CFD MODELING OF THERMAL DISTRIBUTION IN INDUSTRIAL SERVER CENTERS FOR CONFIGURATION OPTIMISATION AND CONTROL

Pierre-Luc Paradis[a,c], Drishtysingh Ramdenee[a], Adrian Ilinca[a], Hussein Ibrahim[a], Abderrazak El-Ouafi [a], Jean-Sébastien Deschênes[a], Alexandre Boudreau [a] Daniel Rousse [b]

[a] Université du Québec à Rimouski, Canada, G5L 3A1
[b] École de technologie supérieure, Canada, H3C 1K3

[c]pierre_lucparadis@hotmail.com

ABSTRACT
Industries and institutions are more and more prone to be equipped with servers to control all required computational and communication tasks. In many cases where the computational requirements of tasks or where data management are enormous, the need for proper configuration of the server cabinets is important to optimize cooling mechanisms and to improve energy efficiency. Specialized software are presently available to model new server centers with optimized configuration. However, there are no clear guidelines in the literature on the best way to build datacenter and on how you could increase efficiency of existing server center. In this paper, we present a Computational Fluid Dynamics (CFD) model made use to evaluate different solutions of cooling in data center and to write guidelines to help owners to improve efficiency of their buildings. Then a look is taken to control strategy that could be develop with those simulations.

Keywords: Flow Simulation, SolidWorks, Data center, Cooling optimization.

1. INTRODUCTION

Telecommunication services mainly those involving IP internet protocol have known enormous growth over the last few years and, thus, servers with enormous computational capacity have become very common. With them, challenges with regard to optimized cooling configuration, have surfaced. These mainly target already existing server centers as newly built ones are designed with specialized software using the cooling criteria as an intrinsic design parameter. Methods involving positioning of the server cabinets according to hot/cold corridor logic and their confinements at different levels exist and can lead to significant energy savings. However, in many cases, the non-uniformity of air inlet and outlet positioning and sizes bring additional challenges. Hence, in an attempt to advice upon proper ventilation to minimize air conditioning cost, a numerical model based on SolidWorks Flow Simulation CFD computational engine has been developed to characterize different situations and scenarios and then provide conclusions about the cooling mechanism in a server center. Furthermore, the model has been made generic as possible to be applicable to a large number of data centers. This paper, first, presents a literature review of this field and then focuses on the characteristics of the model. Following the model presentation, the article then ponders on the validation of the numerical model on a test bench case followed by results and conclusions obtained for an industrial server center case study. The article concludes by providing recommendations for optimization of the server arrangement and cooling mechanisms as well as an insight of future development in predictive control. A 2D lumped model is built for a simplified server centers and a state model is built in a temperature predictive algorithm control.

2. LITERATURE REVIEW AND CHOICE OF SOFTWARE

In order to better understand the intrinsic engineering analysis that are required into thermal modeling of server centers, the intrinsic mechanism of server centers and related cooling systems and to analyse improvement possibilities with regard to existing tools, a literature review has been performed. One of the essential conclusions of our research has been that simulation tools should be used for positioning optimisation (of server cabinets and cooling systems) of to be constructed data centers. In other words, the tools inscribe themselves as an architectural tool into the development of new datacenter. However, we need to develop a tool to provide guidelines in improving energy efficiency of already existing server centers where different constraints need to be respected. When it comes to modeling of distinct isolated thermal energy generating electronic devices, there have been recent developments in the availability of thermal modeling tools for architectural studies in the academic/research community. One such tool is HotSpot (K. Skadron and al, 2003) for microprocessors, which models temperature using thermal resistances and capacitances derived from the layout of micro-architectural structures that has been validated using finite element simulations. Rather than detailed thermal simulators for processors, quick estimation using convective energy dissipation techniques are used after calculating a processor's energy consumption using event counters in (F. Bellosa and al, 2003; A. Weissel and F. Bellosa, 2004). Such estimation has been used for developing temperature aware scheduling (A. Weissel and F. Bellosa, 2004). There have also been thermal modeling studies for individual disks (S. Gurumurthi, A. Sivasubramaniam, and V. Natarajan, 2005) and disk arrays (R. Huang and D. Chung, 2002), with the former providing a tool which also integrates with a disk performance simulator for architectural studies. These tools, which allow integrated performance and power/thermal studies, have been facilitating research contributions (K. Skadron and al, 2003; D. Brooks and M. Martonosi, 2001; M. Gomaa, M. D. Powel and T. N. Vijaykumar, 2004; L. Shang and al, 2004; S. Gurumurthi and al, 2003; Y. Kim and al, 2006; E. Pinheiro and R. Bianchini, 2004) in the architecture community for reducing power/temperature. All these tools are useful when studying and optimizing individual components. However, in this paper, we are interested in studying complete data center systems, where there could be interactions between different components. A recent tool (T. Heath and al, 2006) proposes using simple equations to calculate temperatures at very specific points in the server center system. While this approach suffices for certain simple "what-if" questions as suggested in (T. Heath and al, 2006), a CFD based model is needed for a more holistic examination of the system under a wider spectrum of static (e.g.: where to place components, computer room air conditioning (CRAC) units) and dynamic (e.g.: how long before the temperature reaches a threshold upon fan failure? what thermal management technique provides the best recourse upon emergency?). Fluid flows need to be modeled accurately for figuring out where components need to be placed and understanding complete system interactions. Thermal modeling of data centers: The importance of cooling high density datacenters/machine-rooms has attracted considerable interest recently (N. Rasmussen; N. Rasmussen; R. K. Sharma, C. E. Bash and C. D. Patel, 2002; C. E. Bash, C. D. Patel and P. K. Sharma, 2003; P. Rodgers and V. Eveloy, 2003; J. Moore, J. Chase and P. Ranganathan, 2006). Most of these studies (J. F. Karlsson and B. Moshfegh, 2005; C. D. Patel and al, 2002; C. D. Patel and al, 2001; C. D. Patel and al, 2003; M. H. Beitelmal and C. D. Patel, 2004; C. D. Patel and A. J. Shah, 2005; S. V. Patankar and K. C. Karki, 2004) have looked at this problem from an engineering perspective of designing CRAC and other cooling systems, placement of racks in machine rooms, etc., with many of them using CFD models. For instance, (R. Sharma and al, 2005) points out that heat recirculation is a limiting factor in existing cooling systems and proposes using fan/coil unit in the ceiling above the rack. Impact of CRAC failures on static provisioning has also been studied using CFD models (C. D. Patel and A. J. Shah, 2005). From the computer science/systems perspective, researchers are starting to use CFD models for workload placement (J. Moore and al, 2002; J. Moore and al, 2005; J. Mooreand al, 2006) across racks of a machine room, and balancing the temperature across these racks (R. Sharma and al, 2005). The different studies supported our choice for a CFD code. The most widely used code in such cases is « FloVENT simulation software » as it comprises of a specialised module in server air conditioning modeling. However, this license is not available at Univesity of Quebec at Rimouski (UQAR) and the choice was made among Comsol multiphysics, Ansys CFX, Ansys Fluent and SolidWorks Flow Simulation. For simplicity and functionality reasons SolidWorks would have been the software of choice for geometry modeling. Furthermore, the aim of the project was to define a tool that allows trend characterisation to advice on cooling, heat sources and heat sinks positioning rather than high accuracy point to point temperature definition. For these reasons, SolidWorks Flow Simulation was chosen.

3. CFD MODEL

The aim of this work is to build a numerical model of a server center such we can evaluate the impact of modifying the air flow, the addition of confinements, etc. on the temperature distribution. The model was built on SolidWorks commercial software and respects the actual geometrical parameters of the real server center. The boundary conditions for a given simulation were also defined according to measured data in the real physical center. The computational domain was bounded by the walls of the actual data centre. The CFD model was based on previous calibration works that evaluated the relative performance of turbulence models, transition models, mesh type and size and the size of the computational domain. The size of our domain in our case is the size of the room. The SolidWorks automatic mesh generator was used. The mesh was successively refined from a completely coarse one and a convergence study was performed. The mesh size relating to relative error of less than 3% between two consecutive simulations was taken. This consisted of 250 000 tetrahedrons. The used turbulence model was a k-episilon one with fully turbulent transition.

4. CFD VALIDATION

The SolidWorks based numerical model's capacities need to be tested and validated in order to be able to assume advices made to the industry viable. For such, the main server room of UQAR was used as a test bench. The geometric model of the UQAR server room was modeled on SolidWorks. For a specific working regime (assumed stationary as the variability is quite small over a short time period), the power output of the different server center components, the inlets and outlets air temperature and flow (pressure velocity and area of exit) as well as the power rating of other different heat sources were measured and entered as boundary conditions in the validation model. The computational domain was the data center room in real size as the simulation was a bounded one and the flow simulation was run as steady. The model mesh size and turbulence models were all configured according to the previously described methodology. Figure 1 illustrates the temperature distribution obtained for a specific situation.

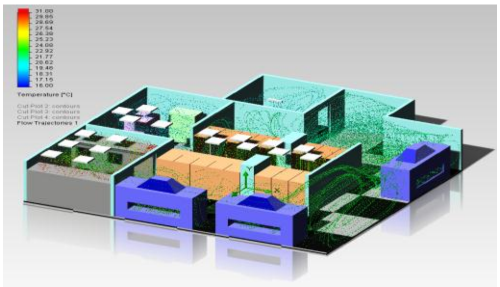

Figure 1: Results with Validation model

In this figure, the grey square represents the air inlet whereas the orange blocks represent the racks. The blue modules are the CRAC units. To validate the quality of the results, the data center was run at the same "regime" and the temperature in a corridor between the racks was measured on a central plane at middle altitude with respect to the two calculation columns lines. Table 1 illustrates the obtained simulated results compared to experimentally measured ones. It is to be noted that Windmate 200 instrument was used to measure both the volumetric flow rate and the air temperature. A windmate 200 is an instrument usually used in sea navigation to measure wind speed, wind direction, temperature, chill and other navigation pertinent parameters.

Table 1: Single Line Table Caption

x [m]	y [m]	z [m]	Fluid Temperature [°C]	Measured [°C]	Error [°C]
3.20	1.49	0.41	24.77	26.1	1.33
2.55	1.49	0.41	24.36	26.0	1.64
1.89	1.49	0.41	23.83	25.6	1.77
0.70	1.49	0.41	23.60	24.6	1.00
0.05	1.49	0.41	23.42	23.2	-0.22
-0.59	1.49	0.41	22.88	21.4	-1.48
-1.23	1.49	0.41	21.58	20.4	-1.18
-1.88	1.49	0.41	21.95	18.8	-3.15
		Mean	23.30	23.26	-0.23
		Standard deviation [%]	1.111	2.79	11.98

We note that the simulated and experimental trend of the temperature variation on the same crucial geometric reference was very similar. We note a mean error of -0.23198 °C and a standard deviation of the two sets of data of 12 %. These can be explained and justified as follows: The transport equation of any parameter, φ, in the computational code is defined as:

$$\frac{\partial \rho\varphi}{\partial t} + div(\rho\varphi\vec{u}) = div(\Gamma grad\varphi) + q_{\varphi} \qquad (1)$$

This equation defines only heat transport via source, convection and diffusive terms. These terms were neglected in our model as the convective factor was assumed much larger than the other thermal transport terms. Furthermore, analysis of simulated results allows us to believe that the flow was intermittently laminar and turbulent (in both space and time). Due to model flow limitations, the simulation was performed as purely turbulent. The k-epsilon model was used. This model works well in thin shear layers where the changes in the flow direction are always so slow that the turbulence can adjust itself to local conditions. In flows where convection and diffusion cause significant differences between production and destruction of turbulence, as it is here the case, a compact algebraic prescription for the mixing length is no longer feasible and the k-epsilon model is a justified choice. However, further analysis of the characterizing equations of the model lead us to believe that a K-omega Shear Stress Transport turbulence model (k- ω SST) would have been more appropriate. The study of the k-epsilon and the k- ω SST models in (D. Ramdenee and A. Ilinca, 2011) lead us to notice that the latter model includes terms that account for stress shear transport along boundaries defined as $\left[2\rho S_{ij}S_{ij} - \frac{2}{3}\rho\omega\frac{\partial U_i}{\partial x_j}\delta_{ij}\right] - \beta^*\rho\omega^2$. In the real server center, such phenomena do occur. Therefore, the modeling would have been more precise with the k- ω SST, which is, unfortunately not available in SolidWorks Flow Simulation. It is, also, interesting to note that the flow is intermittently laminar and turbulent, such that the use of a transition model including an intermittency factor would have improved precision. Apart from the modeling aspect, it is pertinent to note that the precision of the Windmate 200 is 1 °C for temperature and 3 % variability for volumetric flowrate. Finally, human presence during measurement is, also, considered to influence experimental results. Therefore, we can consider our model very accurate whilst recognizing improvement opportunities in future versions

5. INDUSTRIAL SIMULATION

The actual industrial data center that was simulated to recommend on thermal optimisation contained 36 cabinets and 4 cooling units. Figure 2 represents the configuration with respective power output and air flow at different data centre cabinets and cooling units.

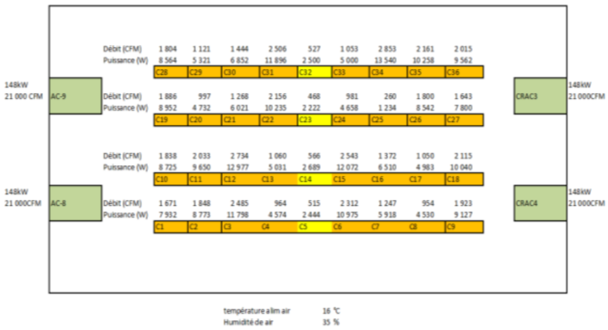

Figure 2: Real industrial model configuration

Following a meeting with the client, the different possible modifications that could be applied to the server center were established and the mandate was to evaluate the impact of the different modifications to advise upon the return rate of investment and energetic consumption impact. 7 different scenarios were identified. The scenarios were a combination of confinement possibilities of the cabinets and the air volumetric flow rate geometric positioning. In the complete study, all the different cases were exhaustively studied and recommendations were based on the analysis of around 6 different simulations. With reference to figure 3, 4 and 5 where the geometries of the different scenarios are illustrated, in this article, we will emphasize on the 1) effect of complete confinement, 2) effect of cooling airflow from the ceiling (down flow) or the floor (up flow), 3) effect of removing all confinements and 4) the effect of only confining the cabinets top with applying down flow cooling effect (from the ceiling). The choice of these results have been based on the need to define scenario(s) that amplify temperature fall and standardization, effect of standard cooling flow direction, recirculation and vortex analysis and the need to characterize the most energy efficient scenario.

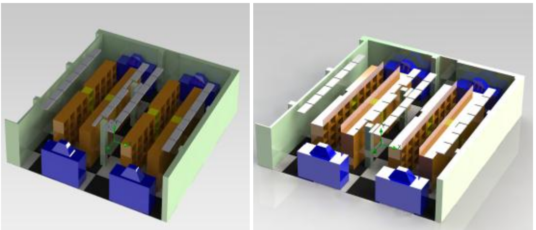

Figure 3: Scenarios representing cool air flow from raised floor and exit in ceiling without confinement and confinement over cabinet only

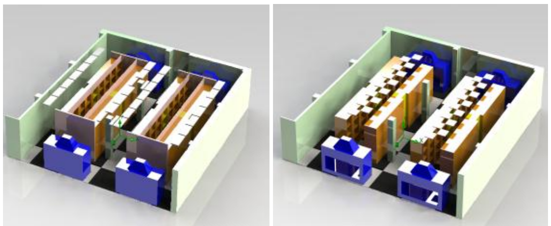

Figure 4: Scenarios representing cool air flow from raised floor and exit in ceiling with full confinement and cool air down flow with exit through air conditioners without any confinement

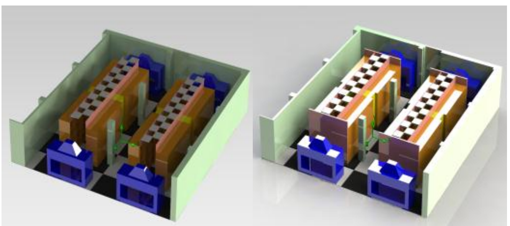

Figure 5: Scenarios representing cool air down flow with exit through air conditioners with confinement only above the cabinets and full confinement respectively

6. RESULTS

6.1. Effect of total confinement

When the confinement is complete, the mean temperature in the hot corridor tends to become uniform and falls by several degrees. We notice from figures 6, 7, 8 and 9 that the temperature range gets reduced from a maximum of 31^0C to 27^0C. This method seemingly better controls the hot air exits and enable more effective channeling of the cool air. The confinement is in fact a vertical lid that connects and bounds any cabinet or wall with the ceiling.

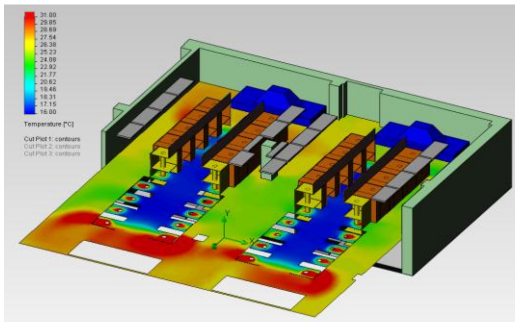

Figure 6: Horizontal temperature distribution for second scenario illustrated in figure 3

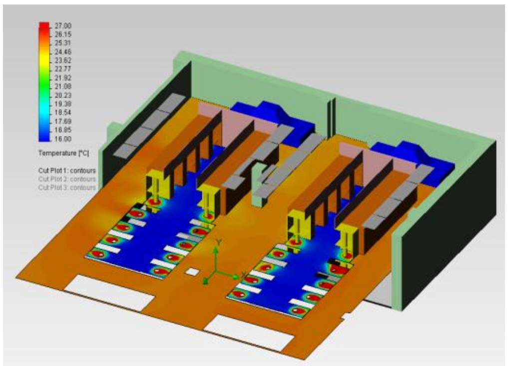

Figure 7: Horizontal temperature distribution for first scenario in figure 4

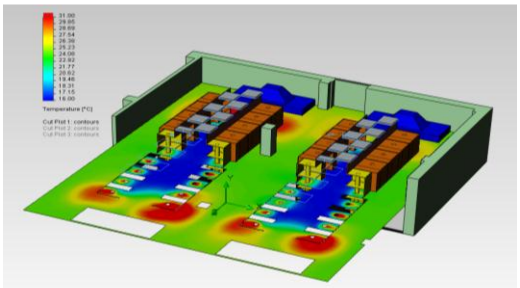

Figure 8: Horizontal temperature distribution for first scenario in figure 5

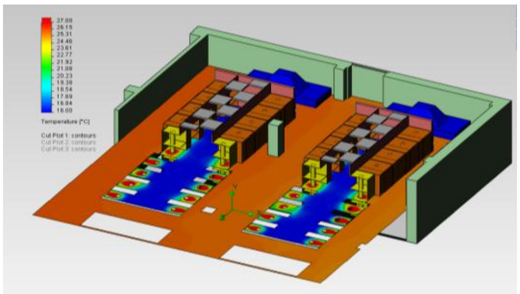

Figure 9: Horizontal temperature distribution for second scenario in figure 5

Figures 6 to 9 illustrate temperature distribution results on the same horizontal plane for diverse cool air flow direction for totally confined and semi confined geometries. It is clearly noticeable that for figures 7 and 9 where the simulations are for different air flow but complete confinement, the temperature is very uniform. Furthermore, analysis of the temperature color range allows seeing rapid temperature drop.

6.2. Effect of up flow or down flow cooling

This section ponders on the impact of the flow being from raised floor to ceiling or vice versa or into air conditioners directly. Being given that the previous section showed marked advantage of total confinement, we will evaluate the importance of cooling scheme with total confinement only. Figures 7, 9, 10 and 11 illustrate temperature uniformity in the cold corridor which is cooled by airflow at 16 $^{\circ}$C. We can, thus, conclude on the fact that when the cold corridor is completely confined, there is no significant difference in the temperature distribution by up flow or down flow cooling.

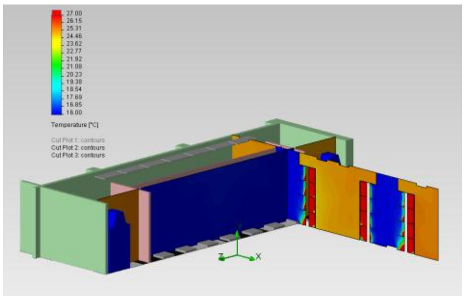

Figure 10: Vertical temperature distribution cool air down flow with exit through air conditioners without any confinement

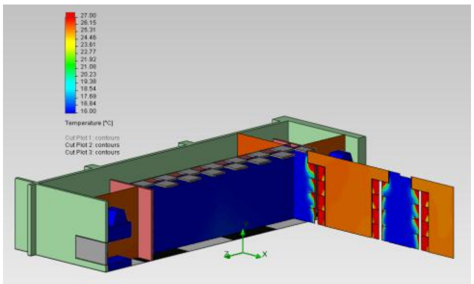

Figure 11: Vertical temperature distribution for second scenario in figure 5

Analysis of figure 7, 9, 10 and 11 allow us to see appreciable homogeneity and comparable low profile temperature range. These two advantages have been already associated to total confinement modification. It was, thus, concluded, that the two cooling sequences (simulated in these 2 cases) do not have any major consequence on the results.

6.3. Effect of removing all confinements

Previous simulations allowed us to see that confinement appreciably improve homogeneity and lower average temperature distribution. Simulation without any confinements has been performed to evaluate the presence of recirculation zones and vortex shedding that reduces homogeneity and adds to zones of “thermal uncertainty”. Such is visible in figures 6 and 8 whereby circular high temperature zones are visible. These have been explained by hot air recirculation which is noticeable in figures 12 and 13.

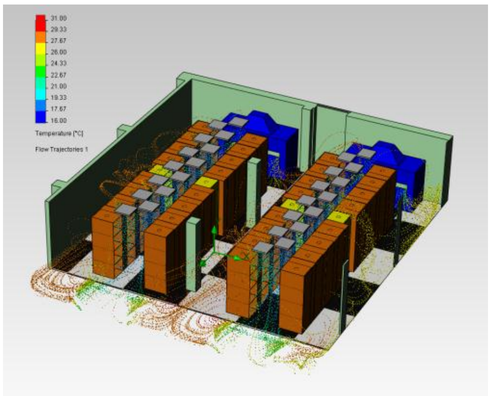

Figure 12: Air flow simulation for second scenario 1 in figure 3

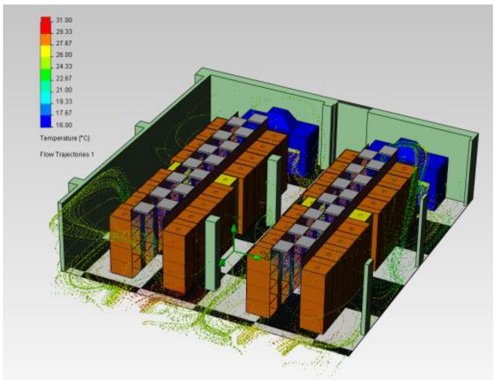

Figure 13: Air flow simulation for second scenario 2 in figure 3

Close analysis of figure 12 allows us to see more high temperature zones due to hot air recirculation as compared to figure 13. This can be explained by the absence of confinements in simulation of figure 12. The confinements in simulation of figure 13 are only over the cabinets.

6.4. High efficiency gain

It has been noticed through several previous simulations that major advantages are achieved with adding confinements above the cabinets. Successive simulations with different flow direction and mechanisms with these confinements have leaded us to advice on the significant energetic gain of down flow cooling from ceiling with confinements above the cabinets. This can be illustrated by comparison of figure 8 with figure 14, and comparison of figure 15 with figure 16.

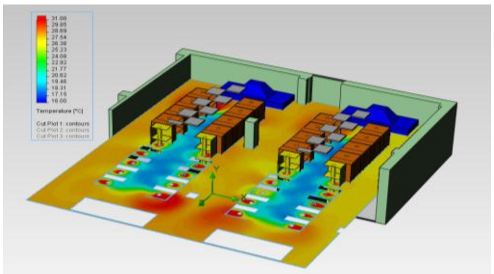

Figure 14: Horizontal temperature distribution for second scenario in figure 4

Comparison of figure 8 and 14 let us notice, that on a horizontal plane for ceiling to air conditioners air flow in both cases, the simulation with confinements (figure 8) propose cooler charged air in the cabinets. Similar conclusions can be made from figures 15 and 16.

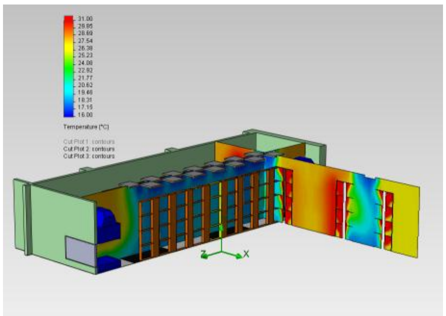

Figure 15: Vertical temperature distribution for cool air down flow with exit through air conditioners without any confinement

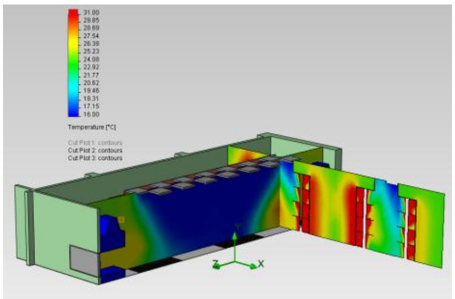

Figure 15: Vertical temperature distribution for cool air down flow with exit through air conditioners with confinement only above the cabinets

7. FUTURE OF THE PROJECT

It is essential to understand that most industrial data centers we have worked upon are two-lines one and the temperature sensors are placed far from the cold corridor found between the server centers lines. This is quite contradictory because the cold central corridor is the most crucial place for temperature control. In this section of the report, we propose an outline of the work being conducted in order to define a refined temperature control algorithm. In the article we develop differential equations and a control algorithm for a 2D lumped model of the cold corridor. This part of the article only describes a methodology that is yet to be modified to the server context. This will be part of the future of this presented project. In this case, we model 4 walls and a thermal power flux as generic as an energy $Q_{ser,}$ This model does perfectly represent the server center model but is used as starting point into applying predictive control to a future 3D model of the server room. Furthermore, we acknowledge that definition of perpendicular heat transport might be an important factor since the server room is not a confined one. In the future, effort will be made on the 2D model to compare predicted results with experimentally measured ones, for the same situations and improve the accuracy of the model as a result. Examination of the complete air conditioning system allows us to see that we can control the alimentation air temperature via valves controlling chilled water flow in the CRAC unit. The volumetric flow of the air can also be controlled. It is essential to keep in mind that, for any control mechanism, the norms of air content must be respected and be a primary condition in the algorithm code. An outline of a typical air conditioning system in a building is illustrated in figure 16, which is an excerpt from (Tabeli and al).

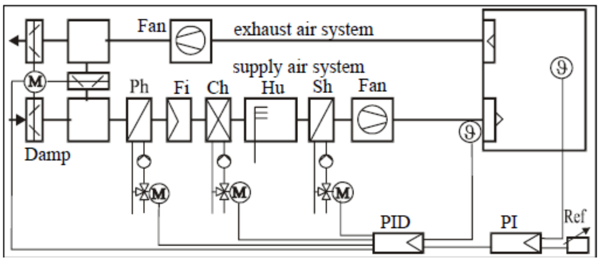

Figure 16: Typical Air conditioning system from (Talebi and al)

The air conditioning system with its control loops for a temperature cascade control is used for education and research on building energy management systems. In its structure it is equivalent to common industrial applications explaining the choice of this example. The supply air system consists of motorized dampers (Damp), a preheated exchanger (Ph), a chilled water exchanger (Ch) and a secondary heating exchanger (Sh) to condition the air temperature. The steam humidifier (Hu) and filter (Fi) are not taken into account for the temperature control (Talebi, R. Daryani and H. Plass, 1998; Talebi, R. Daryani, and H. Plass, 1998). In order to predicatively evaluate the temperature distribution and

accordingly adjust alimentation temperature, a primary 2D model has been built. Despite the advent of higher accuracy techniques, a lumped method has as advantage the implementation of a simple model that allows real time simulation to propose predictive control mechanisms.
Predictive control will be applied on a 2D model expressed via state equations defined by differential equations. In this section, we will describe the equations for a simple four walled room with a given disturbance due to the data centers Q_{server}. Equations are only developed for one partition. In the future, such will be improved and adjusted to the real server centre room. Let T represent the temperature, Q, the heat flux, A, the surface are, X the thermal transmittance in $Wm^{-2}k^{-1}$, the subindex "i" representing interior, "o" representing exterior and "v" representing ventilation. Furthermore, "w" represents wall with subindexes 1, 2, 3 and 4 expressing the four different walls. "f" refers to the floor, "c" refers to the ceiling and "p" to a partition. "C" refers to capacitance. Finally, "t_{iair}" and "t_{oair}" represent the internal and external air temperature. The different temperatures shall be measured by different sensors. Application of state equations with known data center disturbance will allow for calculation of temperature gradient and hence predict future temperature. Henceforth, ventilation flaps and valves will be accordingly controlled to keep optimized temperature. The definition of state equations will be implemented in Simulink Linear library model. Derivatives will be integrated using a 4th order Runge Kutta method with a 0.1 hour integration interval. (This is assumed sufficient considering the macro variability scheduled in the data centre system). The simple four walls, one partition model is first schematized in an equivalent electrical diagram illustrated in figure 17 for simplicity reasons.

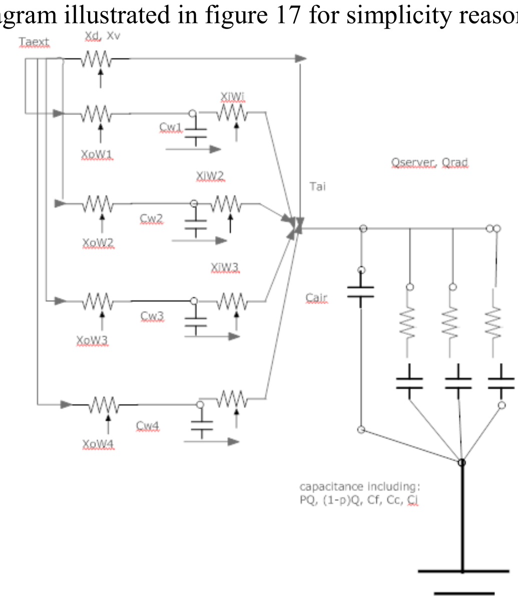

Figure 17: Equivalent Electrical schematic of thermal loads

Following equations describe the different thermal transfers at the different parts of the system: wall, partitions, etc.

Wall 1:

$$\frac{dT_{w1}}{dt} = \frac{Aw_1}{Cw_1}[X_i w_1(\mathrm{t_{iair}} - T_{w1}) + X_0 w_1(\mathrm{t_{oair}} - -T_{w1})] \tag{1}$$

Wall 2:

$$\frac{dT_{w2}}{dt} = \frac{Aw_2}{Cw_2}[X_i w_2(\mathrm{t_{iair}} - T_{w2}) + X_0 w_2(\mathrm{t_{oair}} - -T_{w2})] \tag{2}$$

Wall 3:

$$\frac{dT_{w3}}{dt} = \frac{Aw_3}{Cw_3}[X_i w_3(\mathrm{t_{iair}} - T_{w3}) + X_0 w_3(\mathrm{t_{oair}} - -T_{w3})] \tag{3}$$

Wall 4:

$$\frac{dT_{w4}}{dt} = \frac{Aw_4}{Cw_4}[X_i w_4(\mathrm{t_{iair}} - T_{w4}) + X_0 w_4(\mathrm{t_{oair}} - -T_{w4})] \tag{4}$$

Floor:

$$\frac{dT_f}{dt} = \frac{A_f}{C_f}\left[\frac{PQ_{ext}}{A_f}_i + X_f(\mathrm{t_{iair}} - -T_f)\right] \tag{5}$$

Ceiling:

$$\frac{dT_c}{dt} = \frac{A_c}{C_c}[X_c(\mathrm{t_{iair}} - T_c)] \tag{6}$$

Single partition:

$$\frac{dT_{ip}}{dt} = \frac{A_{ip}}{C_{ip}}\left[\frac{(1-P)Q_{ext}}{A_{ip}}_i + X_{ip}(\mathrm{t_{iair}} - -T_{ip})\right] \tag{7}$$

Internal Air:

$$\frac{dt_{iair}}{dt} = \frac{1}{C_a}\Big[Q_p + Q_{ext} + \left(A_g X_g\right) + (\mathrm{t_{oair}} - \mathrm{t_{iair}}) + A_{w1}X_{iw1}(T_{w1} - \mathrm{t_{iair}}) + A_{w2}X_{iw2}(T_{w2} - \mathrm{t_{iair}}) + A_{w3}X_{iw3}(T_{w3} - \mathrm{t_{iair}}) + A_{w4}X_{iw4}(T_{w4} - \mathrm{t_{iair}}) + A_f X_f\left(T_f - \mathrm{t_{iair}}\right) + A_c X_c(T_c - \mathrm{t_{iair}}) + A_{ip}X_{ip}\left(T_{ip} - \mathrm{t_{iair}}\right)\Big] \quad (8)$$

The differential equations are then written in matrix form of the type:

$$\dot{T} = AT + Bi \quad (9)$$

$\dot{T}$ is a vector of derivatives, A and B are matrices of coefficients, T is a vector of states and i is a vector of inputs. This leads us to the following matrix equation.

$$\dot{T} = \begin{bmatrix} \dot{T}_{w1} \\ \dot{T}_{w2} \\ \dot{T}_{w3} \\ \dot{T}_{w4} \\ \dot{T}_f \\ \dot{T}_c \\ \dot{T}_{ip} \\ \dot{t}_{iair} \end{bmatrix} \quad (10)$$

$$A = \begin{bmatrix}
\frac{-A_{w1}}{C_{w1}}[X_{iw1+}X_{ow1}] & 0 & 0 & 0 & 0 & 0 & 0 & \frac{A_{w1}X_{w1}}{C_{w1}} \\
0 & \frac{-A_{w2}}{C_{w2}}[X_{iw2+}X_{ow2}] & 0 & 0 & 0 & 0 & 0 & \frac{A_{w2}X_{w2}}{C_{w2}} \\
0 & 0 & \frac{-A_{w3}}{C_{w3}}[X_{iw3+}X_{ow3}] & 0 & 0 & 0 & 0 & \frac{A_{w3}X_{w3}}{C_{w3}} \\
0 & 0 & 0 & \frac{-A_{w4}}{C_{w4}}[X_{iw4+}X_{ow4}] & 0 & 0 & 0 & \frac{A_{w4}X_{w4}}{C_{w4}} \\
0 & 0 & 0 & 0 & \frac{-A_f}{C_f} & 0 & 0 & \frac{A_f X_f}{C_f} \\
0 & 0 & 0 & 0 & 0 & \frac{-A_c X_c}{C_c} & 0 & \frac{A_c X_c}{C_c} \\
0 & 0 & 0 & 0 & 0 & 0 & \frac{-A_p X_{ip}}{C_{ip}} & \frac{A_{ip}X_{ip}}{C_{ip}} \\
\frac{-A_{w1}X_{w1}}{C_a} & \frac{-A_{w2}X_{w2}}{C_a} & \frac{-A_{w3}X_{w3}}{C_a} & \frac{-A_{w4}X_{w4}}{C_a} & \frac{A_f X_f}{C_a} & \frac{A_c X_c}{C_a} & \frac{A_p X_{ip}}{C_a} & \frac{-1}{C_a}[A_g X_g + X_v + A_{w1}X_{v1} + A_2 X_2 + A_3 X_3 + A_4 X_4 + A_f X_f + A_c X_c + A_{ip}X_{ip}]
\end{bmatrix} \quad (11)$$

$$T = \begin{bmatrix} T_{w1} \\ T_{w2} \\ T_{w3} \\ T_{w4} \\ T_f \\ T_c \\ T_{ip} \\ t_{iair} \end{bmatrix} \quad (12)$$

$$B = \begin{bmatrix}
0 & 0 & 0 & \frac{A_{w1}X_{0w1}}{C_{w1}} \\
0 & 0 & 0 & \frac{A_{w2}X_{0w2}}{C_{w2}} \\
0 & 0 & 0 & \frac{A_{w3}X_{0w3}}{C_{w3}} \\
0 & 0 & 0 & \frac{A_{w4}X_{0w4}}{C_{w4}} \\
0 & 0 & \frac{p}{c_f} & 0 \\
0 & 0 & \frac{1-p}{c_{ip}} & 0 \\
0 & 0 & 0 & 0 \\
\frac{1}{C_a} & \frac{1}{C_a} & 0 & \frac{A_g X_g + X_v}{C_a}
\end{bmatrix} \quad (13)$$

$$i = \begin{bmatrix} Q_d \\ Q_{ser} \\ Q_s \\ T_{a0} \end{bmatrix} \quad (14)$$

These equations make up the state equation. Inputs i are entered via WPPT like systems including historical and probabilistic data of data centre operation as well as sensor measured external data. This allows the differential equations inscribed in the matrices to predict the temperature gradient and temperature distribution. Henceforth, a controller (as a modification of the state equations) can be added to the system that will accordingly vary the CRAC unit commands to optimise temperature distribution. The *algorithmic* layout of this control algorithm is presented in figure 18.

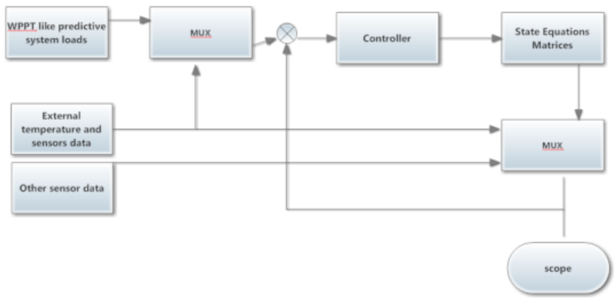

Figure 18: Preliminary layout of control algorithm

However, this model does not allow for direct control since the variable that we wish to control is the mean temperature of the cold air used to cool the server centers in the cold corridor. In practice, in order to take action on this temperature, two parameters are usually altered: the opening of the water valve in the heat exchanger coil of the cooling unit and the frequency of the CRAC engine. Since, $\dot{Q} = \dot{m}C_p\Delta T$, we can control the mean temperature in the cold corridor by acting on $\dot{m}$ which is the volumetric flow rate and the input cold air temperature to vary the cooling level of the data centers. The control loop is established with respect to a temperature sensor that measures the air temperature of the air returning to the cooling system or with respect to a thermostat situated in the cold corridor. Therefore, the control algorithm will make use of a large number of steady state simulations with three varying parameters. These are: the temperature of the input cold air, the volumetric flow of the cold air and the power output of the server centres. Analysis of the different simulations will allow us to define within a multi variable choice algorithm a correlation between the temperatures measured by the sensor to propose a control via correction from results obtained from the different simulations. Furthermore, the inclusion of statistical model providing information on the server centers power output and a model allowing quantification of energy and cost efficiency by varying the volumetric flow rate of the cold air, in the simulation-sensor control model, will allow real time optimisation according to different criteria. The state model can be used as a gradient matrix to verify simulation results at chosen simulation time steps.

8. CONCLUSION

In this paper we have presented work performed on an industrial data center in an attempt to define, according to given inputs, the temperature distribution in the cold and hot corridors. Emphasis has been laid in the modeling of the cold corridors. Moreover, different scenarios, including flow direction and partition have been evaluated using CFD methods. The CFD model has been calibrated as far as possible and the standard deviation between the measured and CFD predicted results have been explained and analyzed from both experimental (measurement errors, etc.) and numerical (turbulence and transport models) point of view. Different results have been illustrated to propose and enable the reader to evaluate the pertinence of such tools in data center and room temperature distribution modeling. Finally, the authors have pondered upon control algorithms and preliminary work based on a simplified 2D lumped model has been illustrated in this paper. In the future, we project to work on an optimized meshed model and propose a 3D predictive control model.

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