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THERMAL PERFORMANCE EVALUATION OF THREE TYPES OF NOVEL END-OF-AISLE COOLING SYSTEMS

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ABSTRACT

Data centers house a variety of compute, storage, network IT hardware where equipment reliability is of utmost importance. Heat generated by the IT equipment can substantially reduce its service life if *Tjmax*, maximum temperature that the microelectronic device tolerates to guarantee reliable operation, is exceeded. Hence, data center rooms are bound to maintain continuous conditioning of the cooling medium becoming large energy consumers.

The objective of this work is to introduce and evaluate a new end-of-aisle cooling design which consists of three cooling configurations. The key objectives of close-coupled cooling are to enable a controlled cooling of the IT equipment, flexible as well as modular design, and containment of hot air exhaust from the cold air. The thermal performance of the proposed solution is evaluated using CFD modeling. A computational model of a small size data center room has been developed. Larger axial fans are selected and placed at rack-level which constitute the rack-fan wall design. The model consists of 10 electronic racks each dissipating a heat load of 8kw. The room is modeled to be hot aisle containment i.e. the hot air exhaust exiting for each row is contained and directed within a specific volume. Each rack has passive IT with no server fans and the servers are cooled by means of rack fan wall. The cold aisle is separated with hot aisle by means of banks of heat exchangers placed on the either sides of the aisle containment. Based on the placement of rack fans, the design is divided to three sub designs- case1: passive heat exchangers with rack fan walls; case2: active heat exchangers (HXs coupled with fans) with rack fan walls;case 3: active heat exchangers (hxs coupled with fans) with no rack fans. The cooling performance is calculated based on the thermal and flow parameters obtained for all three configurations. The computational data obtained has shown that the case 1 is used only for lower system resistance IT. However, case 2 and Case 3 can handle denser IT systems. Case 3 is the design that can consume lower fan energy as well as handle denser IT systems. The paper also discusses the cooling behavior of each type of design.

NOMENCLATURE

Q - Total heat dissipated (W)
m_w - Mass flow rate of water (kg/sec)
m_a - Mass flow rate of air (kg/sec)
C_{pw}- Specific heat of water (kJ/kgK)
C_{pa} - Specific heat of air (kJ/kgK)
ΔT_w- Temperature difference on water side (K)
ΔT_a- Temperature difference on air side (K)
dP - Static pressure drop across the server(Pa)
$\dot{v}$ - Volumetric flow rate (m3/sec)
$\dot{Q}$ - Volumetric flow rate (CFM)
ΔP - Fan static pressure drop (in w.c.)
μ_f - Fan efficiency (%)

INTRODUCTION

Data centers house a variety of compute, storage, network it hardware where equipment reliability is of utmost importance. The US environmental protection agency (EPA) defines a data center as "primarily electronic equipment used for data processing, data storage, and communications. Collectively, this equipment processes, stores and transmits digital information and specialized power conversion and backup equipment to maintain reliable, high quality power, as well as environmental control equipment to maintain the proper temperature and humidity for the ICT equipment " **[1]**. Some of the world largest internet companies like Google, Facebook, Yahoo, Amazon, Ebay, Microsoft, twitter etc. constitute large scale enterprise-class data centers. Typically, an enterprise-class data center consists of more than 100,000 servers with a facility size more than 5000 sq.ft.

Data centers are managed mostly managed by overly conservative thermal management approaches [2]. The typical thermal solution for data centers is perimeter computer air handlers (CRAH). The traditional cooling method includes supplying air through raised floor plenum that is blown by the CRAH blowers and the chassis fans typically located at the rear end of the IT pull the air across the server collecting the heat and the exhaust air from fans is again collected by the CRAH return. This architecture provides adequate cooling for rack densities below 5kW. However, as the rack densities raise beyond 5kW the removal using CRAH based cooling becomes challenging [3]. In order to address this issue, data centers have adopted the concept of close-coupled cooling i.e. bringing the cooling source closer to the IT equipment [4]. Some of the close-coupled cooling solutions are overhead cooling, rear-door heat exchanger cooling, in-row cooling, bottom located cooling etc. The key objectives of close-coupled cooling is to enable a controlled cooling of the IT equipment, flexible as well as modular design architecture, containment of hot air exhaust from the cold air.

Overhead cooling is a row-based cooling design where the cooling coils are placed on top of the IT rack which removes the heat by circulating the air across the row. Steady and transient behavior of overhead cooling systems has been conducted to compare the cooling effectiveness and energy efficiency of overhead downward flow and overhead upward flow [5]. InRow Cooling is another row-based cooling design where the cooling coils coupled with blowers are placed at specified locations in between the cabinets and in-line with the rows. In Row Coolers are also used to complement the CRAH based cooling which is referred as hybrid cooling; detailed CFD analysis is conducted to show how In Row cooling coupled with CRAH unit can provide uniform cooling energy distribution [6]. Rear door heat exchanger is a water cooled rack based cooling solution where cooling coils are mounted on the rear side of the cabinet [7]. Transient models for cross flow heat exchangers have been developed in order to understand the dynamic response of the heat exchanger during varying operating conditions [8]. Close-coupled cooling solutions are considered to be best fit for data centers high density racks where retrofit upgrades are needed [9]. In terms of reliability, the different types of in-row, above-row and rear door HXs are capable of providing continuous redundant cooling more effectively compared to CRAC based cooling [10]. Various basic heat removal methods have been compared to cool IT and it has been found that rack and row based cooling solutions are more efficient compared to CRAH based cooling; however, these close-coupled solutions highly rely on server or rack fans to operate [11]. While comparing the annual electric cost for different close-coupled solutions, it has been found out that row-based cooling coupled with hot air containment showed lowest costs while room-based traditional CRAC-based design showed highest costs and also the average rack power also played an important role where an increased rack density of around 12 kW that is cooled using rack based cooling solution showed lowest dip in the annual electricity costs [12]. Various factors such as agility, system availability, serviceability, total cost of ownership, system availability and so on should be considered while selecting appropriate close-coupled cooling solution.

MODEL DEFINITION

A computational model of a small size data center room will be developed based on the typical data center room designs. The servers are modeled to be passive components with no in-built fans. Larger axial fans are selected and placed at rack-level which constitute the rack-fan wall design. Active as well as passive heat exchangers are selected based on the commercial designs available. The cooling performance is calculated based on the thermal and flow parameters obtained for all three configurations. There is no raised floor plenum.

The primary focus of the close-coupled cooling methods is to bring cooling closer to the heat source which is the IT rack thereby improving the heat dissipation process along with controlled air flow management in the data center room. The objective of the current study is to analyze the thermal performance of a new kind of close-coupled cooling solution for small data center cooling room using computational software.

MODELING OF ROOM LEVEL DATA CENTER:

A small sized data center room has been modeled using commercial CFD software called 6SigmaRoom [13]. The model consists of 10 electronic racks each dissipating a heat load of 8kW. There are two rows with five racks placed on each row. The room is modeled to be hot aisle containment i.e. the hot air exhaust exiting for each row is contained and directed within a specific volume. . Each rack has passive IT with no server fans and the servers are cooled by means of rack fan wall. The cold aisle is separated with hot aisle by means of banks of heat exchangers placed on the either sides of the aisle containment as shown in figure 1. There is no raised floor plenum. The air flow movement includes the cold air being pulled in by the rack fans across the servers and the exhaust air then takes a turn passing through the heat exchanger units placed on either side of the room containment.

$$Q = m_w C_{pw} \Delta T_w = m_a C_{pa} \Delta T_a \quad (1)$$

Passive heat exchangers with rack fan wall shows that the data center room consists of 10 42U rack (1 rack U =1.72 inch). The dimensions of the rack are 2004x597x1008 mm³. The rack slots are filled with 1U servers. The servers are modeled in passive type by defining the system resistance in terms of viscous and inertial resistance coefficients as shown in figure 3. The server dimensions are assumed to be 711.4x444.5x43.9 mm³. Each server dissipates 190.5 W of heat.

$$dP = Viscous\ res\ Coef * \dot{v} + Inertial\ res\ Coef * \dot{v}^2 \quad (2)$$

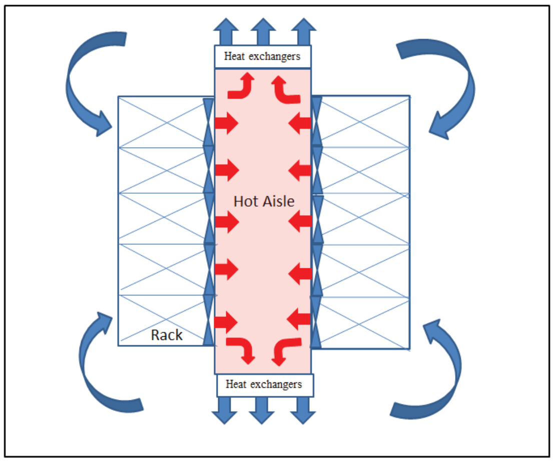


Figure 1: Top view schematic of the close-couple cooling solution

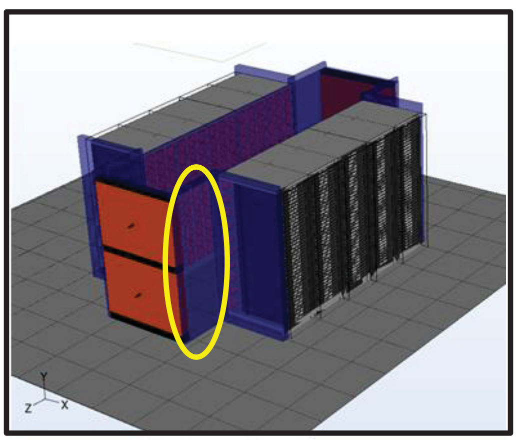


Figure 2: Isometric View of data center CFD model

The hot aisle and cold aisle widths are considered as 1500 mm. The range hot aisle width in industry practice varies between 3' and 6' [14] The hot aisle entrance width (circled space in figure 2) of 838 mm is designed to accommodate the space used to access into the hot aisle typically used for servicing or maintaining of the IT/ cooling equipment.

MODELING OF THE HEAT EXCHANGER UNITS

The heat exchangers are the primary cooling sources that cool the return air exhaust and the cool air supply coming out will be pushed back into the cabinet racks. The heat exchanger design is considered to be a water to air finned tube model. Commercially available heat exchanger designs are selected in order to consider the cost, practicality and feasibility of the cooling design. The heat exchanger system resistance, for each configuration, is obtained using the coil selection software [15]. The system resistance curve of server measured for case 1 is shown in figure 4. The coil selection software has been leveraged to estimate the performance of the selected heat exchanger design for the given air side and water side boundary conditions. The heat exchanger effectiveness has been calculated using the effectiveness-NTU method. The air side and water side volumetric flow rates are specified based on equation 1 where the water and air side temperature difference is assumed based on typical industry practice.

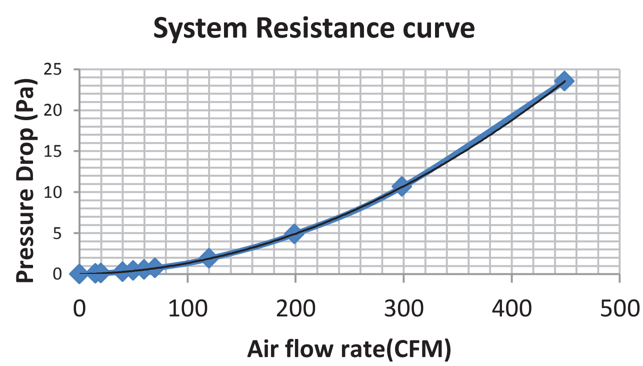


Figure 3: System Resistance curve of the 1U Server

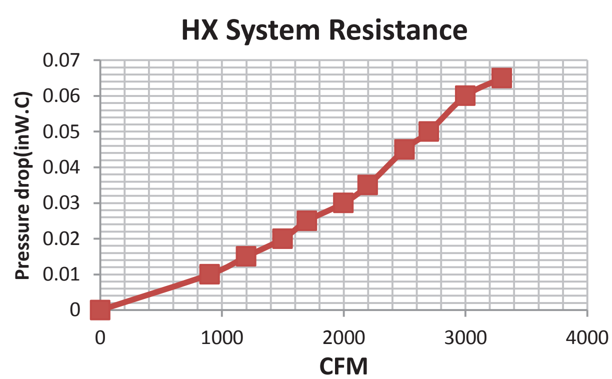


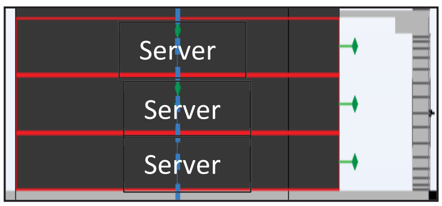
Figure 4: Water-to-air finned tube heat exchanger system resistance

The heat exchanger dimensions are 33x47x5 in³ based on the available commercial designs. The geometry of the heat exchanger design is also influenced by the size of the hot aisle (width and height). There are total four heat exchanger banks each assumed to handle a sensible heat load of 20kW. The entering water and leaving water temperatures are assumed to be 17°C and 22°C respectively according to typical industry practices used to make sure the water temperature is about dew point temperature of ambient air. The effectiveness values calculated based on these boundary conditions is 0.65. The water flow rate based on energy equation to dissipate 20 kW is 15.2 GPM. The entering air temperature is 38 °C. These are the values that are defined for modeling the performance of the heat exchanger unit.

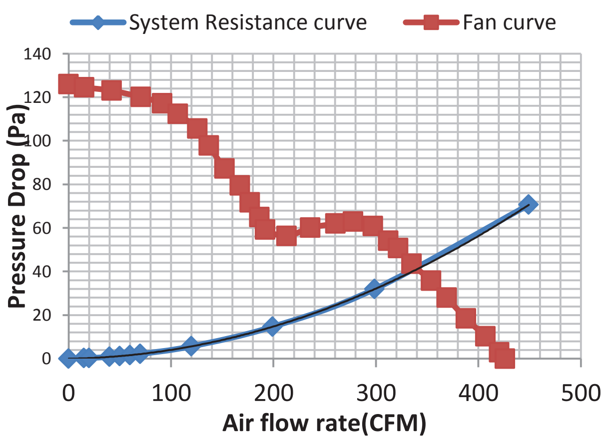
RACK FAN WALL SELECTION:

Case1:

The total air flow requirement based on the air side energy balance equation (1) for 8kW total load and a ΔT of 10°C is 1375 CFM per rack. Each server needs 32.7 CFM of air flow rate. A total of 42 pieces in terms of 14 rows of 3 120 mm parallel fans have been grouped.



(a)



(b)

Figure 5: a) Server stack with one row of 3 120 mm fans b) System resistance and derived fan curve the server stack and fan system

Based on the geometry of the rack space and the server system resistance, the servers and fan system are arranged as shown in Figure 5 (a). The Fan curve for three parallel fan curves is calculated based on fan curve information of single 120 mm fan curve [16] as shown in Figure 5 (b). Even though the derived fan curve and system resistance curve shows the fans are operating in the low resistance points, the following system has been considered to account for additional resistance that will be offered by the heat exchanger banks and the air flow movement.

RESULTS

The model is studied for steady state conditions. Standard K-ε turbulence model has been chosen. The total grid generated is 23 million cells. The conformal meshing technique known as grid control method is used in the software to mesh certain geometry such as IT servers and cabinets which develops finer mesh. This helps in achieving higher accuracy during the iterative solver procedure opted by the CFD.

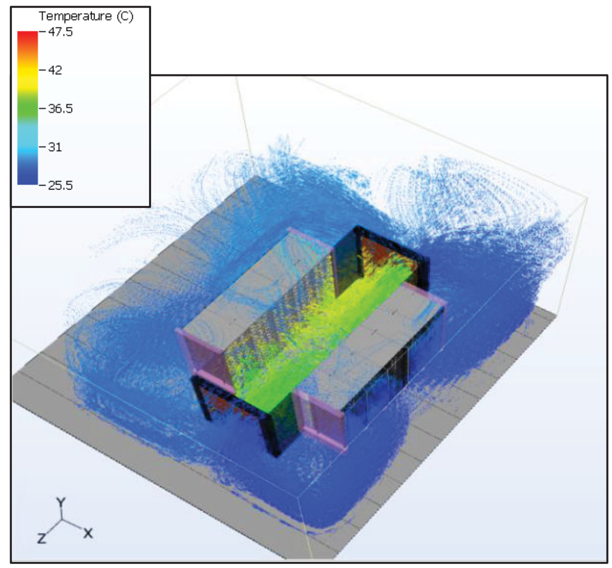


Figure 6: Temperature Streamlines of the air flow movement

The temperature streamlines shown in figure 6 demonstrate the air flow movement in the room. The average temperature different across each cabinet is around 10 (±1) °C as shown in figure 7.

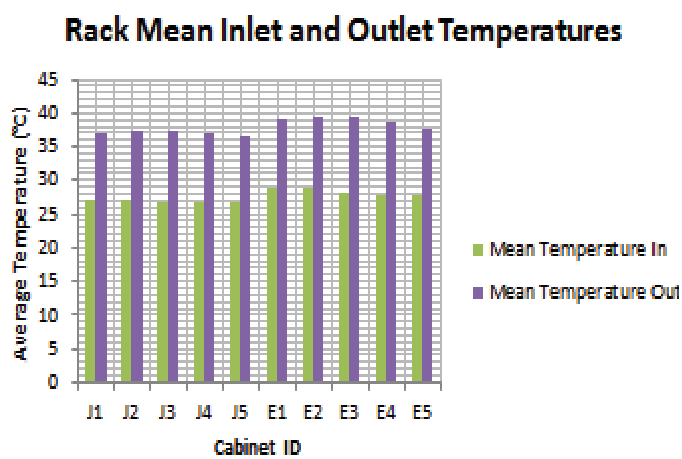


Figure 7: Rack Average Inlet/Outlet temperatures

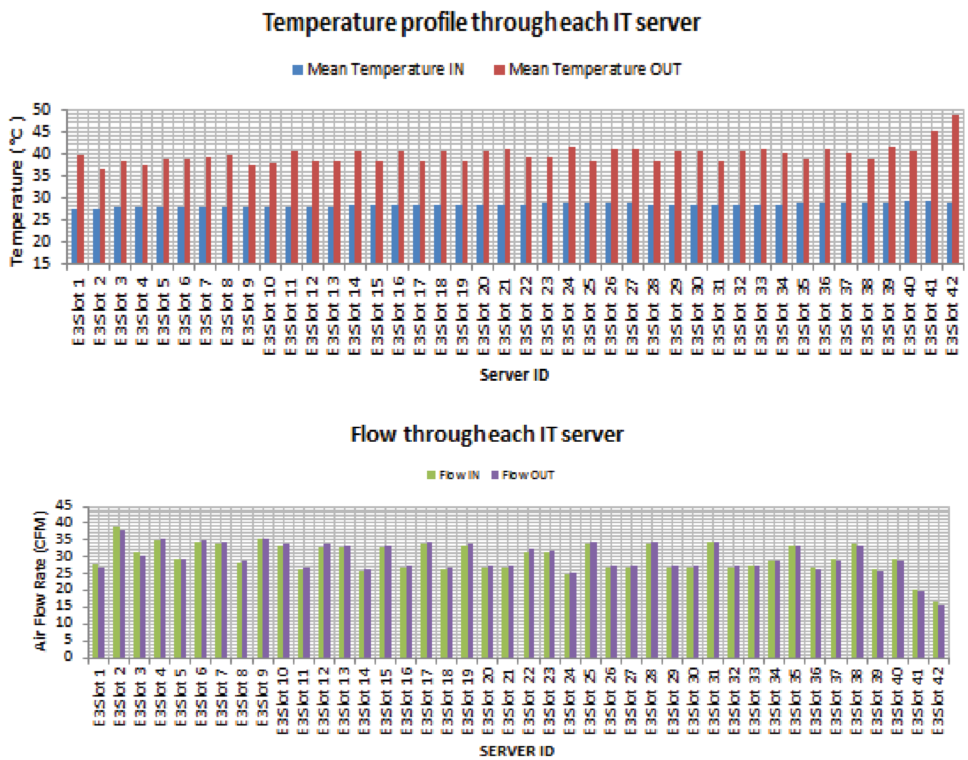


Figure 8: Server parameters

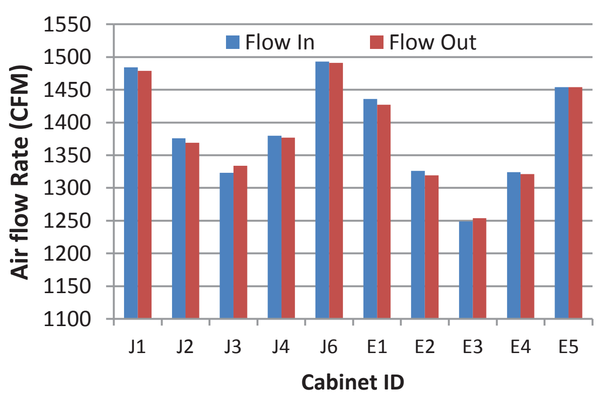


Figure 9: Individual Cabinet Flow Rates

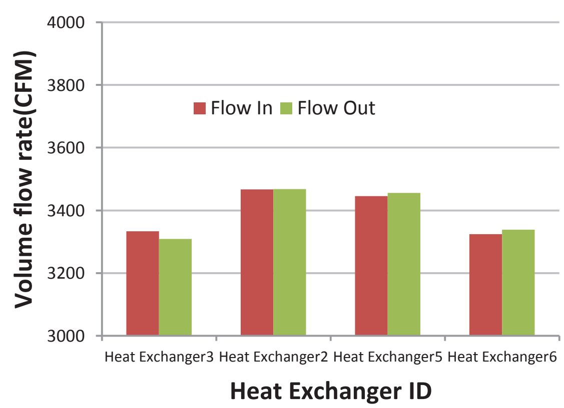


Figure 10: Individual Heat exchanger Flow Rates

The rows are labeled as Row J and Row E with 5 cabinets in each row. J1, J5, E1 and E5 represent the outermost cabinets on each row. From figure 9, it can be seen that the middle cabinets do not receive sufficient air flow. The reason can be attributed to the air flow path in the specific design and this data helps understand the limiting factor for the number of cabinets in each row. Currently, the model works well with 5 cabinets in each row, because the middle cabinet receives around 150 CFM lesser air flow rate than the peripheral cabinets. The middle cabinets did not exhibit relatively higher temperatures due to lesser flow rates.

The air flow rates through individual heat exchangers are reported in figure 10. The heat exchanger 3 and heat exchanger 6 are located at the bottom section on either side of the aisle. It can be seen that these heat exchangers receive lower air flow rates. Because of the tendency of the hot air to rise within the aisle, the units located at the top section receive higher flow rates. This factor can influence in sizing the heat exchangers to different capacities. The difference between the top and bottom heat exchanger flow rate is 135 CFM. The hot air entering the top section heat exchangers is around 1°C higher compared to the bottom section heat exchangers.

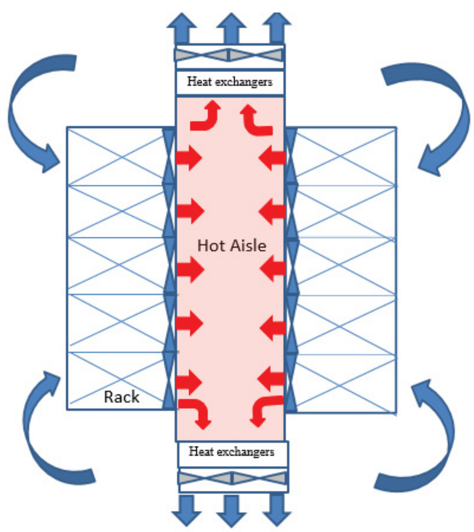
The flow going in and out of cabinets and heat exchanger is the same which ensures that there is no recirculation within the system.

Because the middle cabinet showed lower inlet temperatures, the cabinet E3 is considered to examine the temperatures and flow rates of individual servers as shown in figure 8. The flow rate varies across the height of the server. However, there is no recirculation. The individual server flow rate did not follow a specific pattern in the flow distribution. However, it is noticed that the top few servers (E39-E42) receive lower flow rates and hence exhibit higher exhaust temperatures. The total fan power in the room is calculated to be 1038 W for 24% fan efficiency based on equation 3.

$$Fan\ Power = \Delta P * \dot{Q} * \frac{0.1175}{\mu_f} \quad (3)$$

It is identified that there are three main variables that mainly influence the cooling design. It is the system resistance of the IT, pressure drop that is overcome by the fans (i.e. the only air movers in the room), temperature difference across the rack.

Case 2:



This case is the same as Case 1 with group of fans arranged in series with the heat exchangers as shown in figure 11. Placing fans in series with heat exchangers enables the design to handle high resistance IT placed inside the room. The fans across the heat exchanger (second-stage fans with 200mm diameter) are selected such that they are geometrically arranged along the same size as HX surface area and in parallel to each other. The server resistance considered for the IT is based on resistance for HPSE1102 [17].

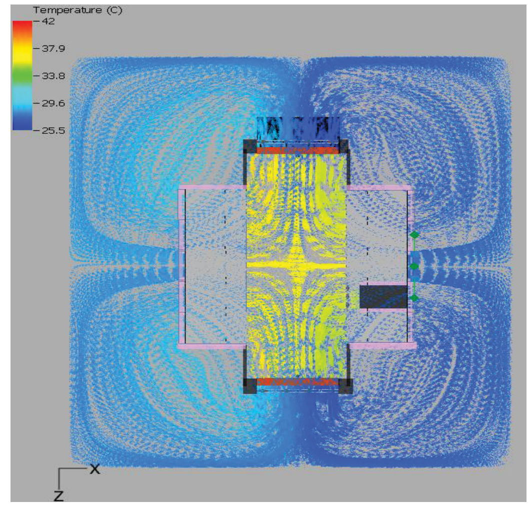


Figure 12: Air flow Streamlines in the room colored by temperature

Multi-stage fans in series should handle the same amount of volumetric capacity as the first stage fans (rack level fans) [18]. Hence, the second stage fans (Φ200 mm) are selected from the commercially available axial fan catalogue based on the total CFM requirement from each heat exchanger-fan system. Figure 12 shows the air flow movement in the room

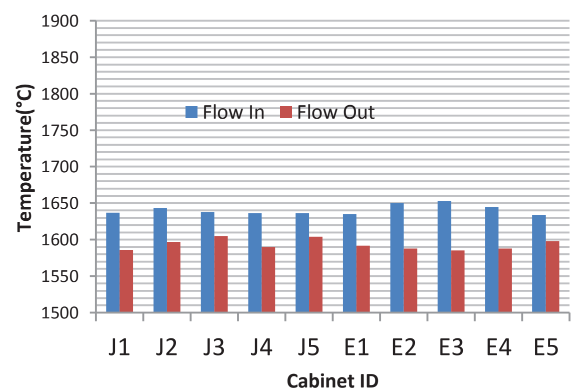


Figure 13: Rack Inlet and Outlet Flow rates

The total rack fan power in the room is calculated as 896.9 W. The total heat exchanger series fan power is 1227.83 W. Rack fans operate at lower pressure drop for wider hot aisle widths. The hot aisle size determines the pressure drop overcome by rack fans. Compared to case 1, the rack fans operate at lower static pressure. However, overall fan power is higher. The main advantage of case 2 over case 1 is that it can handle high resistance IT. The average temperature difference observed across the rack is 8.8°C. The average rack flow rate is 1640.7 CFM.

Since the cooling design can more handle volumetric capacity the rack heat load is also scalable for increased loads.

The fan operating speeds between first stage and second stage fans is important. The fan control algorithm of the rack level fans typically controls the fan speeds based on the maximum operating temperature of a group of servers (in our case 3 servers shown in figure 5). The coordination for fan speeds between multiple stages should be such that the when rack level fans are running idle the second stages should also run at lower speeds.

Case 3:

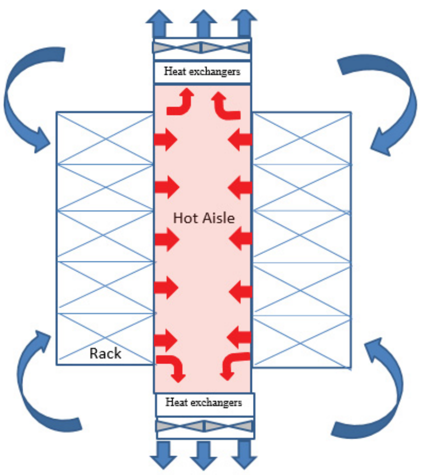


Figure 14: Cooling design schematic for case 3

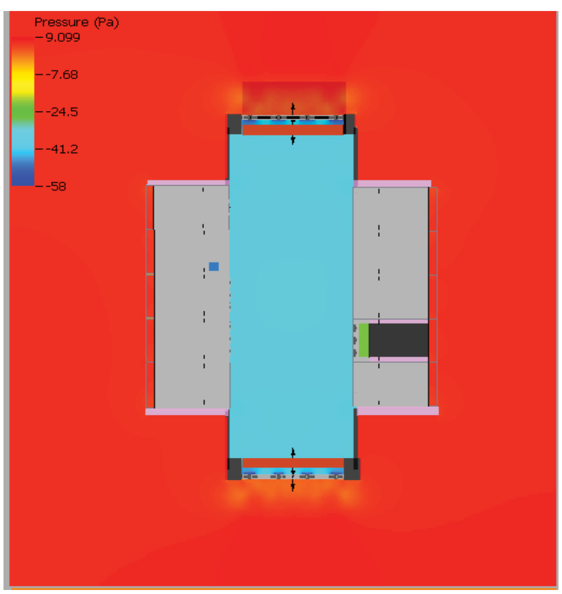


Figure 15: Pressure plot for case 3-Top view of the room with cut section running in the through the center of the rack height

In this case, there are no rack level fans; the fans (Φ200 mm) are located only in series with the heat exchanger as shown in figure 14. This cooling design also handles higher IT system resistances in the room. The heat exchanger fans would operate at higher static pressure compared to the previous case because they are the only air movers in the room. The average temperature difference observed across the rack is 9.3°C. The average rack flow rate is 1572.9 CFM.

If the fan movers are only located across heat exchanger (as in case 3) the room is more pressurized compared to case 2 as seen from figures 15 and 16. This kind of over pressurized systems might lead to an increased risk of leakage between hot aisle and cold aisle. The total heat exchanger fan power consumption in the room is 1010.55 W. The overall fan power consumption is reduced compared to case 2 and comparable to case 1.

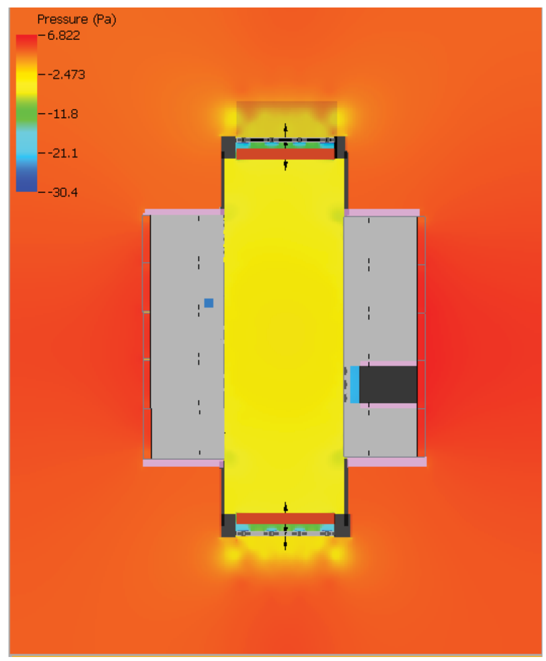


Figure 16: Pressure plots from case 2- Top view of the room with cut section running in the through the center of the rack height

CONCLUSION

The study evaluates a new type of end-of-aisle cooling system for a small data center room using CFD. The methodology for modeling passive IT, selection of rack fan wall and heat exchanger designs including the design boundary conditions based on industry practice has been presented. The system showed acceptable temperature and air flow profile based on the given boundary conditions. The volume resistances considered for case1 design are for low resistance IT systems. However, it has been seen that such a system if had air movers arranged in series with the IT rack fans or in series with heat exchangers can handle higher system resistance. Larger fan systems that can overcome higher resistance systems should be used to for denser IT components. The heat exchanger units are placed such that it would eliminate the risk of having cooling coils at the top of the electronic racks. The top section of servers showed higher exhaust temperatures but overheating of the IT is not observed. The fan power consumption for all three cases have been quantified. Case 2 has been an over pressurized system compared to other two cases. This can be addressed by increasing the hot aisle volume. It has also been noticed that the distance between heat exchangers and the axial fans in series for case 2 and case 3 is of importance. If the fan systems are closer to heat exchangers, then they would pull the air fasters across heat exchangers coils reducing the thermal performance of the heat exchanger. Further analysis is needed to determine the best case cooling design. The analysis would include response of the system during failure conditions, system air flow behavior when it is not isolated i.e. if neighboring IT rows are present, using blowers instead of axial fans design. Overall, the current study proposes a new cooling design that addresses the challenges for placing cooling coils at the top of the IT or at the bottom of the IT. The CFD analysis comparing the thermal performance of all three cases gave multiple design considerations to be made by opting to certain case of design.

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