LOW CARBON LIVING CRC

RP 1033 - Mainstreaming High Performance Commercial Building HVAC

Stage 3 Report

Authors	Mehrdad Farshchimonfared, Alistair B Sproul, Jose I Bilbao,
Title	Stage 3 Report- RP 1033 High Performance Commercial Building HVAC
Date	20/06/2019

Australian Government
Department of Industry, Innovation and Science

Business
Cooperative Research Centres Programme

Acknowledgements

This research is funded by the CRC for Low Carbon Living Ltd supported by the Cooperative Research Centres program, an Australian Government initiative

Disclaimer

Any opinions expressed in this document are those of the authors. They do not purport to reflect the opinions or views of the CRCLCL or its partners, agents or employees.

The CRCLCL gives no warranty or assurance, and makes no representation as to the accuracy or reliability of any information or advice contained in this document, or that it is suitable for any intended use. The CRCLCL, its partners, agents and employees, disclaim any and all liability for any errors or omissions or in respect of anything or the consequences of anything done or omitted to be done in reliance upon the whole or any part of this document.

Peer Review Statement

The CRCLCL recognises the value of knowledge exchange and the importance of objective peer review. It is committed to encouraging and supporting its research teams in this regard.

The author(s) confirm(s) that this document has been reviewed and approved by the project's steering committee and by its program leader. These reviewers evaluated its:

- originality
- methodology
- rigour
- 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).

Contents

List of Tables

List of Figures

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

NPC Net Present Cost

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 3 of the LCL-CRC research project RP 1033 "High Performance Commercial Building HVAC".

This study utilised the knowledge gained form Stages 1&2 and examined the opportunities for further improving the HVAC design of a high-performance commercial office building (referred to as the case study building investigated in Stage 2). The focus of this study is air handling systems as it is a major component of HVAC energy consumption and the energy consumption is significantly influenced by design practice. Hence, substantial opportunities for fan energy improvement identified in Stage 2 such as optimizing duct sizes and air face velocity of coil or air filter were further analysed to reduce the pressure drop of air handling plants. This approach has the potential for significant carbon emission reductions.

This was done by investigating various design/ sizing methods for a large and a small air handling system in the case study building in terms of system energy use and cost. The results showed that the optimum duct sizing method is based on a duct pressure gradient ($\Delta P/L$) that varies between 0.4 and 0.8 Pa/m. Further, it was found that the AHU filters should be cleaned once the filter pressure drop exceeds 100 Pa. Moreover, an optimum air face velocity of 1.30 m/s was determined for selecting AHU coils and filters.

Subsequently, a validated IES model for the case study building and its associated high efficiency HVAC design (the case study HVAC design) was utilised to model two other HVAC designs (the improved HVAC design and the BCA compliant HVAC design) for the same building. The BCA compliant HVAC design is a design which just complies with the minimum HVAC energy requirements of the BCA. However, the improved HVAC design was created by developing the air handling systems of the case study HVAC design based on the adoption of the above methods for optimizing duct and designing coils/filters. Eventually, all IES models were simulated and the results were compared. It was identified that the fan power consumption of AHU's for the improved design is less than the usage of the same fans for the case study and BCA compliant HVAC design. Particularly, significant fan power savings occur in the large AHU's which have maximum fan power consumption of greater than 10 kW. Further, it was found that the total electrical energy consumption of AHU fans for the improved HVAC design is 29% and 45% less than the case study and BCA compliant HVAC design respectively. Finally, it was determined that the total electrical energy consumption of HVAC systems for the improved HVAC model is 13% and 24% less than the same metric for the case study and BCA compliant HVAC model respectively.

In conclusion six strategies were found to be technically and economically cost effective in reducing the energy requirements of AHUs for a large commercial building HVAC system.

- Duct systems were sized by utilizing an optimized duct sizing method – lowering the pressure gradient design criteria from 1 Pa/m to 0.4 - 0.8 Pa/m.
- Even larger energy savings (~40%) with only a slight increase in overall costs (~2%) were achieved for the largest AHU investigated with a pressure gradient design criterion of 0.2 – 0.4 Pa/m.
- More efficient fittings (fitting with lower loss coefficient factors) were utilized. For example, utilising bends with turning vanes instead of normal bends.
- The optimal maximum AHU filter pressure drop was found to be 100 Pa, which requires regular cleaning of filters to maintain this figure – a 100 Pa reduction from typical operational practice.
- The optimal air face velocity of AHU coils and filters was found to be 1.3 m/s – significantly lower than the AIRAH recommendation (2.25 m/s for cooling coils and 3.5 m/s for heating coils).
- A pressure drop of 25 Pa was assumed for VAV boxes based on the recommendation from REHVA [5].
- A pressure drop of 20 Pa was used for AHU outside air louvres considering larger and more efficient louvres.

Overall – these design strategies produced air handling systems with pressure drops on average 500 Pa lower than typical systems complying with the Building Code of Australia and industry recommended design rules. If such high efficiency air handling designs were adopted across all Australian non-residential buildings this approach would deliver CO_2 emissions reductions of 1.6 MT per annuum and financial savings of $255 M per annum [9].

.

1 Introduction

Standard HVAC systems are designed by adopting tried and tested methods to reduce the cost risk of design errors. these methods are often based on how they have been done for years, derived from rules-of-thumb and meeting the minimal energy efficiency provisions of the National Construction Code [1]. The Australian Building code has included minimum energy efficiency requirements for HVAC systems for commercial buildings since 2006 [1]. Since 2006 performance requirements have been adjusted and tightened with an updated version released in 1st May 2019. In parallel to this, industry has seen increased uptake on the use of energy and sustainability ratings as a way of differentiating buildings in a competitive market. Thus, many buildings that pursue such ratings exceed minimum energy efficiency requirements. In Stage 2 of this research, a case study high energy efficiency HVAC system design was compared to an HVAC system designed to meet the minimum BCA HVAC energy requirement (BCA compliant HVAC design). Table 1 represent the comparison of the electrical energy use of the HVAC components for the case study and the BCA compliant HVAC designs as described in the Stage 2 report (Mainstreaming High Performance Commercial Building HVAC- stage 2 Report). From Table 1, it can be seen the air handling units & return air fans (AHU's & RA Fans), chillers and pumps & water distribution systems are the most energy consuming elements in both HVAC system designs. Additionally, it is evident that the major reduction in the electrical energy consumption for the case study HVAC design is mainly due to the lower energy usage for the (AHU & RA) fans as well as utilization of more efficient chillers. Energy usage of AHU & RA fans is dependent on the fan efficiencies and pressure drops of ductwork systems and other air handling elements (i.e. filters, VAV boxes). Note that fans, chillers and other off-the-shelf equipment such as pumps have already received attention through ongoing manufacturer product innovation. This means that there is a much room for a better design and selection of the air handling system components in HVAC systems. Hence, it is important to examine various economically viable design methods for further improving the design of air handling systems in HVAC systems.

Table 1. Electrical energy use of the case study and BCA compliant HVAC design described in previous stage (stage 2) of this research

Major HVAC components	BCA compliant HVAC design		Case study HVAC design	
	Electrical energy use (MWh/yr)	Percentage of total electrical energy use of HVAC system	Electrical energy use (MWh/yr)	Percentage of total electrical energy use of HVAC system
AHU's & RA Fan Systems (Including Air Distribution Systems)	692	38.7%	530	33.3%
Chillers	593	33.1%	444	27.9%
Pumps and Water Distribution Systems	261	14.6%	316	19.9%
General Mechanical Ventilation systems (including carpark systems)	130	7.3%	130	8.2%
Cooling tower fans	57	3.2%	122	7.6%
Miscellaneous small AC & packaged units	56	3.1%	49	3.0%
Boilers	1.1	0.1%	1.1	0.1%

There has been significant literature focused on the energy reduction of air handling systems for HVAC systems (i.e. [2], [3], [4] and [5]) as shown in Table 2. It is notable that the elements listed in Table 2 are the most important elements of an air handling system which can have significant impacts on the total pressure drop of a system and consequently the energy use of an AHU fan. Some of these elements including ductwork, duct fittings, plenum box, VAV box, registers, diffusers, sound attenuators and louvres are located outside an AHU's casing and called external elements. However, other elements (i.e. filter and coils) are located inside an AHU casing and called internal elements. Some literature presented in Table 2 uses the tried and tested design manuals (i.e. [2] and [3]). Others have been released recently as green guidebooks to provide further recommendations and reduce energy use of HVAC systems (i.e. [4] and [5]). Further, Table 2 represents the air handling system design approaches adopted in the case study HVAC design investigated in the previous stage of this research (stage 2).

Table 2. Summary of literature review

Air handling system elements	Recommendations from previous literature	Case study HVAC design approaches
Supply riser and main duct	Maximum air velocity of 11 m/s [2]-using equal friction method the maximum pressure drop per length of duct should be 1 Pa/m of duct [2], Air velocity of 5.6- 8.1 m/s [3], Max pressure gradient of 0.65 Pa/m of duct run [4], Total pressure drop in Supply should be within the rage of 100-115 Pa [5].	Duct sized based on maximum pressure gradient of 0.8 to 1 Pa/m.
Return duct	Air velocity of 6 m/s [2]-maximum pressure gradient should be less than 1 Pa/m of duct [2], Max pressure drop of 10 Pa for duct run [4].	Maximum air velocity of 6 m/s
Branch riser and duct	Air velocity of 4.1- 4.6 m/s [3], Air velocity of 3.1-3.6 m/s [3]	Duct sized based on maximum pressure gradient of 0.8 to 1 Pa/m
Flexible duct	Air velocity of 3.5m/s [2]	20 pa
Duct fittings	The angle of a reduction transition should be no more than 45 degrees (if one side is used) or 22.5 degrees (if two sides are used) [4], The angle of expansion transitions should be no more than 15 degrees (laminar air expands approximately 7 degrees) [4]	Various duct fittings utilized
Plenum box	Maximum pressure drop in a plenum box should be 30 Pa [5].	Maximum pressure drop in a plenum box should be 20 Pa.
Cooling coil	Air face velocity of 2.25 m/s pressure drop of 100 to 250 Pa [2], Air face velocity of 1 to 4 m/s [6] Air pressure drop of 60 Pa [5], Air face velocity of 1.5 m/s [5]	Pressure drop 20 to 215 Pa. Air velocity of 1.5 to 2.3 m/s
Heating coil	Air velocity of 3.5 m/s pressure drop 50 to100 Pa [2], Air pressure drop of 40 Pa [5], Face velocity of 1.5 m/s [5]	Pressure drop 5-15 Pa/ air velocity of 1.5 to 2.5 m/s
Filter	Face velocity of 1.8 to 2.5 m/s [2], Minimum filter efficiency based on ISO 16890-1:2016, Air velocity of 1.5 m/s [3], Filter F3 bag 50 Pa, Filter M5 bag 75 Pa, Filter F9 bag 110 Pa [5]	A value of 200 Pa considered for the filter pressure drop allowance
VAV boxes	Maximum pressure drop of 25 Pa [5]	Up to 50 pa
Fans	Air face velocity of 1.8 to 2.5 m/s [2], Fan efficiency design based on [AS/NZ ISO 1275: 2013], Fan outlet duct should be sized at an air velocity of 6.6-10.2 m/s [3], 65% fan efficiency, 93% motor efficiency, and 95% variable-speed drive (VSD) efficiency [4]	Fan outlet duct sized at an air velocity of 6-8 m/s
Diffusers, Registers and Louvres	2.5 m/s for face velocity of supply air registers, Louvers-intake (2.5 m/s velocity through free area), Louvers- Exhaust (2.5 m/s velocity through free area) [2], Maximum static pressure of 20 Pa [5]	Diffusers and registers between 10 and 40 pa depending on the type of the outlet or inlet. Louver pressure drop was assumed to be 50 Pa

From Table 2, it is can be seen that there are different design recommendations for air handling system components. For example, using the duct design method recommended by AIRAH [2] leads to a duct size which is not equal to the duct size selected based on the method recommended by an international handbook such as ASHRAE [4]. Also, in some cases it is observed that the design method adopted in the good practice case study HVAC design is far better than the method recommended by the local design manuals. For instance, AIRAH [2] recommends an air face velocity of 3.5 m/s for sizing heating coils while the air face velocity of heating coils selected for the case study HVAC design is around 1.5 m/s. Another example is the different filter pressure drop allowances reported in Table 2.

Further, the comparison of HVAC energy consumption for a BCA compliant HVAC design and a good case study HVAC design showed that there is a potential for improving air handling system design by reducing energy use of the AHU fans. Hence, there is a need for a research study which examines major elements of air handling systems for commercial buildings in terms of energy and cost in Australia.

This study focuses on improving the design of air handling systems for a good practice air-water HVAC system in a case study building with the aim of reducing AHU’s fan energy consumption. The main HVAC system of the case study building 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 move conditioned air to the different zones of an office building via VAV boxes. The building also consists of some supplementary HVAC systems.

Initially, using data of a large and a small AHU system from the main HVAC systems some analysis were carried to identify the optimum design/sizing methods for some air handling elements (i.e. duct, filter and coil). Then, an improved HVAC system is modelled by adopting the optimized duct systems and AHU’s internal elements for the existing verified HVAC model of the case study building in IES. Finally, the simulation results of the improved HVAC design model are compared with the case study and the BCA compliant HVAC models created in the previous stage (stage 2) of this research work.

2 Method of optimizing air handling systems

A large and a small air handling system implemented in the case study building were investigated in this work. The large air handling system has an air flow rate of 21482 l/s and consists of a large AHU (AHU-L03-PE) located in the plantroom (level 3 of the building) and its associated ductwork system which serves seventeen levels of the case study building. The small air handling system has an air flow rate of 1486 l/s and consists of a small AHU (AHU-L03-L1C1) and a small ductwork system which serves one level of the case study building. Sections 2.1 & 2.2 describe the method used to optimize duct systems and AHU's elements.

2.1 Method to optimise ductwork system and other external AHU's elements

Using various duct design methods described in Appendix A, various ductwork systems were designed for the AHU. Then the net present cost (NPC) of each duct system was calculated using the following equation [7]:

$$NPC = C_0 + PWF \times C_e \qquad (1)$$

where C_0 is the capital cost of duct system, C_e is the annual fan electricity cost for moving air through the duct system and PWF is the present worth factor and is given by [7]:

$$PWF = \frac{\left(1-\left(\frac{1+f}{1+i}\right)^n\right)}{\left(\left(\frac{1+i}{1+f}\right)-1\right)} \qquad (2)$$

where i, f and n are the interest rate and the inflation rate and number of years for the duct system lifespan respectively. A value of 6% and 3% were assumed for i and f [8] and a value of n = 30 years was assumed [2].

The values of C_0 in Eq.(1) were estimated based on the data received from AECOM [9] and the Rawlinson's Australian Construction Handbook [10]. Additionally, the value of C_e in Eq. (1) was calculated by:

$$C_e = W_{fan} \times D_f \times h_d \times N_d \times H_{elec} \qquad (3)$$

where W_{fan} , D_f, h_d, N_d and H_{elec} are the fan power required for pumping air through the air distribution system, diversity factor for the fan operation, the fan operation hours per day, number of days that the fan operates in a year and the average electricity tariff respectively. Table 3 shows the assumptions for calculating the cost of electricity.

Table 3. Assumptions for some parameters used in calculation of energy cost

Parameter	Assumption	Parameter	Assumption
D_f	0.5	h_d	8 hours
H_{elec}	0.15 $/kWh	N_d	260 days (5 days per week and 52 weeks)

Also, the value of W_{fan} was given by (Cengel et al. [11]):

$$W_{fan} = \frac{\dot{V} \times \Delta P_{ads}}{\eta_{fan\,\&\,motor}} \qquad (4)$$

where $\dot{V}$ is the air quantity of the system, ΔP_{ads} is the pressure drop of the air distribution system and $\eta_{fan\,\&\,motor}$ is the combined fan and motor efficiency of the fan assumed to be 50% based on the data received from AECOM [9].

The value of ΔP_{ads} was determined by [2]:

$$\Delta P_{ads} = \Delta P_{duct} + \Delta P_{fitting} \qquad (5)$$

where ΔP_{duct} is the total pressure drop due to the straight ducts and $\Delta P_{fitting}$ is the total pressure drop associated with duct fittings.

The value of ΔP_{duct} can be calculated using the following expression [12]:

$$\Delta P_{duct} = \sum_{i=1}^{n} f_i \frac{L_i}{D_{h_i}} \frac{\rho\, V_{i\,avg}^2}{2} \qquad (6)$$

where L_i is the duct length of the i^{th} duct, D_{h_i} is the hydraulic diameter of the i^{th} duct, f_i is the friction factor of the i^{th} duct, and $V_{i\,avg}$ is the average velocity of the air in the i^{th} duct.

The hydraulic diameter of a duct is given by [13]:

$$D_h = \frac{4A}{P} = \frac{2DW}{(D+W)} \qquad (7)$$

where A, P, D, and W are the duct area, the duct perimeter, the duct depth, and the duct width, respectively. The friction factor for turbulent flow can be calculated using Eq. (8) [12].

$$\frac{1}{\sqrt{f}} = -2\log\left[\frac{2.51}{Re\sqrt{f}} + \left(\frac{\varepsilon_r/D_h}{3.7}\right)\right] \qquad (8)$$

where ε_r is the roughness and Re is the Reynolds number given by [13]:

$$Re = \frac{V_{avg} D_h}{\nu} \qquad (9)$$

where ν is the kinematic viscosity of air, which depends on the air temperature [13]. The value of $\Delta P_{fitting}$ is given by:

$$\Delta P_{fitting} = \sum_{i=1}^{n} K_i \frac{\rho\, V_{i\,avg}^2}{2} \qquad (10)$$

To optimise the duct design method, the *NPC* for different air distribution systems associated with different duct sizing methods was investigated. The duct design method which produced the minimum *NPC* was identified as the optimum duct design.

2.2 Method for optimum design and maintenance of AHU internal elements

The large and the small AHU systems described earlier were used to examine the optimum design and maintenance of the major internal elements (coils and filters) of the AHU's.

Initially, a broad range of filter pressure drop were investiagted (ΔP_f= 75 to 250 Pa) for the large and the small AHU's described earlier and the net filter maintenance costs were estimated using Eq. (11) for each case.

$$NPC_{filter} = (C_{cl} + C_{e-f}) \times PWF \quad (11)$$

where C_{cl}, is the annual cleaning cost of a filter, C_{e-f} is the estimated electricity cost for moving air through filter.

The value of C_{Cl} was identified by the following expression:

$$C_{cl} = C_{cl-typ} \times \frac{\Delta P_{f-typ}}{\Delta P_f} \quad (12)$$

where C_{cl-typ} is the typical annual cost of cleaning filter, ΔP_{f-typ} is typical filter pressure drop allowance and ΔP_f is the filter pressure drop allowances considered in this study. In this study, a value of 250 Pa was assumed for ΔP_{f-typ} based on the data received from AECOM for a standard AHU. Additionally, a cost of \$500 is estimated for C_{cl-typ} based on the information received from maintenance contractors.

The value of C_{e-f} was calculated using Eq. 13,

$$C_{e-f} = W_{fan-f} \times D_f \times h_d \times N_d \times H_{\ elec} \quad (13)$$

where W_{fan-f} is the fan power required for pumping air through the filter and is calculated using Eq. 14. Table 3 shows the values used for other variables in Eq. 13.

$$W_{fan-f} = \frac{\dot{V} \times \Delta P_f}{\eta_{fan\,\&\,motor}} \quad (14)$$

A value of 5 years was assumed for n to calculate PWF considering five years lifespan for filters based on AIRAH [2]. However, the value of other variables in Eq. (2) is same as the values estimated in Table 3.

To identify the optimum filter pressure drop allowance, sensitivity analysis of NPC_{filter} for different filter pressure drop allowances (ΔP_f) was investigated. The value of (ΔP_f) which caused the minimum NC_{filter} was identified as the optimum filter pressure drop allowance.

Another parameter which has a significant impact on the pressure drop variation of AHU internal elements is the air face velocity of AHU as described earlier. The large and the small AHU systems described earlier were used again to examine the optimum air face velocity of AHU's filter and coils. For this examination, the net present cost of AHU's was calculated by using Eq. (15) and assuming various values for the air face velocity V_f for the AHU's coils and filters between 0.8 to 2 m/s. Note that the heating/cooling capacity of coils were kept constant. In order to reduce the value of V_f in this analysis, AHU face areas were increased.

$$NPC_{AHU} = C_{0\_AHU} + PWF \times C_{e-AHU} \quad (15)$$

where C_{0-AHU} is the capital cost of AHU, C_{e-AHU} is the annual AHU fan energy cost for moving air through the AHU's filter and coils.

The value of C_{0-AHU} is identified by Eq. (16).

$$C_{0-AHU} = C_{AHU-avg} \times A_{AHU} \quad (16)$$

where $C_{AHU-avg}$ is the average capital price of AHU per surface area and calculated based on the data provided by AECOM [9] for the AHU's in the case study building, A_{AHU} is the surface area of AHU's calculated by using Eq. (17).

$$A_{AHU} = A_{face} \times L_{AHU} \quad (17)$$

where A_{face} is the face area of AHU and L_{AHU} is the length of the AHU.

The value of C_{e-AHU} was calculated using Eq.(18).

$$C_{e-AHU} = W_{fan-AHU} \times D_f \times h_d \times N_d \times H_{elec} \quad (18)$$

where $W_{fan-AHU}$ is the fan power required for pumping air through the filter and coils and is calculated using Eq. (19). Also Table 3 assumptions were used for other variables in Eq. (18).

$$W_{fan-AHU} = \frac{\dot{V} \times \Delta P_{AHU}}{\eta_{fan\,\&\,motor}} \quad (19)$$

where ΔP_{AHU} is the total pressure drop inside the AHU due to the losses in filter and coils and is identified as below:

$$\Delta P_{AHU} = \Delta P_{coils} + \Delta P_f \quad (20)$$

Also, the pressure drop in the coils and filter can be calculated using Eq's. (21) & (22).

$$\Delta P_{coils} = K_{coils} \times \frac{\rho\, V_f^2}{2} \quad (21)$$

$$\Delta P_f = K_f \times \frac{\rho\, V_f^2}{2} \quad (22)$$

For filters it is assumed that the filter type, material and depth are the same for all cases of each AHU design. Hence, the AHU filter loss coefficients (K_f) were considered to be equal to the loss coefficient of each AHU in the case study design. Note that the loss coefficient of all AHU filters in the case study design was provided by AECOM [9]. In addition, the loss coefficient of AHU coils (K_{coils}) was calculated based on the methods shown in Appendix C.

To identify the optimum air face velocity of AHU's, a sensitivity analysis of NPC_{AHU} was investigated considering different values of the air face velocity for AHU's coils and filters (V_f). The value of (V_f) which caused the minimum NPC_{AHU} was identified as the optimum face velocity of AHU's.

3 Method to investigate improved HVAC system

The three different HVAC system designs were evaluated using an existing IES model of the case study building and its associated HVAC system, provided by AECOM [9].The case study building is a commercial office building with a net lettable area (NLA) of 39,803 m2. 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) [14] 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 [9]. 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. 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 B.1 in Appendix B provides further details for the major components of the HVAC system in the case study building.

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 details of the second model can be found in the stage 2 report of this research. Further, the existing IES model of the case study building and its associated HVAC system was developed to create a third model which is called an improved HVAC design model.

The following elements remained constant for the case study and the improved HVAC design models:

- Building components
- Occupancy loads and schedules
- Lighting and office equipment loads and usage schedules
- Chillers
- Pumps and water distribution systems
- Cooling tower fans
- Miscellaneous small AC and Packaged Units

Additionally, the improved HVAC system consists of the same type/number of HVAC equipment considered for the case study design. However, the air handling system elements (i.e. ducts, risers and AHU dimensions) for the improved design are different from the case study design. Note that the flow rates of the fans were considered to be equal for all three HVAC design types. Hence, the total pressure drop for the air distribution systems differs in the improved HVAC design compared to the case study design due to the different duct sizes, different types/numbers of duct fittings and different size of AHU filters and coils utilized for these designs. Table 4 shows design criteria of the air handling system for all three HVAC design options investigated in this study.

Table 4. Design criteria of air handling systems for HVAC design options in this study

HVAC Design	Design Criteria
Case Study HVAC Design	• Duct sizing methods were based on methods shown in Table 2. • Usually, it was attempted to use efficient fittings (fitting with lower loss coefficient factors). However, some standard fittings (i.e. bend without turning vanes) were selected in some cases. • Maximum AHU filter pressure drop was assumed to be 200 Pa. • The AHU coils and filters were selected to have a face air velocity between 1.5 to 2.3 m/s. • A pressure drop of 50 Pa was assumed for VAV boxes. • A pressure drop of 50 Pa was used for AHU outside air louvres.
BCA Compliant HVAC Design	• Ductwork sized by utilizing constant pressure gradient method ($\frac{\Delta P}{L} = 1\ \text{Pa/m}$). Other duct system elements (i.e. duct fittings) were sized to have fan power consumptions complied with the Specification J5.2a from BCA [1]. • Maximum AHU filter pressure drop was assumed to be 200 Pa • The AHU coils and filters were selected to have a face air velocity between 1.5 to 2.3 m/s. • A pressure drop of 50 Pa was assumed for VAV boxes. • A pressure drop of 50 Pa was used for AHU outside air louvres.
Improved HVAC Design	• Duct systems were sized by utilizing optimized duct sizing method identified based on the results of optimization work in this study. • More efficient fittings (fitting with lower loss coefficient factors) were utilized. For example, utilising bends with turning vanes instead of normal bends. • Maximum AHU filter pressure drop was assumed to be equal to the optimum filter pressure drop allowance identified in the first part of this study. • Air face velocity of AHU coils and filters was assumed to be equal to the optimum air face velocity identified in the first part of this study. • A pressure drop of 25 Pa was assumed for VAV boxes based on the recommendation from REHVA [5]. • A pressure drop of 20 Pa were used for AHU outside air louvres considering larger and more efficient louvres (see Appendix D for an example)

4 Results and discussions

4.1 Optimization of duct sizing methods in air handling systems

The Figure 1a and 1b show the sensitivity analysis results of the costs and the average hydraulic diameter for different ductwork system designed based on the various constant pressure gradient methods ($\frac{\Delta P}{L}$ ranging from 0.2 to 1 Pa/m). Note that Figure 1a illustrates the results for a large air handling system with an air flow of 21482 l/s while Figure 1b demonstrates the results for a small air distribution system with an air flow of 1486 l/s as described in the methodology section.

From Figure 1a, it can be seen that constant pressure gradient method produces average hydraulic duct diameters (D_h) ranging from 0.98 to 1.4 m). Also, it is observed that using the method which considers a constant value of 0.4 Pa/m for $\frac{\Delta P}{L}$ leads to a lower Net Present Cost (*NPC*) compared to other constant values of $\frac{\Delta P}{L}$ (i.e. AIRAH [2] and ASHRAE [4] recommend $\frac{\Delta P}{L}$ = 1 Pa/m and $\frac{\Delta P}{L}$ = 0.65 Pa/m for sizing ducts respectively). Note that for a design utilizing the standard method of 1 Pa/m, the D_h is 1 m and the NPC is \$70,000 (ductwork \$35,000 and electricity costs \$35000) compared to a D_h is 1.2 m and NPC of \$60,000 (ductwork \$42,000 and electricity costs \$18000) for a 0.4 Pa/m design.

Figure 1b, shows that using the same duct sizing methods for the small air distribution system leads to a duct system with smaller average hydraulic diameters (D_h =0.47 to 0.67 m) compared to the large system. This is due to the significant lower air quantity in the small system as described earlier. Further, it is evident that using a higher constant pressure gradient ($\frac{\Delta P}{L}$ ≥0.6 Pa/m) leads to a lower value of *NPC* (i.e. where duct system is designed based on AIRAH [2] and ASHRAE [4] methods). The reason is that for a small air distribution system, the cost of fan energy is significantly less than the initial cost of ductwork, insulations and fitting used for that system. Hence, the *NPC* of ducts for smaller air handling systems is mainly dependent on the initial cost of those systems.

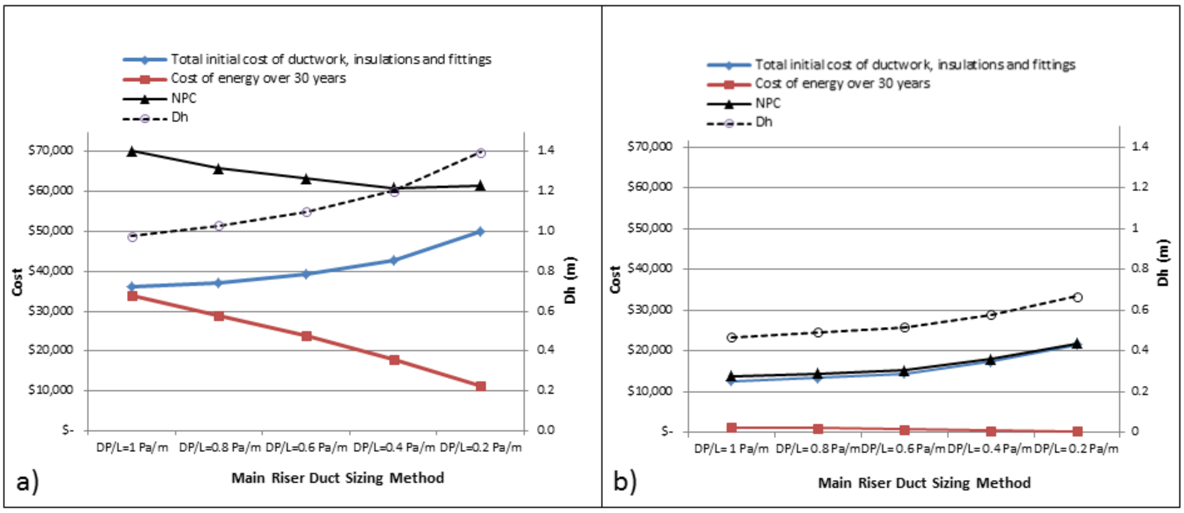


Figure 1. Sensitivity analysis for various costs and average hydraulic diameter of different ductwork systems sized based on different constant pressure gradient methods for a) a large air handling system with an air flow of 21,482 l/s and b) a small air handling system with air flow of 1486 l/s

Other design methods were explored to further examine the best method for sizing ductwork in a large or small air distribution system. Figures 2 and 3 show the results of adapting various methods described in Appendix A (excluding constant pressure gradient method shown in Figure 1) for sizing the main ducts of a large system. From Figure 2, it is evident that using these methods, the average hydraulic duct diameter (D_h) of the large system only increases by 0.35 m (D_h between 0.9 to 1.25 m).

This has a very small impact on the net lettable area of the occupied space served by this system. Also, it can be seen that the *NPC* is minimized when the duct system is sized using the method allows the duct pressure gradient ($\Delta P/L$) varies between 0.4 and 0.8 Pa/m. This is due to a lower energy cost over 30 years and a low initial cost of duct system sized based on this method compared to other ductwork systems sized based on the other methods illustrated in Figure 2.

Additionally, comparing Figures 1a and 2, it can be seen that the minimum *NPC* shown in Figure 2 ($\Delta P/L$ = 0.4 – 0.8 Pa/m) is around \$10,000 less than the minimum *NPC* shown in Figure 1a ($\Delta P/L$ = 0.4 Pa/m). For the $\Delta P/L$ = 0.4 Pa/m design it has an average main riser duct diameter of 1.2 m (Figure 1a), while the $\Delta P/L$ = 0.4 – 0.8 Pa/m design has a smaller average hydraulic duct diameter of 1.09 m (Figure 2). Despite this average duct diameter difference, note that the energy costs for both designs are similar. This is because for the $\Delta P/L$ = 0.4 Pa/m, constant friction gradient design, (Figure 1a), after each take off, a transition is required on the main riser which introduces additional pressure drops. For the $\Delta P/L$ = 0.4 – 0.8 Pa/m design (Figure 2), the main duct has far fewer transitions in the main duct riser. Hence, despite the smaller duct diameter for this design, the fewer transitions means that the energy costs are lower, and the duct costs are significantly lower than the $\Delta P/L$ = 0.4 Pa/m design.

Interestingly, for the $\Delta P/L$ = 0.2 – 0.4 Pa/m design (Figure 2), the average duct diameter is just over 1.2 m, and the energy costs are further reduced, by approximately 40%, with a similar rise in duct cost to give an *NPC* of $51,652 – a marginal increase on the *NPC* of $$50,524 for the $\Delta P/L$ = 0.4 – 0.8 Pa/m design (Figure 2).

From Figure 3, it can be seen that the average hydraulic duct diameter (D_h) of the small system varies by 0.4 m (D_h between 0.45 to 0.57 m). Again, this increase in the main duct riser size has a small impact on the NLA of the space conditioned by this system. Also, it can be seen that the *NPC* is minimized when the duct system is sized using the method allows the duct pressure gradient ($\Delta P/L$) varies between 0.6 and 1 Pa/m. This is due to a low cost of energy over 30 years and a low initial cost of duct system sized using that method compared to other methods illustrated in Figure 3.

Additionally, looking at Figures 1b and 3, it can be seen that the minimum *NPC*'s shown in both Figures are essentially similar due to the similar average hydraulic diameters and similar number of transitions for their associated duct system. For the duct sizing method which allows the $\Delta P/L$ to vary between 0.4 and 0.8 Pa/m, this leads to only a small increase in the *NPC* of about $1000 over 30 years compared to the optimum *NPC* shown in Figure 3. As the electricity usage of AHUs in a large building would be dominated by the larger AHU systems, the method of $\Delta P/L$ varying between 0.4 and 0.8 Pa/m is proposed for sizing ductwork in all air distribution systems. However, for large air handling systems similar to the case studied in Fig. 2 (~20,000 l/s), similar NPCs can be achieved with an even lower pressure design such as $\Delta P/L$ = 0.2 – 0.4 Pa/m. Appendix E shows further details for the duct systems sized based on the different methods examined in this study.

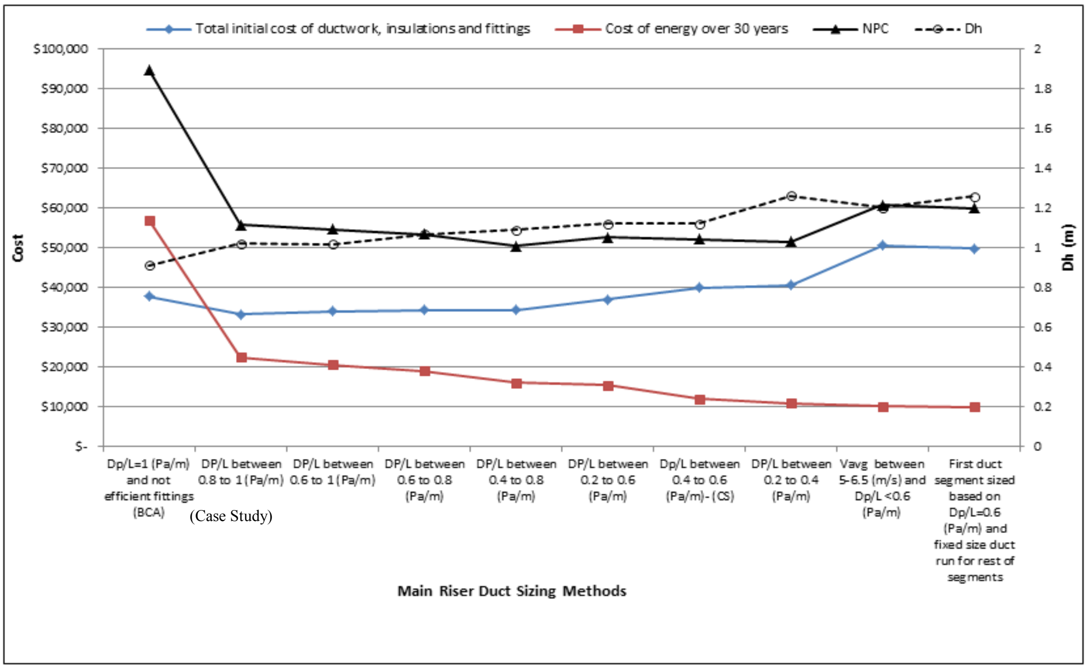


Figure 2. Sensitivity analysis for various costs and average hydraulic diameter of different ductwork systems sized based on different duct sizing methods in a large air distribution system

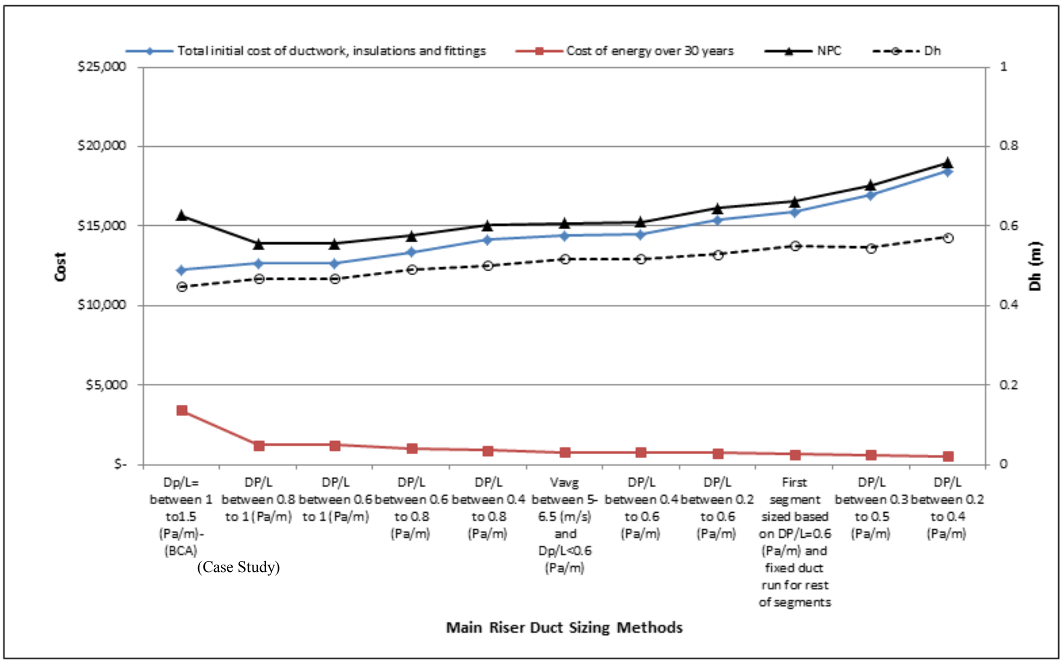


Figure 3. Sensitivity analysis for various costs and average hydraulic diameter of different ductwork systems sized based on different duct sizing methods in a small air distribution system

4.2 Recommendations to reduce AHU fan energy for moving air through filters and coils

4.2.1 Cleaning AHU Filters

Figures 4a and 4b demonstrate the sensitivity of *NPC*'s for AHU filters over 5 years, annual energy cost for moving air through the filter as well as annual cost for cleaning filters considering different filter pressure drops (ΔP between 75 Pa to 250 Pa) in the large and the small air distribution systems described in section 2. From Figure 4a, it is observed that the *NPC* over 5 years is minimised when the filter pressure drop does not exceed 100 Pa. Therefore, it is recommended to clean the filter for a large AHU when the differential pressure drop (ΔP) across the filter reaches 100 Pa. Note that despite lower cleaning cost for higher ΔP (i.e. 200 Pa) the annual fan energy cost associated with the filter is increased substantially.

From Figure 4b, it is evident that by cleaning the filter at 175 Pa the *NPC* over 5 years became minimised for the small AHU. Further, it can be seen that the *NPC* varies slightly for regular cleaning of the filter for a wide range of ΔP between 100 to 250 Pa. For example, if the value of 100 Pa is used for cleaning a filter in a small AHU, the *NPC* over 5 years will increase only by $170. This is due to the low annual energy cost as well as low cleaning cost for filters in a small AHU compared to these costs in a large AHU. From the above discussions, it is recommended to clean an AHU filter when the pressure drop across the filter reaches to 100 Pa. Note that this pressure drop allowance is significantly less than the typical allowances considered for AHU filters in the case study design (ΔP= 200 Pa).

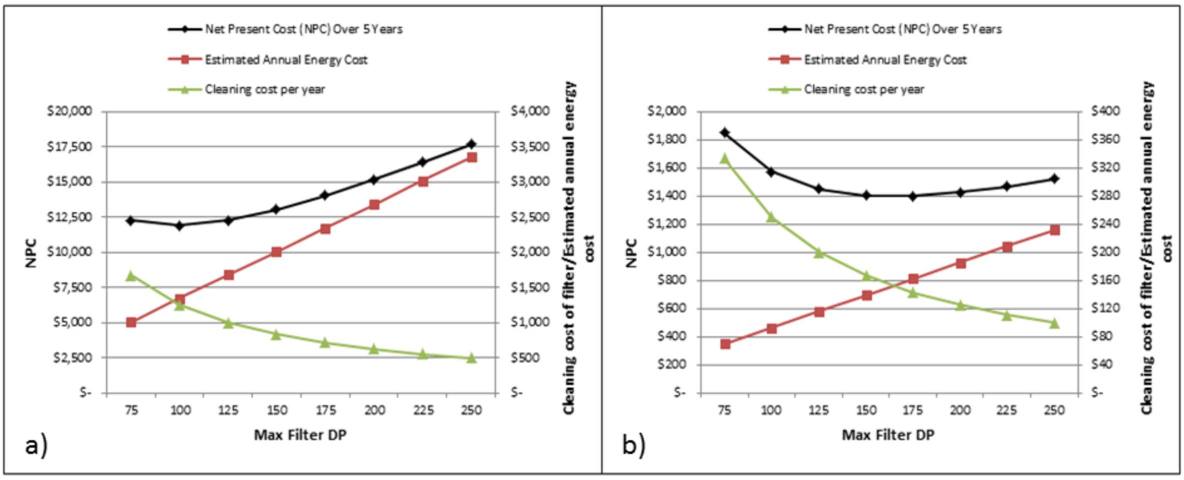


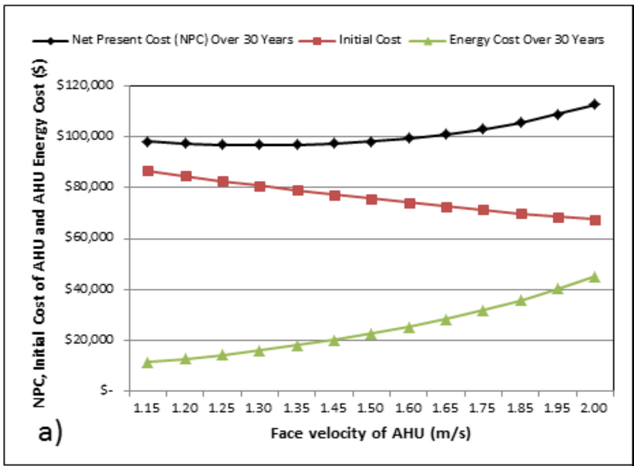
Figure 4. Sensitivity analyses of NPC, annual energy cost and filter cleaning cost for cleaning filters at different conditions for a) a large air handling systems with an air flow of 21,482 l/s and b) a small air handling system with air flow of 1486 l/s

4.2.2 Decreasing the face velocity of AHUs

Figures 5a and 5b demonstrate the results of sensitivity analysis for the AHU's fan energy cost, the AHU's initial cost and the *NPC* considering various air face velocities through the coils and filters for the large and the small air distribution system described earlier.

From Figure 5a, it is observed that the *NPC* over 30 years is minimised when the air face velocity of coils and filters is equal to 1.30 m/s. Note that for higher air face velocities (V_f ≥1.50 m/s), the AHU fan energy consumption for moving the air through the coils and filters increases significantly (see Eqs. (19 to 22). Further, for the lower velocities (V_f ≤1.50 m/s), the initial cost of AHU (including cost of bigger coils and filter) increases significantly. However, the NPC values are essentially the same for the lower velocities due to the significant decreases in the fan energy costs. Therefore, an air face velocity of 1.30 m/s for AHU coils and filters within a large AHU serves a large air distribution system is recommended. Note that Table E.2 from Appendix E shows further details for the AHU coils/filters sized based on the different air face velocities presented in this study.

From Figure 5b, it is evident that the optimum values of V_f for a small AHU serves a small air distribution system is around 0.87 m/s. Further, it is observed that the *NPC* values of AHU's change slightly for a broad range of air face velocities (V_f between 0.83 to 1.35 m/s). For example, the NPC of AHU's chosen based on the above range of V_f only varies by approximately $500 over 30 years. This is due to the small variations in the energy cost and the initial cost of the small AHU's selected. Hence, the air velocity of 1.30 m/s recommended earlier can also be used for design of coils and filters in small AHU's. Note that the minimum air face velocity recommended from the previous literature listed in Table 2 is 1.5 m/s (REHVA recommendation [5]).



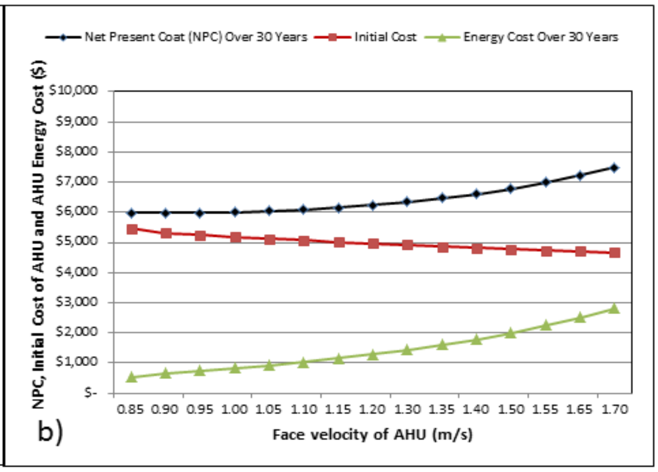


Figure 5. Sensitivity analyses of NPC, annual energy cost and initial cost for AHU's with various face velocities through the AHU coils and filters for a) a large air handling systems with an air flow of 21,482 l/s and b) a small air handling system with air flow of 1486 l/s

4.3 Comparison of improved HVAC system with the case study and BCA design

Figure 6 shows the AHU fan power consumption for the three HVAC design investigated in this study. From Figure 6 it is observed that using the improved HVAC design approaches, the AHU fan power consumption is reduced significantly compared to fan power consumption of AHU's selected for the case study and the BCA compliant HVAC design. Additionally, it can be seen that significant fan power savings occur in the large AHU's (AHU's have maximum fan power consumption of greater than 10 kW) selected for the improved HVAC design. This is due to the higher range of air quantities in the larger AHU's which lead to the higher value of total system pressure drop.

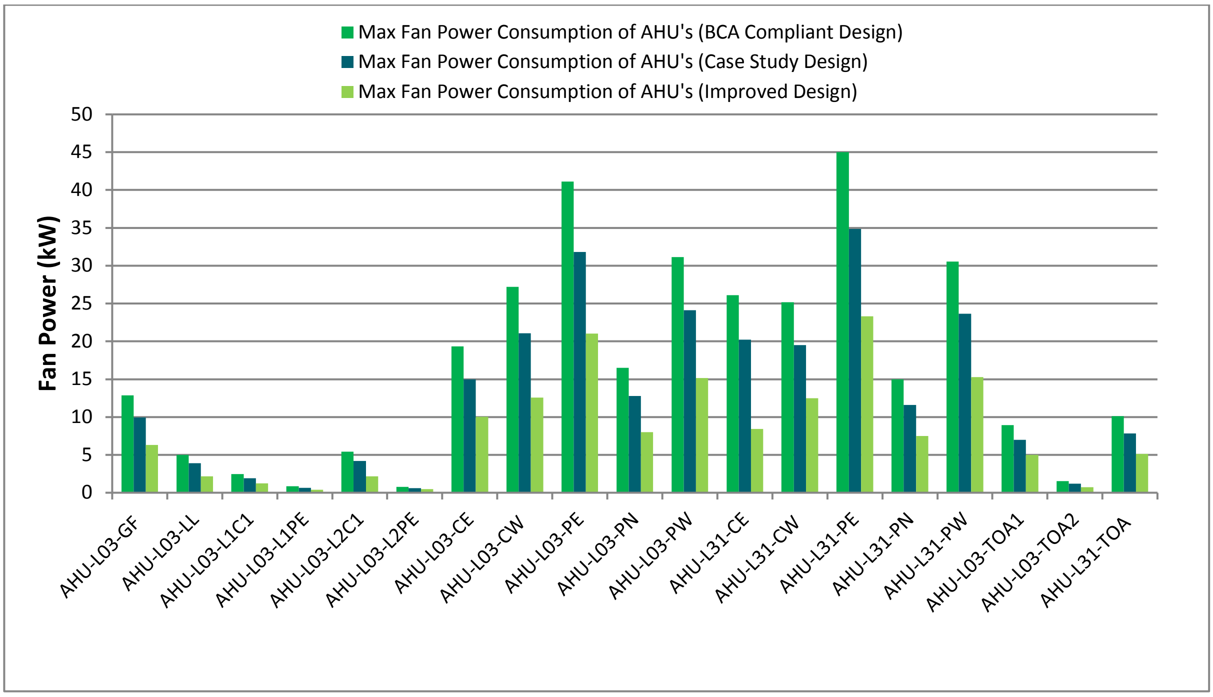


Figure 6. Various AHU fan power consumption for different HVAC design options in this study

Table 4 shows the total energy consumption of the HVAC and AHU's and average pressure drop for the three designs from the IES models. Looking at Table 4, it is evident that the total electrical energy consumption of all HVAC systems in the improved HVAC design is 13% and 24% less than the case study and the BCA compliant HVAC design respectively.

In addition, the electricity use of the AHU's in the improved HVAC design is 29% and 45% less than the case study and BCA compliant HVAC design respectively. Also the average pressure drops of the AHU's in the improved design are 260 and 500 Pa lower than the case study and BCA compliant HVAC design respectively.

If these simple design changes were made for all HVAC systems in non-residential buildings in Australia this would deliver CO_2 emissions reductions of 1.6 MT per annum and financial savings of \$255 M per annum. These calculations are based on data provided by AECOM [9] and using \$0.15/kWh for electricity.

Hence, it is recommended to adopt the identified improved HVAC design methods for sizing air handling systems of HVAC systems (particularly for larger AHU's that have maximum fan power consumption of greater than 10 kW).

Table 4. Electrical energy consumption of HVAC and AHU systems and average pressure drop for different HVAC designs.

	BCA Compliant HVAC Design	Case Study HVAC Design	Improved HVAC Design
Total Electricity Use of HVAC System (MWh/year)	1790	1574	1376
Total Electricity Use of AHU Systems (MWh/year)	692	529	380
Average Pressure drop (Pa)	940	700	440

5 Conclusions

Initially, this study optimised the method of sizing/design for the major elements of air handling systems within the HVAC systems of the commercial buildings. This was done by investigating various design/ sizing methods of a large and a small air handling system for an exemplar office building (case study building) in terms of system energy use and cost. The exemplar office building located in Sydney achieved NABERS energy 5.5 star rating. The HVAC system of the exemplar building is called the case study HVAC system. 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 the 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).

Subsequently, two other HVAC designs (improved and BCA compliant HVAC design) were modelled using the existing verified IES model of the case study building and its associated HVAC system, provided by AECOM. 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 third model is called the improved HVAC model and was created by developing the air handling systems of the case study HVAC model. This was achieved by using the optimized methods determined in the first part of this study to redesign/resize the major elements of air handling systems. Finally, the above models were simulated in IES and the energy consumption of the HVAC systems and AHU's for all model were compared. The following items are the major conclusions from the review and comparison of the simulation results for all HVAC design models in this study:

- Power consumption of AHU fans for the improved HVAC design is reduced significantly compared to power consumption of the AHU fans selected for the case study and BCA compliant HVAC design. Additionally, significant fan power savings occur in large AHU's (AHU's which have maximum fan power consumption of greater than 10 kW).
- Total electrical energy consumption of AHU fans for the improved HVAC design is 29% and 45% less than the case study and BCA compliant HVAC design respectively.

Total electrical energy consumption of HVAC systems for the improved HVAC model is 13% and 24% less than the same metric for the case study and BCA compliant HVAC model respectively.

In conclusion six strategies were found to be technically and economically cost effective in reducing the energy requirements of AHUs for a large commercial building HVAC system.

- Duct systems were sized by utilizing an optimized duct sizing method – lowering the pressure drop design criteria from 1 Pa/m to 0.4 - 0.8 Pa/m.
- Even larger energy savings (~40%) with only a slight increase in overall costs (~2%) were achieved for the largest AHU investigated with a pressure gradient design criterion of 0.2 – 0.4 Pa/m.More efficient fittings (fitting with lower loss coefficient factors) were utilized. For example, utilising bends with turning vanes instead of normal bends.
- The optimal maximum AHU filter pressure drop was found to be 100 Pa, which requires regular cleaning of filters to maintain this figure – a 100 Pa reduction from typical operational practice.
- The optimal air face velocity of AHU coils and filters was found to be 1.3 m/s – significantly lower than the AIRAH recommendation (2.25 m/s for cooling coils and 3.5 m/s for heating coils).
- A pressure drop of 25 Pa was assumed for VAV boxes based on the recommendation from REHVA [5].
- A pressure drop of 20 Pa was used for AHU outside air louvres considering larger and more efficient louvres.

Overall – these design strategies produced air handling systems with pressure drops on average 500 Pa lower than typical systems complying with the Building Code of Australia and industry recommended design rules. If such high efficiency air handling designs were adopted across all Australian non-residential buildings this approach would deliver CO_2 emissions reductions of 1.6 MT per annuum and financial savings of $255 M per annum [9].

6 References

[1] Australian Building Codes Boards (ABCB). National Construction Codes-Building Codes of Australia Volume one. 2016.

[2] Australian Institute of Refrigeration Air Conditioning and Heating. Technical Handbook. 4th Edition. Melbourne: 2007.

[3] Grondzik WT. Air-Conditioning System Design Manual. Second Edi. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE); 2007.

[4] American Society of Heating Refrigerating and Air-Conditioning Engineers. Advanced energy design guide for small to medium office buildings; achieving 50% energy savings toward a net zero energy building. Atlanta, GA, USA: Ringgold Inc; 2011.

[5] Brelih, Njec; Seppanen, Olli; Bertilsson, Thore; Maripuu, Marii-Liis; Lamy, Herve; Borre A Vanden. REHVA Guidebook No. 17 - Design of Energy Efficient Ventilation and Air-Conditioning Systems. Brusseles, Belgium: Federation of European Heating Ventilation and Air-Conditioning Association (REHVA); 2012.

[6] American Society of Heating Refrigerating and Air-Conditioning. ASHRAE Handbook - Heating, Ventilating, and Air-Conditioning Systems and Equipment (SI Edition). Atlanta, GA, USA: 2012.

[7] Bullinger CE. Engineering Economic Analysis. New York: McGraw-Hill; 1950.

[8] Reserve Bank of Australia 2018.

[9] AECOM Pty Ltd 2018. https://www.aecom.com/.

[10] Rawlinson Australian Construction Handbook. Perth: Rawlinson; 2017.

[11] Cengel Y, Turner R. Fundamentals of Thermal-Fluid Sciences 4th Edition. McGraw-Hill; 2012. doi:10.1115/1.1421126.

[12] American Society of Heating Refrigerating and Air-Conditioning. ASHRAE Handbook - Fundamentals (SI Edition). Atlanta, GA, USA: 2013.

[13] Cengel Y, Cimbala J. Fluid Mechanics: SI Unit: Fundamental Sciences. 2nd edition. New York: McGraw-Hill. 2009.

[14] NABERS 2018. https://nabers.gov.au/public/webpages/home.aspx.

[15] McQuay Duct Sizer 2019. https://hvac-works.blogspot.com/2018/10/mcquay-duct-sizer.html.

7 Appendix A- Further details for ductwork design methods

There are various common methods to size ductwork in an air distribution system as shown in Table 2 in Section 1. Some design handbooks (i.e. [2] and [3]) recommend the velocity reduction method to size ducts based on a specified velocity for each duct segment. Note that few of these handbooks suggest an air velocity range for this duct design method (i.e. air velocity of 5.6-8.1 m/s for supply ducts based on [3]). Whilst, other guidebooks recommend selecting the air velocity on the basis of designer experience (i.e. [2]). As an example, Table A.1 compares different parameters in two duct systems (a large and a small system) sized based on an air velocity of 6.5 m/s through ducts using McQuay duct selection program [15]. From Table A.1 it is observed that using this method for the small system leads to the smaller hydraulic duct diameter compared to the large system due to the greater air quantity for the large system. Hence, for a given duct length and air velocity, the pressure drop of the smaller duct becomes less than the pressure drop of the bigger duct (see Eq.(6)).

Table A.1. Example of duct sizing for two systems based on air velocity of 6.5 m/s through duct

Air delivery system Type	Duct Design Method	Air quantity (l/s)	Velocity (m/s)	Duct width (mm)	Duct depth (mm)	Hydraulic Diameter (mm)	Friction factor	Reynold Number	Pressure drop per metre (Pa/m)
Large System	Velocity Method	20000	6.5	2100	1700	1879	0.01372	865888	0.177
Small System	Velocity Method	1000	6.5	450	375	409	0.01879	193473	1.07

Another common duct design method is the constant pressure gradient method. In this method a constant pressure gradient (pressure drop per unit length of straight duct) is used to size each duct segment. The recommended value of the constant pressure gradient varies for various design handbooks (see Table 2 in Section 1). It is notable that adopting this method for duct sizing, the duct size is needed to change after a change in the air flow rate. Hence, a duct transition is required after each duct take-off from the main ducts sized based on a constant pressure gradient method. The pressure drop of duct transitions would be problematic and increase substantially where the main duct serves several sub-branches and there are several duct transitions. Therefore, it may be better to keep the duct size constant for some duct segments and reduce number of duct transitions. This occurs where the duct segments are sized based on a reduction pressure gradient method (i.e. DP/L between 0.4 to 0.8 Pa/m) rather than a constant pressure gradient method. For example, Figure A.1 shows the schematic of a main duct riser sized based on a duct design method which allows the value of $\Delta P/L$ varies between 0.4 to 0.8 Pa/m. As shown in Figure A.1 and Table A.2, the first duct segment is sized to have a pressure gradient of 0.8 Pa/m. Then the duct size remains constant for other segments as long as the value of $\Delta P/L$ is equal or greater than 0.4 Pa/m (duct size changes on Level 11). This sizing process is repeated for the rest of duct segments.

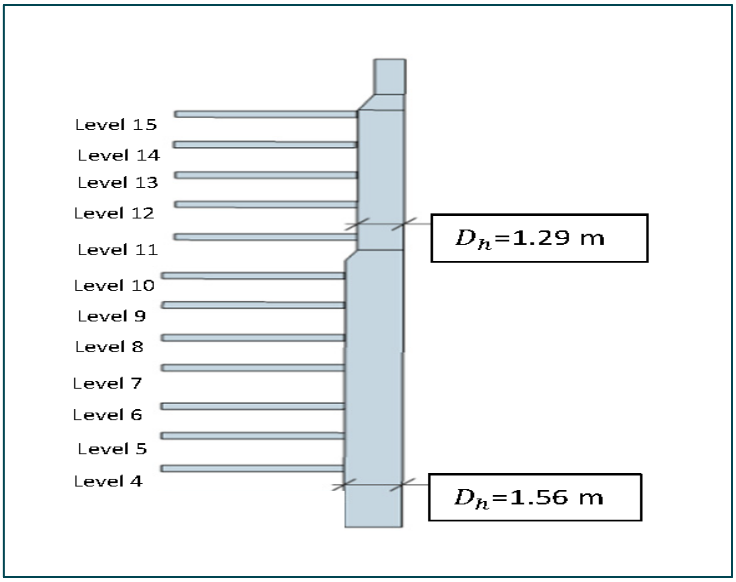


Figure A.1. Example of duct schematic sized based on a reduction pressure gradient method - ($\Delta P/L$ between 0.4 to 0.8 Pa/m)

Table A.2. Further details for a duct system shown in Figure A.1

Duct Run	Duct Segment	Air flow (l/s)	Duct Width (mm)	Duct Depth (mm)	*Dh* (mm)	Pa/m	Duct length (m)	Duct Run (m)
Duct to L4	1	30,000	1,700	1,450	1565.079	0.793	3	3
Duct to L5	2	28,500	1,700	1,450	1565.079	0.718	3	6
Duct to L6	3	27,000	1,700	1,450	1565.079	0.647	3	9
Duct to L7	4	25,500	1,700	1,450	1565.079	0.579	3	12
Duct to L8	5	24,000	1,700	1,450	1565.079	0.515	3	15
Duct to L9	6	22,500	1,700	1,450	1565.079	0.455	3	18
Duct to L10	7	21,000	1,700	1450	1565.079	0.398	3	21
Duct to L11	8	19,500	1,700	1050	1298.182	0.806	3	24
Duct to L12	9	18,000	1,700	1050	1298.182	0.691	3	27
Duct to L13	10	16,500	1,700	1050	1298.182	0.584	3	30
Duct to L14	11	15,000	1,700	1050	1298.182	0.486	3	33
Duct to L15	12	13,500	1,700	1050	1298.182	0.397	3	36
Duct to L16	13	12,000	1,700	750	1040.816	0.791	3	39

Considering the above discussion, various duct sizing methods listed in Tables A.3 were used in this study.

Table A.3. Description of methods used in this study

Method No	Duct sizing method- Description
1	Constant pressure gradient method - ($\Delta P/L = 0.2$ Pa/m)
2	Constant pressure gradient method - ($\Delta P/L = 0.4$ Pa/m)
3	Constant pressure gradient method - ($\Delta P/L = 0.6$ Pa/m)
4	Constant pressure gradient method - ($\Delta P/L = 0.8$ Pa/m)
5	Constant pressure gradient method - ($\Delta P/L = 1.0$ Pa/m)
6	Constant pressure gradient method- BCA compliant design method
7	Combination of reduction velocity method and constant pressure gradient- (V = 5 to 6.5 m/s and $\Delta P/L \leq 0.6$ Pa/m)
8	First duct segment sized based on $\Delta P/L = 0.6$ Pa/m and fixed size duct runs for the rest of segments
9	Reduction pressure gradient method- Case study design method - ($\Delta P/L$ between 0.8 to 1 (Pa/m))
10	Reduction pressure gradient method - ($\Delta P/L$ between 0.6 to 1.0 (Pa/m))
11	Reduction pressure gradient method - ($\Delta P/L$ between 0.6 to 0.8 (Pa/m))
12	Reduction pressure gradient method - ($\Delta P/L$ between 0.4 to 0.8 (Pa/m))
13	Reduction pressure gradient method - ($\Delta P/L$ between 0.2 to 0.6 (Pa/m))
14	Reduction pressure gradient method - ($\Delta P/L$ between 0.4 to 0.6 (Pa/m))
15	Reduction pressure gradient method - ($\Delta P/L$ between 0.2 to 0.4 (Pa/m))

8 Appendix B-Further details for case study building and associated HVAC system

Table B.1 shows further details for the case study HVAC system described earlier in section 2:

Table B.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°C and 14°C 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 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.
Supplementary Heating/ Cooling	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

System	System Components	Type	Description
Systems			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 comprise 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.

9 Appendix C- Loss coefficient calculations for AHU coils

The pressure drop of a AHU coil can be calculated using the following equation based on ASHRAE Heating, Ventilation, Air Conditioning Systems and Application [6]:

$$\Delta P_{coil} = \left(\frac{\Delta P_{st}}{r}\right) \times N_r \quad \text{(C.1)}$$

where $\left(\frac{\Delta P_{st}}{r}\right)$ is the coil air-side pressure drop per coil row and the N_r is the number of row.

Assuming two identical coils (Coil#1 and 2) with different coil rows ($N_{r2} < N_{r1}$) and the same air face velocity for the both coils, the following equation can be written by using Eq. (C.1):

$$\frac{\Delta P_{coil2}}{\Delta P_{coil1}} = \frac{N_{r2}}{N_{r1}} \quad \text{(C.2)}$$

The Eq. (C.2) can be written as below by substituting Eq. (21) for ΔP_{coil} and assuming the same face area for both coils:

$$\frac{K_{coil2}}{K_{coil1}} = \frac{N_{r2}}{N_{r1}} \quad \text{(C.3)}$$

Additionally the sensible thermal capacity of a coil is expressed by the following equation [6]:

$$q_{td} = U_o F_s A_f N_r \Delta T_m \quad \text{(C.4)}$$

where U_o is the overall coefficient of sensible heat transfer between airstream and coolant fluid per unit area, A_f is the air face area, ΔT_m is the mean temperature difference between airstream and coolant fluid and F_s is calculated using Eq. (C.5) [6]:

$$F_S = Ao/A_f N_r \quad \text{(C.5)}$$

where Ao is the heat transfer surface area of the coil.

A third coil (Coil#3) which is similar to the Coil#2 and has a larger face velocity is also assumed.

Table C.1 shows details of all assumptions for Coils #1to3.

Table C.1. Parameter assumptions for Coils #1 to 3

Coil	N_r	q_{td}	F_s	***Ao***	A_f	***Uo***	ΔT_m	K_{coils}
Coil #1	N_{r1}	q_{td1}	F_{s1}	A_{o1}	A_{f1}	U_{o1}	ΔT_{m1}	K_{coils1}
Coil #2	$N_{r2} < N_{r1}$	$q_{td2} < q_{td1}$	$F_{s2} < F_{s1}$	$A_{o2} < A_{o1}$	$A_{f2} = A_{f1}$	$U_{o2} = U_{o1}$	$\Delta T_{m2} < \Delta T_{m1}$	$K_{coils2} \neq K_{coils1}$
Coil #3	$N_{r3} = N_{r1}$	$q_{td3} = q_{td1}$	$F_{s3} = F_{s1}$	$A_{o3} = A_{o1}$	$A_{f3} > A_{f1}$	$U_{o3} = U_{o1}$	$\Delta T_{m3} = \Delta T_{m1}$	$K_{coils3} = K_{coils2}$

Considering Table C.1 data and using Eq. (C5) the following equation can be given for Coils #1 & 3:

$$\frac{A_{f1}}{A_{f3}} = \frac{N_{r3}}{N_{r1}} \quad \text{(C.6)}$$

Eq. (C.6) can also be expressed as below based on Table C.1 data:

$$\frac{A_{f1}}{A_{f3}} = \frac{N_{r2}}{N_{r1}} \quad \text{(C.6)}$$

Further Eq. (C.7) can be given by substituting K_{coils3} in Eq. (C.3).

$$\frac{K_{coil3}}{K_{coil1}} = \frac{N_{r2}}{N_{r1}} \quad \text{(C.7)}$$

Also, taking into account the Eqs. (C.6 &7), the equation below can be written:

$$K_{coil3} = K_{coil1}\left(\frac{A_{f1}}{A_{f3}}\right) \quad \text{(C.8)}$$

As descried in Section 2.2, this study examines various AHU's coils with different values of air face velocities while keeping the heating/cooling capacity of coils constant. This case is exactly same as what was assumed for Coil #1 and Coil#3. Hence, having the loss coefficient (K_{coil1}) of the smaller coil (Coil#1) and the air face area of both coils, the loss coefficient of the larger coil (K_{coil3}), can be calculated by the Eq. (C.8). Therefore, this equation is used for the calculation of loss coefficient of various coils in this study.

10 Appendix D- An example of utilizing larger and more efficient outside air louvre

Figure D.1shows the back dimension of an AHU as well as the layout of ducts connected to the back end of the AHU. Note that this AHU is one of AHU's installed in the case study HVAC systems. When the AHU operates on the economy cycle mode the outside air is delivered to the system via a separate duct pathway called ECO O/A. However, in the normal operation of the system the outside air (O/A) is provided from the other branch showed as MIN O/A in Figure D.1. These outside air ducts (ECO O/A and Min O/A ducts) are connected to two outside air (O/A) louvres. The O/A louvre dimensions are same as O/A duct dimensions (1500 mm X 1000 mm and 700 mm X 700 mm). However, instead of two O/A louvres a bigger louver with a size of 2610 mm X 1000 mm can be selected to provide outside air for both of these branches. This is suggested to reduce the average air velocity through the louvre which causes smaller pressure drop in the louvre. Additionally, Figure D.2 shows the selected louvre for the improved model. Note that a louvre with this blade profile and a free area ratio of 80% leads to smallest value of loss coefficient based on the AIRAH handbook [2].

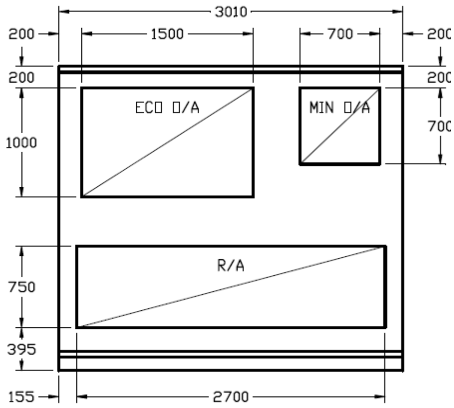
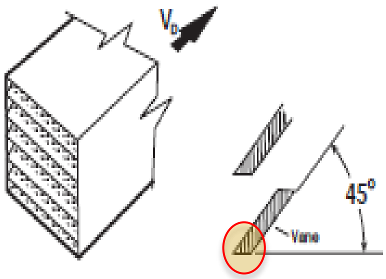


Figure D.1. Back view of one AHU



Free area ratio	0.4	0.5	0.6	0.7	0.8
k_T	6.3	3.6	2.2	1.4	0.9

Figure D.2. Blade shape and free area of louvres selected for the improved model [2]

11 Appendix E-Further detailed outputs for ductwork system sized based on different methods

Table E.1 shows the total pressure drop and maximum fan power consumption of duct systems for a large air handling system (as described in section 2).

Table 5. Total pressure drop and maximum fan power consumption of duct systems sized based on various methods of this study

Method No	Duct sizing method- Description	Total pressure drop of duct system (Pa)	Maximum fan power (kW)
1	Constant pressure gradient method - ($\Delta P/L = 0.2$ Pa/m)	86	3.7
2	Constant pressure gradient method - ($\Delta P/L = 0.4$ Pa/m)	136	5.8
3	Constant pressure gradient method - ($\Delta P/L = 0.6$ Pa/m)	179	7.7
4	Constant pressure gradient method - ($\Delta P/L = 0.8$ Pa/m)	217	9.3
5	Constant pressure gradient method - ($\Delta P/L = 1.0$ Pa/m)	256	11
6	Constant pressure gradient method- BCA compliant design method	430	18.5
7	Combination of reduction velocity method and constant pressure gradient- ($V = 5$ to 6.5 m/s and $\Delta P/L \leq 0.6$ Pa/m)	77	3.3
8	First duct segment sized based on $\Delta P/L = 0.6$ Pa/m and fixed size duct runs for the rest of segments	76	3.2
9	Reduction pressure gradient method- Case study design method - ($\Delta P/L$ between 0.8 to 1 (Pa/m))	169	7.2
10	Reduction pressure gradient method - ($\Delta P/L$ between 0.6 to 1.0 (Pa/m))	156	6.7
11	Reduction pressure gradient method - ($\Delta P/L$ between 0.6 to 0.8 (Pa/m))	144	6.2
12	Reduction pressure gradient method - ($\Delta P/L$ between 0.4 to 0.8 (Pa/m))	121	5.2
13	Reduction pressure gradient method - ($\Delta P/L$ between 0.2 to 0.6 (Pa/m))	117	5
14	Reduction pressure gradient method - ($\Delta P/L$ between 0.4 to 0.6 (Pa/m))	91	3.9
15	Reduction pressure gradient method - ($\Delta P/L$ between 0.2 to 0.4 (Pa/m))	83	3.6

Table E.2 demonstrates further results for increasing of the air face velocities for a large AHU system with the air quantity of 21482 l/s as described in section 2

Table E.2. Further results for increasing air face velocity in a large AHU

Air Face Velocity (m/s)	$\Delta P_{Cooling\ coil}$ (Pa)	$\Delta P_{Heating\ coil}$ (Pa)	ΔP_{Filter} (Pa)	$W_{fan-AHU}$ (kW)
1.15	22.5	1.7	62.0	3.7

Air Face Velocity (m/s)	$\Delta P_{Cooling\ coil}$ (Pa)	$\Delta P_{Heating\ coil}$ (Pa)	ΔP_{Filter} (Pa)	$W_{fan-AHU}$ (kW)
1.2	26.0	2.0	68.4	4.1
1.25	30.1	2.3	75.4	4.6
1.3	34.8	2.7	83.1	5.2
1.35	40.3	3.1	91.6	5.8
1.45	46.7	4.0	101.0	6.5
1.5	54.0	4.2	111.4	7.2
1.6	62.5	4.8	122.8	8.2
1.65	72.4	5.6	135.4	9.2
1.75	83.8	6.4	149.2	10.3
1.85	97.0	7.46	164.5	11.6
1.95	112.3	8.6	181.4	13
2	130	10	200	14.6