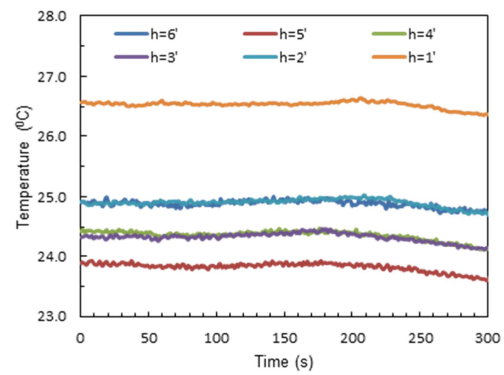

Fig. 9 Temporal variations of average air temperature in the hot aisle at different heights.

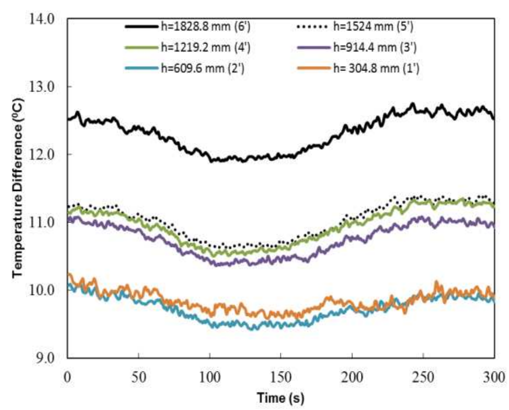

Fig. 10 Temporal variations of average temperature difference across the test rack for different rack heights.

Fig. 11 shows the computed velocity field in the plane y= 304.8 mm (1"), which is at the middle of the computational domain. The details of the computational domain are listed in Table 2.

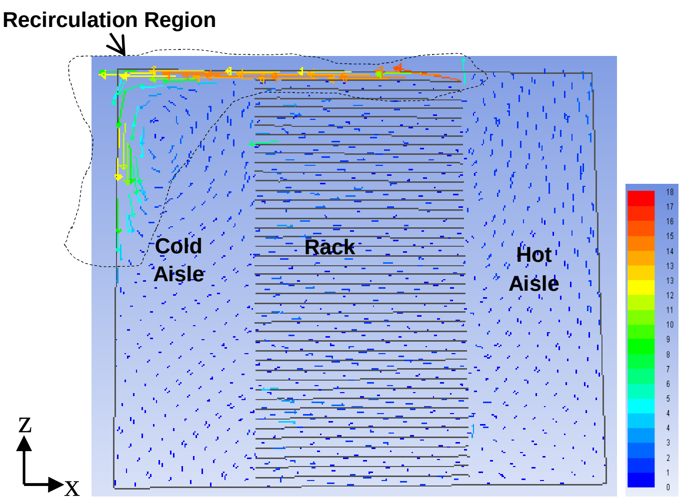

Fig. 11 CFD-predicted flow field at y=12 inches in the test rack, cold aisle, and hot aisle.

Fig.11 suggests that significant air recirculation occurs via the void over the headnode. Such recirculation entrains hot air near the exhaust to the cold aisle and has a significant impact on the forced convective temperature field. Also, it shows the expected turning of cooling air, coming out from the perforated tile, into servers.

During the experimental study, a deviation from normal conditions inside a typical data center exists because the confinement system is used. To estimate the effect of the confinement system on air temperature field, we monitor the air temperatures with and without the confinement system. Fig. 12 shows average air temperatures in cold aisle at different heights, with and without confinement. Fig. 12 reveals the average air temperature in the cold aisle is lower in the presence of the confinement system because the confinement system blocks entrainment of warm room air into the cold aisle. For both arrangements, air temperature is the highest near the top of the test rack. Such a trend can be explained by hot air recirculation. For the arrangement with the confinement system, hot air recirculation faces considerable blockage which explains moderate rise in temperatures near the top in comparison to a sharper rise in the arrangement without the confinement system.

For results reported so far, measurements are conducted with a fully populated test rack, N=42. The server numbers were changed from N=42 to N=32, 22, and 12 by removing servers from the top. Fig. 13 shows the different configurations with varying N.

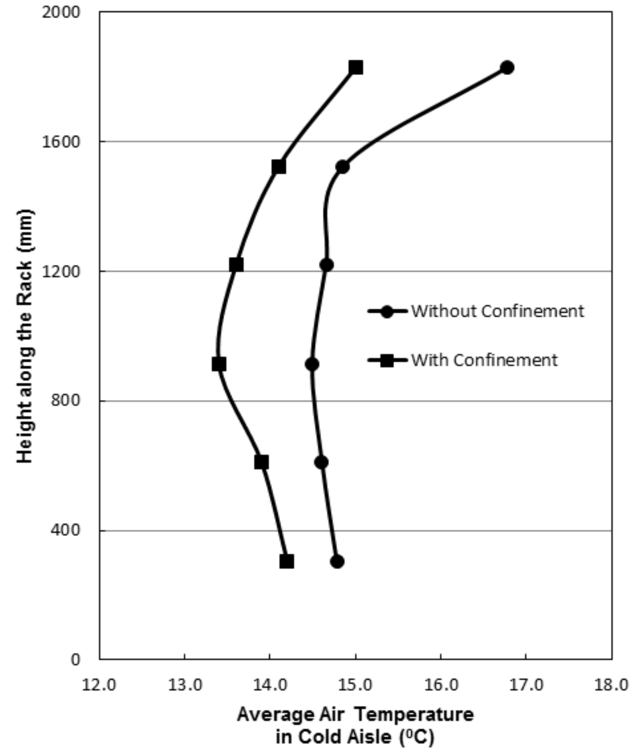

Fig. 12 Variation of average cold aisle temperatures with the height along the rack for the arrangements with and without confinement.

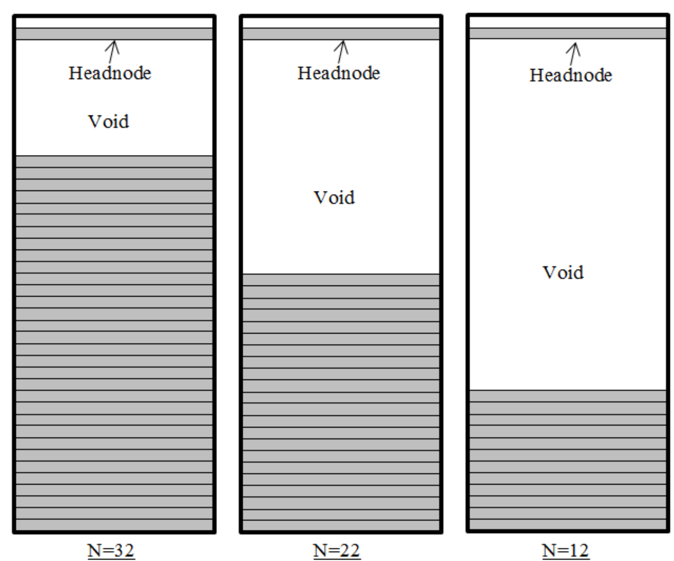

Fig. 13 Different configurations of the test rack with different server populations.

Changes in server population modify airflow path, overall power load, and thermal capacity of the test rack. As a result, air temperatures in the confinement system, shown in Fig. 3, change. Fig. 14 shows temporal variations of exhaust temperatures for different numbers of servers inside the test rack. The exhaust temperatures are the average temperature in the exhaust plane (as shown in Fig. 3). Fig. 14 suggests that the exhaust temperature remains steady. The slight fluctuations appear in Fig. 14 are within measurement uncertainty ($\pm 0.2\ ^{0}C$) of temperature sensors. Fig. 14 indicates as N increases, exhaust temperature also increases. This increment is quite obvious because a higher number of servers would dissipate larger heat rates. The reduction in the number of servers decreases the rack heat load, as shown in Table 3. Also, increasing the void fraction in the rack without any pressure differential induced by server fans significantly impacts the local temperature field.

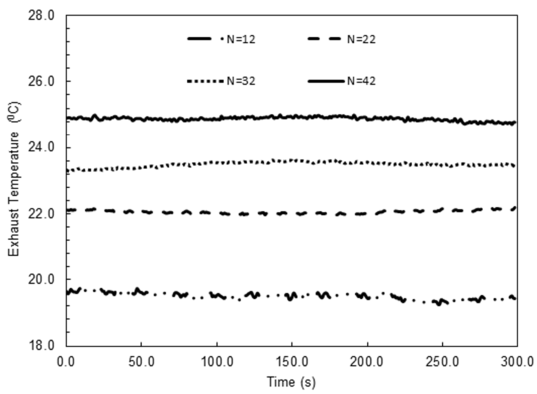

Fig. 14 Temporal variations of average exhaust temperatures for different populations.

Table 3. Variation of rack heat load with different N

N	Heat Load of Test Rack (W)
42	8871
32	6902
22	4885
12	2708

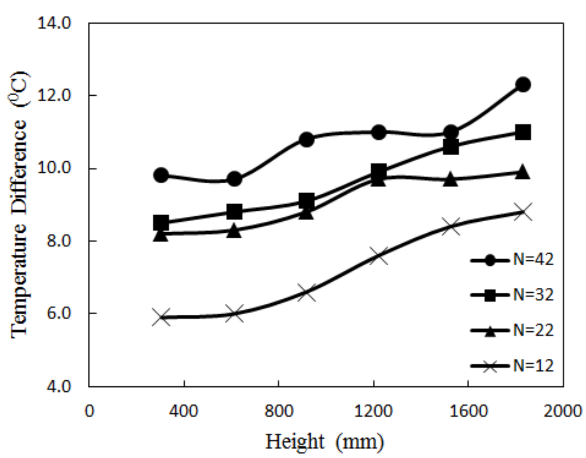

Fig. 15 Effect of server number on the variation of the temperature difference with the height along the test rack.

Fig. 15 shows the effect of server population on temperature difference across the test rack. The temperature difference is defined as difference in average temperatures in hot and cold aisles at a given height. The average temperature in a particular aisle is calculated by taking average of 21 temperature data obtained at that height. In the experimental set up, air temperatures in cold and hot aisles at six different heights: h= 305 mm (1'), 610 mm (2'), 914 mm (3'), 1,219 mm (4'), 1,524 mm (5'), and 1,829 mm (6') are measured. Clearly at all heights, the temperature difference increases with the number of servers inside the test rack.

The results reported so far suggest that server population has a significant impact on rack-level air temperature. It is also found that voids inside a rack significantly influence air flow and temperature fields. The effect of the location of voids is investigated by placing 12 servers at different locations in the test rack. Fig. 16 shows different configurations, as 12 servers are placed at selected locations. Fig. 17 lists the corresponding power consumptions. For these experiments, no confinement system was used.

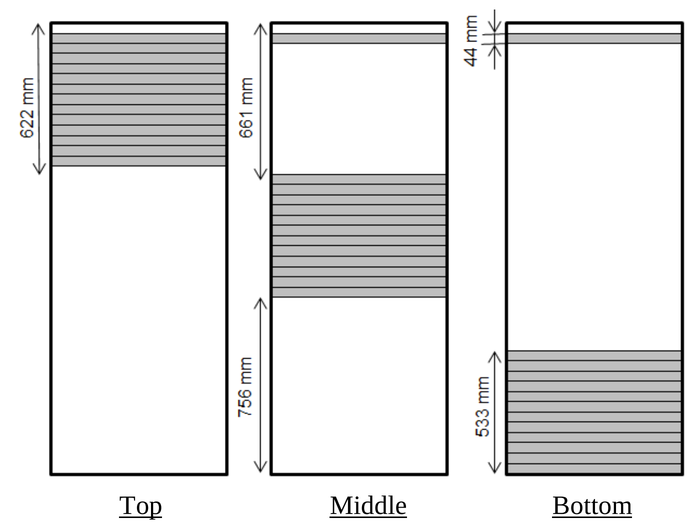

Fig. 16 Various arrangements with 12 servers and a headnode to estimate the effect of location of servers.

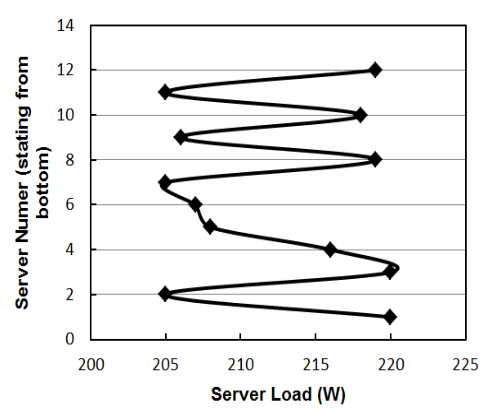

Fig. 17 Server power variation with server number (starting from the bottom of the stack)

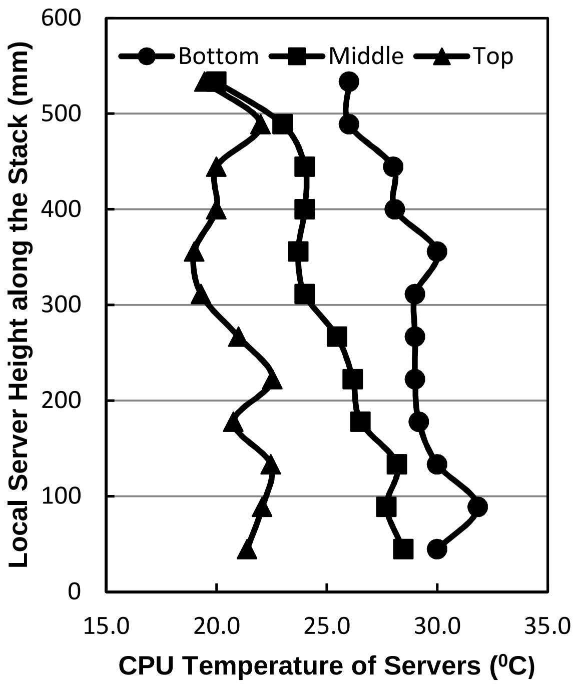

Fig. 18 Variation of CPU temperatures with local server height along the stack. Height=0 indicates a location where the stack begins.

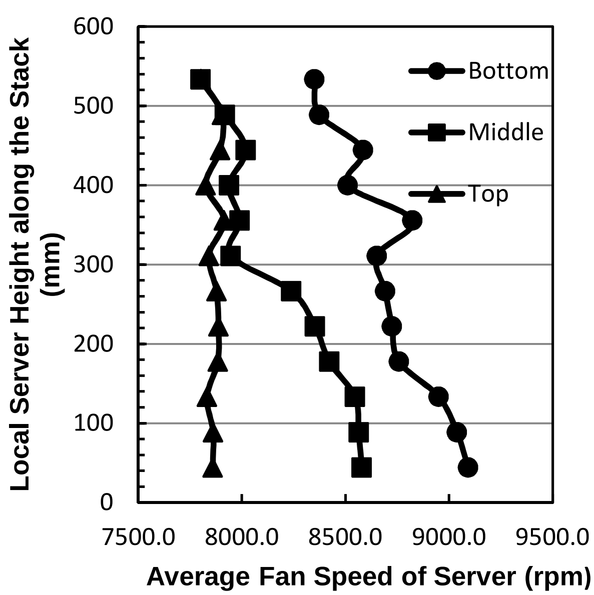

Fig. 19 Variation of average server fan speeds with local server height along the stack. Height=0 indicates a location where the stack begins.

For different arrangements, the CPU temperatures and fan speeds of various servers were monitored. Fig. 18 shows the variation of CPU temperatures with server heights along the stack for various server arrangements namely Top, Middle, and Bottom. Similarly, Fig. 19 shows the variation of average server fan speeds with various server arrangements. Height zero indicates the lowest position in the stack. As Fig. 18 indicates, for all servers CPU temperatures are lowest for Top arrangement and then increase for Middle and Bottom arrangements, Also, Fig. 19 suggests that the average fan speed is highest for Bottom arrangement, followed by Middle, and Top. Evidently, server fans consume least power in Top arrangement keeping the CPUs of the servers at the lowest temperatures. Therefore, keeping a server at the highest possible location in a rack would be more energy-efficient practice in this case.

CONCLUSIONS

This paper investigates the effect of the server population on air temperatures at the rack-level. It is observed that the temperature difference across the rack increases with the height. CFD-based numerical simulations near the test rack suggest significant air recirculation from the hot aisle to the cold aisle. Such a secondary flow has a critical impact on rack-level air temperatures. It is also observed that decrease in the number of servers reduces the temperature in the exhaust plane. A confinement system is implemented and its effect on air temperatures is examined.

In addition, a recommendation is provided on filling out a server rack. It is revealed that the location of a server affects its fan speed and power consumption. Placing a server at the highest location possible improves energy efficiency. In that context, the results reported supports development of effective thermal management strategies for data centers that are undergoing various rack-level upgrades.

Acknowledgments

The authors acknowledge support for this work from IBM Corporation, with Dr. Hendrik Hamann as the Technical Monitor. Acknowledgements are also due to the United States Department of Energy as the source of primary funds. Additional support from the National Science Foundation award CRI 0958514 enabled the acquisition of some of the test equipment utilized.

References

[1] C. E. Bash, C. D. Patel, and R. K. Sharma, "Efficient thermal management of data centers—Immediate and long-term research needs," *HVAC&R Research*, vol. 9, no. 2, pp. 137-152, 2003.

[2] C. L. Belady. "In the data center, power and cooling costs more than the it equipment it supports," http://www.electronics-cooling.com/2007/02/.

[3] K. V. Vishwanath, A. Greenberg, and D. A. Reed, "Modular data centers: how to design them?" 1st ACM workshop on Large-Scale System and Application Performance, pp. 3-10, 2009, New York, NY, USA.

[4] T. Daim, J. Justice, M. Krampits *et al.*, "Data center metrics: An energy efficiency model for information technology managers," Management of Environmental Quality: An International Journal, vol. 20, no. 6, pp. 712-731, 2009.

[5] L. Wang, G. Von Laszewski, J. Dayal *et al.*, "Towards thermal aware workload scheduling in a data center." 10th International Symposium on Passive Systems, Algorithms, and Networks (ISPAN), pp. 116-122, 2009 , Rochester, NY, USA.

[6] www.osisoft.com/software-support/what-is-pi/what_is_pi_.aspx.

[7] G. M. Nelson, "Development of an Experimentally-Validated Compact Model of a Server Rack," Master's Thesis, Geogia Institute of Technology, 2007.