

Thermal Performance Evaluation of a New Close-Coupled

Cooling Solution including Cooling Failure Analysis

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Abstract: 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 Information Technology (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 Computational Fluid Dynamics (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. 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 from hot aisle by

means of banks of heat exchangers (HXs) placed on 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 during cooling failure conditions. During failure conditions, case 2 showed better cooling redundancy than case 1 and case 3.

Nomenclature:

: Total heat dissipation (W)

m_w : Mass flow rate on water-side (kg/sec)

C_p : Specific heat of water (KJ/Kg-K)

ΔT_w : Temperature difference on the water-side ($^{\circ}$ C)

m_a : Mass flow rate on air-side (kg/sec)

C_p : Specific heat of air (KJ/Kg-K)

ΔT_a : Temperature difference on the air-side ($^{\circ}$ C)

d : Static pressure drop (Pa)

$\dot{v}$: Volumetric flow rate (m^3 /sec)

ΔP : Fan Pressure drop (Pa)

Q : Fan flow rate (CFM)

μ : Overall fan efficiency (%)

1 Introduction: Close-Coupled Cooling Solutions for IT PODS

Data centers are 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]. 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 [5].

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 [6]. 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 [7]. Rear door heat exchanger is a water cooled rack based cooling solution where cooling coils are mounted on the rear side of the cabinet [8]. 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 [9]. Bottom-located cooling unit is compared with perimeter based cooling system in terms of air flow energy using CFD [10]. Computational study analysis has been conducted in developing innovative server rack design with bottom located cooling unit [11]. Close-coupled cooling solutions are considered to be best fit for data centers high density racks where retrofit upgrades are needed [12]. 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 [13]. 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 [14]. 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 [15]. 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 [15]. The current

solution is aimed to serve as an alternative to existing designs as it avoids placing the coils at the bottom of the racks and also avoid the leakage risks while placing coils at the top of the racks.

2 Modeling of Room level data center

2.1 Model Definition

A computational model of a small size data center room is developed based on the typical data center room designs. The servers are modeled to be passive components with no built-in 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. 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.

A small sized data center room has been modeled using commercial CFD software called 6SigmaRoom [16]. The software is essentially designed to conduct numerical simulation of heat transfer and fluid flow which are obtained by solving a set of second order, partial differential equations. The variables that are solved for are the velocity vectors in x, y, z direction, pressure and temperature. The test chamber space is the volume where the data center model is built. The domain/test chamber space is discretized to generate finite volume cells/ grid cells. The conservation equations for mass, momentum and energy are solved in this domain by means of

numerical integration. The current 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 an enclosed 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 from the hot aisle by means of banks of heat exchangers placed on either side of the aisle containment as shown in figure 1. 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. Air density is assumed constant in the model.

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

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 equation 2. Equation 2 is static pressure vs. volumetric flow curve i.e. unique to the server and is determined by a wind tunnel test. The data points derived from the test represent the system resistance curve which is quadratic in nature. The coefficients of the equation are obtained from the system impedance curve of the server. The server dimensions are assumed to be 711.4x444.5x43.9 mm³. Each server dissipates 190.5 W of heat.

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

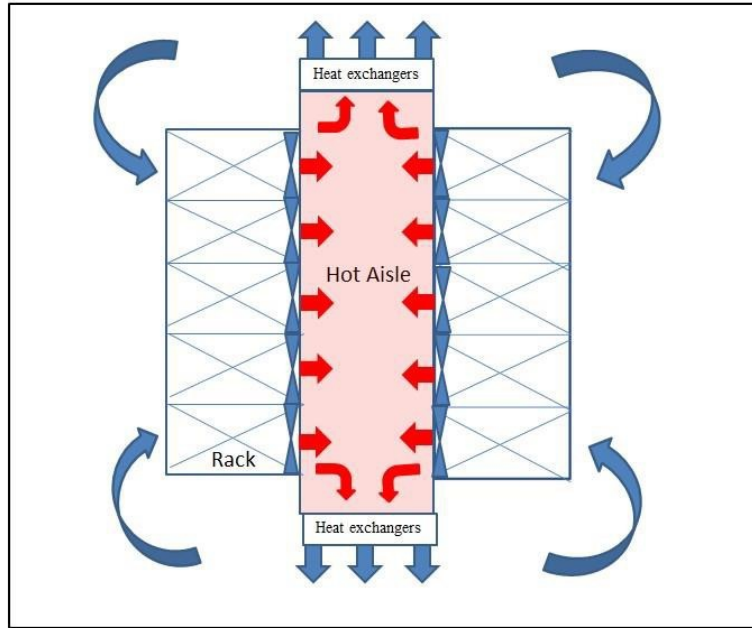


Figure 1: Top View of the Close-coupled Cooling Solution

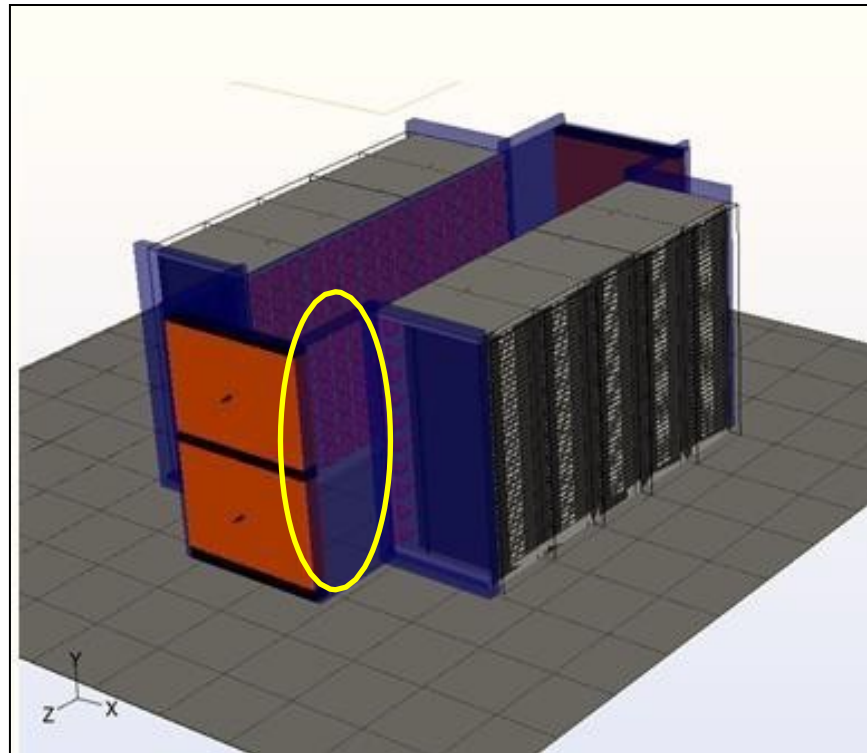


Figure 2: Isometric View of data center CFD model

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' [17] 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.

3. 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 [18]. The system resistance curve of server measured for case 1 is shown in figure 4. The curve is obtained by placing the server in an air flow bench and measuring multiple static pressure difference across the server for multiple known volumetric flow rate points. 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.

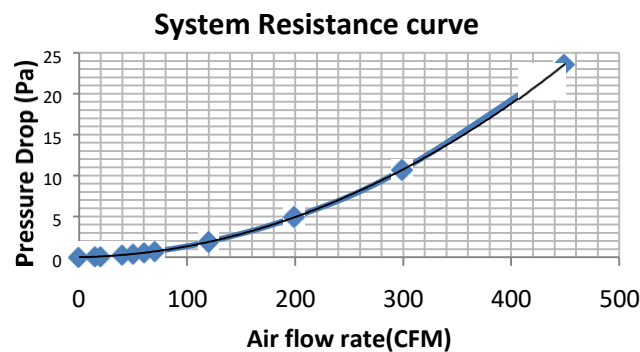


Figure 3: System Resistance Curve of 1U server

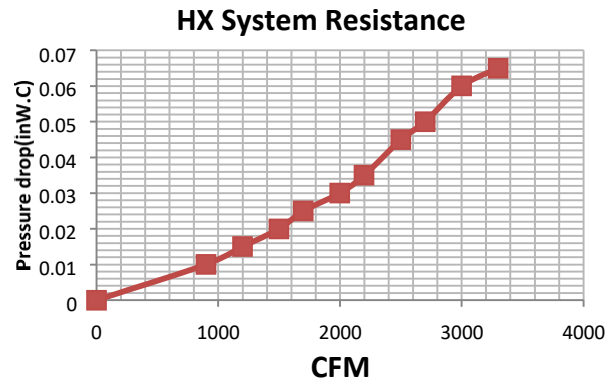
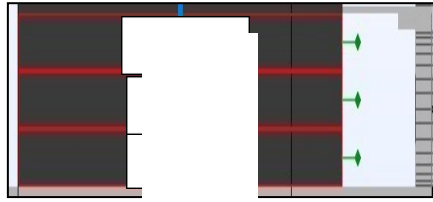


Figure 4: Water-to-Air Finned Tube Heat Exchanger System Resistance

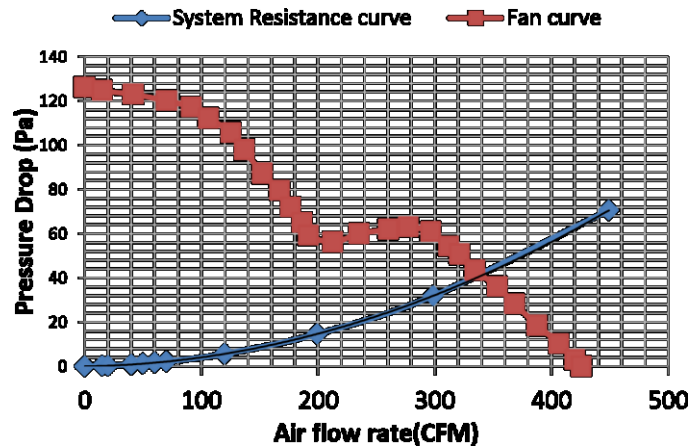
The heat exchanger dimensions are 33x47x5 in³ based on the available commercial designs. The geometry of the heat exchanger design is also influenced by the size of the hot aisle (width and height). There are total four heat exchanger banks each assumed to handle a sensible heat load of 20kW. The entrance and exit 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 above dew point temperature of ambient air. The effectiveness values calculated based on these boundary conditions is 0.65. The water flow rate based on water side 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.

4. Case 1 Rack Fan Wall Selection

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 three 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 [19] 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.

4.1 Case 1 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.

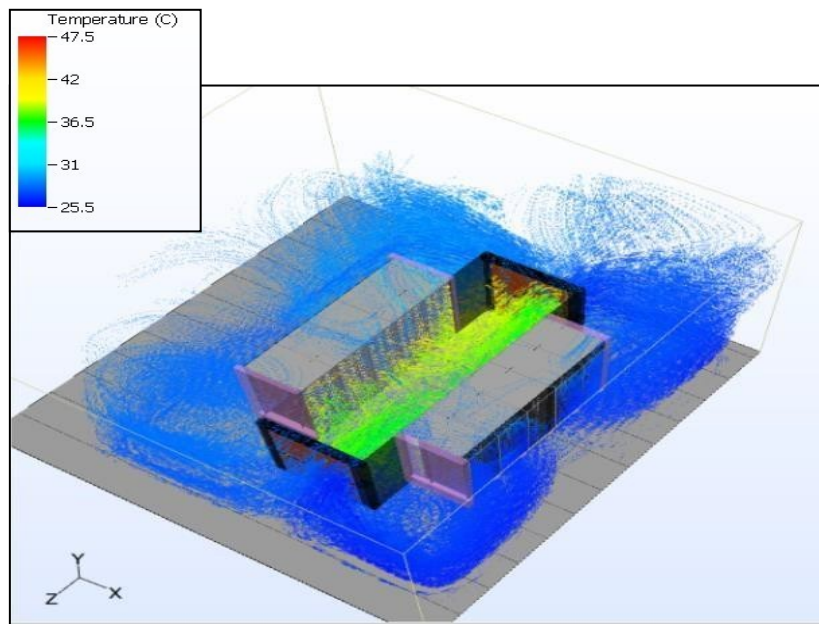


Figure 6: Temperature Streamlines of the Air Flow Movement

The temperature streamlines shown in figure 6 demonstrate temperature difference between hot aisle and cold aisle.

The average temperature different across each cabinet is around 10 (± 1) °C as shown in figure 7.

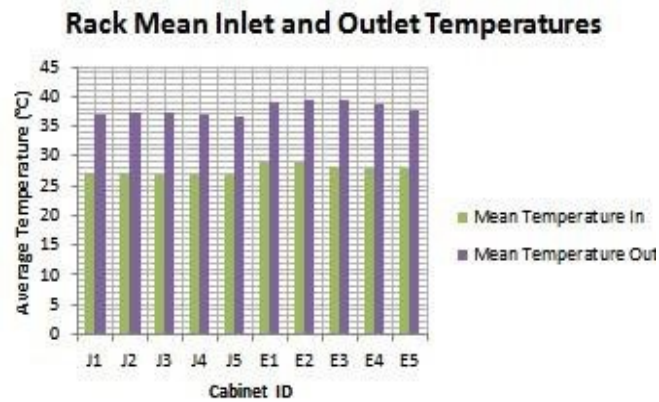


Figure 7: Rack Average Inlet/Outlet Temperature

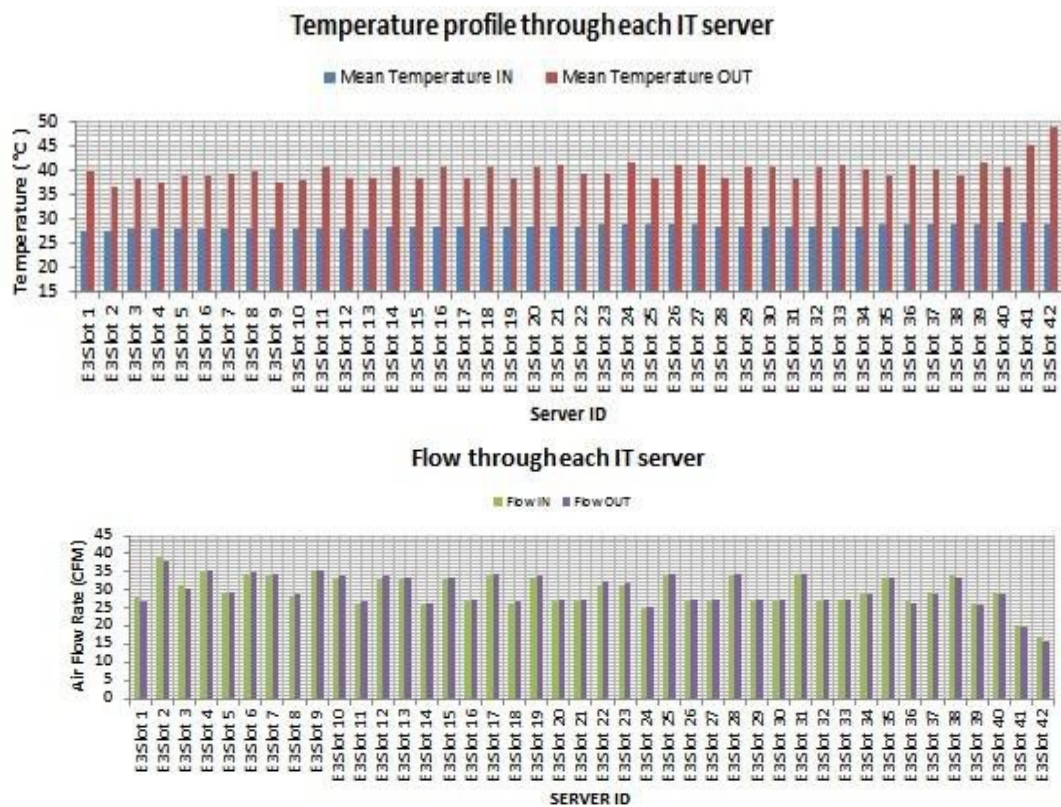


Figure 8: Rack Average Inlet/Outlet Temperature

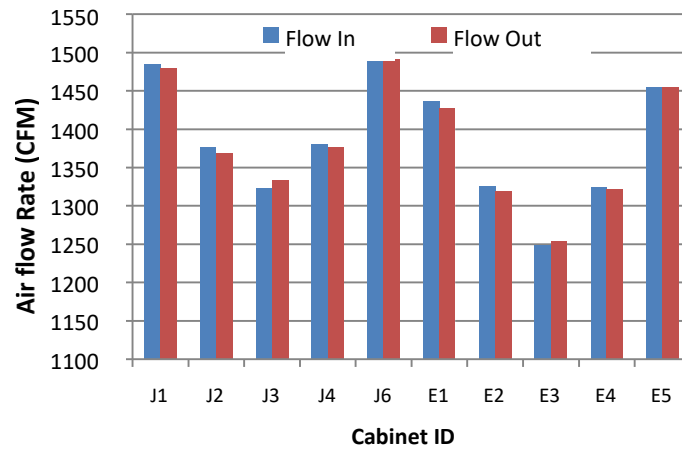


Figure 9: Individual Cabinet Flow Rates

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 less air flow rate than the peripheral cabinets. The middle cabinets did not exhibit relatively higher temperatures due to lesser flow rates because the difference in the rack

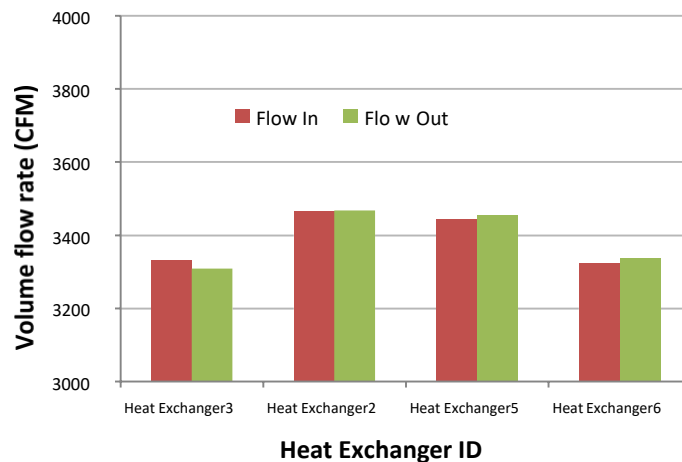


Figure 10: Individual Heat Exchanger Flow Rates

air flow rate would not have significant impact on the individual server CFM. The air flow rates through individual heat exchangers are reported in figure 10. The heat exchangers 3 and 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. The negligible difference between flow in and flow out in the plot is because of the marginal error in absolute values reported by the software.

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.

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

It is identified that there are three main variable 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.

5. Case 2

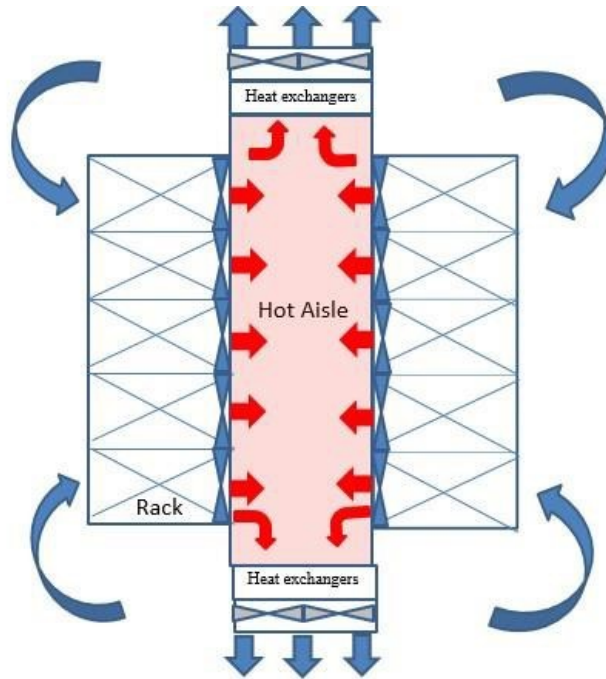


Figure 11: Schematic Layout of Case 2 with Active heat Exchangers and Rack level fans

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 172 mm 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 [20].

Multi-stage fans in series should handle the same amount of volumetric capacity as the first stage fans (rack level fans) [21]. Hence, the second stage fans ($\Phi 172$ 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. The temperature on the slightly higher on left hand side because of the rack flow rates are lower in this row.

5.1 Case 2 Results:

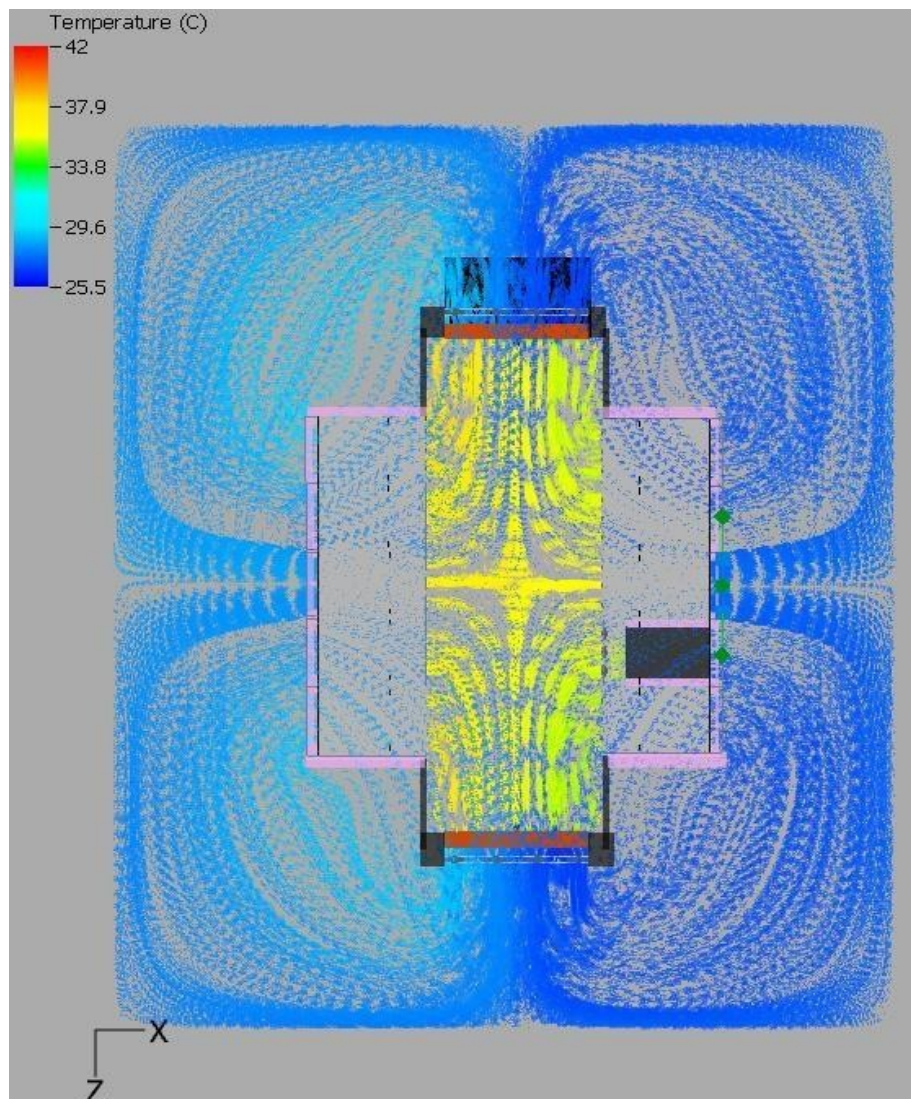


Figure 12: Air Flow Streamlines colored by temperature

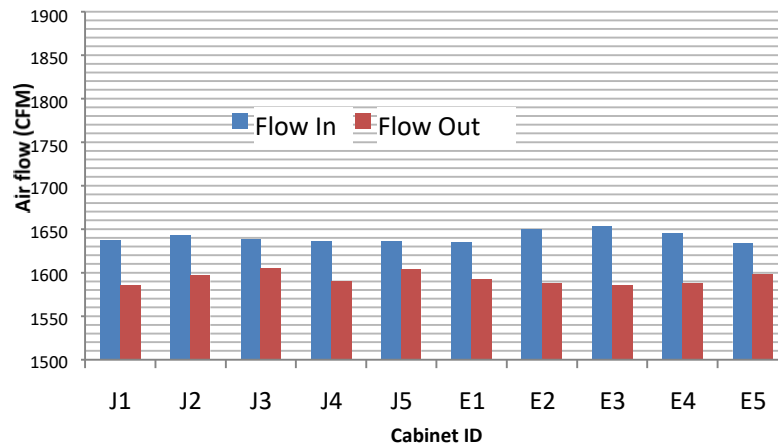


Figure 13: Rack Inlet and Outlet Flow rates

The total 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 (23 times higher static pressure). 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 handle higher 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.

6. Case 3

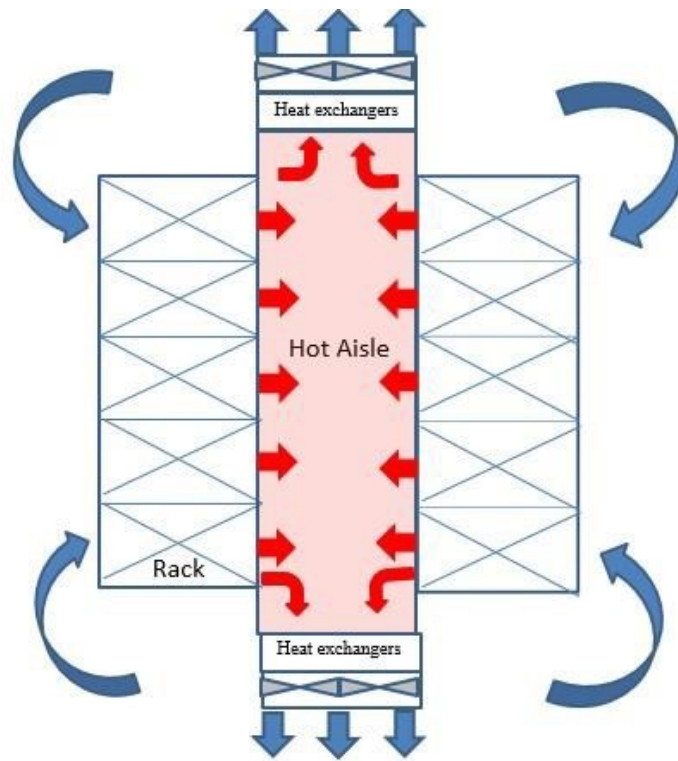


Figure 14: Cooling Design Schematic for Case 3

In this case, there are no rack level fans; the fans (Φ 172 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.

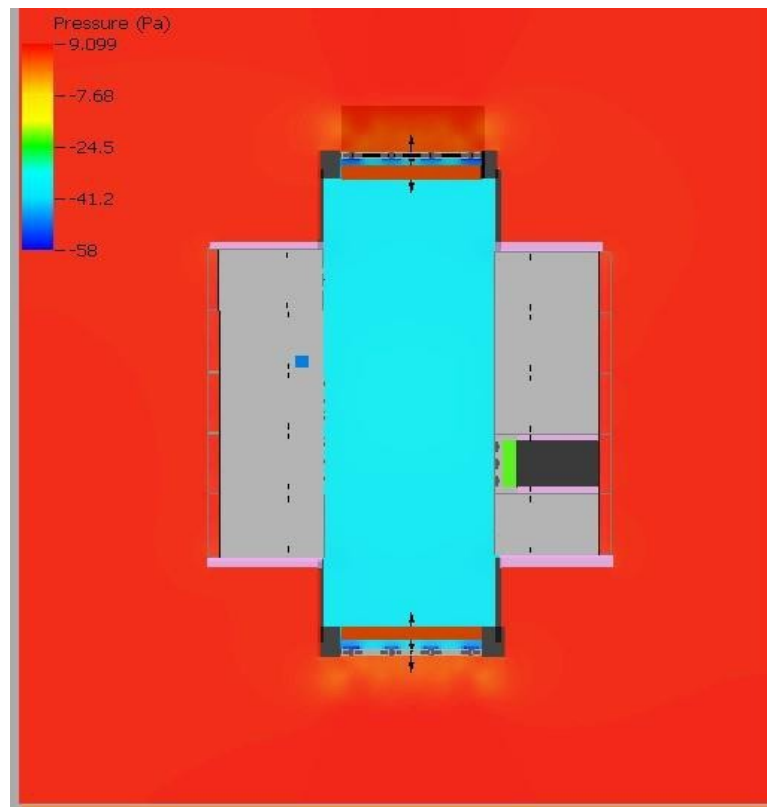


Figure 15: Pressure Plot for Case 3-Top view of the room with cut section running through the center of the rack height

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. These 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.

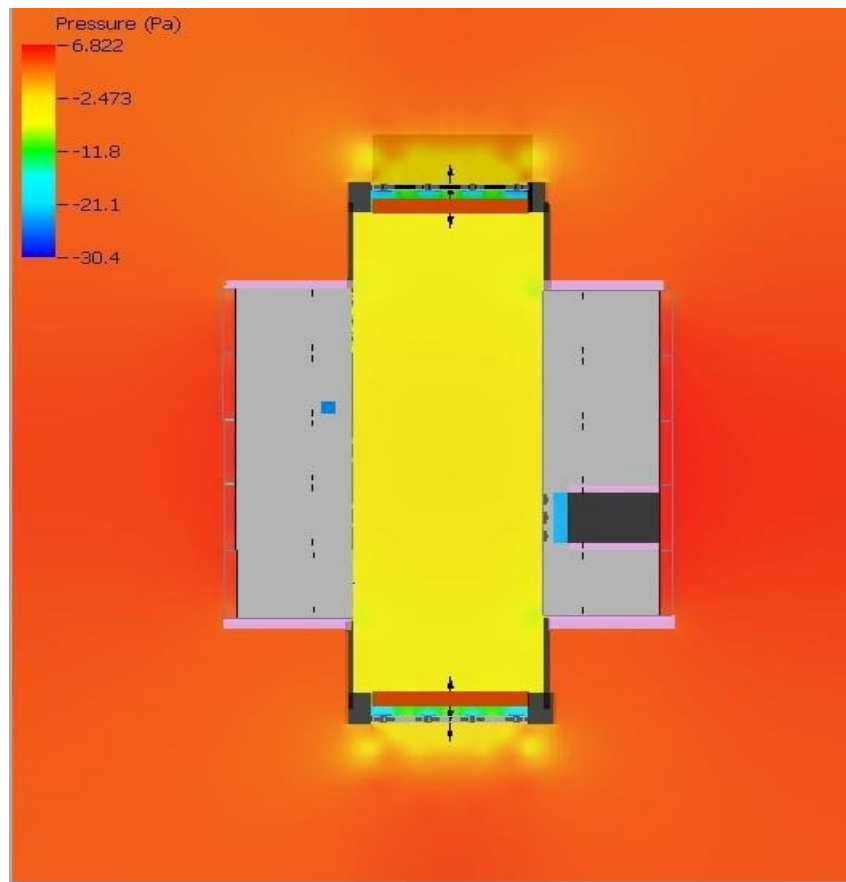


Figure 16: Pressure Plots from case 2- Top view of the room with cut section running through the center of the rack

The cold aisle pressure values in the model are in agreement to the typical ranges seen in a data center room. Case 2 HX fans carry 3.2 times less static pressure due to the presence of rack fans as shown in figure 17. Case 2 average cold aisle pressure: 0.013" H₂O. Case 3 average cold aisle pressure: 0.031" H₂O. Case 3 holds higher risk of leakages between hot aisle and cold aisle. Case 1 average cold aisle pressure: 0.0013" H₂O. In Case 1, hot aisle is at positive pressure unlike case 2 and case 3. Low velocity region has formed in the middle of the hot aisle as shown in figure 18. This did not affect the rack outflow. Having higher number of racks will be affected by the high pressure region creating a back flow in to the racks.

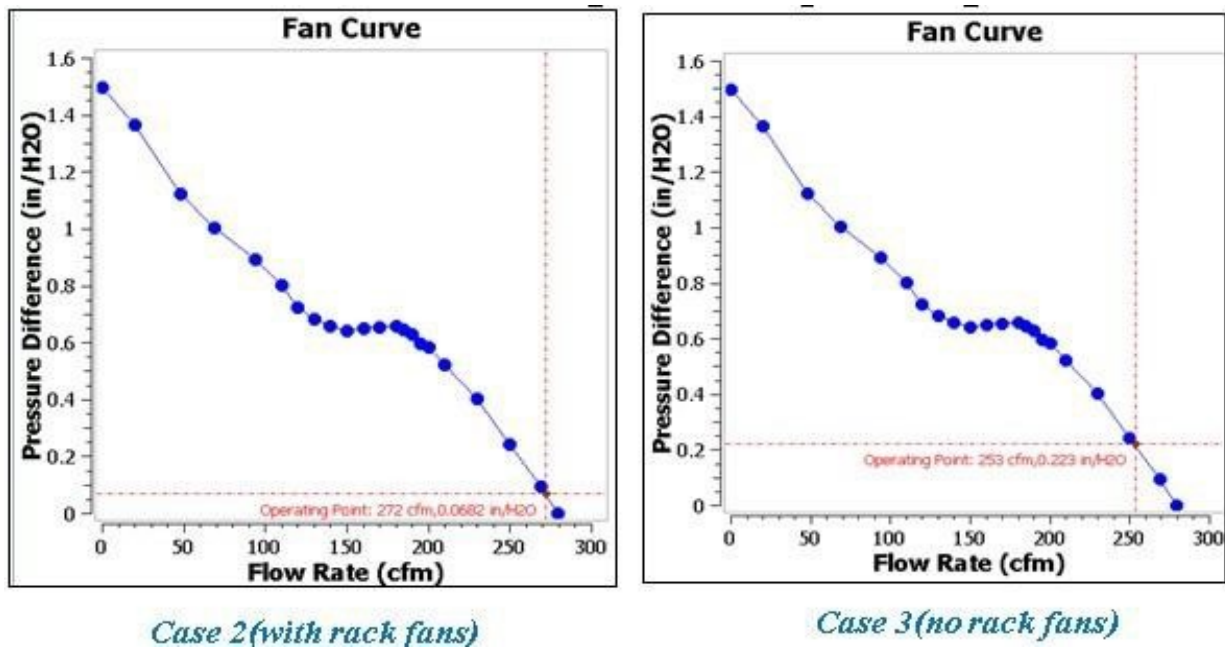


Figure 17: Active Heat exchanger fan operating point

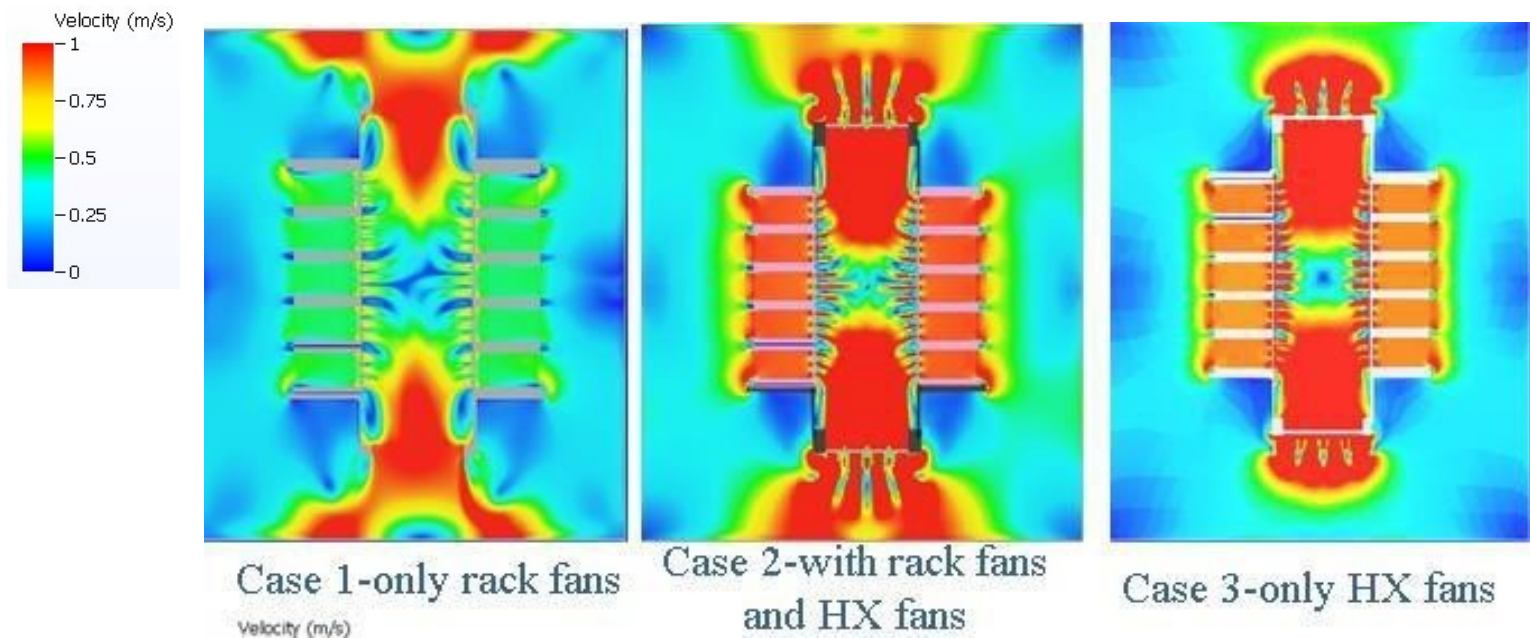


Figure 18: Velocity plots with cut-plane across the middle of the rack height

7. Cooling Failure Scenarios

For all three different cases, different failure cases have been simulated by turning off heat exchangers and rack fans and heat exchanger fans in various scenarios as shown in the figure 19 below. In typical data center set up, the IT equipment is connected to Uninterrupted Power Supplies (UPS) that turn on during a loss of utility power to the facility. However, the CRAC/CRAH systems, chilled water pumps, chillers are often not connected to the UPS [22]. The objective is to quantify the increase in inlet temperatures and reduction in the supply rack air flow in the room due to any imposed failure conditions. The scenarios will be discussed individually as well as in comparison against default designs or other failure scenarios for all three cases. The following cases considered to be in steady state conditions.

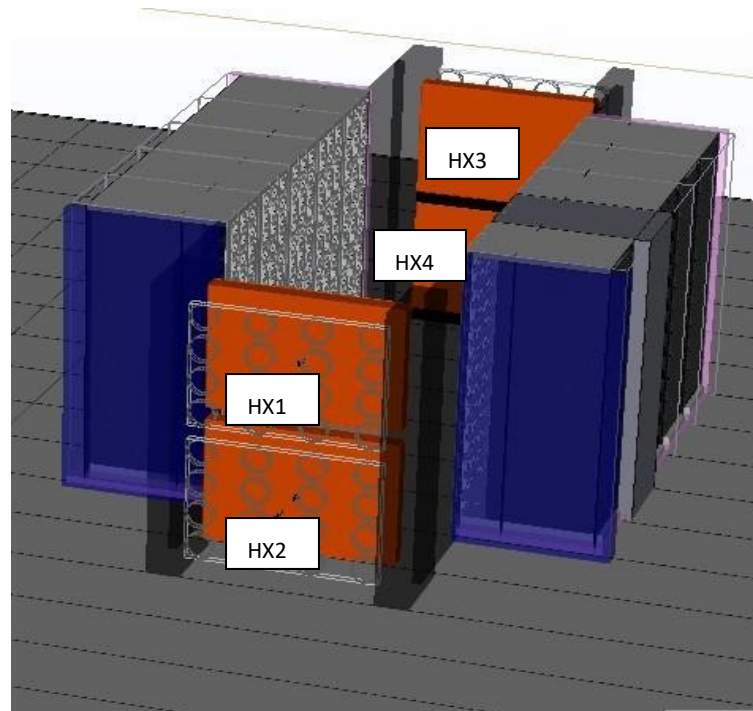


Figure 19: Isometric View of the room

Case	Status of Rack Fans	Status of Heat Exchanger fans	Status of Heat Exchangers
Case 3A	OFF	HX1-OFF	HX1-OFF
Case 3B	OFF	HX1-OFF;HX2:OFF	HX1-OFF;HX2-OFF
Case 3C	OFF	HX1-OFF;HX2:OFF	ALL-ON
Case 3D	OFF	All-ON	HX1-OFF;HX3-OFF
Case 2A	ON	HX1-OFF;HX2:OFF	HX1-OFF;HX2-OFF
Case 2B	ON	All-ON	HX1-OFF;HX3-OFF
Case 1A	ON	-	HX1-OFF;HX3-OFF
Case 1B	2 middle Rack fans OFF	-	ALL-ON

Table 1: Different failure scenario simulations

Case 3A: This case has all rack fans turned off, with one top heat exchanger turned off and the corresponding heat exchanger fans have been turned off as well.

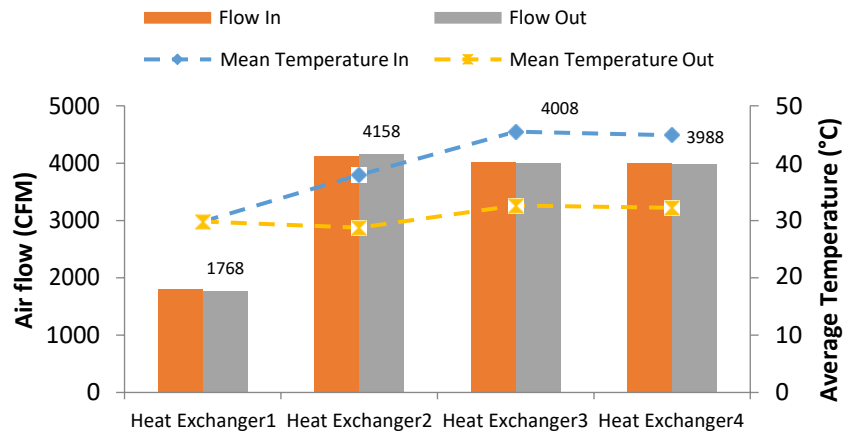


Figure 20: Case 3A Heat Exchanger Flow rate and Temperature parameters

As anticipated, the supply air and return air temperatures of the failed heat exchanger are equal as shown in figure 20. The air flow rate going into the failed heat exchanger is merely due to the pressure difference between the hot aisle and the cold air but not due to induced air flow. The average reduction in the air flow rate is 2286.7 CFM for failed HX1. HX2 has shown reduced temperature difference between supply and return air i.e. 9.2 °C compared to the temperature difference across HX3 and HX4 being i.e. 12.6 °C. The supply air temperature across HX3 and HX4 has been increased by 4.9 °C. The reason for this is being the corresponding heat exchanger fans draw the same amount of air flow not uniformly from all the rack but disproportionately. If we consider the air side energy balance equation, these heat exchanger fans (of HX3 and HX4), by drawing the same amount of air flow and increased heat load, [i.e. imposed due to the failed heat exchanger (HX1)], the corresponding average air side temperature difference has been increased, along with rise in both supply and return air temperatures.

Case 3B: This case has both top heat exchangers (HX1 and HX3) disabled and the corresponding fan systems have been disabled with overall rack fans in the room disabled.

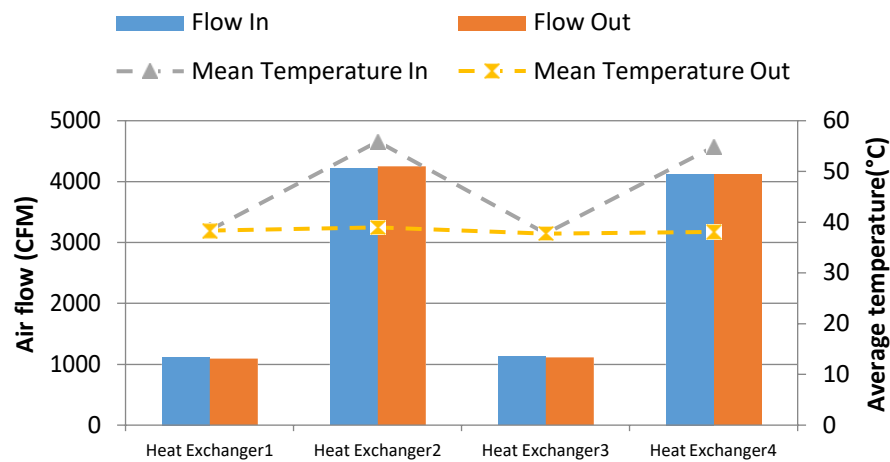


Figure 21: Case 3B Heat Exchanger Air Flow and Temperature Parameters

The heat exchangers that have been disabled are drawing 3101.5 CFM lower than the other two heat exchangers. The mean inlet temperatures of HX3 and HX4 are higher by 17.3 °C as shown in figure 21. As explained in previous section (for case 3A), the reason for this is that the HX2 and HX4 that draw the same amount of air flow; with increased heat load on them, the air side temperature difference increased along with the return and supply air temperatures. This rise in inlet temperatures can be mitigated by employing fan control over individual heat exchanger fan systems. The heat exchanger fan pulse width modulations (PWM) (second-stage fan systems) must be modulated accordingly based on the return air temperatures. This type of fan control will help the fans draw in more air flow thereby reducing the overall supply air temperatures in the room in case of failure.

In terms of the rack air flow rates and the supply temperatures, there is uniformity. The average rack air flow rate is 624.06 CFM. The average temperature difference across the racks has increased to 22.9 °C

by ~2.5 times compared to the default case (Case 3). The corresponding server air flow rates have decreased. However, there is no major non-uniformity even within the rack in terms of air flow rates.

Additional case has been run as a part of case 3B, where heat exchangers on one side of the aisle have been disabled i.e. HX1 and HX2, it has been noticed that the top heat exchanger (HX1) has higher return air temperature compared to HX2 by 6 °C. On the active side (HX3 and HX4), the top heat exchanger, HX3, has highest return air temperature which is 28% more than the HX4 return air temperature as shown in figure 3-21. The reason for this can be attributed to the fact that hotter air rises and quickly being drawn by the active fans of top located HX3. The temperature distribution across the racks fluctuated by about ± 1.5 °C for the inlet and outlet temperatures. The average rack flow rate in the room is 627.5 CFM.

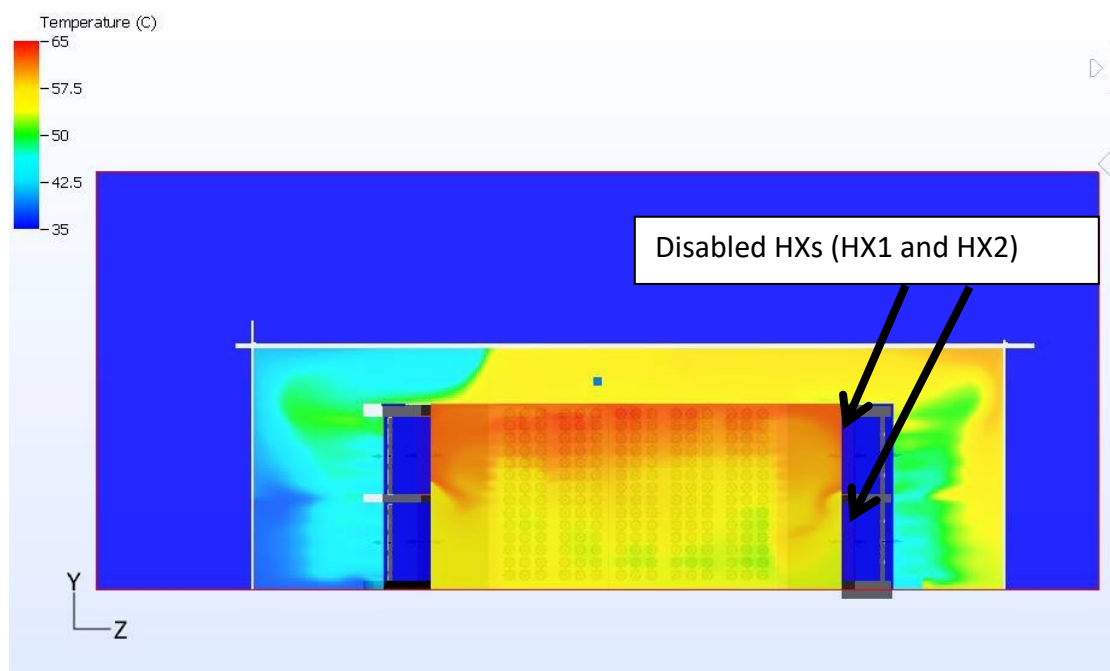


Figure 22: Temperature Plot when HXs on one side were disabled and hot air accumulation in the hot aisle. [the cut plane is running through the center of hot aisle in Y-Z plane]

Case 3C: This case represents the case where two heat exchangers on one side (HX1 and HX2) are disabled but all the heat exchanger fan systems are turned on.

This scenario can be represented as worst case failure scenario, since the fans are all turned on. The expectation is that hot exhaust is quickly recirculated across the room as the corresponding heat exchanger has failed. HX1 and HX2 showed inlet and exhaust temperatures (being the same) to be 58.8 °C and 53.9 °C which is about ~2.3 times more than the default case as shown in figure 24. The return air temperatures of HX3 and HX4 are 36.8 °C and 39.9 °C respectively. The heat exchangers located at the top once again received higher return air temperatures. There is an average air side temperature difference of 18 °C across HX3 and HX4. Since all heat exchanger fans are working, the average HX fan air flow is 3840.8 CFM.

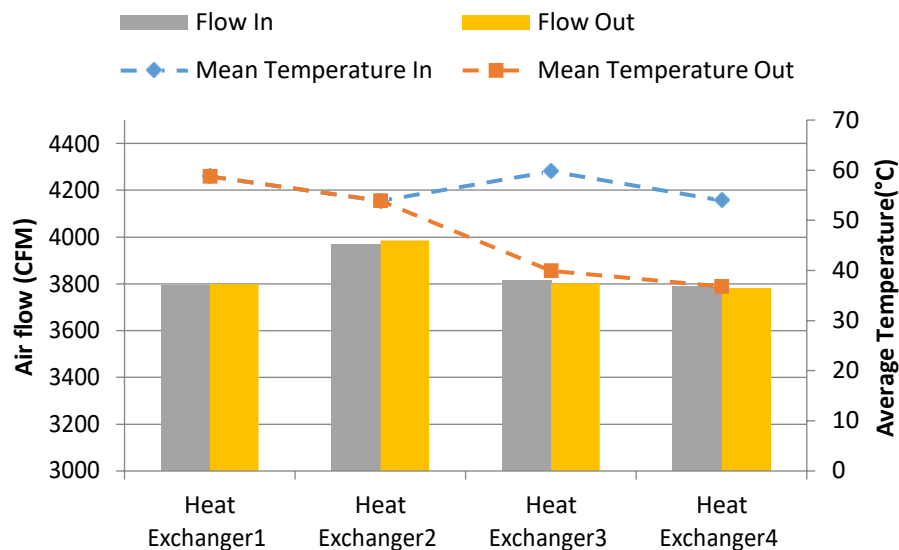


Figure 23: Heat Exchanger Air Flow and Temperature Parameters

The cabinet had a mean inlet temperature of 47.5 °C and temperature difference of 9.23 °C across the rack. The inlet temperatures in this case are the highest compared to the default design (Case 3) and

other failure scenarios. Hence, the heat exchanger fan PWM control has to be adjusted according to the return air temperature at the inlet as mentioned in the previous case as well.

Overall, for case 3 design scenario, it has been noticed that the system is redundant with one of the heat exchanger failed, there is only a 4.3 °C increased in the inlet temperatures. If the maximum return air temperature limit is assumed to be 55 °C, then the case 3B with heat exchangers failing on either side is reaching the maximum limit. However, the relatively worse case is heat exchangers along with their fan systems failing only on one side. The worst case scenario is heat exchangers failing on one side and fans are running at a higher PWM (above 50% in our case).

Case 2A: This case corresponds to the simultaneous heat exchangers and attached fan systems failures with all rack fans turned on.

HX1 and HX2 air flow rate is about 3.3 times lower compared to the other two active heat exchangers. The air temperature across HX1 and HX2 is 52.1 °C which is very close to the assumed higher limit i.e. 55 °C as shown in figure 25. The air temperature across HX3 and HX4 is about 12.6 °C higher than the default case. The return air temperatures have reached their maximum limits. So even with the presence of rack fans to move the air in the heat exchanger failure on one side imposes a huge risk of rising inlet temperatures in the room. The average rack air flow is 1215.1 CFM. Maximum rack inlet temperature is 43.2 °C. Maximum hot air temperature across the rack is 55 °C.

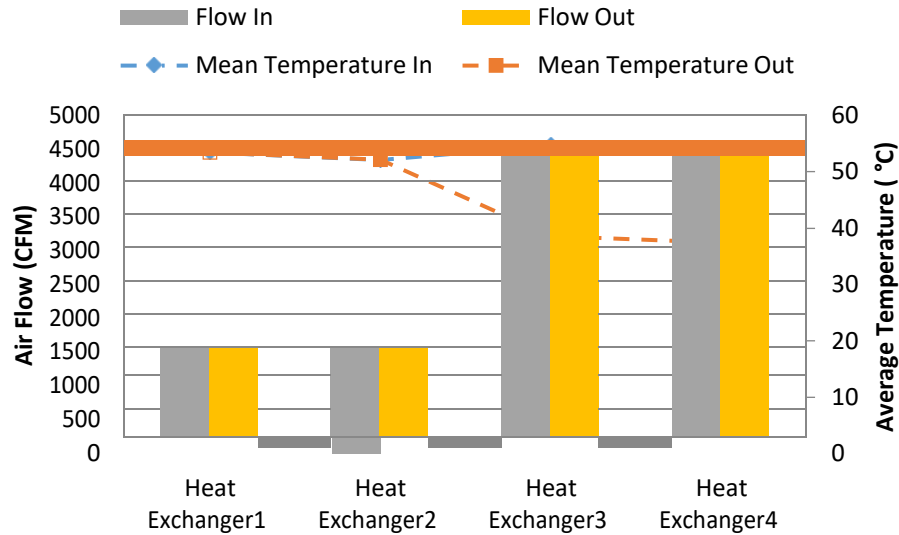


Figure 24: Heat Exchanger flow and Temperature Parameters

Case 2B: This case describes the scenario where the top two heat exchangers are turned off and but all the heat exchanger fan systems are running. The rack fans are all turned on.

The average heat exchanger air flow is 4188.75 CFM. The air temperature across the failed heat exchangers is 1.5 times higher than the active heat exchangers. The HX1 and HX2 show 62.6 °C and 49.1 °C return/supply air temperatures respectively shown in figure 25. The heat exchanger failure without fan failure is the worst case scenario to happen. Compared to 3C, this case shows that a return air temperature across the heat exchangers as well as higher inlet air temperatures to the rack (46.7 °C) is not very different. The presence of rack fans (running at the same PWM as the default design-Case 2) neither worsen nor improve the hot air exhaust temperatures in the room. The average rack air flow is 1606.6 CFM. This is also not very different from the case 3C (1565.7 CFM).

Pressure plots in Case 3C and Case 2B are vary significantly. Case 2B has hot aisle more pressurized compared to cold aisle (figure 28) and Case 3C has the room at high pressure compared to the hot aisle (figure 29).

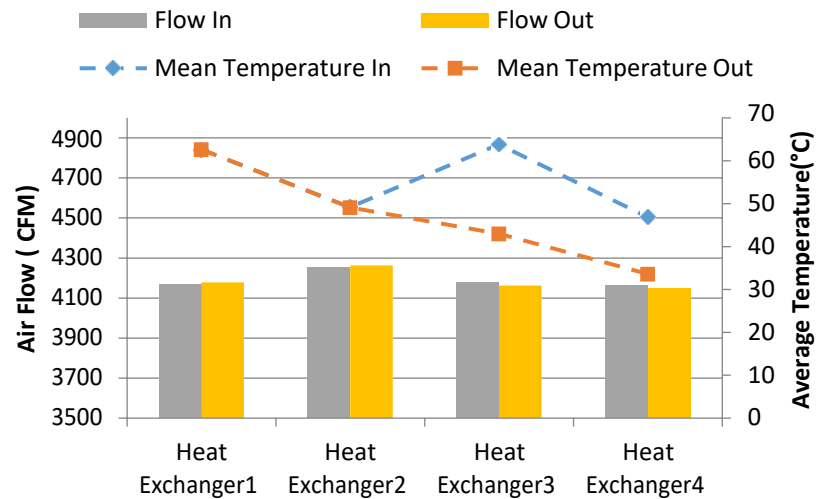


Figure 25: Heat Exchanger Flow and Temperature Parameters

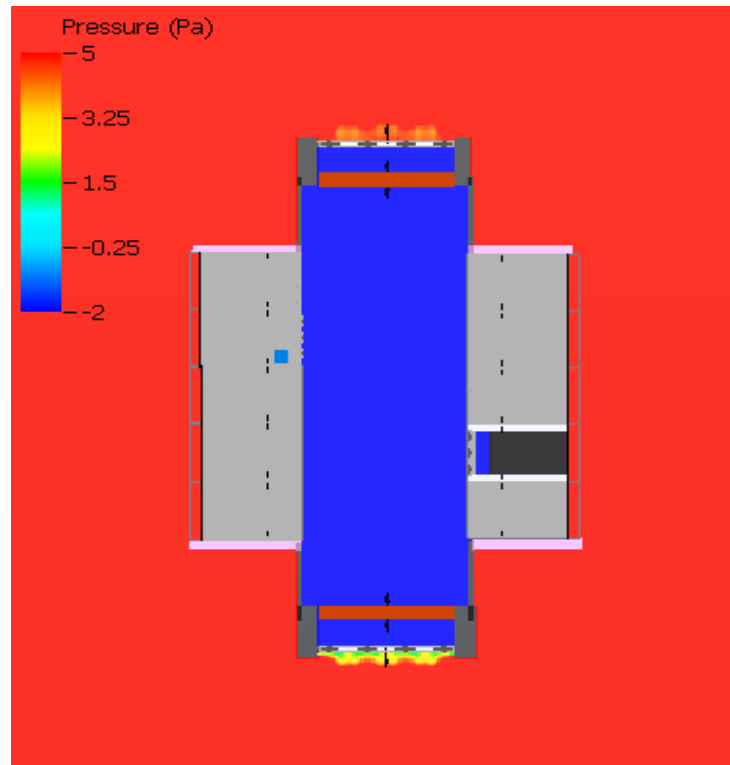


Figure 26: Pressure Plot for Case 3C (for same legend)

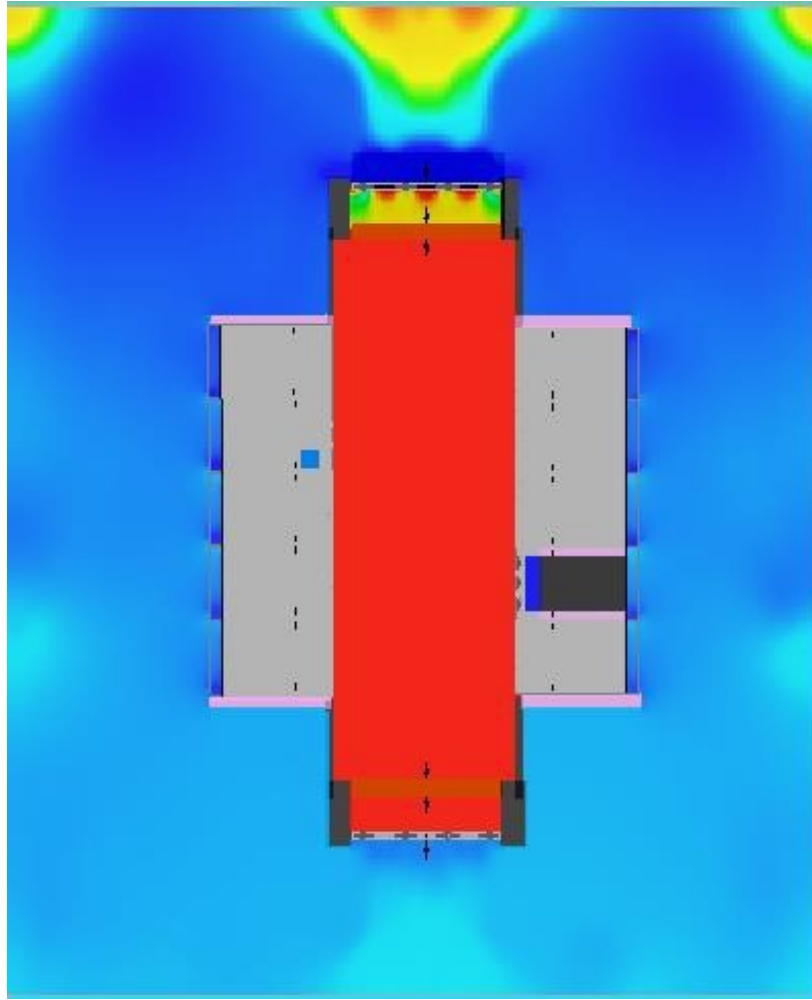


Figure 27: Pressure Plot for Case 2B (for same legend as figure 27)

The presence of rack fans in Case 2B blowing high pressure air in the hot aisle is driving the flow in this case. That is why the air flow across the heat exchangers is higher than that of Case 3C case.

Case 1A: In this case, the top two heat exchangers HX1 and HX3 are turned off with all rack fans turned on and no heat exchanger fans present.

It was seen that the temperatures in the room did not reach a steady state during simulation. The fan temperatures especially for the fans located at the upper half section of the rack exponentially increased

with each iteration. The average inlet temperature into the racks is varying. Since, only top section heat exchangers have been disabled, the top half of the server received inlet temperatures higher than 120 °C and the first half bottom section of the server received temperatures of about 36 °C inlet and second half of the bottom section received about 58.6 °C. The inlet temperature is ranging between 29.7 °C and 170 °C. The location of the heat exchanger failure influences the section of servers being affected by extreme over heating temperatures.

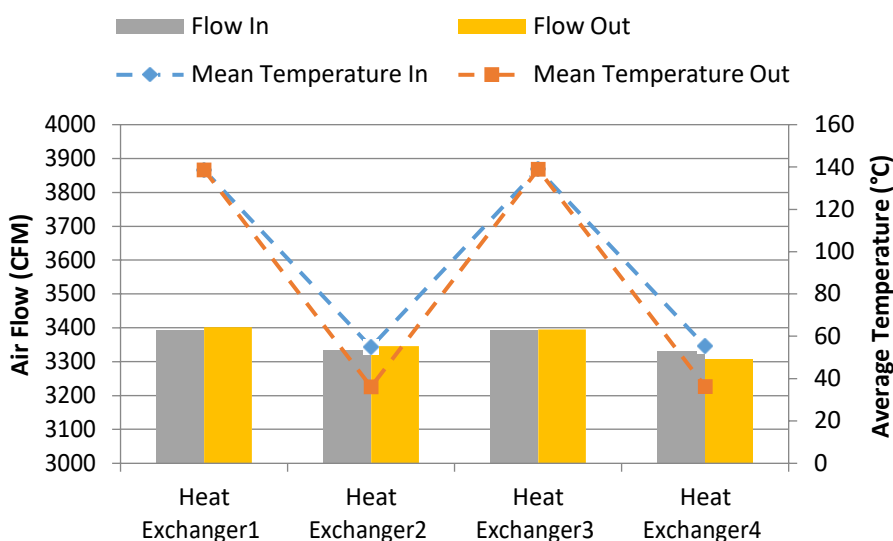


Figure 28: Heat Exchanger Flow and Temperature Parameters

The average temperature difference across the heat exchanger for HX2 and HX4 is about 19.2 °C as shown in figure 28. The average rack air flow rate is 1367.7 CFM. Unlike case 2 and Case 3 failure scenarios, in this case, the rack fans should modulate based on the supply air temperature of the rack (or heat exchanger) not only based on CPU based temperature.

Case 1B: This section considers both the top heat exchangers to be turned off and the rack fans of the middle racks (which are typically the racks receiving lower flow rate compared to the exterior racks). It has been noticed again that during the simulation the fan temperatures of the upper section of each rack are increasing with every iteration. The average air flow across the heat exchanger is 2809.5 CFM. The inlet temperature for HX2 and HX4 is 32.7 °C. The inlet temperature from disabled HX is about four times higher. The middle racks J3 and E3 have their fans turned off showing lower rack air flow rates as shown in figure 29. Because of lack of pressurization in J3 and E3 racks the hot air exhaust from hot aisle is entering the neighboring racks from them as shown in Figure 30.

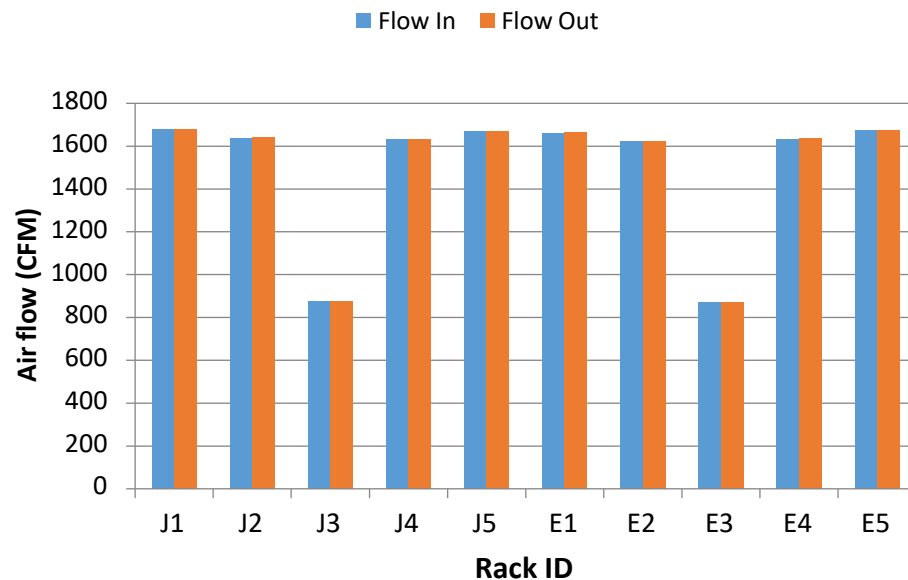


Figure 29: Rack Air Flow Rates Showing Lower Flow Rates for J3 and E3

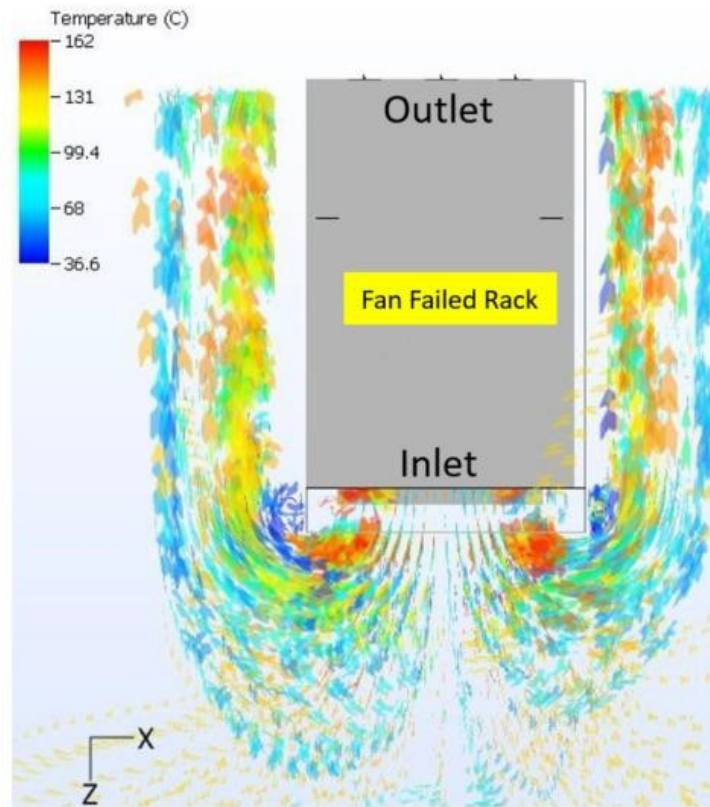


Figure 30: Hot Air Exhaust from E3 rack entering neighboring racks

The average rack inlet temperature is 82.9 °C. This scenario can be considered the worst case. The room temperature will quickly escalate to overheating temperatures. The racks J3 and E3 have showing higher temperatures because of the recirculation taking place from the hot air exhaust. The pressure plots shown in figure 31 demonstrate the low pressure regions in the hot aisle across servers J3 and E3 that are re-circulating the air into the neighboring cabinets.

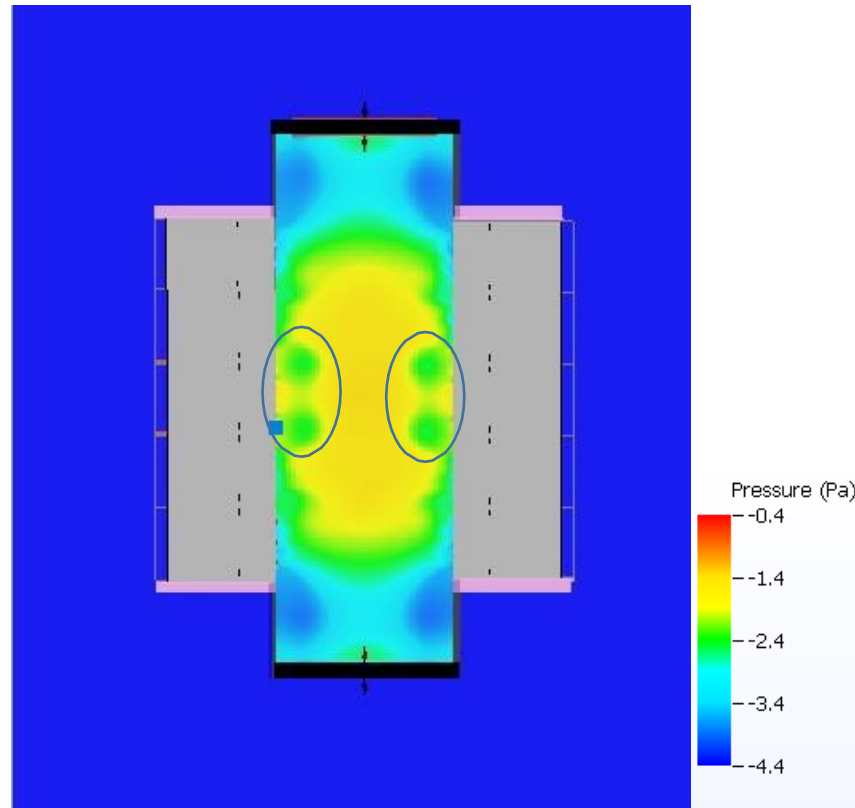


Figure 31: Pressure Plot in the room with cut plane across mid height of the racks (circled areas)

Additional case with only one top heat exchanger, HX1, failure has been simulated. It shows that the system is not over heating and the return air temperatures i.e. 47.7 °C are below the maximum limit. The failed heat exchanger would transport air temperature of 46.4 °C. The racks J2, J3 and E2, E3 have received lower CFM compared to other racks. There is non-uniformity in the temperature distribution across the racks. There is about 4 °C variation in the different rack inlet temperatures. But the system is not overheating unlike other failure scenarios Case 1A and Case 1B. Unlike failure scenarios of case 2 and case 3, if the rack fans are running in regular mode, there are no pressure losses in the hot aisle due to heat exchanger failure. Only the temperatures are impacted not the pressure profiles. The system is redundant for one heat exchanger failure during operation.

Overall, in all three cases of failure scenarios, it has been noticed that if two heat exchangers fail on one side it will be the worst case scenario. The case 1 failure scenarios have so far reported about 40 °C higher temperatures than the assumed upper hot aisle temperature (55 °C) which was the worst case of overheating observed. Case 1 and case 3 are redundant with one heat exchanger failure as shown in figure 21. However, Case 2 is redundant even with two heat exchanger failures due the presence of multi-stage fans for better air flow movement in the room. Case 2 and Case 3 shows similar results during worst case failure conditions (when both heat exchangers on one side without fan failure). In case 1C, the rack fans of the middle racks were turned off. As a result, the hot aisle exhaust that is at higher pressure is entering failed rack into the cold aisle and that hot air is again being circulated into the adjacent racks as shown in figure 22. Transient analysis is needed to estimate for each worst failure condition in all three different cases how much time it takes for the return air temperature in hot aisle to reach the maximum limit i.e. 55 °C.

Case	Status of Rack fans	Status of HX fans	Status of Heat exchangers	Overheating
Case 3A	-	HX1-OFF	HX1-OFF	No
Case 3B	-	HX1-OFF; HX2-OFF	HX1-OFF; HX2-OFF	Yes
Case 3C	-	All-ON	HX1-OFF; HX2-OFF	Yes

Case 2A	ON	HX1-OFF;HX2-OFF	HX1-OFF; HX2-OFF	No
Case 2B	ON	All-ON	HX1-OFF; HX3-OFF	Yes
Case 1A	ON	-	HX1-OFF;HX3-OFF	Yes
Case 1B	2 middle rack fans Off	-	All-ON	Yes

Table 2: Table showing cases that are overheating during HX and HX fan failure scenarios

8.1 Fan Control considering the failure scenarios

In all the cases, the rack fan control algorithm works in such a way that the rack fan speed/ revolutions per minute (RPM) should increase with rise in the CPU temperature. This is one general way of controlling the overheating of CPUs/ major processors in data center racks. However, beyond a certain higher operating limit on the CPU temperature, the server would shut down. Since the servers do not include any fans within them, the rack fan RPMs should slow down (as a part of secondary fan control scheme) with the increase in rack inlet temperature so that there will be more time for the room to reach unacceptable (i.e. greater than 55 °C) return air temperatures. Considering case 2 and Case 3 i.e. default designs, the second stage heat exchangers can modulate to achieve the desired supply air temperature. Also, the active fan heat exchangers should always maintain a certain minimum PWM such that they would not transfer the fan energy on to rack level fans to move the air flow in the room. However, when the heat exchangers fail without fan failure, the active fans which would otherwise run at higher RPMs (to reduce the liquid coolant temperature), could

distribute the hot air quickly across the room. This would lead to room reaching an unacceptable temperature affecting the reliability of the equipment. So, beyond certain coolant inlet temperature (greater than 45 °C) the active HX fans should shut down or run at idle PWM. The overall idea is that the effectiveness of the cooling design is also highly dependent on the effectiveness of fan control algorithm mainly for the heat exchanger fans in case 2 and case 3 and its corresponding failure scenarios.

Both the rack level fans (first stage fans) and the heat exchanger fans (second stage fans) need primary and secondary fan control algorithms. For the first stage fans it is based on the CPU temperature and rack inlet temperature. For the second stage fans, it is based on the supply air temperature and the coolant temperature.

8. 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 shown that such a system with 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 has 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 faster across heat exchangers coils reducing the thermal performance of the heat exchanger. Various factors differentiating all three designs have been tabulated in table 3. 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. The comprehensive cooling failure simulations helped to analyze how the cooling failure effect all three cases differently. Case 2 has shown good cooling redundancy with two heat exchangers failed unlike case1 and case 3. The presence of multi-stage fans will significantly help in controlling the overall air flow movement in the room. Further experimental validation and transient CFD analysis can study the 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. Based on the steady state CFD study, case 2 shows a better balance of design considerations (i.e. better scope for multi-stage fan control, better cooling redundancy) compared to case1 and case 3.

Factors	Case 1	Case 2	Case 3
Type	Rack level fans	Rack fans+HX fans	HX fans
Average Rack inlet temperature(°C)	10.3	8.4	9.3
Average Rack flow rate(CFM)	1382.5	1640.7	1572.9
Average HX flow rate(CFM)	3392.5	4229.5	3922.5
Average server flow rate(CFM)	32.8	35.9	37.8
IT equipment density	Low	Denser (1U HP server)	Denser (1U HP server)
Heat exchanger redundancy	1HX failure	2HX failure	1HX failure
Total fan power(W)	1038.2	2124.7	1572.9
Scalability (No.of racks)	Less	High	High

Table 3: Table showing the differences in temperature, pressure, and flow parameters in all three case

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