LOW CARBON LIVING CRC

RP 1033- Mainstreaming High Performance Commercial Building HVAC

Stage 2 Report

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- compliance with ethical guidelines
- conclusions against results
- conformity with the principles of the Australian Code for the Responsible Conduct of Research (NHMRC 2007),

and provided constructive feedback which was considered and addressed by the author(s).

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Acronyms

AHU Air Handling Unit

BCA Building Codes of Australia

COP Coefficient of Performance

CHWP Chilled Water Pump

CWP Condenser Water Pump

HHW Heating Hot Water Pump

HVAC Heating, Ventilation and Air Conditioning

NABERS National Australian Built Environment Rating System

NPV Net Present Value

NLA Net Lettable Area

PWF Present Worth Factor

RA Return Air

VAV Variable Air Volume

VRV Variable Refrigerant Volume

VSD Variable Speed Drive

Executive Summary

This report presents a summary of the findings and activities of Stage 2 of the LCL-CRC research project RP 1033 “High Performance Commercial Building HVAC”.

This research examined the HVAC design of a high performance commercial office building (hereafter referred to as the case study building) and compared its electricity consumption to that of a standard BCA compliant HVAC design for the same building. The main HVAC system designed and installed in the case study building as a low temperature VAV, air-water HVAC system, which is one of the most common designs currently utilised for commercial buildings. The case study building is based on a project completed by AECOM where they were engaged as both the Mechanical Services and ESD consultant.

This study was used to gain a better understanding of HVAC design used in high performance buildings verses that required for minimum code compliance. This work is part of a broader CRC project - Mainstreaming High Performance HVAC, which is looking at factors associated with facilitating increased industry uptake of high performance HVAC design. The focus of this study is to examine the potential energy savings from improved duct and pipe system design.

A validated IES model for the case study building and its associated high efficiency HVAC design (the case study HVAC design) was utilised in this work. The following two scenarios were modelled and compared:

- A HVAC design that met the minimum BCA HVAC requirements (BCA compliant HVAC design)
- The case study HVAC design

The simulation results showed that the annual electricity consumption of the case study HVAC system was 13% less than the consumption for the BCA compliant HVAC system. This was mainly due to energy usage reductions in the air handling units (AHU’s) and return air (RA) fans (24% reduction), the chillers (25% reduction) and the chilled water pumps (28% reduction) for the case study HVAC design compared to the BCA compliant HVAC design.

The energy reduction for the AHU’s fans for the case study HVAC design was achieved by using larger ducts (around 10% larger hydraulic diameter), more efficient duct fittings as well as reducing the number of duct fittings. Further, the energy reduction in the chillers and their associated chilled water pumps for the case study HVAC design was achieved by selecting more efficient chillers and utilizing larger pipes for the chilled water pipe network (around 7% larger hydraulic diameter).

Other aspects of the case study HVAC design did not achieve as great an energy reduction as the three components described previously. As such, for all elements of the case study HVAC design, the overall reduction in electricity consumption achieved was 13% in comparison to the BCA compliant HVAC system. Note that this reduction in the electricity consumption for the case study building leads to an approximate 171 tons reduction in annual CO_2 emissions.

Additionally, an economic analysis was performed to determine the cost-benefit of the case study HVAC design compared to the BCA compliant HVAC design. The results showed that the initial cost of the case study HVAC design is only 1% greater than the capital cost of the BCA compliant HVAC design. Further, a discounted payback period of 4 years was calculated for the additional cost invested on the case study HVAC design instead of the BCA compliant HVAC design.

The widespread adoption of the design criteria utilised for the case study HVAC design is recommended for other low temperature VAV, air-water HVAC systems. If adopted by HVAC designers, this approach would lead to significant reductions in the energy usage and greenhouse gas emissions of these systems.

1 Introduction

1.1 HVAC Energy Consumption and Minimum Energy Requirement in Australia

HVAC systems typically represent a substantial portion of a commercial buildings energy use and are therefore important areas to target when designing low energy buildings. Figure 1 shows the breakdown of energy consumption for typical Australian office buildings. It can be seen that the HVAC system accounts for 39% of the total building's energy consumption [1].

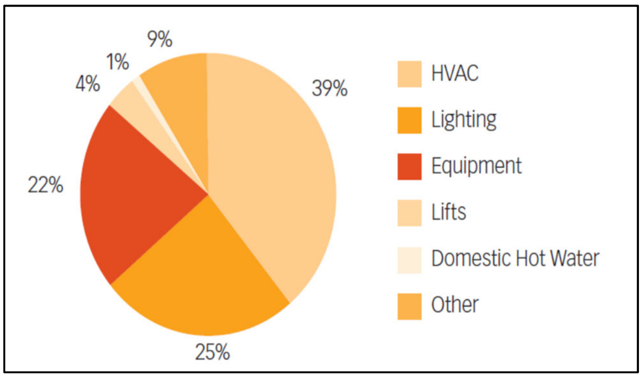

Figure 1. Typical energy consumption breakdown in an office building [1].

Further, as shown in Figure 2, closer examination of the HVAC segment of energy shows that fans and pumping consume around 50% of HVAC energy while heating and cooling consume less at 44% [1].

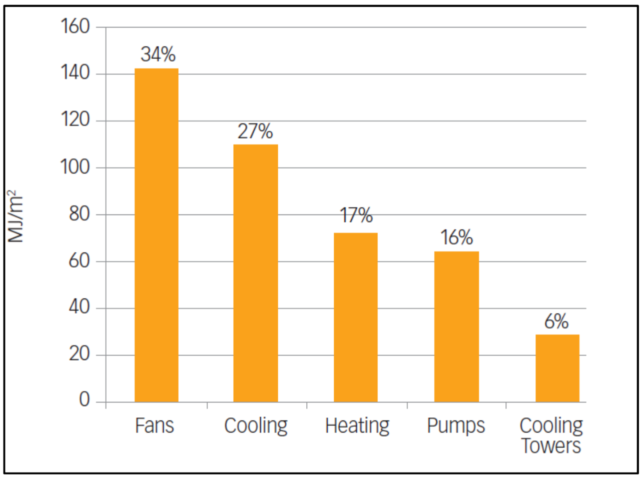

Figure 2. Typical HVAC end use breakdown [1].

These breakdowns of energy usage are averaged across Australia and different components of the system will dominate depending on the building's location. For example if a building is located in a tropical climate, then there will be greater energy expended on cooling due to humidity and high temperatures [2].

However, in temperate regions with lower heating and cooling demands, the simple movement of air and fluids through ducts, pipes, past heating and cooling coils and filters in air handling units will dominate the energy consumption.

When buildings are constructed, the construction contract usually goes to the lowest price tenderer. To achieve lowest price, contractors concentrate on assembling buildings in the shortest time possible to minimise labour costs. Low competitive margins mean there is little room for error. And so contractors will price certainty by employing tried and tested methods to reduce cost risk. Innovation is often talked about but rarely employed by the construction industry. Consequently the methods employed by the building industry are often based on how it has been done for years, derived from rules-of-thumb and meeting the minimal energy efficiency provisions of the National Construction Code [3]. The Australian Building code has included minimum energy efficiency requirements for HVAC systems for commercial buildings since 2006 [3]. Since 2006 performance requirements have been adjusted and tightened with an updated version due for release in 1st May 2019. In parallel to this, industry has seen increased uptake on the use of formal energy and sustainability ratings as a way of differentiating buildings in a competitive market. Thus, many buildings that pursue such ratings, particularly Property Council of Australia (PCA) premium and grade A commercial office buildings include design features that routinely exceed minimum energy efficiency requirements.

There has been significant academic research into high efficiency HVAC systems and designs [4–6]. ASHRAE - the American Society of Heating, Refrigerating, and Air – Conditioning Engineers – has published excellent guidelines that point to the design of buildings and associated HVAC systems with energy performance 50% better than code [7]. However for example, these designs only aim for 900 Pa static pressure for their air handling systems. Current best practice in industry is already achieving these sorts of static pressures and from AECOM's experience further reductions in static pressure are possible [8]. As such further research is required to compare the technical and economic performance of a case study building with a high efficiency HVAC system design in comparison with a BCA compliant HVAC system design.

HVAC systems can be designed as all-air, air-water or all-water systems [9]. The HVAC system investigated in this study is an air-water, low temperature VAV system similar to the system shown in Figure 3. Such a system is a very common design which is currently being installed in many commercial office buildings in Australia. Figure 3 shows the air schematic of an air-water system which includes VAV boxes to deliver the conditioned air to different zones of a building. Note that AHU shown in Figure 3 is connected to chilled and hot water systems which are not shown in Figure 3.

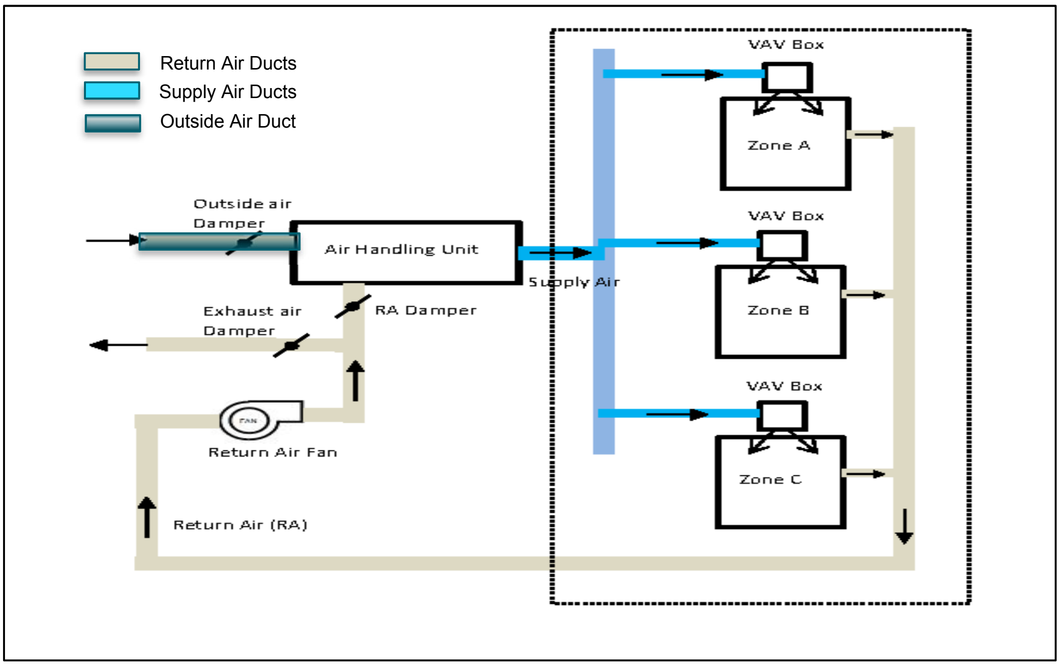

Figure 3. Simple air schematic of an air-water HVAC system

1.2 Scope of Work:

This study focuses on the comparison of 1) a case study high energy efficiency HVAC system design versus 2) a system designed to meet the minimum BCA HVAC energy requirement. Note that the main focus of this study is on comparing air/water handling system design of the above HVAC systems and the associated potential to deliver lower energy outcomes. The main HVAC systems in the case study is an air-water, low temperature VAV system and consists of a Chilled Water (CHW) cooling system and a gas-fired Heating Hot Water (HHW) plant connected to air handling units (AHU's). The AHU's pump conditioned air to the different zones of an office building (case study building) via VAV boxes (the system is similar to what was described in Figure 3). The case study building also consists of some supplementary HVAC systems (i.e. split/packaged AC units, car park and general mechanical ventilation systems).

For the main HVAC system, the BCA compliant HVAC system is again an air-water, low temperature VAV systems serving the same building (with the above supplementary systems) and consists of essentially the same equipment. However, the air/water distribution system components (i.e. duct riser and duct fitting sizes) and equipment efficiencies for the BCA compliant design are different from the case study design. The analysis has been completed with the aid of building performance modelling software IES (Integrated Environmental Solutions). Further details are presented in Sections 2 and 3.

From the detailed examination of this case study HVAC system in comparison to the BCA compliant design aims to investigate the potential for cost effective designs that can deliver energy and carbon savings. The aim of this study is to facilitate increased industry uptake of high efficiency HVAC systems in Australia.

2 Case Study Building and Associated HVAC System

The case study building is PCA Grade A commercial office building with a net lettable area (NLA) of 39,803 m^2. The building is located in Sydney and achieved an energy star rating of 5.5 based on the National Australian Built Environment Rating System (NABERS) [10] and a 5 Star Green Star Office V3 As Built Rating The building consists of four levels of basement car parking areas, thirty levels of commercial office spaces, low-rise plant level, rooftop plant level and a lobby area on the ground floor. There is a retail/restaurant tenancy located on the ground level and a smaller café tenancy on the lower ground level. The building has a validated IES model created by AECOM [11]. The maximum thermal cooling and heating loads of the building are 4160 kW and 1970 kW respectively. The main HVAC system design of the building is an air-water system similar to that described in Figure 2. The HVAC system consists of a Chilled Water (CHW) cooling system and a gas-fired Heating Hot Water (HHW) plant connected to the air handling units (AHU's). Additionally, the building consists of some supplementary HVAC systems (i.e. split/packaged AC units, car park and general mechanical ventilation systems). Table 1 provides further detail of the HVAC system major components.

Table 1. Further details for case study HVAC system

System	System Components	Type	Description
Chilled Water System	Chillers	Water cooled VFD screw chillers	The chiller mix includes two equally sized high load chillers (1805 kWr) and a low load machine (550 kWr) sized to be capable of operating stably at part load serving the minimum after hours' zone area. The chilled water supply and return design temperatures are 6℃ and 14℃ respectively. Additionally, a staging strategy is considered for the chillers to operate in relation to the building part load condition
	Chilled water pumps	Variable speed pumps	Chilled water is provided by a single primary loop from the chillers consisting of four variable speed chilled water pumps (CHWP's); three on duty and one on standby.
	Cooling tower	Induced draft counter flow cooling towers	There are two, roof mounted, induced draft, counter flow cooling towers. Each cooling tower has 3 cells and each cell has a fan with an input power of 15 kW. The cooling towers are sized to provide heat rejection from the chillers as well as the office tenant condenser water loop and retail condenser water provision
	Condenser water pumps	Variable speed pumps	Heat rejection from the chillers is via a condenser water system which consist of cooling towers and four variable speed condenser water pumps (CWP's); three on duty and one on standby.
Heating Hot Water System	Boilers	Gas fired hot water generators (boilers)	Heating is provided by four gas fired hot water generators (boilers) split in pairs between the Level 3 and Level 31 plantrooms. The boiler mix for each pair includes a high load boiler (720 kW) and a low load boiler (300 kW). Also, a staging strategy is designed for running the boilers and their associated pumps based on the building part load.
	Heating hot water pumps	Variable speed pumps	Heating water is provided from each set of boilers through primary loops, each provided with three variable speed primary pumps; two on duty and one on standby.
Air Handling Systems	AHU's on level 3 and 30 AHU's on level 30	Low temperature variable air volume air handling system	The AHU's are located at level 3 and 31 plant rooms and provide conditioned air to the variable air volume (VAV) boxes located in the different thermal zones of the building from the ground level to level 30. Each AHU has air economy cycles (which utilizes outdoor air for free cooling of the associated thermal zones when ambient conditions are favourable). The thermal zones consist of the East, North and Western façade orientations and two central zones. An additional South perimeter zone is provided for, from one of the central zones

System	System Components	Type	Description
			with a dedicated VAV box & re-heat coil per floor. The AHU's on Level 3 serve ground to level 17 and those on Level 31 serve Level 18 to 30. The spaces on Level 1 and 2 are served by four AHU's; two per floor that serve two dedicated thermal zones of the East facade and the central zone. It is notable that the AHU's and their associated air delivery systems were designed with the aim of reducing the total system pressure drops. Tables A.1 to A.3 in the Appendix A show all the thermal zones served by the AHU's
Supplementary Heating/ Cooling Systems	In-slab and trench heating system	In slab-electric heating system	The ground floor is served by a 100% outdoor air, under floor air distribution system to the central area in combination with an in-slab centre zone hydronic heating system and perimeter zone trench heating system.
	Packaged air conditioning units	Water-cooled PAC units	Water cooled packaged unit air conditioning systems are utilised to serve ground floor retail tenants and the lift motor rooms on Level 23 and Level 31.
	Air conditioning units	Air cooled split DX systems	The units are utilised to serve various small spaces throughout the levels below ground level.
Car park and General Mechanical Ventilation Systems	Car park and mechanical ventilation fans	Some of these fans have VSD's.	The carpark system includes two exhaust fans and four supply fans. The general mechanical ventilation systems comprises of toilet exhaust fans, kitchen exhaust fans, stair pressurisation exhaust and supply fans as well as other general supply/exhaust fans. Many of these fans have VSD's.
Tenant Condenser Water System	Tenant condenser water pumps	Variable speed pumps	Tenant condenser water is a closed loop system circulated by the tenant condenser water pumps rejecting heat to the main condenser water system via heat exchangers. Heat rejection is combined with the chiller plant heat rejection through the main cooling towers. Heat is inserted into the closed tenant condenser water circuit as needed from a heat exchange with the HHW loop.

3 Methodology

The two different HVAC system designs were evaluated using an existing IES model of the building and the HVAC system, provided by AECOM.

The original design of the HVAC system is called the case study HVAC design which is a high efficiency HVAC design. A second IES model was developed to model another HVAC design which just complies with the minimum HVAC energy requirements of the BCA (the "BCA compliant HVAC design"). The following elements remained constant for both HVAC design models:

- Building components
- Occupancy loads and schedules
- Lighting and office equipment loads and usage schedules

Additionally, The BCA compliant HVAC system consists of the same number of HVAC equipment considered for the case study design. However, the air/water distribution system elements (i.e. ducts and risers) and equipment efficiencies for the BCA compliant design are different from the case study design. For example, for the BCA compliant design, the fans were selected considering the requirements of Tables 3a, 4a and 4b from Specification J5.2a in the BCA [3]. Note from Table 3a, the energy requirements can be interpreted differently as described in Appendix C. Various interpretations will have different outcomes for the fan energy consumptions as described in Appendix D.

Further, ductwork and pipe systems for the BCA compliant HVAC system design were selected by changing the size/diameter of ducts and pipes as well as considering different types of fittings. Note that the flow rates of the fans and the pumps were considered to be equal for both HVAC design types. Hence, the total pressure drop for the air/water distribution systems differs in the BCA compliant HVAC design compared to the case study design due to the different duct/pipe sizes and different types of fittings utilized for these designs. Further details for the calculation of the total pressure drops in the air/water distribution systems are described in Appendix C. Table 2 provides a summary for both HVAC designs modelled in this study.

Table 2. Summary of design criteria for two HVAC design options in this study

HVAC Design	Design Criteria
Case Study HVAC Design	• Air and water distribution systems were designed with the focus of reducing the total pressure drops through those systems. Further details of the method of pressure drop calculation is described in Appendix B. • High efficiency chiller plants and boilers were utilized. • High efficiency fans and pumps were utilized. • VSD and fan controllers were utilized to reduce fan/pump energy consumption. • Packaged and Split AC units with high energy efficiency ratio were utilized.
BCA Compliant HVAC Design	• The maximum fan motor power consumption of the air handling units and other mechanical ventilation systems were calculated considering the requirements of Tables 3a, 4a and 4b from Specification J5.2a in the BCA [3]. • The chillers were selected based on the minimum energy performance standards (AS/NZS 4776) as required by the BCA [3]. • The cooling towers were designed based on Table 3b from Specification J5.2a in the BCA[3]. • All the pumps were designed to comply with the maximum pump power consumption recommended in Table J5.2 from the BCA[3]. • The VSD's and the fan controllers were assumed to be the same VSD's and controllers designed in the case study HVAC design. • Air and water distribution systems of the HVAC system were designed so that the maximum power consumption of the associated fans and pumps complied with the Specification J5.2a and Table J5.2 from the BCA[3] respectively. • Packaged and split AC units were designed to comply with the minimum energy efficiency ratio (EER) value specified in the Australian standard (AS/NZ 3823.2) [12].

4 Review and comparison of case study HVAC design vs BCA compliant HVAC design

4.1 Results

Figure 4 shows the simulation results for electrical energy consumption of the HVAC components for both HVAC designs. It can be seen the major reduction in the electrical energy consumption for the case study HVAC design is because of a better design for the air handling units (AHU) and return air (AHU & RA) fans as well as utilisation of more efficient chillers. This was due to the focus of the case study design in reducing the pressure drop of the air distribution systems and selecting equipment with higher efficiencies. Looking at Figure 4, it is evident that there is not much difference in the electrical energy consumption of other HVAC components in comparison to the BCA compliant design. Note that in some cases the case study design consumes more energy than the BCA design, however overall the case study design delivers a reaction of 13% in energy. Hence, a Performance Solution was proposed to achieve the BCA compliance. Note that the BCA allows a choice of Deemed-to-Satisfy Solution or Performance Solution for a building design compliance [3]. For a Deemed-to-Satisfy Solution, Part J1 to J7 of Deemed-to-Satisfy Provisions must be used [3]. Alternatively, the compliance with the Performance Solution is verified when it is determined that the annual energy consumption of the proposed building with its services is not more than the annual energy consumption of a reference building [3].

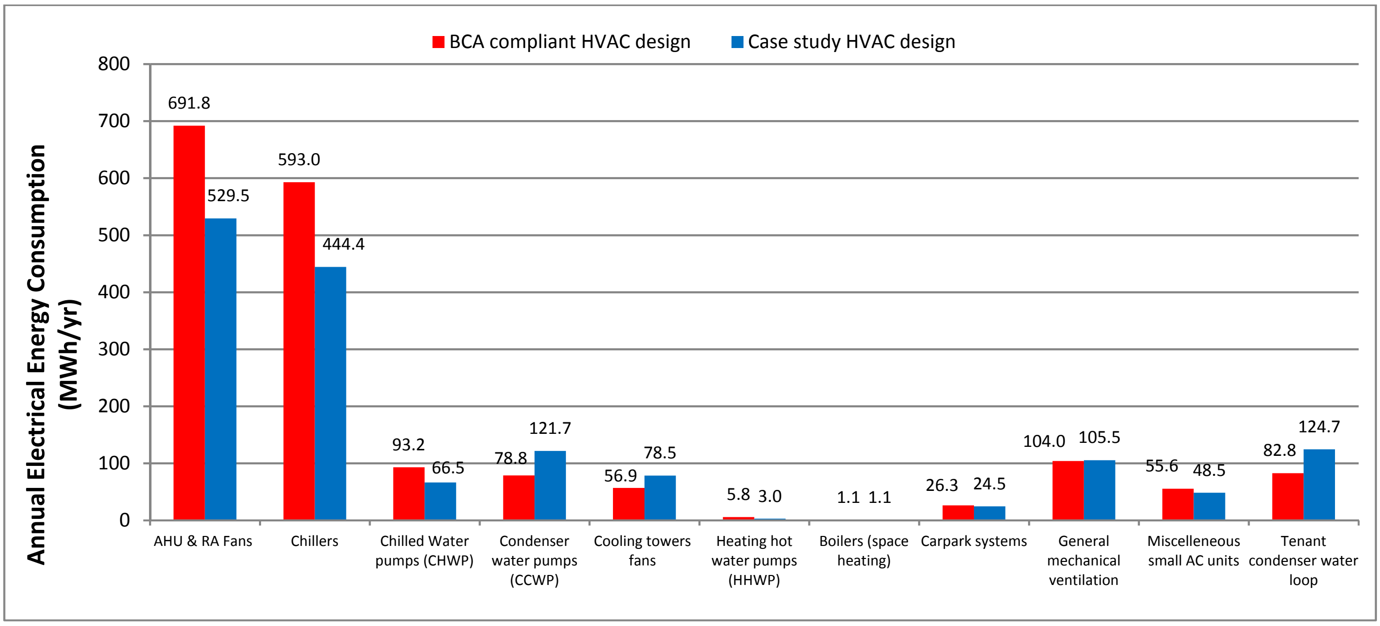

Figure 4. Comparison between the electrical energy consumption of HVAC components for both HVAC design

Table 3 shows the simulation results for the annual energy consumption of the major HVAC subsystems for the both HVAC design models in this study (including HVAC components for each subsystem). A more detailed discussion of Table 3 will be given in the Section 4.2.

From Table 3, summing all electricity consumption, the total electrical energy required for the BCA compliant design is 1789.6 MWh and for the case study design it sis 1547.2 MWh, which is a reduction of 13% in electricity usage due to the improvements in the HVAC design.

Note that the avoided emissions and generation tool (AVERT) [13] recommends an emission factor of 7.07×10^{-4} CO_2/kWh to convert reductions of kilowatt-hours into avoided units of carbon dioxide emissions. Hence, the above percentage reduction in the electricity consumption for the case study building leads to an approximate 171 tons reduction in annual CO_2 emissions.

Also, looking at the subtotal energy consumption of the various subsystems shown in Table 3, it is clear that the chilled water loops and AHU's and RA systems are the biggest energy consumers for both HVAC design systems investigated in this study. Further results and discussions are presented in Section 4.2.

Table 3. HVAC subsystems energy consumptions

AHU's and RA systems electrical energy consumption (MWh/yr)			
Components	BCA compliant HVAC design	Case study HVAC design	Comments
AHU's fans	546	393	For the case study HVAC design, these results were achieved by utilising larger main ducts (10% increases in the hydraulic diameter of ducts), reducing number of ductwork fittings as well as utilizing efficient ductwork fittings.
RA's fans	146	136	
Subtotal	692	529	

Chilled water loops electrical energy consumption (MWh/yr)			
Chilled water loops components	BCA compliant HVAC design	Case study HVAC design	Comments
Chillers	593	444	More efficient chillers were selected for the case study HVAC design
Chilled water pumps	93	67	The energy reduction was achieved by increasing pipe sizes of the chilled water distribution systems in the case study HVAC design (7% increases in the pipe diameters).
Condenser water pumps	79	122	Higher pressure drop of condenser water piping network in the case study HVAC design compared to the BCA compliant HVAC design.
Cooling tower fans	57	78	Lower efficiency of the cooling tower fan used in the case study HVAC design compared to the BCA compliant HVAC design.
Subtotal	822	711	

Hot water loop electrical energy consumption (MWh/yr)/ gas energy consumption (GJ/yr)					
Hot water loops components	BCA compliant HVAC design (electrical energy consumption)	BCA compliant HVAC design (gas energy consumption)	Case study HVAC design (electrical energy consumption)	Case study HVAC design (gas energy consumption)	Comments
Boilers	1.1	1675.6	1.1	1685.5	Lower R-Value of ducts and an increase in the hot water pipe sizes in the case study HVAC design led to a slightly greater heat loss during heating seasons. Hence, the gas boiler needed to operate longer for the case study HVAC design.
Hot water pumps	5.8	N/A	2.9	N/A	The energy reduction was achieved by utilizing 8% larger hot water pipe diameters for the case study HVAC design compared to the BCA compliant HVAC design.
Subtotal	6.9	1675.6	4	1685.5	

Carpark and general mechanical ventilation system electrical energy consumption (MWh/yr)			
Car park and general mechanical ventilation system components	BCA compliant HVAC design	Case study HVAC design	Comments
Car park systems	26.3	24.5	Similar fan efficiency and design for both HVAC designs
General mechanical ventilation systems	104	105.5	
Subtotal	130.3	130	
Miscellaneous air cooled split AC units electrical energy consumption (MWh/yr)			
Subtotal	BCA compliant HVAC design	Case study HVAC design	Comments
	55.6	48.5	More efficient AC units were selected for the case study HVAC design
Tenant condenser water loop system electrical energy consumption (MWh/yr)			
Subtotal	BCA compliant HVAC design (MWh)	Case study HVAC design	Comments
	82.8	124.7	Higher pressure drop of tenant condenser water piping network in the case study HVAC design compared to the BCA compliant HVAC design.
Grand total electrical/ gas energy consumption			
Electrical energy/Gas use	BCA compliant HVAC design	Case Study HVAC design	Comments
Electrical energy consumption (MWh/yr)	1789.6	1547.2	The energy reduction for the case study design is because of lower pressure drops in the air and water distribution systems as well as utilizing equipment with higher efficiencies
Gas energy consumption (GJ/yr)	1675.6	1685.2	Lower R-Value of ducts and an increase in the hot water pipe sizes in the case study HVAC led to a slightly greater heat loss during heating seasons. Hence, the gas boiler needed to operate longer for the case study HVAC design.

4.2 Discussion

4.2.1 AHU's & RA Fan Systems (Including Air Distribution Systems)

From Table 3, it can be seen that the total energy consumption of all supply and return air fans for the air handling units (AHU & RA systems) in the case study and the BCA compliant HVAC design are 529 MWh and 692 MWh respectively (excluding chiller and boiler energy). Hence, there is an approximate 24% less energy consumption for these fans in the case study HVAC design. This percentage reduction was due to the following reasons:

- The total average pressure drops of the AHU's in the case study HVAC design is around 700 Pa. However, the total average pressure drops of the AHU's for the BCA compliant design is around 940 Pa. For the case study HVAC design this result was achieved by sizing the main ductwork to have a low pressure gradient (mainly between 0.8 and 1 Pa/m) and utilizing efficient ductwork fittings (i.e. round bend with splitter). Figure 5 shows the sketch drawing of a riser duct section in one of the AHU's (AHU-L03-PW). From Figure 5, it is clear that for the BCA compliant HVAC design, the duct cross section area is reduced after each branch take-off in the main riser. However, this does not necessarily occur in a riser duct for the case study HVAC design. Consequently, fewer duct transitions were utilized in the air handling systems for the case study HVAC design. Additionally, looking at duct sizes shown in Figure 5, it is evident that hydraulic diameter of ducts in the case study design are around 10% greater than the same metric for the ducts in the BCA compliant design. Further, a majority of the duct bends (especially those ones used in the riser ducts) in the case study HVAC design have splitters or turning vanes which reduce air turbulence through those fittings. For the air delivery systems of AHU's in the BCA compliant HVAC design, typical bends without any vanes or splitters were utilised in the design.

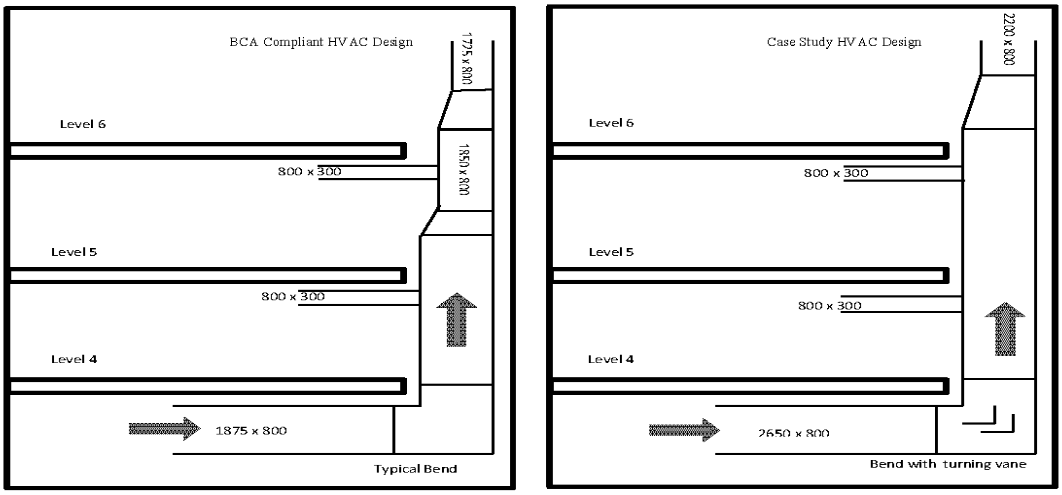

Figure 5. Sketch of the riser duct from level 3 to level 6 for AHU-L03-PW in both HVAC design

- The total average pressure drop of the return air (RA) fans in the case study HVAC design is 325 Pa. However, the total pressure drop of the same fans for the BCA compliant design is around 346 Pa. Note that 10% larger diameter ducts, smaller number of duct fittings and more efficient duct fittings were utilised in the case study HVAC design compared to the BCA compliant HVAC design.

Additionally, Figure 6 shows the breakdown of the fan motor energy consumption for the different major elements of the AHU's for the case study and the BCA compliant HVAC designs respectively.

From Figure 6, it is evident that the AHU fans in the BCA compliant HVAC design consume a significant amount of energy for moving the air through the ductwork and other external elements. This is mainly due to the greater pressure drops for the riser ductwork and fittings utilized in the BCA compliant design as shown in Figure 6.

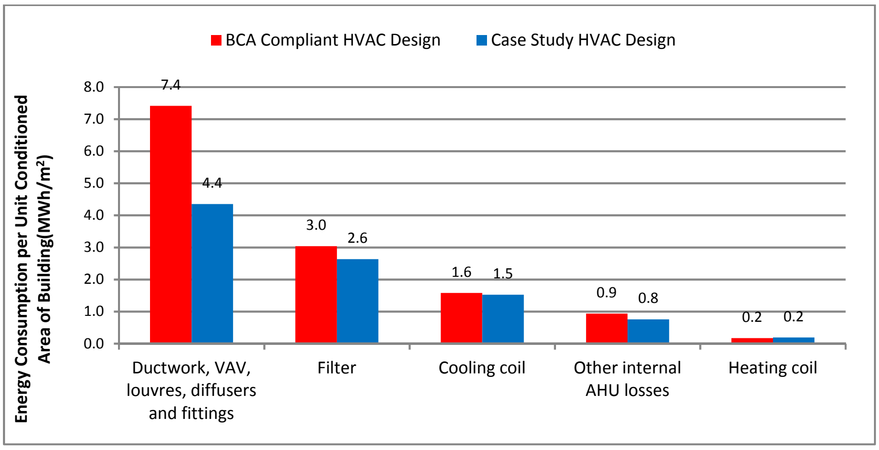

Figure 6. Breakdown of fan motor energy consumption for AHU's in both HVAC design

Moreover, Figure 7 compares the average pressure drops of the AHU's ductwork and other external components for the case study and the BCA compliant HVAC designs.

Looking at Figure 7, it can be seen that the pressure drops for the riser and plantroom ductwork for the case study HVAC design are less than the pressure drops for the same elements in the BCA compliant HVAC design. This is due to using the larger duct cross section areas for the case study HVAC design (as shown in Figure 5). Note that by selecting larger ducts the hydraulic diameters are increased. Also, the air velocity is reduced through larger ducts where the air quantity is constant. Note that the VAV boxes, outside air (OA) louvres, branch ducts and diffusers were considered to be the same for both types of HVAC designs; hence they have the same pressure drop for both designs.

Additionally, it is evident that the pressure drops for the ductwork fittings (riser & plantroom fittings and branch fittings) in the BCA compliant HVAC design are greater than the pressure drops for the same components in the case study HVAC design. This is due to utilizing less duct reducer transitions and selecting ductwork fittings with smaller values of the loss coefficient for the case study HVAC design compared to the BCA compliant design as shown in Figure 5.

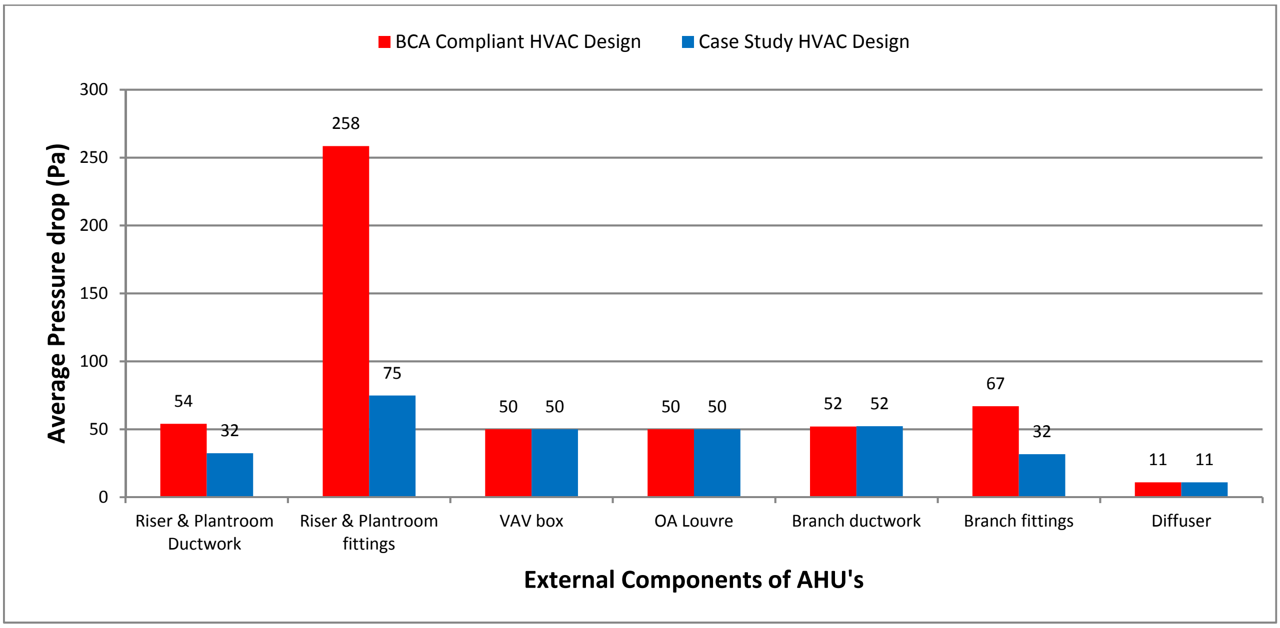

Figure 7. Breakdown of average pressure drop of ductwork and associated external components in both HVAC designs (excluding pressure drops associated with the AHU e.g. filters, coils and other internal AHU pressure drops)

From the above discussions, it is concluded that reducing the air velocity through the external elements (i.e. ductwork) and AHU's can reduce energy consumption of as such air handing systems significantly. This solution however typically requires additional plant room and riser to accommodate increased ductwork.

This can add to the capital cost of the project and also reduce the available NLA which reduces revenue from ongoing rent. Hence, performing an economic analysis is recommended where it is aimed to select larger ductwork (larger than typical) for an HVAC system. For example, for the case study HVAC design a 10% larger diameter ducts could deliver around 24% saving energy. For this increased duct sizes, the size of plantroom, riser and available NLA does not change dramatically. Therefore, the additional costs discussed earlier could be negligible.

4.2.2 Pumps and Water Distribution Systems

Chilled water pumps

From Table 3, it is observed that the energy consumption of the chilled water pumps (CHWP's) in the case study HVAC design is 28% less than the energy usage of chilled water pumps in the BCA compliant HVAC design. This percentage reduction was achieved by utilizing 7% larger chilled water pipe diameters for the case study HVAC design compared to the BCA compliant HVAC design. Note that the maximum chilled water pump power consumption per floor area of conditioned area should be equal to 1.9 W/m^2 (based on Table J5.2 of the BCA) [3]. However, for the case study HVAC design the total value of chilled water pump power consumption per floor area of conditioned area is 1.39 W/m^2. This means that for the case study HVAC design, the chilled water pump distribution systems have been designed far better than BCA minimum requirement.

Additionally, Figure 8 shows the breakdown of the energy consumption for the chilled water pumps in the case study HVAC design. From Figure 8, it is evident that around 86% of energy consumption in the chilled water pumps for the case study HVAC design is due to the energy required to move the water through the chilled water pipework, fittings and cooling coils. Additionally, it can be seen that less energy is consumed for pumping chilled water into the chilled water pipework and fittings in the case study HVAC design compared to the BCA compliant HVAC design. This is because of sizing the pipe systems with lower pressure gradient (between 100- 360 Pa/m) in the case study HVAC design in comparison to 600 Pa/m for the BCA design.

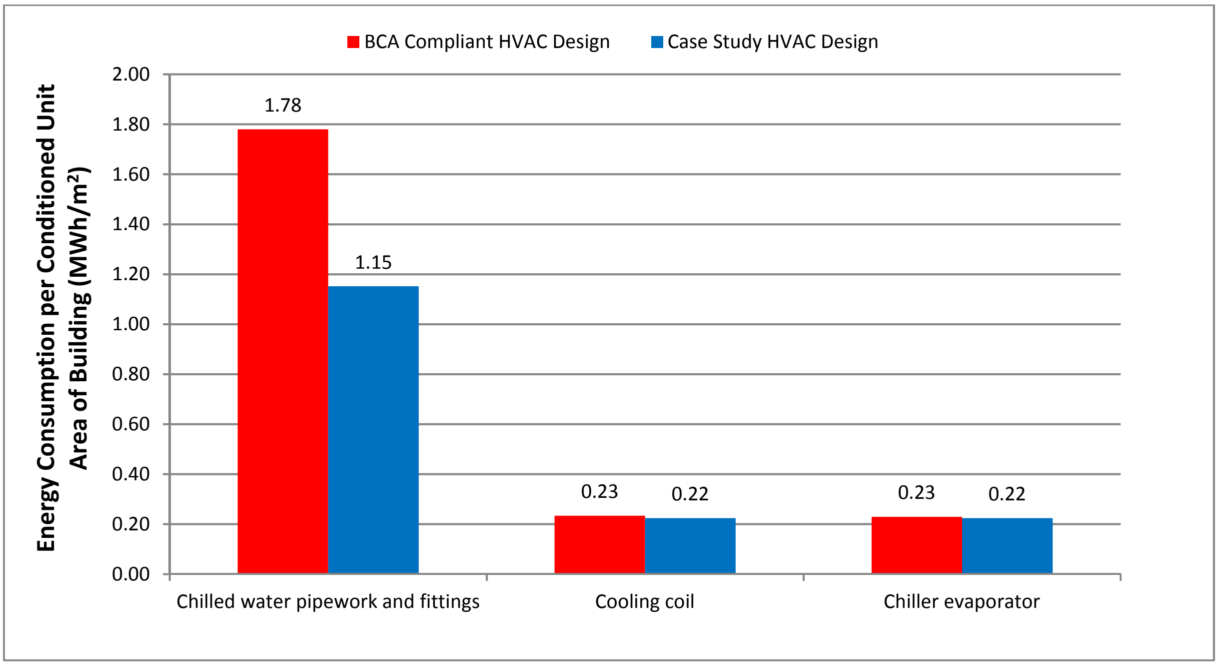

Figure 8. Detailed breakdown for energy consumption of chilled water pumps (CHWP's)

Hence, a better design of the piping equipment can reduce the energy consumption from the chilled water pumps.

Condenser Water Pumps

As shown in Table 3, the energy consumption of the chiller condenser water pumps (CWP's) in the BCA compliant HVAC design is 35% less than the energy usage of condenser water pumps in the case study HVAC design. This was due to the higher pressure drop in the condenser water piping network for the case study design. Note that the method of sizing condenser water pipes for the case study HVAC design was same as the method used for selecting chilled water pipes (pressure gradient of between 100- 360 Pa/m), however the maximum recommended power consumption for condenser water pumps should be less than this metric for chilled water pumps (based on Table J5.2 of the BCA) [3]. The maximum power consumption per floor area for condenser water pumps and chilled water pumps should be equal to 1.2 W/m^2 and 1.9 W/m^2 respectively (based on Table J5.2 of the BCA) [3]. Also, note that for the case study HVAC design the total value of pump power consumption per floor area of conditioned area is 1.96 W/m^2. This means that the power consumption of condenser water pumps for the case study design does not meet the minimum requirement of Table J5.2 of the BCA.

Figure 9 illustrates the breakdown of energy consumption for the condenser water pumps for the case study HVAC design. From Figure 9 it can be seen that around 53% of the energy consumption in the condenser water pumps for the case study HVAC design is due to the energy required to move the water through the condenser water pipework and fittings. Additionally, it can be seen that greater energy is consumed for pumping water into the condenser water pipework and fittings in the case study HVAC design. To achieve the BCA minimum requirement for the condenser water pumps, the condenser water pipe diameters in the case study HVAC design should increase by around 9%.

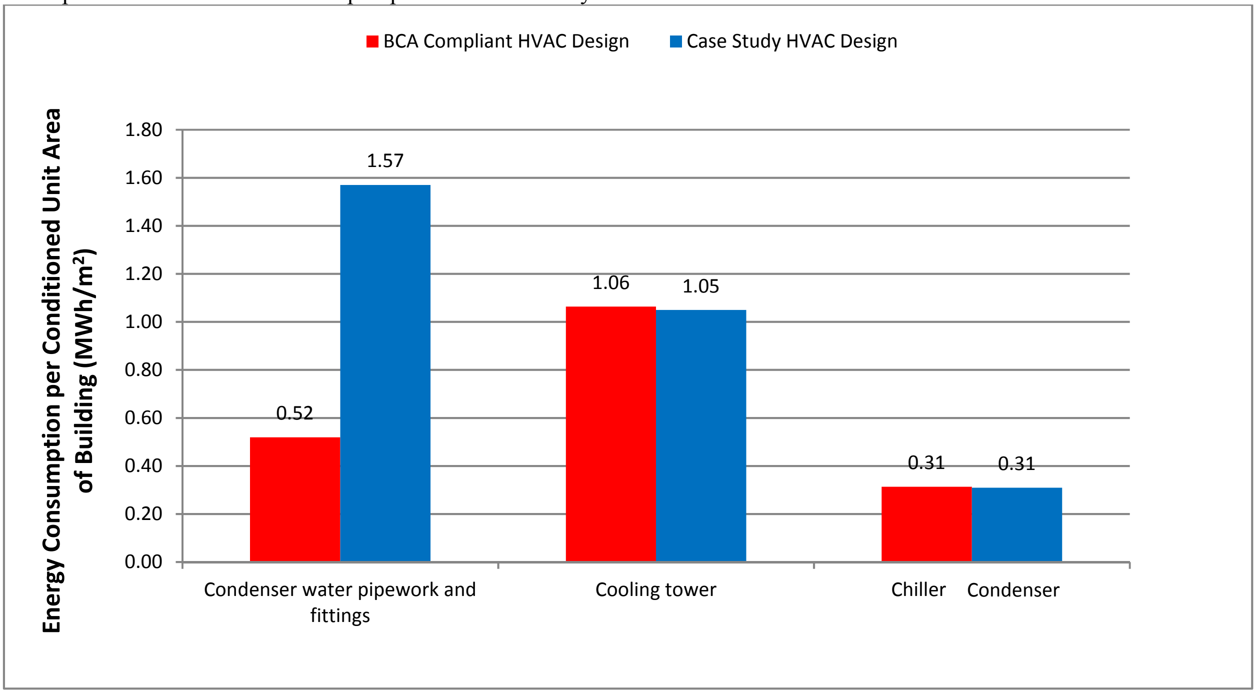

Figure 9. Detailed breakdown of energy consumption for condenser water pumps

Heating Hot Water Pumps

From Table 3, it can be seen that the energy consumption of the heating hot water pumps (HHWP's) in the case study HVAC design is 34% less than the energy usage of HHWP's in the BCA compliant HVAC design. This percentage reduction was achieved by utilizing 8% larger hot water pipe diameters for the case study HVAC design compared to the BCA compliant HVAC design. Note that the method of sizing heating hot water pipes for the case study design was the same as the method used for selecting chilled water pipes (pressure gradient of between 100- 360 Pa/m). Note that the minimum heating hot water pump power consumptions per floor area of conditioned area should be equal to 1.3 W/m^2 (based on Table J5.2 of the BCA) [3]. However, for the case study HVAC design the total value of pump power consumptions per floor area of conditioned area is 0.27 W/m^2.

4.2.3 Chillers, Boilers, Cooling Tower fans and Miscellaneous AC Units

As shown in Table 3, the energy consumption of the chillers and AC units in the case study HVAC design are 25% and 13% less than the energy usage of the chillers and AC units in the BCA compliant HVAC design respectively. This is simply due to more efficient chillers and AC units being selected for the case study HVAC design as shown in Table 4. Additionally, the boilers in the case study HVAC design have a slightly higher efficiency (see Table 4). However, the electrical energy and gas consumption of the boilers are almost equal for both HVAC designs due to a slight difference in the operation of the boilers for each design. Note that a slightly lower R-Value of ducts and an increase in the hot water pipe sizes in the case study HVAC led to a slightly greater heat loss during the heating season. Hence, the gas boiler was required to operate longer for the case study HVAC design.

Note that Table 3 of Specification J5.2b in the BCA recommends the R-values of 1.2 W/m^{2}K and 2 W/m^{2}K for insulation of ductwork located in the conditioned and unconditioned spaces not exposed to the direct sun light respectively [3]. However, for the case study HVAC design, the R-values of 1 W/m^{2}K and 1.5 W/m^{2}K were utilized for insulation of ductwork located in the occupied spaces and unoccupied spaces respectively.

Further, from Table 3 it can be seen that the total energy usage of the axial fans for the cooling towers in the BCA compliant

HVAC design is around 38% less than the usage of the cooling towers fans in the case study HVAC design. This was due to a lower efficiency cooling tower fan being used in the case study HVAC design.

In order to achieve the BCA minimum requirement, the value of the cooling tower fan power consumption per water flow rate should be equal to 310 W/ (L/s) (based on Table 3b from Specification J5.2a) [3]. Note that, the value of cooling tower fan per condenser water flow rate is 441 W/ (l/s) for the case study HVAC design.

Table 4, Major HVAC Equipment Efficiencies

Equipment	COP/ Efficiency (BCA Compliant HVAC Design)	COP/ Efficiency (Case Study HVAC Design)	IPLV[1] (BCA Compliant HVAC Design)	IPLV (Case Study HVAC Design)
Large Chillers	6	5.9	6.5	9.6
Small chillers	5.1	5.6	6	8.7
Boilers	80%	85%	N/A	N/A
AC Units	3.1	3.6	N/A	N/A

1. The Integrated Part Load Value (IPLV) is a common term used to describe the performance of chillers based on chillers capability for different capacities

4.2.4 Carpark and General Mechanical Ventilation Systems

From Table 3, it is evident that the energy usage of the car park general mechanical ventilation systems in the case study HVAC design is only slightly different from the energy consumption of the same systems in the BCA compliant HVAC design. This is due to a slight difference in the fan motor efficiency for those systems.

The average combined fan and motor efficiency for the car park fans in the case study HVAC design was 54%. However, for the BCA compliant HVAC design, an average combined efficiency of 51% was utilized for the car park fans in order to satisfy the minimum requirement of the BCA (Table 4b from Specification J5.2a) [3].

5. Economic Analysis

In this study, an economic analysis was carried out by comparing the net present values (NPV's) for both HVAC designs. The difference in the NPV of two HVAC design options can be calculated as follows [14]:

$$NPV_{BCA-CS} = (C_{CS} - C_{BCA}) + PWF \times (S_{BCA} - S_{CS}) \qquad (4)$$

where C is the capital cost, S is the annual running cost (energy only), PWF is the present worth factor and the subscripts BCA and CS designate the BCA design and case study design respectively.

The values of C_{BCA}, C_{CS} were estimated based on the data received from AECOM, the Rawlinson's Australian Construction Handbook [15] and equipment manufacturer (i.e. Carrier[16]). Additionally, the value of S_{BCA} and S_{CS} were calculated by multiplying the annual energy consumption of each option by the associated energy cost. Note that the annual energy consumptions were presented in Table 2 earlier. Also, an average electricity price of 0.15 $/kWh was used in this work based on the actual average rate of the electricity at the time.

The present worth factor PWF is given by [14]:

$$PWF = \frac{\left(1-\left(\frac{1+f}{1+i}\right)^{n}\right)}{\left(\left(\frac{1+i}{1+f}\right)-1\right)} \qquad (5)$$

where i, f and n are the interest rate, the inflation rate and number of years for the HVAC life span.

As shown in Table 5, additional capital cost was needed for the larger ductwork and pipes utilized in the case study HVAC design. Also, as discussed earlier the chillers, cooling towers and AC units are not the same for both HVAC designs investigated in this study. Table 5 details the capital cost differences for the HVAC design options in this study. Further, it was assumed that the capital cost of all HVAC equipment and services except the capital cost of equipment listed in Table 5 are the same for both designs. In addition, Table 6 shows the assumptions used for the economic analysis in this study.

Although not included in the economic analysis presented here it is proposed that the rental revenue of office spaces for both HVAC design options would be similar. This is because despite a slight reduction in the building NLA for the case of utilizing the case study HVAC design, .for a higher star rating building, the rental income per unit area would be higher offsetting the loss in NLA.

Table 5 shows the assumptions used for the economic analysis in this study.

Table 5. Details of capital cost differences for HVAC design options

Item number	**Item**	$C_{CS} - C_{BCA}$
1	Chillers	\$120,152
2	Cooling towers	-\$49,889
3	Package and split A/C units	\$7,949
4	Ductwork and insulation	\$30,515
5	Pipes and insulation	\$26,400
6	Total	\$135,112

Table 6. Parameter assumptions for economic analysis

Parameter	**Value**
Interest rate at the time (i) [17]	6%
Electricity inflation rate at the time (f) [17]	3%
HVAC system lifetime (n)[18]	20
Capital cost difference (from Table 5)	\$135,126.12
Running cost difference (considering annual energy saving only)	\$36360

Considering the assumptions shown in Table 6, a value of 14.99 was calculated for the PWF. Subsequently a value of $410,215 was calculated for NPV_{BCA-CS} taking into account the above assumptions. This means that utilizing the case study HVAC design leads to a significant reduction in total cost of HVAC system over 20 years compared to the BCA compliant HVAC design. Also, it is noticeable that the additional investment for the case study HVAC design was almost 1% of total capital cost of the BCA HVAC. Further, a discounted payback period of 4 years was calculated for the case study HVAC design investment compared to the BCA compliant HVAC design.

6. Barriers for Widespread Adoption of Case Study HVAC Design Methods constraint

As discussed for Figure 4 and Table 3, the energy consumption of the case study HVAC systems is significantly less than the energy usage of the BCA compliant HVAC systems mainly due to the significant reduction in the hydraulic losses of AHU's and utilizing energy efficient HVAC equipment. However, an energy efficient HVAC design such as the case study HVAC design may not be adopted widely by the designers for other similar building due to the following potential barriers based on anecdotal evidence:

- The developer / building owner seeks a design that meets a performance target but provides largest possible NLA.
- Utilizing larger ductwork and pipes may create space limitations for other services (i.e. hydraulic).
- Some HVAC designers and contractors prefer tried and tested methods to design and construct HVAC systems due to a desire to minimise possible errors and labour costs. Consequently, the methods employed by the building industry are often based on how it has been done for years, derived from rules-of-thumb and meeting the minimal energy efficiency provisions of the National Construction Code.

7. Conclusions

This study compared a high efficiency case study HVAC design with a standard BCA compliant design for an exemplar office building in Sydney achieved NABERS energy 5 star rating. The case study HVAC system is an air-water, low temperature VAV system and consists of a Chilled Water (CHW) cooling system and a gas-fired Heating Hot Water (HHW) plant connected to air handling units (AHU's). The AHU's pump conditioned air to the different zones of an office building (case study building) via VAV boxes. The case study HVAC system also consists of some supplementary HVAC systems (i.e. split/packaged AC units, car park and general mechanical ventilation systems).The validated model of the exemplar building with the high efficiency case study HVAC system was used to perform the analysis in this work. To model the BCA HVAC design, the case study HVAC design of the IES model was changed considering the minimum HVAC requirements of the BCA. Additionally, two possible interpretations with two different methods (lumped and individual) for the calculation of the maximum fan power consumption of AHU's in the BCA case were evaluated. Further, a method was proposed to split the combined total fan power of an AHU system calculated from Table 3a, Specification J5.2a in the BCA. It was identified that using the interpretation leads to applying lumped method provides greater stringency for the maximum total fan power of AHU's in this work. So, the lumped method should be used to investigate the compliance of AHU's fans of a HVAC system in a building.

In addition, the results of the building model simulations with the BCA compliant HVAC design and the case study HVAC designs were reviewed and compared. The following items are the major conclusions from the review and comparison of the simulation results in both HVAC designs:

- For a low temperature VAV, air-water BCA compliant HVAC system, the five largest energy consuming HVAC components sorted from the highest to the lowest usage are AHU's fans, chillers, boilers, pumps, cooling towers and general mechanical ventilation systems. While the five largest energy consuming HVAC components for this case study HVAC designs are boilers, chillers, AHU's fans, pumps, cooling towers and general mechanical ventilation systems.
- The annual electrical energy usage of the case study HVAC system is around 13% less than the annual electrical energy usage of the BCA compliant HVAC system.
- The total energy consumption of all supply and return air fans for the air handling units in the case study HVAC design is around 24% less than the consumption of the same fans in the BCA compliant HVAC design because of the lower pressure drops in the AHU's ductwork for the case study HVAC design. It is notable that in the case study design the main ductwork systems were sized considering a low pressure gradient (mainly between 0.8 and 1 Pa/m), reducing the transitions and fittings where it was possible and utilizing efficient ductwork fittings (i.e. round bend with splitter).
- The annual energy usage of the chillers and chilled water pumps for the case study HVAC design are around 25% and 28% less than the annual usage of the same components for the BCA compliant HVAC design respectively. This is because of utilizing more efficient chillers and sizing the pipe systems with lower pressure gradient (between 100- 360 Pa/m) in the case study HVAC design compared to the BCA compliant HVAC design. However, the annual energy consumption of cooling tower fans and pumps are greater than the energy consumption of the same components in the BCA compliant HVAC design. A simple solution can be selecting a cooling tower with a higher efficiency (i.e. fans with EC motors) and reducing the pressure drops of condenser water pipe networks (i.e. consideration for the location of the cooling tower and the condenser water pipe lengths). These initiatives can be investigated in future work.
- The energy usage of the packaged and split AC units for the case study HVAC design are approximately 13 % less than the units energy usage for the BCA compliant HVAC design due to the greater COP value of the units selected for the case study design.
- The energy consumption of other systems including heating hot water systems, carpark systems and general mechanical ventilation systems are essentially the same for both systems.

Finally an economic analysis was carried out for both HVAC designs. It was determined that the additional investment for the case study HVAC design is almost 1% of total capital cost of the BCA HVAC. Additionally, a discounted payback period of 4 years was calculated for the case study HVAC design investment compared to the BCA compliant HVAC design. Consequently, the widespread adoption of the design criteria applied for case study HVAC design is recommended for other low temperature VAV, all air HVAC systems in other similar buildings.

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Appendix A- Further Details of the Case Study Building HVAC System

Table A.1. Thermal Zoning – L4 to L30 Centre East, Centre West and North Perimeter Thermal Zones [5]

Level	AHU Ref.	Model Zones Served		AHU Ref.	Model Zones Served	AHU Ref.	Model Zones Served
Level 4	AHU-L03-CE (Centre East)	L4 CE		AHU-L03-CW (Centre West)	L4 CW	AHU-L03-PN (North Perimeter)	L4 N1
Level 5		L5 CE	L5 S1 CE		L5 CW		L5 N1
Level 6		L6 CE	L6 S1 CE		L6 CW		L6 N1
Level 7		L7 CE	L7 S1 CE		L7 CW		L7 N1
Level 8		L8 CE	L8 S1 CE		L8 CW		L8 N1
Level 9		L9 CE	L9 S1 CE		L9 CW		L9 N1
Level 10		L10 CE	L10 S1 CE		L10 CW		L10 N1
Level 11		L11 CE	L11 S1 CE		L11 CW		L11 N1
Level 12		L12 CE			L12 CW		L12 N
Level 13		L13 CE			L13 CW		L13 N
Level 14		L14 CE			L14 CW		L14 N
Level 15		L15 CE			L15 CW		L15 N1
Level 16		L16 CE			L16 CW		L16 N1
Level 17		L17 CE			L17 CW		L17 N1
Level	**AHU Ref.**	**Model Zones Served**		**AHU Ref.**	**Model Zones Served**	**AHU Ref.**	**Model Zones Served**
Level 18	AHU-L31-CE (Centre East)	L18 CE	L18 S1 CE	AHU-L31-CW (Centre West)	L18 CW	AHU-L31-PN (North Perimeter)	L18 N1
Level 19		L19 CE	L19 S1 CE		L19 CW		L19 N1
Level 20		L20 CE	L20 S1 CE		L20 CW		L20 N1
Level 21		L21 CE	L21 S1 CE		L21 CW		L21 N1
Level 22		L22 CE	L22 S1 CE		L22 CW		L22 N1
Level 23		L23 CE	L23 S1 CE		L23 CW		L23 N1
Level 24		L24 CE	L24 S1 CE		L24 CW		L24 N1
Level 25		L25 CE	L25 S1 CE		L25 CW		L25 N1
Level 26		L26 CE	L26 S1 CE		L26 CW		L26 N1
Level 27		L27 CE	L27 S1 CE		L27 CW		L27 N1
Level 28		L28 CE	L28 S1 CE		L28 CW		L28 N1
Level 29		L29 CE	L29 S1 CE		L29 CW		L29 N1
Level 30		L30 CE	L30 S1 CE		L30 CW		L30 N1

Table A.2. Thermal Zoning – L4 to L30 East and West Perimeter Thermal Zones [5]

Level	AHU Ref.	Model Zones Served			AHU Ref.	Model Zones Served
Level 4	AHU-L03-PE (East Perimeter)	L4 E1	L4 E2+3	L4 EN2	AHU-L03-PW (West Perimeter)	L4 W
Level 5		L5 E1	L5 E2+3	L5 EN2		L5 W
Level 6		L6 E1	L6 E2+3	L6 EN2		L6 W EDH
Level 7		L7 E1	L7 E2+3	L7 EN2		L7 W
Level 8		L8 E1	L8 E2+3	L8 EN2		L8 W
Level 9		L9 E1	L9 E2+3	L9 EN2		L9 W
Level 10		L10 E1	L10 E2+3	L10 EN2		L10 W
Level 11		L11 E1	L11 E2+3	L11 EN2		L11 W
Level 12		L12 E				L12 W
Level 13		L13 E				L13 W
Level 14		L14 E				L14 W
Level 15		L15 E1	L15 E2+3	L15 EN2		L15 W
Level 16		L16 E1	L16 E2+3	L16 EN2		L16 W
Level 17		L17 E1 EDH	L17 E2+3	L17 EN2		L17 W
Level	**AHU Ref.**	**Model Zones Served**			**AHU Ref.**	**Model Zones Served**
Level 18	AHU-L31-PE (East Perimeter)	L18 E1	L18 E2+3	L18 EN2	AHU-L31-PW (West Perimeter)	L18 W
Level 19		L19 E1	L19 E2+3	L19 EN2		L19 W
Level 20		L20 E1	L20 E2+3 EDH	L20 EN2		L20 W
Level 21		L21 E1	L21 E2+3	L21 EN2		L21 W
Level 22		L22 E1	L22 E2+3	L22 EN2		L22 W
Level 23		L23 E1	L23 E2+3	L23 EN2		L23 W
Level 24		L24 E1	L24 E2+3	L24 EN2		L24 W
Level 25		L25 E1	L25 E2+3	L25 EN2		L25 W
Level 26		L26 E1	L26 E2+3	L26 EN2		L26 W
Level 27		L27 E1	L27 E2+3	L27 EN2		L27 W
Level 28		L28 E1	L28 E2+3	L28 EN2		L28 W
Level 29		L29 E1	L29 E2+3	L29 EN2		L29 W
Level 30		L30 E1	L30 E2+3	L30 EN2		L30 W

Table A.3. Thermal Zoning – L0 (Ground) to L2 Thermal Zones [5]

Thermal Zone	AHU Ref	Model Zone		
		Level 0	Level 1	Level 2
Lobby Centre	AHU-L03-GF			
Lobby North Perimeter	NA (Trench heating)	L0 Trench		
Lobby West Perimeter		L0 W Trench		
Lift Lobby	AHU-L03-LL	L0 LL	L1 LL	L2 LL
Centre Zone	AHU-L03-L1C1		L1 C	
	AHU-L03-L2C1			L2 CE
				L2 CW
East Perimeter	AHU-L03-L1PE		L1 E	
	AHU-L03-L2PE			L2 E

Appendix B- Method of pressure drop calculations for ductwork/pipework systems

In this study, the total pressure drop of each fan or pump was calculated by adding up all the pressure drops in the associated index run (the ductwork or pipework path with the greatest total pressure drop). For each ductwork or pipework system the total pressure drop is calculated by combining the straight duct/pipe pressure drop and duct/pipes fitting pressure drop [18].

The pressure drop of each straight duct/pipe was calculated by multiplying the pressure gradient (Pa/m) of that straight duct/pipe and the duct/pipe length. Note that the pressure gradient of each duct/pipe can be obtained from the duct/pipe friction chart where the duct/pipe size and the fluid flow rate are known.

For fittings and other ductwork/pipework elements, the total pressure drop is given by [18]:

$$P_t = K_{loss} \times P_v \qquad \text{(B.1)}$$

where P_t, P_v, K_{loss} are the total pressure drop, the velocity pressure and the loss coefficient of the fitting or other ductwork/pipework elements respectively. Additionally, the velocity pressure is calculated by the following equation [18]:

$$P_v = 0.5 \times \rho \times v^2 \qquad \text{(B.2)}$$

where ρ and v are the fluid density and the average fluid velocity respectively.

For the case study HVAC design, the total pressure drop for each fan/pump was calculated by considering duct/pipe sizes, duct/pipe lengths, fittings and other ductwork/ pipework elements in the associated index run. AECOM provided the detailed data for the total pressure drop calculation in the index run of each air/water distribution system designed for the case study HVAC system.

For the BCA compliant HVAC design, the total pressure loss of each fan/pump was calculated by the following equation:

$$P_t = (\eta \times \dot{W}_{in})/\dot{V} \qquad \text{(B.3)}$$

where η, $\dot{W}_{in}$ and $\dot{V}$ are the combined fan/pump and motor efficiency, the fan/pump input power and the volume flow rate of the fluid respectively.

Note that the fluid flow rates and the efficiencies for all fans/pumps in the BCA compliant HVAC design were assumed to be the same as the value of these parameters used in the case study HVAC design. Additionally, the values of $\dot{W}_{in}$ were calculated considering the minimum requirements of the BCA for the fans and pumps as described in Table 2.

Appendix C- Possible Approaches for Calculating Fan Power of AHU Systems in BCA Compliant HVAC Design

Various Interpretations for Table 3a, Specification J5.2a in the BCA:

The BCA has provided the following statement for fan power consumption of an air conditioning system [3]:

"An air-conditioning system must be designed so that the fan motor power of the supply and return air fans as a combined total is in accordance with Table 3a"

Table 3a of the BCA is reproduced below (Table C.1).

Table C.1. NCC Table 3a. Maximum fan motor power of air-conditioning systems- supply and return [4]

Air-conditioning sensible heat load (W/m2 of the floor area of the conditioned space)	Maximum fan power (W/m2 of the floor area of the conditioned space)	
	For an air-conditioning system serving not more than 500 m2	For an air-conditioning system serving more than 500 m2
Up to 100	5.3	8.3
101 to 150	9.5	13.5
151 to 200	13.7	18.3
201 to 300	22.2	28.0
301 to 400	30.7	37.0
More than 400	See Note	

Note: Where the air conditioning sensible heat load is more than 400 W/m^2, the maximum fan power must be determined

a) in a building of not more than 500 m^2 floor area, using 0.09 W of fan motor power for each Watt of air conditioning sensible heat load and

b) in a building of more than 500 m^2 floor, using 0.12 W of fan motor power for each Watt of air conditioning sensible heat load.

The designers may interpret the above NCC statement in the following ways where the building air conditioning system consists of several air handling units (AHU's):

- **Method 1 (lumped method):** Some designers check the summed total fan motor power of all supply and return fans for all AHU's divided by the total conditioned floor area of the building against the associated value from Table 8. For example, for the Case Study building the value of maximum fan power per conditioned floor area, from Table 8 was 8.3 W/m^2, as the building load density was less than 100 W/m^2.
- **Method 2 (individual method):** Some other designers check the combined total value of the supply and return fans for each AHU individually against the associated value from Table 3a. Using this approach, the maximum fan power allowance from Table 3a varies depending on the load density of the conditioned space. For example, in the case study building the value of maximum fan power per conditioned floor area for one of the AHU's in one perimeter zone was equal to 28 (W/m^2), as the load density of that area served AHU is around 250 m^2. Note that calculating the maximum supply/return fan power allowed for individual AHU would be ambiguous if one return fan serves a number of AHU's. Unfortunately, there is a lack of information regarding splitting the total fan power between return and supply fans in the BCA. So, it is very important to have a standard method for splitting the power between the supply and return fans of an AHU.

Appendix D- Implication of Various Interpretations from Table 3a, Specification J5.2a in BCA:

Various interpretations of Table 3a (detailed in Appendix C) may have significant impact on the total fan power consumption allowed for an HVAC system of a building. For example, considering the lumped method for the case study building HVAC systems, the average building sensible load density is less than 100 W/m^2 and the conditioned floor area is 41,562 m^2. Therefore, the value for the maximum fan motor power allowed per conditioned floor area is 8.3 W/m^2 based on Table 3a of the BCA. Hence, the total fan power consumption of all supply and return fans for all AHU's will be approximately 345 kW.

However, if the second method (individual method) is utilized for the same building, the maximum fan power allowed per conditioned floor area of the AHU's will be different. This is due to the variation of the load densities for different zones served by AHU's as shown in Figure D.1. From Figure D.1, it can be seen that the maximum total fan power of the AHU's in the case study building can reach be as high as 28 W/m^2 when the sensible load density increases to 250 W/m^2 for one of the AHU's. Consequently, the maximum supply and return fan power of each AHU can be calculated considering the area served by that AHU (black dot point shown in Figure D.1). Using this approach and summing up all of the fan powers, the total fan power consumption of all supply and return fans for all AHU's will be around 479 kW.

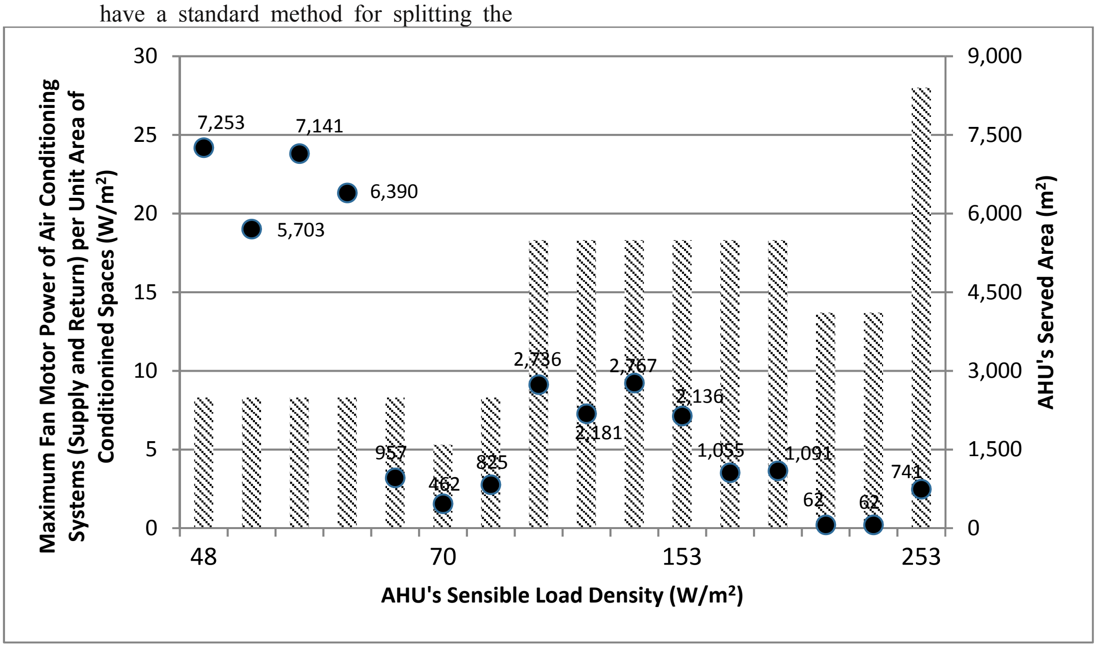

Figure D.1. Maximum fan motor power of air conditioning systems (supply and return) per unit area of conditioned spaces (bars), AHU's served zone areas (dot points) plotted against the AHU's sensible load density- individual method.

Additionally, the IES model was simulated considering the maximum fan powers calculated using the lumped and individual methods. Note that AECOM used lumped method for calculating maximum fan power for this project. Table D.1

presents the total annual energy consumption of supply and return fans using the lumped and individual methods for the BCA compliant HVAC design. From Table D.1 it is evident that using the lumped method provides greater stringency for the maximum total fan power of AHU's in this work. Therefore, the lumped method was adopted to calculate the max fan power from Table C.1 data in the rest of this study for the BCA compliant HVAC design.

Table D.1. Comparison between the fan energy consumptions of supply/return fans for AHU's using both methods

Method	Total energy usage of supply air fans for AHU's (MWh)	Total energy usage of return air fans for AHU's (MWh)	Total energy usage of supply and return air fans for AHU's (MWh)	Comments
Lumped Method	546	146	692	Using lumped method leads to almost 17% reduction energy usage compared to the individual method
Individual Method	667	162	829	