CFD SIMULATIONS COMPARING HARD FLOOR AND RAISED FLOOR CONFIGURATIONS IN AN AIR COOLED DATA CENTER

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ABSTRACT

An increasing number of companies and organisations have started to outsource their data storage. Although the potential of future investments in data centers is prosperous, sustainability is an increasingly important factor. It is important to make sure that the server racks in data centers are sufficiently cooled whereas too much forced cooling leads to economical losses and a waste of energy. Computational Fluid Dynamics (CFD) is an excellent tool to analyze the flow field in data centers. This work aims to examine the performance of the cooling system in a data center using ANSYS CFX. A hard floor configuration is compared to a raised floor configuration. When a raised floor configuration is used, the cold air is supplied into an under-floor space and enters the room through perforated tiles in the floor, located in front of the server racks. The flow inside the main components and the under-floor space is not included in the simulations. Boundary conditions are applied to the sides where the flow goes out of or into the components. The cooling system is evaluated based on a combination of two different performance metrics. Results show that the performance of the cooling system is significantly improved when the hard floor configuration is replaced by a raised floor configuration. The flow field of the air differs in the two cases. It is considered to be improved when the raised floor configuration is used as a result of reduced hot air recirculation around the server racks.

INTRODUCTION

The establishment of data centers is one of the most important growth factors in the IT business. One of the reasons for the growth is that an increasing number of companies and organisations have started to outsource their data storage. Although the potential of future investments in data centers is prosperous, sustainability is an increasingly important factor. In 2012, the total electricity used by data centers was about 1.4% of the total worldwide electricity consumption with an annual growth rate of 4.4% [1]. The main components in an air cooled data center are Computer Room Air Conditioner (CRAC) units and server racks. The server racks dissipate heat and need to be cooled in order to make sure that the electronics operate in the temperature range recommended by the manufacturer. Otherwise there is a risk of overheating, resulting in malfunction or shut down to prevent hardware damages. This interruption is costly for business and needs to be prevented. It is important to make sure that the data centers are sufficiently cooled whereas too much forced cooling leads to economical losses and a waste of energy. The CRAC units supply cold air into the data center. The cold air is supposed to enter through the front of the server racks and hot air will exit through the back. Depending on the distribution of CRAC units and server racks in the data center, there is a risk that the cold air does not necessarily reach all the server racks to the desired extent. Therefore, the cooling of data centers is crucially dependent on the flow field of the air. Computational Fluid Dynamics (CFD) is an excellent tool to provide detailed information about the temperature and flow field in a data center. CFD modeling can be used to analyze both existing configurations and proposed configurations before they are built [2]. It is important that the simulations are carried out with quality and trust.

The cooling system in an air cooled data center might be based on a hard floor or raised floor configuration. When a raised floor configuration is used, the CRAC units supply cold air into an under-floor space and the cold air enters the room through perforated tiles in the floor [3]. The perforated tiles are removable and can be replaced by solid tiles which makes the configuration flexible. Strategies for the design in data centers are often based on hot and cold aisles where the server racks are placed into a serie of rows. Cold aisles are formed between the front sides of two rows of server racks and hot aisles are formed on the other sides. This design strategy has become the standard when raised floor configurations are used. Perforated tiles are placed in the cold aisles and solid tiles are placed in the hot aisles. The purpose of the design strategy is to prevent hot air exhausted by the back of a server rack to enter the front of another server rack. Hot air recirculation over and around the server racks should be avoided if possible [4]. A ceiling return strategy can be used in order to further prevent hot and cold air from mixing. The hot air is then guided back to the CRAC units through a void space with ceiling vents or ducts located above the hot aisles [5]. Another strategy that can be used for the same purpose is aisle containment where either the hot or cold aisles are isolated [6]. Recommended and maximum allowable air temperatures at the front of the server racks are specified by ASHRAE's thermal guidelines. The recommended temperature range is 18-27°C and the allowable temperature range is 15-32°C [7].

The aim of this work is to examine the performance of the cooling system in an existing data center with a hard floor configuration and to compare it to a raised floor configuration. A room return strategy without aisle containment is used in the existing data center and the same applies to the simulations. The supply air flow rate is the same in both configurations. The commercial CFD software ANSYS CFX is used to perform the simulations.

NOMENCLATURE

A_{tile}	[m^2]	Area of a perforated tile
c_p	[J/kg·K]	Specific heat capacity
e	[J/kg]	Thermal energy per unit mass
k	[J/kg]	Turbulent kinetic energy
$\dot{m}$	[kg/s]	Mass flow rate
p	[Pa]	Pressure
$\bar{p}$	[Pa]	Mean pressure component
q	[W]	Heat load
Q	[m^3/s]	Volumetric flow rate
S_i	[kg/m^2·s^2]	Body force in tensor notation
t	[s]	Time
T	[°C]	Temperature
u_i	[m/s]	Velocity component in tensor notation
$\bar{u}_i$	[m/s]	Mean velocity component in tensor notation
u_i'	[m/s]	Fluctuating velocity component in tensor notation
V	[m^3]	Volume
x_i	[m]	Cartesian coordinate in tensor notation

Special characters

δ	[-]	Kronecker delta symbol
ε	[J/kg·s]	Rate of dissipation of turbulent kinetic energy
κ	[W/m·K]	Thermal conductivity
μ	[kg/m·s]	Dynamic viscosity
μ_t	[kg/m·s]	Turbulent viscosity
ρ	[kg/m^3]	Density
σ	[%]	Percentage open area of a perforated tile
Φ	[kg/m·s^3]	Dissipation function

THEORY

The governing equations of fluid mechanics are the conservation laws for mass, momentum and energy. For Newtonian fluids, they can be expressed as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{1}$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_j u_i)}{\partial x_j} = S_i - \frac{\partial p}{\partial x_i} + \mu\frac{\partial^2 u_i}{\partial x_j \partial x_j} \tag{2}$$

$$\frac{\partial(\rho e)}{\partial t} + \frac{\partial(\rho u_j e)}{\partial x_j} = -p\frac{\partial u_i}{\partial x_i} + \Phi + \frac{\partial}{\partial x_i}\left(\kappa\frac{\partial T}{\partial x_i}\right) \tag{3}$$

The governing equations of momentum (2) are also called the Navier-Stokes equations. In turbulent flow, the time averaged properties are often of greatest interest. The governing equations of the steady mean flow are called the Reynolds-Averaged Navier-Stokes (RANS) equations. They are obtained by introducing the Reynolds decomposition which means that the flow variables are decomposed into steady and fluctuating components. The RANS equations are obtained by using the Reynolds decomposition in the Navier-Stokes equations and taking the time average. The RANS equations read as follows:

$$\frac{\partial(\rho\bar{u}_i)}{\partial t} + \frac{\partial(\rho\bar{u}_j\bar{u}_i)}{\partial x_j} = \bar{S}_i - \frac{\partial\bar{p}}{\partial x_i} + \mu\frac{\partial^2\bar{u}_i}{\partial x_j \partial x_j} - \frac{\partial(\rho\overline{u_i'u_j'})}{\partial x_j} \tag{4}$$

As shown by the first term on the left hand side, depending on the choice of time averaging scale, transitions and larger fluctuations in the flow are still allowed. As compared to the Navier-Stokes equations, the RANS equations include additional stress terms called the Reynolds stresses. A turbulence model is needed to account for the Reynolds stresses and close the system of equations. The *k-ε* model is a turbulence model that solves additional transport equations for the two quantities turbulent kinetic energy and rate of dissipation of turbulent kinetic energy. It assumes isotropic turbulence and is based on the Boussinesq approximation which assumes that the Reynolds stresses are proportional to the mean velocity gradients according to:

$$-\rho\overline{u_i'u_j'} = \mu_t\left(\frac{\partial\bar{u}_i}{\partial x_j} + \frac{\partial\bar{u}_j}{\partial x_{ij}}\right) - \frac{2}{3}\rho k\delta_{ij} \tag{5}$$

The velocity approaches zero close to a solid wall and there is a viscous sublayer. Far away from the wall, inertia forces dominate over viscous forces. Before the wall distance reaches zero there is a region where both inertia forces and viscous forces are important. This region is called the log-law layer because of the logarithmic relationship between the dimensionless expressions for velocity and wall distance [8]. The *k-ε* model includes a wall function approach in ANSYS CFX that can be used for cases when it can be assumed that the entire boundary layer does not need to be resolved. The near-wall grid points should then be located at a dimensionless wall distance within the range where the log-law layer is valid [9].

METHOD

The main components in the existing data center are two CRAC units and twelve server racks which are arranged in four parallel rows based on the design strategy of hot and cold aisles. Figure 1 illustrates the geometry for the hard floor configuration. The dimensions of the data center are 10.10 x 3.74 x 2.54 m^3. A hexahedral mesh is used since the geometry of the computational domain is relatively simple. It was concluded that a mesh with maximum cell size equal to 0.05 m is appropriate based on previous studies on grid size in CFD modeling of data centers [10]. The aspect ratio is kept as close as one as possible but there are deviations at some locations as a result of the dimensions of the geometry.

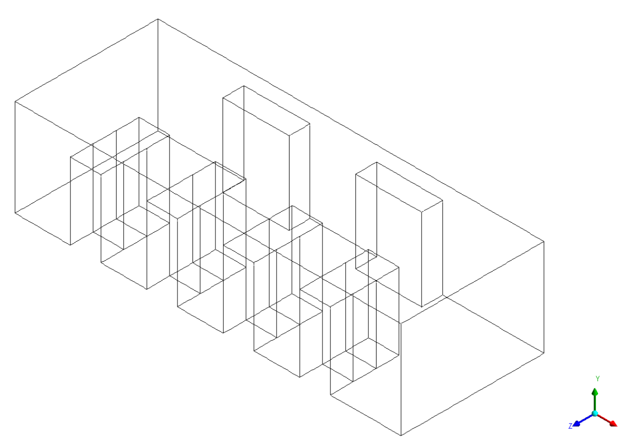

Figure 1 Geometry of the existing data center

The flow inside the CRAC units and the server racks is not resolved and all components are modeled as "black boxes". Boundary conditions are applied to the sides where the flow goes out of or into the components. For the server racks, it means that the flow enters the front and then reappears at the back. A uniform velocity normal to the surface is defined as boundary condition on the front of the server racks. The velocity is set to 0.58 m/s in order to obtain the measured flow rate of 0.7 m^3/s. The mass flow rate should be kept constant between the inlet and outlet located at each server rack, respectively. It is thus assumed that it is the internal fans that draw air through the front. The temperature distribution at the front of the server racks is taken from the adjacent upwind cells. Each server rack generates a heat load of 5700 W at full capacity. The full capacity of all server racks is considered in this study even though it is not currently used in the existing data center. The heat load generates a temperature difference between the back and the front of the server racks. The temperature increase is determined as:

$$\Delta T = \frac{q}{\dot{m} c_p} \tag{6}$$

The black box model has previously been compared to more detailed server rack models in order to investigate the extent to which the temperature and flow field in a data center is affected by the choice of server rack model. It was shown that there are no significant differences when using different server rack models [11, 12]. The flow rate supplied by the CRAC units can be varied. However, in this study it is set equal to the flow rate demanded by the server racks. It results in a velocity of 1.21 m/s. The measured supply air temperature is equal to 17°C. The mass flow rate should be kept constant between the inlet and the outlet located at each CRAC unit, respectively.

Figure 2 illustrates the geometry for the raised floor configuration. The under-floor space is not included in the simulations. In addition to the main components that already has been described, twelve perforated tiles are now included. The dimensions of a perforated tile are 0.6 x 0.6 m^2. They are located right in front of the server racks and supply air at a velocity of 1.93 m/s since the supply air flow rate should be the same for both configurations. It is assumed that all perforated tiles supply air at the same flow rate. However, this is a simplification [13]. The same grid size as for the hard floor configuration is used.

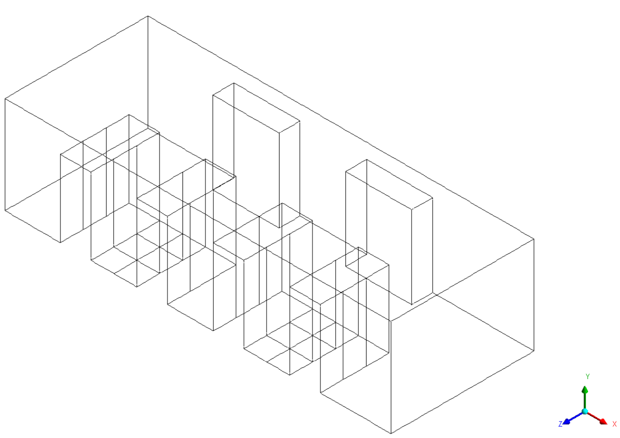

Figure 2 Geometry of the raised floor configuration

The perforated tiles in a raised floor configuration have a large number of small openings which requires a very fine grid to resolve the flow when using CFD modeling [14]. The percentage open area is what characterizes a perforated tile and 25% is the most common value. Due to grid size limitations, the perforated tiles have often been modeled as fully open tiles with constant velocity. The reasoning behind this simplification is that since the openings in the perforated tiles are very small, the small jets will quickly merge into a single large jet [15]. In reality, when air flows through the perforated tiles, it is accelerated through the small openings which results in a momentum increase in the vertical direction and also in losses [16]. The velocity through a fully open perforated tile can be set equal to Q/A_{tile} in order to obtain the correct flow rate and the resulting momentum flux is then equal to $\rho Q(Q/A_{tile})$. The velocity is lower than the correct jet velocity through the small openings. The average of the jet velocity through the small openings in the perforated tile is equal to $Q/\sigma A_{tile}$ where σ is the percentage open area and the resulting momentum flux is equal to $\rho Q(Q/\sigma A_{tile})$. Therefore, the correct momentum flux is not obtained if the perforated tile has a percentage open area less than 100%. A body force model is used to model the perforated tiles in the CFD model. It is applied in order to compensate for the low momentum fluxes of the jets that enter the data center through the perforated tiles when they are modeled as fully open. The flow area is still considered to be the full area of the tile. When the body force model is used, a body force term is added in the momentum equation in the vertical direction to correct for the momentum deficit. The body force per unit volume in the vertical direction is expressed as:

$$S_y = \frac{1}{V} \rho Q \left(\frac{Q}{\sigma A_{tile}} - \frac{Q}{A_{tile}} \right) \tag{7}$$

where V is the volume of the region above each of the perforated tiles where the body force is distributed [17]. The region above a perforated tile is modeled as a subdomain with the same area as the perforated tile and a height equal to 0.1 m.

The walls, the floor and the ceiling are assumed to be adiabatic. It is also assumed that there is no leakage of air out of or into the data center. The *k-ε* model with scalable wall functions is used. The dimensionless wall distance from the first interior node is kept well below the upper limit of the range where the log-law layer is valid everywhere except for above the CRAC units. It has been shown that buoyancy often should be included in CFD modeling of data centers and how the importance of including buoyancy can be evaluated [18]. The full buoyancy model in ANSYS CFX is consequently used in this study.

Performance metrics

A combination of Rack Cooling Index (RCI) and Capture Index (CI) is used to evaluate and compare the results of the configurations. The RCI has two parts, one that focuses on the high end of the temperature range and one that focuses on the low end. Only the high end of the temperature range is considered here. Over-temperature is what it is called when the maximum value of the temperature at the front of a server rack exceeds the maximum recommended temperature. A summation of the over-temperature across all the server racks is called the total over-temperature. The RCI is defined as follows:

$$\mathrm{RCI} = \left(1 - \frac{\text{Total over-temperature}}{\text{Max. allowable over-temp.}}\right)100[\%] \qquad (8)$$

where the maximum allowable over-temperature is a summation of the difference between the maximum allowable and maximum recommended temperature across all the server racks. If the temperature at the front of all server racks is below the maximum recommended temperature, the RCI is equal to 100% which is the ideal value. If the temperature at the front of at least one server rack exceeds the maximum recommended temperature, the RCI is less than 100%. If the temperature at the front of at least one server rack exceeds the maximum allowable temperature, the value is marked with a warning flag "*" [19].

The CI has two parts, one that focuses on the cold aisles and one that focuses on the hot aisles. Only the CI for cold aisles is considered here. The CI is then defined as the fraction of air that enters the front of a server rack that originates from the CRAC units or the perforated tiles. All the server racks are assigned individual values. If all the air that enters the front of a server rack originates from the CRAC units or the perforated tiles, the CI is equal to 100% which is the ideal value. However, a low value does not necessarily imply that the server rack is insufficiently cooled. Most of the air that enters the front of a server rack might originate from the surrounding room environment instead of from the CRAC units or the perforated tiles. The CI will be low but if the temperature of the surrounding room environment is low enough, the server rack will be sufficiently cooled. This situation is more complex and unpredictable. Therefore, high values of CI are preferable in data centers even though it might not be necessary. To be able to evaluate the CI, a volumetric additional variable is defined and a transport equation is solved for it in the CFD model. The transport of this additional variable is a purely convective process. The concentration is set equal to one at the inlets located on the CRAC units or the perforated tiles and equal to zero at the inlets located on the server racks. The concentration is then evaluated at the front of the server racks and it represents the fraction of air that originates from the CRAC units or the perforated tiles [20].

RESULTS

The temperature distribution at the front of the server racks in the hard floor configuration is illustrated in Figure 3. The temperature scale ranges from 17°C which is the temperature of the air supplied by the CRAC units to 32°C which is the maximum allowable temperature at the front of the server racks according to ASHRAE's thermal guidelines. The temperature exceeds the maximum allowable temperature at large parts of some of the server racks. It means that the cooling system needs to be revised if the server racks in the existing data center would generate full heat load. The temperature is far above the maximum recommended temperature of 27°C at even larger parts of the server racks. Therefore, the value of RCI is very poor and is equal to 37%*.

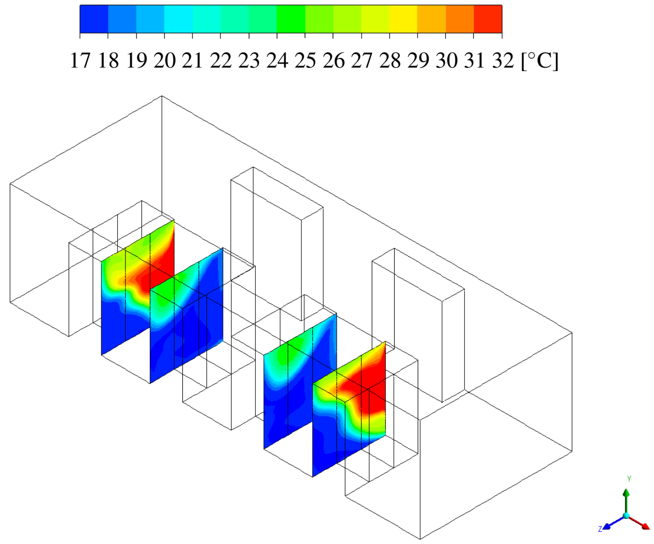

Figure 3 Temperature at the front of the server racks in the hard floor configuration

The temperature distribution at the front of the server racks in the raised floor configuration is illustrated in Figure 4. The temperature scale is the same as in Figure 3 in order to simplify comparison of the results. However, a much smaller temperature scale could have been used for this configuration. The temperature does not exceed either the maximum allowed temperature or the maximum recommended temperature at any part of the server racks. It means that the cooling system based on a raised floor configuration would have been more than sufficient when the server racks generate full heat load. The supplied air flow rate is the same for both configurations which means that the raised floor configuration performs much better than the hard floor configuration based on the temperature at the front of the server racks. The fact that no part of the server racks exceeds the maximum recommended temperature at the front results in an ideal RCI value of 100%.

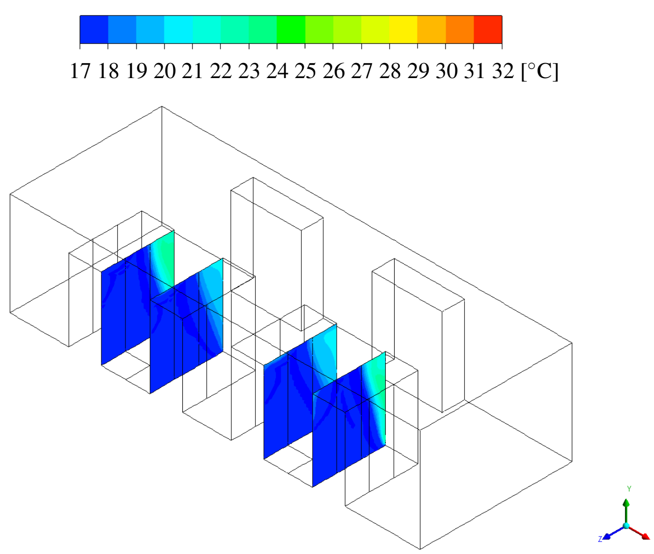

Figure 4 Temperature at the front of the server racks in the raised floor configuration

In Figure 3, it is seen that it is mainly the two outermost rows that are insufficiently cooled in the hard floor configuration. The flow field of the air for this configuration is illustrated in Figure 5 and the color of the streamlines represents velocity. The flow field in the data center is divided into two almost equal parts where each CRAC unit supplies cold air to one of the two cold aisles, respectively. The air that is exhausted by the two innermost rows meet in a hot aisle and is pushed up towards the ceiling before it returns to the CRAC units. Some fraction of the air supplied by the CRAC units returns directly to the CRAC units without cooling any server racks. In Figure 5, this is illustrated for one of the CRAC units by the curved arrow. Since the flow rate supplied by the CRAC units is equal to the flow rate demanded by the server racks, it results in hot air recirculation around some of the server racks. This is what happens at the end of the two outermost rows. The air exhausted by the back of the server racks is recirculated and enters the front of the server racks in the same row again, resulting in temperatures exceeding both the maximum recommended temperature and the maximum allowable temperature.

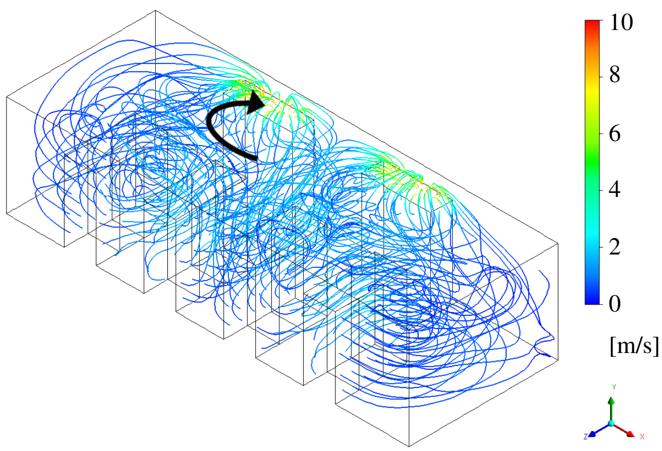

Figure 5 Streamlines of the flow in the hard floor configuration

The CI for the hard floor configuration is illustrated in Figure 6. The values of the CI support the reasoning based on the flow field. It can be seen that the values of CI are higher for the server racks closest to the wall. The server racks at the end of the two outermost rows have the lowest values of CI as a result of the hot air recirculation that occurs there.

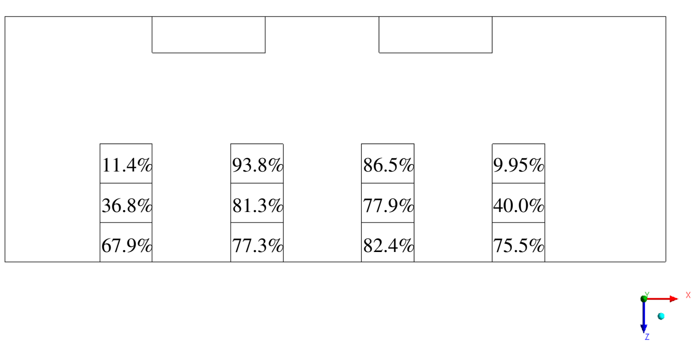

Figure 6 CI for the hard floor configuration

The flow field in the existing hard floor configuration can be compared to the flow field in the raised floor configuration in order to further explain why the temperature distribution at the front of the server racks is improved in the latter case. The flow field of the air in the raised floor configuration is illustrated in Figure 7. It can be seen that the flow field in the data center is divided into two almost equal parts again. The air that is exhausted by the two innermost rows meet and is pushed up towards the ceiling before it returns to the CRAC units. Each CRAC unit approximately receives the hot air exhausted from two of the rows with server racks, but the CRAC units do not supply the cold air directly into the data center anymore. Instead, it is supplied into the under-floor space. The air enters the data center through the perforated tiles located right in front of the server racks which implies that the cold air reaches the server racks to a much higher extent than before. The problem with air that returns directly to the CRAC units is not as extensive in this configuration. There is some hot air recirculation at the end of the rows, but this is to such a low extent for the raised floor configuration that it does not cause any problems.

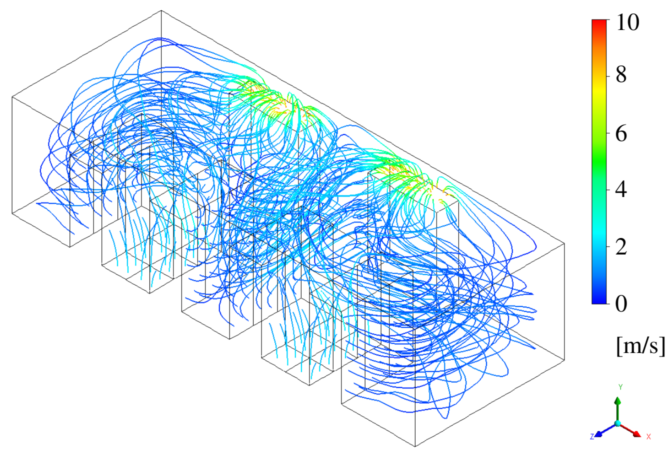

Figure 7 Streamlines of the flow in the raised floor configuration

The CI for the raised floor configuration is illustrated in Figure 8. It supports the reasoning about the flow field. The recirculation that occurs in this setup does not affect the temperature distribution at the front of the server racks so much that it exceeds the maximum recommended temperature and it is kept well below the maximum allowable temperature.

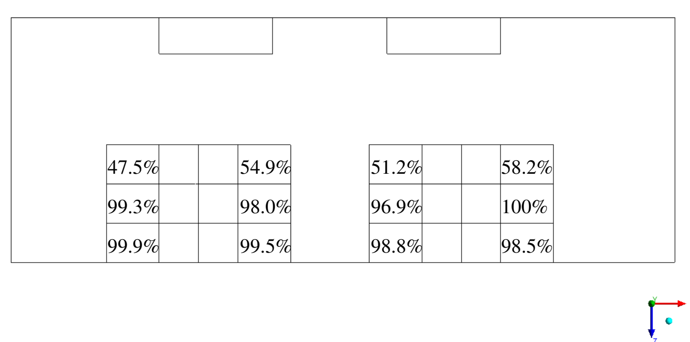

Figure 8 CI for the raised floor configuration

CONCLUSIONS

The performance of the cooling system in an existing data center with a hard floor configuration has been examined and compared to a raised floor configuration. A combination of the performance metrics RCI and CI has been used. Results for the hard floor configuration show that the cooling system needs to be revised if the server racks would generate full heat load. The cold air does not reach all the server racks to the desired extent which results in insufficient cooling for some of them. The temperature at the front of some of the server racks exceeds both the maximum recommended temperature and the maximum allowable temperature. Some fraction of the air supplied by the CRAC units returns directly to the CRAC units again which implies that hot air recirculation is present around the end of the two outermost rows of server racks.

The benefits of replacing the hard floor configuration in the existing data center by a raised floor configuration are clear. Results for the raised floor configuration show that the performance of the cooling system is significantly improved compared to when the hard floor configuration is used. Both the value of RCI and the values of CI for ten of the twelve server racks are increased. The flow field of the air differs between the two configurations. The main difference is that the perforated tiles supply the cold air right in front of the server racks. As a result, most of the air that enters the front of the server racks originates from the perforated tiles which in turn results in temperatures close to the supply temperature as well as almost ideal CI values for most of the server racks. The hot air recirculation that is present in the hard floor configuration is reduced when the raised floor configuration is used. Therefore, the flow field is considered to be improved.

It could be investigated how either increasing the flow rate of the supplied air to a value that excess the flow rate demanded by the server racks or decreasing the temperature of the supplied air would affect the performance of the cooling system of the data center considered in this study. However, it should be noted that ASHRAE's thermal guidelines not only specifies maximum allowable and maximum recommended temperatures but also minimum allowable and minimum recommended temperatures at the front of the server racks. The minimum allowable temperature at the front of the server racks is equal to 15°C. Therefore, it exists a limit to how much the supply temperature can be used to improve the performance of the cooling system. As a final remark it should be stated that a smarter distribution of the cold air in a hard floor configuration may be proven as efficient as a raised floor configuration. Such optimizations are topics for future work.

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