DYNAMIC SIMULATION STUDY
COMPARISON OF VAV, CB & UFAD SYATEMS

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Conducted by: Team Catalyst
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www.arbsfoundation.com.au

CONTENTS

1 EXECUTIVE SUMMARY

This research study commissioned by the ARBS Foundation, is a comparative dynamic simulation analysis of commonly used Heating, Ventilation and Air Conditioning (HVAC) systems, namely Variable Air Volume (VAV), Under Floor Air Distribution (UFAD) and Passive Chilled Beam (PCB) systems from an energy efficiency perspective. These are the key system configurations prevalent in Australian buildings.

The study has been carried out by Team Catalyst, and to the best of our knowledge, this is the first study where the energy performance of all three systems has been carefully modelled, when applied to the same hypothetical building. The study would be of interest to the HVAC industry in that it has been carried out by experienced practising engineers, involved in the design, specification and ongoing monitoring of HVAC systems as installed in energy efficient buildings.

It is noted that the authors have presented the predicted energy efficiency outcomes in for each of these systems (summarised in Table 1 below) as modelled for the Sydney climate using system configurations and control strategies detailed in this report. It is not the intention of the authors to promote or discourage use of any one type of HVAC system. It is recognised that there are many variations of these system types, and combinations of systems that are installed in buildings; and that there are many factors that need to be considered in the HVAC system selection process for a specific building project.

Table 1: Summary of predicted HVAC system energy intensities

SYSTEM TYPE →	*UFAD*		*STD-VAV*		*PCB*	
HVAC END USE ↓	*Electricity [kWh]*	*Natural Gas [kWh]*	*Electricity [kWh]*	*Natural Gas [kWh]*	*Electricity [kWh]*	*Natural Gas [kWh]*
HVAC totals	197,817	19,098	195,294.0	19,261.1	253,333.4	43,858.4
Energy intensity, kWh/m2-yr	*24.1*		*23.8*		*33.0*	

The predicted performance of all three HVAC systems modelled above is estimated to be well with the operational requirement of a 5 star Base Building NABERS rating for office buildings.

This report, in following sections, provides a break down of the predicted energy consumption for each HVAC sub-system (eg., pumps, fans, chillers etc), along with a series of system level insights from the modelling exercise, and from anecdotal observations made by experienced HVAC engineers that are consistent with the simulation study outcomes.

In addition to presenting the predicted performance of these three ‘typical’ HVAC system configurations, the study sets the framework for further investigations in the future of other system variations that can also evaluate combinations of various HVAC system types, a trend which seems to be gaining momentum in many recent ‘green’ buildings.

2 INTRODUCTION

Choosing an air conditioning system for a building project is primarily based on capital investment costs, and in many cases, on a life cycle costing (LCC) basis. Whilst historical costs may be available for maintenance and replacement of components associated with each type of system, energy costs are not readily available for an LCC analysis, since applications and operational periods vary significantly. Energy costs are a function of the annual energy consumption of the HVAC system configuration and on the system type.

Three Heating Ventilation and Air Conditioning (HVAC) system types, prevalent in larger commercial buildings in Australia, are:

- Variable Air Volume (VAV) systems
- Under Floor Air Distribution (UFAD) systems, and
- Chilled Beam (CB) systems

It is acknowledged that there are variations in the applications of these systems to meet specific requirements for each client.

VAV installations are in a majority of buildings in Australia's major cities, followed by CB systems that came into vogue some 20 years ago. UFAD systems have become prominent in recent developments. It is appreciated that with each of these systems there are variations in terms of applicability. For example, there is a low temperature VAV system variation utilising low supply air temperature; UFAD systems incorporating VAV terminals in the under floor air supply arrangement, etc.. Some of the more recent building projects have installed combinations of these HVAC systems such as VAV and CB systems. Different combinations adopted in chilled water plant configurations include use of heat exchangers or dedicated chillers in the supply of "high temperature" chilled water to supply CB installations. However, this study has not covered any of these variations; it has primarily concentrated on predicting the performance of 'typical' VAV, UFAD and CB systems. These are described in detail in later sections of this report.

Whilst there are various previous evaluations comparing energy efficiencies of VAV vs CB, and VAV vs UFAD, these have been limited to specific buildings and in most instances carried out by Consultants responding to specific project briefs. From an academic viewpoint there are various published papers comparing different aspects of these HVAC systems. However, from carrying out a literature review it is apparent that there has not been a study comparing the energy intensities for these systems for the same building. We have modelled these three HVAC systems as incorporated in a hypothetical building that has been described with a high performance building envelope, internal loads from NCC Section-J (building envelope) and NABERS operational schedules.

The study models a practical, and, as far as possible, a "typical" configuration of each of these systems, and these have been described in some detail in the report. The systems have been applied to a 10 storey high tower building located in Sydney, with identical ("typical") floor plate organisation. For modelling purposes, each floor is divided into five thermal zones; four perimeter zones in each cardinal orientation, and one interior zone. These "whole building" simulation models are used to predict the annual energy performance for each of the HVAC systems. The results have been reviewed and summarised, and conclusions are reported here. The intention has been to present the energy intensities as per the modelling results without 'rating' the systems in any order.

The dynamic simulation analysis is carried out using the well respected EnergyPlus (https://energyplus.net/) simulation engine developed and maintained by the USDOE (US Dept of Energy). The engine is freely available in the public domain and is extremely well documented. It combined[1] that best features of DOE-2.1E and BLAST, the previous generation of dynamic simulation engines. Many well respected research laboratories and universities around the world have contributed to it's development. For this study we have used the DesignBuilder GUI (https://www.designbuilder.co.uk/) to access the power of EnergyPlus.

We believe this study is unique whereby for the first time, results of comparative energy efficiency analyses with a practical approach are being delivered to the industry. It is to be noted that the authors have presented the outcomes for each of these systems. It is not the intention of the authors to promote or discourage use of any one type of HVAC system. It is recognised that there are many other factors that need to be considered in the selection process for a specific building or project.

The study is also somewhat unique in that this comparison (in Australia) has been carried out by a team of practising consulting engineers with practical experience and expertise in the design, review and delivery of energy efficient HVAC systems in larger buildings. Many studies focus on the energy efficiency of the building envelope (which is the first step) but do not consider the practical limitations of systems and components when modelling environmental control (HVAC) systems. Predicted results in such cases can lead to optimistic outcomes. A concerted effort has been made to incorporate practical control strategies and incorporate system and component limitations into the modelled system representations.

There is also constant discussion of the pros and cons of each of these systems from all sectors of the building industry. This study compares the technical aspects of these systems, and predicts their energy efficiency performance when applied to the same hypothetical building. The authors have also made some comments on other aspects, for example, thermal comfort performance of each these systems, but these have not been explored in any detail as part of the study (although the data has been generated along with other outputs).

[1] Crawley, D. B., Lawrie, L. K., Pedersen, C. O., Liesen, R. J., Fisher, D. E., Strand, R. K., ... Huang, Y. J. (1998). *Beyond DOE-2 and BLAST: EnergyPlus, the New Generation Energy Simulation Program*. In Commercial Buildings: Technologies, Design, and Performance Analysis: Proceedings of the 1998 Summer Study on Energy Efficiency in Buildings (Vol. 3, pp. 3.89-3.104). Washington, DC: ACEEE

3 METHOD

3.1 BUILDING MODEL DESCRIPTION

A hypothetical building model (similar to the reference building type "A" used for the National Construction Code, Section-J stringency analysis developed by Team Catalyst for other projects) has been used. The general arrangement of the building model is described below:

- square floor plate, 30m on all sides
- oriented to cardinal directions
- greenfield site (no shading from adjacent buildings)
- 10 levels (ground plus 9 levels), plus an unconditioned basement floor
- a floor-to-floor height of 4.0 m, ceiling at 2.7m
- 2m high vision glazing at 700mm sill height (façade WWR = 50%)
- high performance glazing systems, SHGC=0.3 and U=3.4 W/m2-K
- external walls modelled to be R1.4
- 200mm concrete roof construction insulated to R2.7
- ceiling below non-conditioned space insulated to R1.35
- exposed floors modelled as uninsulated slab
- 200mm thick concrete floors

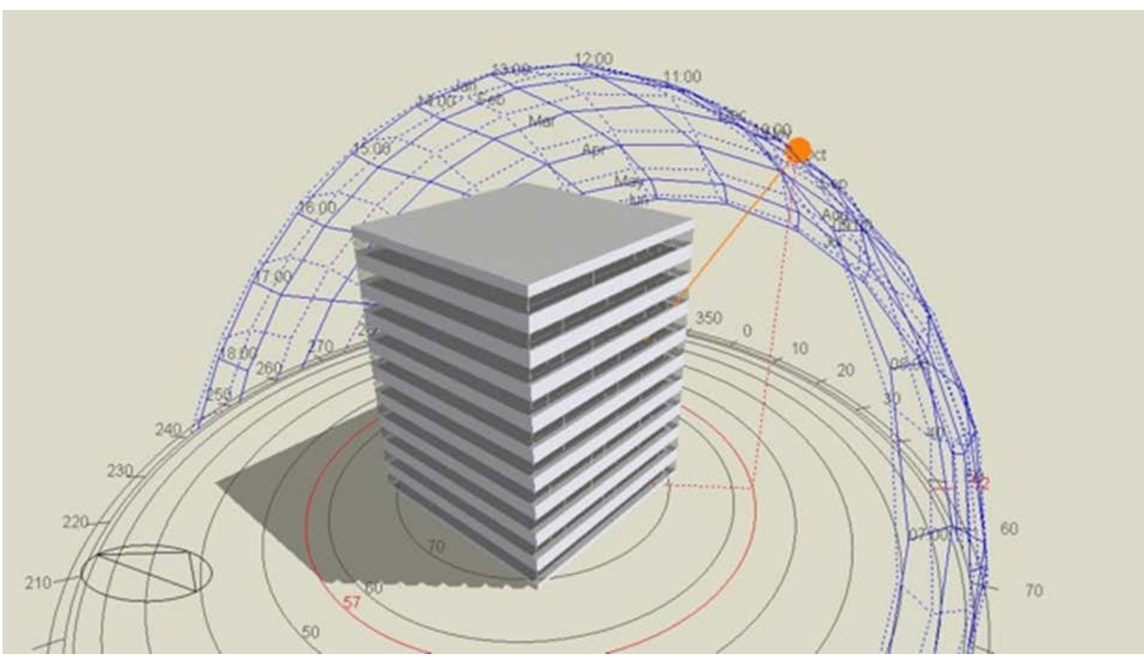

Figure 1: 3D visulation of the building with superimposition of annual sunpath for Sydney

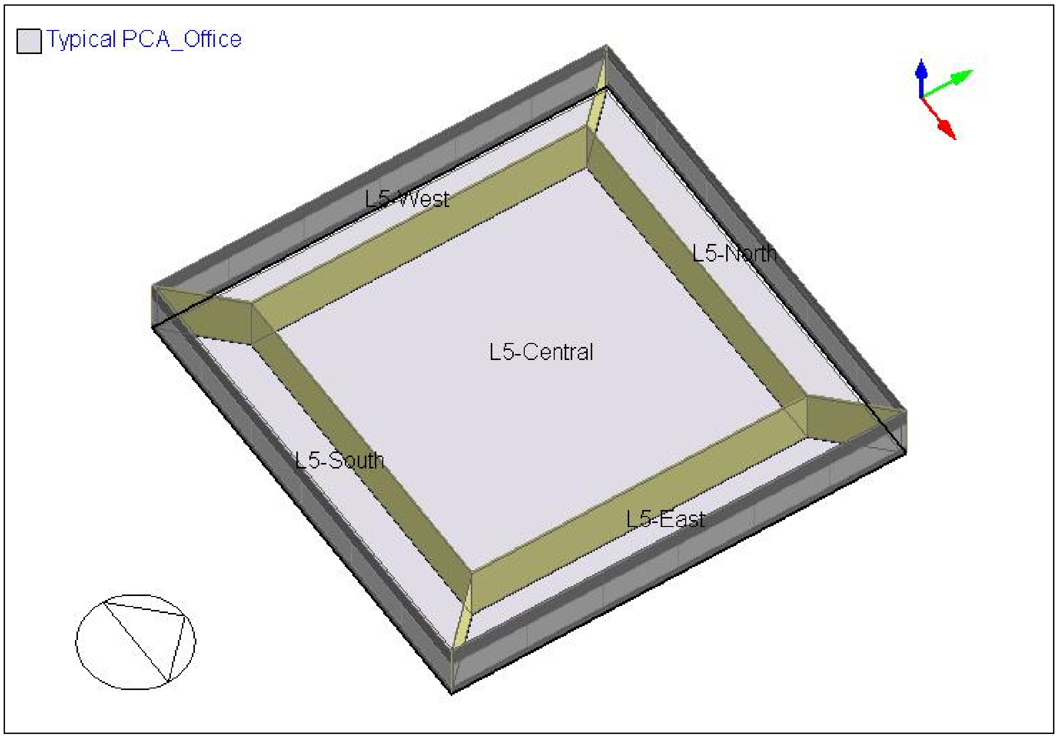

Figure 2: Zoning for typical floors; the zones are separated by “virtual walls”, ie., heat transfer boundaries, typically used in HVAC design for open plan offices

Table 2: Internal loads

Internal Load	• Occupancy density 1 per 10 m^2 as per PCA Guide (2012) Standard office occupancy • Occupant load of 130 watts per person (55W latent/ 75W sensible) • Lighting load: 8 watts per m^2 (this is 1 W/m2 less than current NCC Section-J, but about 2 W/m2 higher than most current designs) • Equipment load: 15 watts per m^2 (as per current NCC Section-J allowance for modelling)
Schedules	• Occupancy schedules: NABERS occupancy schedule, starting at 8am (at 15% occupancy) to 6pm • HVAC schedule: 7am to 6pm (1 hour prior to occupancy) • Lighting and equipment schedules in accordance with NABERS schedules for lighting and equipment
Outside Air Rate	10 L/s per person as per AS 1668.2 without filtration for offices
Indoor Temperature	22.5C +/- 1.5 (generally 21C for heating and 24C for cooling)
Design Criteria	0.4% confidence level, ASHRAE monthly design criteria, dry bulb priority (listed in a later section of this report)

From the descriptions above, we note that the building envelope (or building fabric) for this hypothetical building is capable of high performance outcomes. Many 4.5 and 5 star NABERS rated building that maintain this level of operational perform year-after-year can be found in all metropolitan cities in Australia with similar levels of building envelope performance and internal loads.

3.2 HVAC PLANT DESCRIPTION

3.2.1 CHILLED WATER PLANT

Large HVAC installations can be separated into the plant (or water) side and the zone (or air) side. All three buildings were modelled with the same configuration of HVAC plant *as far as practically possible*, as described below. The total thermal load for the building has been estimated to be in the range of 1,000kW. Therefore, the chilled water plant (CHWP) loop for all the three air-side systems was modelled with three water cooled chillers in parallel configuration. The chillers were explicitly sized at 2 X 400kW + 1 X 200kW (low load) water cooled chillers, with a minimum unloading ratio of 20% and identical part load performance curves which considered the impact of condenser water temperatures, chilled water temperatures and chiller part load ratio in estimating chiller efficiency (COP) at each time step. Each chiller has been modelled design coefficient of performance (COP) of 5.09 and with minimum part load ratio of 30%. Each chiller has been modelled with a dedicated chilled water pump. The pumping arrangement is set as constant volume primary flow with a 250 kPa head.

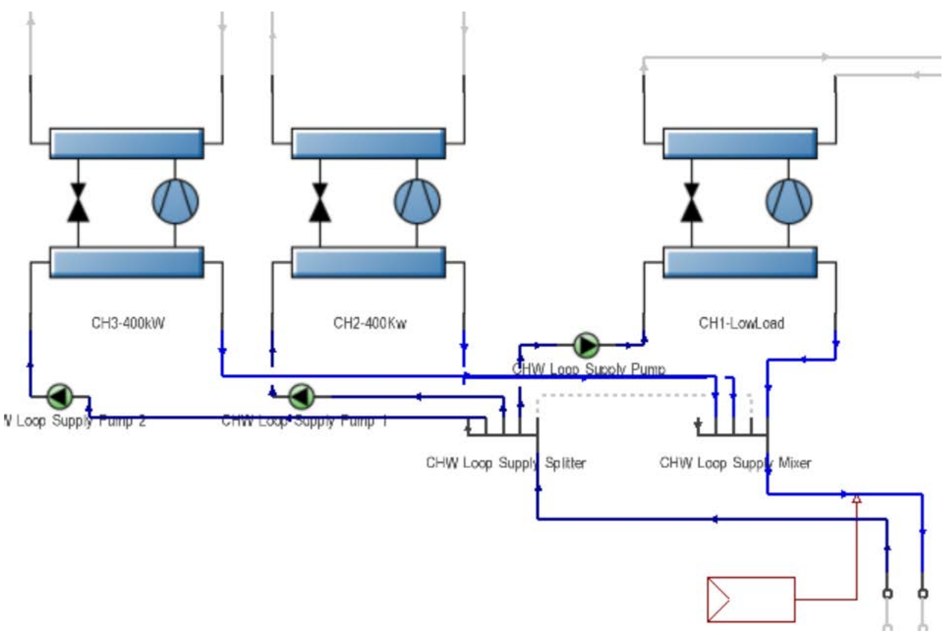

Figure 3: HVAC plant – chilled water (CHW) loop

It is noted that the selected chiller sizes (400 kW and 200 kW) are probably too small for practical centrifugal machines, and are at the lower limit for screw chiller technology. However, they are appropriate for the simulation study at hand, since:

- identical chiller machines are being used for all three systems
- part load performance of the chillers being modelled are identical, with respect to chilled water and condenser water reset, and chiller part load ratio
- the equations are scalable, and not impacted by absolute chiller size.

The above strategy allows us to review the performance of the air/zone side HVAC system and its impact on the chilled water plant and other electrical and gas consuming components.

Chiller sequencing has been modelled using a “swing chiller” sequencing strategy.   The low load chiller carries the building till it’s capacity is exhausted, when CH2 is energised (and the low load chiller is switched off). When CH2 capacity is exhausted, the low load chiller is also energised. When the capacity of these two chillers is exhausted, then CH3 is energised and the low load chiller is switched off. In the final stage all three chillers will run if required.

The chilled water loop split has been modelled to be a fairly typical 6/12C for the VAV system configuration.

The chilled water plant configuration for the CB system modelled uses a heat exchanger (HX) to reset (increase) chilled water temperature to the passive chilled beams (PCB) in the conditioned spaces. Pumps on both sides of the HX are used to control the temperature of water supplied to the PCB to 16C and return at 18C (design). The main chilled water loop itself runs at the standard 6/12C split.

The chilled water plant for the UFAD configuration runs at an 8/14C split, since the AHU is trying to maintain a supply of 19C from the underfloor plenum. Realistic plenum constructions have been modelled, and the AHU supply is required to be set to be 2C lower (about 17C) to compensate for the heat gain from underfloor plenums.

3.2.2 HEAT REJECTION PLANT

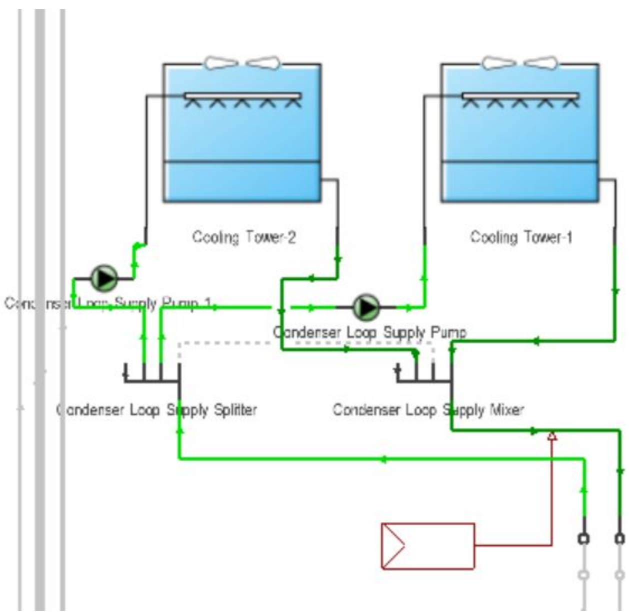

Figure 4: HVAC plant – heat rejection or condenser water (CW) loop

The heat rejection system for the HVAC plant has been modelled to be two equal sized cooling towers, connected in parallel with a simple single speed fan and cycling control operation. The sump water is controlled to follow ambient wet bulb temperature with an approach of 3C. Each cooling tower has it's own dedicated constant volume pump designed to meet a 180 kPa head. Cooling tower sizing is based on a 29.5C/35C loop split.

3.2.3 HEATING HOT WATER PLANT

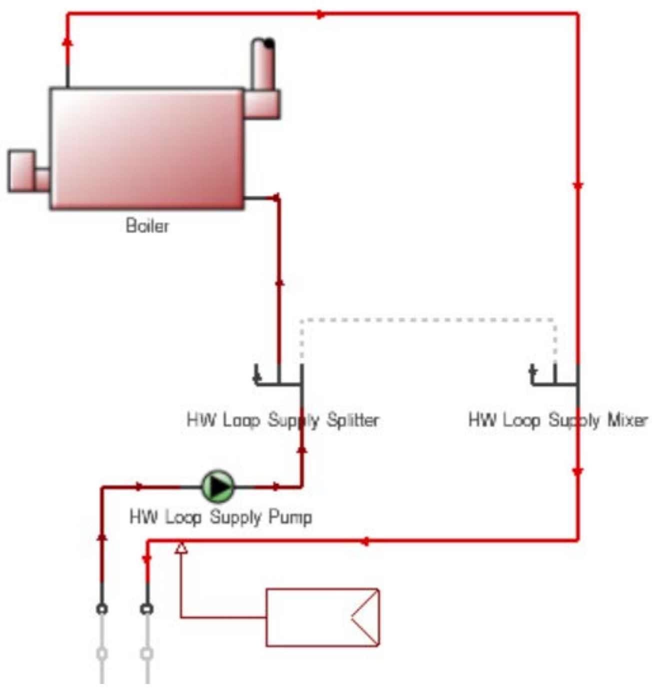

Figure 5: HVAC plant – heating hot water (HHW) loop

The heating hot water loop has been modelled to be a single natural gas fired non-condensing boiler with an 80% efficiency. A constant volume pump circulates the hot water across the system and it is designed to meet a head of 250 kPa.

3.3 VAV SYSTEM DESCRIPTION

The Variable Air Volume (VAV) is modelled as a series of air loops (AHUs) which serve various thermal zones. Each thermal zone has a VAV terminal unit that is fitted with a hot water reheat coil.

A “face zone” system has been modelled, where each “face” or façade of the building is supplied by a dedicated air loop (AHU). Therefore, there are four (4) perimeter air loops with dedicated variable volume AHUs for each orientation (North, South, East, West) and an internal zone air loop with it’s own dedicated variable volume Air Handling Unit (AHU). AHUs have been designed to meet a Total Static Pressure (TSP) of 600 Pa, including the resistance for internal coils and filter sections.

The Supply Air Temperature (SAT) is allowed to vary between 13C to 18C based on cooling load. The simulation employs a “warmest setpoint manager” that resets the cooling supply air temperature according to the cooling demand of the warmest zone. For each zone in the system at each system timestep, the manager calculates a supply air temperature that will meet the zone cooling load at the maximum zone supply air flow rate. The lowest of the possible supply air temperatures becomes the new supply air temperature setpoint, subject to minimum and maximum supply air temperature constraints. The resulting SAT setpoint is the highest SAT that will meet the cooling requirements of all the zones.

Zone level VAV boxes are equipped with heating hot water coils that can reheat the supply air to a maximum reheat temperature of 29°C. VAV boxes in the internal zones are set with minimum turn down capability to 40% of design air flow, and those in the perimeter zones to 30% of design air flow.

“Low temp” variants of VAV systems have not been modelled for this study, and could be dealt with in a future exercise.

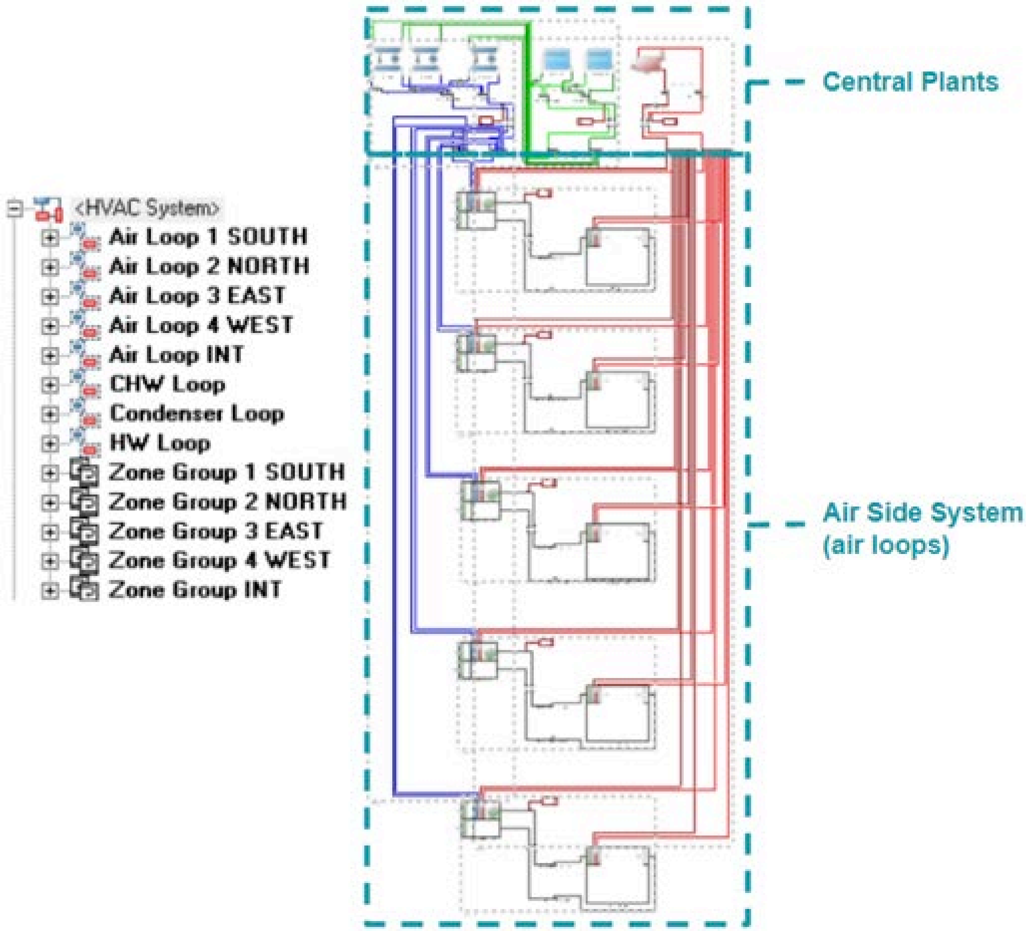

Figure 6: Schematic of variable air volume (VAV) system as modelled

3.4 CHILLED BEAM SYSTEM DESCRIPTION

Chilled beam systems meet the cooling requirement in a space using a combination of air and water. The air system provides tempered outside air to the air-conditioned space. Chilled water is reticulated to “chilled beams” in the space to deal with sensible cooling load. Chilled beams come in two flavours, active (ACB) or passive (PCB) beams. This study uses the passive chilled beam system configuration where high(er) temperature water is reticulated to flat metal plates at ceiling level. Local convection currents from the PCBs provide sensible cooling of the space.

Two AHUs provide “primary” tempered outside air to the spaces, one to the interior zone, and another to all the perimeter zones. The AHU cooling coils are supplied with chilled water at the standard 6C/12C split, and these are referred to as the “low temperature” cooling coils. AHUs have been designed to meet a Total Static Pressure (TSP) of 600 Pa, including the resistance for internal coils and filter sections. The cooling coil is designed to provide dehumidification and cooling, limiting the supply air absolute humidity to 0.012 kg/kg. This control strategy eliminates the risk of condensation in the space, the greatest operational risk for a PCB system. Reheat and space heating load is also met by the AHUs, and a heating coil has been included for this purpose. AHU supply temperature is allowed to vary between 14C and 29C SAT using a “warmest” control strategy as described earlier, that is, the highest possible SAT is used to provide the most energy efficient operation of the system. AHU cooling coils are sized based on ventilation load.

The passive chilled beams (PCB) in the tenant spaces are supplied with high(er) temperature chilled water, with a design loop split of 16C/19C. A heat exchanger is used to control the inlet water temperature to the PCBs. A variable speed pump controls the flow of chilled water to the passive chilled beams. The PCBs are sized to take up the bulk of the “space” load.

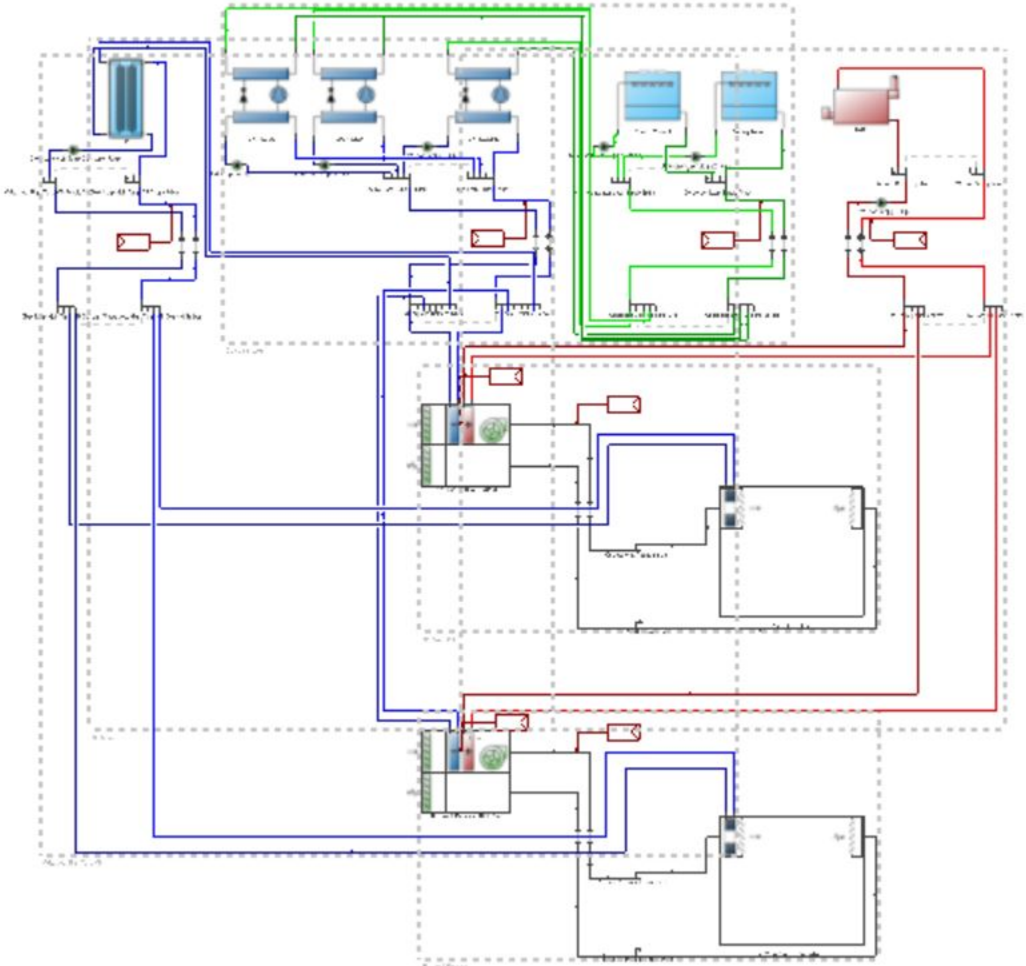

Figure 7: Schematic of passive chilled beam system as modelled

3.5 UNDERFLOOR AIR DISTRIBUTION SYSTEM DESCRIPTION

The Under Floor Air Distribution (UFAD) system is an off-shoot of the original Displacement Ventilation (DV) system. Conditioned air, in the temperature range of 19C to 21C is delivered to the zones to be conditioned from an underfloor plenum. The underfloor plenum is under a small positive pressure, typically 30 Pa, and this allows for reductions in fan energy use. Each floor plate has been zoned in a manner similar to the VAV design, with four perimeter zones and one interior zone. The underfloor plenums have also been zoned with baffles, so the supply air from each AHU is kept separate until delivered to the conditioned areas. The AHU fans have been modelled to 200 Pa to account for pressure loss from filters and coils, and the duct run to the low pressure plenum. Supply air temperature is controlled at the AHU outlet so that the underfloor plenum can supply the space at about 19C. Since air has to be delivered at such a warm temperature (compared to the other systems), a large quantity of air has to be supplied to meet the cooling load. If care is not taken to design low static pressure fan systems, then all the advantages of the UFAD system would be nullified. A reheat coil has been placed in the supply path to provide heating. There is no active control of humidity.

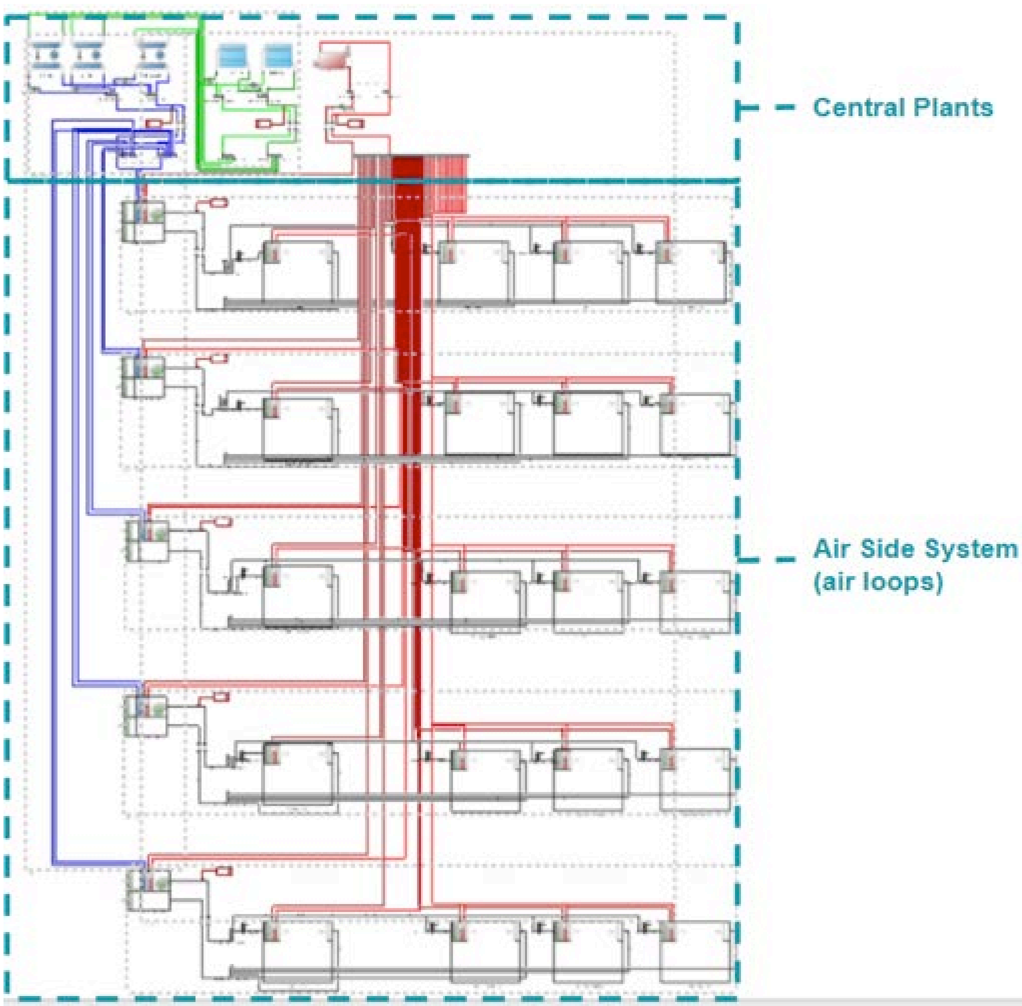

Figure 8: Schematic of Under Floor Air Distribution system as modelled

4 RESULTS

4.1 LOAD CALCULATIONS

Table 3: ASHRAE Design Day Temperatures for 0.4% confidence limits

Design temperature period	2-Multiple design months
Monthly Design Temperatures	
0.4% Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	
Monthly Design Dry Bulb	
Jan (°C)	35.1
Feb (°C)	35.1
Mar (°C)	31.2
Apr (°C)	29.4
May (°C)	25.3
Jun (°C)	22.1
Jul (°C)	22.0
Aug (°C)	25.2
Sep (°C)	30.0
Oct (°C)	33.9
Nov (°C)	35.0
Dec (°C)	35.4
Mean Coincident Wet Bulb Temperatures	
Jan (°C)	20.9
Feb (°C)	22.2
Mar (°C)	20.6
Apr (°C)	17.3
May (°C)	16.3
Jun (°C)	14.3
Jul (°C)	12.8
Aug (°C)	13.9
Sep (°C)	15.9
Oct (°C)	17.3
Nov (°C)	19.3
Dec (°C)	19.9
Monthly Minimum Dry Bulb	
Jan (°C)	24.7
Feb (°C)	25.5
Mar (°C)	21.6
Apr (°C)	18.9
May (°C)	15.8
Jun (°C)	12.6
Jul (°C)	11.7
Aug (°C)	13.5
Sep (°C)	16.8
Oct (°C)	20.2
Nov (°C)	23.0
Dec (°C)	24.3

Load calculations were carried using the latest ASHRAE Heat Balance[2] calculation procedures, which have been encoded into EnergyPlus. The calculations were done for Dry Bulb Priority using the 0.4% monthly confidential level (Table 1).

Table 4a and 3b: Cooling load for each zone at the selected design conditions for a typical mid level floor

Zone peak (NON-COINCIDENT loads, typical floor)	*Total Cooling Load (kW)*	*Sensible (kW)*	*Latent (kW)*	*Air Temperature (°C)*	*Humidity (%)*	*Time of Max Cooling*	*Design Cooling Load Per Floor Area(W/m2)*	*Outside Dry-Bulb Temperature at Time of Peak Cooling Load (°C)*
04Lvl5XMltpldBy7:L5XWest	103.4	92.4	11.0	22.5	53.1	Feb 16:00	148.4	34.5
04Lvl5XMltpldBy7:L5XNorth	74.1	74.1	0.0	22.5	49.9	Apr 13:00	106.4	28.9
04Lvl5XMltpldBy7:L5XEast	96.1	85.2	10.9	22.5	53.2	Feb 09:00	137.9	29.8
04Lvl5XMltpldBy7:L5XCentral	244.6	205.8	38.8	22.4	54.4	Feb 06:30	69.6	26.0
04Lvl5XMltpldBy7:L5XSouth	64.1	54.0	10.1	22.5	54.5	Feb 14:00	91.9	35.1

Zone peak (COINCIDENT loads, typical floor)	*Total Cooling Load (kW)*	*Sensible (kW)*	*Latent (kW)*	*Air Temperature (°C)*	*Humidity (%)*	*Time of Max Cooling*	*Design Cooling Load Per Floor Area (W/m2)*	*Outside Dry-Bulb Temperature at Time of Peak Cooling Load (°C)*
04Lvl5XMltpldBy7:L5XWest	99.9	88.8	11.1	22.5	53.2	Feb 15:30	143.3	34.8
04Lvl5XMltpldBy7:L5XNorth	70.8	60.6	10.2	22.5	54.1	Feb 15:30	101.6	34.8
04Lvl5XMltpldBy7:L5XEast	81.1	70.7	10.4	22.5	53.6	Feb 15:30	116.4	34.8
04Lvl5XMltpldBy7:L5XCentral	226.9	185.5	41.5	22.5	54.8	Feb 15:30	64.6	34.8
04Lvl5XMltpldBy7:L5XSouth	63.2	53.2	10.0	22.5	54.5	Feb 15:30	90.7	34.8

From *Table* 3a (see shaded columns) it is seen that the typical West facing perimeter zone has a design cooling peak load per floor area of 148.4 W/m2 which occurs at 16:00 hours on the Feb design day. This value is 137.9 W/m2 for the typical East perimeter zone, peaking at 9:00am on the Feb design day. It is noted that these are total cooling loads and are the sum of room and ventilation loads (typically, mechanical engineers list room loads and ventilation loads separately).

It is clear from *Table 3a* that it is a listing of the non-coincident zone peak cooling loads for each individual zone. *Table 3b* lists the typical zone loads at the time of the Coincident Building Peak, or the highest cooling load that the building will present to a central chilled water plant. Clearly these loads are lower when compared to the zone non-coincident peaks.

The Total Cooling Load, or the sum of the non-coincident peaks for the modelled building is 824.4 kW and the Coincident Building Peak (also called the Chiller Block Load) is 762.1 kW, i.e., a diversity of 92.4%.

[2] Pedersen, C.O., Fisher, D.E., Liesen, R.J. 1997. *Development of a Heat Balance Procedure for Calculating Cooling Loads*, ASHRAE Transactions, 103 (2), pp459-468

4.2 PREDICTED ANNUAL COOLING AND HEATING LOADS

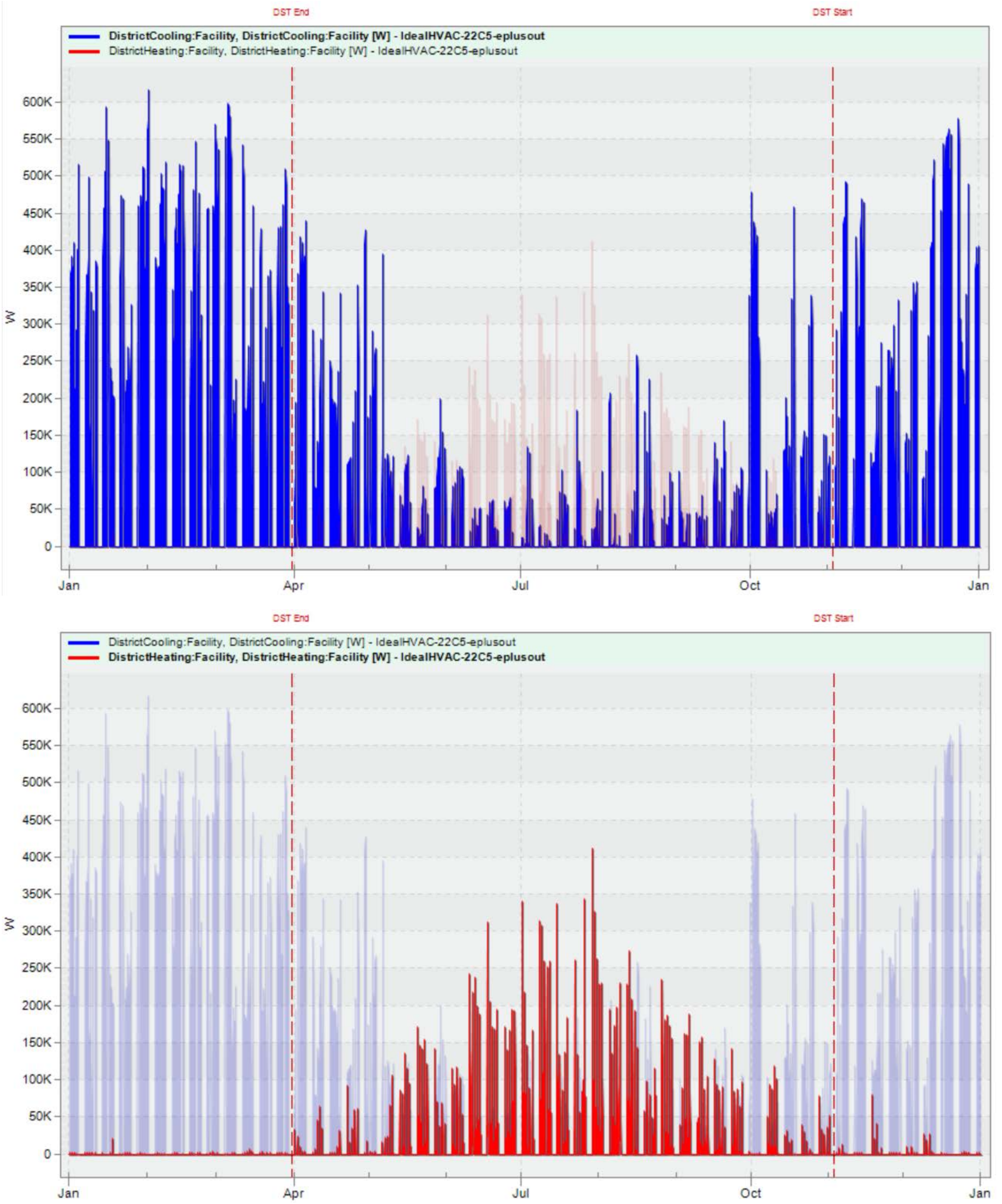

Figure 9a and b: Predicted building annual thermal cooling (above) and heating (below) hourly loads to be met by the HVAC systems being modelled, generated for a 22.5C setpoint. These loads are generated for the assumed operational profiles and are lower than the design day peak loads

Figure 9a is a graph of the predicted hourly cooling and heating loads (*Figure 9b*) for the whole building. These loads have been generated based on the building fabric and the internal loads varying as per the operational schedules modelled for occupancy, lighting and office equipment. Compared to the predicted Coincident Building Peak demand of 762.1 kW for cooling, it is seen that, in operation, the building is predicted to present a peak cooling load of about 620 kW *once* in the year, and more than 550 kW about 10 times in the year. Such differences in the peak design load, and the peak operational load (which can be verified by reviewing chiller electrical trend logs in real buildings for the previous 12 months) are common. Energy efficient HVAC systems are those that can meet such varied loads across the year in a manner that consumes the least amount of electricity and gas to drive system components like chillers, boilers, pumps and fans.

4.3 PREDICTED ENERGY (ELECTRICTY AND GAS) CONSUMPTION

Table 5: HVAC sub-system end use, and annual energy intensity for the 3 systems modelled

SYSTEM TYPE →	*UFAD*		*STD-VAV*		*PCB*	
HVAC END USE ↓	*Electricity [kWh]*	*Natural Gas [kWh]*	*Electricity [kWh]*	*Natural Gas [kWh]*	*Electricity [kWh]*	*Natural Gas [kWh]*
Heating	-	19,098	-	19,261.1	-	43,858.4
Cooling	84,599	-	76,986.9	-	120,257.2	-
Fans	64,706	-	58,322.8	-	25,839.0	-
Pumps	38,587	-	48,052.4	-	91,099.3	-
Heat Rejection	9,925	-	11,931.9	-	16,138.0	-
HVAC totals	197,817	19,098	195,294.0	19,261.1	253,333.4	43,858.4
Energy intensity, kWh/m2-yr	*24.1*		*23.8*		*33.0*	

The annual energy intensities, for the three systems modelled in an identical hypothetical building located in Sydney, are listed in Table 4 above. Predicted annual energy consumption for each HVAC sub-system (fans, pumps, etc) are also listed, and indicate the spread of energy use for these sub-systems for each HVAC system configuration.

Based on the results in Table 4, it can be inferred with some confidence that all three HVAC system configurations are predicted to be able to meet a 5 star Base Building NABERS level performance in metropolitan Sydney. An estimate that proves this statement is as follows:

- The highest energy intensity is that of the PCB system, at 33 kWh/m2-yr
- Using kgCO2/kWh of 0.95 for electricity and 0.23 for gas, the GHG intensity for the HVAC component of the base building for the CB system is 27.9 kgCO2/m2-yr
- Based on our experience with high performing buildings, we can reliably assume that HVAC emissions are about half the Base Building GHG emissions; the remainder being from other end uses like carpark lighting and ventilation, miscellaneous ventilation, common area and security lighting, end of trip facilities, lift energy use, etc
- Therefore, Base Building GHG emissions for the CB building can be estimated at about 55.7 kgCO2/m2-yr, which suggests almost a 20% buffer to the 70 kgCO2/m2-yr benchmark for 5 star NABERS Base Building performance in NSW for a typical office building

As noted previously, it is not the intention of the authors to rank the performance of the systems. System selection for a project is not determined by technical performance alone, but by a host of other decisions. However, some insights from the modelling exercise, and indeed from our practical experience, are discussed in the next section. When reviewing these results (or those from other studies using models) it is good to remember George Box's famous saying – "all models are wrong, but some are useful[3]…"; the carefully modelled representations here are useful, and should be used appropriately.

4.4 SYSTEM LEVEL INSIGHTS

We list below a series of system level insights from the modelling exercise, and from anecdotal observations made by experienced HVAC engineers that are consistent with the simulation study outcomes:

- Lowest fan energy predicted for CB (outside air only), followed by VAV (all air system) and UFAD systems (low pressure plenum). Design decisions on static pressure, fan efficiency and air quantities in real systems will determine the real rankings)
- VAV and UFAD employ recirculation of air, the PCB (Passive Chilled Beam) system is modelled as having once through outside air
- CB systems are predicted to have the highest pumping energy since high(er) temperature chilled water is being pumped to the PCBs located in conditioned spaces
- Heating energy use, and cooling energy use, is predicted to be highest in this configuration of (P)CB systems due to application of an explicit control strategy for dehumidification and reheat of primary air to eliminate the risk of condensation. Sydney has many low ambient temperature, high humidity days, resulting in higher energy use for dehumidification and reheat. CB systems maybe considered less appropriate for warm and humid climates, as they require carefully setup controls to manage the risk of condensation
- Our practical experience indicates that predicted heating energy use, particularly for VAV and UFAD systems maybe somewhat optimistic. Design equipment loads are modelled at 15 W/m2, however "plug" loads in modern office buildings are around half that value. Modern office lighting system designs are averaging less than 6 W/m2, with good designs getting below 4 W/m2. Hysteresis loads for the heating hot water loop are modelled with a series of defaults, and these can be better calculated for real projects. All of these reasons would increase predicted heating energy to values that better align with monitored values
- A "pure" configuration of the PCB system (without the dehumidification and reheat control employed in this modelled version), can be limited in it's capacity to provide cooling at times of high ambient humidity due to having to reset the primary air supply temperature upwards to be above space dewpoint
- CB systems are slower in responding to system perturbations, and can take longer to achieve space temperature due to unscheduled outages, or consecutive extreme weather events (beyond design day conditions)

[3] *Box, G. E. P. (1976), "Science and Statistics",* Journal of the American Statistical Association*, **71**: 791–799*

- CB systems require careful consideration of sensible and latent loads in the space in relation to CB configuration; and may not be appropriate for high occupancy areas like meeting rooms
- Full economy cycle is not feasible in CB systems, therefore constant chiller (or cooling tower) operation is required to meet any building cooling needs even in the cooler months
- However, CB systems are designed for full outside (ventilation) air, and the once through systems may offer better indoor air quality
- CB systems can have quieter operation as fan noise can be greatly used with careful design
- Smaller ducts (ventilation air quantities only) can mean lower floor-to-ceiling heights, can lead to more NLA for same height of buildings
- CB systems require chilled water reticulation in ceiling/tenant spaces; any leaks would impact tenant space. Maintenance personnel may need to access/intervene in the tenancy spaces to carry out repair works
- Whilst it provides flexibility in the ability to provide cooling closer to the occupants, the UFAD system has less controllability. A review of the hourly simulation outputs indicates the highest variability in space temperature for the UFAD system (compared against VAV and CB), along with high relative humidity levels in the space in summer (50 – 70%).
- UFAD systems have limited dehumidification capability due to higher off-coil temperatures, which leads to high relative humidity levels in the conditioned spaces. Dehumidification could be provided by use of enthalpy wheels; however, this solution will bring with it fan energy penalties and additional maintenance requirements
- Heat gain from plenums requires AHU supply temperatures to be reduced (by about 2C in this configuration and Sydney climate) to maintain the desired 19C at the supply diffuser. In practice, the uniformity of space temperatures will also vary due to air flow adjustments carried out by occupants by changing diffuser positions
- Both UFAD and PCB systems require high performance building façade that limits perimeter peak cooling loads; UFAD cooling capability is limited by the “ankle freezing” syndrome placing a limit to SAT, and PCB systems are limited by beam performance and dewpoint control
- CB systems have stable temperatures in the interior zones, and these are seen to remain between 22 and 23C. This is another reason that can explain the high annual cooling energy consumption predictions
- VAV systems are predicted to respond quickly to external conditions and can maintain very stable temperatures that are close to the upper end of the deadband in cooling mode (24C) and the lower end of the deadband (21C) in heating mode
- VAV systems are robust, are a common configuration and there is depth of knowledge in service industry to maintain and tune them
- Design refinements that can lead to higher operational efficiency, for example, low temp VAV with terminal turn down ratios

- VAV systems pose no risk to humidity/condensation in space; however, air in the conditioned spaces is fully mixed
- Plant restricted to service areas (no water circulated in tenant spaces) for VAV systems; allows the ability to carry out the bulk of maintenance without accessing tenant spaces
- VAV systems can offer better flexibility for tenancy modification to space arrangements

4.5 COMMENTARY

The oil crisis of the 1970s and early 1980s and the subsequent rise in electricity charges forced a fundamental rethink of adoption of different types of air conditioning systems in commercial building. Many of the older buildings in Australia that had constant volume, dual duct and multi-zone type HVAC systems were gradually replaced by VAV systems that offered lower energy consumption by virtue of providing cooling or heating to match the varying heat load requirements. During that period VAV systems became the norm in a majority of new commercial developments. In many cases these VAV systems still continue to operate satisfactorily, and offer a great opportunity to have their performance enhanced by taking advantage of improved technologies and digital control. The climate change and the 'green' movement in the 1990s with the Kyoto Protocol to the United Nations Framework Convention on Climate Change coming into force in 1997, gave rise to further technological evolutions in HVAC systems in Australia, such as chilled beams and UFAD system types mostly in pursuit of higher energy efficiencies.

Whilst overall HVAC energy efficiencies continue to show significant improvements, there have been notable factors that have contributed to HVAC systems not achieving optimum levels. Firstly, there is the issue of "right-sizing" of HVAC systems and associated equipment to match requirements as established by modern heat load calculations. In far too many installations there is evidence[4] that the HVAC equipment is oversized – particularly the central chilled water plant. This continues to lead to equipment not operating at peak efficiencies. Secondly, time for commissioning of HVAC systems is often compressed due to the way many commercial buildings are delivered. Therefore correct setting up of the equipment and systems, in particular HVAC controls, is not carried out satisfactorily. Apart from over cooling and reheating in many instances, there are also air circulation issues resulting in uncomfortable indoor conditions. Thirdly, there is a knowledge gap between designers and operators of these often fairly complicated HVAC systems. Hence these systems are not able to reach optimum energy efficiency levels in operation. Therefore it is reiterated that the values presented in this report may vary from observed or monitored outcomes from similar system types in practice.

[4] Thomas, PC and Moller, Steven. *HVAC system size – getting it right.* In Clients Driving Innovation : Moving Ideas into Practice, CRC for Construction Innovation, 12-14 March 2006

5 CONCLUSIONS

A dynamic simulation analysis of three types of HVAC systems has been carried out between:

- Passive Chilled Beam (PCB) system
- Under Floor Air Distribution (UFAD) system, and
- Variable Air Volume (VAV) system.

Comparative results have been shown for same building and operating conditions.

The study does not make any judgements to encourage or discourage the use of a particular system, but strives to provide a series of objective criteria that engineers can consider in their selection of systems for practical projects. We reiterate that the selection decision for a particular system (or indeed a combination of systems, or a “hybrid” configuration of systems) can be arrived at due to a combination of technical, physical, financial or perception constraints, and hope the study is useful for practicing HVAC design and construction engineering teams to evaluate their choice of systems.

It is also reiterated that this is a first step in the research process and future studies can delve into further detail of operation, or indeed, into the performance prediction of hybrid systems, or combinations of systems. Reviewing these and other results using future climate scenario weather files would also be of value.

6 RESEARCH TEAM

This study has been carried out by:

PC Thomas, Director, Team Catalyst. PC has been using building energy simulation tools for more than 30 years, and provides training in the use of building performance simulation tools to students and colleagues in the industry in Australia and internationally. Among his other experiences PC helped found the ESD team at Arup, Sydney, in the early 2000's. He is a member of the International Education Committee with IBPSA (the International Building Performance Simulation Association), and is currently Adjunct Associate Prof at the University of Sydney.

GS Rao, Director, Team Catalyst. GS provided high level advice and review for modelling of each system configuration, with particular attention to the practicability, component performance limitations and application of control strategies. He has more than 30 years of HVAC systems knowledge and experience having worked variously with York, Carrier and Trane over the course of his career, and having designed and delivered major HVAC installations in a number of countries.

Ms Noni Nuriani, Senior Sustainability Consultant, NDY, helped to develop the HVAC inputs to the DesignBuilder models with inputs from PC and GS. Noni has been involved in the performance modelling of a number of large commercial building projects over the last few years, and is one of a new breed of young engineers who regularly use such tools in their work.

Ms Ayshvarya Venkatesan, Associate, Team Catalyst developed the building envelope model that formed the basis of the study. The model was developed by Aysh for other projects that Team Catalyst previously carried out for other research projects, including projects for the Australian Building Codes Board. Aysh has a background in Architecture and is passionate about delivering occupant comfort and reducing GHG emissions in the built environment.