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Evaluation of cooling air distribution system at Amirkabir University's high-speed data processing center

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ABSTRACT: In the present paper, the distribution of air and temperature contours at the Amirkabir University of Technology's high-speed data processing center has been studied using numerical simulation with 6Sigmadcx software. To solve the governing equations the finite volume method and to create a turbulent effect k-epsilon model has been used. The purpose of the management of air distribution systems is to solve the problems in the cooling of data centers, including minimizing the hot air recirculation, cold air bypass. To solve these problems, the data center once has been simulated with a cold aisle enclosure and again with a hot aisle enclosure with the return flow from the ceiling plenum and the results of dimensionless temperature indices were compared with the results of the main model of the data center. The most important index that is first reviewed is the rack cooling index high which measures the difference between overheating temperatures. According to the results, this index for the main model is 82, for the cold aisle enclosure model is 95 and for hot aisle enclosure with the return flow from the ceiling plenum is 99. Also, by considering the optimum value of 100, the hot aisle enclosure with the return flow from the ceiling plenum model is a more suitable model for the data center and all racks are cooled in ASHRAE (2011) standard range.

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I NTRODUCTION

A data center is a place used to arrange computer systems and related equipment, such as storage and communications systems. The cooling systems in the data centers should work all the time (24 hours and 7 days a week) [1]. Cooling systems consume 38% of the total energy needed in data centers, which accounts for a high percentage of energy consumption. So, it is important to use a proper cooling system that is both energy-efficient and decent air distribution in the data center [2]. Each server must receive a certain amount of cold air according to its heat transfer rate. Therefore, the solution to ensure the proper operation of this equipment is that the cool air must be distributed properly and satisfies the needs of each area in the data center [3]. Increasing computational capabilities is consistent with increasing the number of racks and power density. Adding the number of cooling units and increasing their power is not a good way to manage airflow in data centers [4]. Many researchers have studied data centers in the context of thermal conditions. Schmidt and Iyengar [5] also examined the airflow conditions for high-density data centers. The result of this study was that the supply of airflow from the floor with the return of air from the ceiling plenum is the best model for airflow conditions. Cho et al. [6] evaluated the air distribution management system in high-density data centers using 6SigmaDCX software. To improve the Return Heat Index (*RHI*) and Supply Heat Index (*SHI*), they proposed an underfloor air distribution system with return flow through the ceiling plenum duct from the hot aisle to CRACs units. Also, to improve the Return Temperature Index (*RTI*) and Rack Cooling Index (*RCI*), they proposed a cold aisle partitioning system with a height of 0.6 m. Yanmei et al. [7] developed a numerical simulation of a data center in Singapore with 6SigmaDCX software and concluded that by cold aisle enclosure with return hot airflow from the ceiling to CRACs Units, by-pass airflow is reduced and the energy consumption of the air conditioner fans decreases by 20-25%. Huang et al. [8] developed a numerical simulation of a data center with three different types of air distribution: An under-floor air supplying system, a row-level cooling system, and rack cooling performance and concluded that the rack level airflow pattern shows the best performance in data center cooling. Nada and Elfeky [9] studied various configurations of cold aisles containment and comparisons between different states. These states were: free open cold aisle, semi-enclosed cold aisles where the aisles are enclosed from the sides, full enclosed cold aisles where the aisles are enclosed from sides and top. This study showed that the percentage of enhancement in data center performance due to using cold aisle containments dramatically increases with increasing the power density. Chao et al. [10] examined the effect of aisle enclosure on the cooling efficiency of data centers with a low cooling load. Song [11] described the application of fan-assisted

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floor perforations to evaluate the cooling operation of a data center configuration and concluded that the use of the well-constructed performance indices and categorical designs, it could significantly enhance efficiencies of data centers. Cho and Woo [12] investigated the thermal performance of an existing room-based cooling system and a new independent row-based cooling system in an experimental measurement and showed that data center cooling systems utilizing the fully containment of cold and hot aisles technique are very effective at removing heat load of IT servers.

The use of temperature indices helps to understand the efficiency and performance of air distribution. Most of the investigations cited in the literature performed numerical simulations for the imaginary data center model. While reviewing these Computational fluid dynamics (CFD) studies, it seems that the effects of an increasing volume of airflow, an increasing number of air cooling units, and decreasing supply air temperature for different configurations of cold aisle enclosure, hot aisle enclosure, and as well as enclosure of both cold and hot aisles to achieve a best thermal performance of data center by comparing temperature indices were not investigated. In this research, the distribution of air and temperature at the Amirkabir University of Technology (AUT)'s high-speed processing center has been studied numerical simulation using 6Sigmadcx software. With over 4000 objects, 6Sigma DCX software has the most comprehensive data center library in any set of CFD packages. Especially, IT equipment objects are simplified representations of equipment that simulate their overall physical geometry and the power (heat) and airflow interactions with their surroundings, and also their internal airflow and heat transfer can be modeled in 6SigmaET or 6SigmaRack by using a model called a chassis. The software provides the ability to control the airflow and heat transfer reflecting the capacity and control settings for the Air Cooling Unit (ACU) and allows for ACUs to be modeled in either simplified or detailed levels. This library has servers and air cooling units from any manufacturer, brand, and business group. So, the CFD-solver of 6Sigma DCX software has higher speed and accuracy than other common CFD solvers because of the ready-made packages and equipment. Then, the results of numerical simulation are investigated and validated according to the temperature measurement in the data center. According to the results, the re-circulation of hot air and cold air by-pass negatively affected the cooling process of the servers. To solve these problems and improve the thermal performance of the air distribution system, the data center once has been simulated with a cold aisle enclosure and again with a hot aisle enclosure with the return flow from the ceiling plenum. Results of dimensionless temperature indices have been compared with the results of the main model of the data center.

2. PRESENT MODEL

Hot air that exhausting from the rear of the server may be returning from above or inside the server to the cold aisle and after mixing with inlet cold air again can enter the server from the front of the server, which is called hot air re-circulation (Fig. 1). Also, the cold flow that passes through the cold aisle without cooling the racks and returns to the CRAC units return valves, is called cold air by-pass (Fig. 2). The goal of air-distribution systems management is to solve the problems in the data centers' cooling, including minimizing re-recirculation of hot air, minimizing cold air bypass, and eliminating excessive cold points. Recirculation flow and bypass flow reduce the efficiency of the cooling system [13].

AUT data center's server room with dimensions of 12.4 m × 8.7 m × 2.5 m has been considered as the physical model (Fig. 3). The height of the plenum of the raised floor is 0.64 m. A total of 12 perforated tiles with dimensions of 0.6 m × 0.6 m and an opening ratio of 50% is installed in the cold aisle for cooling the racks. This center includes 19 racks with dimensions of 1.97 m × 0.75 m × 1 m, arranged in two rows with a spacing of 1.8 m between the two rows. The first row has 12 racks, where the distance of this row from the sidewall (longitudinal) and the transverse wall respectively is 1.24 m and 2.35 m. The racks of this row with the servers and switchers of them in different slots of each cabinet have been shown in Fig. 4. The second row has 7 racks, where the distance of this row from the sidewall (longitudinal) and the transverse wall respectively is 3.25 m and 2.35 m. The racks of this row with the servers and switches of them in different slots of each cabinet have been shown in Fig. 5. The total heat load of the server room that calculated by the software automatically is 65 kW, of which 42.2 kW is IT equipment (sum of all available server heat from the library of 6SigmaDCX), 13.2 kW for cooling systems (kW consumed by all air cooling units) and 9.6 kW for data center's lighting (24 lamps of 400 Watts). The heat load per cabinet has been shown in Fig. 6. Both of hot aisle and cold aisle are enclosed and a circular vent with 60 cm diameter has been installed on the east wall of the room to exhaust hot air at a flow rate of 1.25 $\frac{m^3}{}$ from the hot aisle enclosure with a circular duct of 60 cm diameter at the overhead of racks 3 and 5 (Fig. 3). There is a 3.2 m gap in the middle of the second row as the cold aisle enclosure door. The total capacity of the cooling systems and air conditioner units is 195 kW. The specifications and boundary conditions for CRAC units according to Fig. 7 are given in Table 1. Note that, these values have been recorded from CRAC units panels in the server room. In Fig. 7 arrangements of CRAC units have been shown.

3. MATHEMATICAL FORMULATION

In this section, firstly governing equations to simulate the incompressible fluid flow and heat transfer are described, then the dimensionless temperature indices which are important for data center analysis in minimizing the by-pass flow and recirculation flow and eliminating excessive cold points and hotspots are introduced.

3.1. Governing Equations And Numerical Method

The calculations and solving of the governing equations of the simulated data center have been used in the 6SigmaDCX software and the fundamental equations have

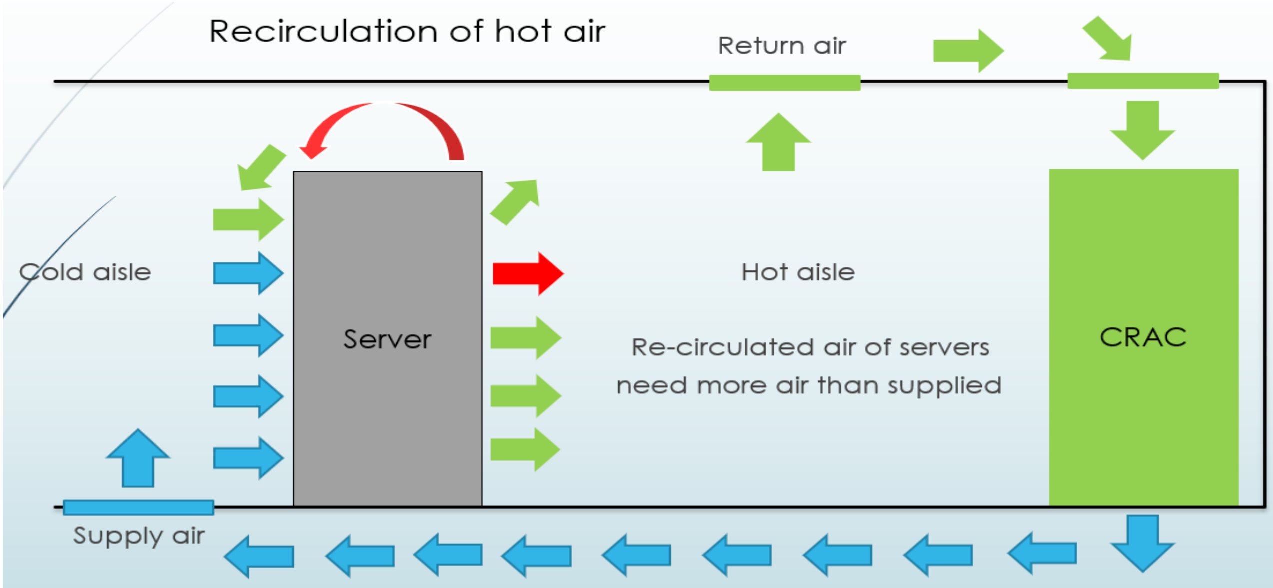

Fig. 1. Re-circulation airflow

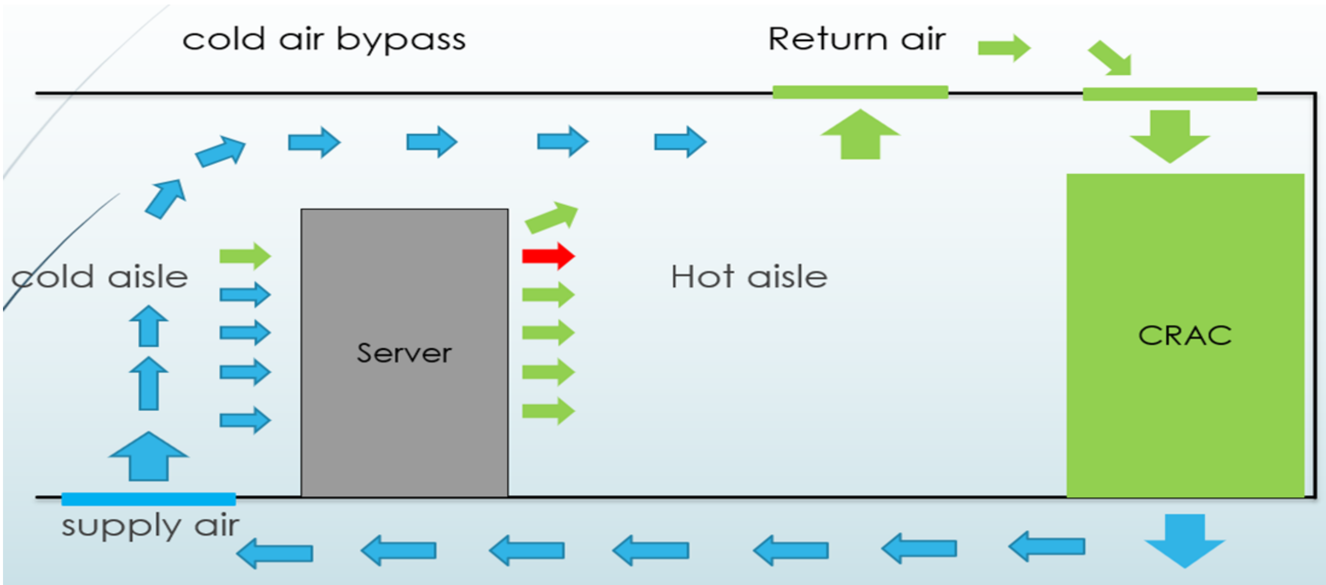

Fig. 2. By-pass airflow

been solved using the code of this program in a steady-state, incompressible, and turbulence 3D domain. In this numerical simulation, to solve the governing equations, the finite volume method and for the accuracy of the computations, the second-order upwind method has been used for the discretization of advection terms. The working fluid is considered to be incompressible concerning the low Mach number. Because the volume of supplying air in the data center does not change and is constant. So, for the dependency of the field of velocity and pressure, the SIMPLE algorithm is used. With these assumptions conversations of mass reduces to:

$$\nabla.(\underline{V}) = 0 \tag{1}$$

The onset of turbulent flow depends on the fluids› speed, its viscosity, its density, and the size of the obstacle it encounters. The Reynolds number can be used to predict the onset of turbulent flow:

$$Re = VL / \nu \tag{2}$$

where, V is the velocity of the air, L is the the characteristics

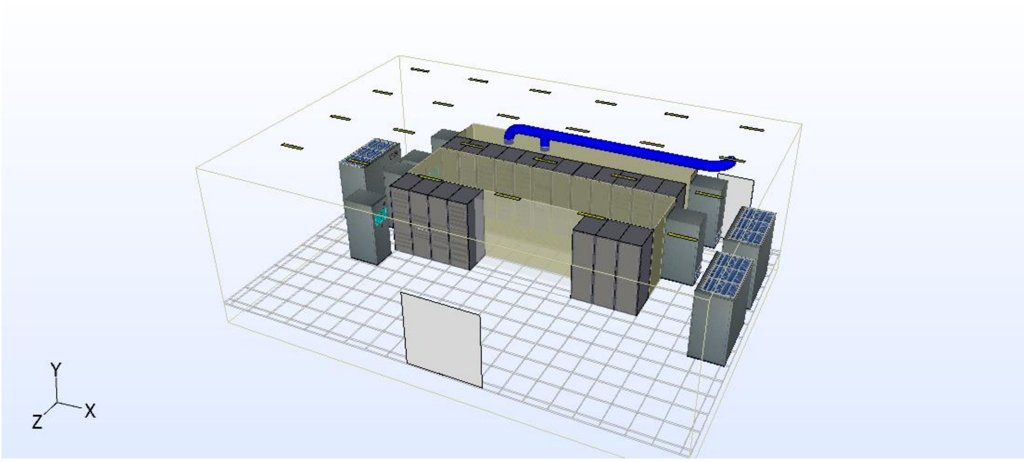

Fig. 3. 3D view of IT server room

Fig. 4. Cabinets of Row-1

Fig. 5. Cabinets of Row-2

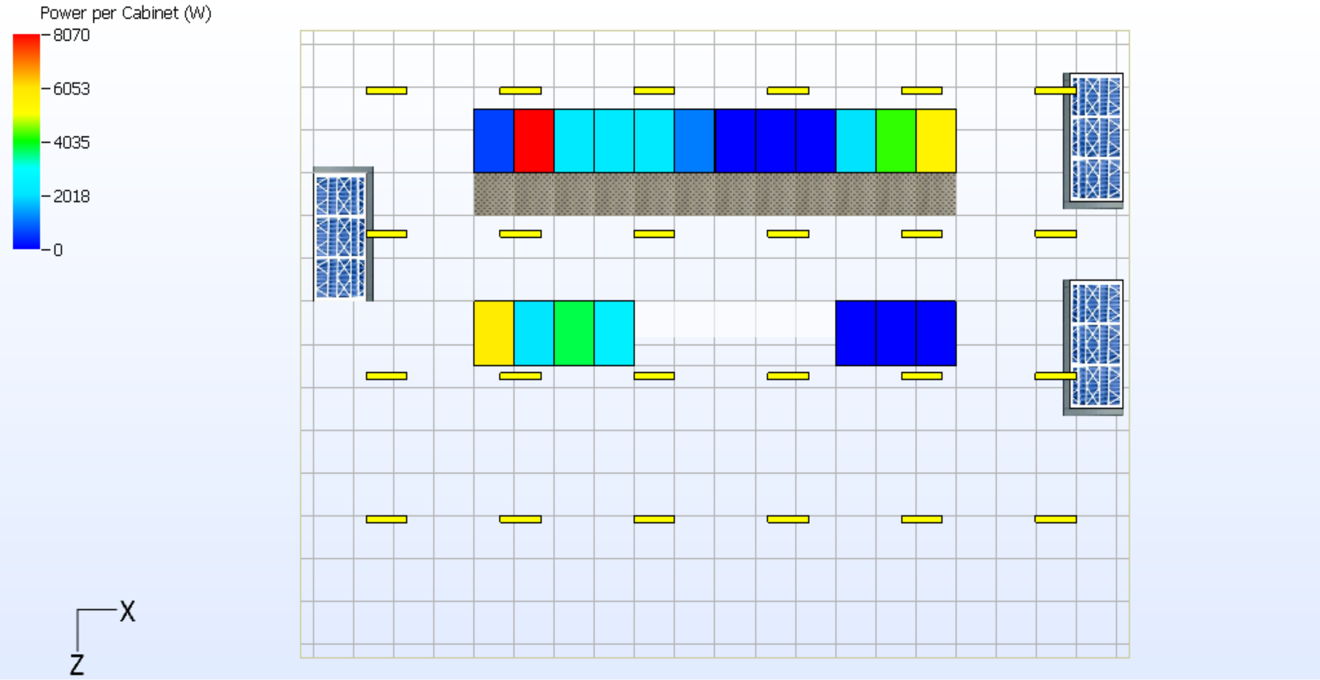

Fig. 6. Power of per Cabinet

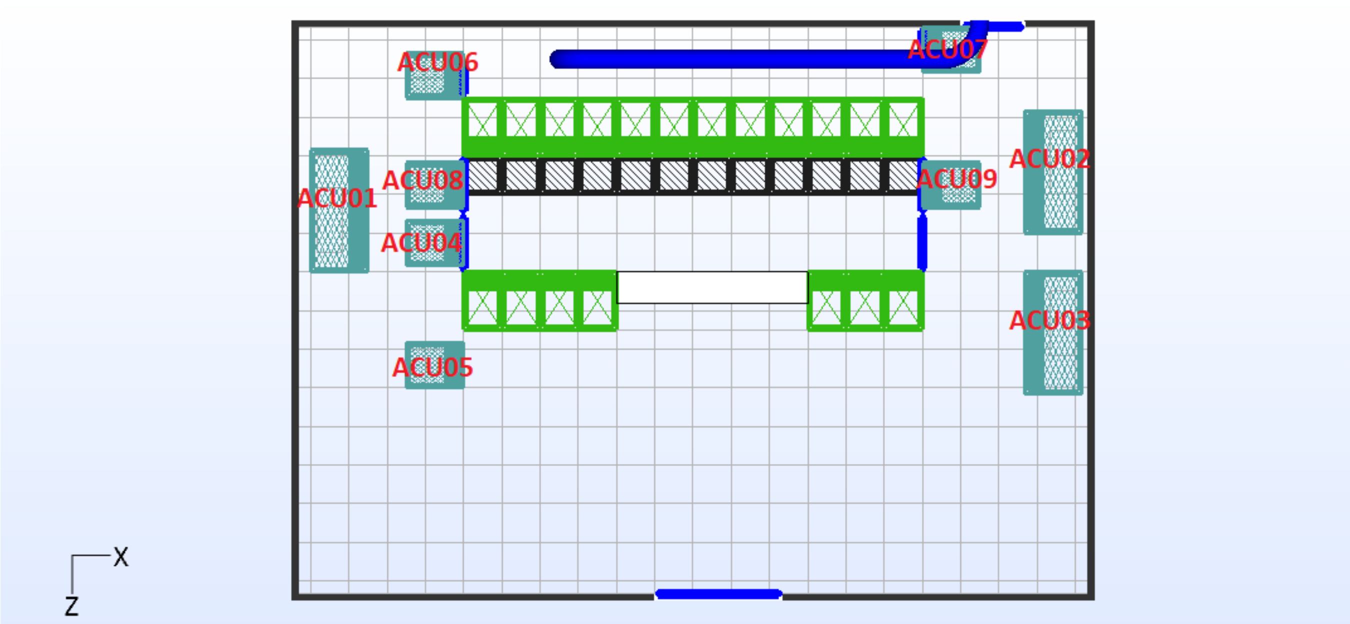

Fig. 7. The numbering of CRAC units

length, and v is the kinematic viscosity of the air that is equal to 1.5111E-5 m^2/s at 20^{0}C temperature and atmospheric pressure. According to the ACU01 panel, the supply velocity of the fans is 2.7 m/s so the Reynolds number in 2.8 meters of the CRAC at the plenum of the raised floor reaches 500,298 and turbulent flow happens. Note that, turbulence appears when the Reynolds number is about 500,000 on a flat plate. Also, for the plenum of the raised floor as a channel, the hydraulic diameter is 1.2 m and the supply velocity of ACU02 is 2.5 m/s so the Reynolds number is 198,531 and for the cold aisle containment, the hydraulic diameter is 2.1 m and the supply velocity of ACU09 is 1 m/s so the Reynolds number is 138,980. Note that, the Reynolds number range to detect turbulence in internal flows as follows:

$$Re_{d_h} \geq 2300 \tag{3}$$

The Reynolds number is most likely high in large parts of the data center and turbulence must then be taken into account when modeling the airflow. The Reynolds Averaged Navier Stokes (RANS) form of the governing equations is used to create the effect of the turbulence transport. The time-averaged conservation of momentum equation for an incompressible and Newtonian fluid is:

$$\rho \frac{D\underline{V}}{Dt} = -\nabla P + \nabla.\left[\left(\nu + \nu_t\right)\nabla \underline{V}\right] + \rho \underline{g} \tag{4}$$

The turbulent form of the conversation of the energy equation in terms of time-averaged variables including server heat source is as follows:

$$\frac{D\underline{T}}{Dt} = \nabla.\left[\left(\frac{\nu}{\Pr} + \frac{\nu_t}{\Pr_t}\right)\nabla \underline{T}\right] + \dot{q} \tag{5}$$

where T is the temperature, Pr is the Prandtl number, and q is volumetric heat source per unit mass.

The above equations are solved numerically with CFD-Solver of 6SigmaDCX software under the boundary conditions mentioned in Table 2. In all simulations, the *k-ε* turbulence model as only and default model in CFD-solver of the software is used to solve the Reynolds stresses. The *k-ε* turbulence model is the most widely used because of its applicability to wide-ranging flow problems and its lower computational demand than more complex models that are available. In this model, it is assumed that the turbulence viscosity (ν_t), is only a function of k and ε, where k is the turbulent kinetic energy, and ε is the turbulent dissipation. Flowchart of the solution method based on input and output

Table 1 Specification of CRAC units

No of CRAC unit	Power(kW)	Flow rate($\frac{\mathrm{m}^3}{\mathrm{s}}$)	Outlet air temperature (°C)
ACU01	40	3.6	22
ACU02	40	3.6	22.4
ACU03	40	3.6	22.7
ACU04	12.5	1.2	22
ACU05	12.5	1.2	25
ACU06	12.5	1.2	29
ACU07	12.5	1.2	31
ACU08	12.5	1.2	22
ACU09	12.5	1.2	24

Table 2 Data center-CFD model boundary conditions

Boundary condition	Value	Description
Supply flow rate	18 $\frac{\mathrm{m}^3}{\mathrm{s}}$	According to Table 1
Servers flow rate	15.3 $\frac{\mathrm{m}^3}{\mathrm{s}}$	Calculated from the library of 6SigmaDCX
Servers heat generation	42.2 kW	Sum of power of the per server
Tile porosity	50%	-
Rack porosity	35%	-
Blower flow rate	1.25 $\frac{\mathrm{m}^3}{\mathrm{s}}$	Air vent on the east wall

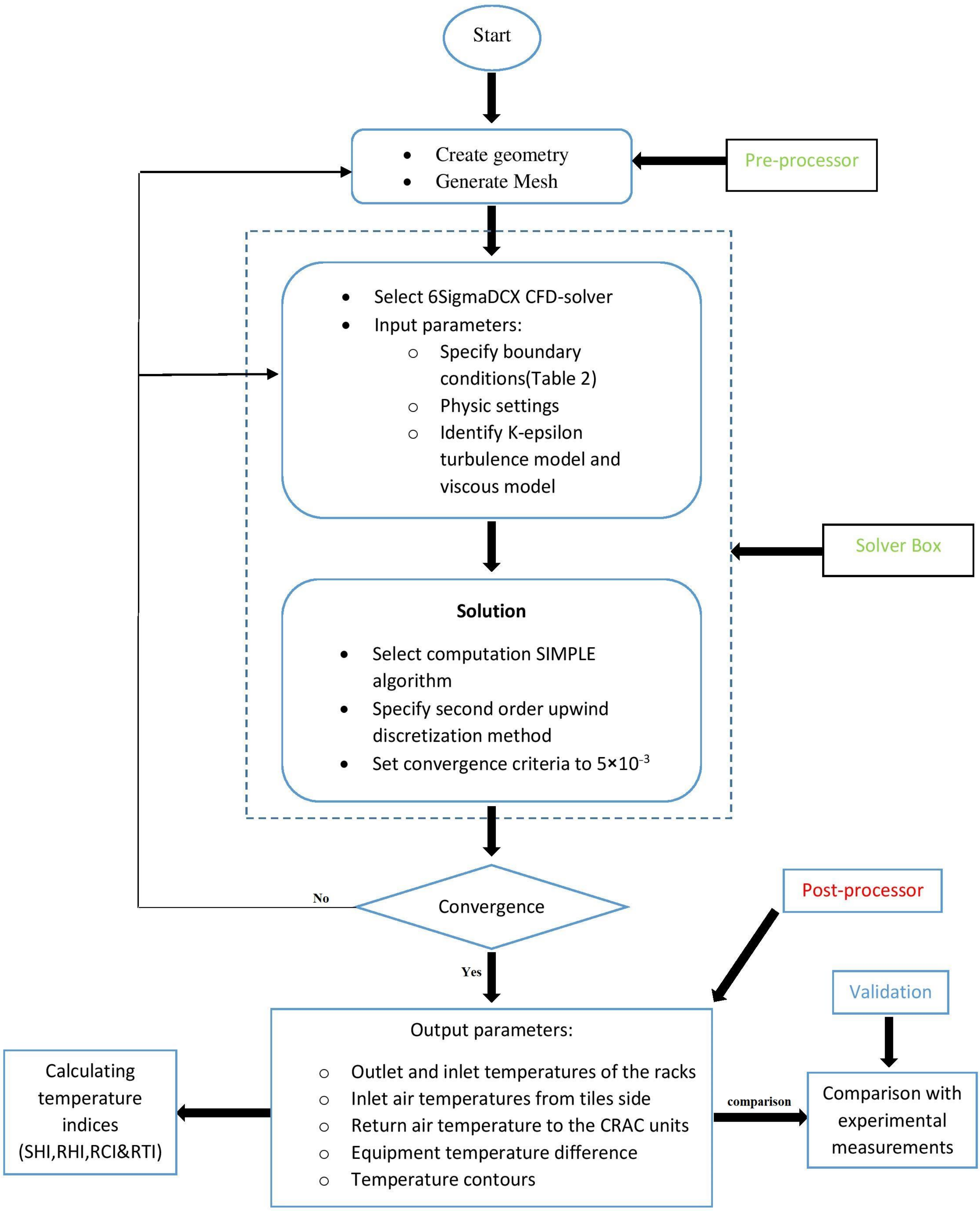

Fig. 8 . Flowchart of the solution method

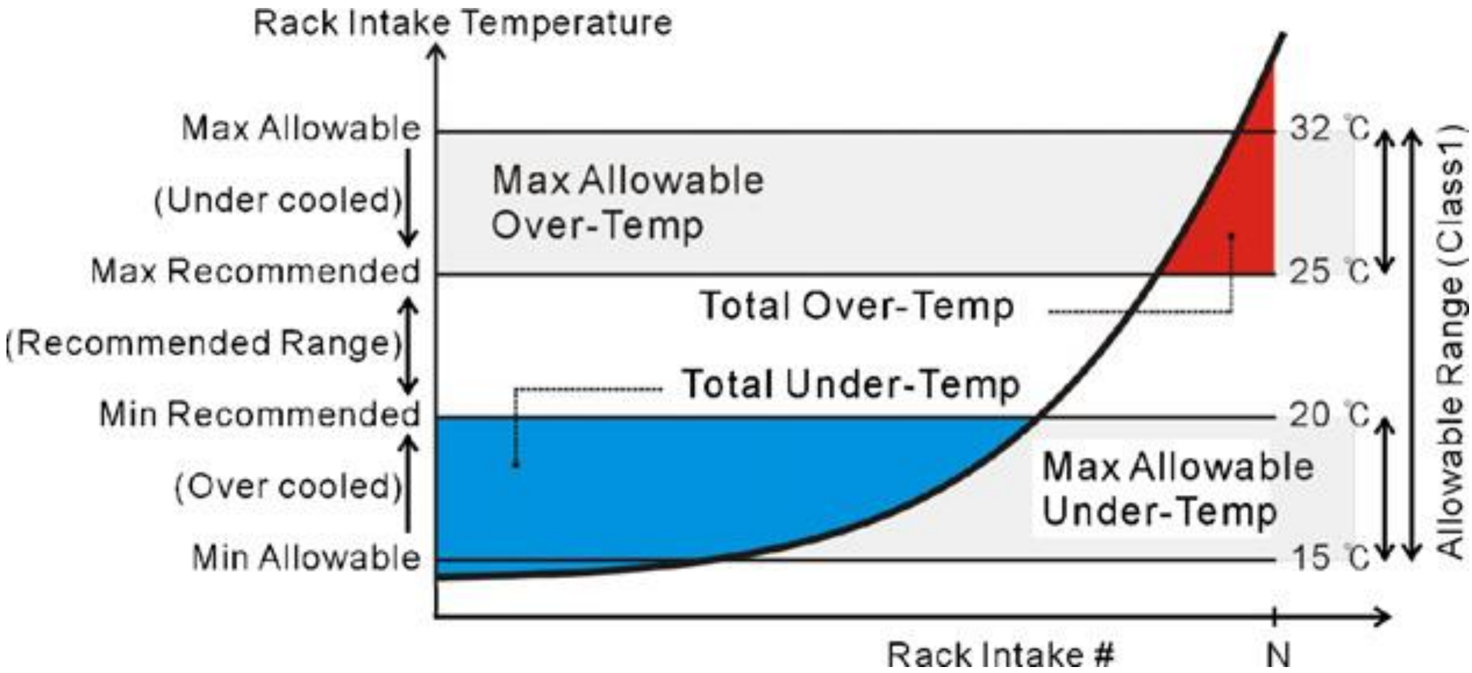

Fig. 9. Temperature ranges according to ASHRAE standard [15]

Table 3. Independency analysis of the grid of temperature indices

Cell Number	*RTI*	*RCI-High*	*RCI-Low*	*SHI*	*RHI*
1682597	73.4	83.5	100	0.502	0.325
2043630	76	80.4	100	0.563	0.422
3998421	76.8	81	100	0.551	0.431
4392839	76.56	81.05	100	0.568	0.428

parameters and post-processing of results are expressed in Fig. 8.

3.2. Performance Parameters Indices

The use of temperature indices helps to understand the efficiency and performance of air distribution. A complete distribution of the *RCI* has been performed by Herrlin [14]. This is a benchmark index that shows the intake temperature to the rack and it is very important for the continuous operation of the data center [14]. Fig. 9 shows the allowable and recommended range for rack intake temperature for the design conditions. Overheating temperature conditions occur when one or more intake temperatures are greater than the maximum recommended temperature. Similarly, under temperature conditions occur when one or more intake temperatures drop below the minimum recommended temperature. The *RCI* index contains two relationships RCI_{High} and RCI_{Low} to measure the difference between overheating temperatures and the difference in lower temperatures, respectively. They are defined in the following relationships:

$$RCI_{HIGH} = 100\times\left[1-\frac{\text{Total overtemprature}}{\text{Max Allowable Overtemprature}}\right] = 100\times\left[1-\frac{\sum_{1}^{n}(T_i - T_{max-rec})_{T_i<T_{max-rec}}}{n\times(T_{max-all}-T_{max-rec})}\right] \tag{6}$$

$$RCI_{low} = 100\times[1-\frac{\text{Total Undertemptarure}}{\text{Max Allowable Undertemprature}}] = 100\times[1-\frac{\sum_{1}^{n}(T_{min-rec}-T_i)_{T_i<T_{min-rec}}}{n\times(T_{min-rec}-T_{min-all})}] \tag{7}$$

where RCI_{High} is 100%, indicates that there are no overheating temperatures and all racks are cooled according to the standard. The lower percentage indicates the probability that racks will experience temperatures above the maximum recommended. A value of RCI_{High} in the range of 91%-96% is an acceptable range for operation [15]. One of the ways to improve this index is to increase the supply airflow rate and lower the supply airflow temperature, which is also due to an increase in energy consumption [16].

RTI is a measure of the energy consumption performance of the air management system. This index is defined as Eq.(8) [17]:

$$RTI = \left[1 - \frac{T_{return} - T_{supply}}{\Delta T_{equipment}}\right] \times 100 \tag{8}$$

The *RTI* is the ratio of total airflow through the CRACs to the total airflow through the IT-equipment. A value of *RTI* above 100% shows hot air re-circulation and the rising in racks intake temperatures. Also, values are less than 100% indicate the cold air by-pass and thus reducing the return temperature to CRACs. Therefore, the optimum value for *RTI* is 100%. With closing the aisle enclosure, this index is optimized.

SHI is the ratio of heat gained by air before entering the racks and heat gain by air leaving rack exhausted. By considering the mass flow rates at the inlet and outlet of each rack which is equal, *SHI* can be written as the ratio of enthalpy rise due to infiltration in cold aisle to total enthalpy rise at the rack exhaust. *RHI* is considered as a complement to *SHI* (*SHI* + *RHI* =1), where the *RHI* is defined as the ratio between total heat extractions by the CRAC units to the total enthalpy rise at the rack exhaust. These indexes defined by equations as follow:

$$SHI = \frac{\delta Q}{Q + \delta Q} \tag{9}$$

$$RHI = \frac{Q}{Q + \delta Q} \tag{10}$$

where,

$$Q = \sum_i^n \sum_j^n m_{i,j}^r c_p [(T_{out}^r)_{i,j} - (T_{in}^r)_{i,j}] \tag{11}$$

$$\delta Q = \sum_f^M \sum_f^M m_{i,j}^r c_p \left[\left(T_{in}^r\right)_{i,j} - T_{ref}\right] \tag{12}$$

where Q is total heat dissipation from components and δQ is the rise in enthalpy of cold air before entering the racks. In Eqs. (9) and (10), $m_{i,j}^r$ is the mass flow of air through the *i*th rack in the *j*th row of the racks, $(T_{out}^r)_{i,j}$ and $(T_{in}^r)_{i,j}$ are the average outlet and inlet temperature from the *i*th rack in the *j*th row of the racks. T_{ref} is the inlet air temperature from the tile side [18]. If *SHI* is high, it indicates that the intake temperature is high which may be due to the re-circulation or mixing of the air before it enters the rack. A low value of *RHI* is an indication of rack exhaust hot air mixing with cold air inside the cold aisle. The ideal optimum value of *SHI* and *RHI* are 0 and 1 respectively, but practical

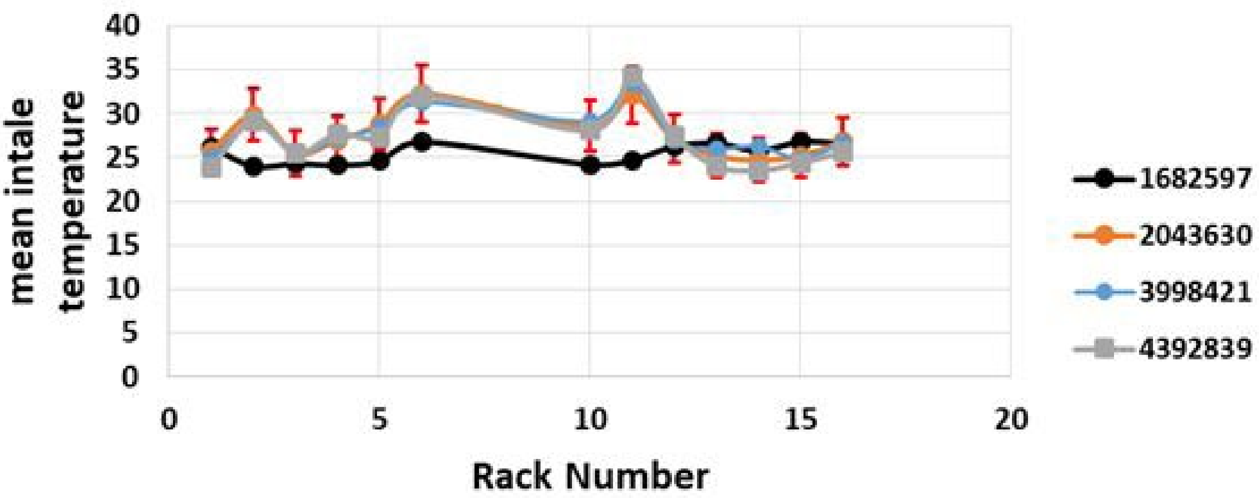

Fig. 10. Mesh analysis of the mean intake temperature of racks

Fig. 11. Testo temperature measurement device

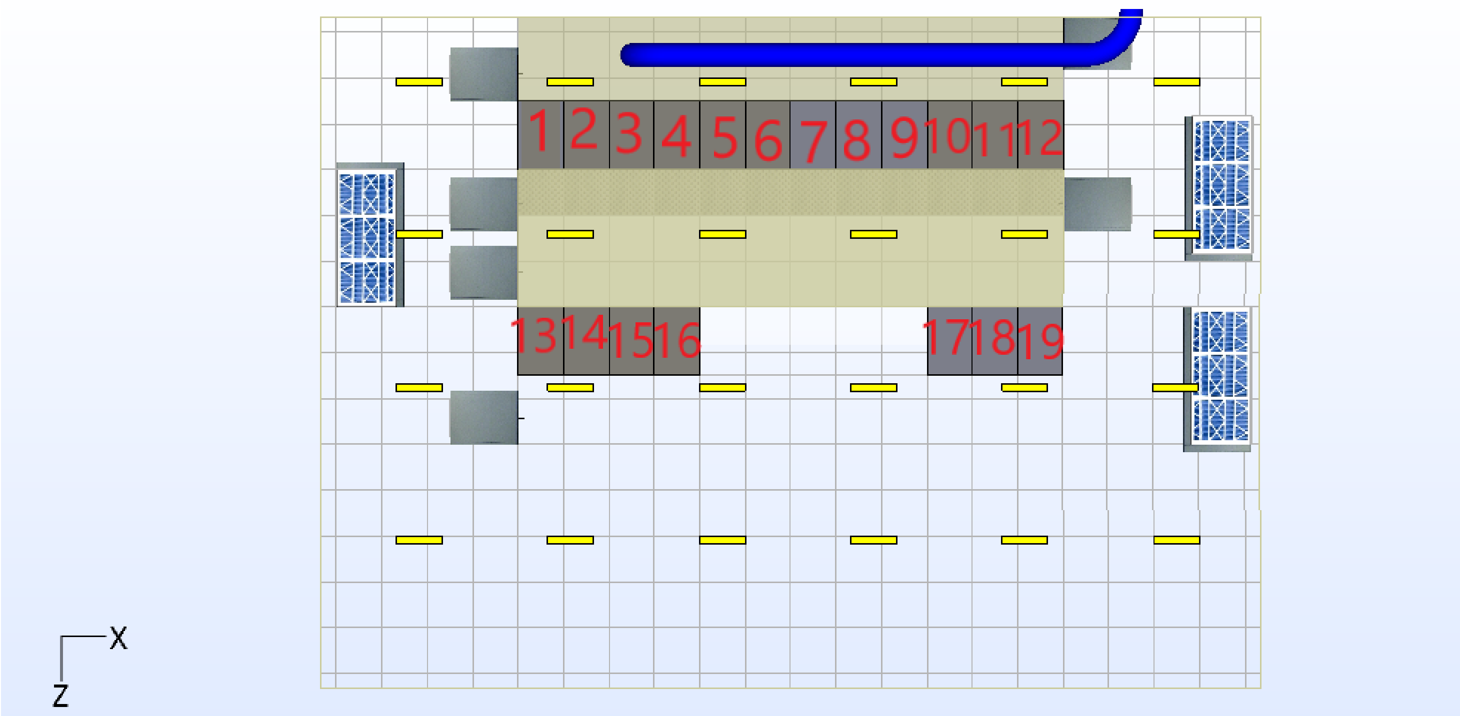

Fig. 12. Numbering of racks

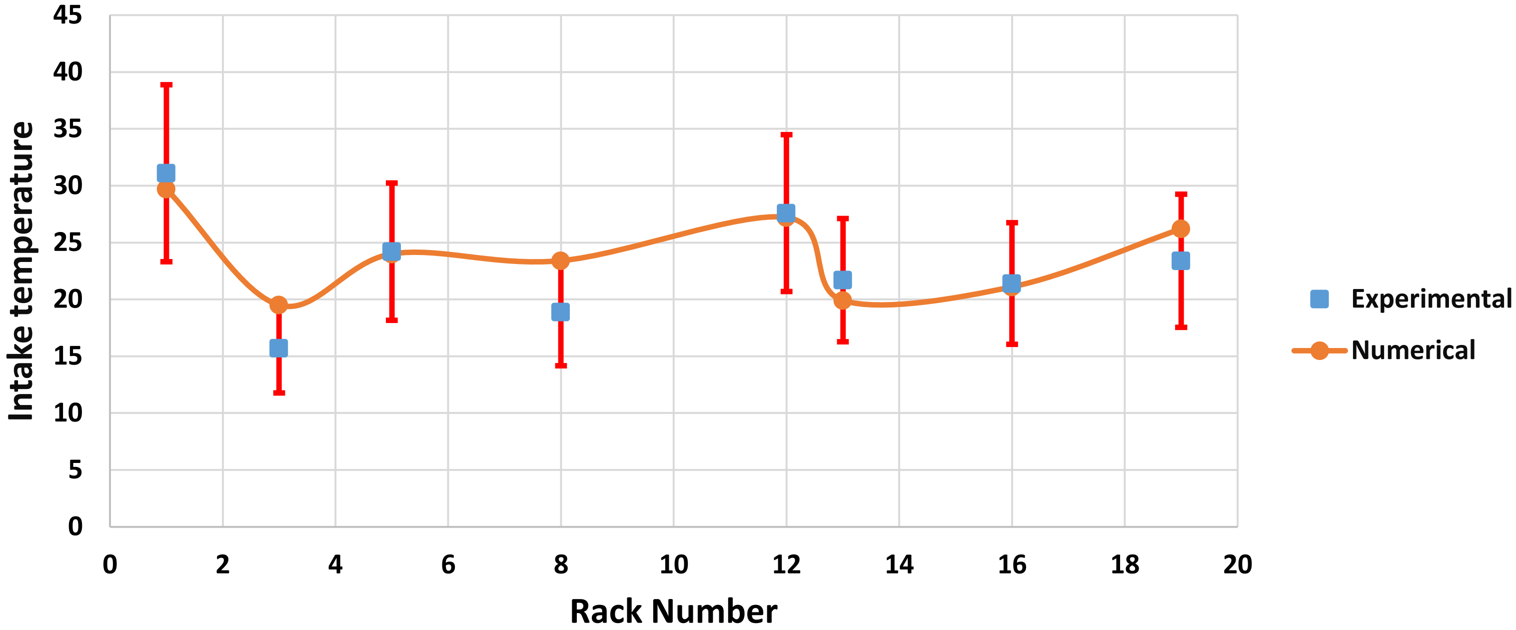

Fig. 13. Comparison of temperature measurement and numerical of intake temperature at the height of 1.8 m

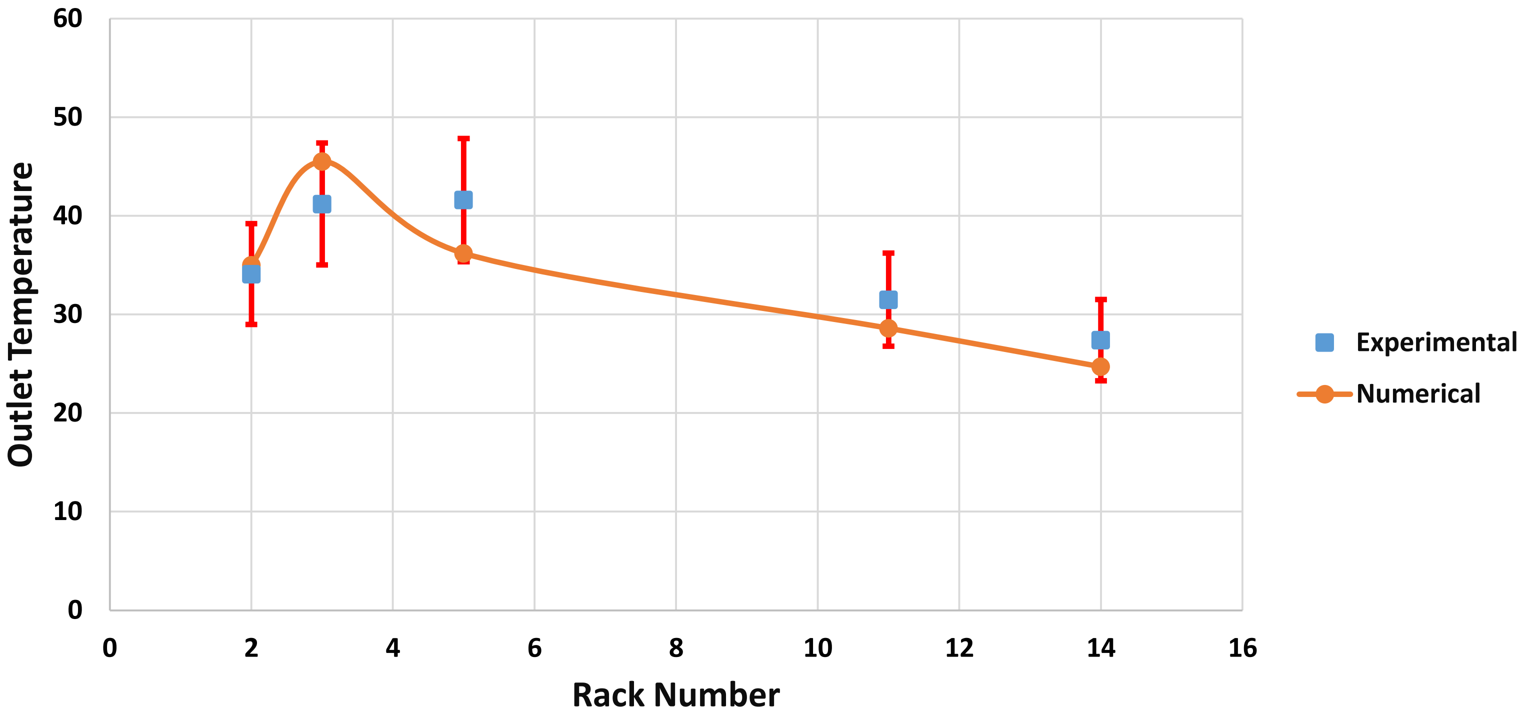

Fig. 14. Comparison of experimental and numerical of outlet temperature at the height of 1.8 m

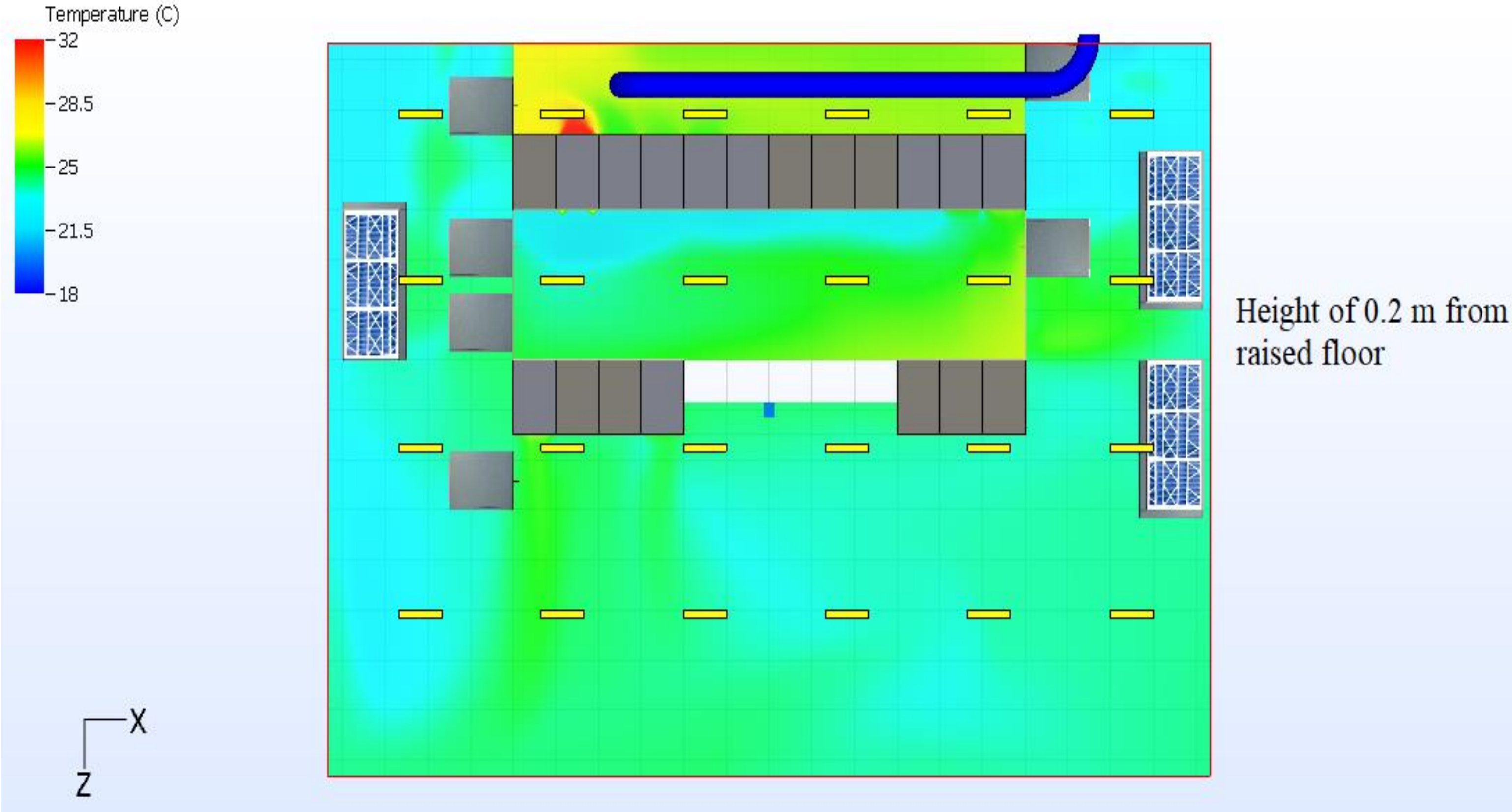

Fig. 15. Temperature contours at a height of 0.2 m

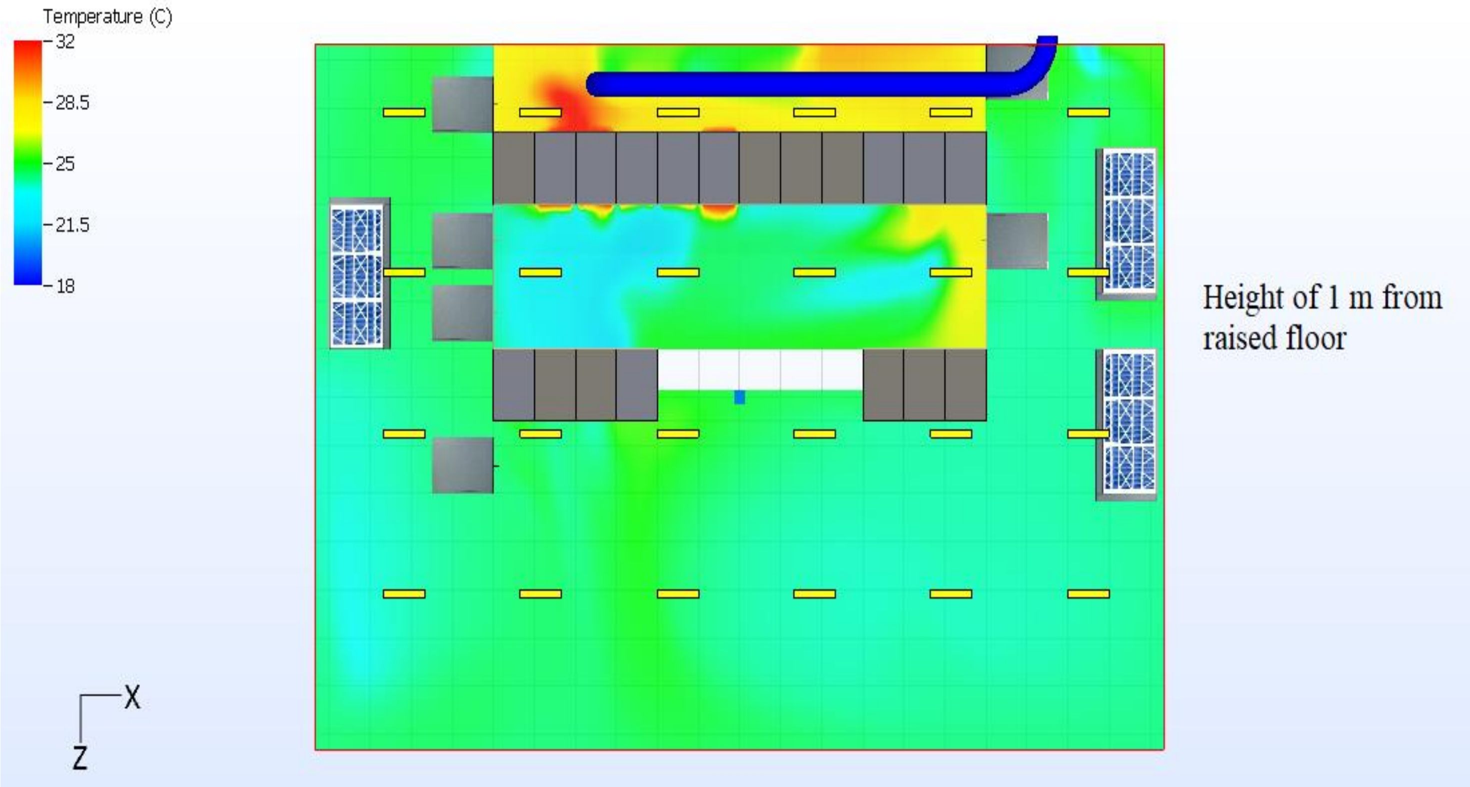

Fig. 16. Temperature contours at a height of 1 m

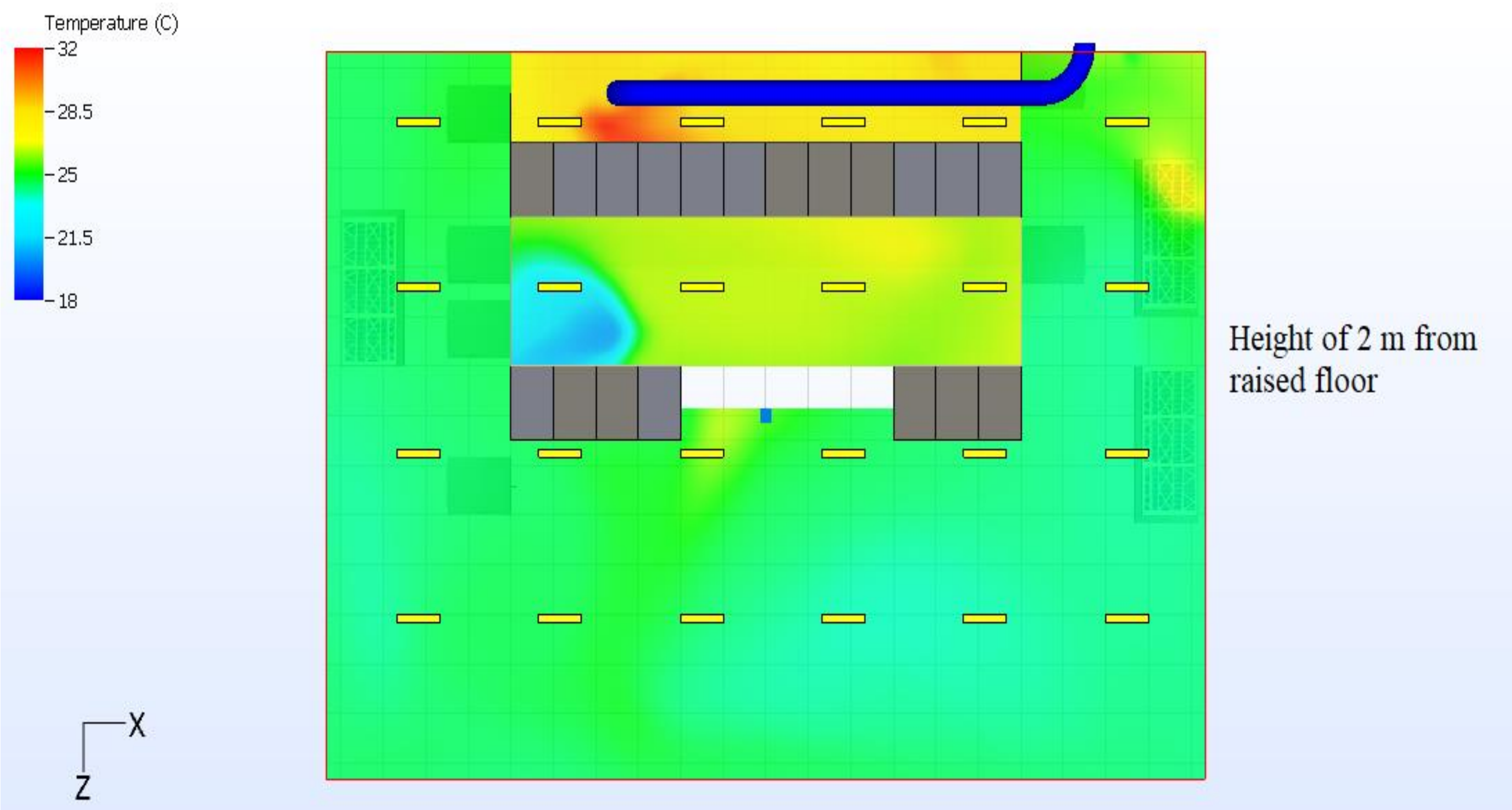

Fig. 17. Temperature contours at a height of 2 m

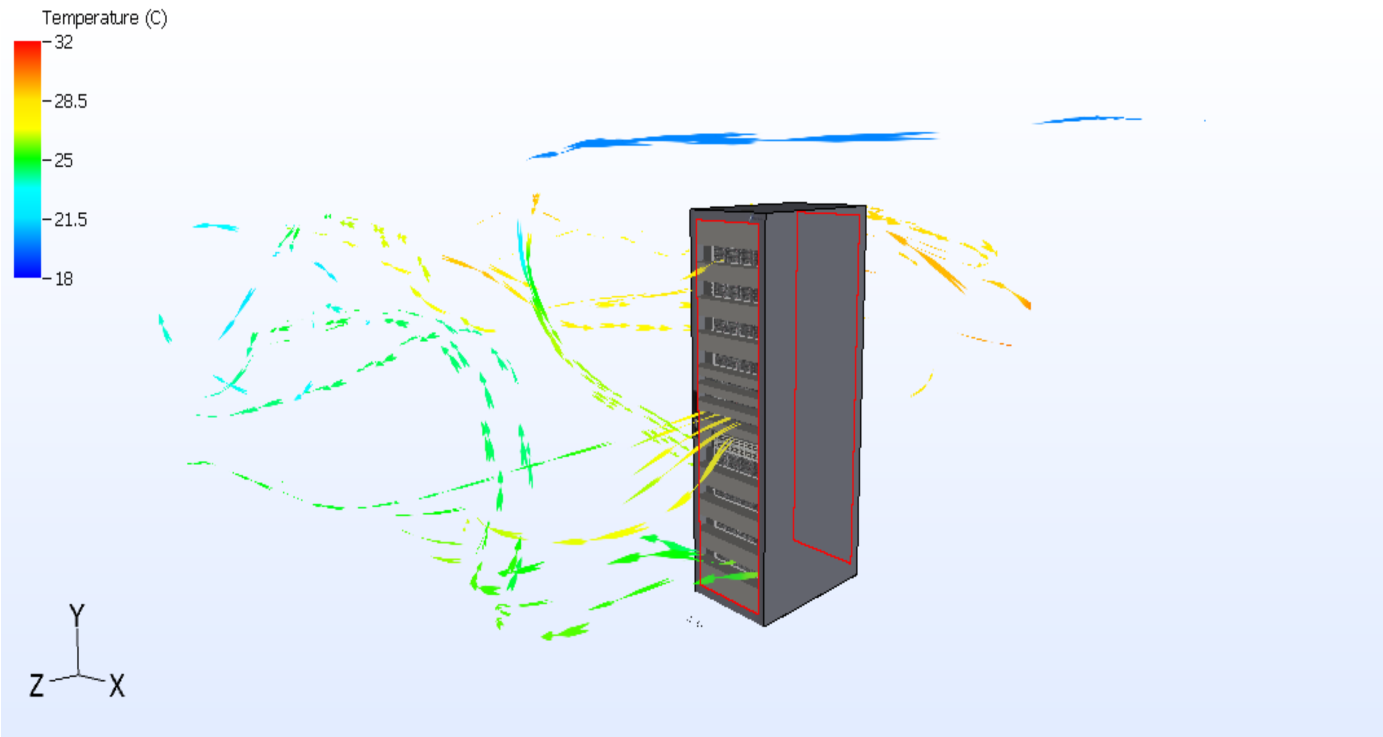

Fig. 18. Velocity contours for rack number 2

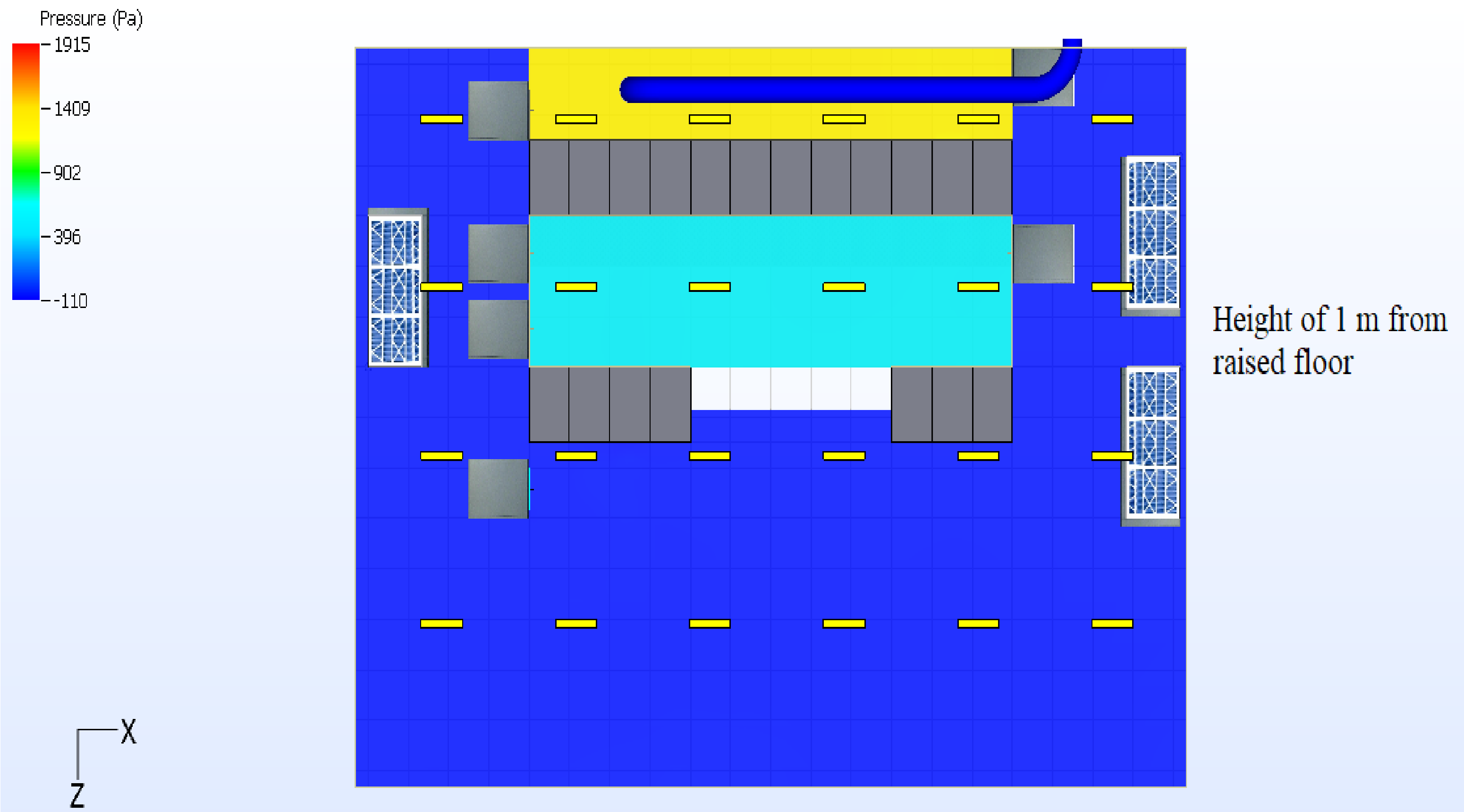

Fig. 19. Pressure contour at a height of 1 m

values are 0.2 and 0.8 [18].

4. GRID INDEPENDENCY STUDY AND MODEL VALIDATION

Grid refinement study is to achieve a minimum size that does not affect the numerical results and has an independent mesh size. Grid independency solutions are modeled in four different mesh sizes and the results of dimensionless temperature indices are illustrated in Table 3 that shows after the cell number of 1682597 there are very few variations between the results. So, in Fig. 10 the mean intake temperature per rack for the cell number 1682597, 2043630, 3998421, and 4392839 are compared where error bars are less than 10% for the cell number of 2043630, 3998421, and 4392839. According to these results, the cell number of 2043630 with a mesh size of 0.2 meters is sufficient to continue the calculations.

To validate the present numerical model, temperature contours obtained from the numerical simulation were analyzed and compared with the values obtained from temperature measurements with Testo 605-H1 digital thermometer (Fig. 11). The racks are numbered as shown in Fig. 12 and the intake and outlet temperatures of racks at the height of 1.8 m from the raised floor that measured with a thermometer are compared with the numerical simulation results. These results are illustrated in Fig. 13 and Fig. 14, as is seen the good agreement between the numerical solution and experimental measurements exists with a minimum and a maximum error of 0.5% up to 14.9% respectively.

5. Results and Discussion

Numerical simulations were conducted for the present model of AUT's high-speed processing center, then for enhancement, the thermal management of data center, cold aisle enclosure with decreasing supply airflow temperature, and hot aisle enclosure with return air from the plenum ceiling is used. In each case, airflow and temperature distributions in the data center are obtained and analyzed. In the following sections, to improve the cooling and airflow performance, the data center performance index matrices that are calculated from the temperature contour results, are compared and discussed between cases. In the end, the more suitable case for the AUT's data center has been introduced.

5.1. Results Of Numerical Solution

The issue has been solved with the help of 6SigmaDCX software to the boundary and initial conditions which were mentioned in section 2. Contours of temperature at the height of 0.2 m, 1m, and 2 m from the raised floor are shown in Fig. 15, Fig. 16, and Fig. 17, respectively.

According to Fig. 16, the recirculation of hot air by 2U-Berbon supermicro1400W(4blade) servers to the cold aisle enclosure occurs at the height of 1 m from the raised floor in racks of the first row numbered 2, 3, 6, 11, and 12. The

Table 4. Room performance indices from the numerical solution

SHI	*RHI*	*RCI-Low*	*RCI-High*	*RTI*
0.568	0.422	100	81.05	76.56

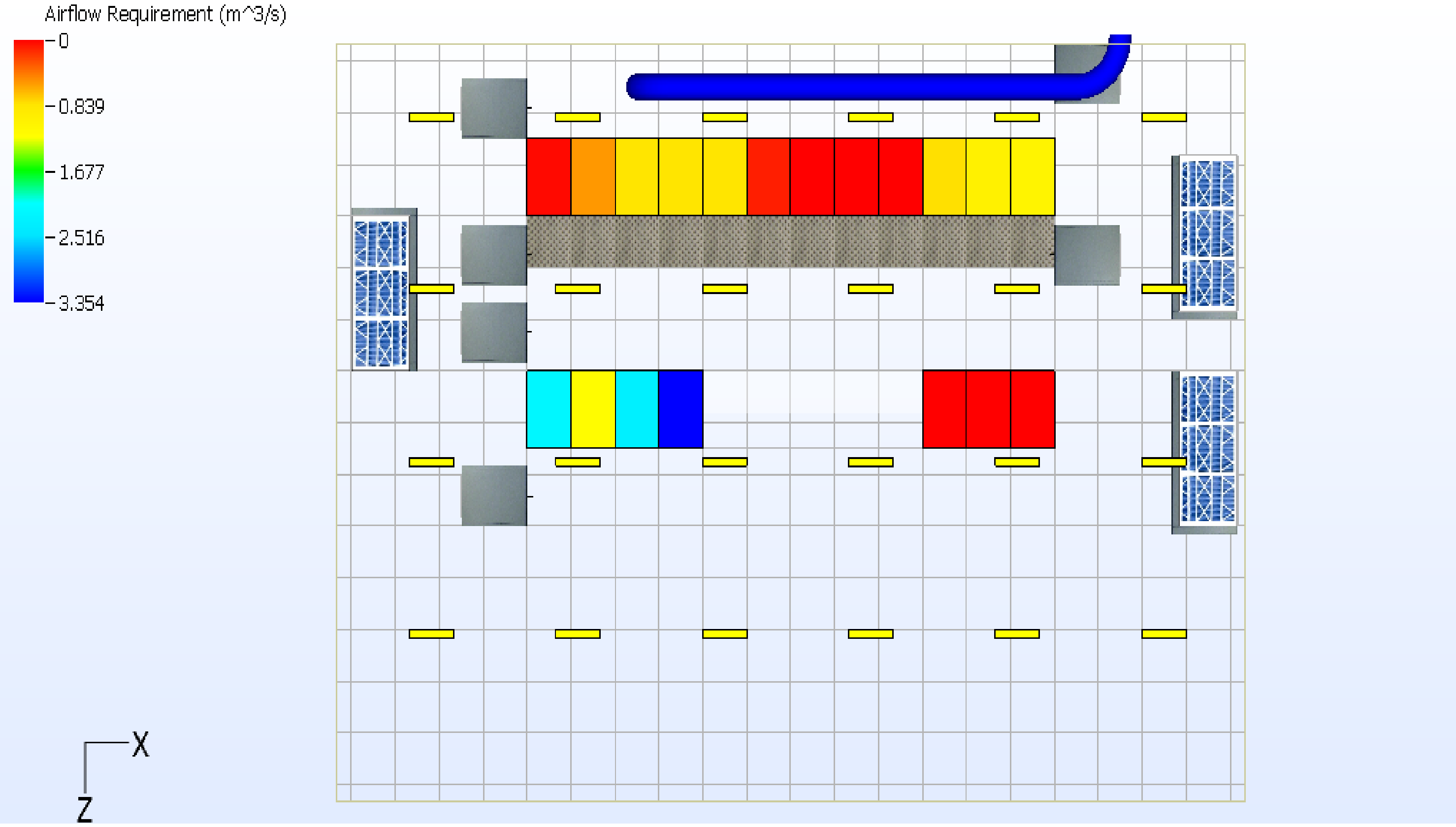

Fig. 20. Airflow requirement for each rack

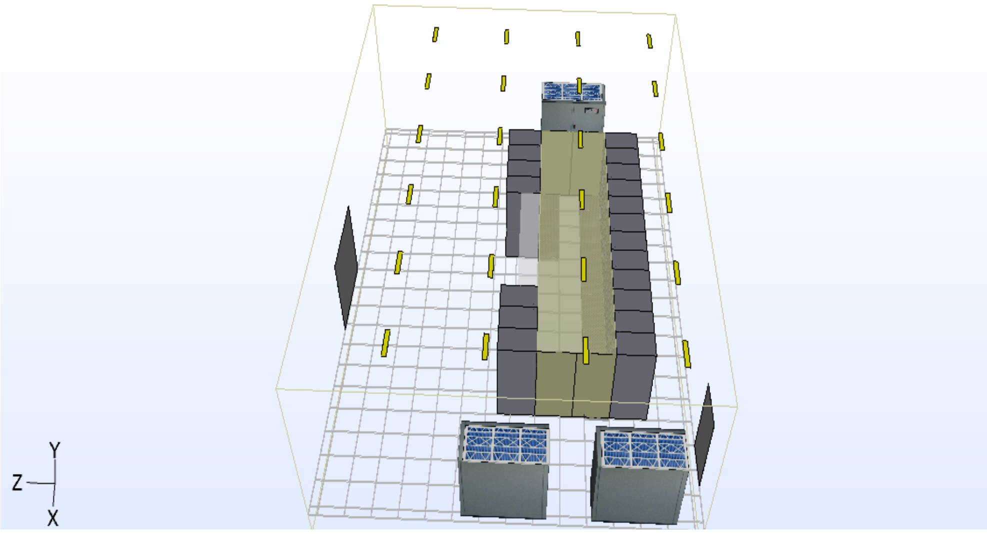

Fig. 21. Cold aisle enclosure model

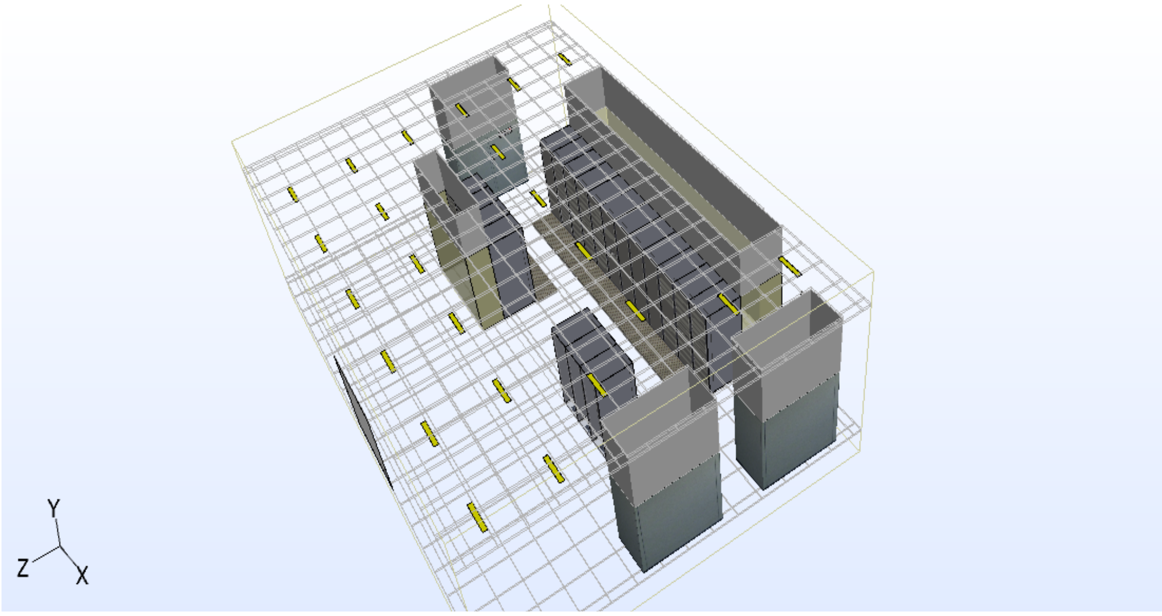

Fig. 22. Hot aisle enclosure with return airflow from ceiling plenum

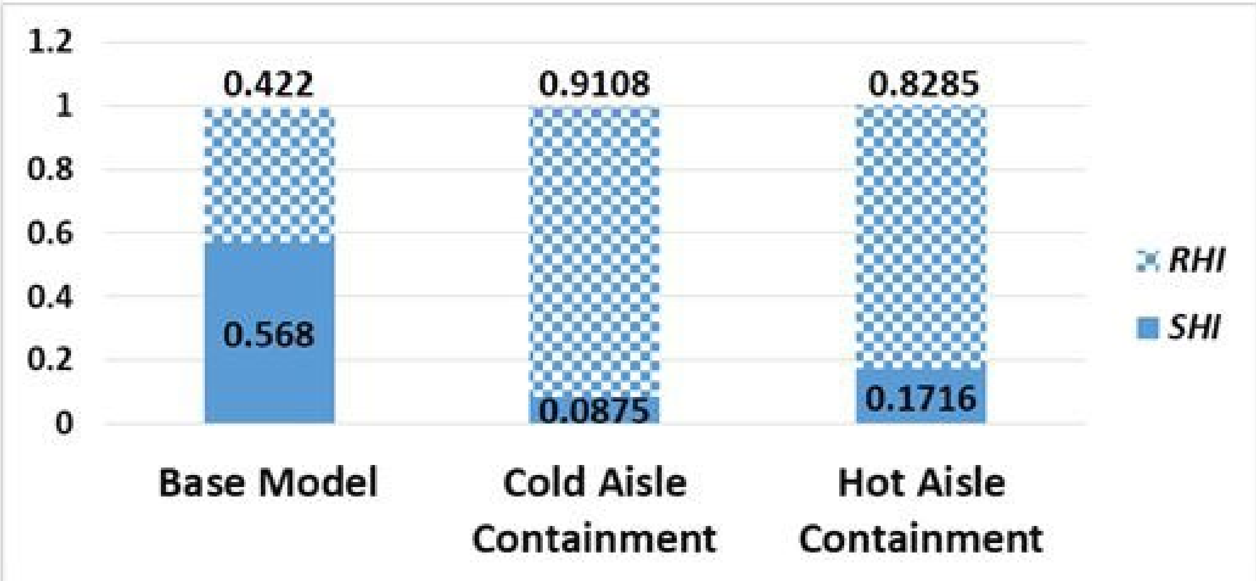

Fig. 23. Comparison of SHI and RHI for the models

recirculation of hot air for rack number 2 is well illustrated with velocity contours in Fig. 18.

The pressure contour is shown in Fig. 19 at the height of 1 m from the raised floor. It is observed that the pressure on the side of the hot aisle enclosure is higher than the pressure on the side of the cold aisle enclosure or the input of the racks, which causes a negative pressure difference on the output of 30 devices of the first row 2U-Berbon super micro servers from the racks and the recirculation of hot air according to Fig. 16.

The data center performance parameter indices are calculated automatically with software (Table 4). The high

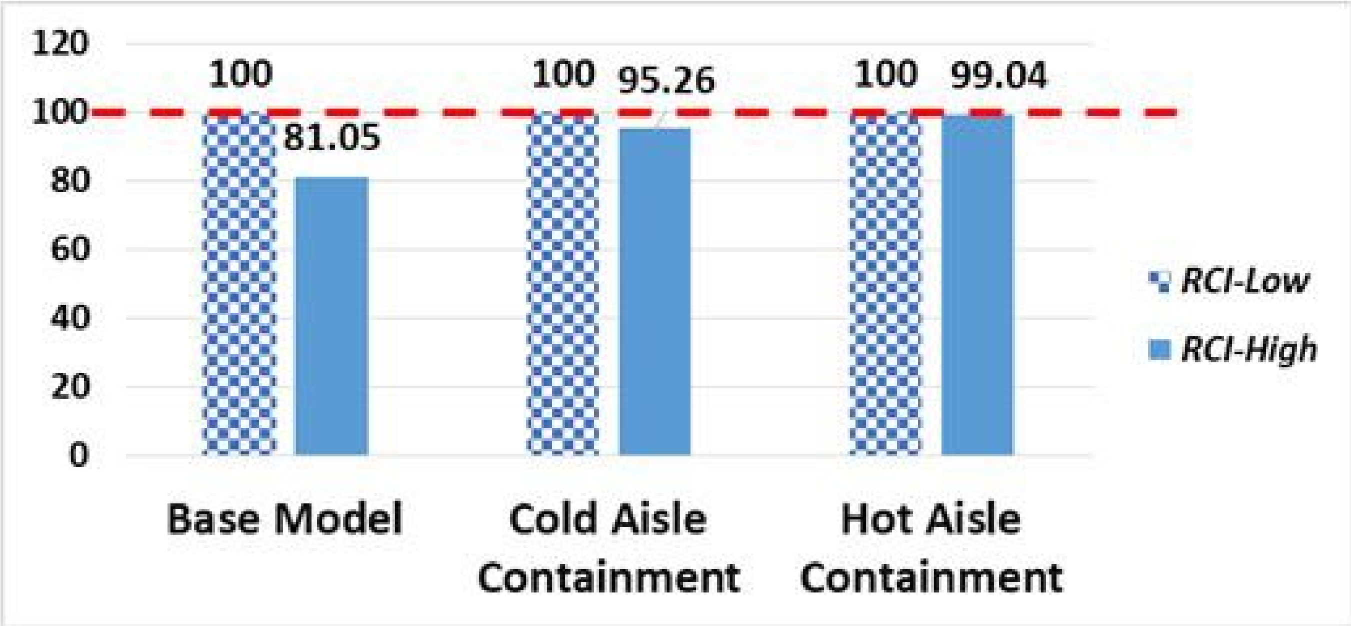

Fig. 24. Comparison of RCI for the models

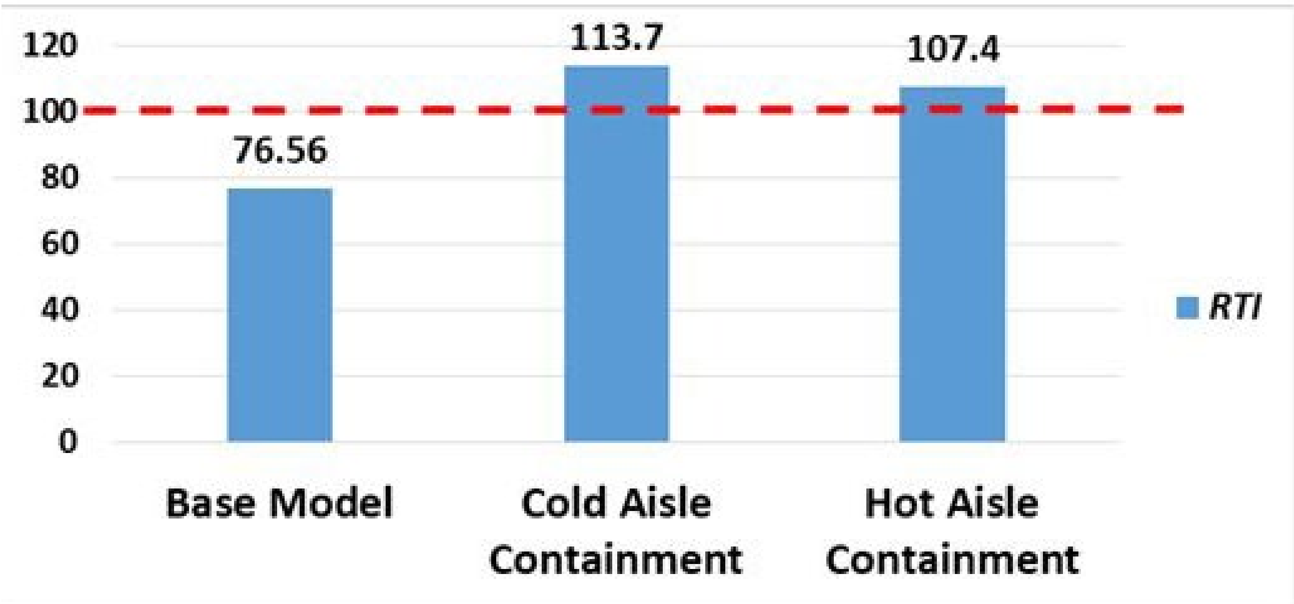

Fig. 25. Comparison of RTI for the models

value of *SHI* (0.568) compared to the optimum value (zero) indicates that the high intake temperature to the racks. Also, *RHI* is below the optimum value of 1, indicating the mixing of hot air from the output of the rack with cold air within the cold aisle. The value of 100 *RCI-Low* indicates that there is no lower temperature than the permissible limit, however, a low percentage of *RCI-High* of the 100 percent, increases the possibility of the racks that experience high intake temperature than the recommended temperature. The low *RTI* from 100, means by-pass cold air.

5.2. Enhancement Of Data Center Thermal Management Using Cold Aisle Enclosure

Fig. 20 shows the amount of air required for each cabinet. Considering that the volume of air required to cool the second-row rack servers is more than the other racks, perforated tiles will be provided in front of the racks 13, 14, 15, and 16 volumes of required airflow in cold aisle enclosure model.

In the cold aisle enclosure model (Fig. 21), to improve the *RCI-High* index and reduce the intake temperature to servers within the standard range of ASHRAE, CRACs that can supply air at a temperature of 18 ^{o}C is replaced and floor standing air conditioners have been removed from the data center. These air conditioner units were added to increase the amount of supply air to improve the *RCI-High* index of the racks in the data center, but this index did not reach its optimum value.

5.3. Enhancement Of Data Center Thermal Management Using Hot Aisle Containment With Return Flow From Ceiling Plenum

To improve the *RTI* and *RCI-High* indexes to reduce the re-circulation airflow and intake temperature to the racks, the data center simulated once more using hot aisle enclosure with return airflow from ceiling plenum with the height of 0.5 m (see Fig. 22). In this case, instead of ventilating hot airflow from the hot enclosure by the blower to the outside, by creating a ceiling plenum and ducting from hot aisle enclosure to the CRAC units return values, hot air returns to the air conditioners. For this model, the same specification of the CRAC units is used as the primary and base model of the data center. Also, in this case, the lights are installed on the ceiling plenum instead of the room ceiling.

5.4. Compare Results For Different Configuration Models Of The Data Center

In this section, the primary and base model of Amirkabir University's server room is compared with the cold aisle enclosure and hot aisle enclosure models. In Figures Fig. 23, up to Fig. 25, the temperature indices *SHI*, *RHI*, *RCI*, and *RTI* for each of the three models are compared with the optimum value of the indexes. Fig. 23 shows a stacked column chart for *SHI* and *RHI*. According to this figure, the cold aisle enclosure and the hot aisle enclosure are the most suitable models in terms of the Supply Heat Index and Return Heat Index. According to Fig. 24, the *RCI-High* index for cold aisle enclosure and hot aisle containment models are 14% and 18% higher than the base model of the server room. Since the reference temperature of the air conditioners is not lower than the minimum recommended temperature of the ASHRAE (18 ^{0}C), the *RCI-Low* index is usually in its optimal value for all three models, which is 100%.

According to Fig. 25, the *RTI* index is approximated to the optimal value for cold aisle enclosure and hot aisle enclosure models compared to the base model. Also, in the cold aisle enclosure model, the *RTI* index has been improved by 6% to reduce the re-circulation of hot air in the data center.

6. CONCLUSIONS

To consider the design performance of Amirkabir University's high-speed data center, after selecting an appropriate cooling system, attentions to the configuration of a server room in the initial design are necessary. In this research, different parameters that affect the design of data center with air distribution system from the raised floor have been changed and the temperature indices of them have been compared to each other, the results are as follows:

1) Different parameters for evaluating air distribution and thermal management in the data center used in this study were: cold aisle enclosure and hot aisle enclosure with venting hot air to the environment, cold aisle enclosure with low reference temperature, and hot aisle enclosure with return flow from the ceiling plenum.
2) General indices for better air distribution performance in data centers are SHI, RHI, RCI, and *RTI*. The most important index that is first reviewed is the *RCI-High* index and if its value is within the acceptable range, other indices are reviewed. The placement of other indices in an acceptable range affects the overall performance of the data center.
3) Performance enhancement solutions for the *RCI-high* index are an increase in the amount of air supplying that applied to the base model of the data center. Also, the *RCI-High* index in the main model is 81.05%, which is far from the optimal value. Therefore, the base and the current model of the data center are not suitable for cooling servers.
4) To improve the cooling of the servers in the cold aisle enclosure model, the intake temperature has been reduced to 18 ^{o}C. In this model, the *RCI-High* index is 95.26% and it is within the acceptable range, but disadvantages of the method are increased energy consumption and increased costs due to the use of air conditioners that can be supply air with a temperature of 18 ^{o}C.
5) According to Fig. 24 and comparison of the temperature indices, the hot aisle enclosure with return flow from the ceiling plenum will be appropriate for improving the air distribution and thermal performance of Amirkabir University's high-speed data center. The only problem with this model is the low height of the server room, which is about half a meter necessary to create a ceiling plenum. But, Cho et al. [19] showed that the ceiling plenum height for hot air returns does not affect the flow of air if there is no problem in the installation of electrical cables.

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