ABSTRACT
During the last years, the rapid increase of cloud computing, high-powered computing and the vast growth in Internet use have aroused the interest in energy consumption and carbon footprint of data centres. The IT room, also known as whitespace, is the main and the most critical space in these unique facilities. The whitespace is an environmentally controlled space that houses IT equipment and cabling directly related to computation and telecommunications systems that due to their operation generate considerable amounts of heat. Air flow management inside IT rooms is a key issue to increase energy efficiency which has led to apply different solutions such as hot/cold aisle containment or integration of liquid cooling technology. The paper presents different modelling approaches for IT rooms using TRNSYS and discusses the advantages and limitations of the proposed modelling techniques, together with the validation with experimental data.

INTRODUCTION
Data centres are facilities used to house computer systems and associated components aim for performance, reliability and security 24 hours a day the 365 days a year, transforming them into a highly energy demand infrastructures. In the last years, the total energy demand of data centres has experienced a dramatic increase which still growing. This is why data centre industry and researchers are undertaking efforts to implement energy efficiency measures and to integrate renewable energy into its portfolio to overcome energy dependence and to reduce operational costs and CO₂ emissions (RenewIT, 2014). Òró et al. (2014) presented a comprehensive overview of the current data centre infrastructure and summarized a number of currently available energy efficiency strategies and renewable energy integration into the data centre industry together with the recent efforts for its characterization using numerical models.

The evaluation of the air cooling performance in data centres plays an important role in the overall energy efficiency often focuses on air management. Since data center energy density starting to increase, air management starts also to be a key factor in the facility. It has the objective to keep Information Technology (IT) equipment inlet conditions within the recommended (ASHRAE, 2011) ranges with the minimum energy consumption. The basic air management starts with air supply from the Computer Room Air Conditioning (CRAC) units or Computer Room Air Handle (CRAH) units to IT servers, heat removal, and ends with hot air return back to the CRAH units. However, in reality, air streams are affected by different phenomena such as bypass, recirculation and pressure air drop, decreasing cooling efficiency and creating vicious cycle of rise in local temperature. (Salim and Tozer, 2010) presented a set of indexes (by-pass, BP, recirculation, R, negative pressure, NP, and balance, B) to characterize air management in IT rooms. The relationship between proposed indexes are shown graphically in Figure 1.

![Figure 1. Data centre air management metrics by (Salim and Tozer, 2010)](image-url)
of flow distribution is difficult because of the turbulent flow conditions, complexity of geometry, flow paths, and blockages in the plenum from cabling and chilled water distribution pipes. In addition, most of the CFD applications are built upon steady-state analysis. Then, these models are impractical to be used for the annual energy dynamic analysis of an entire facility.

Many researchers have been investigated heat and mass transfer phenomena inside data centre rooms using their own numerical tools and models, as (Zhou and Wang, 2011). Recently, (Demetriou and Khalifa, 2012) developed and validated experimentally a simplified thermodynamic model that can be used for exploring optimization possibilities in air cooled data centres. The results of this analysis highlight the trade-off between low air supply temperature and increased air flow rate to be considered when optimizing the operation of an air cooled infrastructure. Later on, they used the model to evaluate parametrically the total energy consumption of the cooling system for data centres that utilizes aisle containment. The analysis showed a potential for as much as 60% savings in cooling infrastructure energy consumption by utilizing an optimized enclosed aisle configuration with bypass recirculation, instead of a traditional enclosed aisle in which all the data centre exhaust is forced to flow to the CRAC units. (Lin et al., 2013) developed a real time transient thermal model to discuss the negative effect of temperature rise in data centres after a cooling power failure. They highlighted some strategies such as placing critical cooling equipment on backup power, choosing equipment with shorter restart time and employing thermal storage to overcome this phenomenon.

Some other researchers have conducted recently investigations of data centre facilities using general energy modeling packages as eQuest or TRNSYS (TRNSYS, 2010) which are extremely flexible software environments used to simulate the behaviour of energy systems. (Lee and Chen, 2013) conducted an study to analyse the energy savings potential of air-side free cooling for data centres using eQUEST. The model for IT rooms were very simple and air inefficiencies were not considered. (Kummert et al., 2009) proposed a TRNSYS model which described a 5 MW chilled water plant used for cooling a data centre in the UK. TRNSYS was used with the aim of studying the impact of perturbations such as chiller failure on water and air temperatures profiles. Moreover, they showed how the thermal inertia of the system either building envelope or storage tanks allowed a suitable design. IT room side heat transfer is modelled by splitting the IT hall into different zones. Each zone represents the notional part of the hall served by each CRAC unit and no interaction is assumed between the different zones. A schematic of a zone and its associated CRAC is represented in Figure 2.

The paper will present three different approaches to model the IT whitespace with the aim to be integrated in complete simulation projects of data centre facilities using TRNSYS. The model will overcome previous proposals considering air management efficiencies and effects of liquid cooling solutions supporting conventional air cooling of the IT equipment. The dynamic energy models are validated with experimental data from one case study and the advantages and limitations of the proposed modelling techniques are discussed.

DESCRIPTION OF THE MODELLING APPROACH

General modelling environment for IT rooms in TRNSYS

With the purpose to integrate in broad models that will be able to simulate the performance of complete data centre energy supply infrastructures, one essential part is the whitespace sub-model. The IT whitespace, also known as IT room, is an environmentally controlled space that houses IT equipment and cabling directly related to computation and telecommunications systems that due to their operation generate considerable amounts of heat. Traditionally, IT equipment in the whitespaces has been cooled by air systems, being airflow management inside IT rooms a key issue to increase energy efficiency. Additionally, for high density data centres implementation of direct or indirect liquid cooling systems is also a common solution. The authors have been developed a general submodel in TRNSYS which aims to simulate the transient performance of data centre white spaces that can be both air and liquid cooled. Figure 3 depicts schematically the sub-model that it is composed by several parts: the water/air circuit to supply cooled liquid directly to the CRAHs units, the white space itself housing the IT equipment which is highlighted in Figure 3, and the liquid circuit to supply cooled liquid directly to the whitespace. Also, the submodel is prepared to be connected with other models to define the environment conditions of the whitespace surroundings that can be the outdoor ones in case that whitespace is in direct contact with external ambient.
The paper focuses on the description of different approaches for the whitespace model. The aim is to use the model to simulate liquid and air cooled configurations, different air management strategies and transient behaviour. Adaptation to several configurations can be done through parameters in the model. The following section describes the tested modelling approaches.

### Whitespace modelling approach #1. Simple lumped capacitance building and lumped mass

The whitespace was modelled as a simple lumped capacitance single-zone building with internal gains (Type 759). This means that the structure is subjected to internal gains (IT and miscellaneous gains) and to building losses, but neglects solar gains. The racks were modelled as lumped mass (Type 963) which means that they can be characterized by the rack temperature and the heat transfer to the environment. Environment means the whitespace surroundings. Here capacitance effects i.e. rack structure and servers are included. The use of this type allows implementing direct cooling technology by means of negative internal gains. Figure 4 depicts a schematic view for the data centre white space model. Each point represents the different parts of the system where air condition (temperature and humidity) are calculated with in the model, together with the environmental conditions ($T_{amb}$) and internal heat gains. Heat gains due to the IT equipment ($\dot{Q}_{IT}$) together with the effects of direct liquid cooling system ($\dot{Q}_{liquid}$), as negative internal gains, are implemented in the IT racks component. Miscellaneous heat gains ($\dot{Q}_{misc}$, i.e. lighting and others) are defined as internal gains in the IT room component. Therefore, it is possible to calculate average temperature of the racks ($T_{rack}$) and internal heat gains. Air relative humidity in each of the mentioned points are also calculated. Sensible cooling power of the air ($\dot{Q}_{air}$) and liquid cooling power ($\dot{Q}_{liquid}$) can be calculated using Equation 3 and Equation 4. Notice that the liquid cooling depends directly on the effectiveness of the heat exchanger ($h_{hx}$).

\[
\dot{Q}_{air} = m_{air} \cdot c_{pair} \cdot (T_{supply} - T_{return}) \tag{3}
\]

\[
\dot{Q}_{liquid} = \dot{m}_{liq} \cdot c_{pliq} \cdot \varepsilon_{hx} \cdot (T_{Rack} - T_{liq, in}) \tag{4}
\]

### Whitespace modelling approach #2. Multizone building and lumped mass

The whitespace is modelled as a four zones model using a detailed multi-zone building model (Type...
A schematic view of the modelling approach is depicted in Figure 5, where the zones are limited with dashed lines. Four thermal zones (air volumes assumed to be at uniform temperature) are identified in a typical IT room with raised floor: the plenum ($T_{\text{Plenum}}$), the cold aisle ($T_{\text{Cold}}$), the zone where the racks are placed and its close surrounding air ($T_{\text{Rack,Out}}$) and the hot aisle area ($T_{\text{Hot}}$). Using a detailed multi-zone building approach for the IT room allows to accurately defining the geometrical features of each thermal zone as well the composition of each of the surrounding walls. The IT racks are modelled with a lumped mass model (Type 759) as in the previous model approach and it is assumed to interact with its surrounding air. Miscellaneous gains are divided between cold and hot zones according to their respective area. This approach also allows considering not only losses through the skin of the IT room, but also solar gains. Moreover, it is also possible to define different boundary conditions for each of the walls.

**Figure 5. Scheme of the whitespace model. Modelling approach #2**

In this approach air inefficiency also is taken into account using the ability of the multi-zone building to define coupling air flows between zones. The bypass ratio ($\epsilon_{bp}$) defines the airflow going from the plenum to the hot zone, and therefore the remaining airflow to the cold zone. Similarly, the recirculation ratio ($\epsilon_{rec}$) is modelled using the airflow from the hot zone to the cold zone.

**Whitespace modelling approach #3. Multizone building and simplified internal gains model**

The third model approach is based also on a three zones model using a detailed multi-zone building model (Type 56) but with a simplified approach regarding how IT racks and IT gains are modelled. A schema of the approach is shown in Figure 6. Capabilities of defining internal gains through convective and radiative gains have been used. So, the IT gains need to be divided between direct radiative and convective gain in the hot thermal zone using the ratio $r_{IT}$ defined in Equation 5.

$$r_{IT} = \frac{Q_{IT,\text{conv}}}{Q_{IT}} \quad (5)$$

With this approach, the capacitance of the IT equipment needs to be added to the capacitance of the air in the hot thermal zone. Notice that using this approach it will not be possible to have the average rack temperature due the fact that the lumped mass component is not used and temperature $T_{Hei}$ is used to estimate the liquid cooling performance.

**DESCRIPTION OF THE CASE STUDY FOR VALIDATION**

In order to prove the robustness and consistency of the different models proposed, they must be validated using real data. To do so, the authors have looked through the literature to find out a case study containing both cooling technologies: air and liquid cooling systems and with measurement data available. The Georgia Institute of Technology Razor HPC cluster has been selected for the validation process. This data centre which is extensively described in (ASHRAE, 2008). The data centre consists primarily of 1,000 blade servers of 178 kW in total. In addition, support configurations of storage, management and networking hardware were required to operate the infrastructure. The total IT load is thus 178 kW and the miscellaneous load is 14 kW. All this equipment was placed in the whitespace with a total area of 93 m². In order to meet the cooling requirements, a rear-door exchanger (RDHX) was also implemented. The device is a copper-tube, aluminium-fin, air to water heat exchanger that replaces the rear panel of a computer rack.

**Figure 6. Scheme of the whitespace model. Modelling approach #3**

**Figure 7. Case study: Data centre layout**
Figure 7 shows the data centre layout. The blade racks are the six exterior racks on either side of the four support hardware racks. Air was directed below the raised-access floor in the direction indicated by the arrows on the four air-conditioning units shown at the top of the figure. By partitioning the entire subfloor area, a dead-head situation was created in the perforated tile area that maximized static pressure and airflow rates. Second, because the CRACs were located in such close proximity to the rack exhausts, direct return of warm air to the unit intakes was ensured to optimize unit efficiency. Finally, the hot/cold aisle principle was taken to the extreme: a wall completely separating the warm and cold sides of the cluster, shown as the thick black line in Figure 7, guaranteed an absolute minimum of warm air recirculation, a problem that plagues many modern-day data centres.

![Data Centre Layout](image)

**Figure 8. Schematic of cooling method and temperature measurements in the validation case study (ASHRAE, 2008)**

A study was conducted in which temperature were measured at multiple locations among racks and CRAHs. For the blade server racks, thermocouples were placed in three regions: equipment inlet ($T_{Eeq,In}$), equipment outlet ($T_{Eeq,Out}$), and heat exchanger outlet ($T_{HX,Out}$), as shown in Figure 8.

**RESULTS AND DISCUSSION**

**Validation process**

This section shows the validation results of the different models presented using the already described case study. Parameters of the model and boundary conditions have been estimated from the available data and set as constants to reproduce steady-state condition from which the measurements are provided through dynamic simulation.

Table 1 summarizes the values of the main parameters used in the simulation models and specifies in which model approach the parameters are used. Knowing that servers and rack chassis are made of cooper and steel, the thermal capacity of a single rack can be estimated using values to determine physical properties and standard weight values. From product’s brochure, rack weight is 125 kg (APC, 2015), and server weight is 18 kg (ORACLE. 2015).

Therefore, thermal capacitance of a single rack fully populated by 42 U-servers is 260 kJ/K. From characteristics of standard servers the area of the surface of exchange between air and the solid material in a single server have been estimated to be 0.54 m$^2$, based on standards U-servers size. Therefore, for this case study, $C_{rack}$ and $A_{rack}$ can be estimated knowing that the whitespace is populated by 16 racks with 42 servers, each. Value of the heat transfer coefficient between racks and air in the IT room, $h_{rack-air}$, has set to 27.8 W/m$^2$·K, based on correlations for turbulent forced convection phenomena. In order to determine the heat transfer coefficient between the servers and the cooling air, the hypothesis that the server is a cavity and the air is circulating under forced convection has been made. From fundamentals of heat and mass transfers (Incropera and Dewitt, 1996), and knowing that the air flow is turbulent inside the servers, preliminary calculations suggested a value of 33 W/m$^2$·K for the convective heat transfer coefficient. This value was calculated under steady state boundary conditions but, after a parametric study using the case study already described, the value of 27.8 W/m$^2$·K for $h_{rack-air}$ correlates better with experimental results.

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>VALUE</th>
<th>UNITS</th>
<th>MODEL</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{Plenum}$</td>
<td>54</td>
<td>m$^3$</td>
<td>#1, 2, 3</td>
</tr>
<tr>
<td>$V_{Cold}$</td>
<td>108 / 135</td>
<td>m$^3$</td>
<td>#2 / #3</td>
</tr>
<tr>
<td>$V_{Hot}$</td>
<td>108 / 135</td>
<td>m$^3$</td>
<td>#2 / #3</td>
</tr>
<tr>
<td>$V_{Rack-Out}$</td>
<td>54</td>
<td>m$^3$</td>
<td>#2</td>
</tr>
<tr>
<td>$U$-value</td>
<td>3.23</td>
<td>W/m$^2$·K</td>
<td>#1, 2, 3</td>
</tr>
<tr>
<td>$A_{sv}$</td>
<td>129</td>
<td>m$^2$</td>
<td>#1</td>
</tr>
<tr>
<td>$UA$</td>
<td>417</td>
<td>W/K</td>
<td>#1</td>
</tr>
<tr>
<td>$C_{sys}$</td>
<td>64,783</td>
<td>kJ/K</td>
<td>#1</td>
</tr>
<tr>
<td>$C_{rack}$</td>
<td>4,160</td>
<td>kJ/K</td>
<td>#1, 2</td>
</tr>
<tr>
<td>$A_{rack}$</td>
<td>363</td>
<td>m$^2$</td>
<td>#1, 2</td>
</tr>
<tr>
<td>$h_{rack-air}$</td>
<td>27.8</td>
<td>W/m$^2$·K</td>
<td>#1, 2</td>
</tr>
<tr>
<td>$C_{Plenum}$</td>
<td>64.8</td>
<td>kJ/K</td>
<td>#2, 3</td>
</tr>
<tr>
<td>$C_{Cold}$</td>
<td>130 / 162</td>
<td>kJ/K</td>
<td>#2 / #3</td>
</tr>
<tr>
<td>$C_{Hot}$</td>
<td>130</td>
<td>kJ/K</td>
<td>#2</td>
</tr>
<tr>
<td>$C_{Rack-Out}$</td>
<td>64.8</td>
<td>kJ/K</td>
<td>#2</td>
</tr>
<tr>
<td>$r_{fr}$</td>
<td>0.9</td>
<td></td>
<td>#3</td>
</tr>
</tbody>
</table>

The heat transfer area from the whitespace to the environment has been estimated considering all the walls but excluding roof and floor. Due to the characteristics of the case study analysed, as hot/cold aisle principle was taken to the extreme, the bypass ratio and the recirculation ratio are set to 0.05 and zero, respectively. Values for the volumes and capacitances for the thermal zones in modelling approaches 2 and 3, the plenum and the cold and hot zones, are indicated in Table 1.
The supply air conditions, \((T_{\text{supply}}, RH_{\text{supply}})\) its volumetric flow \((V_{\text{air}})\), and the liquid conditions \((\dot{m}_{\text{liquid}} \text{ and } T_{\text{liquid},\text{in}})\) are shown in Table 2. Notice that in the case study presented water was used as the liquid cooling fluid. Environmental temperature surrounding the IT room is fixed at 20 °C, although not detailed information has been able to extract from the case study description for this boundary condition.

### Table 2

**Boundary conditions for the case study validation**

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>VALUE</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_{\text{supply}})</td>
<td>15.2</td>
<td>°C</td>
</tr>
<tr>
<td>(V_{\text{air}})</td>
<td>40,000</td>
<td>m³/h</td>
</tr>
<tr>
<td>(e_{\text{bp}})</td>
<td>0.05</td>
<td>-</td>
</tr>
<tr>
<td>(e_{\text{rec}})</td>
<td>0.00</td>
<td>-</td>
</tr>
<tr>
<td>(e_{\text{hx}})</td>
<td>0.35</td>
<td>-</td>
</tr>
<tr>
<td>(RH_{\text{supply}})</td>
<td>60</td>
<td>%</td>
</tr>
<tr>
<td>(T_{\text{amb}})</td>
<td>20.0</td>
<td>°C</td>
</tr>
<tr>
<td>(\dot{m}_{\text{liquid}})</td>
<td>18,600</td>
<td>kg/h</td>
</tr>
<tr>
<td>(T_{\text{liquid},\text{in}})</td>
<td>16.0</td>
<td>°C</td>
</tr>
<tr>
<td>(Q_{\text{IT}})</td>
<td>178</td>
<td>kW</td>
</tr>
<tr>
<td>(Q_{\text{misc}})</td>
<td>14</td>
<td>kW</td>
</tr>
</tbody>
</table>

### Figures 9 and 10

Figures 9 and 10 compare experimental and numerical data from the simulations for the scenario already described. Experimental data is available for each of the 12 blade racks and the 4 support hardware racks in the IT room. Figure 9 shows the comparison between experimental and numerical data of the air inlet temperature of the racks. While experimental data ranges from 14.2 °C to 15.8 the prediction of model 1 \((T_{\text{Rack,In}})\) is 15.2 °C which is the same of the imposed average supply temperature. Regarding the results from models 2 and 3, simulated temperatures are higher than expected. Both models consider that the cold zone is affected by miscellaneous heat gains and by heat transfer from the surrounding environment, including the plenum zone. As environment temperature has been set to 20 °C, so results of slight higher temperatures than the supply, are completely logical. Differences between the experimental results and the simulated ones are attributed to the uncertainty in the configuration of the IT room walls and the temperature of the surroundings, which are completely guessed in the models. Figure 10 compares the experimental and simulated temperature at the outlet of the IT equipment. Notice that for model 3, there is not a temperature node which can be compared with this temperature. The numerical results from model 1 and 2 shows a good agreement between predicted and the average experimental value, knowing that loads at the racks are different.

### Table 3

**Case study. Main temperatures and cooling powers. Comparison with experimental and simulated data**

<table>
<thead>
<tr>
<th>VALUE</th>
<th>EXP</th>
<th>MODEL #1</th>
<th>MODEL #2</th>
<th>MODEL #3</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_{\text{supply}}) (°C)</td>
<td>15.2</td>
<td>15.2</td>
<td>15.2</td>
<td>15.2</td>
</tr>
<tr>
<td>(T_{\text{Plenum}})</td>
<td>-</td>
<td>-</td>
<td>15.3</td>
<td>15.4</td>
</tr>
<tr>
<td>(T_{\text{Cold}}) (°C)</td>
<td>15.1</td>
<td>15.2</td>
<td>15.8</td>
<td>16.1</td>
</tr>
<tr>
<td>(T_{\text{Rack}}) (°C)</td>
<td>-</td>
<td>29.6</td>
<td>29.4</td>
<td>-</td>
</tr>
<tr>
<td>(T_{\text{Rack,Out}}) (°C)</td>
<td>22.0</td>
<td>22.1</td>
<td>21.8</td>
<td>-</td>
</tr>
<tr>
<td>(T_{\text{return}}) (°C)</td>
<td>21.5</td>
<td>22.6</td>
<td>21.8</td>
<td>24.2</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{air}}) (kW)</td>
<td>85</td>
<td>89</td>
<td>89</td>
<td>122</td>
</tr>
<tr>
<td>(\dot{Q}_{\text{liquid}}) (kW)</td>
<td>108</td>
<td>103</td>
<td>101</td>
<td>62</td>
</tr>
</tbody>
</table>

Table 3 shows the comparison between the main node temperatures in the models with the parameters specified to reproduce the case study. Some temperatures are not available for some nodes due to the intrinsic features of each model approach, i.e. the rack temperature in model 3, as not lumped mass
component is present in the model. Experimental values at the inlet and the outlet of IT equipment are average values and the 3 return temperatures in the table for \( T_{\text{return}} \) correspond with the inlet temperatures for each of the 3 CRAC’s in the case study. Air and liquid cooling powers are calculated using Equations 3 and 4, respectively. Comparison between experimental measurement and simulated results shows good agreement with differences less than 6\%, except for model 3 which is not able to estimate accurately the liquid cooling power.

**Effects varying air management ratios**

One of the aims of the proposed models is to reproduce the effect of having different air management strategies, which will result on different values of the by-pass and the recirculation ratio. Figure 11 compares the air cooling power, \( \dot{Q}_{\text{air}} \), and the liquid cooling power, \( \dot{Q}_{\text{liquid}} \), for the case study configuration with a recirculation ratio equal to 0.25 and for different values of the by-pass ratio. Figure 11 shows that the value of \( \dot{Q}_{\text{air}} \) decreases as by-pass ratio increases, in models 1 and 2. This is consequence of the different values of temperature in the vicinity of the racks dealing to an increase of the liquid cooling. As model 3 is not able to differentiate between temperatures near the racks and the one in the hot aisle (affected by bypass air flow), liquid cooling is not computed in a correctly.

![Figure 11. Variation of air and liquid cooling power versus by-pass ratio. Comparison between modelling approaches with simulated data. \( \varepsilon_{\text{rec}} = 0.25 \)](image)

Figure 12 shows the variation of temperatures at the inlet of the IT equipment for various combinations of recirculation ratio and bypass ratio, using models 1 and 2.

**Transient simulation results**

The developed models are able to simulate transient states and them they can be integrated in simulation models to compute whitespace performance affected by dynamic boundary conditions, which can be variable environment conditions, different IT heat gains over the time due to variations of IT load and/or different conditions in the supply of liquid and air systems. Figure 13 shows the results for model 1 and model 2 of the variation of \( T_{\text{Rack}} \) and \( T_{\text{return}} \) for the reference case study configuration. Further investigation is needed to acquire transient data to validate the proposed models in short transient periods.

![Figure 13. Transient simulation based on case study. Results of \( T_{\text{Rack}} \) and \( T_{\text{return}} \) for 6 minutes; initial state = 20°C. Comparison between models #1 and #2.](image)

**CONCLUSION**

The research presented in this paper led to develop three models for IT rooms in data centres using available TRNSYS components. Models aim to characterize different air management strategies as well to model liquid cooling solution inside the IT room which complements air cooling. The models have been validated with experimental data which are available from the literature for a HPC cluster room showing good agreement. The models have been used to test different air management strategies varying the values of by-pass and recirculation ratios. Results highlights that one of the modelling approaches (model 3) is not appropiarte for modelling both air and liquid cooling in IT rooms. Finally, some results are presented to show the ability of the models to reproduce transient behaviour in the IT equipment and the air. Then, models 1 and 2 have proven to be suitable to reproduce thermal performance of IT rooms with enough accuracy and can be easily integrated in more complex simulations of entire data centre facilities. However, further investigation is needed to validate the models with transient data and to test the ability of the model to be...
adapted to several configurations of IT rooms: with and without air containment, with not raised floor, without liquid cooling, etc.

**NOMENCLATURE**

- $A_{rack}$ = area between IT equipment and air
- $A_{ws}$ = external whitespace area
- $C_{Cold}$ = thermal capacitance of the Cold Zone
- $C_{Hot}$ = thermal capacitance of the Hot Zone
- $C_{Plenum}$ = thermal capacitance of the Plenum zone
- $C_{rack}$ = thermal capacitance of the IT equipment and air
- $C_{air}$ = thermal capacitance of air
- $h_{rack-air}$ = heat transfer coefficient between IT and air
- $Q_{IT}$ = heat gains due to the IT equipment
- $Q_{IT, conv}$ = convective IT heat gains
- $Q_{air}$ = air cooling power
- $Q_{liquid}$ = liquid cooling power
- $Q_{misc}$ = heat gains due to miscellaneous
- $RH_{supply}$ = air supply relative humidity
- $r_{IT}$ = ratio of convective IT heat gains
- $T_{amb}$ = environmental temperature
- $T_{Cold}$ = temperature in the cold aisle zone
- $T_{Hot}$ = temperature in the hot aisle zone
- $T_{rack}$ = average temperature of IT equipment
- $T_{rack,in}$ = inlet temperature to IT equipment
- $T_{rack,out}$ = outlet temperature from IT equipment
- $T_{return}$ = air return temperature
- $T_{supply}$ = air supply temperature
- $UA$ = overall heat transfer coefficient
- $U-value$ = wall heat transfer coefficient
- $V$ = volume of the white space
- $V_{Cold}$ = volume of the Cold Zone
- $V_{Hot}$ = volume of the Hot Zone
- $V_{Plenum}$ = volume of the Plenum zone
- $\varepsilon_{bp}$ = by-pass ratio
- $\varepsilon_{rec}$ = recirculation ratio
- $\varepsilon_{hx}$ = effectiveness of liquid heat exchanger

**ACKNOWLEDGEMENT**

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