DYNAMIC THERMAL MODELLING AND CFD SIMULATION TECNIQUES USED TO INFLUENCE THE DESIGN PROCESS IN BUILDINGS

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ABSTRACT

This application paper outlines some innovative building simulation methodologies used to predict thermal performance of complex energy efficient systems using commercially available softwares. Industry case studies are presented to demonstrate how simulation can influence the design process with requirements varying from zero carbon emissions to optimum thermal comfort.

Simplifications used to reduce computational time and handle software limitations are assessed in regards to model accuracy and the ability to influence the design decision process. Building simulation models are presented with a fit for purpose approach from simpler models to detailed HVAC models used for system performance and thermal comfort assessments.

Keywords: thermal simulation, CFD, IES VE, HVAC, Desiccant, Solar, In-slab cooling.

INTRODUCTION

Computer modelling has been extensively used in the green building industry to predict energy consumption, thermal comfort and to size HVAC equipment. The integrated modelling of innovative green building elements and HVAC systems can reduce risks, validate and optimise design and predict thermal performance required for green building and energy efficiency benchmark for certification such as Green Star (NZGBC,2011) and NABERS (NABERS,2011) (National Australian Built Environment Rating System).

The industry-oriented approach of the case studies presented in the paper show the ability to model systems appropriately despite software limitations. Simulation models are presented using a fit-for-purpose approach; for example, where simplifications are applied to keep computational time and costs low or accuracy levels are set depending on design stage. Thus, a simpler and faster model is used when testing concepts while a more detailed model is used for sizing equipment.

The IES Virtual Environment (VE) software is used for the building simulation analysis coupled with solar energy simulations performed in the TRNSYS software for Photo Voltaic (PV) panels and CPC solar evacuated collectors sizing and performance assessment. Some of the software limitations and their solutions presented in this paper are outlined in Table 1.

Table 1
Limitations of the software used and approach applied to minimise their impact

SOFTWARE	LIMITATION	APPROACH TAKEN
IES VE Apache HVAC	No component that simulates desiccant wheels	Multiplex looping system to simulate performance data
CFD module of IES VE (Microflo)	Cartesian Grid mesh, steady-state,	Segment curves, use dynamic simulation for boundary conditions
IES VE - In-slab cooling	Adjusted U-value of concrete slab cannot be calculated directly	calculated separately in the LBNL software THERM 6.3

Case studies to illustrate success of approaches to software limitations

1 Coleman St – London

The first case study consists of a server room located in the commercial building at 1 Coleman St, London, UK. The objective of this assessment was to determine the cooling capacity and layout of back-up DX units serving the Comms Room, through the use of CFD and dynamic thermal simulation. The CFD simulation software used for this assessment was the Microflo module of the IES VE software and the dynamic thermal simulation was carried out on Apache Sim module of IES VE software.

The primary Comms Room Air- Conditioning (CRAC) system serving the room consists of 6 DX units and one spare DX unit supplying conditioned air at a total flow rate of 19.8m^3/s. The conditioned air is supplied to a sub-floor void that delivers air via multiple floor grilles located between server cabinet rows. In the event of the primary system breakdown, the system will continue to operate in order to circulate air via floor grilles. The secondary backup system is then enabled to maintain the server room temperature below 30°C for a minimum of 4hrs to allow for repairs of the main system to be completed. The layout of the primary and secondary cooling system is shown in Figures 1 and 2.

The total maximum server load is 116 kW and lighting load of 12W/m^2 with a layout as shown in the CFD model (Figure 3): The total conditioned area of the room is 667 m^2 and total volume is 635m^3.

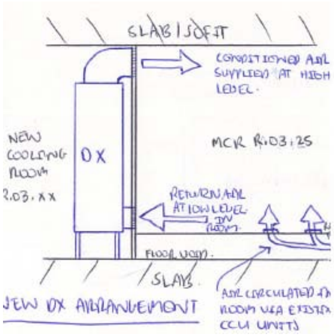


Figure 1 Up flow DX Units configuration

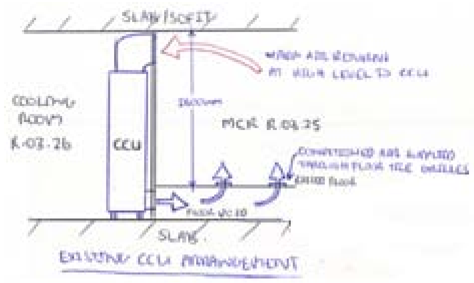


Figure 1 Down flow DX Units configuration

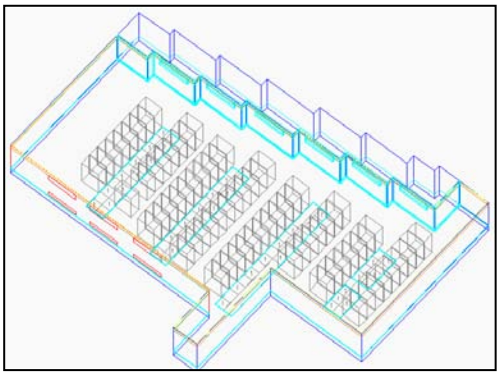


Figure 3 Microflo CFD model of the server room

<u>Main approach, lessons and outputs of the modelling</u>

The limitations of the meshing geometry simulation in IES VE Microflo drove some simplifications of the model to allow for convergence of residual values within a reduced timeframe:

- Cartesian grid meshing: Sloping walls of server room were designed with a stepped geometry to maintain grid cell aspect ratio under 10:1
- CFD Microflo module can only run steady state analysis which required dynamic thermal simulation done on a simpler Apache model
- Floor grille geometry was not feasible to be executed in Microflo with an integrated subfloor void space and air recirculation through chiller DX units. The limitation faced in the creation of this CFD geometry was the impossibility of having multiple floor holes that were required to add fans as boundary conditions. It was imperative that entire geometry was congruent and aligned with grid meshing.

The simplified geometry was developed with larger floor holes between server cabinet rows and a series of fans added as boundary conditions. Fans were added in the same location as the floor grilles and with a supply air flow rate of 0.47m3/s which is equivalent to the total air flow rate of 19.7m3/s divided by the total number of floor grilles. The cubic equation for the fan performance curve was simplified with all coefficients set to zero to create a constant velocity profile independent of pressure.

The areas within the holes created on the floor partitions where there are no fans have been filled by highly dense porous baffle boundary condition to simulate a solid floor surface. The pressure drop equation used for the porous baffle in the model followed the Darcy's Law (IESVE,2011):

$$\Delta p = -a\left(\frac{\mu}{\alpha} + \frac{1}{2}C\rho|\vartheta|\right)\vartheta \qquad (1)$$

where a is the baffle thickness, C is the pressure jump coefficient and α is the permeability. The fluids viscosity, density and velocity are given by ρ, μ and ϑ respectively. The above equation is defined in IES VE Microflo in the following form:

$$\Delta p = (C_1 + C_2|\vartheta|)\vartheta \qquad (2)$$

The equation used in this model had the following coefficients to maintain a high-pressure drop in any condition in order to simulate a solid barrier:

$$C_1 = 100\ Pa * \frac{s}{m}\ ;\ C_1 = -1Pa * \frac{s^2}{m^2} \qquad (3)$$

Air recirculation set-up via sub-floor void connected to adjacent room where CRAC units are located. All room are joined in IES VE Microflo using 'create multi-partition' function with air supply and extract grilles are modelled as openings on surfaces.

The objective of this assessment was to evaluate two configurations of conditioned air flow supply: an upflow direction where air is supplied at ceiling level and down flow where conditioned air is supplied at above floor level. In addition to that, an optimisation of the air flow rate was required to be determined in order to maintain a maximum average air temperature of 30°C for a minimum of 4 hours. Due to the impossibility of simulating dynamical thermal behaviour in the Microflo module, it was necessary to run the dynamic thermal simulation in the Apache Sim module prior to the CFD simulation to determine critical room hourly temperature profile and associated boundary conditions. The below diagrams shows the server room layout with primary and back up cooling units location (Figures 4,5 and 6).

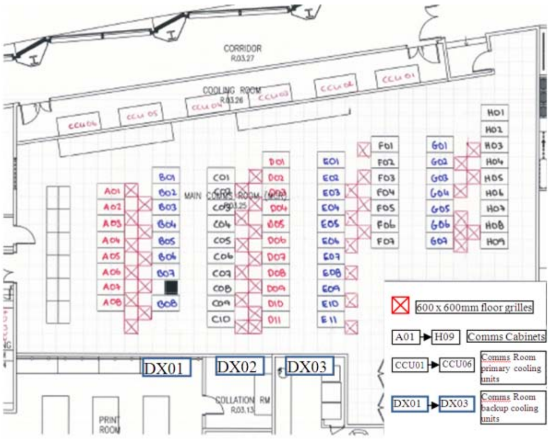


Figure 4 Primary and secondary cooling layout

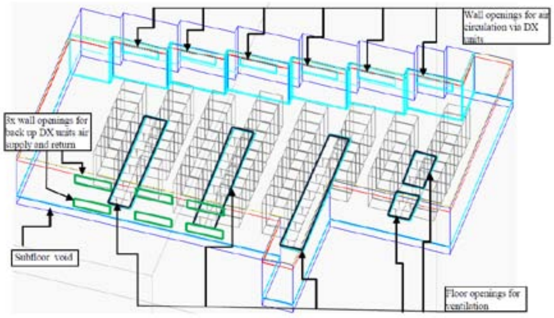


Figure 5 Floor grille diffusers configuration

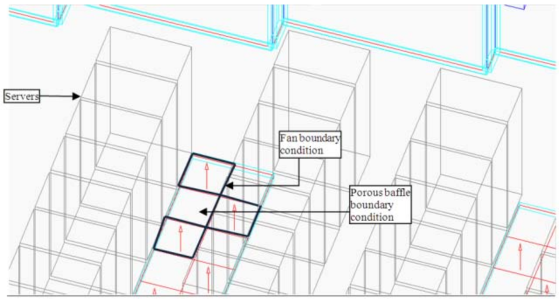


Figure 6 Fan and Porous baffle setup

The CFD analysis had a high level of complexity due to the configuration of two cooling systems with air recirculation provided by primary system at floor level through multiple floor diffusers and cool air supplied by secondary system. The grid meshing had to be set up very fine with minimum cell size of 0.07m to be able to detail air flow passing between server cabinets and through floor diffusers. The model had also to be simplified with a stepped approximation of the north sloping wall. The total number of iterations to achieve acceptable levels of numerical convergence was 6,000 iterations.

The CFD results indicated that the layout of the DX cooling system was not initially efficiently designed to deliver cool air in a balanced way to the comms room space. The reason behind the poor air flow distribution is the close proximity between the supply and return air grilles that leads to short circuiting the air flow of cool air. However, this system can be acceptable if requirement is only to operate secondary cooling system for a minimum number of hours to allow for repair of the primary chiller system. The CFD plots indicated that the upflow DX unit configuration is the most efficient to cool the room when back-up system is required. The total supply air flow of the secondary back-up system was optimized to 9,600L/s with a total cooling capacity of 111.9kW and supply air temperature of 15^{o}C.

Figure 7 and 8 below demonstrates the output presented to the client in the final report of this particular project, illustrating the secondary cooling units operation during a primary cooling system breakdown.

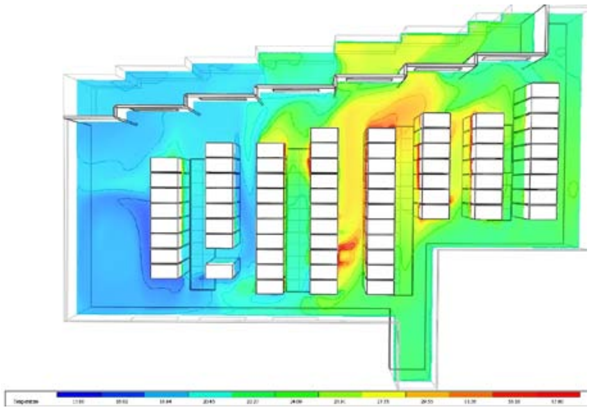


Figure 7 Temperature plot at z plane 1.5m above finished floor: upflow configuration

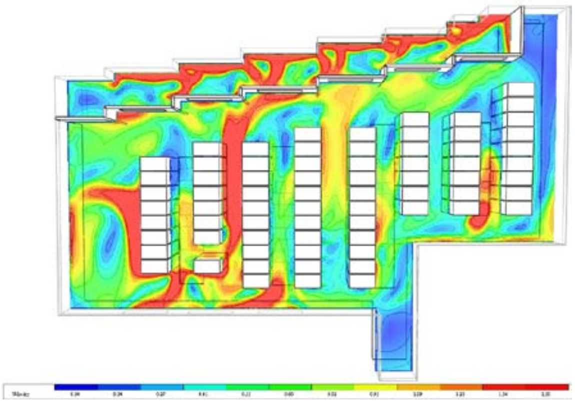


Figure 8 Air velocity contour plot at z plane 1.5m above finished floor: upflow configuration

This case study demonstrated that a simplified CFD model using the limited software resources available for this application could be sufficient to influence design decision significantly. The sizing of the secondary back-up DX units, air flow rates and location were optimized and design validated to operate according to client specifications.

<u>2 University of Queensland – Global Change Institute</u>

The Global Change Institute building is located in the University of Queensland campus and has been designed with several solar passive and renewable energy solutions to achieve a zero or possibly a negative carbon operational capability by exporting electricity back to the grid.

The key features of this landmark project are:

- Thermal labyrinth to pre-cool outside air;
- An air-handling unit with integrated latent and sensible heat recovery wheels and a desiccant wheel for cooling of the process air. Desiccant wheel heat supplied by hot water delivered by Evacuated Compound Parabolic Collectors (CPC) solar collectors;
- Cooled air delivered at subfloor level via floor grilles;
- In-slab cooling and passive chilled beams with chillers energy supplied by solar PV panels located on rooftop;
- Natural ventilation and wide temperature band.

<u>Main approach, lessons and outputs of the modelling</u>

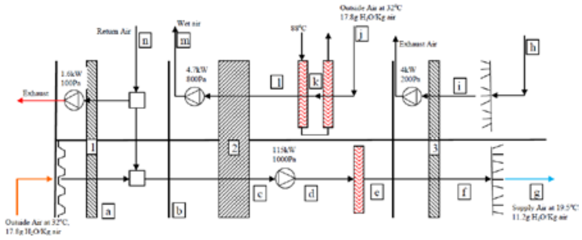
The University of Queensland Global Change Institute presented challenges to model and integrate all the HVAC elements due to the complexity of the systems and the unavailability of a solar hot water collector and desiccant wheel model in IES VE Apache HVAC module, which limited a full dynamic simulation.

The air handling unit in this building consists of a latent and sensible heat recovery wheel, followed by a desiccant wheel and a sensible heat recovery wheel. Fresh intake air is pre-cooled by an underground labyrinth and further cooled by air handling unit before being delivered to conditioned spaces at subfloor level via floor grilles. Desiccant wheel heat is supplied by hot water from roof top CPC evacuated solar collectors with an array of 305m2.The diagram of the entire system is shown on the image below. The air temperature and moisture content shown on the diagram and points (a) to (m) is only one point of the performance curves of this system.

The proposed CPC (compound parabolic collector) evacuated collectors with a total array of 412m2 and a supply return water flow rate of 1.6L/s were modelled in TRNSYS 16 to size stratified hot water storage tank , optimise solar collector area and solar collector installation angle. The objective of this solar energy model was to ensure supply of water at 90^{o}C to the desiccant wheel in order to provide maximum cooling during peak summer loads. Based on the solar collector simulation results, a simplified annual profile was created in IES VE to simulate the input water temperature into the desiccant wheel. The annual hot water temperature profile was based around discretized temperatures of 60^{o}C, 70^{o}C, 80^{o}C and 90^{o}C to reduce the number of points generated for the desiccant wheel's performance curve.

Figure 9 below shows the detailed diagram of the complete air-handling system:

(1) Energy recovery wheel (1) with a sensible effectiveness of 86.2% and latent effectiveness of 82.6%;
(2) Desiccant wheel (2) with a sensible effectiveness of 65.2% and a latent effectiveness of 0% (moisture added by cooling coil/ heating coil combination instead);
(3) Sensible heat recovery wheel (3) with a sensible effectiveness of 82% and a latent effectiveness of 0%;
(4) Humidifiers and heating coils from CPC evacuated solar collectors' hot water system.



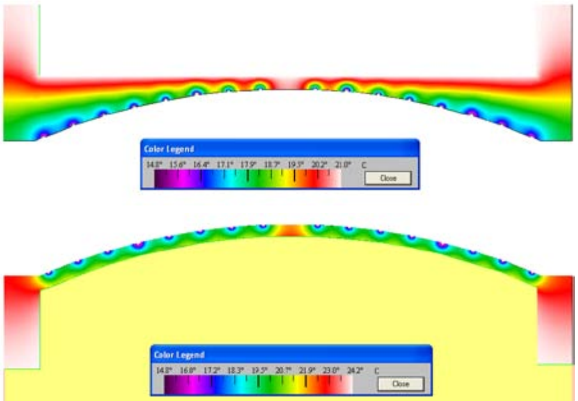
Point (b): Process air at 26.7°C and 13.3 g H_20/ Kg air (tempered with return air)
Point (c): Process air past desiccant wheel (outlet air) at 45.6°C, 7.4 g H_20/ Kg air
Point (d): Process air heat pick up after fan at 46.6°C and 7.4 g H_20/ Kg air
Point (e): Process air cooling by outside air at 43.2°C and 7.4 g H_20/ Kg air
Point (f): Process air past Energy Recovery Wheel at 29°C and 7.4 g H_20/ Kg air
Point (g): Supply air condition cooled by humidification process (80% RH)
Point (h): Outside air at a flow rate of 11,000L/s, 32°C and 17.8 g H_20/ Kg air
Point (i): Outside air cooled by humidification to 26.2°C and 20.3 g H_20/ Kg air
Point (j): Outside air at a flow rate of 4,400L/s, 32°C and 17.8 g H_20/ Kg air
Point (k): Outside air heated by Solar Hot Water to 39°C and 17.8 g H_20/ Kg air
Point (l): Outside air heated by Solar Hot Water to 82.6°C and 17.8 g H_20/ Kg air
Point (m): Wet air exhausted at 47.9°C and 28 g H_20/ Kg air
Point (n): Return Air at 26°C and 12.6 g H_20/ Kg air

Figure 9 –Air Handling system schematics

The desiccant wheel was modelled with a sensible heat recovery component and a cooling and heating coil to dehumidify the air. Manufacturing performance test data for 140 points were supplied by Seibu Giken (2011) with varying conditions of process air inlet temperature and moisture content and solar collector hot water temperature. Seibu Giken (2011) performance data was based on manufacturing testing and simulation results.

The desiccant wheel performance data was entered in Apache HVAC with a Multiplex loop of 140 adiabatic rooms. A controller measuring inlet process air temperature and moisture content, and hot water temperature diverts the air flow in the correct air stream of the Multiplex desiccant wheel loop. Two dummy rooms had to be added to the geometry of the model to allow for temperature and moisture content readings in different points of the system. The temperature and moisture content of the process air outlet immediately after the desiccant wheel cooling were specified based on manufacturers performance data and calculated using a weekly profile and a formula derived from a linear regression of combined points with same process air inlet temperature.

Additional cooling is provided by in-slab cooling coils and passive chilled beams with cold water supplied by chillers powered by PV panels. The undulated variable thickness of the slabs required a finite element calculation of the equivalent conductivity of the material, using THERM 6.3 from the Lawrence Berkeley National Laboratory (LBNL) (shown in Figure 10). The chilled slabs were modelled as individual thermal zones with a maximum height of 1mm to minimise the effect of air volume. Chilled Ceiling components from Apache HVAC were used to simulate the water cooling effect by setting the hydronic cooling component to 100% convective. This methodology for in-slab cooling was developed from the report of Moore (2008).



5.1137 Bottom Thermal transmittance or U-value of slab and tubes at 10 K delta-T, W/m^2-K
0.1956 Overall resistance, including water film, tubing, and air film at exposed surface, m^2-K/W
0.1067 Air film resistance on exposed cooling surface, m^2-K/W
0.0003 Water-film resistance at tubing inside surface (forced convection, 1.8 m/s flow), m^2-K/W
0.0886 Resistance for slab plus Polybutylene tubing without air or water films, m^2-K/W

3.3166 Top Thermal transmittance or U-value of slab and tubes at 10 K delta-T, W/m^2-K
0.3015 Overall resistance, including water film, tubing, and air film at exposed surface, m^2-K/W
0.1067 Air film resistance on exposed cooling surface, m^2-K/W
0.0003 Water-film resistance at tubing inside surface (forced convection, 1.8 m/s flow), m^2-K/W
0.1945 Resistance for slab plus Polybutylene tubing without air or water films, m^2-K/W

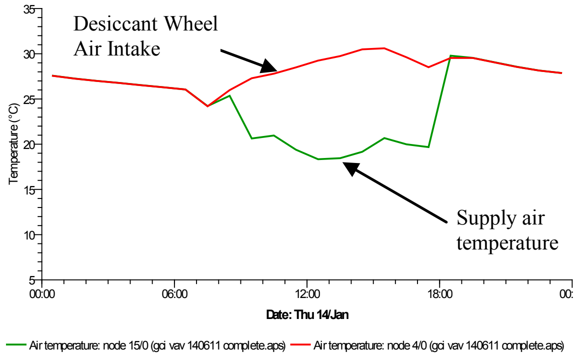
Adjusted conductivity using THERM 2-D heat transfer model **GCI**

12.0 Hydronic tubing inside diameter, mm
2.5 Hydronic tubing wall thickness, mm
0.134 Hydronic tubing spacing in the slab, m
0.384 Hydronic tubing conductivity (Polybutylene tubing), W/m-k
1.40 Slab conductivity, W/m-K
2.0118 Total Thermal transmittance or U-value of slab and tubes at 10 K delta-T, W/m^2-K
0.4971 Overall resistance, including water film, tubing, and air film at exposed surface, m^2-K/W
0.2135 Air film resistance on exposed cooling surface, m^2-K/W
0.0006 Water-film resistance at tubing inside surface (forced convection, 1.8 m/s flow), m^2-K/W
0.2830 Resistance for slab plus Polybutylene tubing without air or water films, m^2-K/W
3.533 Conductance for slab plus Polybutylene tubing without air or water films, W/m^2-K
0.295 Slab thickness at perpendicular heat transfer path (center of tubes to cooling surface), m
$\boxed{1.042}$ Adjusted conductivity for chilled slab material in VE model, W/m-K
89% Faction of adjusted conductivity value obtained using simple spreadsheet model

Figure 10 –U-value calculation of chilled slab

The results of the Global Change Institute HVAC model were assessed with regards to the simulation accuracy of the desiccant wheel model to the manufacturers' performance curve data. The simulation results of the in-slab cooling radiant output into conditioned zones were also used to calculate thermal comfort indicators required for Green Star submission (ASHRAE 55-2004 thermal comfort criteria for naturally ventilated spaces) (ASHRAE,2004).

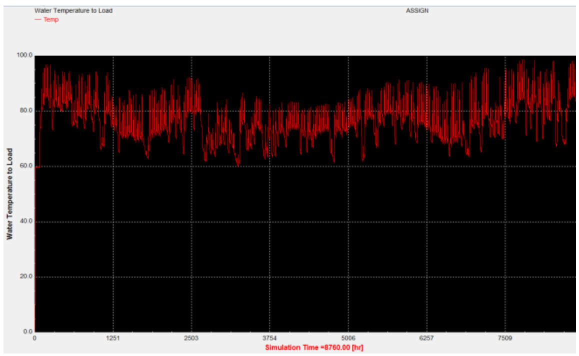
Graph 1 below shows the operation of the desiccant wheel on a typical summer day with delta temperature of air inlet and outlet to desiccant wheel system.



Graph 1 –Plot of supply and ambient air temperature

The IES VE HVAC model of the University of Queensland Global Change Institute was a crucial part of the system design and sizing due to little industry experience with the integration of the solar thermal collectors with the desiccant wheel system (which does not have constant latent heat transfer effectiveness). Therefore the energy model has influenced the design greatly and gave great confidence to the design team that the thermal comfort levels of a proposed naturally ventilated Zero Carbon building, without an active air conditioning system, would be acceptable for an educational institution. The desiccant wheel model in itself dealt with the software limitation of constant latent heat effectiveness by mapping the performance curve of 140 points in a multiplex loop with cooling and heating coils to dehumidify the air. The deviation of the modelled desiccant wheel system to the manufacturers testing data was 3.7% for the air temperature leaving the desiccant wheel and 12% for the relative humidity at the same point.

The solar thermal collectors (CPC evacuated tubes) simulation has also influenced the sizing of the system and validated the requirements of hot water supply to the desiccant wheel. An annual temperature plot of the hot water temperature simulated in the TRNSYS 16 model is shown below, in Graph 2, with clear indication that water temperature would be peaking above 90°C from December to February (summer months in the Southern Hemisphere).



Graph 2 –TRNSYS hot water temperature to desiccant wheel

The U value adjustment of the chilled concrete slabs had also to be simplified due to the sloping cross section. The U-value adjustment calculated by THERM 6 via finite element analysis was used to calculate the conductivity value (k) of the pre-cast concrete material dividing the overall cross section area by its length to obtain an average height.

c) Quad5 - Auckland International Airport

The Quad5 Auckland International Airport building is a landmark green building project aiming for a 5 star Green Star New Zealand Office 2009 (NZGBC,2011) certification in 2011. This 5 storey building has a total Green Star assessable area of 4024 m2 consisting of office spaces in the upper floors, cyclist facilities and change rooms in the basement level and a café/reception in the first level.

The base building HVAC model presented in this paper is a VRF system with sensible heat recovery integrated to outdoor air handling units. The VRF condenser heat recovery and part load performance model was entirely modelled in the Apache HVAC module of IES VE. A detailed HVAC model was required to maximise the number of credit points achieved under Energy and GHG emissions by reducing the margin of error in the HVAC system model.

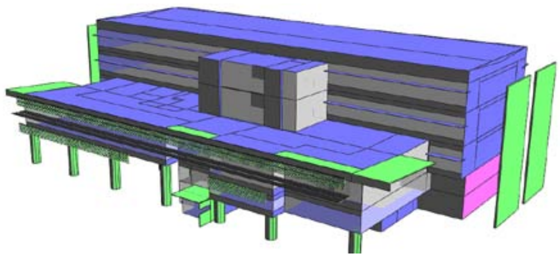


Figure 11 - IES VE model geometry of Quad5 - Auckland International Airport

Main approach, lessons and outputs of the modelling

This case study shows an example of the simplification of the architecture geometry created in REVIT with core and perimeter zones, as shown on Figures 13, 14, 15 and 16. The HVAC model has a fresh air intake tampered with return air via a sensible heat recovery element. The HVAC system comprises of a Variable Refrigerant Flow (VRF) system with condenser heat recovery loops. The part-load performance of the VRF outdoor units were entered in the IES VE Apache HVAC model as a DX cooling component and heat pump connected to cooling and heating coils respectively.

The interior spaces of this building presented a high level of complexity with spiral staircases, enclosed offices throughout and a high degree of complexity of the exterior shading devices. As an example, the detailed architecture level 2 floor plan, shown on Figure 12 below, was changed to a much more simplified energy model with core, north and south perimeter zones. Exterior shading devices on the other hand have been drawn as per Revit architecture model to minimise uncertainties in the solar gains and shading calculations.

Figure 12 - Quad5 Level 2 Floor Plan

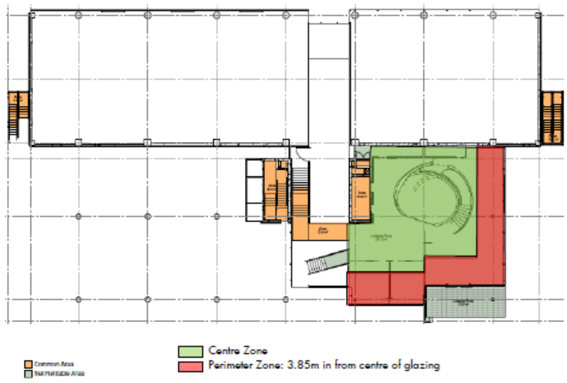


Figure 13 - Quad5 Level 1 - Green Star assessed area: 435m2

Figure 14 - Quad5 Level 2 - Green Star assessed area: 1748m^2

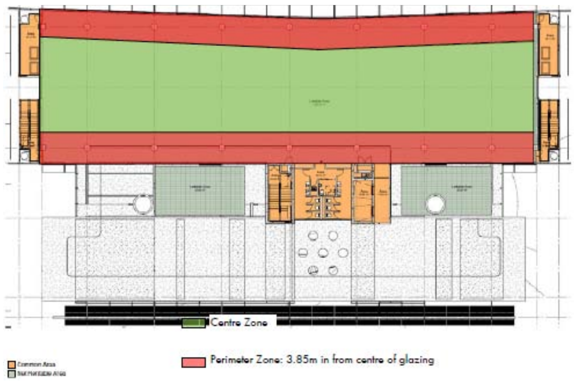


Figure 15 - Quad5 Level 3 - Green Star assessed area: 1374m^2

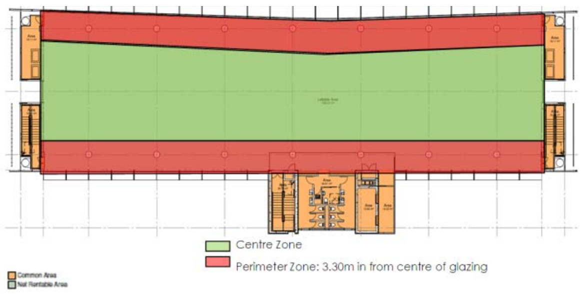


Figure 16- Quad5 Level 4 - Green Star assessed area: 1250m^2

The Apache HVAC model shown below (Figure 17) has a series of 5 outdoor heat pumps used for the heating side of the VRF system and air handling units and are connected to the heating coils elements. The cooling side of the VRF system and air handling units comprises of DX components connected to the cooling coils. The air-conditioned zones are multiplexed around the supply and recirculation air loop with air return from conditioned zones directed to ceiling void spaces. A series of controllers are used to specify air flow rates and required temperature after coils and heat recovery temperature targets.

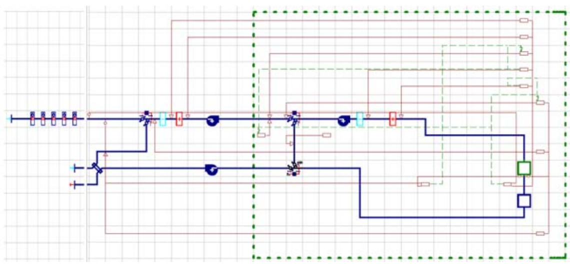
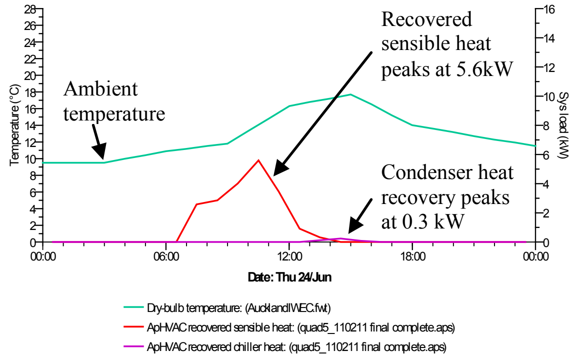


Figure 17-IES VE Apache HVAC model of VRF system with condenser heat recovery, outside air sensible heat recovery and economy cycle

The energy report of this project was a requirement for the Green Star New Zealand 2009 (NZGBC,2011) submission and therefore the VRF system was modelled as per manufacturers' part-load performance testing data. The results were significantly important to the client as it indicated the VRF system was not shifting heat from cooling zones to heating zones due to overlapping with sensible heat recovery in early mornings and late afternoons. There was not sufficient overlapping of heating and cooling requirements to justify a VRF condenser heat recovery strategy.

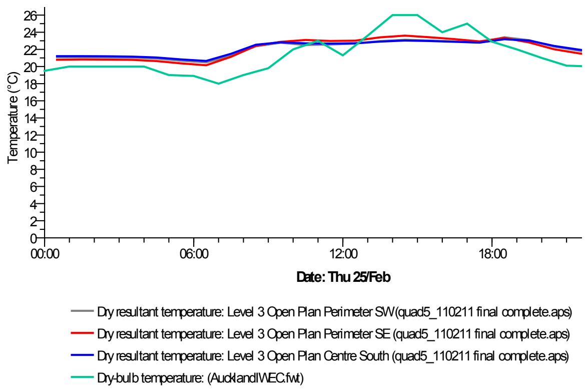
PMV analysis was also carried out to assess thermal comfort credit. Based on IES VE model, changes to the original HVAC layout were introduced to improve PMV levels from within range of -1 and 1 to -0.75 and 0.75.The design was fully validated final energy model result predicted a 5 star Green Star rating.

Graph 3 shows a typical winter day with results showing a small VRF heat recovery of 0.23kW at 2.30pm and a much greater sensible heat recovery from heat recovery wheel that peaked at 5.6kW at 10am.



Graph 3 – Sensible and condenser heat recovery plot

Graph 4 below shows the dry resultant temperature of the two perimeter zones and the core zone on level 3 for a typical summer day. The perimeter zone facing the northeast direction has a higher predicted temperature during the day due to higher solar gains through windows. This is the reason the model was simplified and divided in core and perimeter zones to avoid averaging of higher temperatures near windows and lower temperatures away from windows.



Graph 4 – Plot of core and perimeter zones temperature for a typical summer day

SUMMARY OF MAIN POINTS OF INTEREST

	CS1	CS2	CS3
Outline	1 Coleman st	UQ – GCI	Auckland Airport
Aim	size system	size system/ thermal comfort	Green Star report
Limitations	Cartesian grid,steady-state	HVAC model limitations	Complex architect's model
Solutions applied	Segment curves, use dynamic simulation for boundary conditions	Multiplex looping system to simulate testing data	Simplified thermal zones to core and perimeter

Main lesson	Congruent geometry	Use of multiplex for multiple efficiency paths	Model actual thermal zones

CONCLUSION

This industry application paper demonstrated the successful development of innovative models using a combination of different simulation software algorithms and outputs to work around software limitations. A case study of a desiccant wheel system, using existing components in IES VE and detailed performance data from manufacturer, illustrated how results could validate system design and influence the design process. The maximum deviation of 12% for the relative humidity of the air leaving the desiccant wheel was due to the approximation of the dehumidification process with cooling and heating coils in series. Further development to the methodology developed by Moore (2008) with regards to in-slab cooling and adjusted concrete U value was also achieved. A complete integration with the solar collectors' performance was achieved by exporting results from TRNSYS into IES VE.

Simplifications of the CFD and HVAC models were necessary to achieve reliable results within a reduced timeframe, which is imperative from a business case and productivity standpoint. The industry-driven approach of the case studies presented in this paper demonstrated the ability to model systems and solutions despite software limitations. Building simulation models are presented with a fit for purpose approach where simplifications are essential to keep computational and development costs low and model accuracy specific for each application. It is demonstrated that a simpler CFD model with Cartesian grid coupled with dynamic thermal models are sufficient to test conceptual solutions performance. On the other far more complex HVAC models that incorporate heat recovery processes, part-load VRF performance, airflow balance to conditioned zones, dehumidification process and control systems are required to predict annual energy consumption for Green Star NZ (NZGBC,2011) and NABERS (NABERS,2011) certification or system sizing.

In all case studies, the approach for modelling was customised to achieve the right outcome with regards to influencing the design decision. The simulations detailed in this paper are industry case studies using commercial softwares and do not have an academic nature in terms of modelling the governing equations of dehumidification, adsorption and heat transfer processes. Future work can be developed to identify errors and risks associated with simplification of models when compared with actual post-occupancy data of thermal and system performance. IES VE simplified CFD and HVAC models could be compared to far more sophisticated simulation tools such as ANSYS FLUENT and Matlab/Simulink for validity of results and accuracy.

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