NCC 2025 Energy Efficiency - Advice on the technical basis

Initial Measures Development: HVAC Services Report

DELTA Q

ABN: 486 288 97870

Document Control:	This is the WCAG Compliant Version
Version:	3.0
Date:	13/03/2024
Prepared For:	Erica Kenna, Michael Dodd
	The Australian Building Codes Board (ABCB)
DeltaQ Reference:	REP01080-B-004

This report was prepared with input from EnerEfficiency, Northrop Consulting Engineers, Grosvenor Engineering Services and DeltaQ Pty Ltd. WRAP Engineering provided simulation peer review services.

For questions regarding this report, contact:

- Grace Foo, Managing Principal Consultant (sg.foo@dqcs.com.au)
- Dr Paul Bannister, Director of Innovation (paul.bannister@dqcs.com.au)
- Eser Monty, Principal Consultant (eser.monty@dqcs.com.au)

Measures investigated in this report include:

- Chillers
- Unitary AC (PACs and VRF)
- Heat pumps
- 4 pipe chillers
- Dewpoint Coolers and Indirect Evaporative Cooling
- VSD applications for fans and pumps
- VSD applications for cooling tower fans
- Economy cycle
- HVAC fans
- HVAC zoning

[Draft regulation text shown in this report was originally provided to the ABCB for consideration and further development. It may not reflect final provisions for public comment. The draft regulation below also may not reflect any changes following feedback received from the ABCB or various industry stakeholders.]

Table of Contents

1 Introduction

Section J of the National Construction Code (Volume One) is undergoing a cyclic review of both stringency and coverage. This report records the analyses for the initial measures development for NCC 2025 pertaining to Heating Ventilation and Air-Conditioning (HVAC) Services.

1.1 Project Context

Section J of the National Construction Code (Volume One) last underwent a significant review for the 2019 edition. Since then, HVAC technologies have advanced in some areas, creating the opportunity for enhanced stringency. Furthermore, external pressures on Code from factors such as net zero targets at Australian government and state government level have added to ambition.

1.2 Purpose of this report

The purpose of this report is to present initial stringency analyses in relation to HVAC technologies. The following technologies are reviewed within the following contexts:

1. Chillers: NCC 2019/22 chiller efficiency levels were linked to proposed (but never implemented) Greenhouse and Energy Minimum Standards (GEMS) levels for chillers. In the meantime, chiller technologies have advanced incrementally and a new generation of chillers using low Global Warming Potential (GWP) refrigerants has entered the market. In this review, a cost-benefit analysis has been undertaken to define stringency levels that reflect the economic framework for Code rather than that of GEMS.
2. Unitary air-conditioning: As with chillers, the analyses in this report are focussed on defining cost-effective efficiency requirements suitable for supporting high efficiency building regulation. A particular emphasis is on the assessment of variable speed compressor technologies, which can deliver significant efficiency benefits.
3. Heat pumps and 4 pipe chillers: These technologies are not mentioned in NCC 2022 but are expected to become more prevalent as more buildings move to all-electric operation as part of future strategies for net-zero emissions.
4. Dewpoint and evaporative cooling: The code currently makes no provisions for the inclusion of these technologies in buildings. Prior evidence suggests that in some situations these technologies can have a significant impact on energy use. As a result, the cost effectiveness and impact of these technologies is investigated.
5. Variable speed drives: Variable speed drives have become ubiquitous in HVAC design for energy efficient buildings, but this is not reflected in Code. The potential for variable speed drives for fans and pumps to be mandated is investigated.
6. HVAC zoning: Archetypes used for the assessment of Code measures tend to assume that the zoning of HVAC systems is matched to good practice. However, there is little in Code to actually require such good practice. Therefore, this measure investigates the potential to include measures that require good zoning practice.
7. Economy cycle: The cost-effectiveness of a potential increase in stringency of economy cycles (to a wider range of situations and flows) is investigated.
8. HVAC Fans: The focus of analysis for this measure was on finding ways to better express current code requirements.

It is noted that in all cases, the assessment presented in this report are initial analyses only, intended to develop draft measures that can be optimised on a whole-of-building basis in the next phase of work. Thus, it should be noted that the stringencies recommended in this report may not be final.

2 Chillers

2.1 Background and context

In the development of NCC 2019, cost benefit analyses were undertaken with respect to the efficiency of chillers, but the results were not adopted into Code due to perceived conflicts between a proposed GEMS standard for chillers and the Code. As a result, the NCC 2019 chiller measures were based on the requirements of ASHRAE 90.1; the resultant standards were not dissimilar to those proposed from the cost benefit analysis.

A key weakness of the ASHRAE 90.1 approach is that it regulates the efficiency of individual chillers rather than the efficiency of chiller plant. While this is an expedient approach, it has the potential to drive poor outcomes, as the mix of chiller sizes in a multi-chiller plant significantly determines the achieved overall efficiency. Furthermore, in a multi-chiller plant, the Integrated Part Load Value (IPLV) metric used to assess part load performance of individual chillers is a poor indicator of the average operating efficiency of the chiller due to the significant dependence of chiller part load hours and condensing conditions on the configuration of the chiller plant as a whole.

In the Whole-of-HVAC Co-efficient Of Performance (COP) project conducted by DeltaQ for the Australian Government, the concept of a whole of HVAC system COP was tested and found impractical. However, a key finding of the work was that there was potential to develop separate compressor technology-independent efficiency standards for air-cooled and water-cooled chillers. Such standards would be based on the part load profile of the entire chiller plant, rather than of individual chillers, thereby improving the relationship between regulation and outcomes. The implementation of this approach was projected to consist of a matrix of whole-plant "Climate Specific Part Load Value (CSPLV)" figures based on the whole of plant COP at 100%, 75%, 50% and 25% load points with weighting factors and condensing conditions calculated separately for each climate zone.

2.2 Relationship to 4 pipe chillers and heat pumps in cooling mode

The possibility of including 4 pipe chillers and heat pumps in cooling mode as an integral part of the chiller stringency analysis was considered. However, it was found that these technologies have a significantly lower efficiency than the equivalent dedicated-function chillers; furthermore, the limited maturity of these technologies meant that the depth of performance information for these technologies was poor by comparison to chillers. As a result, it was decided to exclude these technologies both from the analysis and from the subsequent provisions. In effect, these are handled under a simple equipment standard under Heat Pumps in Section 4 and, 4 pipe Chillers in Section 5.

2.3 Data measurement standards

Data gathered for this analysis was either explicitly or implicitly assessed under the conditions specified under AHRI551/591, SI edition. The most recent version of this is dated 2023.

2.4 Methodology

The outline methodology is illustrated in Figure 1.

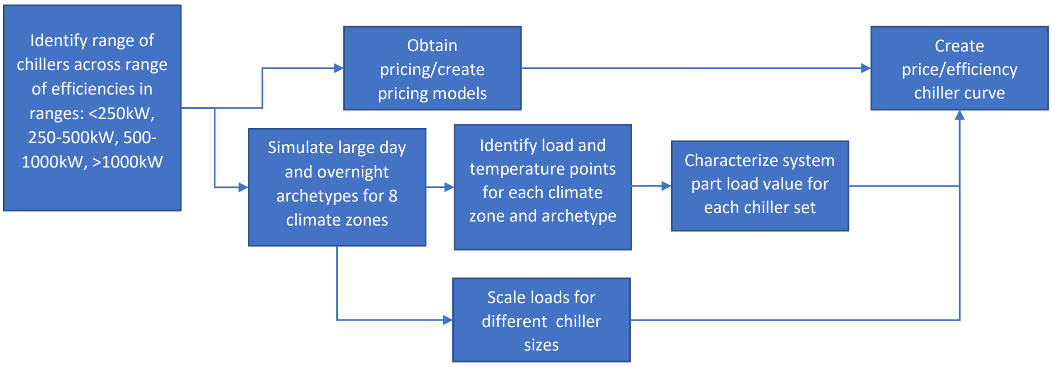


Figure 1: Outline methodology for chiller analysis

2.4.1 Data collection and review – Water-cooled chillers

Chiller capacity, compressor technology, Energy Efficiency Ratio (EER), IPLV, cost and footprint data for water-cooled chillers were collated via contact with multiple chiller manufacturers/suppliers as listed in Table 1.

Table 1. Summary of water-cooled chiller data collected

Chiller type	Number of manufacturers	Number of chiller models
Water-cooled centrifugal	5	29
Water-cooled screw	7	27
Water-cooled scroll	1	1

The spread of capacities, compressor technologies and efficiencies are illustrated in Figure 2. Desensitized chiller data is listed in the Appendices, 12.1.

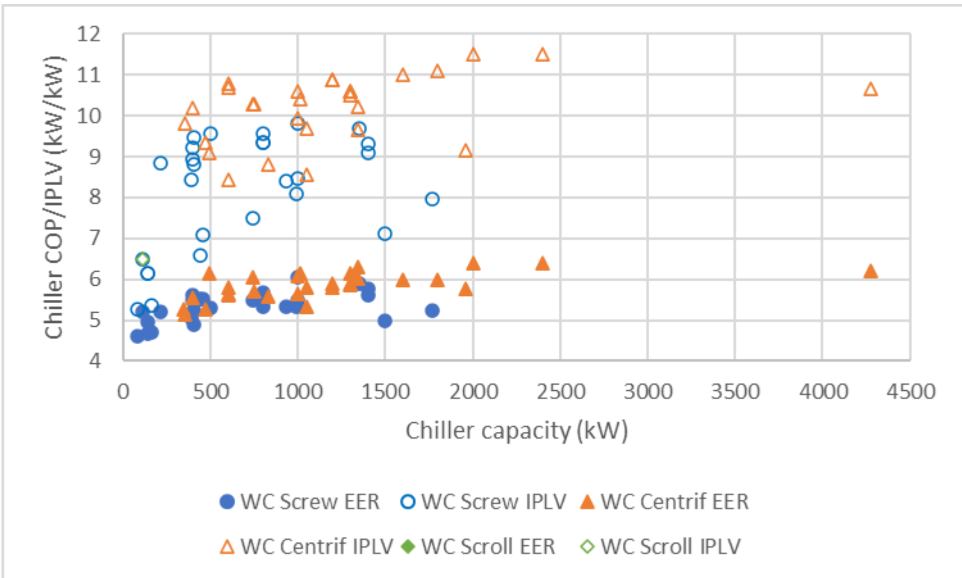


Figure 2. Range of water-cooled chiller efficiencies and capacity data collected for standard rating condition CHW LWT 7°C and condenser entering water EWT 30°C.

It can be seen from Figure 2 that in general, centrifugal[1] chillers have marginally higher EER and significantly higher IPLV figures than the screw chillers, although there is some overlap. It can also be seen that there is a broadly increasing trend in both EER and IPLV with capacity.

The cost of water-cooled chillers was found to be approximately linearly related to capacity, as shown in Figure 3. It can also be seen that centrifugal chillers are generally more expensive than screw chillers. This is reflected in the cost/efficiency relationships shown in Figure 4 and Figure 5. It can be seen that the range of EER figures around the capacity-adjusted average EER is moderate, with most chillers falling within a ±10% band; however, for IPLV the range is significantly wider.

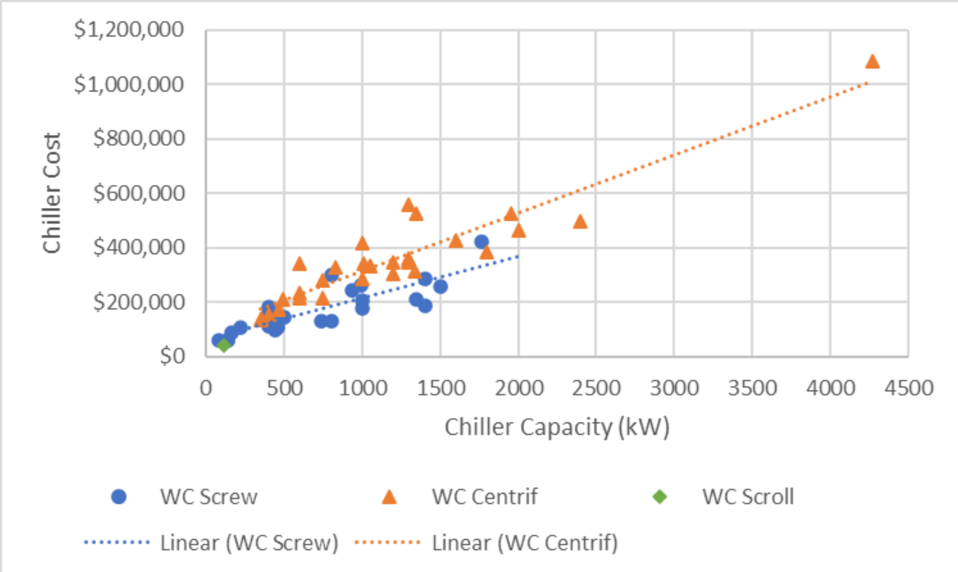


Figure 3. Water-cooled chiller costs

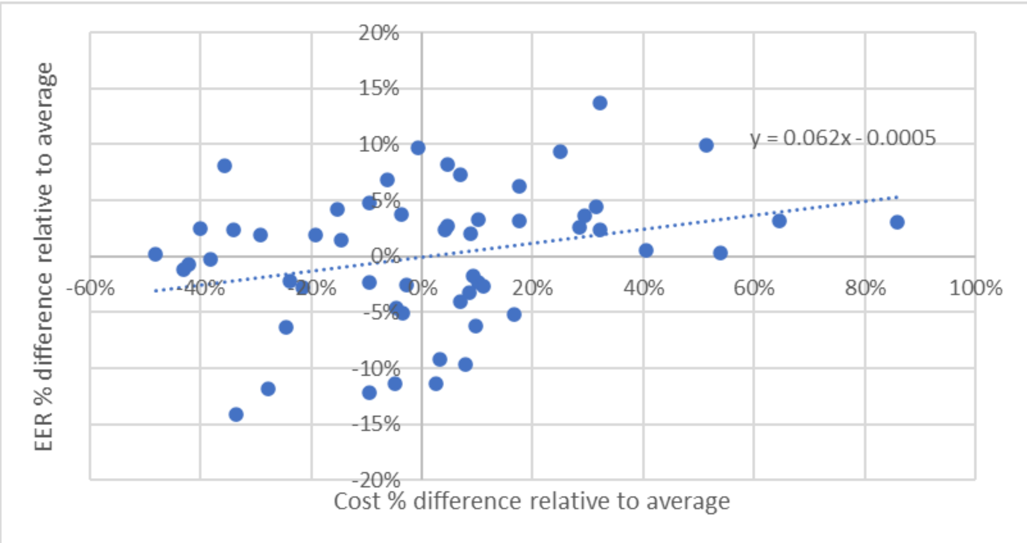


Figure 4. Percentage difference in **EER** versus percentage cost difference for water-cooled chillers; both figures have been corrected for capacity.

[1] Centrifugal chillers for the purposes of this study also included oil-free magnetic bearing chillers.

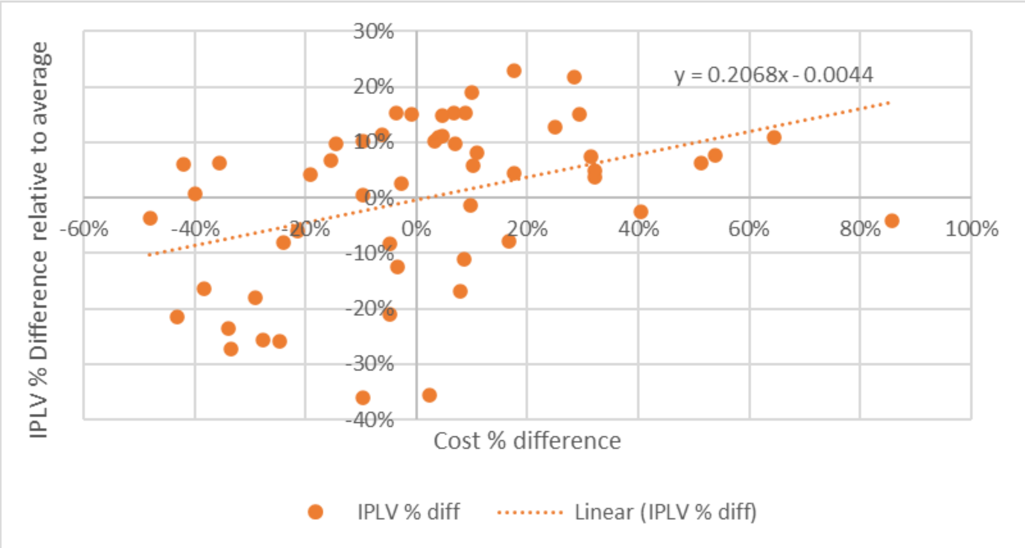


Figure 5. Percentage difference in **IPLV** versus percentage cost difference for water-cooled chillers; both figures have been corrected for capacity.

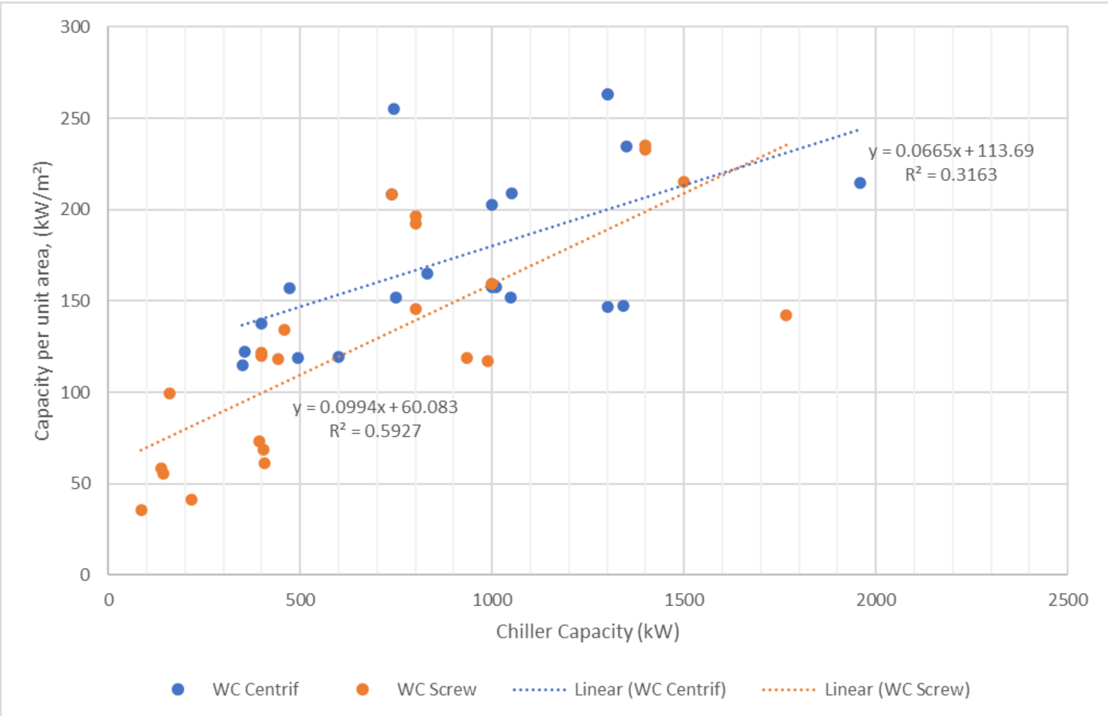


Figure 6: Chiller capacity per unit area kW/m² vs capacity for water-cooled chillers

The spatial allowance for each chiller was assessed based on twice the footprint area of the chiller. Results are shown in Figure 6. Costs associated with the provision of plantroom space were evaluated at \$3800/m² and added to the chiller costs for the purpose of the economic analysis.

2.4.2 Data collection and review – air-cooled chillers

Chiller capacity, compressor technology, EER, IPLV, cost and footprint data for air-cooled chillers were collated via contact with multiple chiller manufacturers/suppliers as listed in Table 2.

Table 2. Summary of air-cooled chiller data collected

Chiller type	Number of manufacturers	Number of chiller models
Air-cooled scroll	4	22
Air-cooled screw	4	31

The spread of capacities, compressor technologies and efficiencies are illustrated in Figure 7. Almost all chillers in the sample are variable speed. Desensitized chiller data is listed in the Appendices, Section 12.1.

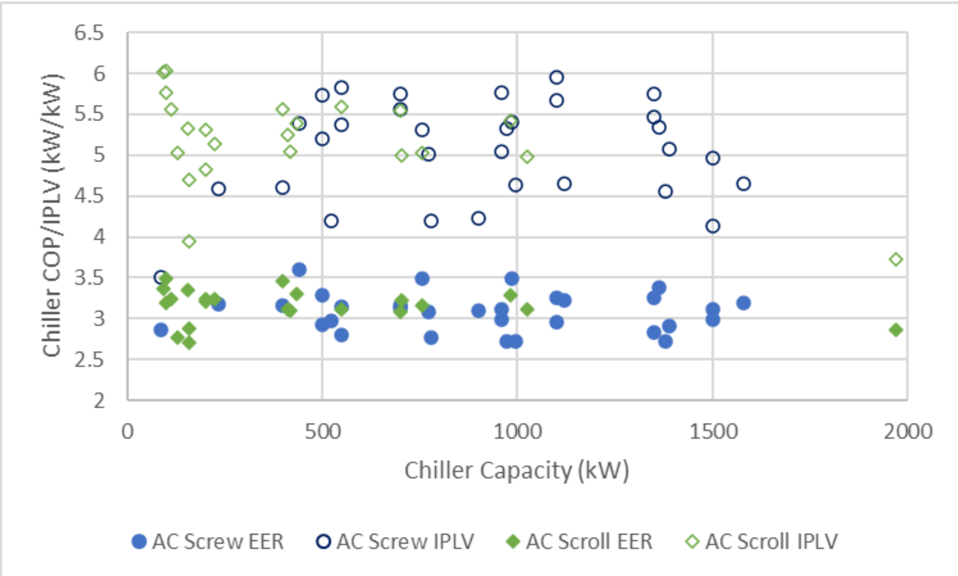


Figure 7. Range of air-cooled chiller efficiency and capacity data collected.

It can be seen from Figure 7 that the EER and IPLV figures for screw and scroll chillers are broadly comparable. There is no significant trend in efficiency with capacity. Screw chillers are available in higher capacities than scroll chillers.

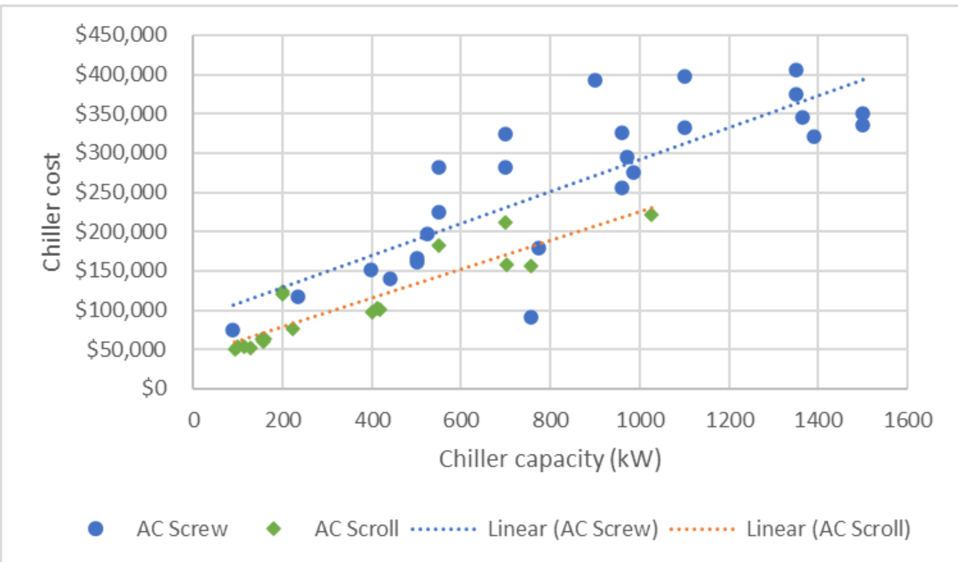


Figure 8. Air-cooled chiller costs.

The cost of air-cooled chillers was found to be linearly related to capacity, as shown in Figure 8. It can also be seen that screw chillers are generally more expensive than scroll chillers.

The relationship between cost and efficiency was weak for both EER and IPLV, as shown in Figure 9 and Figure 10.

Figure 9. Percentage difference in EER versus percentage cost difference for air-cooled chillers; cost figures have been corrected for capacity, average EER was calculated to be 3.114 at all capacities.

Figure 10. Percentage difference in IPLV versus percentage cost difference for air-cooled chillers; cost figures have been corrected for capacity, average IPLV was calculated to be 5.114 at all capacities.

As air-cooled chillers are generally installed outside plant rooms, no allowance for additional spatial costs has been used in the analysis.

2.4.3 Simulation Analysis Methodology

The impacts of chiller efficiency on energy use were calculated as follows for two archetypes, being the Large Office (C5OL) and Large Hospital (C9A).

1. Four EER and IPLV combinations were selected for each chiller type, bounding the majority of the available EER and IPLV for that chiller type. The bounding points were as listed in Table 3.

Table 3. EER/IPLV combinations used to characterise each chiller technology.

Chiller Type	$(EER, IPLV)_1$	$(EER, IPLV)_2$	$(EER, IPLV)_3$	$(EER, IPLV)_4$
Water-cooled (WC) Centrifugal	(5.1, 9.8)	(5.6, 8.4)	(6.2, 9.5)	(6.4, 11.5)
WC Screw	(4.5, 6)	(4.9, 9.5)	(5.5, 6.5)	(6.1, 10)
Air-cooled (AC) Scroll	(2.7, 5)	(2.9, 4.6)	(3.4, 6)	(3.5, 6.5)
AC Screw	(2.7, 5.4)	(3.1, 4.5)	(3.3, 6.6)	(3.6, 5.4)

2. Typical part load curves were identified for each chiller type. For all types other than AC Screw, these were based on data obtained from manufacturers, averaged across multiple models. Basic part load curves used are listed in Appendix 12.
3. The typical part load curves were manipulated to match the EER/IPLV combinations listed in Table 3.
4. The simulation was undertaken for each climate zone using 2 identical chillers sized at 60% of building peak load for that climate zone.
5. Simulation energy results were fitted to an averaged 2 dimensional plane of equation $E/C = aEER + bIPLV$, separately for each archetypes, climate zone and chiller type where E is the annual energy use in kWh and C is the chiller thermal capacity in kW. Similar equations were derived for peak electrical demand.
6. For each chiller, the derived equations were used to calculate a scaled energy consumption and peak demand.
7. Chiller capital cost plus plant rooms space allowance costs were combined to produce a total technology cost.
8. Capital cost figures were used to determine the lowest cost (NCC 2022 J6D11 Option 1 or Option 2) compliant chiller in each capacity range for compliance (see Table 4 and Table 5). The cost/efficiency relationships determined from the chiller data sets were used to determine a cost reflective of the average cost for a minimally compliant chiller.
9. NPV results were calculated for each actual chiller (in each climate zone and for each archetype) versus the baseline chiller of the same size (see Section 2.5.1), providing a benefit cost ratio.
10. For each climate zone and archetype, either:
    a. If there were no chillers with a Benefit Cost Ratio (BCR)>1, the base case chiller was used to provide the EER and IPLV thresholds.
    b. If there were multiple chillers with a BCR>1 then the average EER and IPLV of the chiller compressor technology with the higher frequency of occurrence in the BCR>1 data set was calculated as the new threshold. If both chiller compressor technologies had the same frequency of occurrence, the chiller technology with the lower energy use was selected.
    c. If the resultant EER or IPLV resulted in a situation where only one or two chillers had BCR>1 chillers and these were also the highest efficiency chillers in the dataset, the resultant EER and IPLV was recalculated as the average of the BCR>1 chillers and the base case chiller, in order to ensure that there was some choice of chillers available to achieve compliance.

Graphs illustrating the results of this step are presented in the Appendices, Section 12.3.

11. The new EER and IPLV thresholds were collated by archetype and annual cooling load and checked for inconsistencies (such as a lack of consistent trend for chiller selections with cooling load). Inconsistencies were corrected using the principle that if a particular selection was identified at two different cooling loads, then that selection should also apply to intermediate cooling loads[2]. Graphs illustrating the results of this step are presented in the Appendices, Section 12.4.
12. Notional EER and IPLV thresholds, with accompanying chiller technologies, were finalised for each archetype and climate zones
13. For each archetype and climate zone, the hourly cooling load data from the simulation was interrogated to identify the total thermal load, average dry bulb temperature, average wet bulb temperature and average chilled water temperatures in the following bands relative to the design cooling load: 0-37.5%, 37.5%-62.5%, 62.5%-87.5%,87.5%-100% (corresponding to nominal 25%, 50%, 75% and 100% load points).
14. Chiller part load curves were adapted to match the nominal EER and IPLV thresholds to calculate the CSPLV as follows:

$$\begin{aligned} CSPLV = {} & \alpha_{25} EER_{25}(T_{db,25}, T_{wb,25}, T_{ChW,25}) + \alpha_{50} EER_{50}(T_{db,50}, T_{wb,50}, T_{ChW,50}) \\ & + \alpha_{75} EER_{75}(T_{db,75}, T_{wb,75}, T_{ChW,75}) \\ & + \alpha_{100} EER_{100}(T_{db,100}, T_{wb,100}, T_{ChW,100}) \end{aligned}$$

Where: α_N is a weighting coefficient based on annual loading in load range N and EER_N is the chiller plant EER (including chillers only, no pumps or cooling towers) at load point N calculated and ambient dry bulb $T_{db,N}$, ambient wet bulb $T_{wb,N}$ and chilled water supply temperature $T_{ChW,N}$ which are representative of the conditions in load range N.

2.5 Results

2.5.1 Baseline chillers

The baseline chillers were selected as the cheapest NCC 2022 compliant chillers in their capacity range, separately for air-cooled and water-cooled chillers (but disregarding compressor technology within these categories).

Table 4. Baseline air-cooled chiller selections. NCC 2022 figures refer to the option 1 or option 2 compliance path EER and IPLV figures met by the selected chiller.

Range min	Range max	EER	IPLV	Type	NCC 2022 EER	NCC 2022 IPLV	NCC 2022 option
0	528	2.885	4.700	AC Scroll	2.866	4.669	2 only
528	>528	3.167	5.030	AC Scroll	2.866	4.758	2 only

[2] The situations where such inconsistencies occur were ascribed to the loss of accuracy in energy values arising from the use of correlated chiller energy estimates.

Table 5. Baseline water cooled chiller selections. NCC 2022 figures refer to the option 1 or option 2 compliance path EER and IPLV figures met by the selected chiller.

Range min	Range max	EER	IPLV	Type	NCC 2022 EER	NCC 2022 IPLV	NCC 2022 option
0	264	4.95	6.14	WC Screw	4.694	5.867	1 only
264	528	5.526	6.59	WC Screw	4.889	6.286	1 only
528	1055	5.5	7.48	WC Screw	5.334	6.519	1 only
1055	1407	5.773	9.322	WC Screw	5.633	8.586	2 only
1407	>1407	6.4	11.5	WC Centrifugal	6.018	9.264	both[3]

It can be seen that the baseline chillers generally outperform the NCC 2022 figures by a significant margin.

2.5.2 Notional threshold chillers

The notional threshold chillers based on benefit cost analysis are listed in Table 6 to Table 9. Note that these are effectively chiller selections in the context of a design load that is twice the chiller capacity divided by 1.2, i.e. based on an assumption of two 60% chillers serving the load.

Table 6. Notional EER and IPLV values - air-cooled chillers, daytime archetype

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
1	0-528kW	3.26	5.40	AC Scroll
1	>528kW	3.37	5.22	AC Screw
2	0-528kW	3.26	5.40	AC Scroll
2	>528kW	3.20	5.02	AC Scroll
3	0-528kW	3.26	5.40	AC Scroll
3	>528kW	3.23	5.00	AC Scroll
4	0-528kW	3.26	5.40	AC Scroll
4	>528kW	3.20	5.02	AC Scroll
5	0-528kW	3.26	5.40	AC Scroll
5	>528kW	3.37	5.22	AC Screw
6	0-528kW	3.26	5.40	AC Scroll
6	>528kW	3.37	5.22	AC Screw
7	0-528kW	3.26	5.40	AC Scroll
7	>528kW	3.37	5.22	AC Screw
8	0-528kW	3.26	5.40	AC Scroll
8	>528kW	3.37	5.22	AC Screw

Table 7. Notional EER and IPLV values - air-cooled chillers, overnight archetype

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
1	0-528kW	3.25	5.42	AC Scroll
1	>528kW	3.39	5.25	AC Screw
2	0-528kW	3.25	5.42	AC Scroll
2	>528kW	3.39	5.25	AC Screw
3	0-528kW	3.25	5.42	AC Scroll
3	>528kW	3.39	5.25	AC Screw

[3] Option 2 EER and IPLV listed.

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
4	0-528kW	3.25	5.42	AC Scroll
4	>528kW	3.39	5.25	AC Screw
5	0-528kW	3.25	5.42	AC Scroll
5	>528kW	3.39	5.25	AC Screw
6	0-528kW	3.25	5.42	AC Scroll
6	>528kW	3.39	5.25	AC Screw
7	0-528kW	3.25	5.42	AC Scroll
7	>528kW	3.39	5.25	AC Screw
8	0-528kW	3.2	5.42	AC Scroll
8	>528kW	3.31	5.25	AC Screw

Table 8. Notional EER and IPLV values - water cooled chillers, daytime archetype.

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
1	0-264	4.95	6.14	WC Screw
1	264-528	5.69	8.73	WC Centrif
1	528-1055	5.99	10.42	WC Centrif
1	1055-1407	6.01	10.30	WC Centrif
1	>1407	6.40	11.5	WC Centrif
2	0-264	4.95	6.14	WC Screw
2	264-528	5.69	8.73	WC Centrif
2	528-1055	5.99	10.42	WC Centrif
2	1055-1407	6.01	10.30	WC Centrif
2	>1407	6.4	11.5	WC Centrif
3	0-264	4.95	6.14	WC Screw
3	264-528	5.69	8.73	WC Centrif
3	528-1055	5.99	10.42	WC Centrif
3	1055-1407	6.01	10.30	WC Centrif
3	>1407	6.4	11.5	WC Centrif
4	0-264	4.95	6.14	WC Screw
4	264-528	5.49	7.88	WC Screw
4	528-1055	5.63	9.57	WC Screw
4	1055-1407	5.77	9.32	WC Screw
4	>1407	6.4	11.5	WC Centrif
5	0-264	4.95	6.14	WC Screw
5	264-528	5.49	7.88	WC Screw
5	528-1055	5.63	9.57	WC Screw
5	1055-1407	5.77	9.32	WC Screw
5	>1407	6.40	11.50	WC Centrif
6	0-264	4.95	6.14	WC Screw
6	264-528	5.49	7.88	WC Screw
6	528-1055	5.63	9.57	WC Screw
6	1055-1407	5.77	9.32	WC Screw
6	>1407	6.40	11.50	WC Centrif
7	0-264	4.95	6.14	WC Screw
7	264-528	5.49	7.88	WC Screw

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
7	528-1055	5.63	9.57	WC Screw
7	1055-1407	5.77	9.32	WC Screw
7	>1407	6.40	11.50	WC Centrif
8	0-264	4.95	6.14	WC Screw
8	264-528	5.49	7.88	WC Screw
8	528-1055	5.63	9.57	WC Screw
8	1055-1407	5.77	9.32	WC Screw
8	>1407	6.40	11.50	WC Centrif

Table 9. Notional EER and IPLV values - water cooled chillers, overnight archetype.

Climate Zone	Capacity range (kW_{th})	EER	IPLV	Type
1,2	0-264	4.95	6.31	WC Screw
1,2	264-528	5.84	7.84	WC Centrif
1,2	528-1055	5.94	10.38	WC Centrif
1,2	1055-1407	6.00	10.46	WC Centrif
1,2	>1407	6.40	11.50	WC Centrif
2	0-264	4.95	6.31	WC Screw
2	264-528	5.84	7.84	WC Centrif
2	528-1055	5.94	10.38	WC Centrif
2	1055-1407	6.00	10.46	WC Centrif
2	>1407	6.40	11.50	WC Centrif
3	0-264	4.95	6.31	WC Screw
3	264-528	5.41	8.60	WC Screw
3	528-1055	5.94	10.38	WC Centrif
3	1055-1407	5.89	9.68	WC Screw
3	>1407	6.40	11.50	WC Centrif
4	0-264	4.95	6.31	WC Screw
4	264-528	5.41	8.60	WC Screw
4	528-1055	5.62	9.51	WC Screw
4	1055-1407	5.77	9.32	WC Screw
4	>1407	6.40	11.50	WC Centrif
5	0-264	4.95	6.31	WC Screw
5	264-528	5.41	8.60	WC Screw
5	528-1055	5.94	10.38	WC Centrif
5	1055-1407	5.77	9.32	WC Screw
5	>1407	6.40	11.50	WC Centrif
6	0-264	4.95	6.31	WC Screw
6	264-528	5.41	8.60	WC Screw
6	528-1055	5.62	9.51	WC Screw
6	1055-1407	5.77	9.32	WC Screw
6	>1407	6.40	11.50	WC Centrif
7	0-264	4.95	6.31	WC Screw
7	264-528	5.41	8.60	WC Screw
7	528-1055	5.62	9.51	WC Screw
7	1055-1407	5.77	9.32	WC Screw
7	>1407	6.40	11.50	WC Centrif
8	0-264	4.95	6.31	WC Screw
8	264-528	5.41	8.60	WC Screw
8	528-1055	5.62	9.51	WC Screw
8	1055-1407	5.77	9.32	WC Screw
8	>1407	6.40	11.50	WC Centrif

2.5.3 Part load evaluation points

Daytime archetype

The following tables list the part load evaluation points for the daytime archetype for each climate zone (CZ).

Table 10. Part load evaluation points - outdoor dry bulb and wet bulb temperature (°C), day time archetype

Load	CZ 1 DB	CZ 1 WB	CZ 2 DB	CZ 2 WB	CZ 3 DB	CZ 3 WB	CZ 4 DB	CZ 4 WB	CZ 5 DB	CZ 5 WB	CZ 6 DB	CZ 6 WB	CZ 7 DB	CZ 7 WB	CZ 8 DB	CZ 8 WB
25%	27	17	21	16	21	12	18	12	19	14	17	12	17	12	14	10
50%	31	24	27	22	32	17	27	17	25	20	26	18	25	17	20	13
75%	32	27	29	25	34	21	31	20	28	23	31	20	29	19	23	15
100%	29	28	29	27	28	23	32	22	27	25	36	24	32	21	25	14

Table 11. Part load evaluation points – chilled water temperature, day time archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	9	10	10	10	10	10	10	10
50%	7	8	9	9	8	9	9	10
75%	6	6	8	8	7	8	9	10
100%	6	6	7	7	6	6	8	10

Table 12. Load weighting factors, day time archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	7%	25%	23%	25%	24%	42%	28%	20%
50%	49%	44%	49%	35%	46%	31%	36%	27%
75%	43%	31%	28%	37%	30%	26%	34%	42%
100%	2%	1%	1%	3%	0%	1%	2%	11%

Overnight archetype

The following tables list the part load evaluation points for the overnight archetype.

Table 13. Part load evaluation points - outdoor dry bulb and wet bulb temperature (°C), overnight archetype

Load	CZ 1 DB	CZ 1 WB	CZ 2 DB	CZ 2 WB	CZ 3 DB	CZ 3 WB	CZ 4 DB	CZ 4 WB	CZ 5 DB	CZ 5 WB	CZ 6 DB	CZ 6 WB	CZ 7 DB	CZ 7 WB	CZ 8 DB	CZ 8 WB
25%	24	19	22	18	21	12	20	14	21	17	19	14	18	14	14	10
50%	29	24	27	22	31	17	29	18	26	21	27	19	26	17	17	13
75%	31	27	29	25	36	20	34	20	29	23	33	21	31	19	22	14
100%	34	29	32	27	39	24	41	23	32	26	41	23	36	20	27	17

Table 14. Part load evaluation points – chilled water temperature, overnight archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	8	9	10	10	9	10	10	10
50%	7	7	9	9	8	9	9	10
75%	6	6	8	8	7	8	9	10
100%	6	6	6	7	6	7	8	10

Table 15. Load weighting factors, overnight archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	7%	31%	23%	43%	41%	59%	40%	16%
50%	31%	40%	44%	39%	35%	24%	31%	32%
75%	61%	28%	32%	18%	23%	15%	26%	49%
100%	1%	2%	1%	0%	1%	1%	3%	3%

2.6 Calculated Climate Specific Part Load Value

Table 16. Climate Specific Part Load Values - Air-cooled chillers

Climate Zone	Design load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
1	0-880kW	4.03	4.08
1	>880kW	3.76	3.9
2	0-880kW	4.93	4.98
2	>880kW	4.56	4.66
3	0-880kW	4.61	4.58
3	>880kW	4.39	4.22
4	0-880kW	5.17	5.35
4	>880kW	4.73	4.92
5	0-880kW	5.26	5.36
5	>880kW	4.94	4.99
6	0-880kW	5.73	5.87
6	>880kW	5.33	5.39
7	0-880kW	5.54	5.6
7	>880kW	5.21	5.21
8	0-880kW	6.05	6.35
8	>880kW	5.88	6.16

Table 17. Climate Specific Part Load Values - Water-cooled chillers

Climate Zone	Design Load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
1	0-440	5.22	5.14
1	440-880	5.95	6.03
1	880-1760	6.29	6.06
1	1760-2345	6.31	6.13
1	>2345	6.73	6.53
2	0-440	5.79	5.82
2	440-880	7.31	6.88
2	880-1760	8.29	7.96
2	1760-2345	8.24	8.03
2	>2345	9.04	8.69
3	0-440	6.27	6.48
3	440-880	8.91	9.5

Climate Zone	Design Load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
3	880-1760	11.14	10.99
3	1760-2345	10.93	10.88
3	>2345	12.5	12.31
4	0-440	6.26	6.64
4	440-880	8.44	9.93
4	880-1760	11.27	11.35
4	1760-2345	10.59	10.85
4	>2345	12.5	12.54
5	0-440	5.99	6.13
5	440-880	7.5	7.94
5	880-1760	9	8.93
5	1760-2345	8.78	8.58
5	>2345	10.21	9.83
6	0-440	6.36	6.68
6	440-880	8.76	10.14
6	880-1760	12.19	11.66
6	1760-2345	11.3	11.1
6	>2345	12.77	12.39
7	0-440	6.34	6.63
7	440-880	8.72	9.99
7	880-1760	12.01	11.45
7	1760-2345	11.17	10.93
7	>2345	13.1	12.75
8	0-440	6.54	6.98
8	440-880	9.86	12.61
8	880-1760	15.96	15.55
8	1760-2345	13.97	14.02
8	>2345	16.72	17.16

2.7 Discussion

2.7.1 Benefits of the proposed approach

The proposed approach is considerably more computationally intensive than the current EER/IPLV based requirement. It is incumbent therefore that there must be adequate benefits to justify this additional complication.

The key benefits of this approach are as follows:

1. Chiller selections are more specific to application. The NCC 2022 chiller provisions do not differentiate climate zones or daytime/overnight applications. This “one-size-fits-all” approach does not reflect the very substantial diversity of loads across these different situations. The proposed approach provides a first-order adjustment to key differences in application and decouples from the limited ability of the IPLV calculation to represent realistic chiller operating conditions.

2. Efficiency is related to the overall chiller plant rather than individual chillers. In practice, most chiller plant have multiple chillers which can be sized and selected to provide optimal efficiency in response the load. The proposed approach encourages designers to consider the selection of chiller plant in this more holistic manner, including consideration of how chillers will be staged.
3. Sizing issues are addressed. The NCC 2022 approach is blind to sizing, which can lead to significantly oversized chillers operating at low efficiencies. The proposed approach is linked to performance of the whole chiller plant in response to the *design load* as opposed to the total chiller capacity; as a result, the designers are forced to consider the impact of their sizing decisions.
4. Improving chiller plant design. The three issues above all together force designers to consider efficiency as part of the overall chiller plant design. This has direct design benefits and may furthermore improve the configuration of chiller staging controls[4].
5. Decoupling from GEMS. Although GEMS for chillers has not been adopted, it was a significant influence on the stringency set in NCC 2019 and continued in NCC 2022. The approach adopted is based on requirements for chiller plant design as opposed to the regulation of individual chillers. As a result, the measure is decoupled from any minimum efficiency standards, although these will continue to set the floor for equipment selection.

2.7.2 Risks of the approach

The viability of the measure in its proposed format depends on two critical factors:

1. The availability of chiller part load data. Our data collection for this project indicated that an all bar one chiller manufacturer was able to provide chiller part load data to meet the requirements of this measure. The exception refused to provide data for commercial reasons rather than out of inability, as far as we can determine.
2. The ability of designers to process the information. Our experience is that designers generally avoid the sorts of questions being addressed by the structure of this measures, so there is no doubt that it will be additional work. The nature of the calculation however is not particularly challenging once the principles are understood.

2.7.3 Stringency 3

The stringencies arrived at via the process described are close to the top end of the range of efficiencies that can be identified without excluding significant portions of the market, as can be seen from the graphs in the Appendices (Section 12.3). Limited improvements are possible in some categories by reducing the BCR limit to a lower figure than 1, but these are heavily constrained by market considerations.

2.8 Proposed Measures

J6D11 Refrigerant chillers

---

[4] Variable speed chillers typically have an optimum stage up point at less than 100% load, while fixed speed chillers generally are optimally staged at 100% load. Unfortunately, it is not uncommon to find variable speed chillers programmed to stage up at 100%, losing a potential efficiency benefit. As designers will be forced to think about staging to undertake the CSPLV calculation, it is possible they may also think about optimising staging to maximise efficiency.

For each connected group of air-conditioning system refrigerant in a building, the Climate Specific Part Load Value (CSPLV) as calculated in Specification NN must -

(a) Be greater than or equal to the values in Table J6D11a for a connected group of water-cooled chillers; or
(b) Be greater than or equal to the values in Table J6D11b for a connected group of air-cooled chillers; or
(c) Be greater than the capacity weighted average A of the values in Table J6D11a and Table J6D11b for a connected group with or air-cooled and water-cooled chillers, calculated as:

$$A = \frac{CSPLV_{AC}C_{AC} + CSPLV_{WC}C_{WC}}{C_{AC} + C_{WC}}$$

Where:

a. where C_{WC} is the full load cooling capacity of water-cooled chillers and C_{AC} is the total full load capacity of all chillers in the group; and
b. $CSPLV_{WC}$ is the value in Table J6D11a for the nominal water-cooled design cooling load $D_{WC} = \frac{C_{WC}}{C_{WC}+C_{AC}}D$ where D is the design cooling load for the total plant; and
c. $CSPLV_{AC}$ is the value in Table J6D11b for nominal air-cooled design cooling load $D_{AC} = \frac{C_{AC}}{C_{WC}+C_{AC}}D$ where D is the design cooling load for the total plant

Table J6D11a. Climate Specific Part Load Values (CSPLV) for water-cooled chiller plant

Climate zone	Design Load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
1	0-440	4.82	4.76
1	440-880	4.87	5.34
1	880-1760	4.88	4.84
1	1760-2345	4.93	4.89
1	>2345	5.16	5.16
2	0-440	5.38	5.38
2	440-880	6.07	6.2
2	880-1760	6.45	6.4
2	1760-2345	6.46	6.47
2	>2345	6.91	6.91
3	0-440	5.2	5.19
3	440-880	5.57	5.69
3	880-1760	5.87	5.82
3	1760-2345	5.89	6.21
3	>2345	6.28	6.28
4	0-440	5.48	5.52
4	440-880	6.36	6.5
4	880-1760	6.95	6.92
4	1760-2345	6.98	6.98
4	>2345	7.52	7.52
5	0-440	5.54	5.58
5	440-880	6.4	6.5
5	880-1760	6.92	7.07
5	1760-2345	6.98	6.98

Climate zone	Design Load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
5	>2345	7.68	7.68
6	0-440	5.65	5.74
6	440-880	6.87	7.26
6	880-1760	7.93	7.89
6	1760-2345	7.83	7.83
6	>2345	8.25	8.25
7	0-440	5.63	5.71
7	440-880	6.74	7.04
7	880-1760	7.63	7.59
7	1760-2345	7.58	7.58
7	>2345	8.27	8.27
8	0-440	5.8	5.95
8	440-880	7.38	8.04
8	880-1760	8.98	8.92
8	1760-2345	8.72	8.72
8	>2345	10.02	10.02

Table J6D11b. Climate Specific Part Load Values (CSPLV) for air-cooled chiller plant

Climate zone	Design load range (kW_{th})	CSPLV – daytime archetypes	CSPLV – overnight archetypes
1	0-880kW	4.03	4.08
1	>880kW	3.76	3.9
2	0-880kW	4.93	4.98
2	>880kW	4.56	4.66
3	0-880kW	4.61	4.58
3	>880kW	4.39	4.22
4	0-880kW	5.17	5.35
4	>880kW	4.73	4.92
5	0-880kW	5.26	5.36
5	>880kW	4.94	4.99
6	0-880kW	5.73	5.87
6	>880kW	5.33	5.39
7	0-880kW	5.54	5.6
7	>880kW	5.21	5.21
8	0-880kW	6.05	6.35
8	>880kW	5.88	6.16

Specification NN

The Climate Adjusted Part Load Value X for a group of chillers shall be calculated as:

$$X = \alpha_{100}\mathrm{EER}_{100} + \alpha_{75}\mathrm{EER}_{75} + \alpha_{50}\mathrm{EER}_{50} + \alpha_{25}\mathrm{EER}_{25}$$

Where

a) The values of the coefficients α_n are determined from Table NN3 for Class 2 common area, class 5,6,7,8 9b, 9a other than a ward area and Table NN6 for Class 3, 9c or 9a ward area.
b) The values of EER_n are determined as the average EER of the chillers operating to meet n% of the design load, allowing for the part load efficiencies of the affected chillers as well as:
    a. The chilled water temperatures as listed in Table NN2 for Class 2 common area, class 5,6,7,8 9b, 9a other than a ward area and Table NN5 for Class 3, 9c or 9a ward area.
    b. The outside wet bulb conditions for water cooled chillers, as listed in Table NN1 for Class 2 common area, class 5,6,7,8 9b, 9a other than a ward area and Table NN4 for Class 3, 9c or 9a ward area, modified by the cooling tower design approach temperature to obtain the entering condenser water temperature
    c. The outside dry bulb temperatures for air-cooled chillers, as listed in Table NN1 for Class 2 common area, class 5,6,7,8 9b, 9a other than a ward area and Table NN4 for Class 3, 9c or 9a ward area.

Table NN1. Part load evaluation points - outdoor dry bulb and wet bulb temperature (°C), day time archetype

	CZ 1	CZ 1	CZ 2	CZ 2	CZ 3	CZ 3	CZ 4	CZ 4	CZ 5	CZ 5	CZ 6	CZ 6	CZ 7	CZ 7	CZ 8	CZ 8
Load	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
25%	27	17	21	16	21	12	18	12	19	14	17	12	17	12	14	10
50%	31	24	27	22	32	17	27	17	25	20	26	18	25	17	20	13
75%	32	27	29	25	34	21	31	20	28	23	31	20	29	19	23	15
100%	29	28	29	27	28	23	32	22	27	25	36	24	32	21	25	14

Table NN2. Part load evaluation points – chilled water temperature, day time archetype.

Load	*CZ1*	*CZ2*	*CZ3*	*CZ4*	*CZ5*	*CZ6*	*CZ7*	*CZ8*
25%	*9*	*10*	*10*	*10*	*10*	*10*	*10*	*10*
50%	*7*	*8*	*9*	*9*	*8*	*9*	*9*	*10*
75%	*6*	*6*	*8*	*8*	*7*	*8*	*9*	*10*
100%	*6*	*6*	*7*	*7*	*6*	*6*	*8*	*10*

Table NN3. Load weighting factors, day time archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	7%	25%	23%	25%	24%	42%	28%	20%
50%	49%	44%	49%	35%	46%	31%	36%	27%
75%	43%	31%	28%	37%	30%	26%	34%	42%
100%	2%	1%	1%	3%	0%	1%	2%	11%

Table NN4. Part load evaluation points - outdoor dry bulb and wet bulb temperature (°C), overnight archetype

	CZ1	CZ1	CZ2	CZ2	CZ3	CZ3	CZ4	CZ4	CZ5	CZ5	CZ6	CZ6	CZ7	CZ7	CZ8	CZ 8
Load	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
25%	24	19	22	18	21	12	20	14	21	17	19	14	18	14	14	10
50%	29	24	27	22	31	17	29	18	26	21	27	19	26	17	17	13
75%	31	27	29	25	36	20	34	20	29	23	33	21	31	19	22	14
100%	34	29	32	27	39	24	41	23	32	26	41	23	36	20	27	17

Table NN5. Part load evaluation points – chilled water temperature, overnight archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	8	9	10	10	9	10	10	10
50%	7	7	9	9	8	9	9	10
75%	6	6	8	8	7	8	9	10
100%	6	6	6	7	6	7	8	10

Table NN6. Load weighting factors, overnight archetype.

Load	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
25%	7%	31%	23%	43%	41%	59%	40%	16%
50%	31%	40%	44%	39%	35%	24%	31%	32%
75%	61%	28%	32%	18%	23%	15%	26%	49%
100%	1%	2%	1%	0%	1%	1%	3%	3%

3 Unitary air-conditioning

3.1 Background and context

Unitary air-conditioning systems (comprising packaged air-conditioners (PACs) and variable refrigerant flow (VRF) systems) form a substantial part of the total HVAC industry and in particular dominate the small to medium building sectors. As such they have a significant impact on the built environment as a whole.

NCC 2019 provides a simple metric for air-cooled unitary systems of an EER of 2.9 or higher (J5.11 (b)), complementary to the relevant MEP requirements for unitary AC[5]. This was a moderate increase in stringency on NCC2016 which was informed by a cost benefit analysis. However, in the past 5 years, variable speed compressor controls have become considerably more prevalent in the market, offering significant efficiency benefits at part load operation relative to fixed speed compressors. Furthermore, while PACs are a substantial market, VRF systems also operate in the same markets and need to be considered in any new analysis.

A further area of interest for possible update in NCC 2025 is the adjustment of efficiency requirements with duty, particularly between climate zones, daytime versus 24/7 operation and with/without economy cycle operation.

3.1.1 Part Load indicators

Australia has used the full load EER as its primary indicator of unitary AC efficiency for some time. Unfortunately, this has meant that equipment suppliers do not hold data for part load indicators. As a major focus of the analysis undertaken for this measure examines the impact of measures that affect part load performance, this is a major impediment. This situation is expected to change over the next few years as reporting of Seasonal EER (SEER) became mandatory under the GEMS (Air Conditioners under 65kW) Determination 2019 and GEMS (Air Conditioners above 65kW) Determination 2022.

As a result, we have been forced to assess and propose a measure for NCC 2025 that uses a combination of EER and technology labels to obtain a viable regulation with the information available. This is a significantly inferior result to what could be achieved if all equipment suppliers published a SEER figure.

3.1.2 Analysis Scope

The analysis scope covered air-cooled PAC units and VRF systems only. No analysis was undertaken in relation to water cooled systems.

3.2 Methodology

The outline methodology is shown in Figure 11. Note that separate analyses were undertaken for PAC units and VRF systems, with the VRF systems only analysed without economy cycle.

[5] Relevant to the ducted units considered in the current analysis, GEMS has a minimum EER requirement of 3.1 below 39kW capacity and 2.9 from 39kW and above.

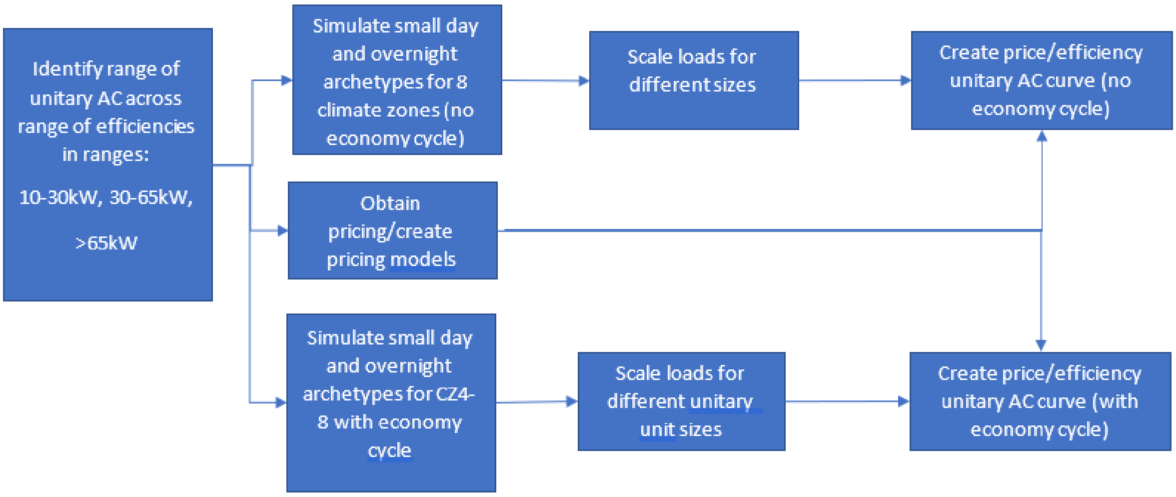


Figure 11. Methodology outline for the unitary AC analysis

3.2.1 Data collection and review – PAC unit

Data was gathered for 76 PAC units across the range 8-203kW cooling capacity, although data above 100kW was relatively sparse (reflecting the market). However, of this data, 22 units were found to not be GEMS (2022) or NCC 2022 compliant and as a result have been excluded from the data set, leaving a reduced data set of 54 points. Furthermore, the small number of units above 100kW were identified as potentially distortionary to the analysis so these have also been excluded, leaving 46 points. It is noted that the general trends depicted in the following figures were also present in the full data set. Desensitized PAC unit data is listed in the Appendices, Section 13.2.

For the reduced data set, unit costs were found to be linearly correlated with cooling capacity as shown in Figure 12 Similarly, EER showed some significant correlation with cooling capacity as shown in Figure 13.

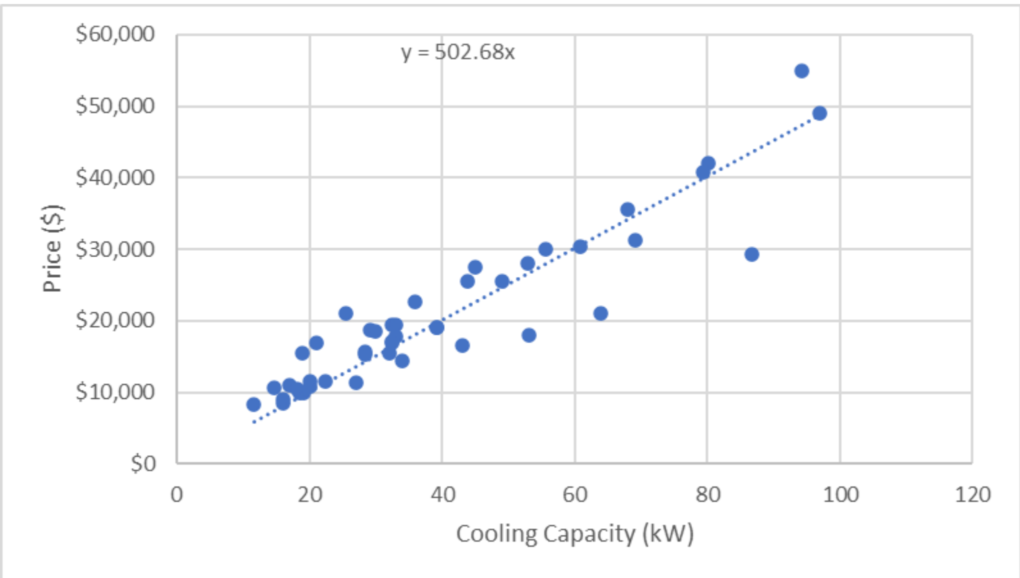


Figure 12. PAC unit cost as a function of cooling capacity.

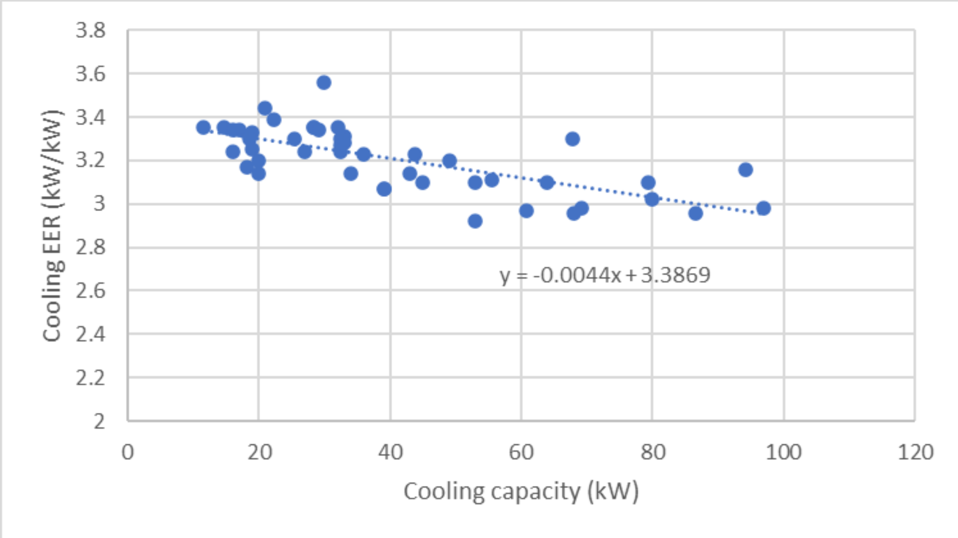


Figure 13. PAC unit Cooling EER versus cooling capacity.

There is positive cost/efficiency relationship based on EER, as shown in Figure 14 However, the limited range of efficiency variation (mostly in the range ±5%) raises questions about the suitability of EER as a basis for a cost/benefit-based efficiency assessment[6].

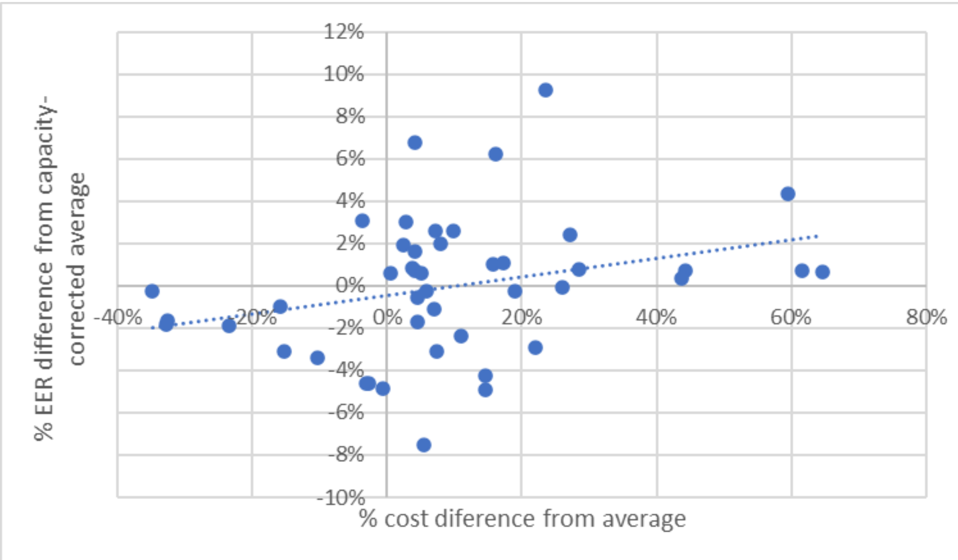


Figure 14. Cost/EER relationship for PAC units

Given data presented in the Section 3.3.1, it is clear that the compressor technology is a larger driver of overall energy use than EER. To this end, cost data has been analysed from the perspective of whether the unit has a fixed speed compressor or a compressor that is either variable speed or has digital capacity modulation. Results are shown in Figure 15, which indicates that digital compressors command a 20% price premium while inverter compressors cost 40% more than fixed speed compressor units.

[6] Although it should be noted that the range of efficiency would have been wider had units not compliant with GEMS/NCC2022 been included in the analysis.

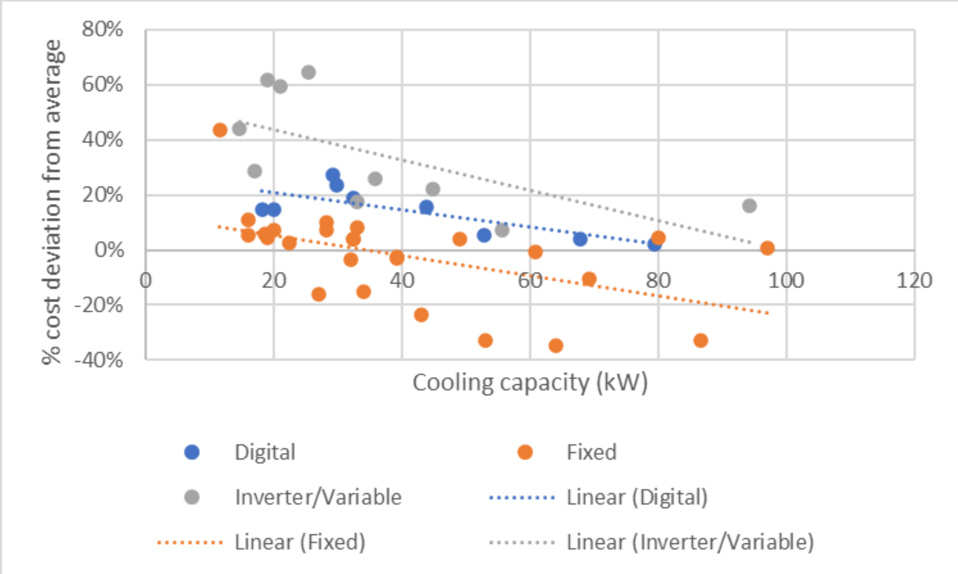


Figure 15. Comparison of PAC unit costs based on compressor control.

3.2.2 Data collection and review – VRF systems

Data was gathered for 215 VRF units across a range from 9-168kW cooing capacity. Unit costs were found to be linearly correlated with capacity, although outliers are present representing ranges from two manufacturers that are significantly cheaper than their competitors as shown Figure 16.

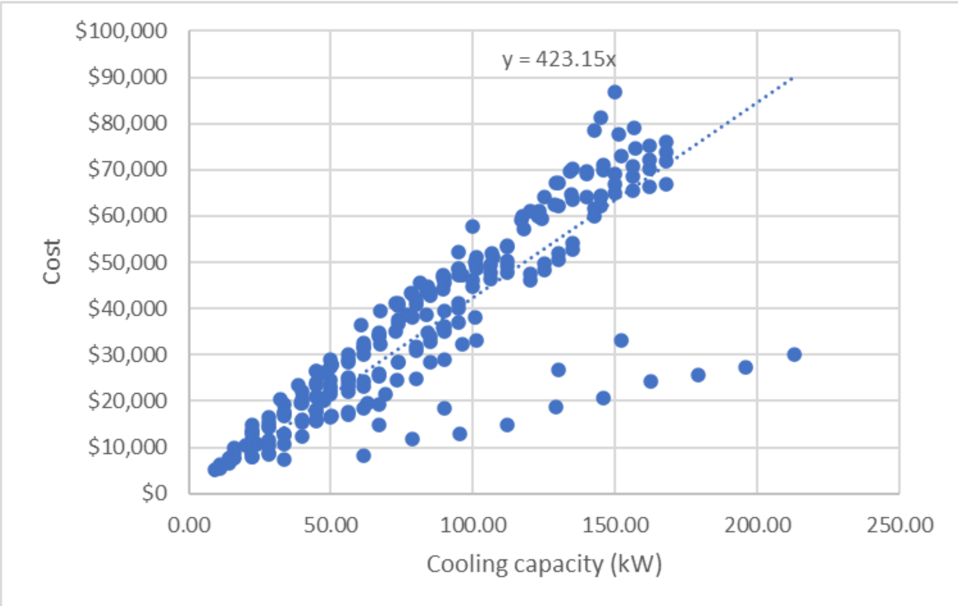


Figure 16. Cost of VRF units as a function of cooling capacity.

The data shows a moderate correlation between capacity and efficiency, as shown in Figure 17. As with PAC units, there is evidence that the MEPS 39kW threshold has an impact on the available units, which indicates that this threshold needs to be considered in the analysis.

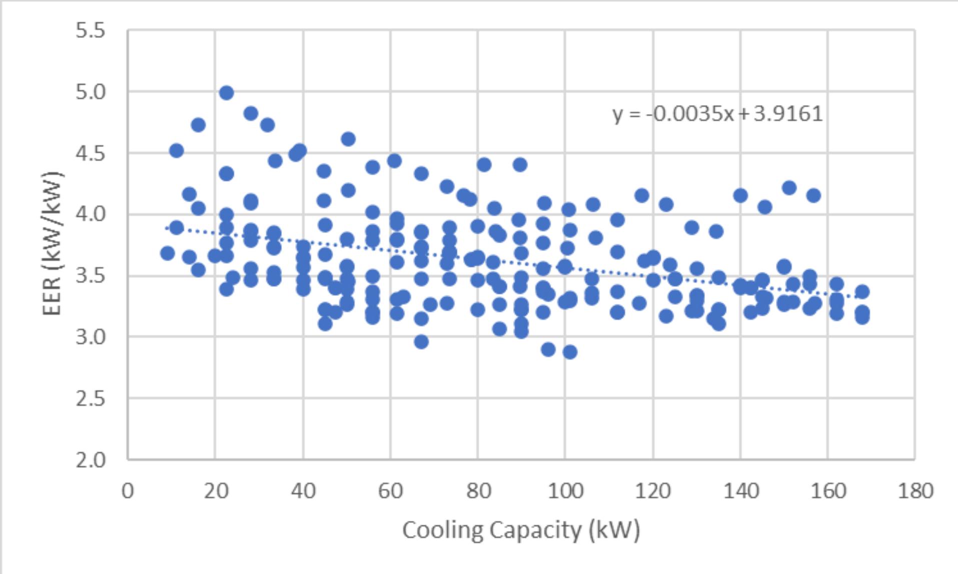


Figure 17. Correlation between VRF capacity and efficiency.

The cost/efficiency relationship is reasonably strong, particularly once the units of exceptionally low cost are excluded, as shown in Figure 18.

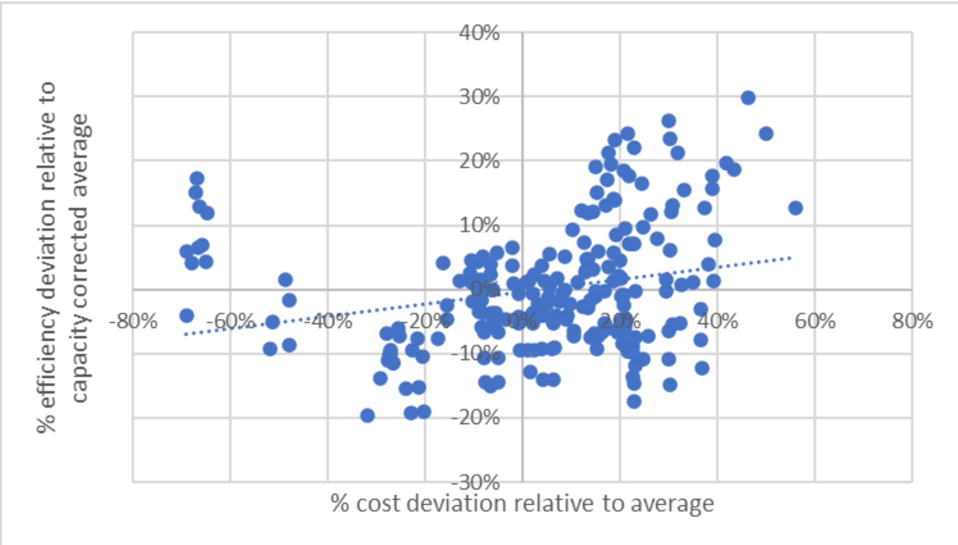


Figure 18. Cost/efficiency relationship for VRF systems including low-cost outliers.

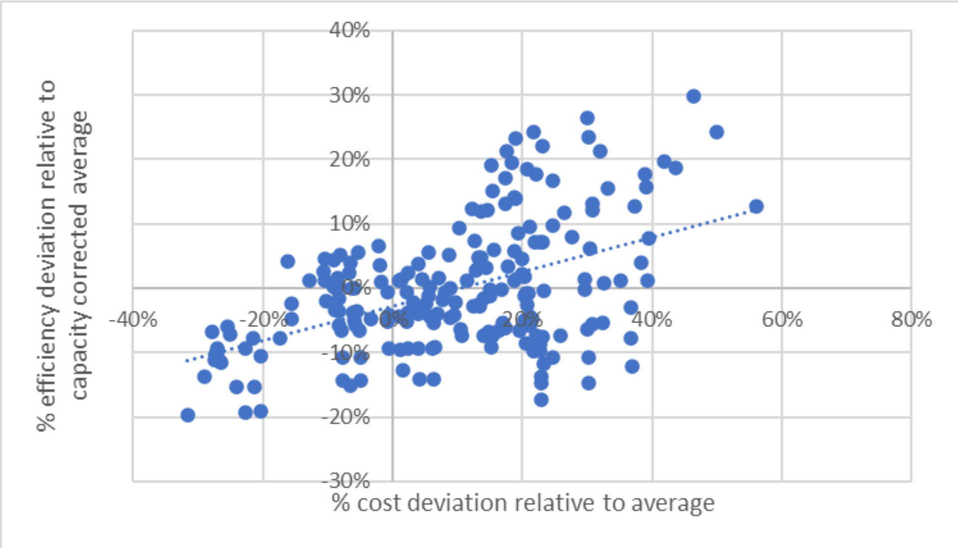


Figure 19. Cost/efficiency relationship for VRF systems excluding low-cost outliers.

3.2.3 Simulation Analysis Methodology

The impacts of unitary AC efficiency on energy use were calculated as follows for two archetypes, being the Medium Office (C5OM) and Aged Care (C9C). All simulations were based on year-round energy use, which covers both heating and cooling modes and resultant energy use and peak demands.

Note that, other than as general context, the simulation analysis was only used for the BCR of fixed speed versus inverter compressor PACs, as discussed in Section 3.3.3.

1. For PACs, three EER levels were simulated, being 2.9, 3.5 and 4.5. These figures span the range from current minimum compliance to the highest EER sourced in the dataset. Two PAC unit types were simulated, being variable speed and fixed speed.
2. For VRF, two EER levels were simulated, being 2.9 and 5, again bounding the market. VRF scenarios were simulated without heat recovery.
3. Typical part load curves were identified for variable speed and fixed speed PAC units. Part load curves were derived from averages of part load curves provided by leading manufacturers for fixed and variable speed units. Basic part load curves used are listed in Appendix *. VRF part load curves were simulated using the IES default curves for this plant type, which are not readily altered[7].
4. The simulation was undertaken for each PAC unit EER in each climate zone for both archetypes, both with and without the presence of an economy cycle. Simulations for VRF were repeated similarly but with no airside economy cycle. Energy results reported included total energy (heating + cooling + defrost).
5. Simulation energy and peak load results were scaled to capacity using an equation of format $\frac{E}{C} = aEER + b$ (where E is Energy use and C is unit capacity) separately for each archetype, climate zone, economy cycle/non-economy cycle and fixed speed PAC/ variable speed PAC/ heat pump VRF.
6. For the PAC unit cost benefit analysis, the NPV of a fixed speed PAC unit at an EER of 3.1 was compared against the NPV of a variable speed PAC unit of EER = 3.1 in each climate zone and

[7] VRF part load data was not available from manufacturers.

with/without economy cycle to calculate the benefit cost ratio for a mandatory variable speed measure.

7. For the VRF cost benefit analysis, the NPV of a minimally compliant VRF system was comparted against the NPV of different grades of higher performing system to calculate benefit cost ratios for different levels of Stringency based on EER.

3.3 Results

3.3.1 Simulation results – PAC units

The simulation results for PAC units (Figure 20 to Figure 23) show a wide range of energy use across the different scenarios. Most notably, the energy use of the variable speed PAC units is typically around 50% (39-57%) lower than that of fixed speed PAC units of the same EER for the office archetype but 64% (38-75%) lower for the Aged Care archetype. The office archetype results are a reasonable match to published results which report 35-50% savings[8]. The higher savings for the Aged Care archetype appear to reflect the nature of operation for this archetype, which has an extensive AC off period in the daytime, followed by high peak loads and then significantly lowered overnight loads, features which would tend to amplify the percentage benefit.

The impact of economy cycle is minor by comparison.

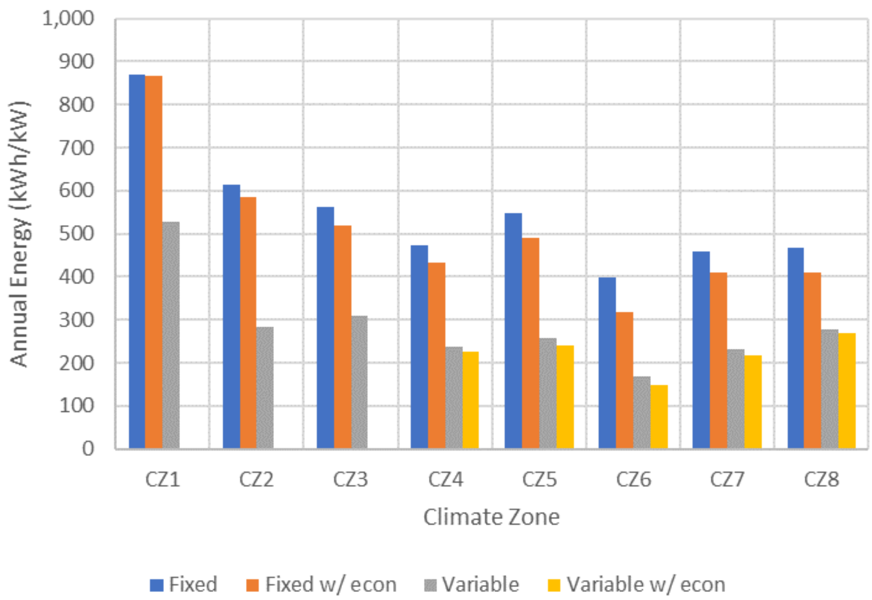


Figure 20. Comparison of fixed and variable speed PAC unit energy use, with and without economy cycle, Medium Office (C5OM) archetype.

[8] "Comparison of energy consumption between non-inverter and inverter-type air conditioner in Saudi Arabia" A. Almogbel, F. Alkasmoul, Z. Aldawsari, J. Alsulami, A. Alsuwailem, *Energy Transitions* (2020) 4:191–197 https://doi.org/10.1007/s41825-020-00033-y reported a 46% saving from laboratory measurements.
"Comparison of Energy Consumption between a Standard Air Conditioner and an Inverter-type Air Conditioner Operating in an Office Building" M. Siriwardhana, SLEMA Journal · September 2017. DOI: 10.4038/slemaj.v20i1-2.5 reported 35% savings.

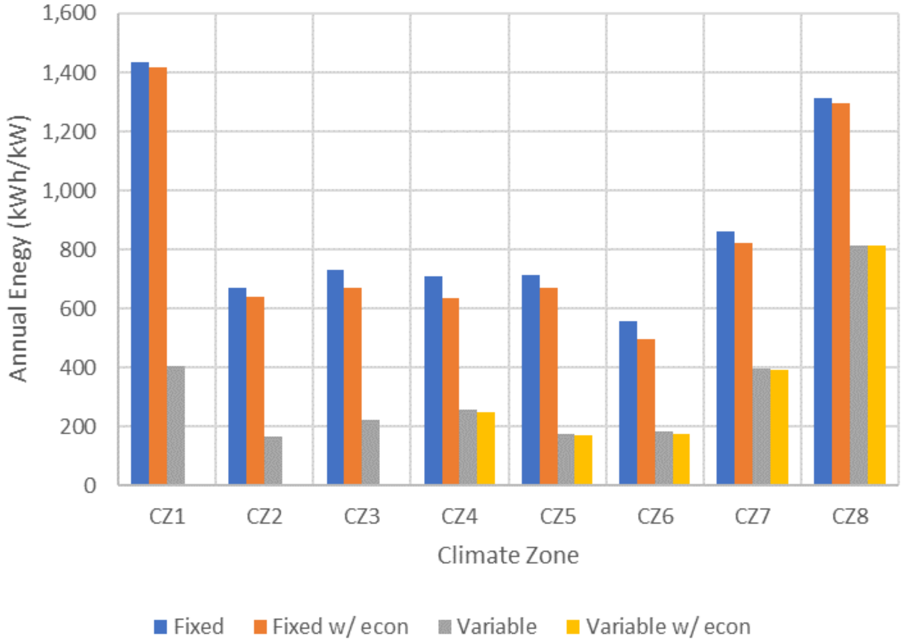


Figure 21. Comparison of fixed and variable speed PAC unit energy use, with and without economy cycle, Aged Care (C9C) archetype.

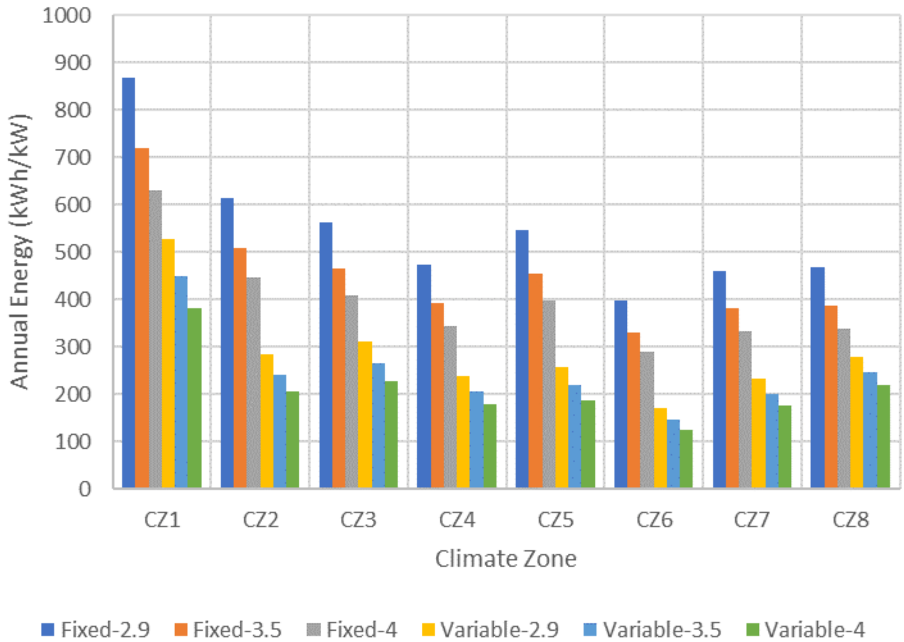


Figure 22. Comparison of fixed and variable speed PAC unit energy use, at different EERs and without economy cycle, Medium Office (C5OM) archetype.

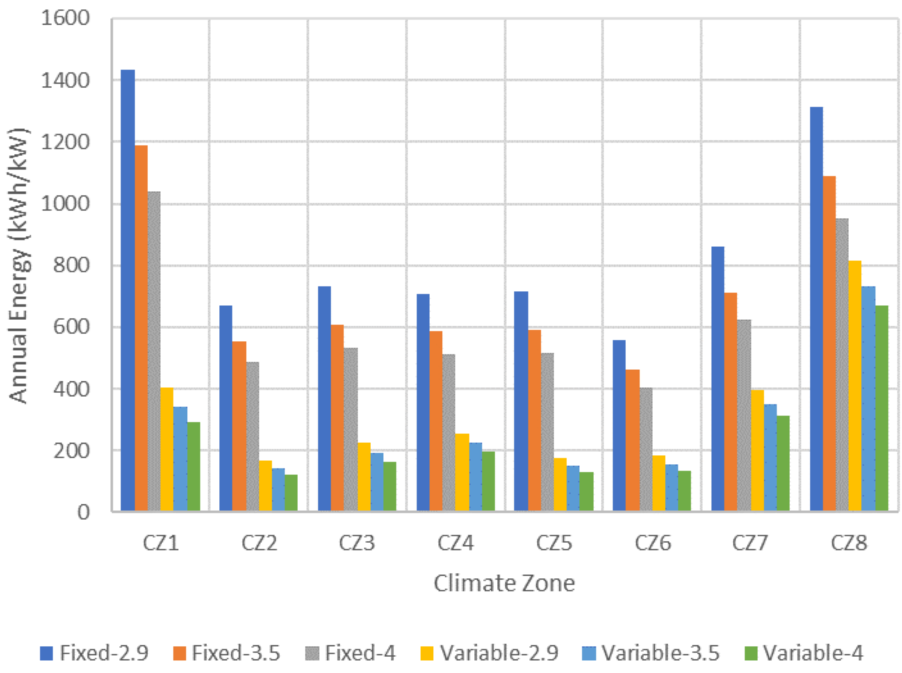


Figure 23. Comparison of fixed and variable speed PAC unit energy use, at different EERs and without economy cycle, Aged Care (C9C) archetype.

3.3.2 Simulation results -VRF

The results for the VRF units are limited to two values at opposite ends of the available EER range. Note that these figures are not directly comparable to the figures for the PAC units owing to differences in the models.

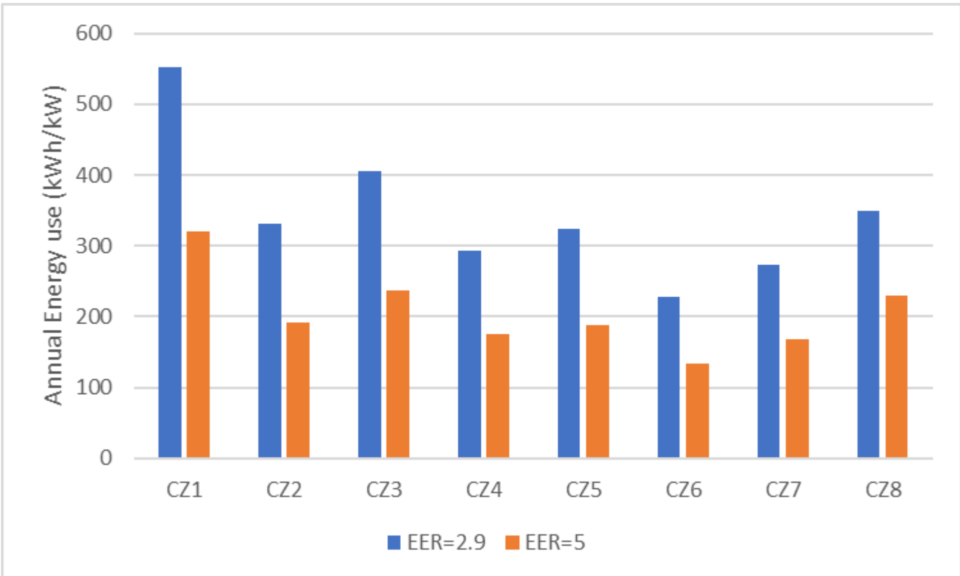


Figure 24. VRF Energy Use per kW capacity - Medium Office (C5OM)

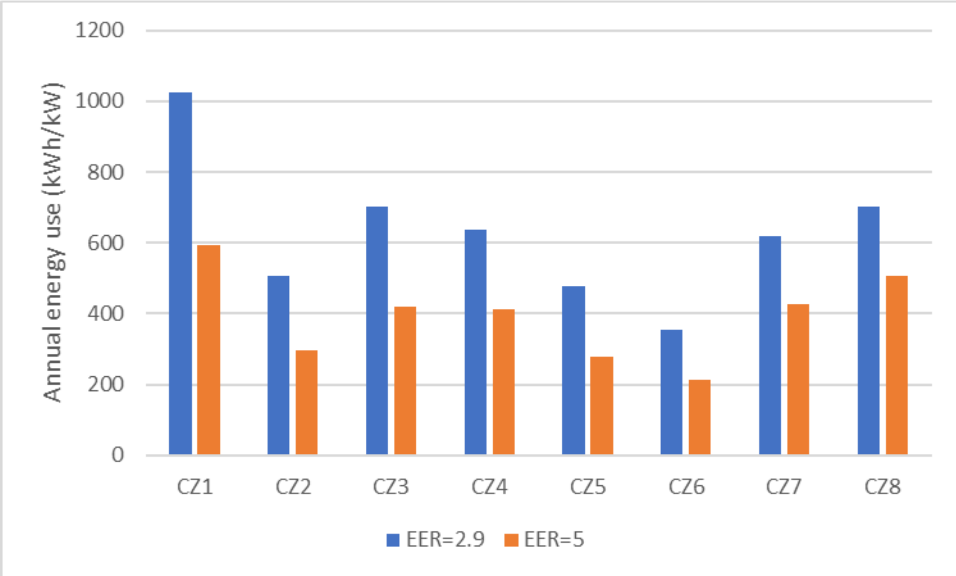


Figure 25. VRF Energy use per kW capacity - Aged Care

3.3.3 BCR analysis results – PACs

Based on the results presented thus far, there are two key questions that appear to merit benefit cost analysis:

1. Is there a meaningful cost-benefit relationship for EER that could form the basis of increased stringency in this indicator?
2. Is there a net benefit in relation to inverter or digital compressor control?

The process for these analyses is laid out in the sections that follow.

PAC unit EER

It is clear from 3.3.1 that the cost-benefit relationship for EER with PAC units is weak at best. This does not support the viability of a significant cost-benefit driven alteration to stringency. However, the lack of a cost-benefit relationship also gives rise to the possibility that a higher stringency could be imposed without incurring a cost. As a result, the primary question for analysis is whether this second approach is viable. This is supported by the fact that the dataset for EER lies significantly above the current requirements, as shown in Figure 26.

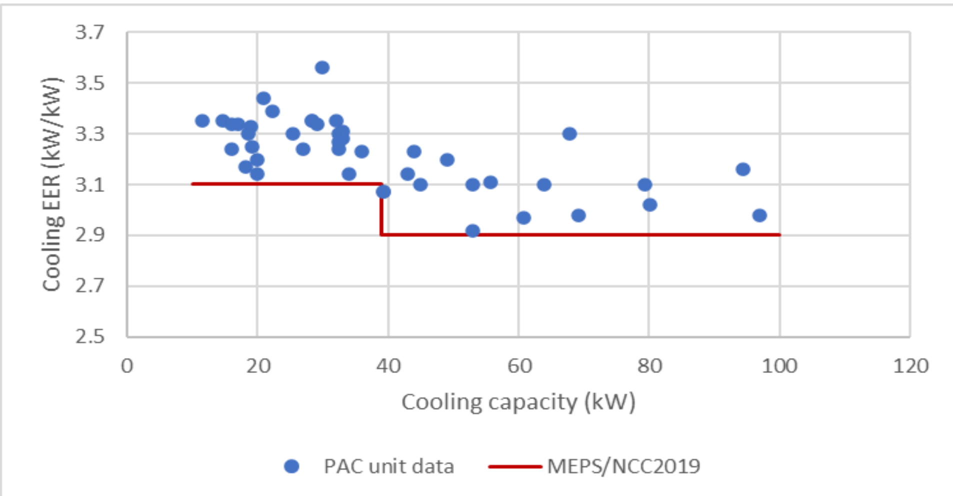


Figure 26. Comparison of PAC unit EER data with Current MEPS and NCC 2019 minimum values

In order to assess this the following process has been adopted:

1. PAC unit data has been divided into two groups, above and below 39kW capacity (which is the threshold for the current MEPS change in stringency and appears to broadly reflect a boundary in available efficiencies).
2. Within each group, the average cost deviation of the lower third of EER points has been calculated.
3. A T-test has been undertaken to compare whether this average cost deviation is significantly different from the average cost deviation for the whole group.

Results are as listed in Table 18 For the T-test results to indicate the average cost deviations are sufficiently different between the lower third of the sample and the sample as a whole, they would need to be less than 0.05; as a result, it can be concluded that none of the tests demonstrated a significant difference. On this basis it can be concluded that a requirement for PAC units to meet the average EER would not impose a significant cost uplift on industry.

Table 18. Comparison of sample average to lower EER third of sample.

Sample	Lower third average EER	Average EER	Lower third average cost deviation	Average cost deviation	T-test result
All units<39kW	3.21	3.30	8%	18%	0.09
Fixed<39kW	3.21	3.29	-2%	5%	0.16
Digital/inverter<39kW	3.22	3.31	19%	33%	0.08
All units>39kW	2.99	3.09	-1%	-2%	n/a
Fixed>39kW	2.98	3.06	-2%	-9%	0.15
Digital/inverter>39kW	3.04	3.13	10%	10%	n/a

On this basis it is concluded that a measure that requires the average EER to be 3.3 for units under 39kW and 3.1 for units over 39kW is viable. This approach does not exclude any of the units within the data set but does mean that designers will need to balance low and high EER units to obtain a qualifying average.

Compressor control

For the assessment of compressor control it is necessary to undertake a cost-benefit analysis. To develop this, the following steps have been taken:

1. To establish the cost of each compressor technology, the cost/kW relationship shown in Figure 12 has been recalculated separately for fixed, digital and inverter PACS. Results are shown in Figure 27.

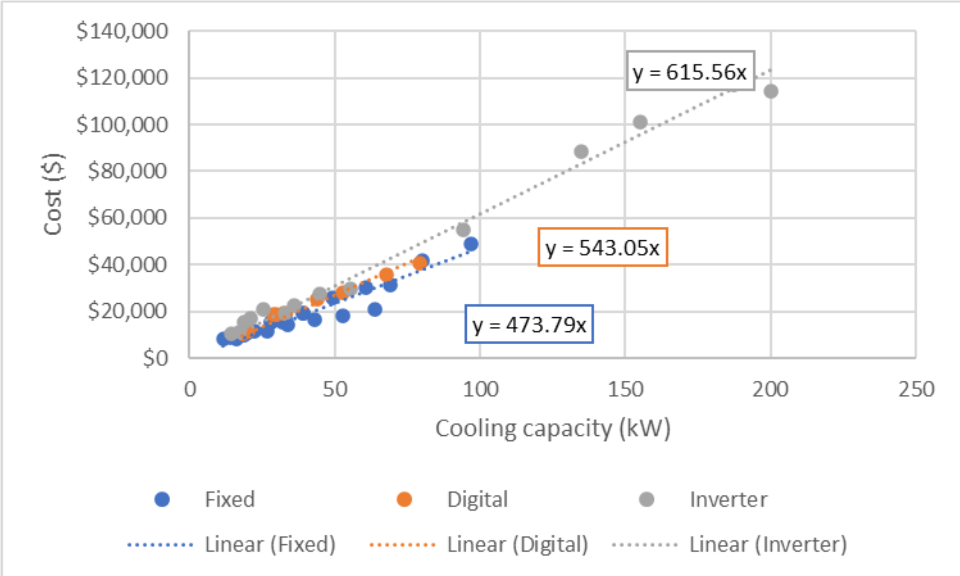


Figure 27. Cost/kW for fixed, digital and inverter PAC units.

2. For the cost-benefit analysis, simulation results from each climate zone and archetype have been applied to the three compressor types, giving a kWh/kW figure for each. EERs have been set at:
    a. Base case: 3.2 <39kW, 3.0>39kW
    b. Stringency case: 3.3<39kW, 3.1>39kW
3. The benefit cost ratio has been calculated for each case (with fixed as the base case) and is shown in Figure 28 to Figure 31 below.

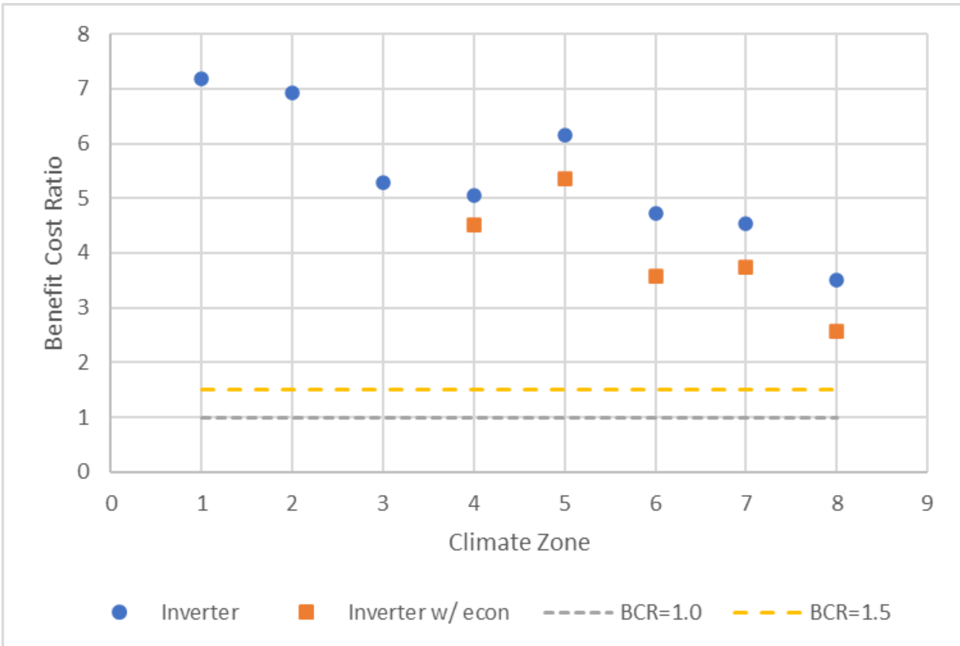


Figure 28. Benefit cost ratios for EER changes plus the use of inverter compressors (<39kW) for the daytime (Medium Office C5OM) archetype.

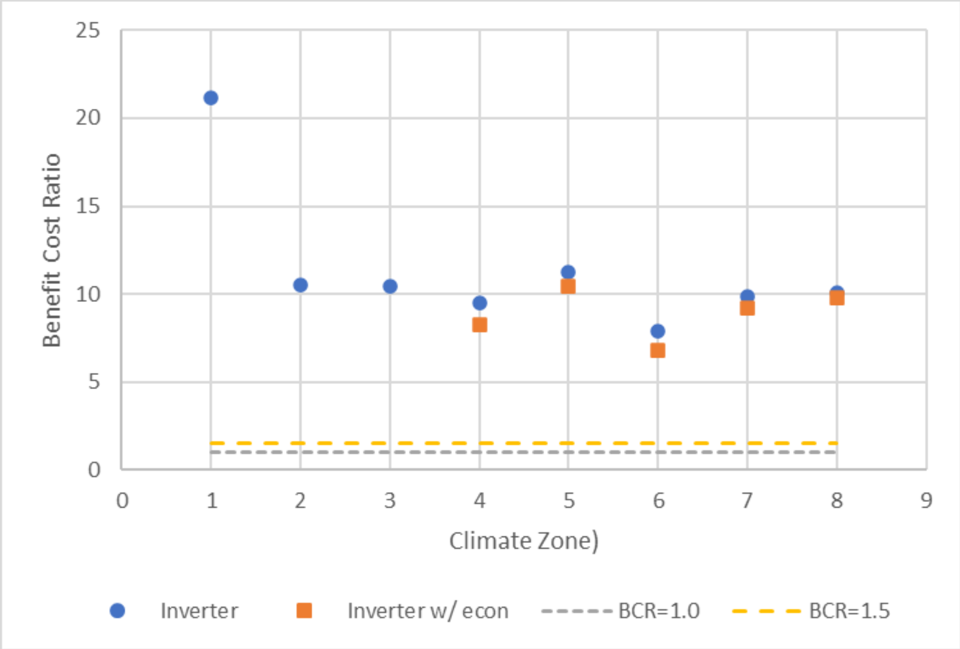


Figure 29. Benefit cost ratios for EER changes plus the use of inverter compressors (<39kW) for the 24/7 (C9C Aged Care) archetype.

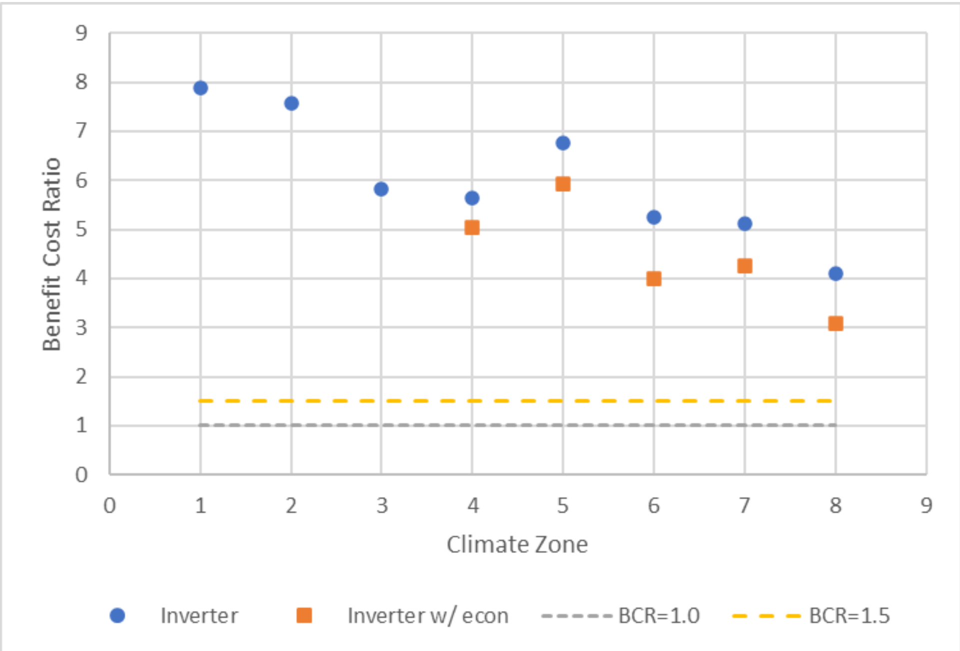


Figure 30. Benefit cost ratios for EER changes plus the use of inverter compressors (>39kW) for the daytime (Medium Office C5OM) archetype.

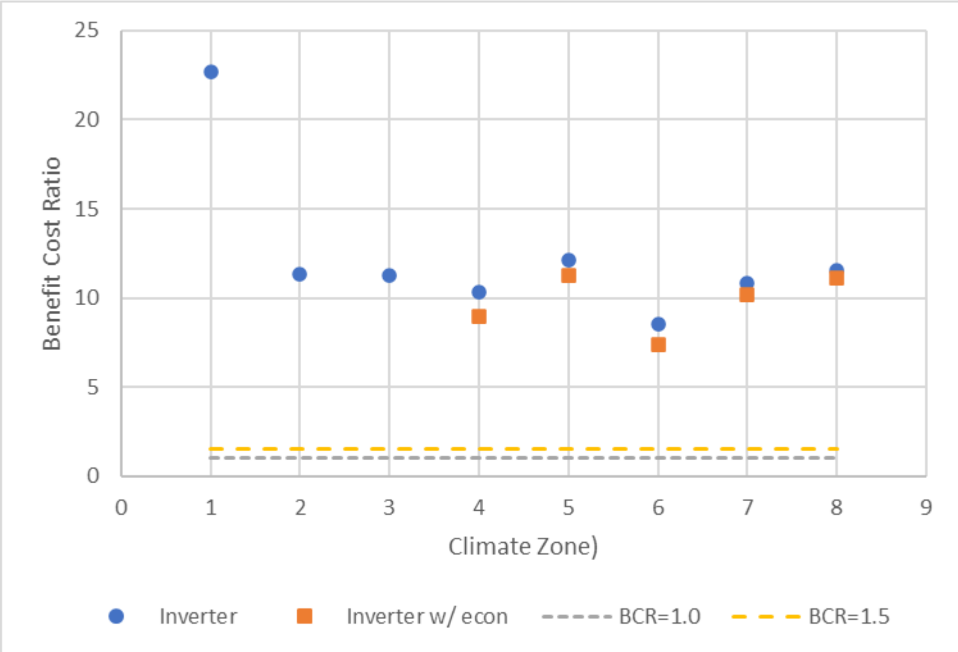


Figure 31. Benefit cost ratios for EER changes plus the use of inverter compressors (>39kW) for the 24/7 (C9C Aged Care) archetype.

3.3.4 BCR analysis results – VRF

The analysis for VRF is somewhat simpler given that there is only one primary technology option to consider. Furthermore, in contrast to PAC units, there is a reasonable cost/EER relationship and a wider range of EERs; the data set is also somewhat larger.

The assessment therefore initially follows a similar process to that used for PAC unit EERs:

1. The data was split at 39kW cooling capacity, with above and below being considered as separate data sets.
2. In each capacity category, the average EER and average cost deviation were evaluated for the whole data set and for individual deciles.

The significance of the average cost difference between EER quintiles and the average was tested using a t-test. Results are shown in Table 19 and

3. Table 20. The results broadly indicate that the quintiles are validly differentiated by cost.

Table 19. Comparison of EER deciles for VRF units above 39kW.

Quintile	Average EER	Average cost, $	t-test vs average
1	3.17	432	0.01
2	3.34	447	0.10
3	3.51	461	0.44
4	3.72	441	0.08
5	4.11	516	0.00

Table 20. Comparison of EER deciles for VRF units below 39kW.

Quintile	Average EER	Average Cost, $	t-test vs average
1	3.48	402	0.03
2	3.68	411	0.04
3	3.86	494	0.45
4	4.16	544	0.11
5	4.68	591	0.01

4. As the intent is to regulate average efficiency rather than minimum efficiency, it is necessary to represent the average cost of units complying with a given average EER. To emulate this, the quintiles Q1-Q5 have been combined into a set of nine rolling averages covering the full range of unit expected to be used to meet a given average efficiency target as follows: Q1, Q1&2, Q1-3, Q1-4, Q1-5, Q2-5, Q3-5, Q4-5, Q5.
5. Based on this we can select the following stringency points for each size range:

Table 21. Stringency test levels for VRF units above 39kW

Band	Average EER	Average Cost, $
Q1	3.17	432
Q1-2	3.25	439
Q1-3	3.34	447
Q1-4	3.43	445
Q1-5	3.57	459
Q2-5	3.67	466
Q3-5	3.78	473
Q4-5	3.91	478
Q5	4.11	516

Table 22. Stringency test levels for VRF units below 39kW

Band	Average EER	Average Cost, $
Q1	3.48	402
Q1+2	3.58	406
Q1+2+3	3.67	435
Q1-4	3.79	463
Q1-5	3.97	488
Q2-5	4.09	510
Q3-5	4.23	543
Q4-5	4.42	567
Q5	4.68	591

6. The relative performance of these stringency test levels can be tested in the cost benefit analysis using the kWh/kW figures from the two archetype simulations, with the lowest cost/EER group taken as the base case, giving results as shown in Figure 32 to Figure 35 below.

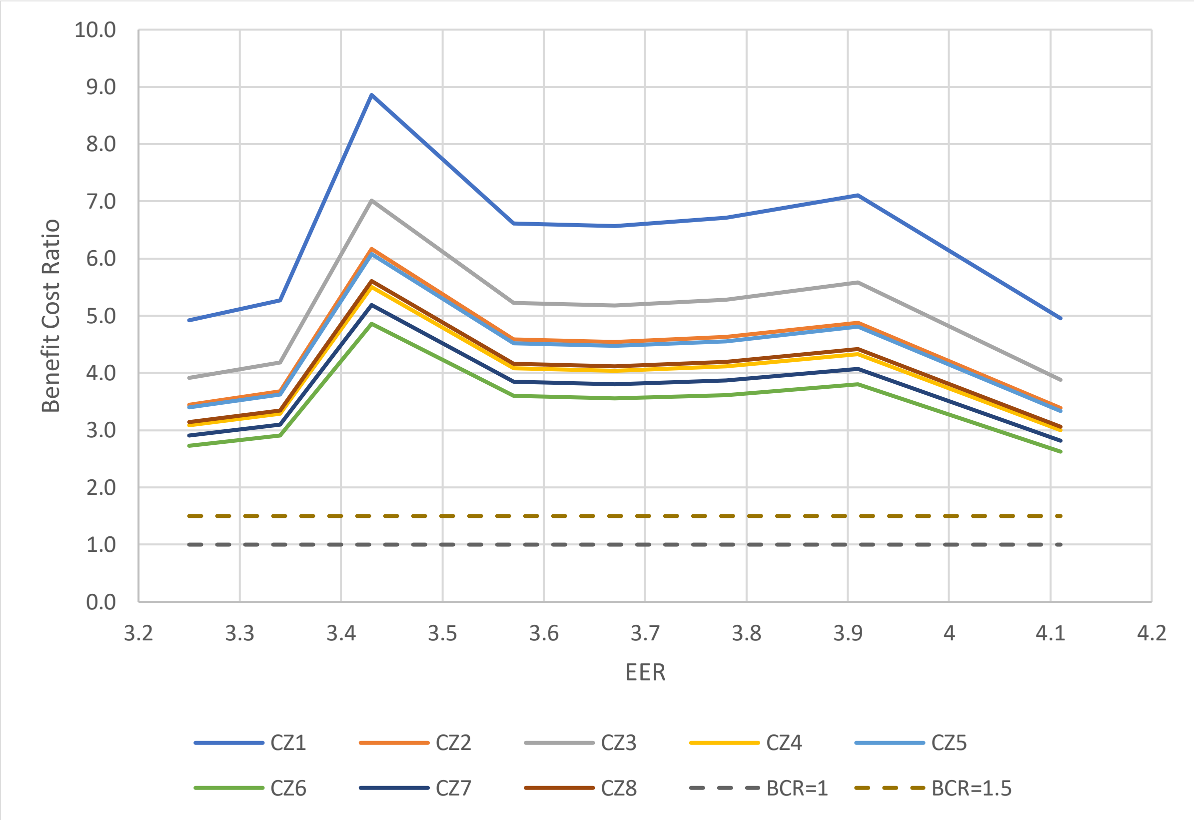


Figure 32. BCRs for VRF units over 39kW, Medium Office archetype (C5OM).

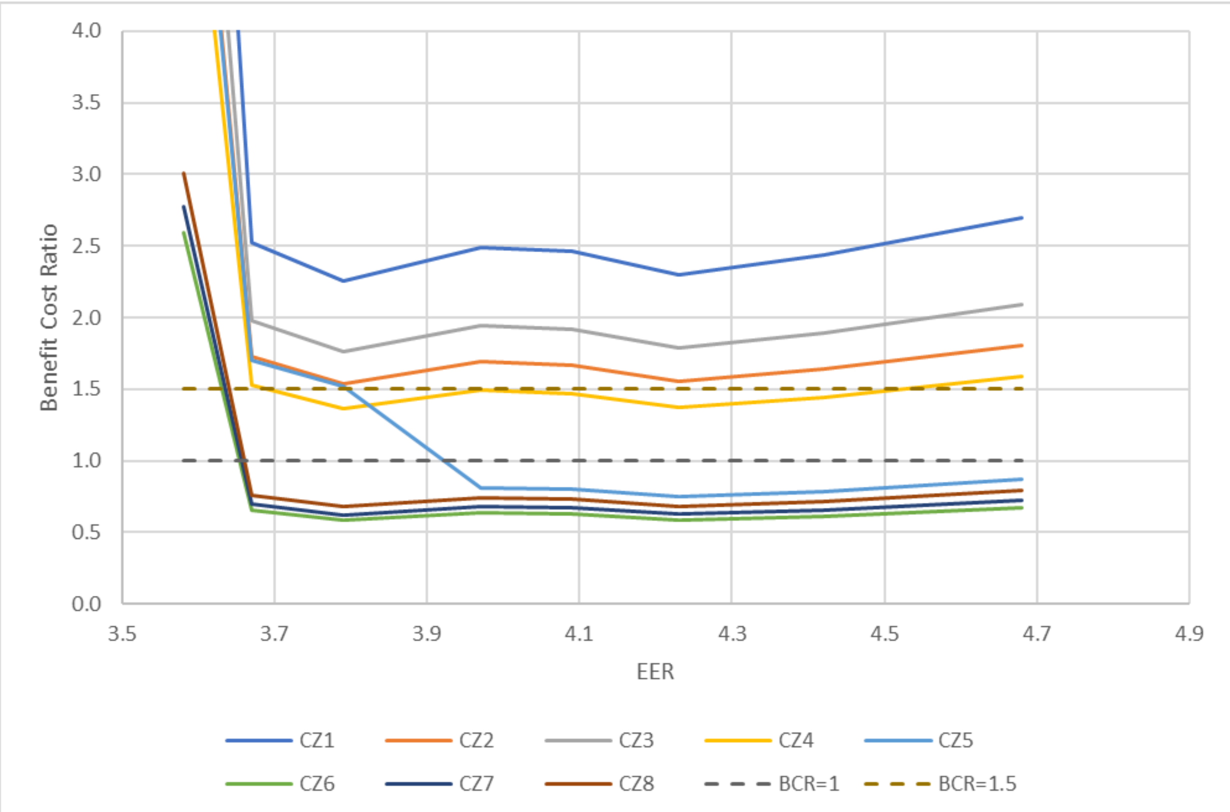


Figure 33. BCRs for VRF units under 39kW Medium Office archetype (C5OM)

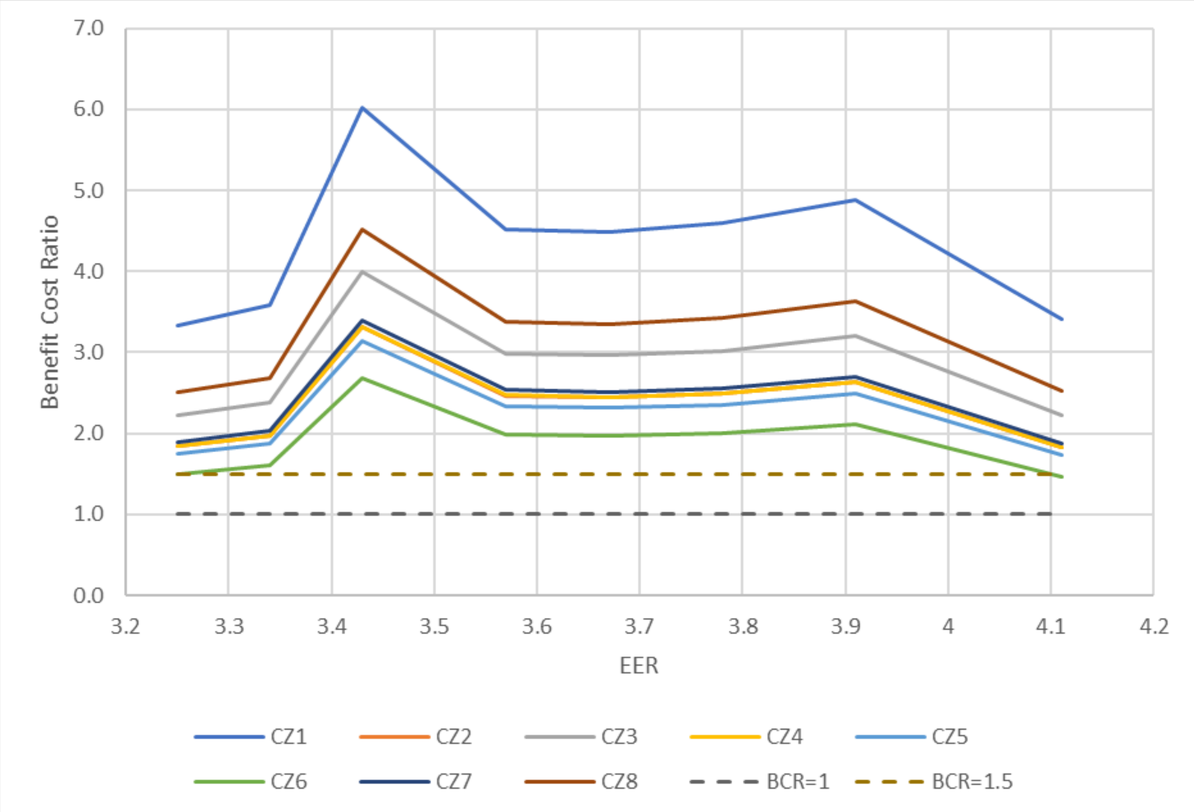


Figure 34. BCRs for VRF units over 39kW, Aged Care archetype (C9C)

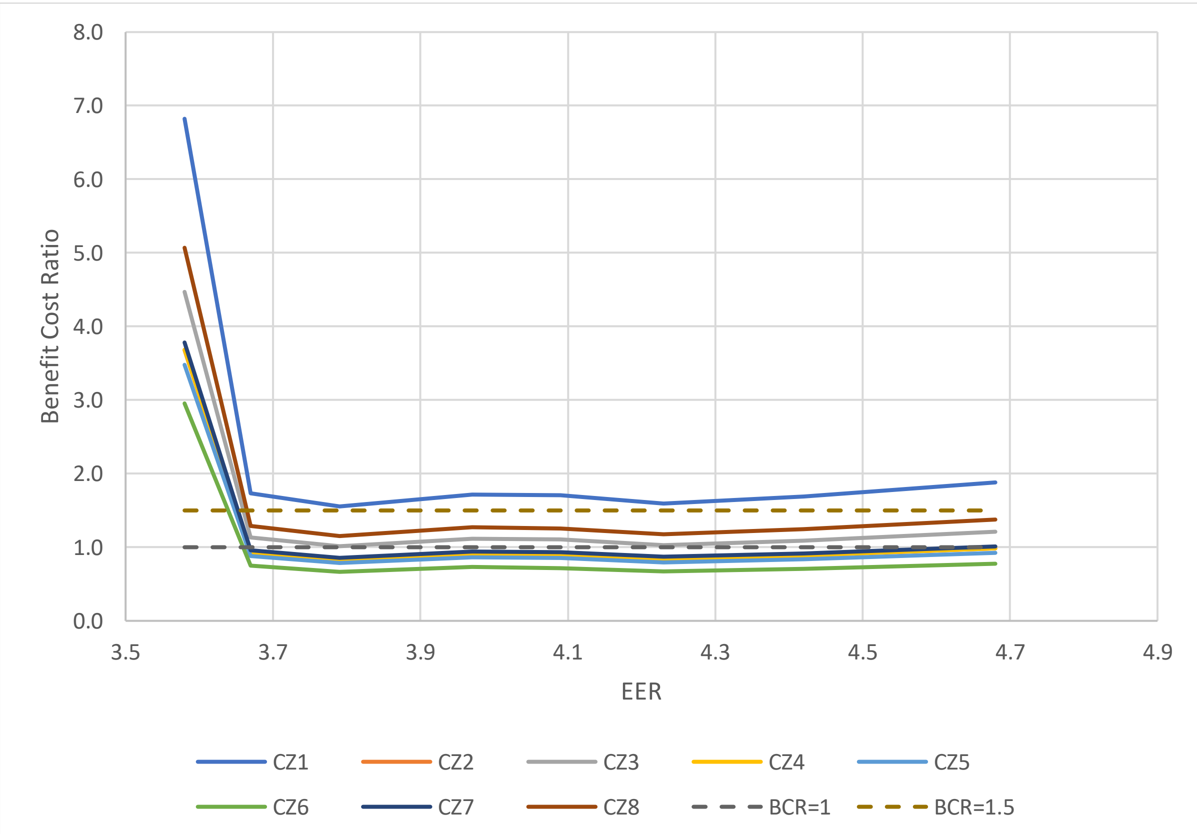


Figure 35. BCRs for VRF units under 39kW, Aged Care archetype (C9C)

3.4 Discussion

3.4.1 PAC units sizing practices

The analysis has been based on an assumption of "good" sizing practice, with EERs being evaluated at nominal peak capacity and matched to loads at this nominal level. However, the data we have received indicates that units are able to operate above nominal capacity but at significantly reduced efficiency – across 4 units in one manufacturer's range the maximum capacity was between 106% and 125% of the nominal capacity but the corresponding EER dropped by between 8.5% and 24%. It is suspected that this is more of an issue for variable speed PACs, as the non-linearity of turndown can mean that a unit complies with an efficiency rating at a reduced capacity.

The real world impact of this is of course variable, as in practice a unit that has been "stretched" to its maximum capacity will in reality spend much of its time operating at part load (although it will of course, spend more time operating in the higher end of its range). Nonetheless it would appear desirable to ensure that units are selected to match load.

This can be achieved through a requirement that the unit EER is evaluated at the design load rather than at the nominal maximum load of the unit.

3.4.2 PAC units vs VRF interchangeability

The significant difference in EER between PAC units and VRF systems somewhat begs the question of whether there is a point at which VRF should be the baseline standard, ahead of the use of PAC units. This would appear to be further supported by the practical interchangeability of the two technologies: in essence the only real difference is that PAC units can be configured for economy cycles whereas VRF systems are normally (but not solely) configured as Fan Coil Unit (FCU) systems that cannot be provided with an economy cycle.

However, running against this proposition are the following factors:

1. We have not undertaken a cost-benefit analysis of replacement of PAC units by VRF.
2. In practice, there are different special considerations for the two system types, with VRF offering a compressed footprint both outdoor and indoor relative to PAC units.
3. There are additional limitations on VRF systems imposed by refrigerant dilution requirements (essentially, there has to be consideration of the management of a refrigerant leak into the occupied space)
4. VRF systems are currently only available with high GWP refrigerants[9] (R410a, GWP=2088) – and contain high volumes of these in a site-assembled network - whereas newer PAC units are moving towards the use of R32 and have less refrigerant which is also contained in a factory assembled unit.

As a result, it is not proposed to integrate the EER requirements for VRF and PAC units into a single overarching requirement.

[9] Our enquiries with suppliers indicated no immediate plans to change from R410a in VRF systems.

3.4.3 VRF unit stringency

The VRF BCR results indicate that the full range of EERs is potentially cost-effective in all cases above 39kW. This means that the BCR is not a useful indicator of a potential regulation, as selection of the maximum EER would be cost effective but would exclude the majority of the market.

The structure of the stringency cases is such that the fifth stringency band (EER 3.57 for units above 39kW) represents the average of the available units on the market; above this stringency the lower quintiles become progressively excluded from a practical combination of units to achieve the required EER, such that the 6th stringency level effectively excludes the bottom 20% of efficiencies, the 7th level 40% and so on. It seems reasonable under these circumstances for the 6th stringency level to be selected as a balance between efficiency and the management of market issues. This corresponds to an average EER requirements of 3.67, rounded to 3.7 for units above 39kW.

For units below 39kW the results are more nuanced, with milder climate zones frequently not achieving a BCR of greater than 1 above an EER of 3.7, while in warmer climate zones the results show a lack of clear relationship between BCR and EER above 3.7kW. Furthermore, for the situations where the BCR is less than 1, the BCR/EER relationship is essentially flat, indicating that above EER 3.7 there is little difference in the economics; furthermore, while the BCRs are less than 1, they are often close to 1. This complex outcome could be used to create a complex regulation, which would be undesirable. As a result, the decision was made to adopt the same approach for determining the initial stringency as for units above 39kW, i.e. based on a limited exclusion of the poorest units. This results in a proposed stringency of 4.09, rounded to 4.1 for units below 39kW. The validity of this approach will be tested in a whole building context in the next phase of the project to ensure that overall economic performance meets the BCR>1 criterion[10].

3.4.4 PAC units' stringency

EER

As demonstrated by the analysis in Section 3.3.3, the proposed measure in relation to EER is for the average EER to be 3.3 for units under 39kW and 3.1 for units over 39kW. This would be evaluated based on a capacity weighted basis.

The existing minimum EER of 2.9 is effectively applied as a floor via GEMS regulation and does not need to be restated in Code text.

Compressor technology

The results indicate a strong case for mandating inverters in almost all situations.

It is noted that no analysis has been undertaken for digital compressors, as part load data for these has not been available in time for the completion of this report. However, their cost lies roughly midway between fixed speed and inverter compressors and general industry understanding indicates that this is true for performance as well. As a result, it is recommended that the measure captures digital compressors as an acceptable alternative to inverters.

[10] The preliminary results of the whole building analysis strongly support the validity of the proposed stringency.

Measure construction

The viability of a methodology that permits mixing of fixed and inverter compressors to achieve an equivalent performance to an all-inverter solution was considered but rejected. This is because the performance gap between fixed and variable speed compressors cannot be matched by an equivalent upgrade in EER, rendering an EER/inverter trade-off unviable.

Impact on Refrigerants

The data set contained PAC units using two refrigerants being R410a (GWP = 2088) and R32 (GWP = 675). There was no significant correlation between EER and refrigerant use, but there was a significant bias towards R32 use in inverter and digital compressor PAC units. As a result, the proposed measure would tend to reduce the GWP of refrigerants installed in the field.

3.5 Proposed Measures

J6D12 Unitary air-conditioning equipment

(a) Water cooled unitary air-conditioning equipment of a capacity greater than or equal to 65 kWr, including packaged air-conditioners, split systems, and variable refrigerant flow systems must have a minimum energy efficiency ratio of 4.0 Wr / Input power for cooling when tested in accordance with AS/NZS 3823.1.3 at test condition T1, where input power includes both compressor and fan input power; or

(b) Air-cooled unitary air-conditioning equipment of a capacity greater than or equal to 10 kWr, including packaged air-conditioners, split systems, and variable refrigerant flow systems must:

(i) have a Unitary Air-conditioning Weighted EER index of greater than the value calculated according to the Specification NN; and

(ii) have variable speed condenser fans and at least one variable capacity compressor in the form of a digital scroll or inverter driven compressor.

Specification NN: Unitary Air-conditioning Weighted EER Index

The Unitary air-conditioning weighted EER index, WEI, shall be calculated according to the equation:

$$\mathrm{WEI} = \frac{\sum_{\text{All unitary systems i}} \mathrm{EER_i D_i}}{\sum_{\text{All unitary systems i}} \mathrm{EER_i^{min} D_i}}$$

Where:

i. EER_i is the energy efficiency ratio of unit i when tested in accordance with AS/NZS 3823.1.2 at test condition T1, where input power includes both compressor and fan input power, evaluated at the design load D_i for the intended application; and
ii. Di is the design cooling capacity of unit i when tested in accordance with AS/NZS 3823.1.2 at test condition T1, where input power includes both compressor and fan input power; and
iii. $\mathrm{EER_i^{min}}$ is the minimum EER for unit i as defined in Table 23.

Table 23. Minimum EER figures for air-cooled unitary air-conditioning systems

Design cooling capacity (thermal)	Unitary systems other than VRF	VRF systems
≤39kW	3.3	4.1
>39kW	3.1	3.7

4 Heat Pumps

4.1 Background and context

The term 'heat pump' applies to a very wide range of equipment. For the purposes of this assessment and measure, a 'heat pump' refers to a single piece of equipment that uses air as a heat source to add heat to water (air-sourced heat pump) at temperatures of up to 60°C using a single stage refrigeration cycle in order to service a space heating application[11]. This type of heat pump is commonly referred to as a 'low temperature air sourced heat pump' and excludes other types such as heat pumps using water as a source, and/or high temperature heat pumps (often using multi-stage, or cascading, refrigeration circuits to generate 80-90°C water temperatures).

Heat pump efficiency is not regulated under NCC 2022, reflecting the fact that until recently this technology was rarely used in Australia. However, with increasing emphasis on degasification of buildings, heat pumps are expected to become a significant component of the building services market over the next 5 years. As a result, it is highly desirable for heat pump efficiency to be regulated under NCC 2025.

It is noted that the vast majority of heat pumps available are reversible, i.e. they can be operated as chillers. This is factored into the current analysis, but priority is given to the efficiency and operation in heating mode.

4.2 Efficiency measurement standards for heat pumps

There is a lack of consistent performance measurement standards for heat pumps in the Australian market. It seems this is due to several factors:

- The relative infancy of the commercial (air-to-water) heat pump market in Australia
- Low demand for the equipment (until recently)
- Equipment being sourced from various locations around the world. Our data gathering exercise showed the majority of equipment to be manufactured in European countries, but some equipment is still developed and manufactured in countries such as the United States (US), China and Japan.

Our data gathering exercise aimed to require suppliers to provide data for heat pumps in cooling mode under AHRI (US) standards, and heating mode under European standards. The intention of this method was to both standardise the data comparison and also 'stress' test the current industry to determine in advance whether one measurement standard would be obviously more appropriate to enforce than the other.

In parallel with our performance data request a review of approximately 12 performance measurement standards was carried out. This was reduced down to two groups as the most appropriate, widely used and applicable to use as the method of rating heat pump performance:

- AHRI Standard 551/591-2023 (SI Edition) – US Standard
- European standards EN 14511 and EN 14825

[11] Heat pumps in domestic hot water applications are not covered under this analysis.

4.2.1 AHRI 551/591

AHRI 551/591 is already referenced by the NCC as it is used to classify and measure the performance of air and water cooled chillers. This standard also contains a method to measure the performance of air to water heat pumps at full load (COP_H). There is no method to measure part load or seasonal performance in heating, such as IPLV or NPLV for chillers. This standard is referenced in the US by ASHRAE 90.1 and was also found to be the basis of some Chinese performance measurement standards.

5.1.2.1 The Heating Coefficient of Performance (COP_H), kW/kW, shall be calculated using Equation 15:

$$COP_H = \frac{Q_{cd}}{W_{input}} \qquad (15)$$

Figure 36: COP_H Calculation as Defined by AHRI 551/591

5.4.2 *Heating Part-load Ratings.* Heat Pump Water-heating Packages and Heat Recovery Water-chilling Packages can be rated at individual part load points. Neither IPLV.SI nor NPLV.SI shall be calculated for such points.

Figure 37: AHRI 551/591 Note for Heat Pumps Regarding Part Load or Seasonal Performance

Due to the simplicity in its performance measurement method, the vast majority of suppliers approached for data were able to provide data in accordance with this Standard.

4.2.2 European standards EN 14511 and EN 14825

The European Standards EN 14511 and EN 14825 and International Standards ISO 19967.2 and ISO 21978 are interchangeable as they are a direct replica of one another. These EN standards are referenced by EU regulations to define energy efficiency requirements in the EU: (EU) 813/2013 for heating, and (EU) 2016/2281 for cooling, for heat pumps <400kW. These standards define a method of measuring performance at both full and seasonal/part load.

While the seasonal method reflects a similar purpose as IPLV for chillers, it is much more detailed, complex, and customizable. The seasonal performance (Seasonal COP, SCOP, for heating and SEER for cooling) is calculated based on a set of annual ‘bin’ temperatures for a given location in Europe (three possible locations classified as warmer, average or cooler), and weighted based on the number of hours that location sustains a given temperature. The calculation also requires ancillary energy consumption (such as compressor sump heaters) to be accounted for, and provides the ability to interrogate a machine’s performance under various operating control strategies:

- FW/FO = Fixed Water Flow, Fixed Outlet Temperature
- FW/VO = Fixed Water Flow, Variable Outlet Temperature
- FW/FO = Variable Water Flow, Fixed Outlet Temperature
- VW/VO = Variable Water Flow, Variable Outlet Temperature

$$SCOP = \frac{Q_H}{Q_{HE}}$$

where

Q_H is the reference annual heating demand, expressed in kWh;

Q_{HE} is the annual energy consumption for heating, expressed in kWh.

Figure 38: EN 14825 Extract - Annual Performance Measurement

$$Q_H = P_{designh} \times H_{HE} \qquad (14)$$

where

$P_{designh}$ is the design heating load of the building the unit is suitable for as declared by the manufacturer, expressed in kW;

H_{HE} is the number of equivalent active mode hours for heating, expressed in h.

Figure 39: EN 14825 Extract - Reference Annual Heating Demand Calculation

$$Q_{HE} = \frac{Q_H}{SCOP_{on}} + H_{TO} \times P_{TO} + H_{SB} \times P_{SB} + H_{CK} \times P_{CK} + H_{OFF} \times P_{OFF} \qquad (15)$$

where

Q_H is the reference annual heating demand, expressed in kWh;

H_{TO}, H_{SB}, H_{CK}, H_{OFF} are the numbers of hours the unit is considered to work in thermostat-off mode, standby mode, crankcase heater mode and off mode respectively, expressed in h;

P_{TO}, P_{SB}, P_{CK}, P_{OFF} are the power inputs during thermostat-off mode, standby mode, crankcase heater mode and off mode respectively, expressed in kW;

$SCOP_{on}$ is the active mode seasonal coefficient of performance,, expressed in kWh/kWh.

Figure 40: EN 14825 Extract Illustrating the Ancillary Energy Consumption Components Included in the Calculation

While far more robust in theory, it has been very difficult to standardise the data gathered on various heat pumps as suppliers either report under different conditions associated with SCOP and EER or appear to not be aware of which method (set of conditions) of SCOP/SEER calculation their data is calculated under. These factors also make it difficult to impose on industry as a measurement standard, particularly as the air to water heat pumps available in Australia are not all sourced from one location/continent.

Ultimately, the availability of performance data for a given machine under a certain measurement standard was highly correlated to the geography of where the machine was either manufactured or developed.

4.3 Methodology

The outline methodology is shown in Figure 41

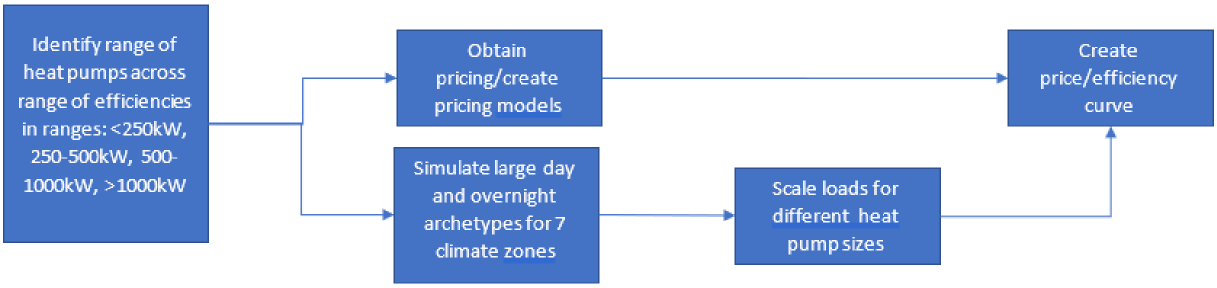


Figure 41. Outline methodology for the heat pumps analysis.

4.3.1 Data collection and review

Data was collected for 49 heat pumps across 7 manufacturers. The full load efficiency of heat pumps within the data sample is shown in Figure 42.

Figure 42. Efficiency of heat pumps

It can be seen that the range of COP and EER is fairly limited and is largely uniform across the range of 100-1200kW. Average heating COP is 3.24, with no significant dependence on capacity.

Part load efficiency data was requested from suppliers but was found to be difficult to obtain, indicating that the market may not be sufficiently mature in this respect. Insufficient data was obtained for any meaningful cost/efficiency analysis based on part load indicators.

4.3.2 Cost/Efficiency relationship

As a result of the lack of part load data, the cost/efficiency relationship has only been analysed based on full load performance indicators, as shown in Figure 43 and Figure 44 for heating and cooling performance respectively.

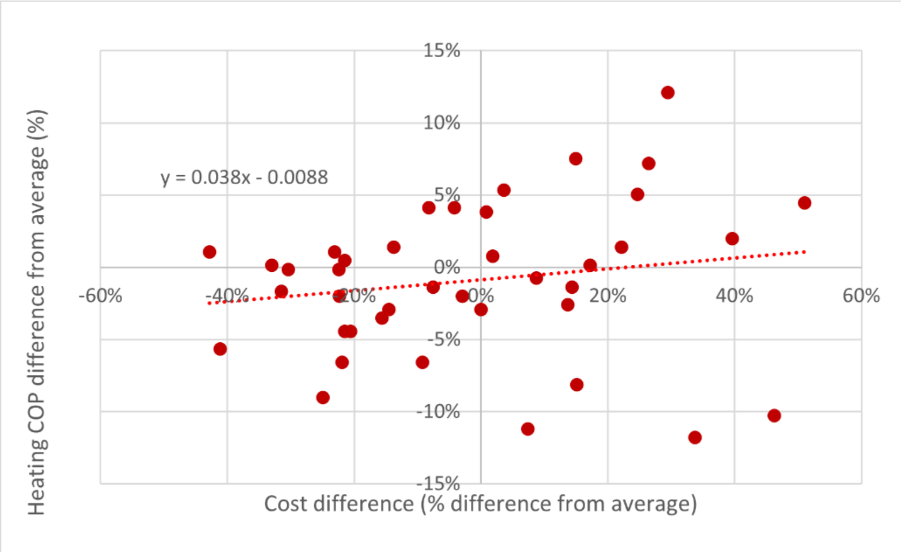


Figure 43. Cost/efficiency relationship for heating COP.

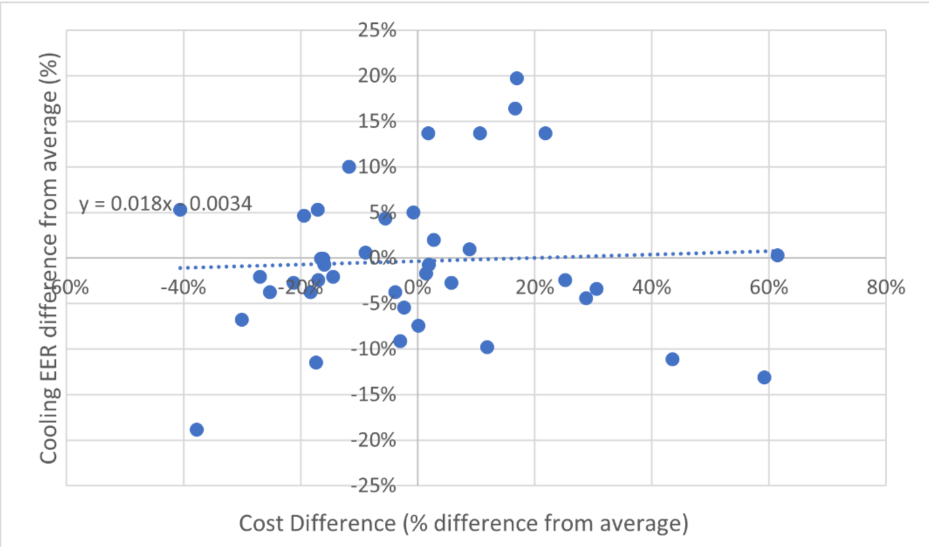


Figure 44. Cost/efficiency relationship for cooling EER.

4.3.3 Manufacturer-efficiency relationship

Given the analyses carried out above are leading to findings of loose correlations for efficiency, a further analysis was carried out to determine whether efficiency is correlated to manufacturers. Figure 45 and Figure 46 illustrate efficiency in heating and cooling modes separately, identified by manufacturer. Figure 47 illustrates the spread of average efficiencies among the various manufacturers. It can be seen that there is little manufacturer/efficiency relationship, with the exception of manufacturer G, whose data consisted of a single high efficiency unit.

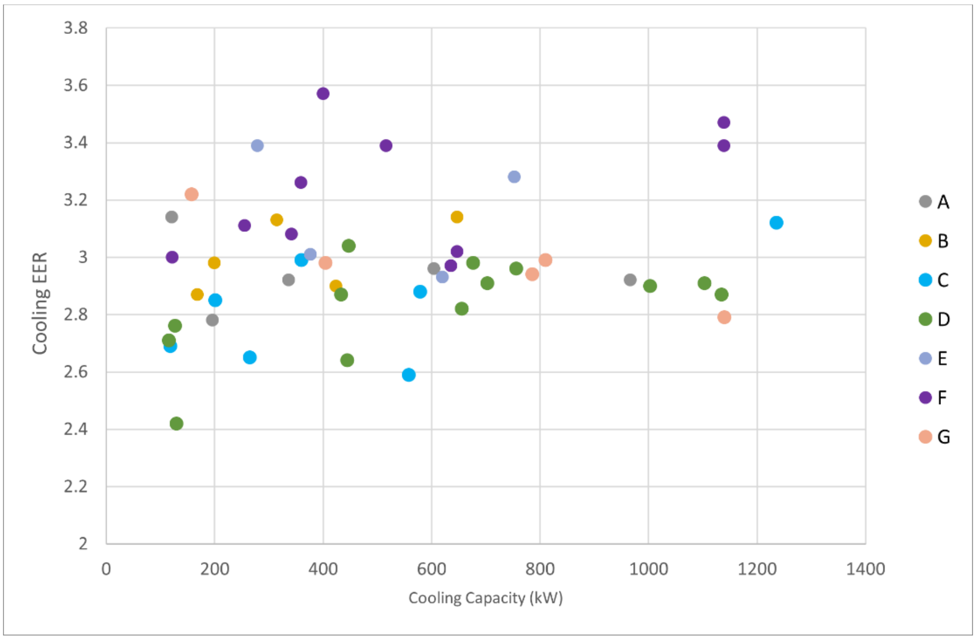


Figure 45: Cooling EER Colour Coded by Manufacturer

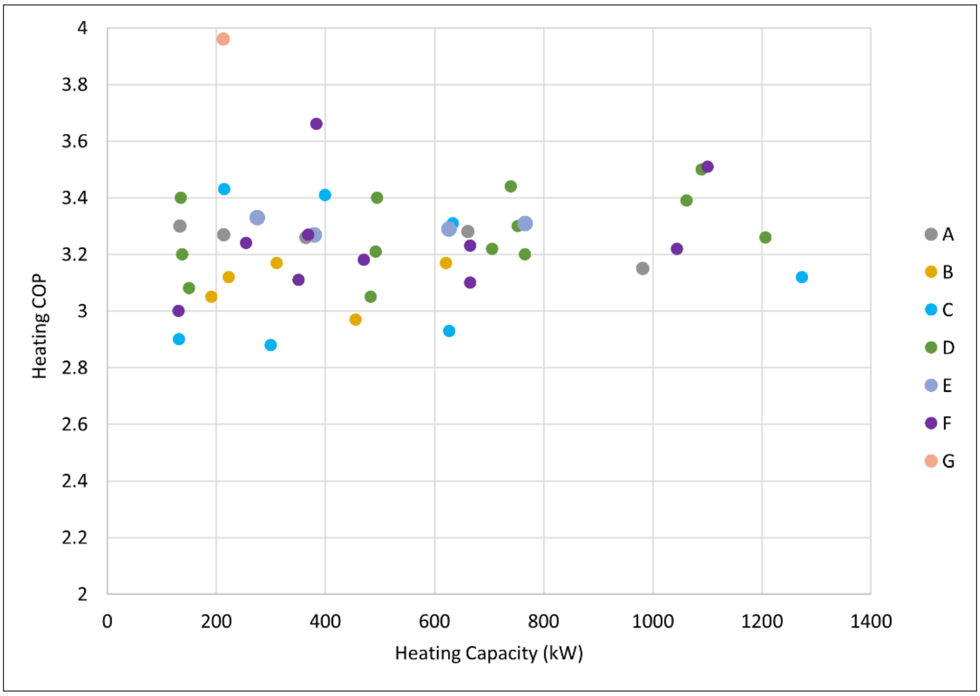


Figure 46: Heating COP Colour Coded by Manufacturer

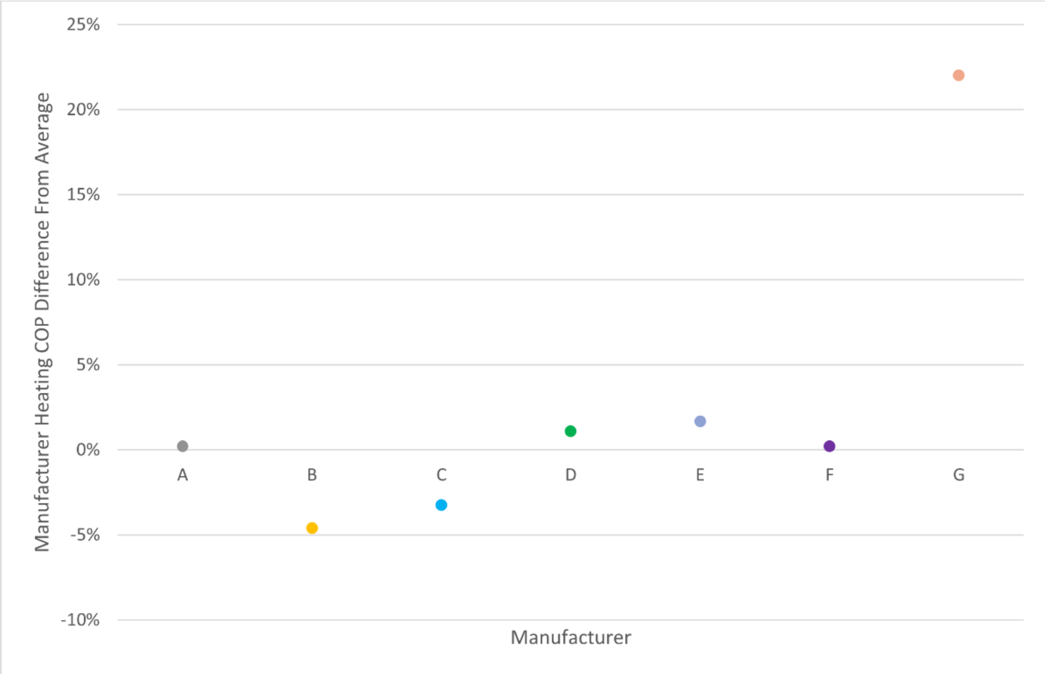


Figure 47: Average Heating COP for Each Manufacturer, Referenced to the Group Average

4.4 Discussion

4.4.1 Measurement standard - heating

With neither of the key performance measurement standards presenting the ideal solution for use in defining a measure in the NCC, it is proposed to generate a 'hybrid' solution.

EN 14511 is proposed in place of AHRI 551/591 primarily due to the following advantages:

- The rating conditions are more applicable to Australian conditions (standard rating at 7°C with options for other temperatures, rather than 8°C and -8°C).
    - While EN 14511 provides various ambient (airside) and waterside temperatures for performance ratings, Table 13 of the standard (extract provided in Figure 48) provides the conditions that will be applicable to this measure:
        - Heating mode (intermediate temperature) relates to waterside conditions of 40°C entering and 45°C leaving temperatures, and
        - Standard rating conditions (using ambient air as a heat source) relates to airside conditions of 7°C dry bulb and 6°C wet bulb (evaporator coil) entering conditions.
- Standard conditions are already used, readily available and published for the vast majority of heat pumps surveyed.
- Generating familiarity in the market with this standard enables the flexibility for extending to part load performance (SCOP) metrics in future versions of the NCC. Currently we have found no information to suggest AHRI are developing a method to rate heat pump performance at part load in heating mode.

Table 13 — Air-to-water(brine) units - Heating mode (Intermediate temperature)

		Outdoor heat exchanger		Indoor heat exchanger intermediate temperature application	
		Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Outdoor air	7	6	40	45
	Exhaust air	20	12	40	45
Application rating conditions	Outdoor air	2	1	a	45
	Outdoor air	−7	−8	a	45
	Outdoor air	−15	—	a	45
	Outdoor air	12	11	a	45

[a] The test is performed with the fixed flow rate or with the ΔT obtained during the test at the corresponding standard rating conditions for units with variable flow rate. If the resulting flow rate is below the minimum flow rate then this minimum is used with the outlet temperature.

Figure 48: Table 13 of EN 14511.2 - 2022 Specifying ambient and waterside temperature conditions for performance rating. Standard rating conditions are as outlined in red

4.4.2 Measurement standard - cooling

It is expected that the vast majority of air-sourced heat pump installations will make use of the equipment's ability to operate in cooling mode. As a result, performance in cooling must be considered as a component of this measure. To maintain a robust chiller stringency, it is proposed to fully integrate reversible heat pump cooling performance requirements into the broader chiller measure, which will require reference to the relevant measurement standards (AHRI 551/591 in NCC 2022).

4.5 Proposed Stringency

4.5.1 Heating

The cost/efficiency relationship for heating shows the very limited range of variation in heating COP figures, with most points within ±5% of average efficiency[12] (which generally holds true for the average performance for each manufacturer). This was tested by dividing the heat pump data into quartiles based on COP and conducting a t-test to establish whether the average % cost deviation relative to average was significant. Results are shown in Table 24, which shows that there is no significant relationship. As a result, no cost-benefit analysis has been conducted.

Table 24. Analysis of cost-efficiency relationship for heat pumps

Quartile	Count	Average COP	Average cost deviation	t-test
1	10	3.01	-4%	0.14
2	10	3.20	-6%	0.43
3	10	3.29	-15%	0.42
4	10	3.45	18%	0.19

[12] A t-test analysis was undertaken to establish whether any statistically significant differences in cost were present between low, medium and high efficiency thirds of the data sample; none were found. As a result, the simple average of available units is suitable as a stringency for a measure based on the average efficiency of installed units.

As with several other technologies, we propose that the stringency is set on the basis of the average efficiency of units installed in the building. This is relevant as most buildings with heat pumps installations will have more than one unit installed. In the absence of a significant cost/efficiency relationship, we propose that the stringency is set at the average heat pump full load COP, which is approximately[13] 3.25. Each manufacturer in the analysed data set has at least one product that is above this level; furthermore, the use of the average COP of installed units rather than imposition of a minimum efficiency for each installed unit means that no currently available product is excluded from the market.

4.5.2 Proposed stringency – cooling

The average EER and IPLV for heat pumps with heating COPs greater than 3.25 are 3.03 and 4.86 respectively. These figures will be used to inform the cooling stringency, as further discussed in Section 5.5.

4.6 Proposed Measures

As the requirements for heat pumps and 4 pipe chillers are substantively similar, it is proposed to integrate the measures for these two technologies into a single measure, which is listed in Section 5.7.1

[13] It is 3.24, but we have adjusted this figure to coordinate with later recommendations regarding 4 pipe chillers.

5 4 pipe Chillers

5.1 Background and context

For the purposes of this section, 4 pipe chillers are defined as water-to-water devices that can produce chilled water and hot water simultaneously. Such products are typically of moderate efficiency in heating-only or cooling-only operation but can produce high efficiencies when providing heating and cooling simultaneously.

4 pipe chiller efficiency is not regulated under NCC 2022, reflecting the fact that until recently this technology was rarely used in Australia. While 4 pipe chillers are a more common technology overseas, Australian buildings rarely operate with large simultaneous heating and cooling loads[14], limiting the potential application of the technology. Nonetheless with increasing emphasis on degasification of buildings, 4 pipe chillers have the potential to be applied in at least some projects in Australia[15]. As a result, it is desirable for 4 pipe chiller efficiency to be regulated under NCC 2025.

5.2 Efficiency measurement standards for 4 pipe chillers

The discussion on performance measurement standards for 4 pipe Chillers is largely equivalent to that of Heat Pumps in Section 4, and therefore not repeated here, but with an additional metric to enable measurement of performance in simultaneous heating and cooling mode:

- COP_{SHC} according to AHRI 551/591
- HRE (Heat Recovery Efficiency) according to EN 14511

However, unlike full load heating or cooling efficiency alone, it was found that neither US nor European jurisdictions mandate a minimum simultaneous heating and cooling efficiency for air-sourced 4 pipe chillers.

5.3 Methodology

The outline methodology for the 4 pipe chiller analysis is shown in Figure 49.

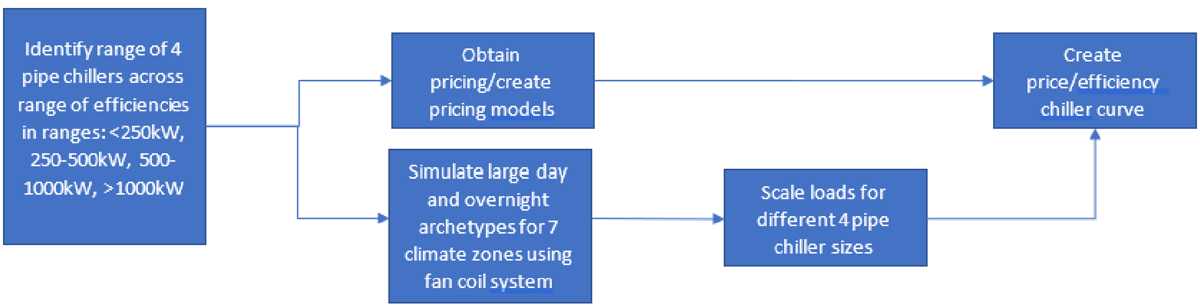


Figure 49. Outline methodology for the 4 pipe chiller analysis.

14 This to some extent reflects Australia's bias towards the use of economy cycles in large buildings. In many overseas markets fan coils are a preferred solution and as a result winter operation often involves simultaneous perimeter heating and centre zone cooling.

15 Examples of possible applications include sites without economy cycles (e.g. buildings using fan coils), with extensive north and south facing glass, with significant dehumidification loads or with non-seasonal process driven loads.

5.3.1 Data collection and review

Data was collected for 66 4 pipe chillers from 7 manufacturers.

The efficiency of 4 pipe chillers can be reported via multiple metrics, including:

1. Cooling EER (cooling thermal output divided by electrical input at full load design conditions)
2. Heating COP (heating thermal output divided by electrical input at full load design conditions)
3. Cooling IPLV (cooling IPLV as calculated based on AHRI)
4. Heating SCOP (seasonal COP – providing a similar purpose as IPLV, however calculated in a much more complex manner)
5. TER (Total Efficiency Ratio – simultaneous heating and cooling thermal output divided by electrical input at full load)

It was found that while the full load measures were available for all 66 models identified, the remaining efficiency metrics were not always available.

Based on the full load design figures, the full load design efficiency of available units is somewhat dependent on capacity, as shown in Figure 50 below.

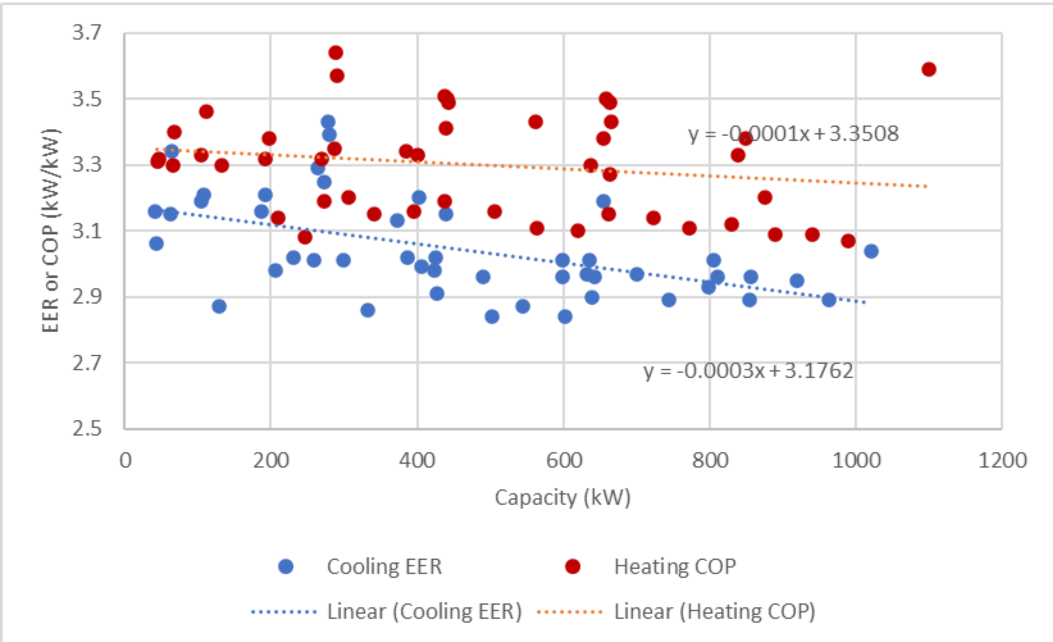


Figure 50: 4 pipe chiller efficiency data – Full load EER and COP

To the extent that part load data was available, SCOP and IPLV data also showed a relationship to capacity, but in the opposite direction to COP and EER, as shown in Figure 51. TER, on the other hand, showed no significant relationship to capacity as shown in Figure 52.

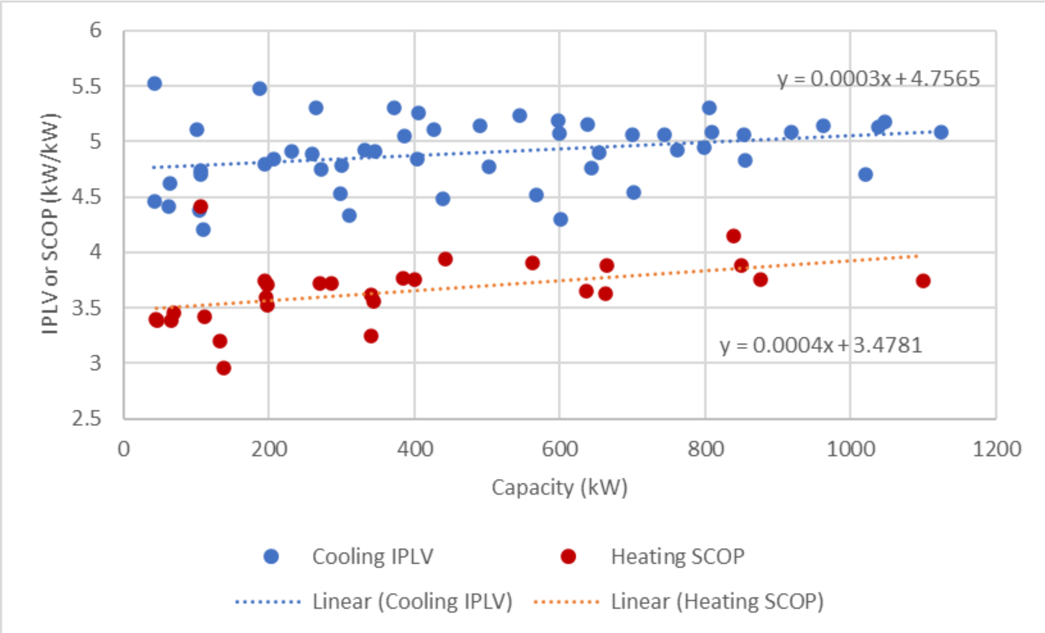


Figure 51: 4 pipe chiller efficiency data – Cooling IPLV and Heating SCOP

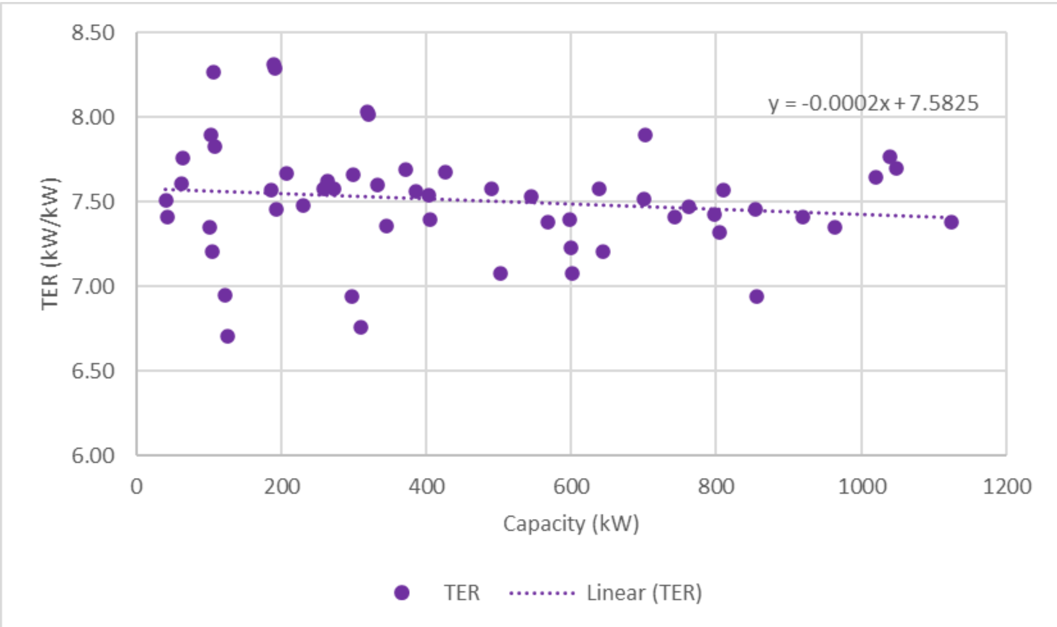


Figure 52: 4 pipe chiller efficiency data - TER

5.3.2 Cost/Efficiency Analysis

Pricing was obtained for 52 out of the 66 4 pipe chillers for which energy data was obtained. The data show a strong cost to capacity relationship, as would be expected.

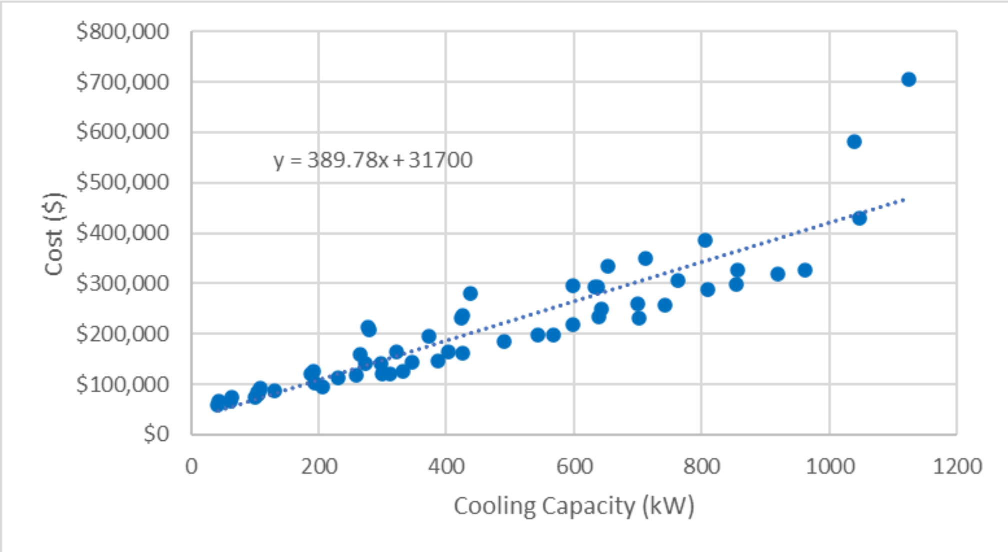


Figure 53: 4 pipe chiller cooling cost/capacity relationship

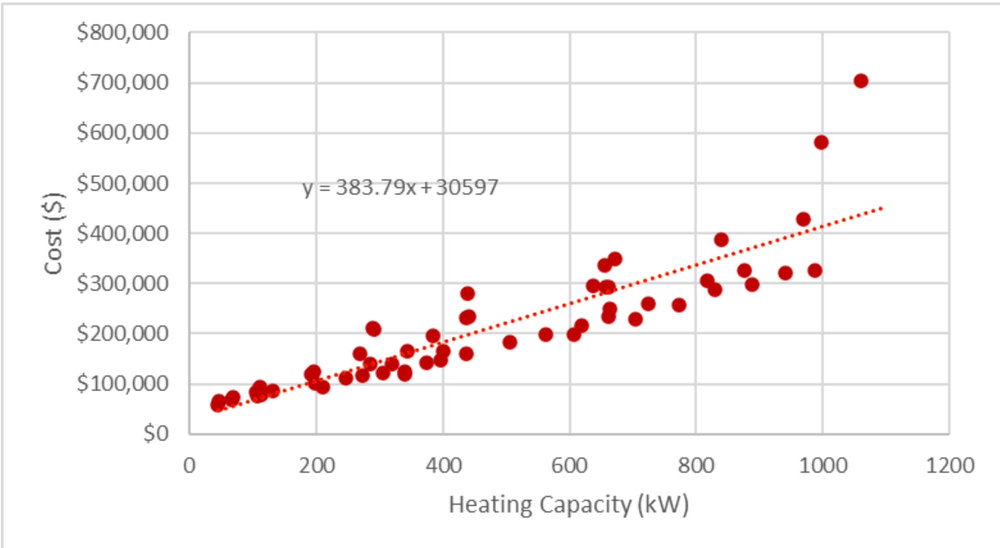


Figure 54: 4 pipe chiller cost/capacity relationship

In order to assess the cost/efficiency relationship, the following steps were undertaken for each chiller:

1. The % change in cost per kW capacity relative to the benchmark cost equations was calculated
2. The % deviation in efficiency indicator relative to the capacity-corrected average efficiency was calculated
3. The results from 1 and 2 were plotted to establish the cost/efficiency relationship.

The results are shown in Figure 55 and Figure 56 below.

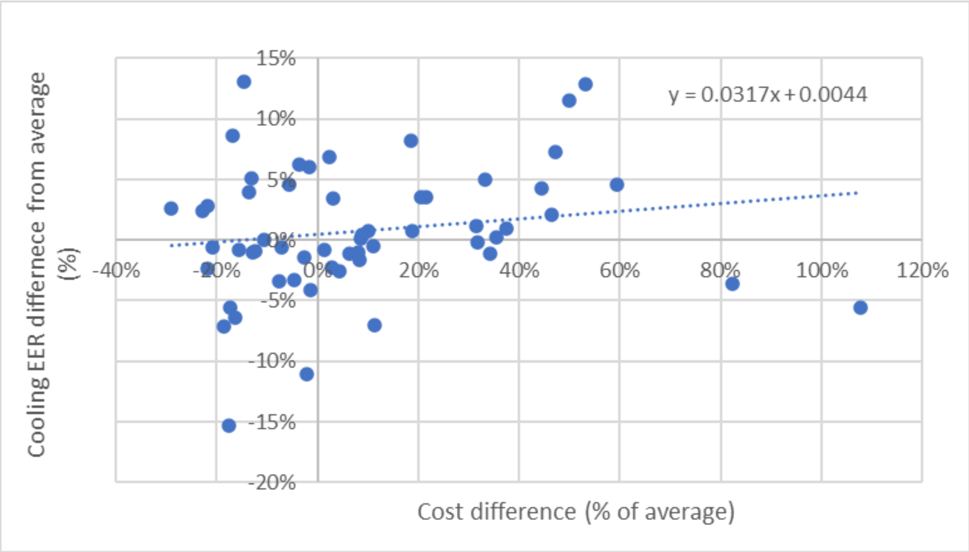


Figure 55: 4 pipe chiller cost/efficiency relationship – cooling EER

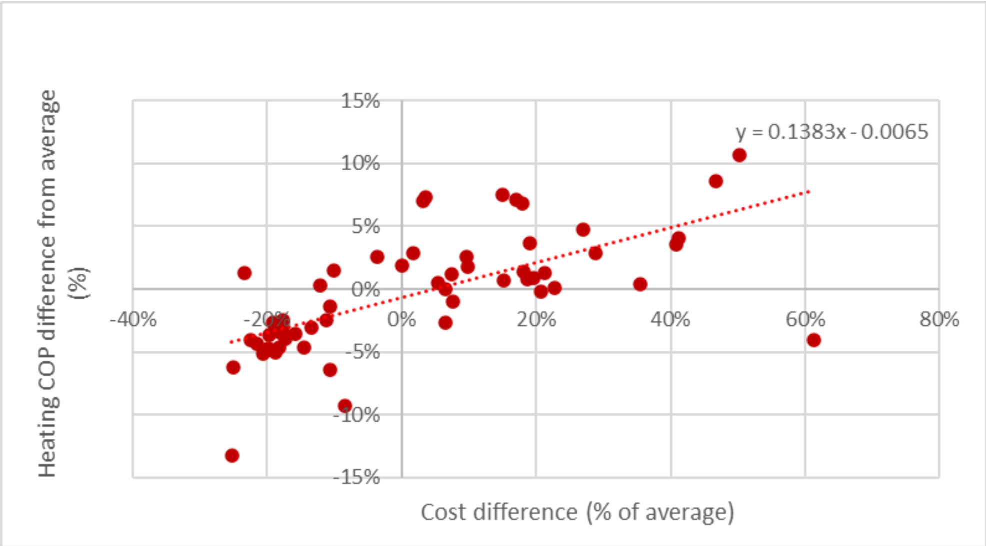


Figure 56: 4 pipe chiller cost/efficiency relationship - heating COP.

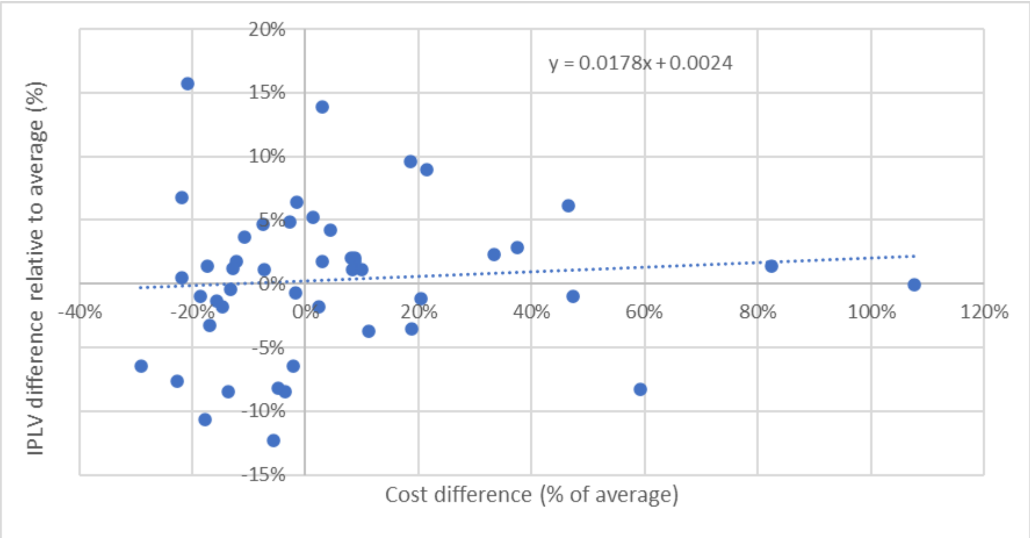


Figure 57: 4 pipe chiller cost/efficiency relationship - IPLV

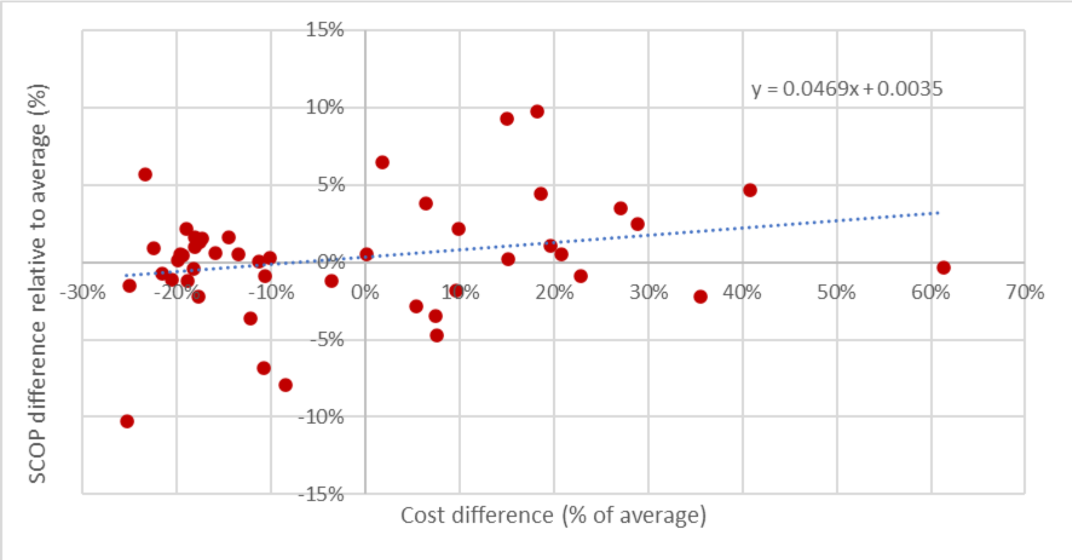


Figure 58: 4 pipe chiller cost/efficiency relationship – SCOP

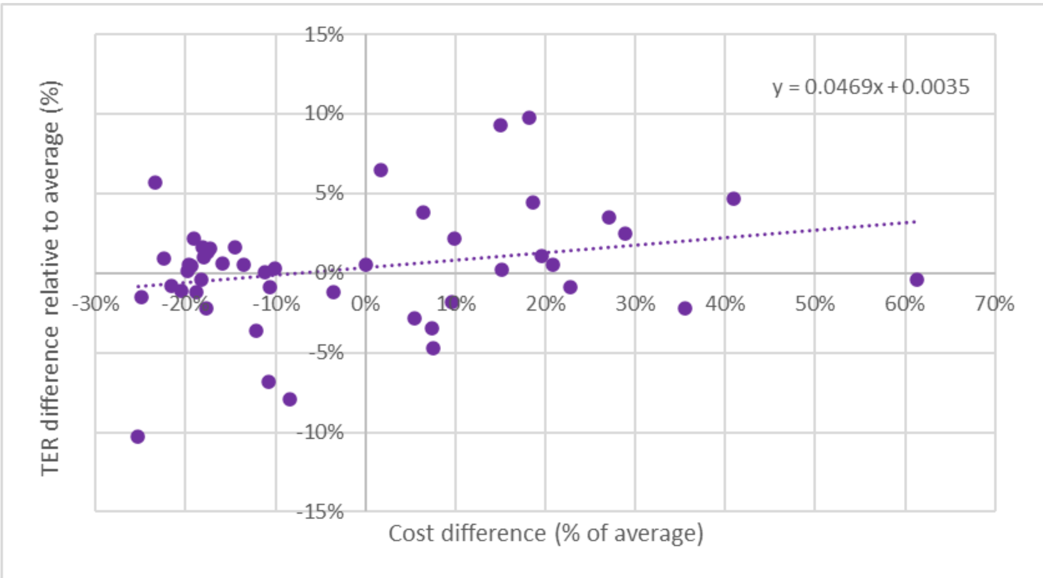


Figure 59. 4 pipe chiller cost/efficiency relationship – TER

5.3.3 TER/EER/COP relationship

While we found no mandatory minimum efficiency requirements for TER in the US and EU, further analyses were carried out to determine whether a standalone minimum TER performance is warranted. Heating COP and cooling EER were separately plotted against TER to determine the level of correlation.

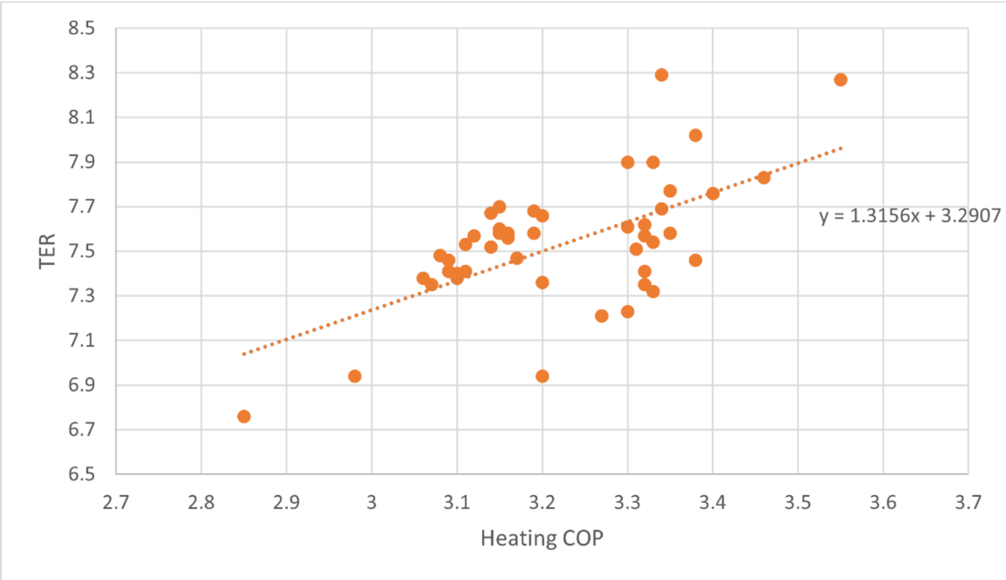


Figure 60: 4 pipe chiller heating COP/TER relationship

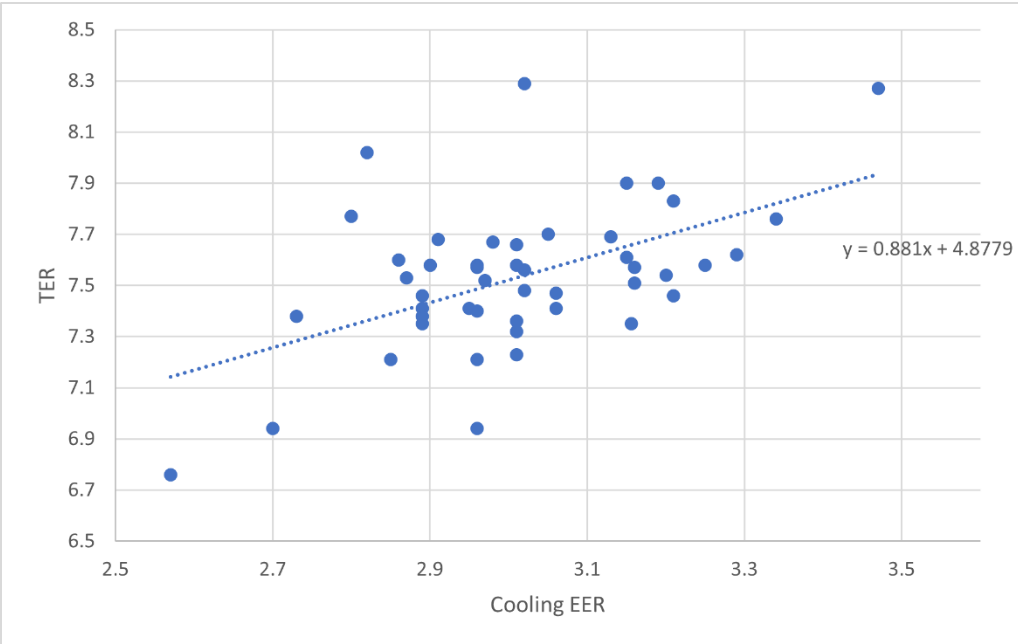


Figure 61: 4 pipe chiller cooling EER/TER relationship

5.3.4 Manufacturer-efficiency relationship

Given the analyses carried out above are leading to findings of either small variations or loose correlations with efficiency, a further analysis was carried out to determine whether efficiency is correlated to manufacturers. Efficiency in heating and cooling modes was plotted separately and identified by individual manufacturers.

Some relationships between manufacturer and efficiency emerges for both heating and cooling performance. Generally speaking, the following conclusions can be seen:

- Higher efficiency: Manufacturers B, C and D
- Lower efficiency: Manufacturer A
- Spread of efficiency: Manufacturer F

- No conclusion: Manufacturer E, as cooling EER is approximately average and heating COP is high

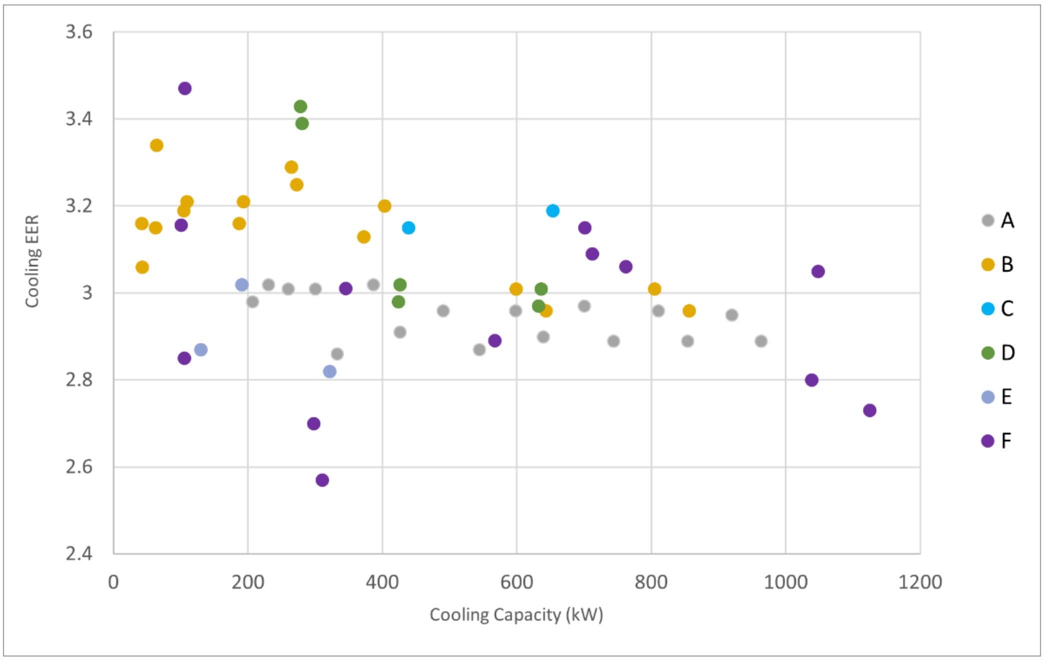


Figure 62: Cooling EER/capacity relationship colour coded by manufacturer

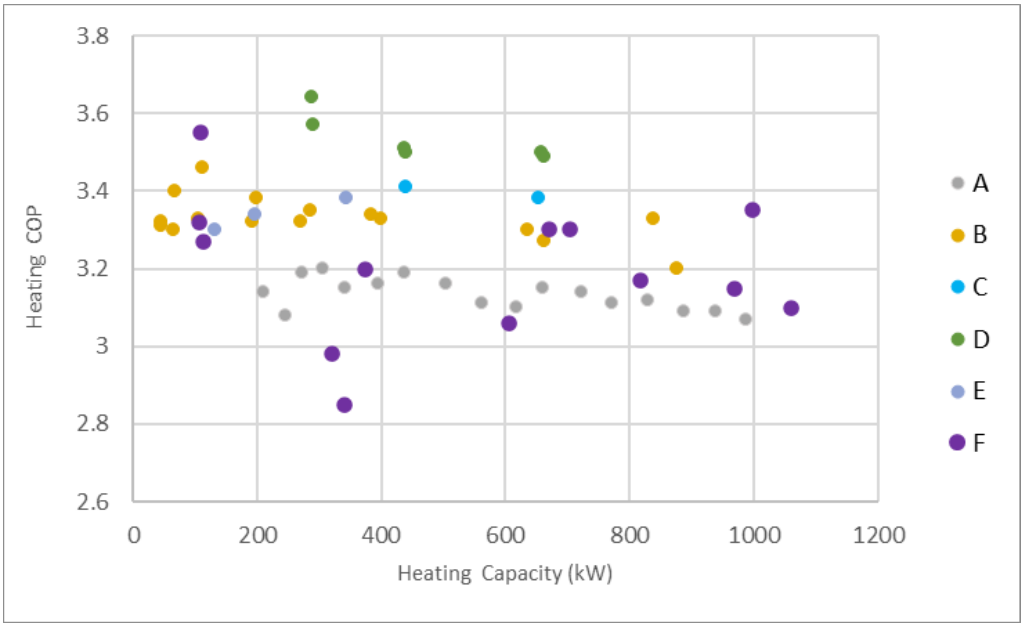


Figure 63: Heating COP/capacity relationship colour coded by manufacturer

5.4 Discussion

5.4.1 Proposed Stringency - heating

The cost/efficiency relationship for heating shows the very limited range of variation in heating COP figures, with most points within ±5% of average efficiency. This was tested by dividing the 4 pipe chiller data into quartiles based on COP and conducting a t-test to establish whether the average % cost deviation relative to average was significant. Results are shown in Table 25, which shows that there is no significant relationship. As a result, no cost-benefit analysis has been conducted.

Table 25. Analysis of cost-efficiency relationship for heat pumps

Quartile	Count	Average COP	Average cost deviation	t-test
1	14	3.07	-12%	0.30
2	14	3.20	-10%	0.28
3	14	3.32	11%	0.30
4	15	3.46	21%	0.17

As with several other technologies, we propose that the stringency is set on the basis of the average efficiency of units installed in the building. Taking the average full load efficiency for the entire collected data set for 4 pipe chiller performance in heating only mode results in a figure of 3.27; applied as an average for units in a building this does not create large exclusions in the market. Only one manufacturer of the six analysed did not offer a product for our assessment that meets the average efficiency.

Further, the strong relationship between TER and heating performance illustrates that a stringency imposed on average heating efficiency inherently removes the worst performing equipment on a TER basis. Applying a meaningful stringency on TER, in conjunction with heating and cooling efficiencies, would exclude a large proportion of market participation. Therefore, the average efficiency in heating mode (COP=3.27, modified to 3.25 to coordinate with heat pumps) is recommended to set the level of stringency in code.

5.4.2 Proposed stringency – cooling

The average EER and IPLV for 4 pipe chillers with a heating COP greater than 3.25 are 3.08 and 4.84 respectively. These figures are input into the discussion in Section 5.6.

5.5 Integration with heat pumps measure

The significant commonalities between heat pumps and 4 pipe chillers, as well as the nearly identical average efficiencies, mean that it makes sense to combine these technologies into a single measure.

The average heating COP for heat pumps was found to be 3.24, while the comparable figure for 4 pipe chillers was 3.27; these numbers are not significantly different. As a result, a common average performance of 3.25 is proposed.

For cooling, the average EER and IPLV for heating COP compliant heat pumps were 3.03 and 4.86 as compared to 3.08 and 4.84 for 4 pipe chillers. Based on these figures, cooling stringency is proposed to be set at an average EER of 3.0 and an average IPLV of 4.85.

5.6 Integration with the chillers measure

A challenge for both heat pumps and 4 pipe chillers is that they are likely to become common plant items that have significant cooling capacity (up to 100% of the required cooling capacity for the building) but a lower efficiency in cooling that is proposed for chillers, as demonstrated in

Table 26.

Table 26. Comparison of cooling EER and IPLV targets for heat pumps, 4 pipe chillers against air-cooled and water-cooled chillers.

	Heat pumps	4 pipe chillers	Air-cooled chillers	Water-cooled chillers
EER	3.03	3.08	3.25-3.48	4.95-6.4
IPLV	4.86	4.84	5.00-5.42	6.14-11.5

In an ideal world, one would fully integrate the heat pump/4 pipe chiller cooling efficiency requirements with the requirements chiller efficiency requirements, thereby ensuring that there is no loss in cooling efficiency arising from the use of heat pumps. IN this context, it is useful to compare the distribution of EER and IPLV figures for heat pumps, 4 pipe chillers and air-cooled chillers on the same chart, as shown in Figure 64 and Figure 65.

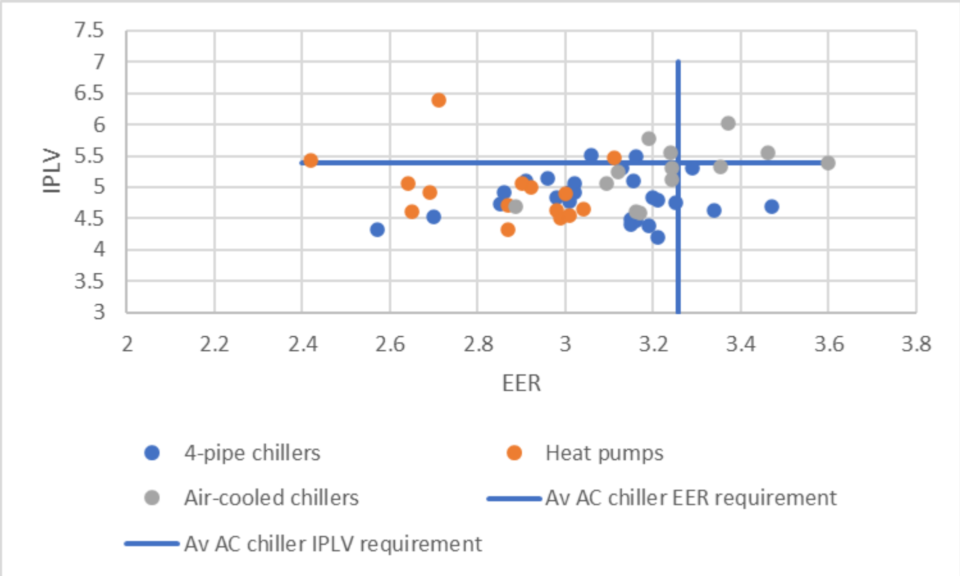


Figure 64. Comparison of 4 pipe chillers, heat pumps and air-cooled chillers deemed cost effective relative to base case – chillers under 528kW.

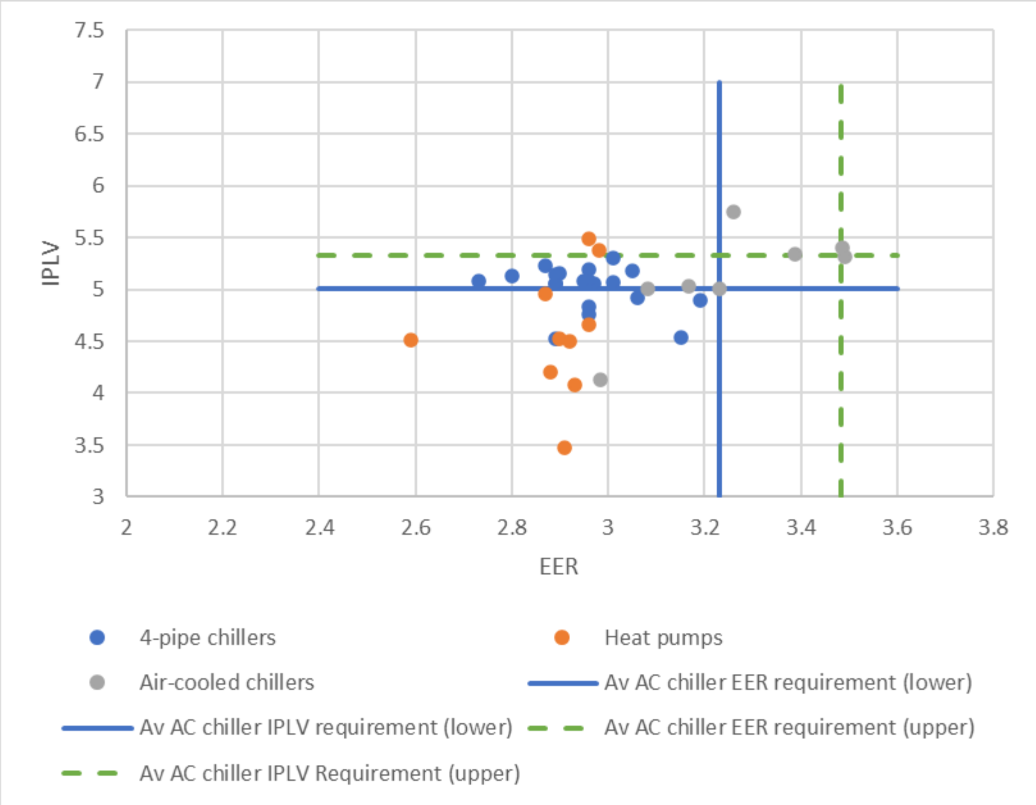


Figure 65. Comparison of 4 pipe chillers, heat pumps and air-cooled chillers deemed cost effective relative to base case – chillers over 528kW.

It can be seen that the inclusion of heat pumps and 4 pipe chillers would significantly constrain the ability of a designer to achieve the required average EER and IPLV figures. As a result, it has been decided not to integrate the cooling performance for heat pumps and 4 pipe chillers with that of air-cooled chillers.

5.7 Proposed Measures

5.7.1 Proposed Code Text

J5XX Heat pumps and 4 pipe Chillers – Coefficient of Performance

(1) Heat pump hot water heaters and 4 pipe chillers operated as part of air-conditioning systems in a building must -
   a) When operated in heating mode, have a capacity-weighted average heating coefficient of performance greater than 3.25 kW/kW when rated to EN 14511 (2022) requirements under the standard rating conditions defined in Table 13 of that Standard and evaluated at the design heating load of the application.
   b) When operated in cooling mode, have -
      i. a capacity-weighted average Energy Efficiency Ratio at full load of 3.0 cooling coefficient of performance greater than 3.0 kW/kW when rated to AHRI 551/591; and
      ii. a capacity-weighted average Integrated Part Load Value of 4.85 kW/kW when rated to AHRI 551/591.

6 Dewpoint Cooler and Indirect Evaporative Cooling

6.1 Background and context

NCC 2022 Section J does not reference the potential to use any form of evaporative cooling. The purpose of this analysis is to test whether this is justified.

Evaporative cooling comes in two basic types:

1. **Direct evaporative cooling.** For direct evaporative cooling, water is vapourised directly into the outside air stream entering a building. The evaporation of the water reduces the dry-bulb temperature of the air, providing a degree of cooling but at the expense of increasing humidity. Direct evaporative cooling is used occasionally in industrial settings and in some residential applications. However, it is ill-suited to broader commercial applications because of the amount of humidity introduced to the building. As a result, it is not considered further in this analysis.
2. **Indirect evaporative cooling.** Indirect evaporative cooling uses water vapourised into an airstream that interacts via heat exchanger with the outside air supply entering the building. This cools the air down without introducing humidity. This comes in two forms:
    a. **Conventional indirect evaporative cooling** uses a heat exchanger between the supply and exhaust airflows for the building, with water being vapourised in the exhaust/relief air stream before it reaches the heat exchanger, cooling the inlet air entering the building via the outside air stream. This has the added benefit of reducing the outside air intake's dry bulb temperature to a degree that is lower than ambient summer design conditions in climate zones 1-7.
    b. **Dewpoint coolers**, on the other hand, require a supply volume of outside air that is roughly twice the amount needed for ventilation to the building. This outside air intake volume is split into two, with half the volume treated with vapourised water. The two air streams are then passed through a common heat exchanger, allowing the water-treated airflow to cool the inlet air to the building. This heat exchange process is repeated multiple times.
    Dewpoint coolers are beneficial because the outside air is subject to an iterative heat exchange process instead of a once-off heat exchange process that conventional indirect evaporative cooling allows. This iterative process enables the outside air stream to be cooled below the wet bulb temperature (the theoretical limit to the evaporative process) and approach dewpoint.

Both indirect evaporative cooling methods described above are fully compatible with conventional air-conditioning system operation because they do not introduce additional humidity. In fact, research by Bannister et al[16] identified reductions in chiller energy from 10% to 100% dependent on climate zone using various combinations of indirect evaporative cooling technology.

[16] "Potential Impact of Evaporative Cooling Technologies on Australian Office Buildings" P Bannister, H Zhang, S White, Ecolibrium (Pub: AIRAH) May 2021.

6.2 Methodology

The methodology for the dewpoint cooler and Indirect evaporative cooling analysis is shown in Figure 66.

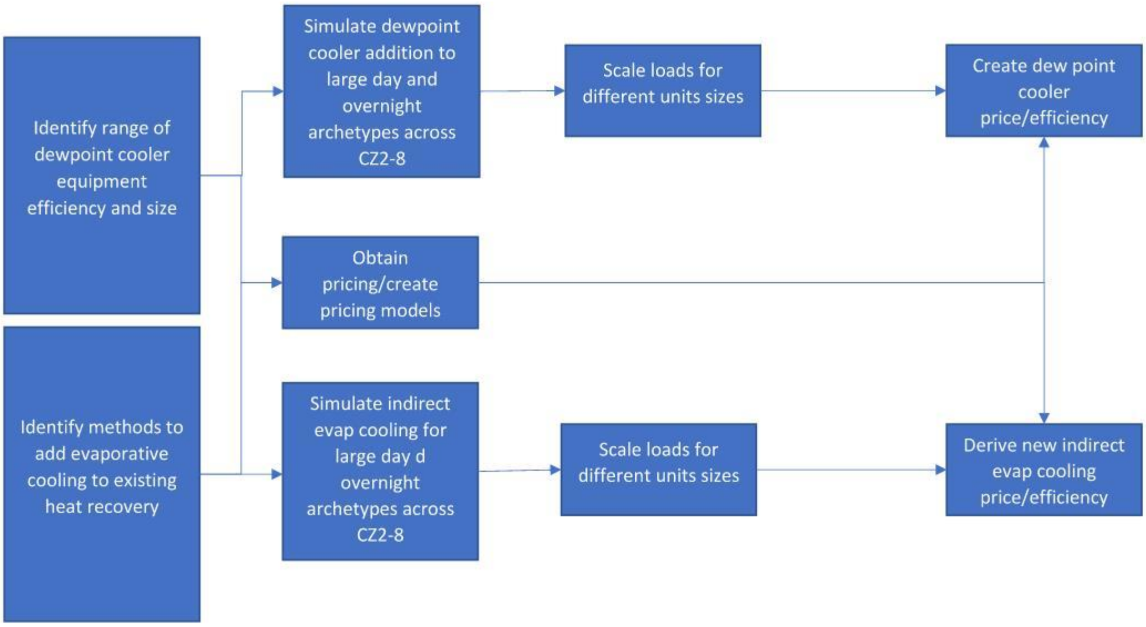


Figure 66. Outline methodology for the dewpoint cooler and indirect evaporative cooling analysis

6.2.1 Archetypes Tested

C5OL archetype

For the C5OL (Large Office) archetype, the archetype building has 5 variable air volume air conditioning systems with central plant, each serving all floors for a single façade aspect (north, east, west, south) or the centre zone. This represents what is generally considered to be best practice.

C9A archetype

For the C9A (Large Hospital) archetype, the modelled building is served outside air via two Air Handling Units (AHUs): one that serves corridors only, and the other serves as a Dedicated Outside Air System (DOAS) to FCUs distributed across the floorplate of each ward floor.

The minimum outside air quantities for each AHU are presented in Table 27.

Table 27: Minimum outside air quantities for tested archetypes

Archetype	AHU	Minimum Outside Air Quantity (L/s)
C5OL	Perimeter west	848
C5OL	Perimeter east	848
C5OL	Perimeter north	848
C5OL	Perimeter south	848
C5OL	Centre zone	4,889
C9A	DOAS	2,700
C9A	Corridor	2,110

6.2.2 Base and test cases

NCC 2022 Section J6D4 currently requires inclusion of demand-controlled[17] ventilation or heat recovery, depending on climate zone and the minimum outside air required for the HVAC system.

Table J6D4: Required outdoor air treatment

Climate zone	*Outdoor air flow (L/s)*	*Required measure*
1	>500	Modulating control
2	Not applicable	No *required* measure
3	>1000	Modulating control
4 and 6	>500	Modulating control or energy reclaiming system
5	>1000	Modulating control or energy reclaiming system
7 and 8	>250	Modulating control or energy reclaiming system

Figure 67: NCC 2022 Section J extract: requirements relating to minimum outside air provision

NCC 2022 provisions set the requirements for the base case for each archetype. The feasibility of dew point coolers is tested using the C5OL archetype only; whereas indirect evaporative coolers are tested using both C5OL and C9A archetypes. These are summarised below:

- C5OL dewpoint cooler:
  - Base case with five AHUs – NCC 2022 compliance
  - Five AHUs in the base case served by two dewpoint coolers: one for the centre zone and one for the four perimeter zones as a group
- C5OL indirect evaporative cooler:
  - Base case with five AHUs – NCC 2022 compliance
  - Five AHUs in the base case served individually by dedicated pre-conditioner air-to-air heat exchangers
  - Five AHUs in the heat exchanger case served by indirect evaporative cooling
- C9A indirect evaporative cooler:
  - Base case with 2 AHUs – NCC 2022 compliance
  - Two AHUs in the base case served individually by dedicated pre-conditioner air-to-air heat exchangers
  - Two AHUs in the heat exchanger case served by indirect evaporative cooling

The dewpoint coolers were modelled for the C5OL archetype initially to establish a benchmark to compare against the indirect evaporative cooler performance. This was chosen in place of the C9A archetype due to the relative incompatibility of the available dewpoint cooler capacities (each of the two C9A Dedicated Outside Air Systems would require either two small dewpoint coolers each, or one very oversized unit each), which would ultimately result in an expensive installation. If the indirect evaporative results show sufficient benefits, the dewpoint cooler modelling would then be extended to the C9A archetype.

The research questions that we sought to answer using the test cases above are as follows:

[17] Achieved via modulation of minimum outside air from the maximum requirement in response to measured CO_2 levels within a building.

1. Is it cost effective to apply indirect evaporative cooling to a test case energy reclaiming system which complies with NCC 2022?

Table 28: Question 1 Base Case Comparisons

	C5OL	C5OL	C9A
Climate Zone	**Centre zone**	**Perimeter**	**Corridor or FCU DOAS**
4	Heat Recovery Ventilation (HRV)/ Heat exchanger (HX)	HRV/HX	HRV/HX
5	HRV/HX	Constant supply airflow	HRV/HX
6	HRV/HX	HRV/HX	HRV/HX
7	HRV/HX	HRV/HX	HRV/HX
8	HRV/HX	HRV/HX	HRV/HX

2. Is it cost effective to add an energy recovery system or indirect evaporative cooling where it is not required under NCC 2022?

Table 29: Question 2 Base Case Comparisons

	C5OL	C5OL	C9A
Climate Zone	**Centre zone**	**Corridor or FCU DOAS**	**Corridor or FCU DOAS**
1	N/A (tested in Question 3)	N/A (tested in Question 3)	Constant supply air flow
2	Constant supply air flow	Constant supply air flow	Constant supply air flow
3	N/A (tested in Question 3)	N/A (tested in Question 3)	Constant supply air flow

3. Is it cost effective to apply an energy recovery system or indirect evaporative cooling in lieu of modulating control?

Table 30: Question 3 Base Case Comparisons

	C5OL	C5OL	C9A
Climate Zone	**Centre zone**	**Corridor or FCU DOAS**	**Corridor or FCU DOAS**
1	Modulating Control (CO_2)	Constant supply airflow	N/A (constant supply air flow)
2	N/A (constant supply air flow)	N/A (constant supply air flow)	N/A (constant supply air flow)
3	Modulating Control (CO_2)	Modulating Control (CO_2)	N/A (constant supply air flow)

6.2.3 Data collection and Review

Dewpoint Coolers

Initial research showed several suppliers offering dewpoint coolers to the Australian market on a commercial scale[18]. However, when approached to provide performance and cost data, it became apparent that only one supplier of dewpoint coolers remained for Australia.

While this supplier offers seven products under the title of indirect evaporative cooler, four of these options involve a direct evaporative cooling stage following the initial indirect stage. As noted in Section 6.1, direct evaporative cooling must be avoided because it introduces humidity to the building. Therefore, the number of applicable products from this supplier was reduced to only three. The nominal performance of these units is summarised in Table 67 in the Appendices under Section 16.1. Table 67 also showcases the nominal performance of a unit with reduced airflow, illustrating the equipment's ability to modulate airflow.

While there exists a large gap in airflow quantities for the three products, the larger two products can modulate and reduce their airflow to suit the duty. The various components of the unit (for example, exhaust and supply airflow streams, dampers and water pumps) modulate in response and achieve very similar performance to the rated/nominal conditions. The modelled arrangements are presented in Table 31.

Table 31: Dewpoint cooler modelling parameters

Archetype	AHU	Minimum Outside Air Quantity (L/s)	Dewpoint Cooler Selection	Adjusted Fan Power Input (kW)
C5OL	Perimeter west	848	Single unit (8,281 L/s)	8.2 kW
C5OL	Perimeter east	848	Single unit (8,281 L/s)	8.2 kW
C5OL	Perimeter north	848	Single unit (8,281 L/s)	8.2 kW
C5OL	Perimeter south	848	Single unit (8,281 L/s)	8.2 kW
C5OL	Centre zone	4,889	Single unit (8,281 L/s)	8.2 kW

Indirect Evaporative Cooling

Indirect evaporative cooling options in this report are tested by adding a direct evaporative pad to the entering exhaust air stream of a traditional Heat Recovery Ventilator (HRV).

Data collected for evaporative pads were used as inputs to size the HRV unit selections. As HRVs and evaporative cooler components are more common than dewpoint coolers (which are effectively a packaged product), more market data was available. Data availability is summarised below:

- Evaporative pads - two suppliers.
- Heat recovery ventilators - three suppliers. However, one declined to provide data, so only data for two suppliers was gathered.

[18] i.e., offering airflow duties in the thousands of L/s rather than hundreds, which would cater more toward residential or very light commercial applications

Evaporative pads

Initially, we aimed to gather data for typical outside air flow rates for heat recovery units in commercial designs. However, it soon became apparent that the minimum module sizes available for evaporative pads were much larger compared to the typical sizes of heat recovery ventilation systems[19]. Therefore, equipment performance at discrete duty points had to be approximated to suit specific modelling inputs for simulation. This adjustment was deemed acceptable because the same rule applies for evaporative pads as with dewpoint coolers - as airflow is reduced, the cooling performance remains consistent.

Evaporative pads in the market are available with various physical properties[20], which ultimately result in different humidification efficiencies. Given this analysis aims to achieve results that prioritise the lowest dry bulb temperature[21], we limited analysis of our gathered data to models with the highest humidification efficiency (95%). Individual product data is available on file but not presented here for confidentiality reasons.

A simplified average value of 17.4°C dry bulb leaving air temperature in the relief/exhaust air stream was also retained in the modelling for energy simulation. The original performance data shows very limited variation in leaving air temperatures between the various evaporative pad models.

A nominal pressure drop of 60 Pa was retained for modelling inputs. As the evaporative pads are modular, and the pressure drops can be seen to somewhat plateau (on average) at airflows above 1,500 L/s.

Table 32: Average Evaporative Pad Performance

Model	Airflow	Entering Air (°CDB/°CWB)	Leaving Air (°CDB/°CWB)	Stated Nominal Humidification Efficiency (%)	Airside Pressure Drop (Pa)
Average	1,600	24 / 17.1	17.4 / NA	95	50.88
Average	3,200	24 / 17.1	17.4 / NA	95	56.32

HRVs

HRV fan powers for the various airflows were available for various HRV make and models. The two manufacturers provided data with different levels of granularity and customisability for each unit. Data for these units is presented in the following tables in coordination with this level of detail. For reference:

- "Series 1" unit selections were highly customisable, with heat exchanger selections and external static pressure adjustments being possible to suit the archetype applications:
  - 20 Pa External Static pressure (ESP) is used to simulate the system pressure drop of each HRV airstream (external of evaporative pad).
  - An additional 60Pa ESP is applied to the exhaust air stream to account for the evaporative pad pressure drop.

[19] for example, 250-500L/s, as per typical in-ceiling HRV applications
[20] Typically, depths and material type varied
[21] Noting that the increased relative humidity is irrelevant, because this air does not transfer moisture into the building

- "Series 2" unit performance was only available at nominal conditions.
    - Please note the sensible heat exchange effectiveness presented in Table 34 is a nominal figure provided by the HRV supplier.
- "Series 3" unit performance was not available, but pricing data was provided (this is presented in Section 6.3.2).

Table 33: "Series 1" heat recovery unit selection data

Model	Supply Airflow (L/s)	Exhaust Airflow (L/s)	Sensible Heat Exchange Effectiveness	Supply Fan Power (kW) @ 20 Pa ESP	Exhaust Fan Power (kW) @ 20 Pa ESP	Exhaust Fan Power (kW) @ 80 Pa ESP
Series 1A	250	200	70%	0.07	0.05	0.07
Series 1B	500	400	65%	0.27	0.17	0.2
Series 1C	850	680	67%	0.53	0.31	0.37
Series 1D	2,110	1,690	67%	0.98	0.55	0.68
Series 1E	4,900	3,920	67%	3.79	2.12	2.45

Table 34: "Series 2" heat recovery unit selection data

Model	Nominal Supply and Exhaust Airflow (L/s)	Sensible Heat Exchange Effectiveness	Total Connected Power (kW)	Total Fan Absorbed Power (kW) @ 250 Pa ESP
Series 2A	250	75% (nominal)	1.5	0.15
Series 2B	550	75% (nominal)	5	0.48
Series 2C	2,000	75% (nominal)	10	1.98
Series 2D	4,000	75% (nominal)	15	3.63

Please note, simulations use a heat exchange effectiveness of 60%. This is because the modelling of the indirect evaporative cooling measure is predicated by an outside air treatment system compliant with NCC 2022[22].

6.3 Construction costs

6.3.1 Dewpoint Coolers

Unit supply and general installation costs were gathered and retained on file (not included here for confidentiality reasons). To align with the C5OL archetype, the installation costs include only ducting required to integrate with an AHU(s).

6.3.2 Conventional Indirect Evaporative Cooling

Costs for the "Series 1", "Series 2" and "Series 3" units were gathered, along with the evaporative pads and installation for each equipment. The results are presented in the following tables, with the heat recovery unit only and indirect evaporative cooling arrangement supply and installation costs provided in Table 35 and Table 36.

[22] For context, see Research Question 1 presented in Section 6.2.3 above

Table 35: Supply and installation costs for heat recovery units (HX)

Equipment Nominal Airflow (L/s)	Installation Cost	HX Equipment Cost - Series 2	HX Equipment Cost - Series 3	HX Equipment Cost - Series 1	HX Equipment Cost - Average	Average Heat recovery Unit Equipment Supply & Install Cost
500	$15,959	$11,520	$10,200	$20,100	$13,940	$29,899
2,000	$22,566	$23,350	$21,600	$44,568	$29,839	$52,405
4,000	$25,794	$37,015	$35,400	$65,136	$45,850	$71,644

Table 36: Supply and installation costs for evaporative pads

Equipment Nominal Airflow (L/s)	Installation Cost	Average Evaporative Pad Equipment Cost	Average Evaporative Pad Supply & Install Cost
500	$8,521	$7,050	$15,571
2,000	$16,269	$11,915	$28,184
4,000	$17,054	$13,925	$30,979

Table 37: Supply and Installation Costs for heat recovery unit supply and installed complete with indirect evaporative cooling

Equipment Nominal Airflow (L/s)	Average Heat recovery Unit Complete with Evaporative Pad Supply & Install Cost
500	$45,470
2,000	$80,589
4,000	$102,623

For comparison against base cases, costs for CO_2 control were also gathered and compiled on a "per CO_2 sensing point" basis, with total costs for the archetype as follows:

Table 38: Costs compiled for CO_2 control

	C5OL
CO_2 Sensor Cost	$1,200
Sensor Install Cost	$1,300
Sensors per Floor	2-3, depending on floor layout
Number of Floors	25
Total Cost	$62,500

For both dewpoint and indirect evaporative cooling equipment, additional maintenance costs are expected to be sustained as an operational expenditure. The complexity and tasks (and therefore costs) involved with maintaining this equipment is similar to an AHU of equivalent airflow capacity. The analysis presented later in this section is compiled initially to exclude these additional costs – maintenance costs are added if BCR results show sufficient benefit and the requirement for further detailed costing analysis.

6.3.3 Plant Capital Cost Savings

For plant capital cost savings calculations associated with any expected peak demand reduction, the data gathered in Section 2 (Chillers) was used and applied according to the technology used for each archetype. The cost summary and calculations are as follows:

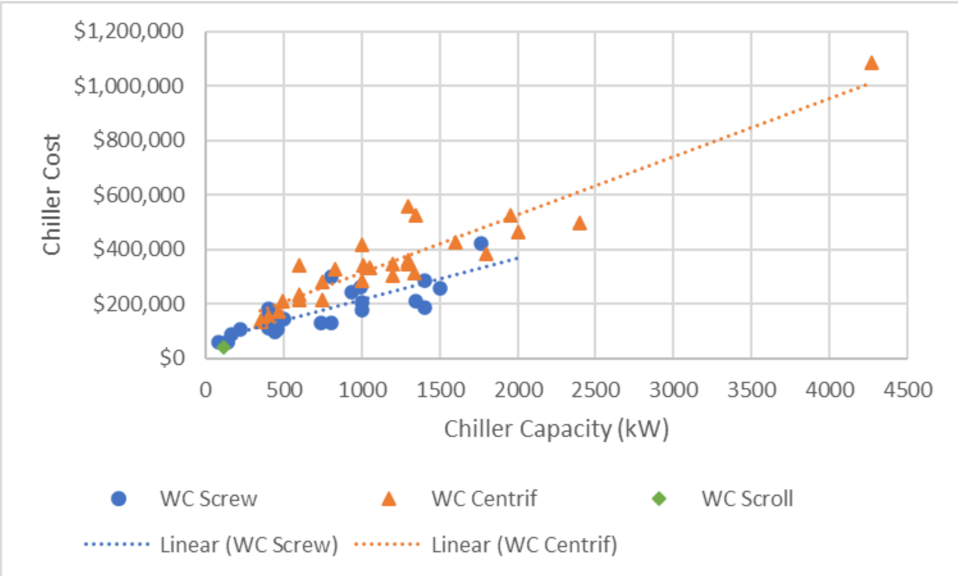


Figure 68. Water-cooled chiller costs

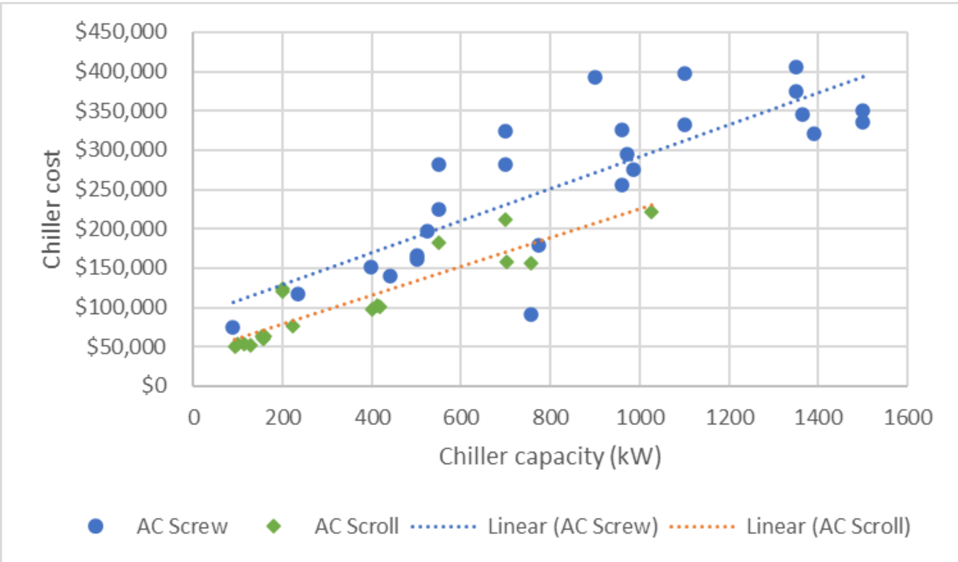


Figure 69. Air-cooled chiller costs.

Table 39: Chiller cost calculations

Archetype / Chiller Technology	Cost Calculation
C5OL (CZ 1-7) / Water Cooled Screw	y = 152.5x + 612929
C5OL (CZ 8) / Air Cooled Screw	y = 203.2x + 88465
C9A / Air Cooled Screw	y = 203.2x + 88465

6.4 Results

6.4.1 Simulation Results – Dewpoint Coolers

Simulations were carried out for the C5OL archetype to firstly test in a cross section of climate zones providing the widest range of dry bulb conditions. As noted in Section 6.2.3, due to the limitation of unit capacities, one large unit was applied to cover pre-treatment as a common system to all five AHUs. It is expected that installation of a dedicated dewpoint cooler to serve each AHU would be cost prohibitive and not representative or a real-world installation. The control applied to the dewpoint cooler cases is as follows:

- Dewpoint cooler is only used in cooling mode, otherwise the minimum outside air will be bypassed. To be consistent with the AHU cooling supply air temperature, the dewpoint cooler operates when average zone temperature is over 22°C for centre zones and high-select zone temperature is over 22°C for perimeter zones.

The results initially showed that the dewpoint cooler was operating at times when ambient conditions resulted in limited cooling effect, which limits the benefits of the dewpoint cooler. A final set of simulations were modelled to explore the effects of limiting operation to times when ambient relative humidity was below 55%, improving the cooling benefit. The results are illustrated in Figure 70, with results described by the following:

- Base, CO_2 – Base case scenario testing with CO_2 modulation on outside air.
- DC – Dewpoint cooler operating via the basic control strategy.
- DC-RH55 – Dewpoint cooler operating via the basic control strategy with the added condition that it operates only when ambient relative humidity is below 55%.

The simulation results are discussed in Section 6.5 below.

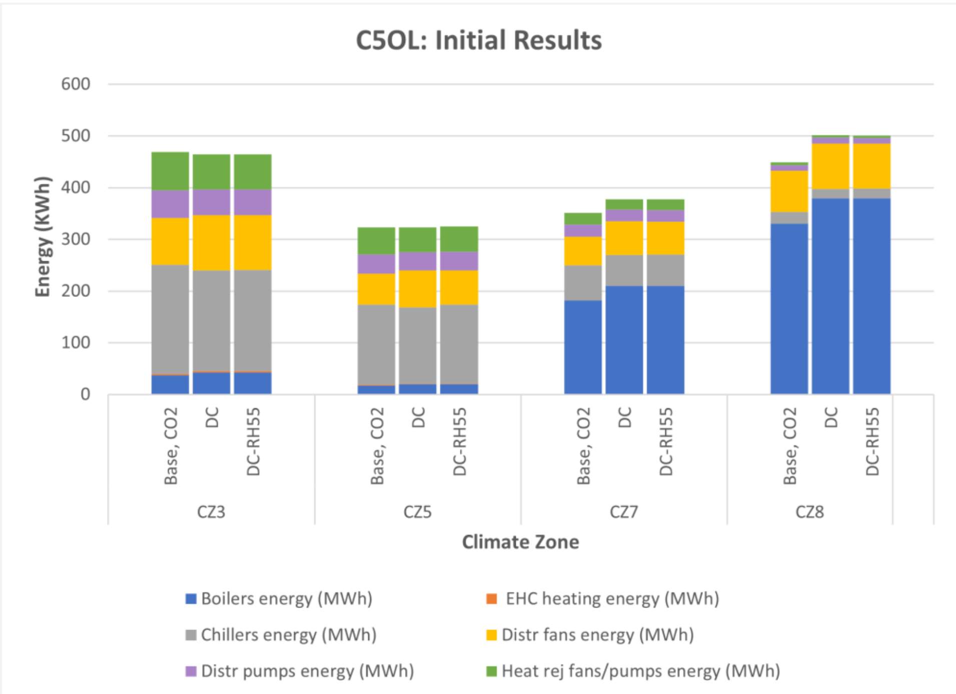


Figure 70: Dewpoint cooler simulation results

6.4.2 Simulation Results – Conventional Indirect Evaporative Coolers

Initial simulations

Simulations were initially carried out for the C5OL archetype as preliminary tests. Heat recovery and indirect evaporative cooling (without modulating CO_2 control) were applied as minimum outside air pre-treatment to NCC 2022 compliant models (including CO_2 control) for the minimum outside air component of the central air handling system. The pre-treatment was initially set to operate under the following conditions:

- When heating was required, pre-treatment is active when the outside air conditions were 2°C below the building return/relief air temperature.
- When cooling was required, pre-treatment is active when the outside air conditions were 2°C above the building return/relief air temperature. For the indirect evaporative cooling cases, the building return/relief air condition was calculated assuming evaporative pad pre-cooling operation is active.

Compared to the CO_2 control cases, the heat recovery and indirect evaporative cooling simulations resulted on an overall increase in energy consumption.

Exploration of potential improvements to the test cases

To explore potential improvements, high level test cases were assessed by comparing approximate chiller COPs with the performance benefit of the heat recovery unit. That is, the pre-requisite to operating the indirect evaporative cooling system is for sufficient heat (sic. *minimum benefit*) removed from the outside air stream, to compensate for the increased energy penalty[23] to operate the indirect evaporative cooler. Increasing the cooling differential temperature signal from 2°C to 8°C was determined to be the minimum benefit.

The results for this revised control strategy were simulated across the test cases. The results are presented in Figure 71. The initial simulation cases were carried out across climate zones 3, 5, 7 and 8, with simulation labels described as follows:

- Base, no CO_2 – (C5OL only) Base case scenario testing without CO_2 modulation on outside air.
- Base, CO_2 – (C5OL only) Base case scenario testing with CO_2 modulation on outside air.
- HX 2ΔT – Heat recovery unit scenario using 2°C differential on both heating and cooling requirements.
- IE 2ΔT – Indirect evaporative cooling scenario using 2°C differential on both heating and cooling requirements.
- HX 8ΔT – Heat recovery unit scenario using 2°C differential on heating and 8°C differential on cooling requirements.
- IE 8ΔT – Indirect evaporative cooling scenario using 2°C differential on heating and 8°C differential on cooling requirements.

[23] due to the internal resistance of the equipment

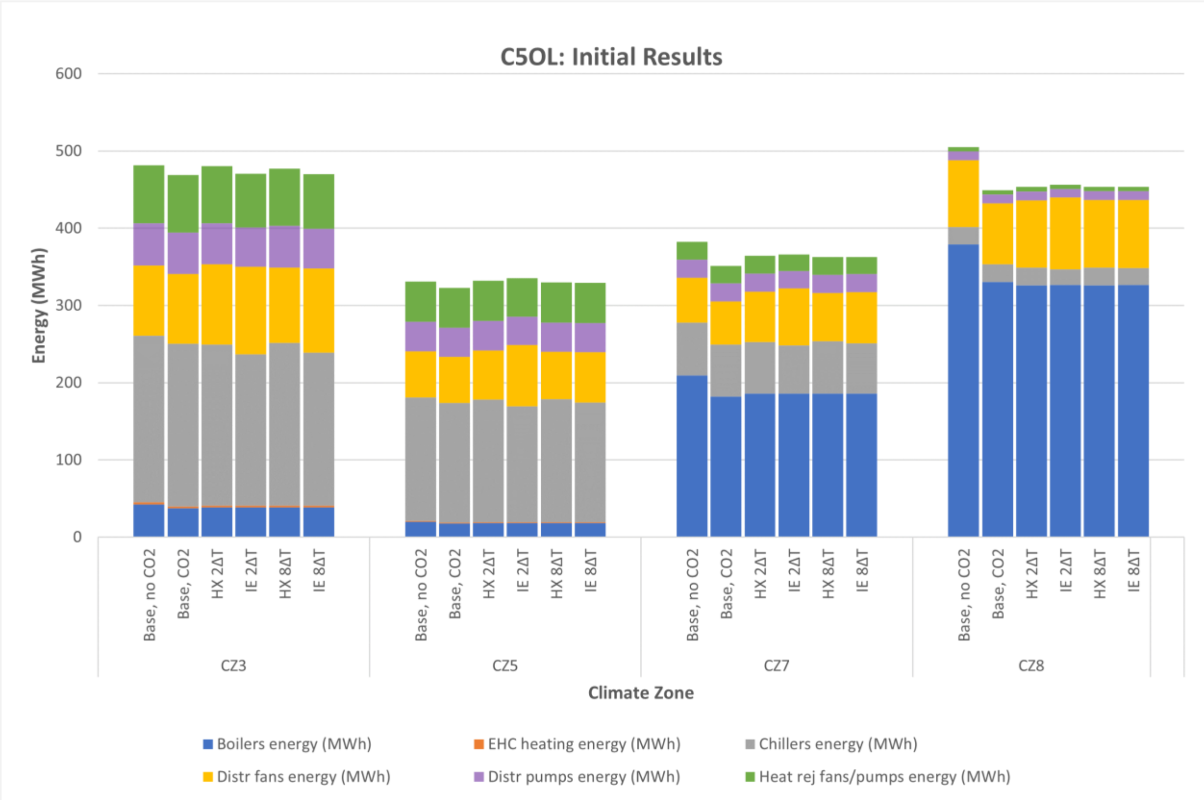


Figure 71: Initial Indirect Evaporative Cooling Simulation Results and Test Cases

Revised simulations incorporating test case improvements

Given the results show that generally the 8ΔT scenarios provide lower overall energy consumption than the 2ΔT cases, these simulations were extended over all climate zones, along with the C9A archetype. The full C5OL results are presented in Figure 72. The C9A results are presented in Figure 73.

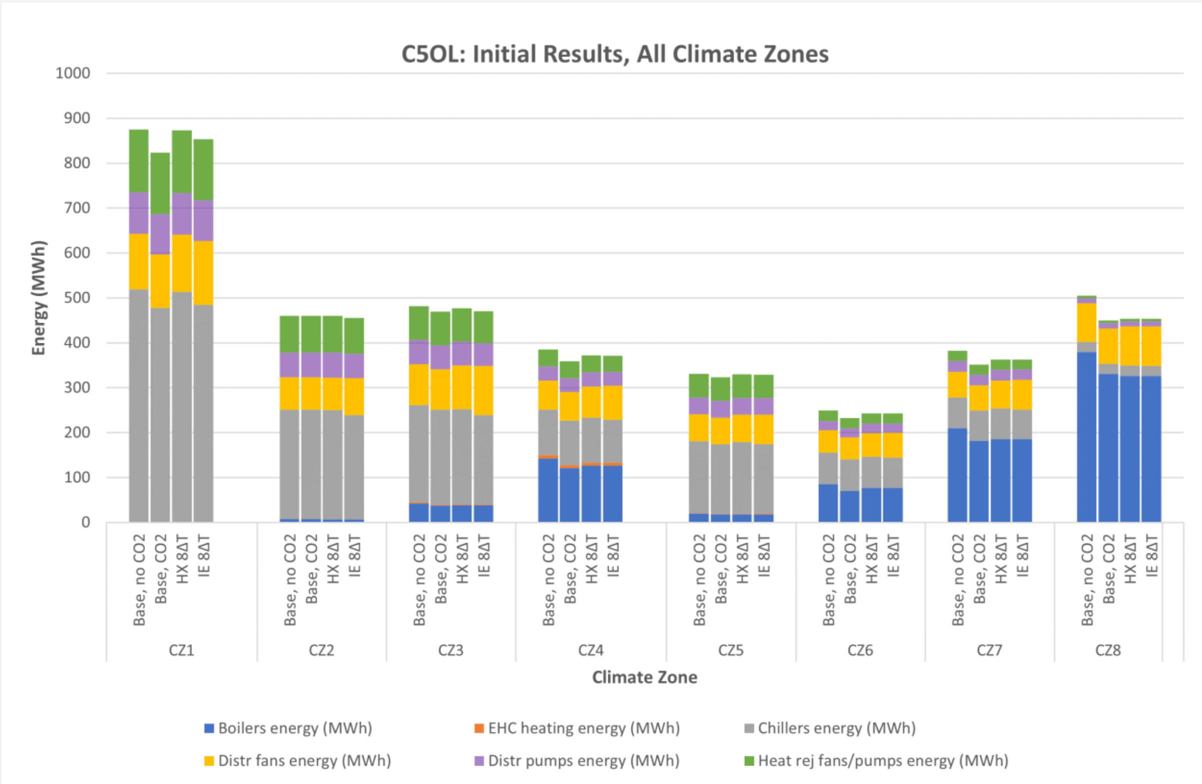


Figure 72: All C5OL climate zones, results extended for the 8ΔT cases

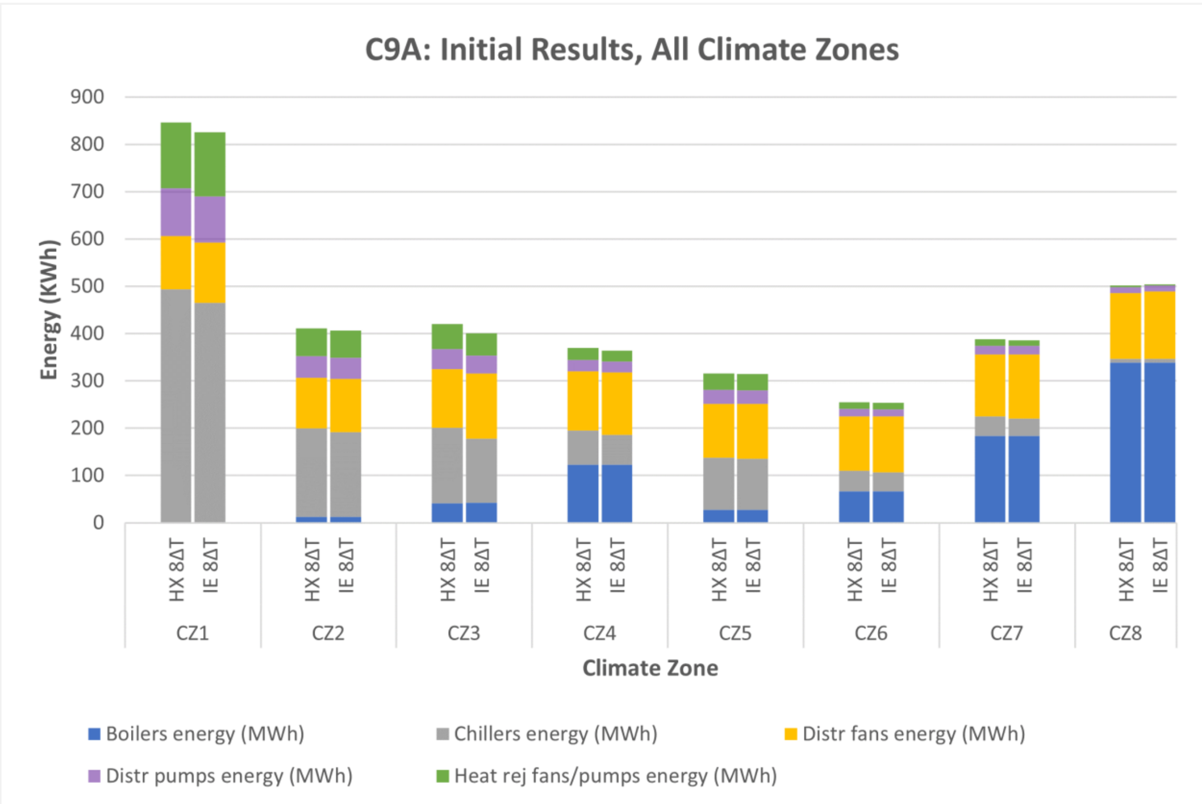


Figure 73: All C9A climate zones, base case compared with 8ΔT heat recovery and indirect evaporative cooling cases

The revised simulations also show a reduction in peak cooling loads in most climate zones, for the heat recovery and indirect evaporative cooling cases in each archetype. The peak cooling loads for each case are presented in Table 40 and Table 41.

Table 40: Peak cooling loads - C5OL

Case (C5OL)	CZ1 Peak Cooling Load (kW)	CZ2 Peak Cooling Load (kW)	CZ3 Peak Cooling Load (kW)	CZ4 Peak Cooling Load (kW)	CZ5 Peak Cooling Load (kW)	CZ6 Peak Cooling Load (kW)	CZ7 Peak Cooling Load (kW)	CZ8 Peak Cooling Load (kW)
Base, no CO2	1707.59	1416.28	1480.21	1028.97	1130.18	963.18	852.99	342.71
Base, CO2	1707.58	1416.28	1480.25	1016.16	1130.17	963.19	831.09	342.74
HX 8ΔT	1707.59	1416.28	1480.26	1019.06	1130.18	963.28	857.42	342.69
IE 8ΔT	1707.58	1416.27	1480.27	1013.61	1130.17	963.52	789.87	336.31

Table 41: Peak cooling loads - C9A

Case (C9A)	CZ1 Peak Cooling Load (kW)	CZ2 Peak Cooling Load (kW)	CZ3 Peak Cooling Load (kW)	CZ4 Peak Cooling Load (kW)	CZ5 Peak Cooling Load (kW)	CZ6 Peak Cooling Load (kW)	CZ7 Peak Cooling Load (kW)	CZ8 Peak Cooling Load (kW)
Base, CO2	512.69	462.46	409.67	356.29	354.31	290.97	258.84	57.10
HX 8ΔT	477.24	462.50	361.74	305.14	348.85	297.52	227.67	56.08
IE 8ΔT	471.12	430.02	347.14	294.38	330.62	290.97	215.40	53.76

6.4.3 BCR analysis results

With some cases and climate zones displaying energy and peak load reduction benefits, both factors were applied to a BCR analysis on a Net Present Value basis using the average equipment costs for each case/archetype combination using the following economic parameters:

- Assessment Timeframe: 40 years
- Equipment Replacement: 25 years
- Discount Rate: 5%

The BCRs were compiled specific for each of the research questions outlaid in Section 6.2.2, repeated here for reference:

- Is it cost effective to apply indirect evaporative cooling to a case using an energy reclaiming system to comply with NCC 2022?
- Is it cost effective to add an energy recovery system or indirect evaporative cooling where it is not required under NCC 2022?
- Is it cost effective to apply an energy recovery system or indirect evaporative cooling in place of modulating (CO_2 based demand control ventilation) control?

Table 42: BCR for application of indirect evaporative cooling to existing energy recovery – C5OL

Simulation	CZ4	CZ5	CZ6	CZ7	CZ8
HX + IE	-	-0.18	-	-0.11	-0.09
HX + IE 8dT	0.04	0.03	-0.02	0.00	0.00

Table 43: BCR for application of indirect evaporative cooling to existing energy recovery where required – C9A

Simulation	CZ4	CZ5	CZ6	CZ7	CZ8
HX +IE 8dT	0.43	0.10	0.10	0.14	-0.15

Table 44: BCR for application of energy recovery or indirect evaporative cooling where there is CO_2 control – C5OL

Simulation	CZ1	CZ2	CZ3
HX, no CO2	-1.13	0.00	-0.17
HX + IE	-	-	-0.01
HX +IE 8dT	-0.41	0.06	0.00

Table 45: BCR for Application of energy recovery or indirect evaporative cooling where there are no requirements - C5OL

	CZ2
HX, no CO2	0.00
HX +IE 8dT	0.05

Table 46: BCR for application of energy recovery or indirect evaporative cooling where there are no requirements – C9A

Simulation	CZ1	CZ2	CZ3
HX, no CO2	0.36	0.12	0.91
HX +IE 8dT	0.78	0.21	1.13

6.5 Discussion

Dewpoint coolers

The additional energy sustained as a result of the dewpoint cooler simulations for climate zones 7 and 8 appear to be a result of the complexities in managing reheat/simultaneous heating and cooling requirements with the reduction in minimum outside air temperatures. Nevertheless, climate zones 3 and 5, which exhibit a more consistent heating consumption baseline, still show very little benefit to cooling energy reduction. Both dewpoint cooler simulations show higher (albeit only slightly) energy consumption incurred from fan operation than the energy saved from chiller and heat rejection.

Indirect evaporative cooling

Comparing the initial simulation cases of CO_2 control against heat recovery and indirect evaporative cooling cases (2°C differential on cooling), the initial results show a consistent overall increase of energy consumption. In climate zone 8 there is a reduction in heating energy, but any reduction in chiller energy consumption is more than offset by additional fan energy required for the HRV to operate and overcome the internal resistance of the equipment.

This led to an assessment of the equipment's effective COP, which was based on a comparison of the energy recovered against the total (supply and exhaust) fan energy input. The objective of this assessment was to determine a more beneficial operating flag for the heat recovery unit. This assessment is summarised for the Series 1 Unit in Table 47, which was modelled as the centre zone unit for the C5OL archetype.

Table 47: Summary of Effective COP Analysis

Centre zone (4,900 L/s)		Heat recovery Unit	Indirect Evaporative Cooler
Total Fan Energy (kW)		5.91	6.24
Effective COP	2ΔT	1.19	1.13
	4ΔT	2.39	2.26
	6ΔT	3.58	3.39
	8ΔT	4.78	4.52

This level of detail in assessment is rarely applied to control sequences for real-world applications and highlights a potential for a major flaw in execution of heat/energy recovery technology. Comparing the results in Figure 72 and Figure 73 highlights the need to consider chiller technology/performance when applying heat recovery control sequences. That is, given C5OL uses more efficient water-cooled chillers, the penalty of additional fan energy for the heat recovery equipment results in a much wider temperature differential being required than the less efficient air-cooled chillers in the C9A archetype. It is anticipated that a large proportion of real-world applications to meet NCC 2022 requirements have been, and will be, implemented with control strategies that lead to the detriment of overall building energy performance.

The results in Table 40 and Table 41 show that the energy savings are far outweighed by the equipment capital costs. The negative results for the HX+IE case (2°C differential) highlight the point discussed above regarding the need to apply control strategies that are tailored to the installation in recognition of chiller performance.

The results in Table 42, Table 43 and Table 44 show that there is no benefit to applying heat recovery or indirect evaporative cooling where there are no heat recovery requirements currently (climate zones 1 to 3), except for potentially in climate zone 3. Table 42 in particular, displaying largely negative results, illustrates the benefit of CO_2/modulating control in comparison to both heat recovery and indirect evaporative cooling solutions. The case presented in Table 44 for climate zone 3 shows potential benefit for adding indirect evaporative cooling to a case where there is no requirement. This BCR result is most influenced by the relatively lower efficiency of the air-cooled chillers, and the equivalent comparison case for the C5OL archetype still shows a very low BCR.

Finally, the results for C5OL illustrated in Figure 73 shows CO_2/modulating control is more beneficial in all cases.

Summary of findings and recommendations

The simulations have tested the application of heat recovery and indirect evaporative cooling against the current NCC 2022 requirements outlined by Section J6D4 and found insufficient benefit primarily due to the high capital cost relative to the energy savings. Further inclusion of maintenance costs are expected to worsen the BCR results.

A secondary finding is due to the potential complexity in execution of controls, which leads to the risk that heat recovery could consume more energy than it saves at times of relatively low ambient air conditions.

Based on the analyses above:

- We do not recommend including indirect evaporative cooling in code.
- We recommend optimising the heat recovery requirements to improve the benefits and application of the technology.

6.6 Proposed Measures

J6D4 Mechanical ventilation system control

(1) General — A mechanical ventilation system, including one that is part of an air-conditioning system, except where the mechanical system serves only one sole-occupancy unit in a Class 2 building or serves only a Class 4 part of a building, must—

(a) be capable of being deactivated when the building or part of the building served by that system is not occupied; and

(b) when serving a conditioned space, except in periods when evaporative cooling is being used—

(i) where specified in Table J6D4, have—

(A) an energy reclaiming system that preconditions outdoor air at a minimum sensible heat transfer effectiveness of 60% and includes heat exchanger bypass functionality that enables heat transfer only when its energy recovery benefits exceed the additional supply/extract air fan energy consumption; or

(B) demand control ventilation in accordance with AS 1668.2 if appropriate to the application; and

(ii) not exceed the minimum outdoor air quantity required by Part F6 by more than 20%, except where—

(A) additional unconditioned outdoor air is supplied for free cooling; or

(B) additional mechanical ventilation is needed to balance the required exhaust or process exhaust; or

(C) an energy reclaiming system preconditions all the outdoor air; and

(c) for an airflow of more than 1000 L/s, have a variable speed fan unless the downstream airflow is required by Part F6 to be constant.

(1) Exhaust systems — An exhaust system with an air flow rate of more than 1000 L/s must be capable of stopping the motor when the system is not needed, except for an exhaust system in a sole-occupancy unit in a Class 2, 3 or 9c building.

(2) Carpark exhaust systems — Carpark exhaust systems must have a control system in accordance with—

(d) clause 4.11.2 of AS 1668.2; or

(e) clause 4.11.3 of AS 1668.2.

(2) Time switches — The following applies:

(a) A time switch must be provided to a mechanical ventilation system with an air flow rate of more than 1000 L/s.

(b) The time switch must be capable of switching electric power on and off at variable pre-programmed times and on variable pre-programmed days.

(c) The requirements of (a) and (b) do not apply to—

(i) a mechanical ventilation system that serves—

(A) only one sole-occupancy unit in a Class 2, 3 or 9c building; or

(B) a Class 4 part of a building; or

(ii) a building where mechanical ventilation is needed for 24 hour occupancy.

Climate zone	*Outdoor air* flow (L/s)	*Required* measure
1	>500	Modulating control
2	Not applicable	No *required* measure
3	>1000	Modulating control
4 and 6	>500	Modulating control or energy reclaiming system
5	>1000	Modulating control or energy reclaiming system
7 and 8	>250	Modulating control or energy reclaiming system

7 VSD Applications: Fans and Pumps

7.1 Background and context

Variable speed drives have become somewhat ubiquitous in modern efficient building design due to the significant reductions in fan and pump energy that can be achieved at reduced speed. To provide context for this, a fan or pump for which the flow is restricted by use of a resistance (valve or damper) the fan or pump energy reduces at the rate of approximately $x^{0.8}$ where x is the flow fraction. The same component with a variable speed drive, with the system operating at fixed pressure, will achieve a turndown of approximately x^2 while if pressure is also allowed to reduce, the turndown[24] can approach $x^{2.7}$. This is illustrated in Figure 74.

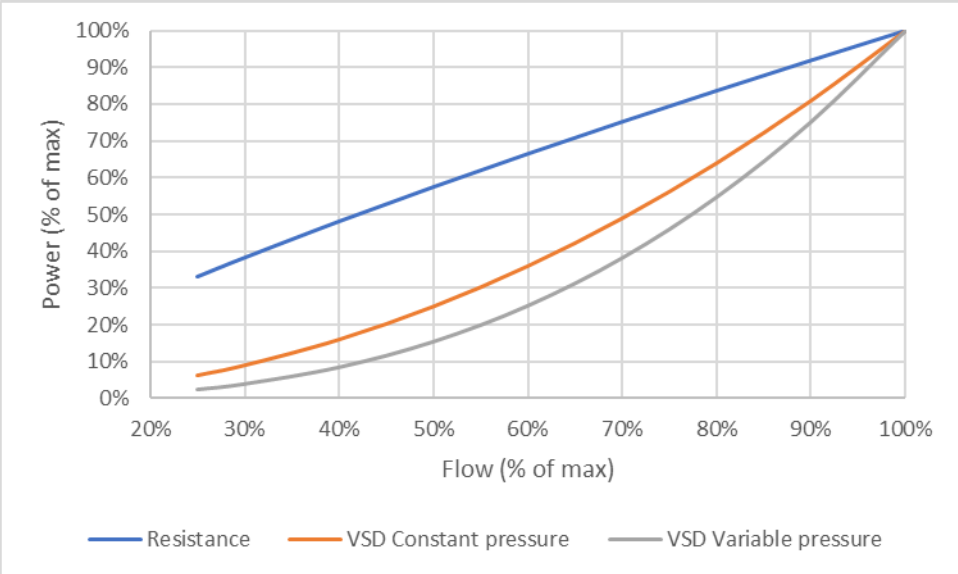


Figure 74: Typical power to flow turndown curves: Resistance ($x^{0.8}$) VSD constant pressure (x^2) and VSD variable pressure ($x^{2.7}$)

NCC 2022 has no requirement for variable speed pumps. Furthermore, while it has a VSD requirement for fans above 1000l/s (J6D4(c)), the use of a flow rather than power limit is not necessarily well attuned to the economics (which are driven by motor size).

Context must be taken into account when considering VSDs. There are a number of scenarios for VSD application that can be considered:

1. Fan/pump static balancing. In this scenario, the base case is that a flow resistance (damper/valve) is used to throttle an oversized fan or pump to achieve a design duty (giving an $x^{0.8}$ turndown), while in the VSD case the motor speed is reduced to achieve the required duty (giving an $x^{2.7}$ turndown, as the fan/pump is selected on the system curve). Typical examples of this case would be an oversized constant flow primary chilled water or heating hot water pump, or an oversized constant flow ventilation fan that requires adjustment to meet design flow requirements.
2. Fan/pump variable duty matching.

a. Fans. The expression of J6D4(1)(c), the current variable speed fan requirement, implies a base case where flow/duty matching is achieved by throttling (also known as riding the fan curve, which is an $x^{0.8}$ turndown), as compared to a variable speed approach with an x^2 (fixed pressure/variable flow) or $x^{2.7}$ (variable pressure/variable flow) turndown. In practice this is not a particularly realistic scenario, as riding the fan curve is not considered acceptable design[25]; we have however provided this analysis for fans in order to align with the current Code expression.
b. Pumps. For pumps, the base case in this scenario is a constant flow system (no turndown, x^0), as constant flow systems (with additional components to facilitate the constant flow) are not uncommon in practice. The VSD case in most cases is variable pressure/variable flow ($x^{2.7}$ turndown) but in some pumping scenarios where there is a fixed static head, a fixed pressure/variable flow scenario may be more realistic (x^2 turndown)[26].

Each of these scenarios is considered separated in this analysis.

7.1.1 Data collection and Review – VSDs

The scope of this measure is limited to the provision of third party VSDs to 3 phase motors for fans and pumps. The smallest 3 phase VSD available is 0.75kW, so this is used as a minimum size in all cases. This is a conservative approach as fans and pumps that are manufacturer-equipped with VSDs are likely to have a lower incremental cost.

While single phase motors have not been covered in the analysis, the 3-phase VSD cost model used here is conservative and can be used to extrapolate to single phase motors at the same sizes. Single phase motors are relatively uncommon in commercial sector fan and pump applications and are thus of limited materiality with the exception, arguably, of fan coil motors (which are generally smaller than 750W).

Costs

Costs for VSDs were provided by two suppliers (Danfoss and ABB) across the range 750W to 55kW, as shown in Figure 75. A linear cost curve was derived based on the motor input size.

[25] This riding the fan curve scenario is equivalent to having a constant speed fan serve a system with downstream control dampers that modulate the total flow demand. Such systems are very poor practice and rarely seen in practice.

[26] While less common in modern design, the design of constant flow pumping systems was once fairly common, with three-way valves used throughout the system. Primary-only pumping designs are still commonly configured to be constant flow (within a stage of chiller/boiler operation) via the use of a bypass valve. Excessive use of three port valves remains an issue in some system designs, undermining the benefits of variable flow design.

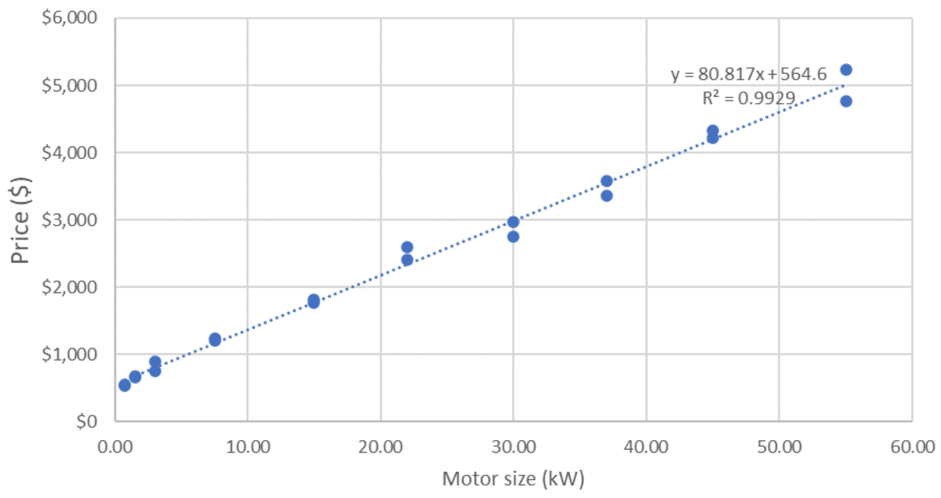


Figure 75: Cost curve for VSDs

Installation costs were derived at $1,750 per VSD, assuming $650 for labour, $700 for materials and $200 for BMS connection works. VSDs were selected at 120% the size of the expected motor draw at 100% load.

VSD efficiency and modelling

VSDs are not 100% efficient, as they have both standing losses associated with controls and power throughout driven losses. In order to assess this, VSD performance data across a range of sizes and operating loads down to 1.6% were obtained from http://www.variablefrequencydrive.org/vfd-efficiency. These were used to determine general operational efficiency by interpolation. Standby losses (at 0% load) were determined by interpolating the performance from 12.5% to 1.6% down to 0%, assuming linear behaviour in that low range of operation. On this basis, VSD losses were determined to be approximately 0.0150*kW + 0.05. It was assumed that VSDs are power downed when the motor being driven is not required; this is normal practice.

7.1.2 Typical duty figures for variable flow systems

In order to assess the economics of a variable flow system, it is necessary to understand both the hours and the average flow of the system. To this end, hourly data was downloaded and analysed from simulations of C5L (Large Office) and C9A (hospital) archetypes. These data represented the following situations:

1. For heating and cooling load, these figures represented the hourly heating/cooling load served by the heating hot water/chilled water systems, such as might be serviced by a secondary pumping system.
2. For the airflow case, these figures represented the airflow of VAV air-handlers.

Note that these scenarios are situations where the flow demand is proportional to the zone heating or cooling demand. Figures do not apply to systems that do not meet this criterion (e.g. car park exhausts, primary pumps in primary/secondary pump systems). The purpose of these figures is to provide baseline turndown figures against which energy/flow savings for various fan and pump installations can be compared. Results are listed in Table 48. A minimum pump/fan speed of 40% was applied to derive the results in the table[27].

Table 48. Average typical duties (% of maximum flow) for pumps and fans during operating hours. Average/min/max figures were determined by consideration of results from all 8 climate zones. No results were generated for airflow in the C9A archetype because this archetype does not have variable air volume air handlers.

	C5L Daytime Average	C5L Daytime Max	C5L Daytime Min	C9A Overnight Average	C9A Overnight Max	C9A Overnight Min
Cooling Load	0.51	0.61	0.44	0.50	0.61	0.43
Heating load	0.45	0.43	0.48	0.47	0.53	0.43
Airflow	0.48	0.61	0.43	n/a	n/a	n/a

Hours run for pumping systems in the archetypes are listed in Table 49.

Table 49. Hours run for heating and cooling pumping systems in the daytime (C5L) and overnight C9A) archetypes.

		CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
Cooling	Daytime (C5L)	2860	2767	2578	2214	2703	2494	2032	1594
	Overnight (C9A)	8760	7169	7030	5336	6744	5401	4640	3043
Heating	Daytime (C5L)	3	159	468	1222	343	969	1495	1723
	Overnight (C9A)	0	569	1117	2920	813	2274	3765	5255

7.2 Methodology

The methodology for this analysis is outlined in Figure 76.

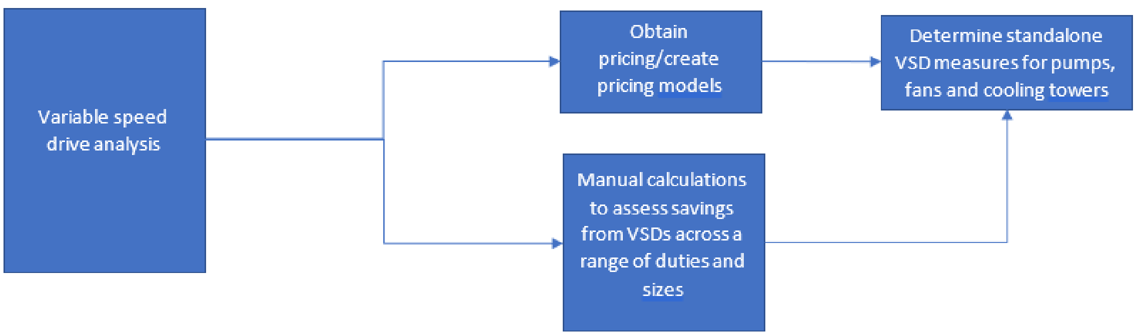


Figure 76. Outline methodology for the VSD analysis.

7.3 Results

7.3.1 Fan/Pump static balancing

For the purpose of this assessment, it has been assumed that the adjustment in duty required for static balancing applications is in the region of 15% with sensitivity figures at 10 and 20%. The incremental cost covers the VSD and installation. The results of the assessment are shown in Figure 77. For the range of operating hours to apply to these figures, see Table 48.

[27] General industry practice is that the lowest speed a motor can be operated at without risk of overheating is 40%. In practice, some configurations with multiple pumps or fans in parallel can achieve a superior turndown but the 40% assumption is conservative.

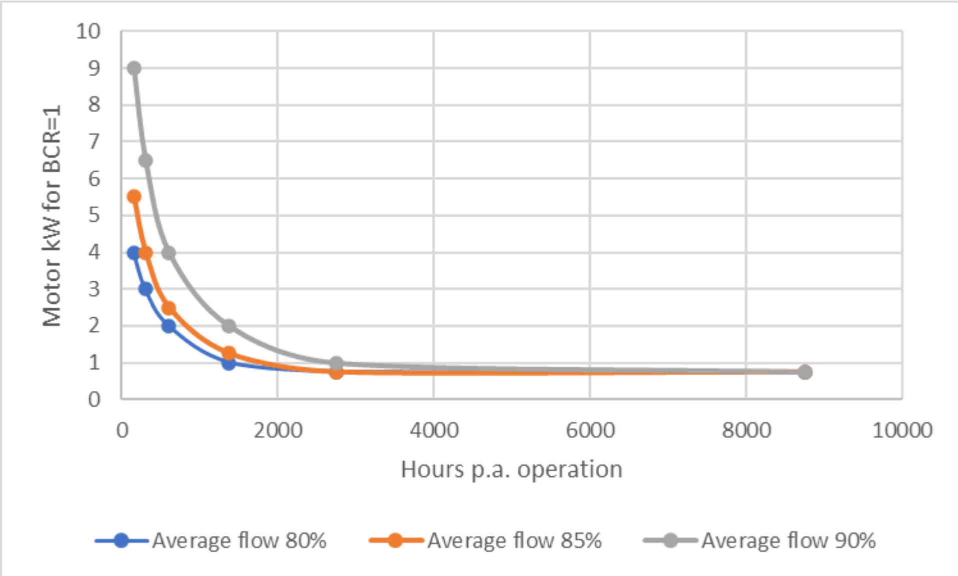


Figure 77. BCR=1 curves for static balancing applications ($x^{0.8}$ base case versus $x^{2.7}$ VSD case).

The results show that for above 3000 hours p.a. operation, the use of VSDs for static balancing can be justified to the minimum drive size (0.75kW). However, at low hours (such as might apply to a heating in a temperate climate), the required motor size is somewhat larger.

7.3.2 Variable versus constant flow

For this assessment a wider range of possible average flows has been considered, as typically a heating or cooling system may operate from 0-100%, although generally pump speeds are limited to a minimum pump speed of 40% due to motor cooling requirements.

From a capital cost perspective, we have based the assessment on VSD supply and installation cost only. This is conservative for pumping systems as in reality a correctly designed constant flow system would have other additional costs such as 3-way valves and/or bypass lines. For a fan system, however, the analysis is incomplete as a variable volume fan system incorporates additional components such as VAV boxes and has additional downstream energy impacts in terms of reheat.

The results of the assessment are shown in Figure 78 for variable pressure systems and Figure 79 for fixed pressure systems, based on x^0-$x^{2.7}$ and x^0-x^2 cases respectively. For the range of expected average flows and hours to apply to these figures, refer to Table 48 and Table 49.

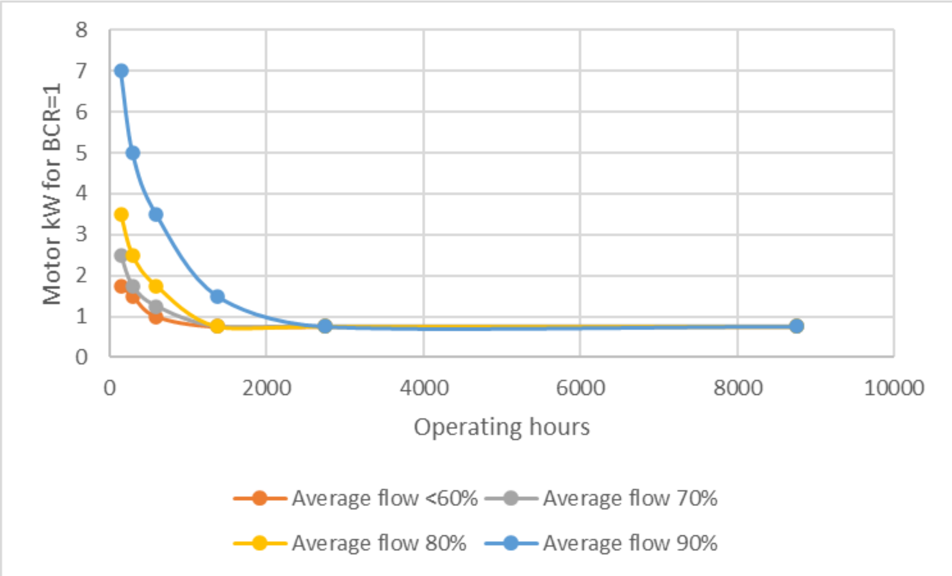


Figure 78. Motor size for BCR=1 in a variable pressure/variable flow system compared to a constant flow system (x^0 base case versus $x^{2.7}$ VSD case). 50% average flow points are equal to the 60% average flow points.

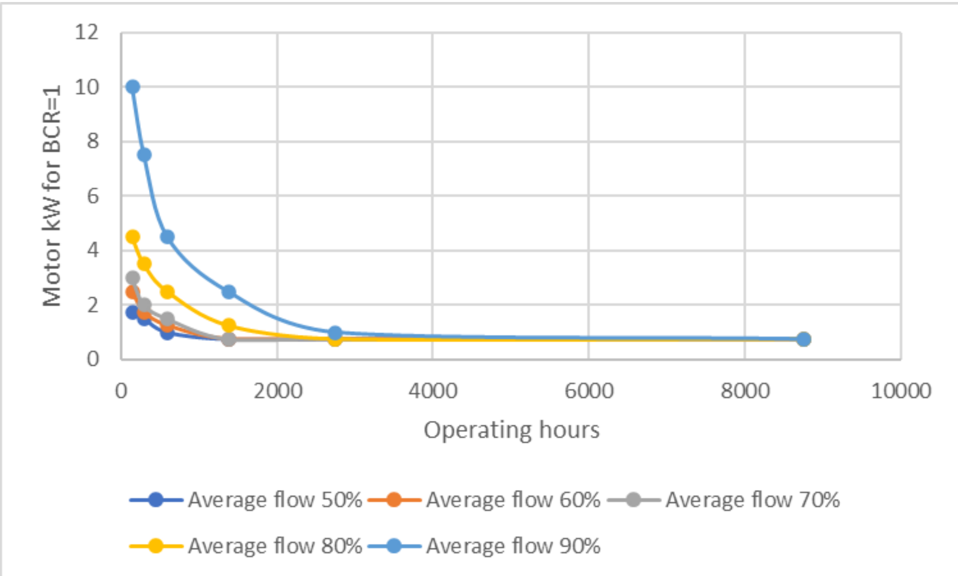


Figure 79. Motor size for BCR=1 in a fixed pressure/variable flow system compared to a constant flow system (x^0 base case versus x^2 VSD case).

7.3.3 Variable duty matching for fans and pumps

A sub-case for variable duty systems is the comparison of allowing the fan or pump to "ride the "curve" against the system resistance as dampers or valves close in the field. This is a marginally realistic technical scenario – it would be rare in practice to design a system to ride the curve. However, it is a necessary analysis to complement the existing NCC 2022 regulation J6D4 (1)(c). This regulation requires all fans above 1000l/s to have VSDs, unless the fan is serving loads that are required to be constant under Part F6. We have extended the coverage of the analysis to include pumps.

For the range of expected average flows and hours to apply to these figures, refer to Table 48 and Table 49.

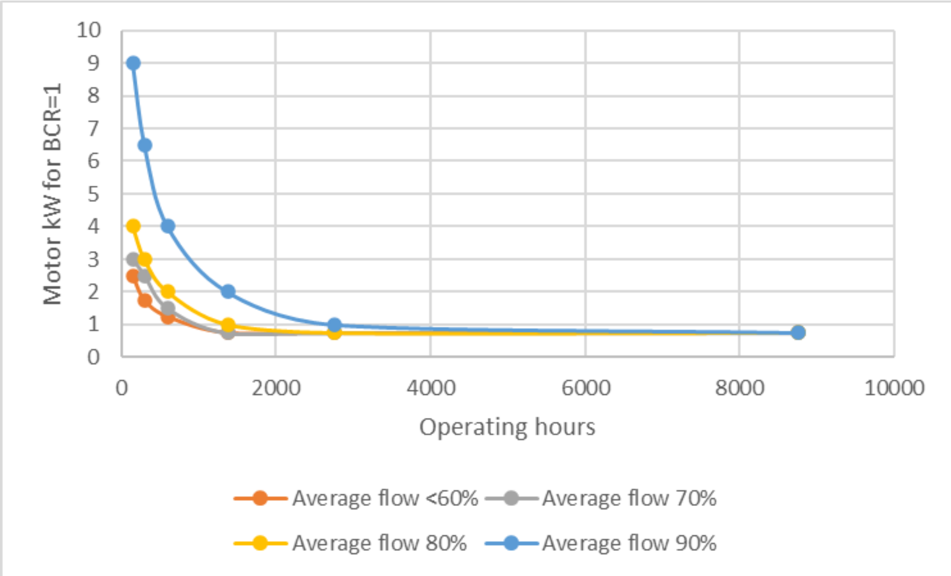


Figure 80. Benefit-cost ratio=1 curves for fans and pumps in the comparison between riding the curve or using a VSD at variable pressure ($x^{0.8}$ base case versus $x^{2.7}$ VSD case).

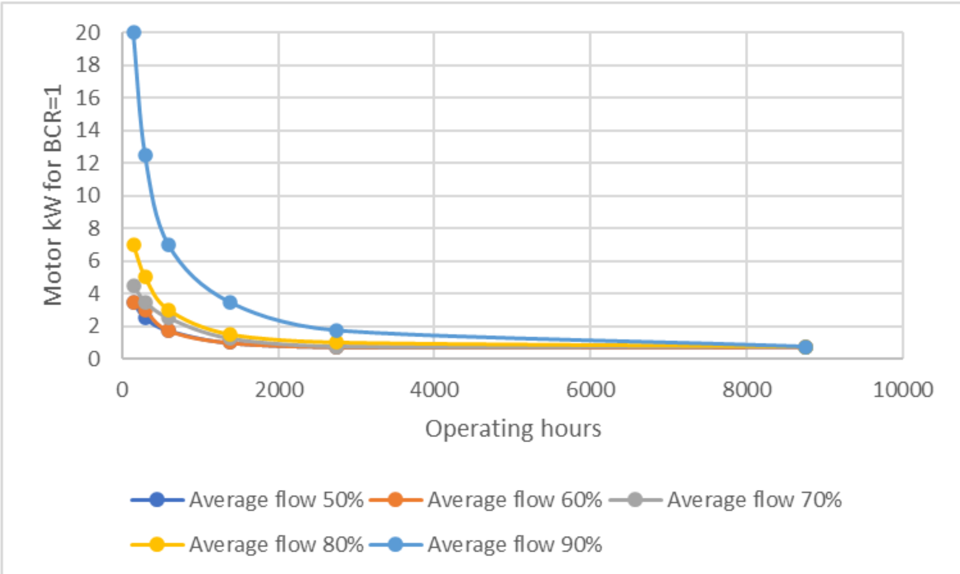


Figure 81. Benefit-cost ratio= 1 curves for fans and pumps in the comparison between riding the curve or using a VSD at constant pressure ($x^{0.8}$ base case versus x^2 VSD case).

7.4 Discussion

7.4.1 Demands and duty hours for pump systems

For pump systems, the VSD comparison needs to be based on x^0-$x^{2.7}$ or x^0-x^2 as shown in Figure 78 and Figure 79. Using the more conservative constant pressure system, it can be seen from Table 48 that the weighted average duty is in the region 0.44-0.68 with an average of approximately 0.52 for cooling applications, and 0.44-0.56 with an average of 0.48 for heating applications.

For cooling systems, Table 49 shows that these run for between 1594 and 2860 hours for the daytime archetype and 3043-8760 hours for the overnight archetype. Applying these figures, and cross checking extreme results for individual climate zones, the threshold for VSD viability is 0.75kW in all cases.

For heating systems, Table 49 shows that these run for between 159 and 1723 hours for the daytime archetype and 546-5255 hours for the overnight archetype. Based on these figures, the threshold for VSD viability is 0.75kW except in climate zone 2,3 and 5 where it is 1.5-1.75kW. These are relatively small deviations from minimum and can be ignored.

7.4.2 Demand and duty hours for fan systems

For fan systems, the VSD comparison is as shown in Figure 80 and Figure 81, representing the somewhat marginal technical case of riding the fan curve versus VSD. For systems servicing heat and cooling loads, the average turndown is in the region 0.5-0.7 depending on pressure control, at which level of is the use of a VSD is economic is economic to 1.25kW and below.

7.4.3 Static Balancing Applications

Standard oversizing practices would include a 10% safety factor and then size with a further margin of around 5%, giving a typical turndown in balancing of 15%. This means that variable speed drives are viable for this purpose between 0.75kW and 5.5kW depending on operating hours.

7.4.4 Fan/pump variable duty matching

Pumps

As noted in Section 7.1.1, the incremental cost between variable flow and constant flow pumped systems is typically less than the cost of the VSD, as the constant flow systems require additional hydraulic components to maintain the constant flow.

This means that in applications where the analysis presented in Figure 78 and Figure 79 shows a positive result, it can be interpreted as being an endorsement of a variable flow system in general, as opposed to merely an endorsement of the pump being variable speed.

Fans

As noted in Section 7.1.1, the situation for a fan system is more complex than for a pump system, as fan systems do not always use dampers to regulate the provision of heating and cooling to zones. This limits the scope of measures that can be recommended in this instance.

For simple single zone systems, the gross variation of flow is a feasible approach in practice although not commonly practiced in design, as there can be concerns about the management of balancing; such concerns are particularly valid once flow drops below 60%, as at that point balancing tends to fail in the absence of VAV boxes or similar[28]. As such systems are inherently variable pressure when subject to variable flow, the results shown in Figure 78 apply, indicating that for any real system (likely to be operating more than 2000 hours and achieving an average flow lower than 90%), variable speed operation is economic. Note that in this application it is necessary to qualify the use of variable flow by the maintenance of satisfactory ventilation. This is because single zone systems may have an essentially passive outside air intake (and frequently, no return fan) with the result that a reduction in

[28] For unitary AC systems with 3-phase fans (generally ducted units), there is an additional airflow constraint dictated by the refrigeration system, which sits typically at around 70% of maximum flow. However, in the deadband, there is no limitation on fan turndown other than is required for ventilation and air movement.

fan speed would lead to a reduction in outside air intake. As no BCA has been undertaken to force such systems to use a controlled outside air flow, it is necessary to exclude cases where variable speed fan operation would lead to unsatisfactory outside air supply.

For multizone systems, gross variation of flow is not feasible without the introduction of VAV boxes or at least dampers to ensure that the flow distribution is maintained. In cases where a system serves zones with differing thermal loads, these would normally be met with reheat and/or VAV boxes. The assessment of the cost-benefit differences between constant volume reheat and VAV systems is beyond scope for this study, although general industry practice has moved away from constant volume reheat in most applications.

For a fan system that already has variable flow demand, the question of whether to use a VSD or allow the fan to ride the curve is somewhat moot as designers very rarely settle for allowing the fan to ride the curve. Nonetheless, a perspective on this can be obtained by inspection of Figure 77 which analyses the comparable situation of dampers versus VSD. Given that in the presence of downstream flow control, flow demand is expected to drop well below 80% on average, it is reasonable to conclude that industry practice reflects cost-effective behaviour, i.e. a VSD should always be used where there is downstream flow control.

7.4.5 Variable pressure versus constant pressure

Traditional practice until around 2010 tended to control fan and pump systems on a constant pressure basis. This approach is simple and robust from a control perspective but leads to the situation where a VSD operates to maintain a pressure setpoint against a bank of mostly closed valves or dampers. This situation not only wastes energy, but also degrades the ability of the valves or dampers to effectively maintain flow control, creating issues with control stability and noise.

The use of variable pressure control has become standard practice in building aiming to achieve higher levels of energy efficiency. This can be achieved by a range of methods, most of which involve adjusting the pressure setpoint dynamically to ensure that a minimum number of valves/dampers are more than a given percentage (typically 90%) open. Simpler but less effective methods include controlling valves to maintain a supply/return temperature difference, and simple resets based on outdoor condition as a proxy for load. The key limitations of the approach are:

1. For pumping systems with a fixed minimum static head, the control cannot drop below that static head setpoint.
2. For fan systems with high induction diffusers or active chilled beams, it is necessary to maintain a minimum pressure setpoint to ensure that the induction feature operates to specification.
3. For fan systems with parallel fan VAV boxes, the VAV fans create a back pressure on the AHU which must be counteracted with a minimum pressure, at least while the parallel fan is operating.

Nonetheless, it is fair to say that in most situations a significant degree of additional energy savings can be achieved through the use of a variable pressure control.

The costs of implementing the variable pressure control relate to the added complexity of programming the pressure reset. However, our enquiries with industry indicate that this is sufficiently common practice that it is not treated as an additional cost in new construction.

7.5 Proposed Measures

7.5.1 Pump Systems

J6D8

(1) General — Pumps and pipework that form part of an air-conditioning system must –

1. either—

    (i) separately comply with (2), (3) and (4); or

    (ii) achieve a pump motor power per unit of flowrate lower than the pump motor power per unit of flowrate achieved when applying (2), (3) and (4) together.

    And, for pumps with 3-phase motors

2. Be equipped with a variable speed drive; and
3. Where they pump circulation systems servicing air handlers, fan coils or other devices for delivering heating and cooling in the building:
    a. Be variable volume in operation with a variable speed pump; and
    b. Operate a to variable pressure to the extent permitted by static head requirements and the requirements of downstream equipment

In addition, provisions for distributive constant speed systems in J6D8 (5) need to be removed.

A further clause can be added to protect the efficacy of variable flow systems:

Uncontrolled bypasses and three port valves shall not be used in a distributive pumping system.

7.5.2 Fan systems

J6D5

(1) Fans, ductwork and duct components that form part of an air-conditioning system or mechanical ventilation system must—

(a) either

a. separately comply with (2), (3), (4) and (5); or

b. achieve a fan motor input power per unit of flowrate lower than the fan motor input power per unit of flowrate achieved when applying (2), (3), (4) and (5) together.

And, for fans motor power input greater than 750W:

(b) Have a variable speed drive; and
(c) To the extent possible while maintaining compliance with the minimum outside air requirements of AS1668.2:
a. Where the system serves a single zone, operate at a reduced flow in the dead band of the zone temperature control; and
b. Where the downstream airflow demand is variable:

i. Be variable volume in operation with a variable speed fan; and

ii. Operate to a variable pressure to the extent permitted by the requirements of downstream equipment

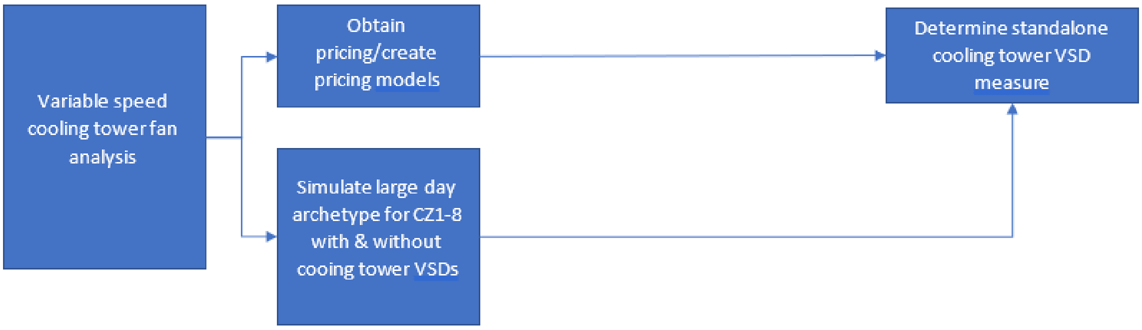
8 VSD Applications: Cooling Towers

8.1 Background and context

Cooling tower fans are used to control the rate of heat rejection from cooling towers. In a fixed speed configuration, the fans are turned on and off on an essentially thermostatic control in order to achieve a fixed leaving temperature. This gives an essentially linear turndown in fan power with load. In a variable speed configuration, the fans are operated at variable speed to maintain the fixed leaving temperature; ideally this is done by staging the fans on in parallel at minimum speed and then ramping the fans together from minimum to 100%, but it can also be controlled with towers staging on and ramping sequentially. Fan turndown in this case is variable pressure i.e. $x^{2.7}$, but the degree of overall savings is determined by a balance between fan energy use and heat transfer characteristics under variable airflow. In general, variable speed fans for cooling towers achieve significant energy savings and also provide better stability of control for the leaving temperature. NCC 2022 has no requirements for variable speed cooling tower fans.

8.2 Methodology

The methodology for this measure is shown in Figure 82.



.

Figure 82. Outline methodology for cooling tower VSD analysis

Capital costing for this measure is taken from Section 7.1.1.

Simulations are necessary for this measure because of the interplay between cooling tower fan speed, airflow and heat transfer. Simulation studies used the Large Office building (C5OL) archetype only[29], with water cooled chiller plant consisting of two equally sized chillers at 60% of design load each and two equally sized NCC 2022 compliant induced draft cooling towers at 60% of design heat rejection capacity each (5°C approach). Fixed speed and variable speed operation were modelled in two control cases, being:

a. Interlocked: Each cooling tower is operated in one-to-one coordination with a single chiller
b. Parallel: Both cooling towers can operate to service either or both chillers.

These controls mimic the sequential and parallel control approaches discussed in Section 8.1.

---

[29] Given the strength of the results for this archetype, it was considered unnecessary to run the simulations for the longer hours archetype as this would be expected to generate even more favourable outcomes.

Note that although the analysis has been performed for cooling towers, it is plausible to extend the validity to other forms of heat rejection, such as air-cooled condensers (other than those covered under chiller and unitary air-conditioning systems) which have similar load ranges to service.

8.3 Simulation results

Relative energy consumption for each case is shown in Figure 83.

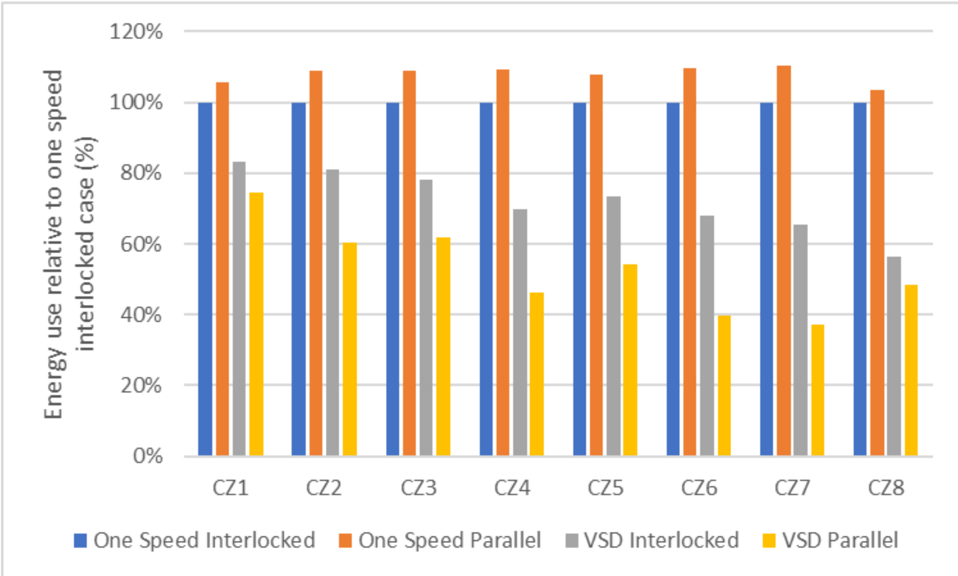


Figure 83. Relative energy use of each cooling tower fan case for each climate zone.

For each simulation, the calculated cooling tower fan energy use was divided by the total cooling tower fan kW to obtain a normalised kWh/kW figure. The difference in normalised energy use was then extrapolated across a range of fan sizes to provide input into the benefit-cost analysis. Normalised figures are shown in Figure 84.

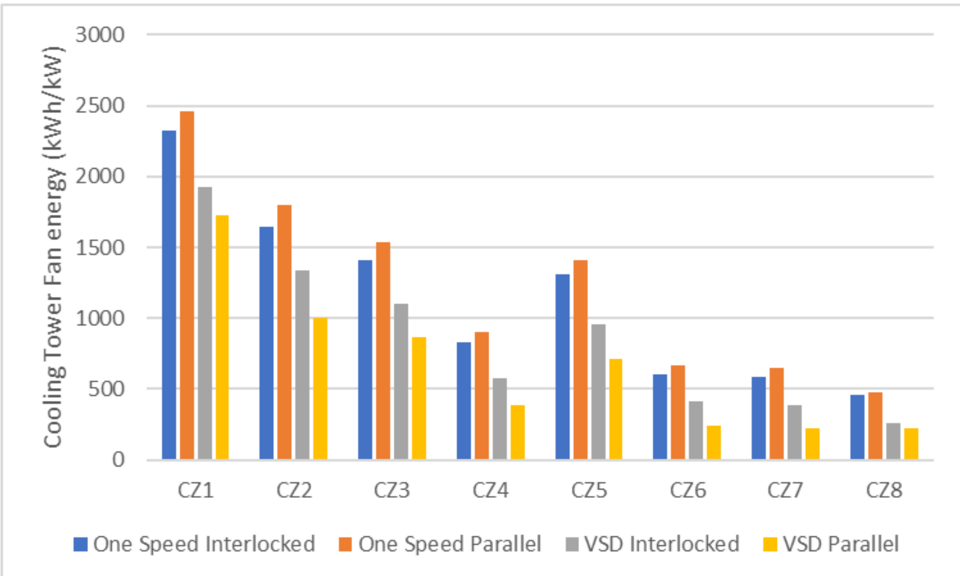


Figure 84. Normalised cooling tower fan energy use.

8.3.1 Benefit costs analysis results

The benefit cost analysis was conducted based on a 15-year equipment lifespan. Results are shown in Table 50.

Table 50. Benefit-cost ratios for the use of variable speed drives on cooling tower fans (interlocked operation) based on Large Office building (C5OL) archetype. BCRs improve at higher motor kW figures. Results illustrate that VSDs are always cost-effective in this application.

	0.75	1	1.25	1.5
CZ1	1.38	1.82	2.25	2.68
CZ2	1.50	1.99	2.46	2.92
CZ3	1.24	1.64	2.03	2.41
CZ4	1.03	1.36	1.68	1.99
CZ5	1.38	1.83	2.26	2.69
CZ6	0.84	1.11	1.37	1.63
CZ7	0.85	1.13	1.39	1.66
CZ8	0.55	0.72	0.89	1.06

Table 51. Benefit-cost ratios for the use of variable speed drives on cooling tower fans (parallel operation) based on Large Office building (C5OL) archetype. BCRs improve at higher motor kW figures. Results illustrate that VSDs are always cost-effective in this application.

	0.75	1	1.25	1.5
CZ1	1.69	2.22	2.75	3.27
CZ2	1.85	2.44	3.01	3.58
CZ3	1.54	2.03	2.51	2.99
CZ4	1.20	1.59	1.97	2.34
CZ5	1.62	2.14	2.65	3.15
CZ6	0.98	1.29	1.59	1.89
CZ7	1.00	1.32	1.63	1.94
CZ8	0.58	0.77	0.95	1.13

It can be seen from Table 50 that VSDs are cost-effective under both control scenarios down to the minimum 3 phase VSD size of 0.75kW other than for climate zones 6-8; and always cost-effective above 1.5kW.

8.4 Discussion

The results show that fans are cost-effective to 0.75-1.5kW motor size dependent on climate zone. In practice, cooling towers have much larger fans than this (a NCC-compliant induced draft cooling tower has a fan size of 1.04kW per 100kW heat rejection capacity, and in practice cooling towers are not installed at low capacity because of the high costs of water treatment). On this basis it can be concluded that cooling tower fan VSDs are almost always economic above 1kW and should be mandated.

One extrapolation has been asserted for this measure - being that the same considerations apply to refrigerant heat rejection equipment. This is reasonable, since the loads will fluctuate similarly to those experienced by cooling towers, when such equipment serves an air-conditioning system (a boutique application given that PAC units and air-cooled chillers are excluded).

8.5 Proposed Measure

J6D13 Heat rejection equipment

(1) The motor rated power of a fan in a cooling tower, closed circuit cooler or evaporative condenser must not exceed the allowances in Table J6D13.

(2) The fan in an air-cooled condenser must have a motor rated power of not more than 42 W for each kW of heat rejected from the refrigerant, when determined in accordance with AHRI 460 except for—

(a) a refrigerant chiller in an air-conditioning system that complies with the energy efficiency ratios in J6D11; or

(b) packaged air-conditioners, split systems, and variable refrigerant flow air-conditioning equipment that complies with the energy efficiency ratios in J6D12.

(3) All fans above 1kW in heat rejection equipment must be variable speed, other than –

a refrigerant chiller in an air-conditioning system that complies with the energy efficiency ratios in J6D11; or

(b) packaged air-conditioners, split systems, and variable refrigerant flow air-conditioning equipment that complies with the energy efficiency ratios in J6D12.

9 Economy Cycle

9.1 Background and context

Economy cycle systems utilise up to 100% fresh air for the supply air. This can be advantageous in milder climates when outside conditions are often suitable for providing cooling without the need to operate chillers. Economy cycle mode should occur any time the outside conditions are at a lower temperature/enthalpy to the return air, resulting in higher cooling effect being provided by the HVAC equipment. Most ducted HVAC equipment can be fitted with economy cycle and can be in the form of an Air Handling Unit (AHU), Fan Coil Unit (FCU), Packaged AC (PAC) unit, or any ducted type refrigerative AC unit. Typically, PAC units are of a more standardised manufacture and installation, with highly replicable on-board controls systems. As such, these units are typically much less costly than more customised FCU, AHU and ducted AC units.

The design and installation of economy cycle systems require larger outside air intakes and ductwork, the sizes of which are similar to the associated HVAC equipment's (AHU, FCU, PAC, or AC unit) supply air ductwork. Without economy cycle, outside air is usually only required for 'minimum' amounts of outside air as per AS 1668.2, which is typically 10-30% of the total supply air amount. The minimum outside air provision required by AS 1668.2 is typically low enough that sufficient relief can be provided by toilet/general exhausts and building leakage. However, the additional outside air introduced into a building by economy cycle often gives rise to the need for building pressure relief. This relief path can be remote to the HVAC equipment (as long as sufficient air can be relieved from a location representative of overall building pressure), but it is commonly most cost effective to add a relief air path by expanding the HVAC return air system. If the HVAC system also incorporates a return air fan, incorporating this equipment into the relief air system enables improved building pressure control.

9.1.1 Current provisions

The current NCC Section J (2022 revision) specifies provisions for air conditioning systems to be installed with an outdoor air economy cycle in Section J6D3 (1) (c), and by reference Table J6D3 (extract below in Figure 86).

(c) which provides the *required* mechanical ventilation, other than in *climate zone* 1 or where dehumidification control is needed, must have an *outdoor air economy cycle* if the total air flow rate of any airside component of an *air-conditioning* system is greater than or equal to the flow rates in Table J6D3; and

Figure 85: Current NCC 2022 economy cycle requirements (clause)

Table J6D3: Requirement for an outdoor air economy cycle

Climate zone	Total air flow rate *requiring* an economy cycle (L/s)
2	9000
3	7500
4	3500
5	3000
6	2000
7	2500
8	4000

Figure 86: Current NCC 2022 economy cycle requirements (reference table)

The 2022 requirements remain unchanged after being first introduced in the 2019 revision of the NCC.

9.2 Methodology

The methodology for economy cycle analysis is shown in Figure 87.

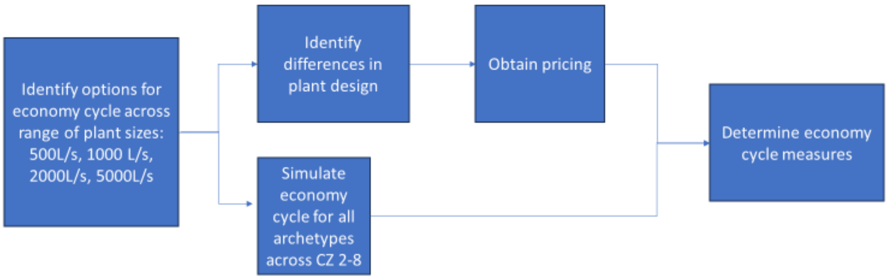


Figure 87. Outline methodology for the economy cycle analysis.

9.2.1 Capital costs

Both site specific installation and ambient weather conditions can impact the costs and benefits of adding economy cycle to a given HVAC system. In order to capture some of this variation, specific economy cycle designs with different configurations were assessed across a range of different archetypes, as summarised in Table 52.

Table 52: Economy cycle cases assessed (AHU/FCUs)

Airflow (L/s)	Building Description	Economy Cycle Impacts
500	School (Day)	• Addition of controlled relief air path • Increase in outside air system • Addition of a motorised return air damper
500	Small Hotel (Night)	As per School (Day)
1,000	Office (Day)	As per School (Day)
1,000	Aged Care (Night)	• Increase in outside and relief air systems • Addition of a motorised return and relief air dampers
2,000	Hospital (Night)	• Addition of controlled relief air path (system) • Addition of an economy mode outside air system (additional to minimum outside air) • Addition of a motorised return air damper
2,000	Small Retail (Day)	As per Hospital (Night)
5,000	Large Office (Day)	As per Hospital (Night)
5,000	Large Retail (Night)	As per Hospital (Night)

In addition to this, the costs of adding economy cycles to PAC units was separately assessed for the cases listed in Table 53. These differ from the economy cycle designs considered in Table 52 because the majority of the required economy cycle equipment is provided as a standardised add-on to the PAC unit design, with very minor external ducting system modifications required on site during installation. This significantly reduces costs.

Table 53: Economy cycle cases assessed (PAC units)

Airflow (L/s)	Building Description	Economy Cycle Impacts
500	School (Day)	• Manufacturer option: ○ Addition of an economy damper and weather hood to suit ○ Modified controls (onboard) • Addition of a relief air damper to return air ductwork.
500	Small Hotel (Night)	As above
1,000	Office (Day)	As above
1,000	Aged Care (Night)	As above
2,000	Hospital (Night)	As above
2,000	Small Retail (Day)	As above
5,000	Large Office (Day)	As above
5,000	Large Retail (Night)	As above

Economy cycle costs for FCUs/AHUs were gathered specific to the unit airflow capacities required and plotted in Figure 88 as the incremental cost of adding economy cycle. The incremental costs based on unit airflow capacity increases at a much faster rate beyond 2,000L/s due to the increased complexity of these units. The larger AHUs include separate dampers for economy cycle and minimum outside air (common to one damper for the smaller FCUs/AHUs), and more sophisticated control functions including supply air temperature and pressure sensors.

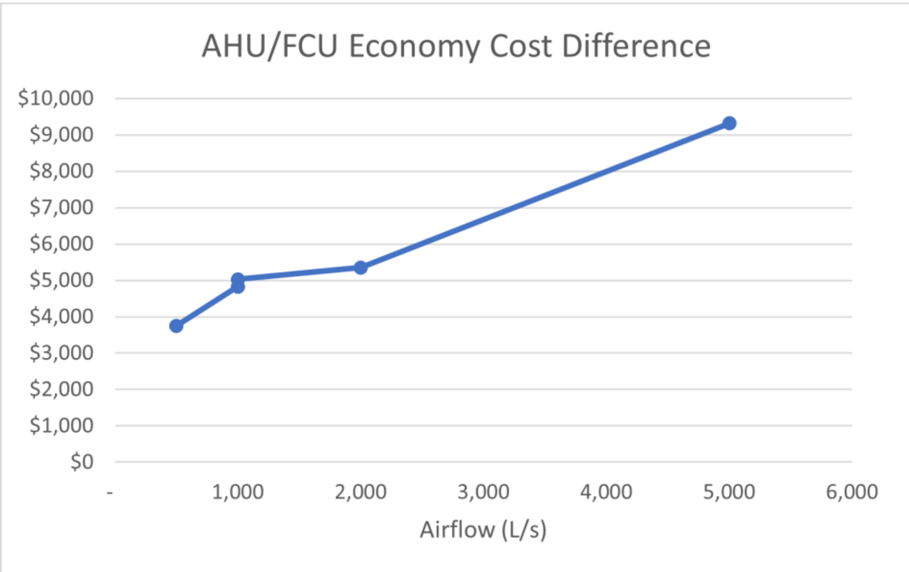


Figure 88: AHU/FCU incremental cost for addition of economy cycle

For clarity, the underlying data has been summarised and tabulated in Table 54.

Table 54: AHU/FCU Economy Cycle Costs and Price Difference

Description (airflow)	Cost With Economy Cycle	Cost Without Economy Cycle	Price Difference (%)
School (500 L/s)	$23,940	$20,190	19%
Small Hotel (500 L/s)	$23,940	$20,190	19%
Office (1000 L/s)	$28,120	$23,290	21%
Aged Care (1000 L/s)	$28,120	$23,090	22%

Description (airflow)	Cost With Economy Cycle	Cost Without Economy Cycle	Price Difference (%)
Hospital (2,000 L/s)	$95,630	$90,280	6%
Small retail (2,000 L/s)	$95,630	$90,280	6%
Large Office (5,000 L/s)	$111,090	$101,770	9%
Large Retail (5,000 L/s)	$111,090	$101,770	9%

For PAC units, Figure 89 shows the costs gathered from suppliers using the available equipment at discrete airflow quantities. The unit sizes required for direct and exact comparison with the nominated airflow quantities was not possible due to the available equipment sizes, so a line of best fit was applied to the gathered costs to enable calculation of the costs associated with specific airflow quantities. The cost trend associated with this equipment is much more linear as equipment remains similar in terms of function and complexity throughout the capacity range.

More detail on the costing build up is provided in the Appendices, Section 18.1.

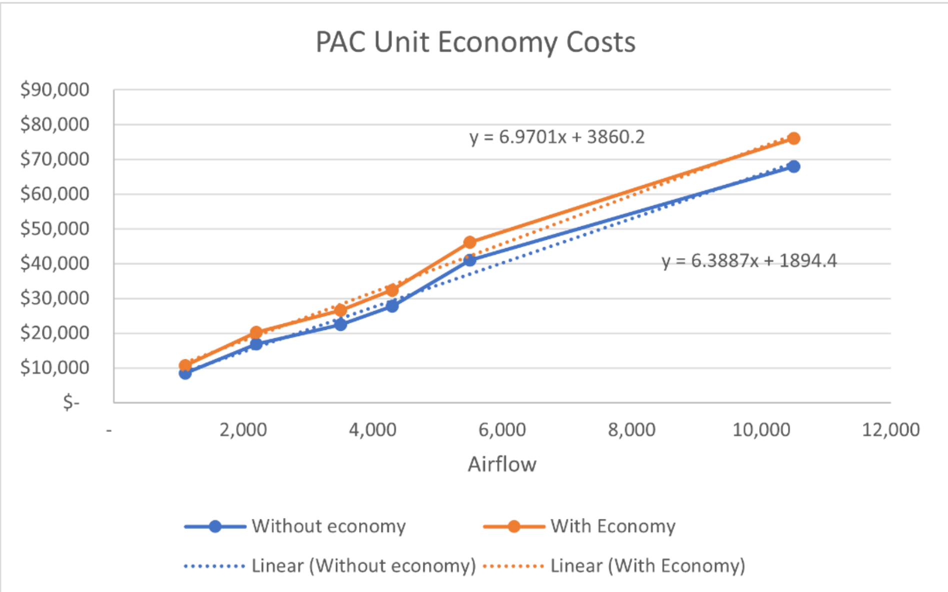


Figure 89: PAC unit costs with and without economy cycle

Again for clarity, this data has been summarised and tabulated in Table 55.

Table 55: PAC Unit Economy Cycle Costs and Price Difference

Airflow	Cost Without Economy Cycle	Cost With Economy Cycle	Price Difference (%)
1,100 L/s	$10,750	$8,400	28%
2,200 L/s	$20,200	$16,800	20%
3,500 L/s	$26,550	$22,500	18%
4,300 L/s	$32,400	$27,800	17%
5,500 L/s	$46,150	$41,000	13%
10,500 L/s	$76,000	$68,000	12%

9.2.2 Simulation studies

Although the costings for AHU/FCU economy cycles were based on multiple archetypes, the simulation analysis was based on a more limited range of archetypes for practical reasons.

The energy impacts of economy cycles were tested via a total of eight simulations per climate zone using the combinations listed in Table 56.

Table 56: Modelling Summary

HVAC Equipment	Operation	Archetype	Economy Cycle
FCU/AHU	Day	C5OL	Yes
FCU/AHU	Day	C5OL	No
FCU/AHU	Night	C9AS	Yes
FCU/AHU	Night	C9AS	No
PAC Unit (inverter only)	Day	C5OM	Yes
PAC Unit (inverter only)	Day	C5OM	No
PAC Unit (inverter only)	Night	C9C (modified to have PAC units serving zones)	Yes
PAC Unit (inverter only)	Night	C9C (modified to have PAC units serving zones)	No

The results were normalised and scaled to represent the archetypes with different equipment airflow quantities to provide a total of 64 individual annual energy consumption results.

The economy cycle controls were modelled using a different strategy depending on the HVAC equipment applicable to each case. The primary difference in equipment functionality relevant to economy cycles is that FCU/AHUs incorporate supply air temperature control.

FCU/AHU control

- Economy cycle is locked out when outdoor ambient conditions are above the following setpoints:
  - Dry bulb temperature: 23°C
  - Dewpoint temperature: 15°C
- The economy cycle is enabled when the above conditions are met and the outside air dry bulb temperature is lower than the building return air dry bulb temperature.
- The economy cycle can modulate to track supply air temperature to the extent possible given the prevailing ambient conditions, as the first stage of cooling.

PAC control

- Economy cycle is locked out above the same outdoor ambient conditions as the FCU/AHU control strategy:
  - Dry bulb temperature: 23°C
  - Dewpoint temperature: 15°C
- The economy cycle is enabled when the above conditions are met and the outside air dry bulb temperature is lower than the building return air dry bulb temperature.

- The economy cycle modulates the outside air/economy damper as the first stage of cooling linearly from 0-100% as the zone temperature increases from 22.5-23.5°C.

9.3 Results

9.3.1 Simulation results

The simulation results show increasing benefits for the cooler climate zones, with negligible benefit for climate zone 1. Overall, unsurprisingly the more nuanced control methodology for the FCU/AHU simulations provide larger savings of up to 20%, compared to PAC units at under 10%. Beyond this, the Large Office archetype presented the highest savings in the coolest climate zones – given this is the daytime archetype this benefit is likely to be attributed to the solar loads which result in building zones requiring cooling sooner and more often at lower ambient temperatures.

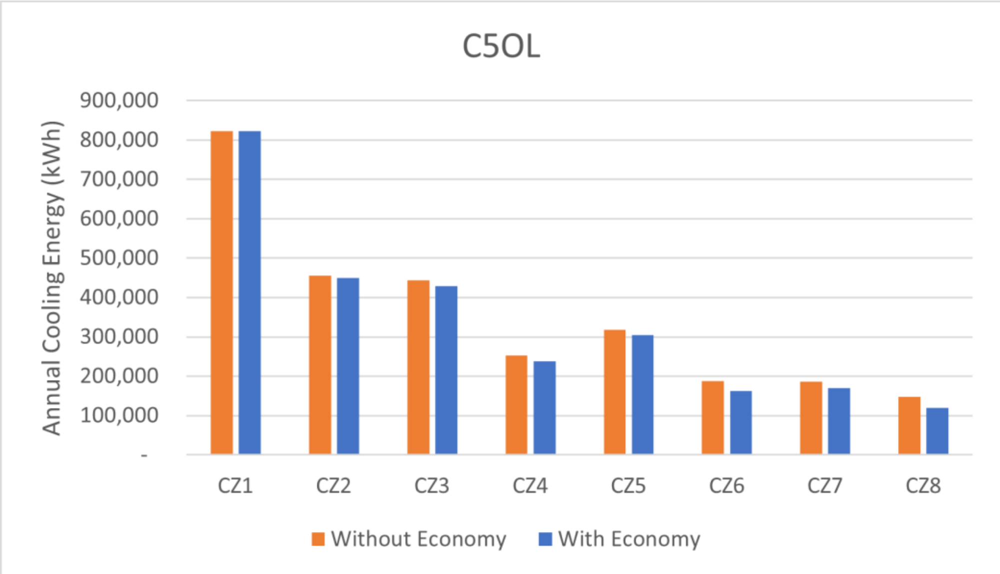


Figure 90: Large Office (daytime) simulation results for HVAC cooling energy only, comparing cases with and without economy cycle.

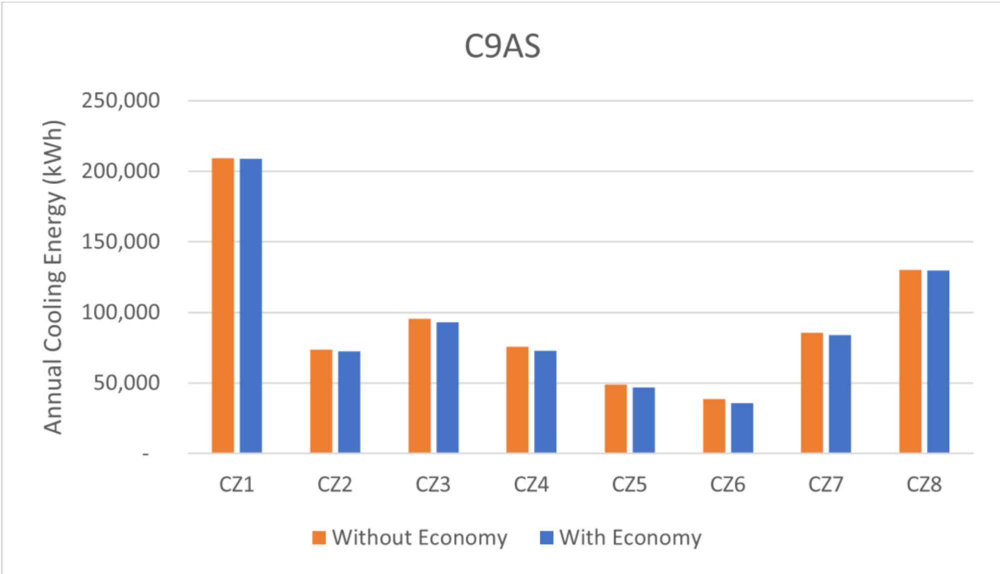


Figure 91: Small Hospital simulation results for HVAC cooling energy only, comparing cases with and without economy cycle.

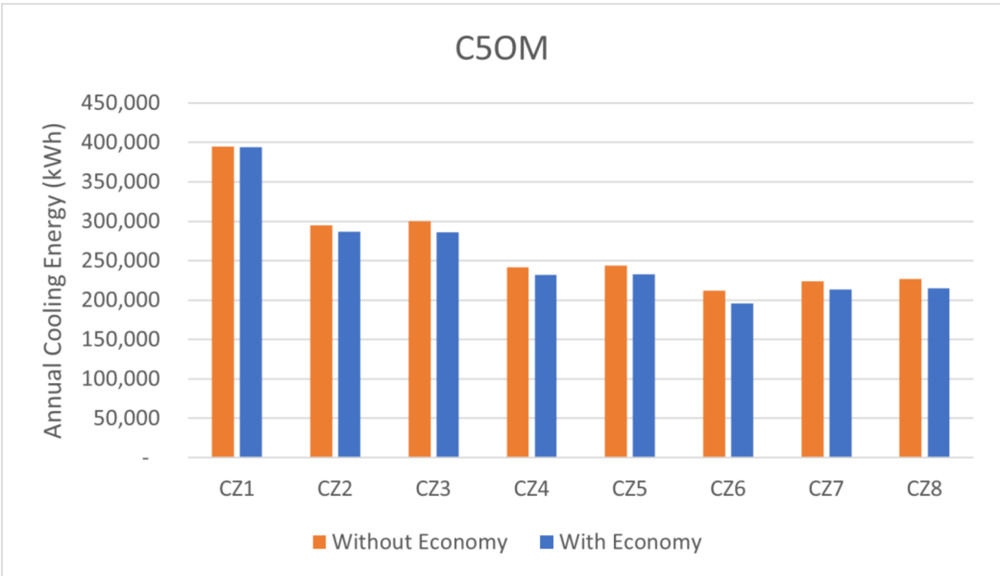


Figure 92: Medium Office simulation results for HVAC cooling energy only, comparing cases with and without economy cycle.

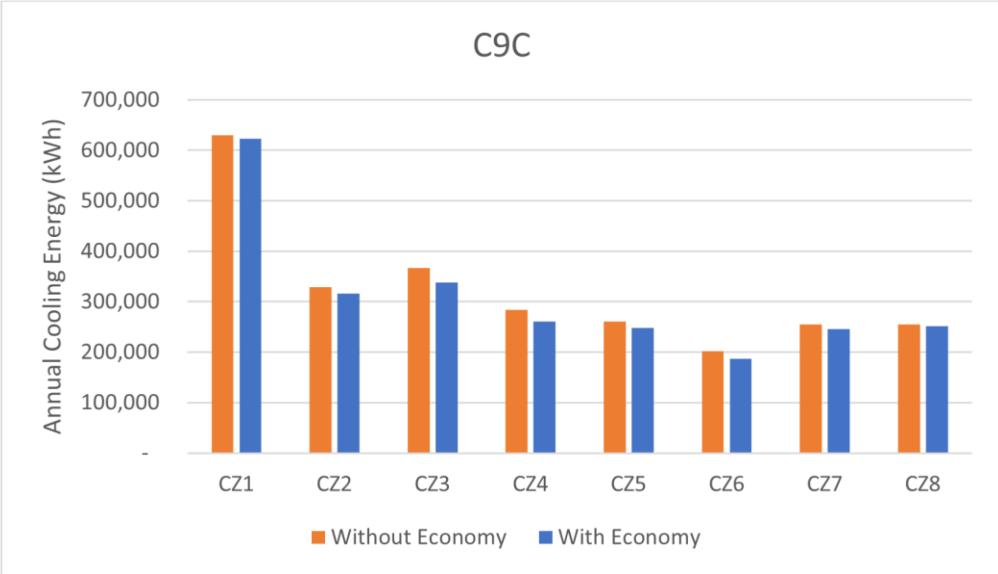


Figure 93: Aged Care simulation results for HVAC cooling energy only, comparing cases with and without economy cycle.

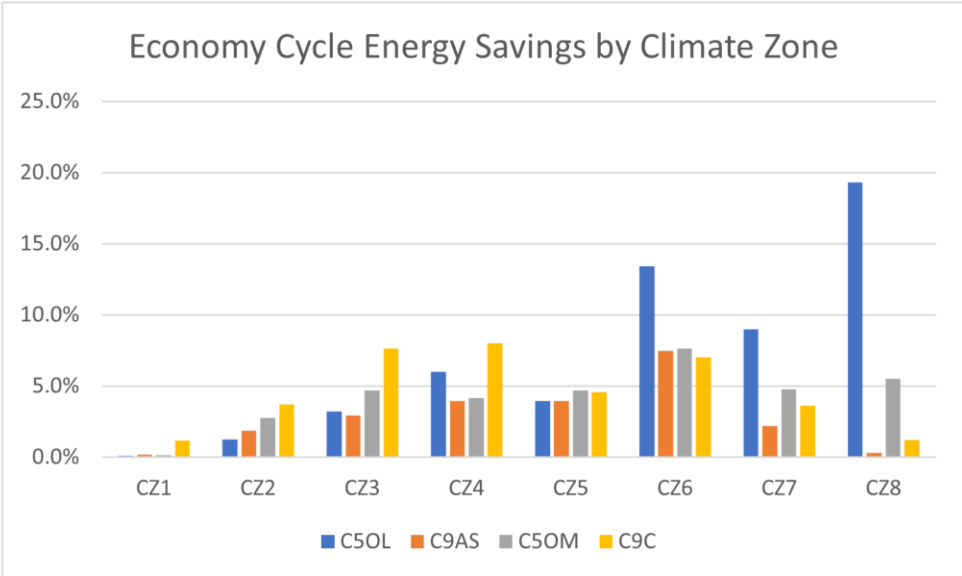


Figure 94: Percentage of HVAC energy saved by adding economy cycle in each climate zone and archetype

Tabulated results for the economy cycle simulations are listed in the Appendices, Section 18.3.

9.3.2 Benefit-cost analysis results

The results from the benefit-cost ratio (BCR) analyses somewhat mask the benefits presented in Section 9.3.1. Although the FCU/AHU economy cycles yield the best energy savings overall, their cost premium to implement does not result in a sufficiently beneficial BCR. Conversely, the significantly lower incremental cost for economy cycle in inverter PAC units yields more beneficial results.

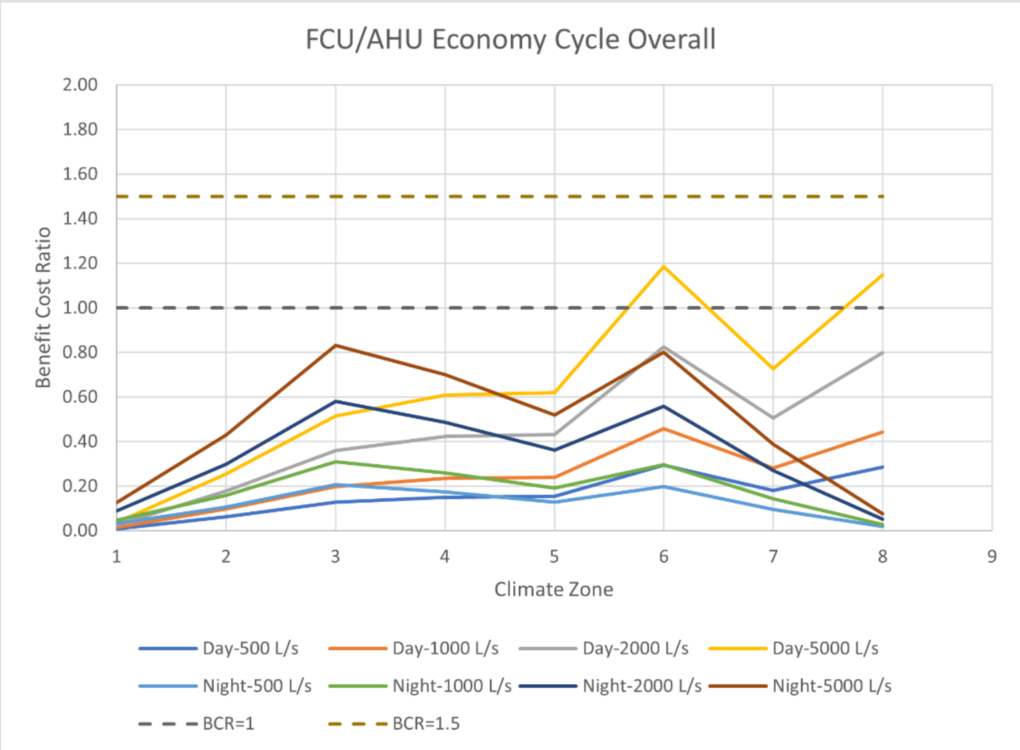


Figure 95: FCU/AHU economy cycle results combined.

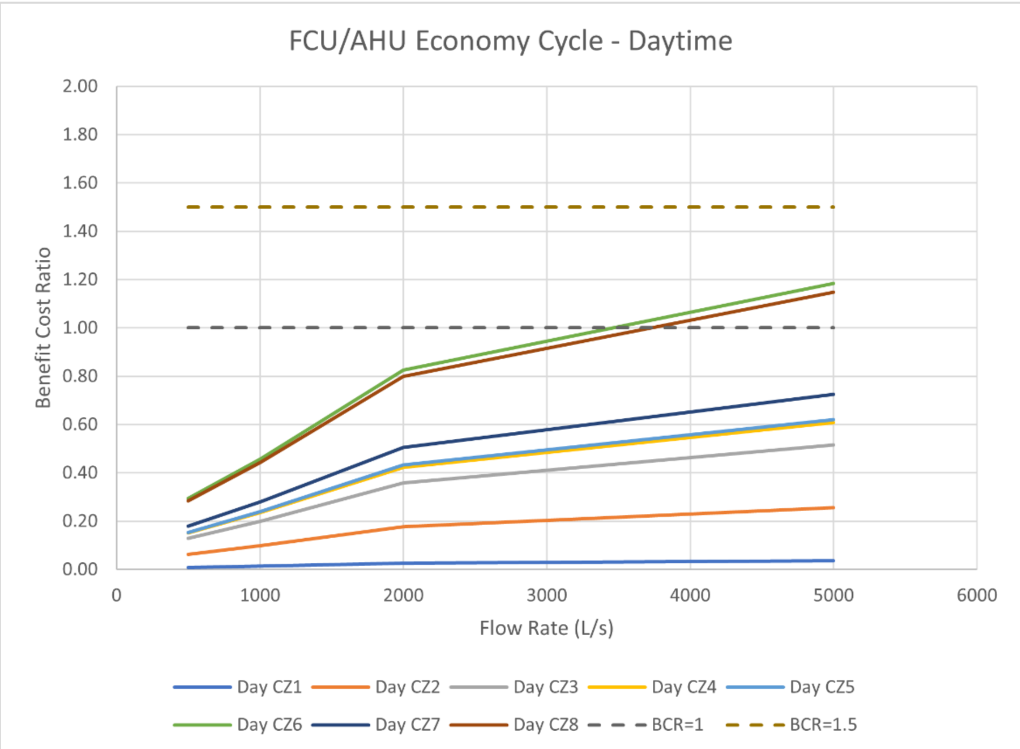


Figure 96: FCU/AHU daytime economy cycle results

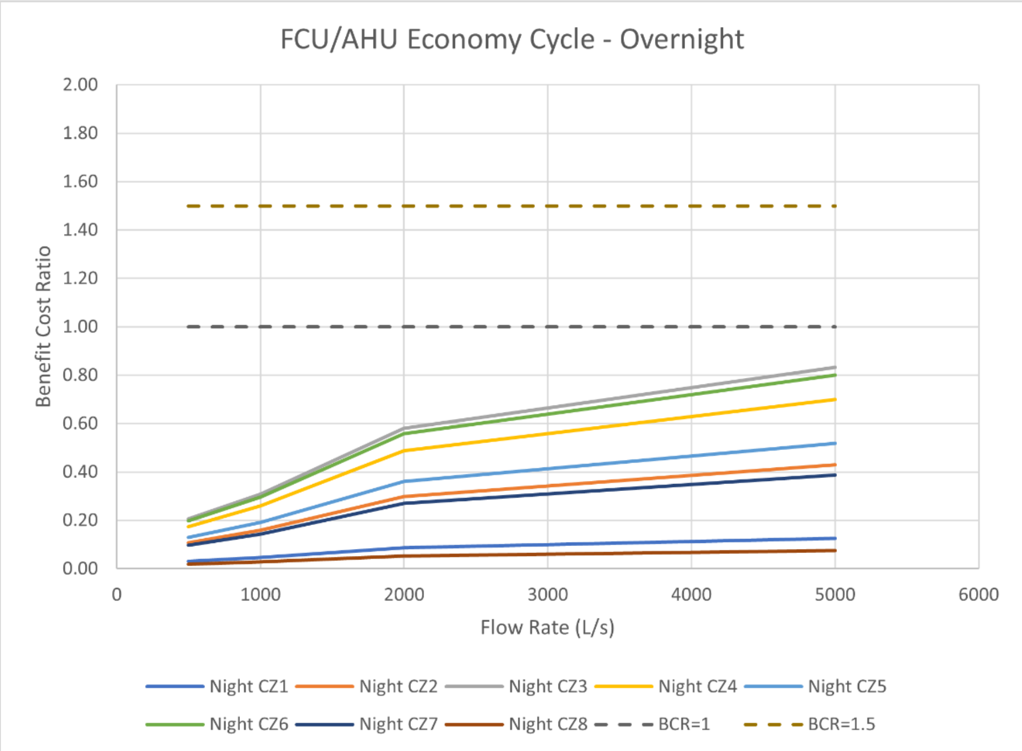


Figure 97: FCU/AHU overnight economy cycle results

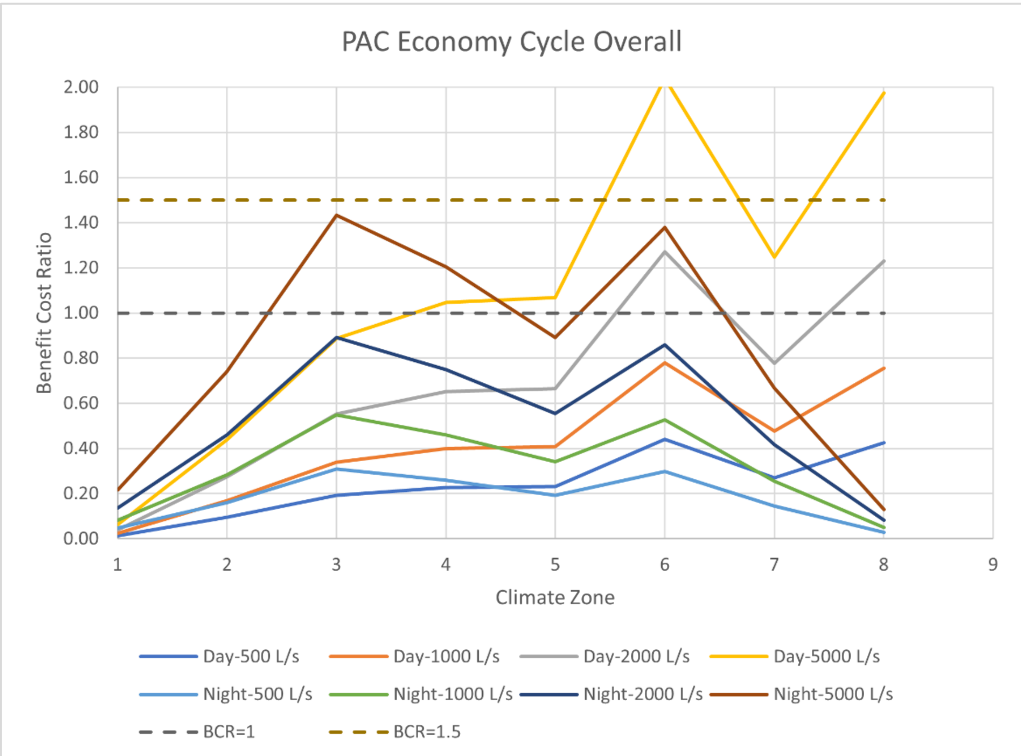


Figure 98: PAC economy cycle results combined

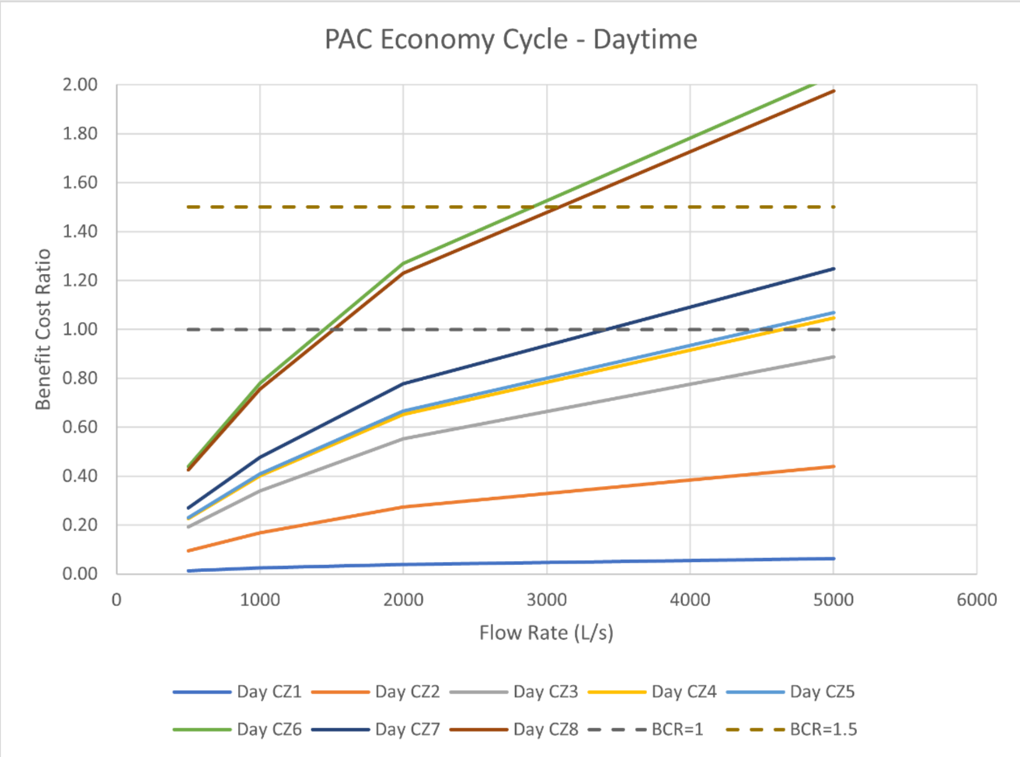


Figure 99: PAC daytime economy cycle results

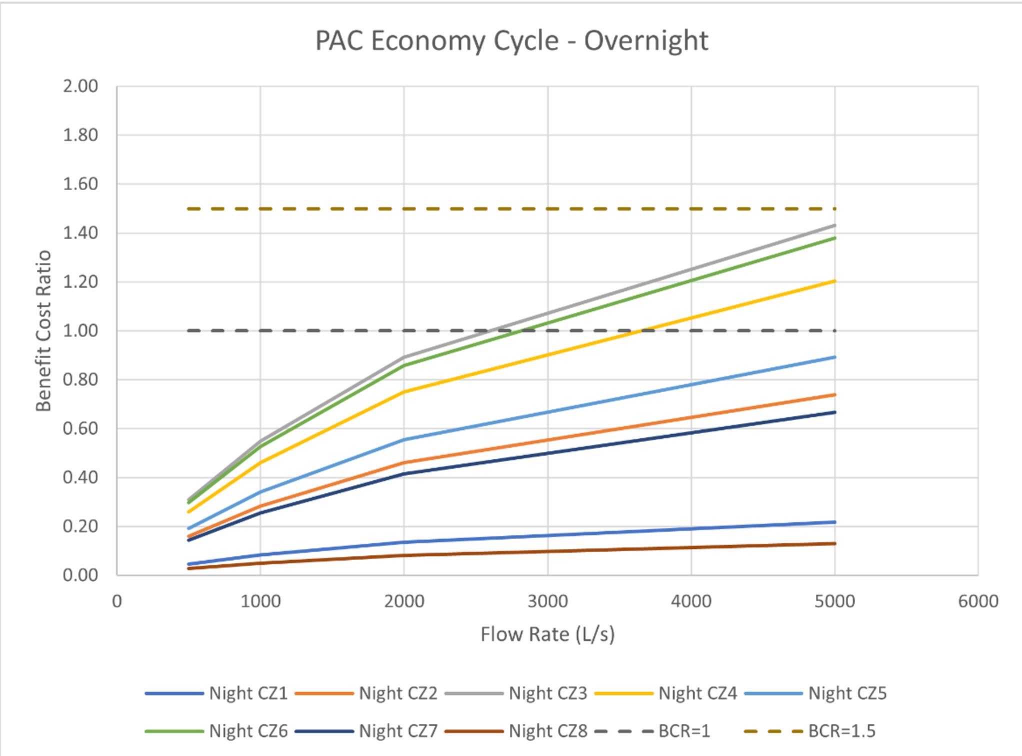


Figure 100: PAC overnight economy cycle results

Tabulated BCR figures for used in Figure 95 to Figure 100 are listed in the Appendices, Section 18.4.

9.4 Discussion

The results show a significant variable to a conclusive outcome: dependency on the type of technology used (FCU/AHU or PAC unit). The fact that FCUs and AHUs require a site and project specific economy cycle design and fabrication increases the installation cost.

The control of economy cycles is executed on a project and site-specific basis, the method of which is typically determined by a combination of project documentation (designers experience) and installer experience. There are ideal methods documented which show a preference to one method over the other[30], so the implementation of economy cycle controls (like many controls strategies) could be standardised.

The other aspect of customisation is dependent on physical installation constraints. Variations in duct sizes based on project design cooling supply air temperature, location of FCU/AHU, and interaction with other HVAC equipment, all have material install-based capital cost implications associated with them. These cost components make costing and resultant BCR somewhat volatile and dependant on the situational cost factors applied to the assessment.

By contrast, PAC units are not subject to the premium associated with customisation and therefore reap the benefit of a much lower incremental cost associated with addition of economy cycle. This benefit extends from standardisation of controls to the benefit associated with installation location. Unlike FCUs (which are typically installed in ceiling spaces of occupied areas) and AHUs (which are typically located in plant rooms) PAC units need to be located external to a building envelop for heat rejection purposes, so access to the ambient air for economy and relief purposes does not add any additional requirements for ductwork, louvres and the like.

Comparing the average daytime results with the 2022 economy cycle provisions for each climate zone individually there exists a variety of conclusions:

- Extrapolating the airflow results for climate zones 2 and 3 should yield a stringency similar to the 2022 code.
- Climate zones 4, 5 and 7 show a slight reduction in stringency compared to 2022.
- Climate zone 6 shows an equivalent stringency compared to 2022.
- Climate zone 8 shows a significant improvement in stringency compared to the 2022 code.

The positive and often significant savings associated with economy cycles make these a key candidate for consideration in the whole building stringency optimisation phase of the project.

9.5 Proposed Measures

The BCR results highlight an increased stringency for Climate zone 8 is appropriate, with the remaining climate zones remaining unchanged. It is recommended that Table J6D3 is updated as follows:

Climate zone	Total air flow rate *requiring* an economy cycle (L/s)
2	9000
3	7500

[30] Taylor, S. T. & Cheng, H. 2010, "Economizer High Limit Controls and Why Enthalpy Economizers Don't Work", ASHRAE.

Climate zone	Total air flow rate *requiring* an economy cycle (L/s)
4	3500
5	3000
6	2000
7	2500
8	2000

10 HVAC Fans

10.1 Background and context

The intended scope of review for this measure was to identify the feasibility of simplifying the measure to a single technology independent efficiency curve. This follows from work in the trajectory project that identified issues with the formulation of the FMA-ANZ promoted single efficiency curve, and proposed reviewing and improving this.

10.1.1 Current regulation

HVAC fans are currently subject to minimum efficiency requirements set out in Clause J6D5, which has the following provisions:

1. A low-pressure efficiency requirement for fans below 200Pa. This is of the format η_{min} = 0.13 x ln(p) - 0.3, where p is the system pressure.
2. A general efficiency requirement for other fans that follows the structure of European regulation EU327[31]. This is of the format η_{min} = 0.85 x (a x ln(P) – b + N)/100) where P is the fan power, a and b are constants and N is the fan efficiency grade, where a, b and N are dependent on the fan technology and configuration. The 0.85 factor is unique to the Australian regulation and is intended to characterise the difference between fan peak efficiency, which is regulated under EU327, and duty point efficiency which is regulated under NCC 2022.

In addition, there is the FMA-ANZ performance solution which extends the equation from (1) above as a general compliance equation at all pressures[32].

10.1.2 Scope of Assessment

Investigation of the available data has somewhat expanded the original scope for this assessment. The revised assessment covers:

1. The suitability of the η_{min} = 0.13 x ln(p) -0.3 formula both in its original role below 200Pa and in its extended role above that pressure.
2. The differences between the NCC 2022 general fan efficiency grade requirements versus those of EU327.
3. Viability of a technology-independent fan efficiency curve.

10.2 Methodology

The originally proposed methodology for this assessment is shown in Figure 101.

[31] EU Commission Regulation 327/2011, Table 2 (Minimum Efficiency Requirements for Fans from 1 January 2015)
[32] Only as applied to static pressure driven configurations, but these dominate in industry selections.

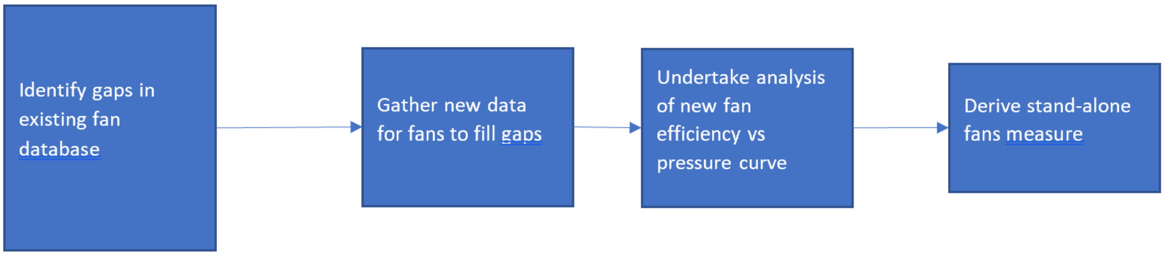


Figure 101. Fan efficiency assessment methodology.

In practice, the methodology changed somewhat due to concerns about the underlying concept of a technology neutral approach, leading to a more broad-ranging exploration of the possibilities, that was however constrained by the initial scope and budget.

10.2.1 Review of existing fan database

The existing FMA-ANZ database was compiled in 2018 and consists of 2641 fans in total, being 1004 axial fans, 1002 backward curved centrifugal fans, 603 mixed flow fans and 28 other centrifugal fans. Fan pressures were predominantly in the range 100-400Pa, as shown in Figure 102.

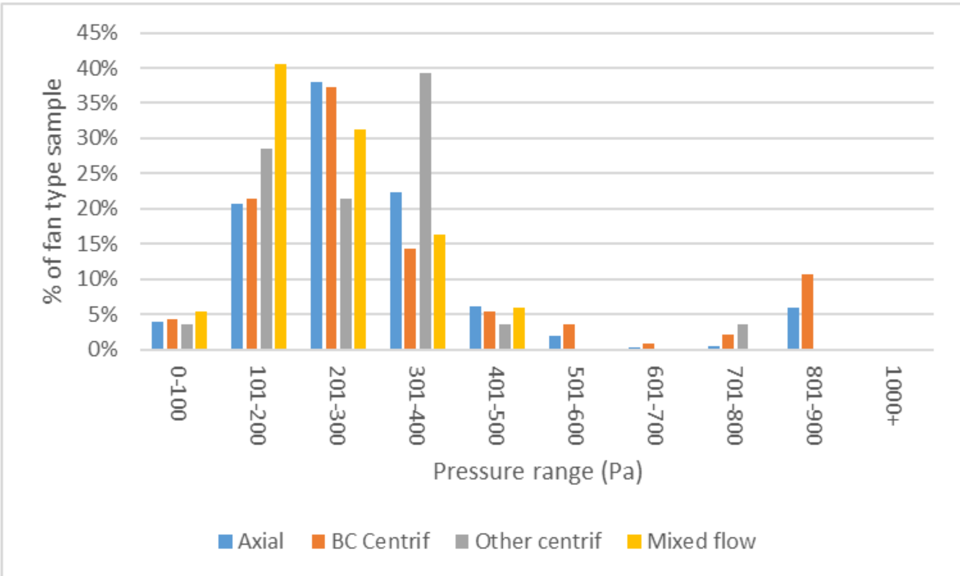


Figure 102. Fan data from FMA-ANZ database by pressure band.

The data within the database shows good representation over low to medium pressure ranges but is somewhat lacking in data in higher ranges. This must be interpreted against an increasing trend (driven, in part by the NCC provisions) of lower pressure system design. The small peak of fans in the 800Pa region may represent non-HVAC applications as pressures in this range are relatively uncommon in building services design. It is important to note that the FMA-ANZ database only provides information on static pressure and efficiency.

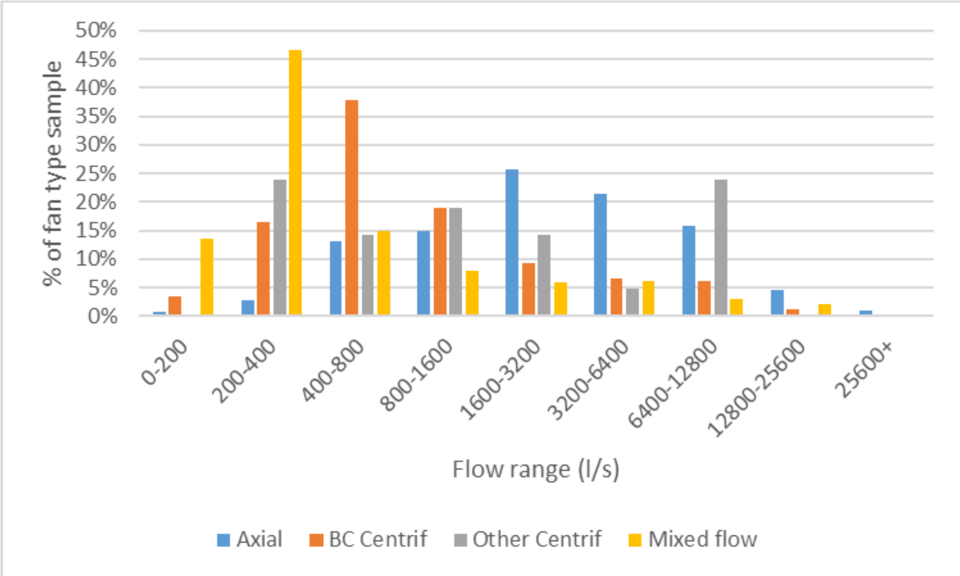


Figure 103. Fan data from FMA-ANZ database by flow range.

The data within the database shows good representation of small to medium fans but has relatively few large fans. In our enquiries with FMA-ANZ, it was noted that there is an increasing trend towards the use of multiple (plug) fans to achieve large flows rather than single large fans.

Based on the above analysis, it was considered that the database provides an adequate starting point for a review of the current provisions. It is noted that additional data was sought at several points of the analysis but did not arrive in time for incorporation in this report.

10.2.2 Suitability of the η_{min} = 0.13 x ln(p) - 0.3 formula below 200Pa (J6D5 2(a))

This formula is understood to have been originally introduced by in response to concerns about excessive stringency of EU 327 at low power levels – noting in this context that EU327 only applies above 125W. In order to test this, the degree of compliance in the FMA-ANZ database was tested against the η_{min} = 0.13 x ln(p) -0.3 formula from J6D5 2(a) and against the EU327 based formulae in J6D5 2(b).

In Figure 104, the two requirements are directly compared for fans in the database. A number of features are visible:

1. For axial fans, which form the lower group of data to the right of the graph, the J6D5 2(a) requirement is more stringent than J6D5 2(b).
2. For Backwards Curved centrifugal (BC centrif) fans, which form the upper group of data to the right of the graph, the J6D5 2(a) requirement is less stringent than J6D5 2(b).
3. For a small number of fans (generally at <50Pa static pressure) the J6D5 2(a) requirement produces exceptionally low efficiency figures, including some figures that are negative.

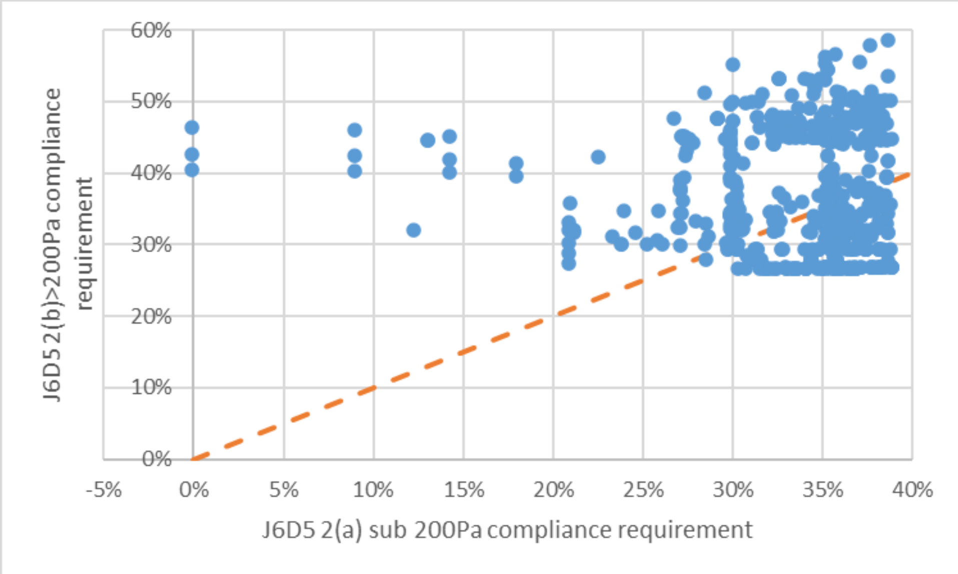


Figure 104. Comparison of compliance requirements for J6D5 2(a) (sub 200Pa) versus the equivalent J6D5 2(b) requirement.

The impact of this can be seen when the compliance of fans is tallied, as shown in Table 57: while compliance for axial fans is marginally poorer under J6D5 2(b), it is far poorer for BC centrifugal fans, and far better for mixed flow fans. The overall number of fans compliant under J6D5 2(b) is marginally higher.

Table 57. Comparison of compliance between J6D5 2(a) and 2(b).

	J6D5 2(a) compliant	J6D5 2(b) compliant	sample size
Axial	113	105	211
BC Centrif	128	21	212
Mixed flow	21	166	250
Forwards Curved (FC) Centrif	0	0	2
Total	262	292	675

Based on the results in Table 57, it would appear that the overall impact of the insertion of J6D5 2(a) into NCC 2022 does not provide any significant benefit relative to the extrapolation of the formulae in J6D5 2(b).

Based on the above, it is concluded that the η_{min} = 0.13 x ln(p) - 0.3 formula from J6D5 2(a) provides no particular value to code and should be removed.

10.2.3 Suitability of the η_{min} = 0.13 x ln(p) - 0.3 formula above 200Pa

The further extrapolation of the J6D5 2(a) formula as a basis for compliance more generally was instituted in 2020 based on work by FMA-ANZ, who used a database of fans to show that the formula represented a reasonable approximation of a mid-point to compliant fans within a database of fans they had developed.

Fundamental to the assessment of the validity of the formula, is the need to understand the relationship between it and the EU327 compliance curves. This relationship is not clear as, although the two formulae have the same structure, one uses pressure as the primary variable while the other uses fan power (which is a function of both pressure *and* flow). This creates the possibility that the pressure-based formula may fail to represent suitable compliance levels across the full range of flows. This was investigated and was found to be a justified concern, as shown Figure 105.

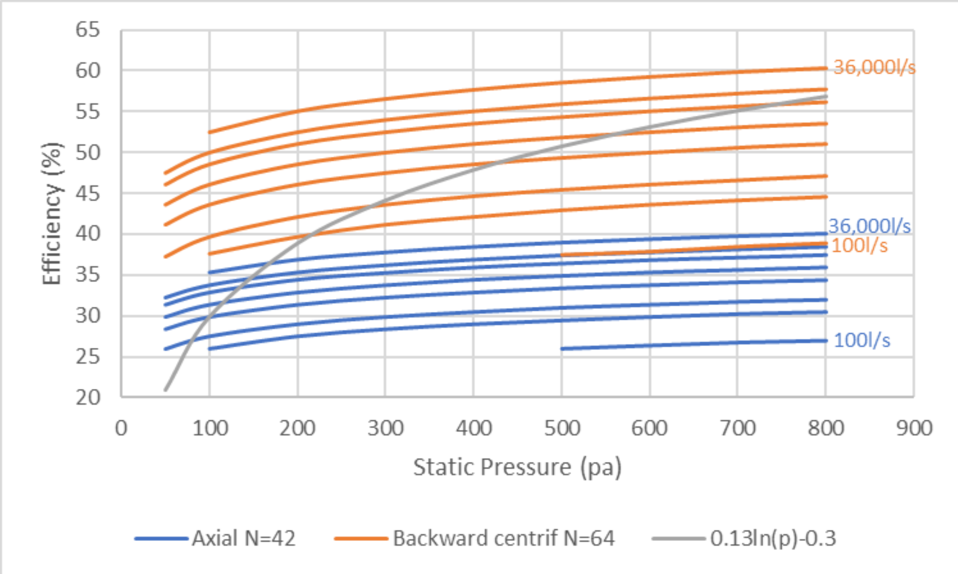


Figure 105. Comparison of NCC 2022 efficiency requirements with the 0.13ln(p)-0.3 formula. Lines area at 100, 500, 1000, 3000, 6000, 12000, 18000 and 36000l/s.

It can be seen in Figure 105 that the NCC 2022 requirements are poorly reflected by the 0.13ln(p)-0.13 formula, with the formula only reasonably representing a small flow range at a given pressure. Importantly, the formula leads to a significantly lower efficiency requirement for large BC centrifugal fans, which are some of the most important and common fans in HVAC applications.

On this basis, it is recommended that the use of this formula is inappropriate and should not be extended to NCC 2025, and indeed should be reconsidered for NCC 2022.

10.3 NCC 2022 vs EU 327

While NCC 2022 follows the structure of EU327, it has a number of differences in fan categorisation and fan efficiency index figures, as summarised in Table 58.

Table 58. Comparison of efficiency grade (N) figures from NCC 2022 and EU327.

Fan type	Installation type A, C - NCC 2022	Installation type A, C - EU327	Installation type B, D -NCC 2022	Installation type B, D - EU327
Axial (AHU/FCU)	46	40	51.5	58
Axial (other)	42	40	61	58
Mixed flow (AHU/FCU)	46	50	51.5	62
Mixed flow (other)	52.5	50	65	62
FC centrif	46	44	51.5	49
Radial centrif	46	44	51.5	49
BC centrif (w/o housing)	64	62	64	-
BC centrif (w/ housing)	64	61	64	64
Cross flow	-	-	-	21

The rational for these differences can be traced to work undertaken by Energy Action in 2017, which in turn used the 2015 Carbon Credits Methodology Determination to propose stringencies that largely reflect the high efficiency category in that determination. The Energy action report identifies these as having been set at 5% higher than EU327 with the exception of FC centrifugal which was set at EU327. The Energy Action report also set the AHU/FCU requirements for mixed flow and axial fans in AHUs and FCUs based on FC centrifugal fans.

The current EU327 requirements date from 2011 and were effective from 2015, and thus are eight years old at the time of writing. Literature review indicates that a draft update of EU327 was prepared but has never been ratified. The draft update proposed the fan efficiency grades listed in Table 59. These higher efficiency figures are generally higher than NCC 2022 figures; as the proposed figures were never ratified, they do not provide a suitable starting point for NCC 2025.

Table 59. Draft fan efficiency grades for an unimplemented update of EU327. α is the fan flow angle.

Fan type	Installation type A, C	Installation type B, D
Axial	50	64
FC/radial centrif<5kW	52	57
FC/radial centrif>5kW	64	67
BC centrif	64	67
Mixed flow	57+7(α-45)/25	67
Cross flow	-	21

In our consultation with FMA-ANZ, suppliers highlighted the desirability of having the NCC reflect EU327 as this allows alignment of their supply chains, which is not an unreasonable request. However, it would appear retrograde to propose reducing the NCC 2022 figures to EU 327.

The major complication in NCC 2022 relative to EU327 is the introduction of the AHU/FCU categories for axial and mixed- flow fans. The purpose of this change is unclear, given that it has variable impacts on the required fan efficiency grade relative to the balance of installations for the given fan types. In discussions with FMA-ANZ, an alternative us using the BC Centrif requirements for air handlers was discussed, which more explicitly reflect the reality that this fan type dominates AHU applications and other fan types can reasonably be argued as unusual; a further benefit is that the use of such an approach does not reduce any efficiency requirement.

In a similar vein, the absence of cross flow fans from NCC 2022 is a further difference with EU 327. This fan type has relatively limited applications (particularly air curtains) and there is a reasonable argument that the application of this fan type should be accordingly limited.

10.3.1 Below 125W

One of the purposes of the J6D5 2(a) formula was to deal with fans below 125W. This is justifiable on the basis that EU327 is only valid from 125W-500kW. There is a reasonable argument that fans below 125W are not material energy consumers in most buildings, which would support a position of excluding these from the analysis, although for certain building types such as fan coil units fan energy can be mores significant. An alternative argument would be that NCC 2022 current does regulate fans below 125W, seemingly without significant disruption. Removal of the J6D52(a) formula would change the nature of compliance, but of 439 fans in the database below 125W, 154 comply under J6D5(2) while 200 comply using the extrapolation of J6 D5 2(b) to all pressures.

On this basis it is proposed to retain the current coverage of fans below 125W.

10.3.2 Technology neutrality

A significant original intent of the current analysis was to move towards a technology neutral methodology for fans. The potential benefit of such an approach would be to avoid the possibility that designers make inappropriate selections of fans for given applications, typically characterised by the use of axial fans in high static applications (where the efficiency difference versus BC centrif is large).

There are a number of hurdles to the definition of such as approach:

1. There may be well defined application reasons for the selection of an axial fan in a high static application. The most significant of these appears to be related space requirements: axial fans are more compact than BC centrifs and in very high-volume applications this can become very significant.
2. There may be cost implications. In general, centrifugal fans are more expensive than axial fans so forcing the use of centrifugal fans has potential cost implications, which are out of scope for the current study.
3. There may be negative impacts. While general industry practice is that axial fans are the preferred selection for low pressure applications, low pressure centrifugal fans are available (as shown in Figure 102). A technology neutral approach that does not effectively mandate the use of such fans would in effect relax the efficiency requirements on such fans.
4. The degree to which inappropriate fan selections are prevalent is unknown.

These factors means that a true technology-neutral approach would be both difficult to create and of unknown benefit[33], other than the simplifying effect upon code text.

10.3.3 Individual fan efficiency versus average efficiency

For equipment such as PAC units and VRF systems, the use of a building-average efficiency has been proposed to overcome some of the challenges of requiring a minimum efficiency. This approach can also be extended to fans in a similar capacity-weighted manner, although the application is marginally more complex.

The benefit of using an average approach is that it makes the individual requirements placed on each individual fan less critical; given the complexity of the issues associated with fan selection, this is highly desirable as it is very hard to assess all unintended consequences.

10.3.4 Selection point versus fan peak efficiency

NCC 2022 currently assesses fans on the basis of the efficiency at the duty point as opposed to the peak fan efficiency. The difference between these two is captured in the 0.85 factor applied, which permits fans to have a duty efficiency of down to 85% of the fan peak efficiency. However, the in-practice application of this requirement appears not to meet the original intent, because the duty point selected by designers is consistently (and often substantially) oversized relative to the expected

[33] The similarity of the requirements for type B/D installations means that definition of a technology neutral requirement is arguably non-controversial. However, it would have little impact precisely because the requirements for different fan types are relatively similar.

duty. For this requirement to have real impact, compliance must be based on the expected fan performance at the expected duty not at the fan selection point.

An example of a fan selection chart is shown in Figure 106 below. In this case, the fan duty was specified as 1200l/s at 260Pa, as shown by the intersecting red lines; the original specification also requested allowance for 10% spare capacity, which is represented by the blue circle. The fan selection, however, was made at 1468l/s at a pressure of 390Pa, which is a considerably higher duty.

In practice, there is an expectation that the commissioning process will address this oversizing by one of two methods:

1. The fan will be equipped with a VSD, which will be speed limited to achieve the required flow. This will move the operating point down the black line to the 1200l/s, 260Pa design duty without compromising the fan's efficiency.
2. A damper will be closed in the system against the fan to force the pressure higher until the flow is limited to 1200l/s. This is equivalent to moving up the blue line until the 1200l/s line is reached. This process decreases the fan efficiency, and increases the fan pressure duty, with the result that the fan power is higher than it would be had a VSD been used.

As a result, a robust NCC measure would include an explicit methodology to ensure that the fan performance at duty point is reflected in the compliance criteria.

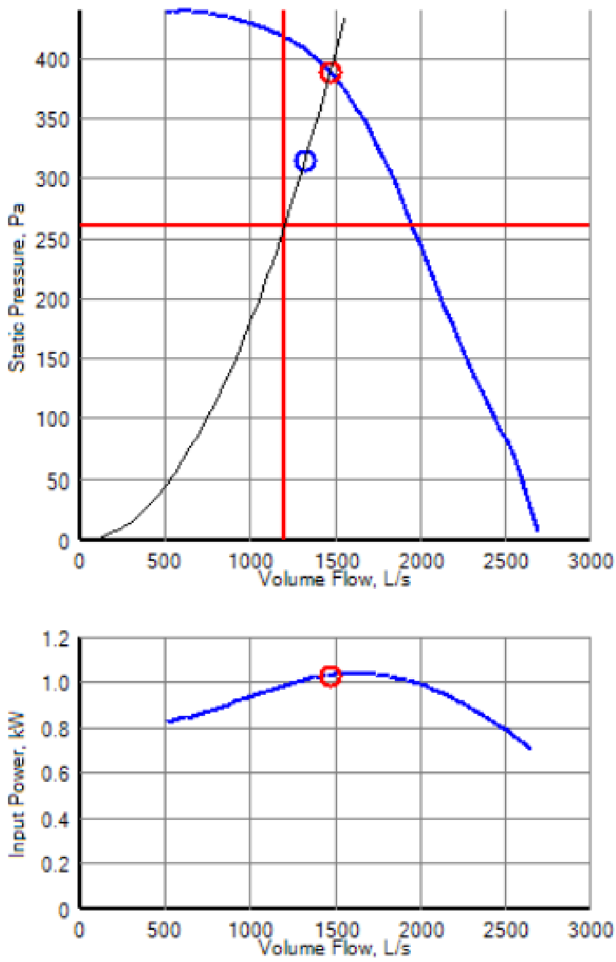


Figure 106. Typical fan selection diagram. The intersection of the red lines represents the expected design duty.

10.3.5 Limitations of EU327/J6D5 2(b)

One of the challenges in interpretation of J6D5 2(b) is that it does not provide a reference for a measurement standard. This is especially important as EU327 utilises a number of modifiers to fan efficiency to calculate the efficiency used in compliance. This means that the efficiency relevant to EU327 is not necessarily the same as the efficiency measured in practice for the same fan. Using local standards, the appropriate standard for evaluation of efficiency in line with EU327 is AS/NZS ISO 12759:2013.

By contrast, the standard used for measurement of efficiency as reported in fan data sheets is ISO5801.

10.3.6 Real fan selections versus theory

As part of the development of this measure, a major fan manufacturer was approached to provide a range of fan selections in order to provide some grounding of realistic selections against the EU327 requirements. The original purpose of the selections was to investigate the potential for technology-neutral fan provisions, which was subsequently discarded as a line of enquiry. Selections were requested in "standard" and "better" categories, with the standard selection representing a normal industry selection compliant with EU327 and the better category representing a more efficient solution for the same application, typically using a different fan technology. The selections, covering 6 flows from 500-24000l/s and 75-550Pa, provide a suitable reference dataset to understand the relationship between EU327 minimum efficiency levels and the real-world fan used to meet these levels.

Fans were selected using normal practices, with the result that there were oversized relative to duty. In processing the data, we have made the important assumption that all fans have VSDs and therefore their efficiency at duty point is the same as their efficiency at selection point. Furthermore, when considering real selection against a minimum standard, it is inevitable that the real selection will be of a higher efficiency by some margin, as fans do not in general have the exact efficiency of the minimum standard.

As a consequence of these two factors[34], it was found that the real fan efficiencies were actually significantly higher, on average, than the EU327 efficiencies evaluated at the same duty power point. This is illustrated in Figure 107.

[34] Note also that as per the discussion in section 10.3.5, the two efficiency figures are measured using different standards, which may also affect results.

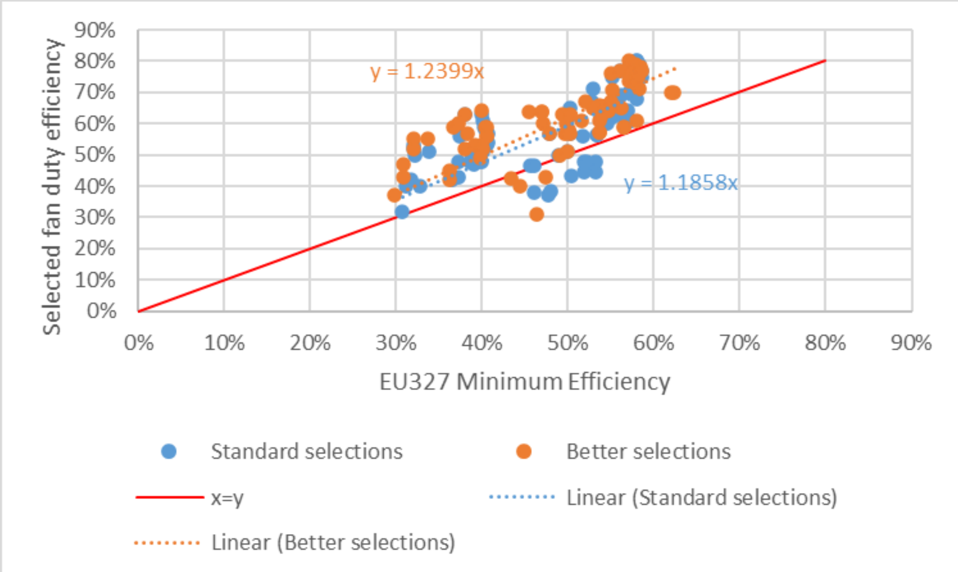


Figure 107. Comparison of selected and minimum efficiency figures for real fan selections

It can be seen in the figure that the standard selections had marginally lower "excess" efficiency than the better selection (which were often effectively standard selections but using different technologies).

If this is re-expressed as a ratio of fan power at duty point, it is the inverse of the efficiency ratio. Using the lower standard selection figure, this means that the average ratio of selected fan power to EU327 reference fan power is 84%.

10.4 Discussion

It is clear that the issue of fan efficiency is complex and that the analysis performed here does not capture all of the issues, and specifically is hindered by the lack of a supporting cost-benefit analysis. However, it appears feasible to make the following key findings:

1. The pressure-based formulation in J6D5 2(a) should be removed from code
2. The efficiency requirement should be split into two components, being:
    a. A minimum standard for fan peak efficiency, which has the effect of ensuring that individual fans meet minimum standards.
    b. A standard for fan efficiency at the expected duty point, which has the effect of ensuring that fan selections are appropriate for the expected duty.
    c. A process that permits averaging of efficiencies across all fans relative to a separate in-duty fan efficiency requirement
3. Some applications should be limited to avoid the worst potential outcomes of the technology dependent nature of measures, in particular:
    a. Fans in air handling units must always achieve compliance equivalent to that of a BC centrifugal
    b. Cross flow fans are only permitted in applications where no other fan can reasonably function, such as air-curtains.

There is clearly potential for further stringency increase based on the data, but in the absence of cost data it is unknown as to whether there is a cost beneficial case in doing so.

10.5 Proposed Code Text

The following modifications to Code text are proposed:

J6D5 Fans and duct systems

(1) Fans, ductwork and duct components that form part of an air-conditioning system or mechanical ventilation system must—

(a) separately comply with (2), (3), (4) and (5); or

(b) achieve a fan motor input power per unit of flowrate lower than the fan motor input power per unit of flowrate achieved when applying (2), (3), (4) and (5) together for each individual system; or

(c) achieve a total fan motor input power per unit of flow rate across all air-conditioning systems lower than the total fan motor input power achieved when applying (2), (3), (4), and (5) together for all air-conditioning systems.

(d) For the purposes of sub clauses (b) and (c), the fan power used must be the fan power P_{dp} at the design duty of the fan as calculated under Specification NN.

(2) Minimum efficiency of individual fans:

(a) The following fans are excluded from requirements (b)-(d) of this clause

(i) Roof mounted ventilation fans with cowling

(ii) Supply air fans provided in association with a unitary air-conditioning system that is covered under the provisions of *

(iii) Fans that are required to be explosion proof

(iv) Fans below 125W

(b) Each fan must have a peak efficiency as measured under AS12759:2013 not less than the efficiency calculated with the following formula:

$$\eta_{min} = a \times \ln(P) - b + N$$

(c) In the formula at (b)—

(i) η_{min}= the minimum required system static efficiency for installation type A or C or the minimum required system total efficiency installation type B or D; and

(ii) P = the motor input power of the fan (kW); and

(iii) N = the minimum performance grade obtained from Table J6D5a; and

(iv) a = regression coefficient a, obtained from Table J6D5b; and

(v) b = regression coefficient b, obtained from Table J6D5c; and

(vi) In = natural logarithm.

Table J6D5a: Minimum fan performance grade

Fan type	Installation type A or C	Installation type B or D
All fans a component of an air handling unit or fan coil unit	61	64
Axial	40	58
Mixed flow	50	62
Centrifugal forward — curved	44	49
Centrifugal radial bladed	44	49
Centrifugal backward-curved	61	64

Table Notes

(1) Installation type A means an arrangement where the fan is installed with free inlet and outlet conditions.

(2) Installation type B means an arrangement where the fan is installed with a free inlet and a duct at its outlet.

(3) Installation type C means an arrangement where the fan is installed with a duct fitted to its inlet and with free outlet conditions.

(4) Installation type D means an arrangement where the fan is installed with a duct fitted to its inlet and outlet.

Table J6D5b: Fan regression coefficient a

Fan type	Fan motor input power < 10 kW	Fan motor input power ≥ 10 kW
Axial	2.74	0.78
Mixed flow	4.56	1.1
Centrifugal forward-curved	2.74	0.78
Centrifugal radial bladed	2.74	0.78
Centrifugal backward-curved	4.56	1.1

Table J6D5c: Fan regression coefficient b

Fan type	Fan motor input power < 10 kW	Fan motor input power ≥ 10 kW
Axial	6.33	1.88
Mixed flow	10.5	2.6
Centrifugal forward-curved	6.33	1.88
Centrifugal radial bladed	6.33	1.88
Centrifugal backward-curved	10.5	2.6

(d) The Fan Power Ratio (FPR) calculated in Specification NN shall be less than 0.84.

Specification NN: Fan Power Ratio (FPR)

1. This calculation includes all fans covered under J6D5 with the exception of those specified in J6D5 2(a) but the inclusion of fans below 125W.
2. For the purposes of this section
    a. The duty point (dp) for a fan shall be the design duty in pressure p_{dp} and flow Q_{dp} that is used as an input to the fan selection process
    b. The selection point (sp) for a fan shall be the pressure and flow at the point at which the fan has been selected
3. Duty point fan power calculation:
    a. For a fan with a variable speed drive the power at design duty point is calculated using η_{sp}, the fan efficiency as measured to ISO5801 at the selection point i.e.
    $$P_{dp} = \frac{p_{dp} Q_{dp}}{\eta_{sp}}$$
    b. For a fan without a variable speed drive the power at the design duty point is calculated using η_{dp*} which is the efficiency of the fan at flow Q_{dp} and pressure p_{dp*} which is the pressure on the fan curve at which the fan flow equals Q_{dp}, i.e.,'
    $$P_{dp} = \frac{p_{dp*} Q_{dp}}{\eta_{dp*}}$$
4. Reference fan calculation
    a. The reference fan power P_{rf} at the design duty point shall be calculated as follows:
    $$P_{rf} = \frac{p_{dp} Q_{dp}}{\eta_{rf}}$$
    Where p_{dp} is the pressure at the duty point, Q_{dp} is the flow at the duty point and η_{rf} is the efficiency of the reference fan at P_{dp}.
    b. The efficiency of the reference fan at duty point shall be calculated using the formulae in J6D5 2 (b) using the input power P_{dp} calculated in step 2 above.
5. Fan Power Ratio Calculation. The fan power ratio (FPR) shall be calculated as the sum of the input power at duty point for all fans divided by the sum of the reference power at duty point for all fans, i.e.
$$FPR = \frac{\sum_{all\,fans} P_{dp}}{\sum_{all\,fans} P_{rf}}$$

11 HVAC Zoning

11.1 Background and Context

NCC 2022 makes no specific reference to how air-conditioning systems are zoned. This means that, to the extent that HVAC energy use is dependent on the configuration of zones, designers may be able to adopt poorly zoned designs that significantly increase energy use. The intent of this measure is to define a baseline for HVAC zoning practice to remove this potential loophole.

The original motivation for this measure comes from earlier work contemplating a whole-of-HVAC COP. As part of that work, it became clear that a baseline assumption has to be made for HVAC zoning in order to calculate a reference HVAC COP. While the whole-of-HVAC thesis was ultimately rejected, there is still validity to the concept of there being explicit baseline assumptions in Code for HVAC zoning practices.

There are interactions between this measure and:

- **Verification methods and reporting:** While this measure deals with zoning of whole air-conditioning systems, the verification methods to some extent assume that simulation models have representative zoning for individual thermal control zones. Some gaming has been reported in relation to the reporting of PMV results in this respect, for instance.
- **Variable speed fans:** As will be demonstrated in this assessment, there are significant differences in the response of variable volume and constant volume air-conditioning systems to different zoning approaches. This to some extent provides an argument in favour of a level of mandatory use of variable volume control.
- **Reheat limits.** Clause J6D3 (b) has requirements that zones that different loads must have separate temperature control and are limited to 7.5°C of reheat. This passively sets some limits on zoning configuration, as discussed below.

The assessment scope for this section is limited, in that no benefit-cost analyses are included.

11.2 Methodology

The methodology for this assessment is shown in Figure 108.

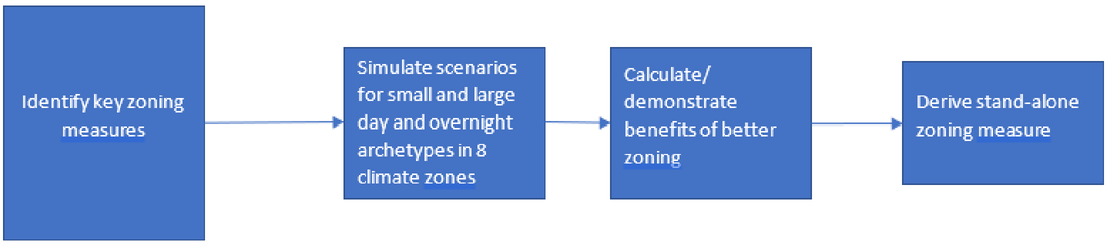


Figure 108. Outline methodology for HVAC zoning measure.

11.2.1 Test cases

C5OL archetype

For the C5OL (Large Office) archetype, the original archetype building has 5 variable air volume air conditioning systems with central plant, each serving all floors for a single façade aspect (north, east, west, south) or the centre zone. This represents what is generally considered to be best practice.

C5OM archetype

For the C5OM archetype, the original archetype has 10 PAC units serving a single floor each of a façade aspect (north, east, west, south) or the central zone. PAC units are modelled as constant volume airflow with reheat.

Test cases

Test cases for both archetypes were as follows:

- 5Z: base case with 5 AHUs
- 1Z: Single AHU – all zones served by a single air-conditioning system.
- NW, SE, C: Three AHUs – serving 1. north and west; 2. south and east; 3. Centre.
- NC, E, W, S: four AHUs – serving 1. north and centre; 2. east; 3. west; 4. south.
- EC, N, W, S: four Ahus – serving 1. east and centre; 2. north; 3. west; 4. south.
- WC, E, N, S: four AHUs – serving 1. west and centre; 2. east; 3. north; 4. south.
- SC, E, W, S: four AHUs – serving 1. South and centre; 2. East; 3. West; 4, north.

The C5OL archetype retained the multi-floor configuration of AHUs in each case; the C5OM archetype retained the single floor configuration of PAC units in each case.

Economy cycles and OA control

As various AHUs/PACs are aggregated, they can cross thresholds for economy cycle control under Table J6D3 and CO_2 control under Table J6D4. This means that in some cases, configurations with aggregated AHUs have economy cycles and/or CO2 control where the disaggregated AHU scenarios do not. This leads to some variability in results, which is realistic from a pure code compliance perspective and thus has been retained. This reduces the differentiation between the disaggregated and aggregated AHU results, and indeed may cause some of the aggregated AHU scenarios to exceed the performance of the base case. In practice, however, designers will tend to maintain a consistent strategy for these design aspects.

Fan efficiency

Fans have been selected to NCC 2022 compliance, which means that for larger aggregated AHUs the selected fans are generally more efficient than for disaggregated AHUs. This further detracts from the expected energy use differences between aggregated and disaggregated AHU scenarios.

11.3 Results

11.3.1 C5OL (Large Office) archetype

The results for the C5OL (Large Office) archetype are shown in Figure 109.

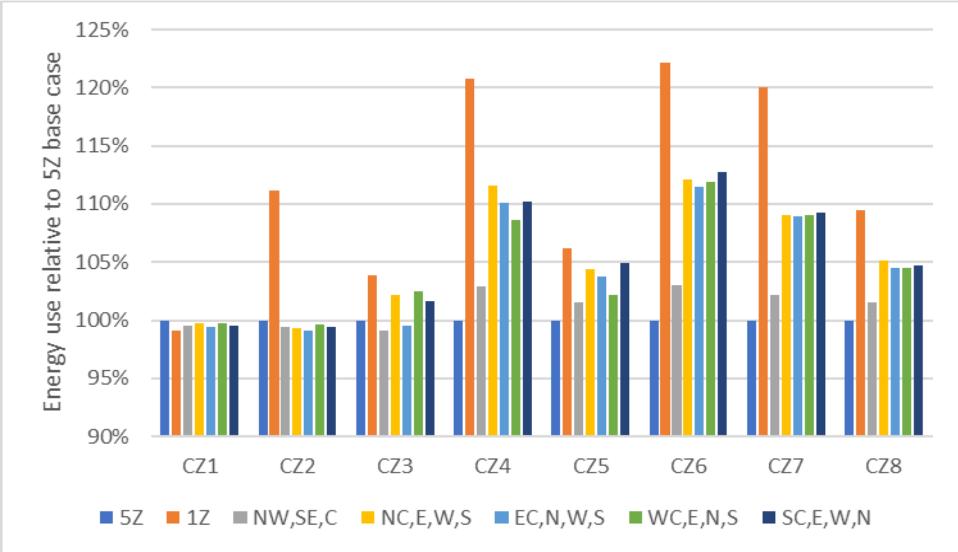


Figure 109. Energy use relative to the base case (5Z) archetype for C5OL (Large Office) across different scenarios of AHU aggregation.

A number of features are visible from Figure 109.

1. Zoning impacts for climate zone 1 are minimal. This is because the warm external temperatures limit the level of differentiation between perimeter and centre zone loads.
2. The single AHU scenario is notably less efficient than the base case for climate zones 2-8
3. The 3 AHU scenario with two perimeter AHUs is less efficient in climate zones 4-8 (which have more substantial heating loads) only.
4. The 4 AHU scenarios generally perform less well than the 3 AHU scenario, particularly in CZ4, 6,7 and 8.

11.3.2 C5OM (Medium Office) archetype

The results for the C5OM (Medium Office) archetype are shown in Figure 110.

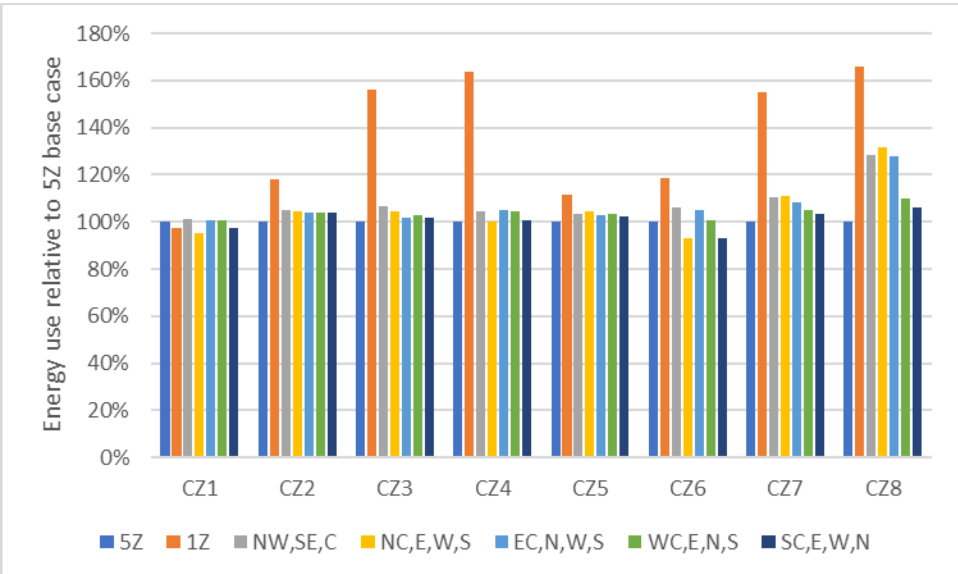


Figure 110. Energy use relative to the base case (5Z) archetype for C5OM (Medium Office) across different scenarios of AHU aggregation.

Some similar trends to those discussed in Section 11.3.1 can be observed, including the lack of impact in CZ1 and the particularly poor performance of the single AHU scenario. Surprisingly, the impact of zoning on energy use appears less pronounced than in the C5OL case. This appears to be because the PAC units, being constant volume, do not have the ability to vary airflow in response to different zone demands. This means that energy use is more consistent across all scenarios, but at a lower level of base efficiency. In some CZ3&4 single zone cases, reheat ΔT figures were higher than the 7.5°C maximum permitted under J6D3 (1)(b)(iii).

11.3.3 Discussion

Simulation results

For climate zone 1, it is clear that there is no merit in an HVAC zoning measure.

For climate zones 2-8 the results are somewhat ambiguous as to the merits of an HVAC zoning measure, except for the single AHU case, which reliably performs poorly. Other cases have variable, if generally poorer, efficiency performance relative to the base case but not in a manner that is easily characterised.

As a result, it is difficult to argue the validity of a HVAC zoning measure based on demonstrable or reliable energy savings.

Interaction with Clause J6D3(b)

It is arguable that this clause already places significant constraints upon zoning, as the 7.5°C limit for reheats significantly limits that ability for an AHU to service zones with radical differences in load, and specifically any configurations where one zone is in heating while others in cooling.

As an example of this, consider the sizing of a reheat element for a combined centre-perimeter AHU. Typically one might expect that the centre zone would be designed for a minimum supply air temperature of 14°C while the perimeter might have a maximum heating temperature of 28-30°C. A diligent designer would want to be able to design the system so that it can service the situation of a densely populated office (thereby, with significant cooling loads in the centre zone) in winter (thereby, with significant heating loads in the perimeter). If the centre zone requires 14°C supply air then Clause J6D3(b) limits the maximum possible air supply temperature in the perimeter to 14+7.5=21.5°C. This is not sufficient to provide heating to the zone[35].

Interaction with variable speed fans

Normal practice for most situations involving an air-conditioning system serving diverse loads is to provide variable air volume control at each zone (with reheat in some cases). Currently Code is silent on this other than the requirement of separate temperature control potentially using reheat underlying Clause J6D3(b). It would appear sensible to echo industry practice in a revised J6D3 by requiring that variable air volume delivery is used ahead of reheat as a means of modulating servicing.

Interaction with verification methods

For both Clause J6D3 and the intent of the JV methodologies to be more consistent and rigorous, there needs to be some baseline identification of what constitutes a thermal zone. Current gaming in this area relies on combining perimeter and centre zones into a single zone.

[35] IES appears to size reheats on a "just-enough" basis using design day data, which will tend to result in smaller reheats. By contrast, designers tend to size more conservatively (i.e. worst case) which is rational from a functional perspective.

Definition of an air-conditioning system

NCC 2022 defines air-conditioning as a service that actively cools or heats the air within a space, which falls somewhat sort of defining an air-conditioning system in the manner necessary for a zoning measure. Subtleties that occur include:

- An AHU serving multiple zones would be considered to be an air-conditioning system and is the core focus of a zoning measure
- An FCU inherently only services a single zone but when combined with central plant may be considered to be an air-conditioning system across multiple zones; however, this is not the focus of a zoning measure.

To resolve this, it is necessary to restrict the application of the zoning measure to situations where zones are attached to a common air handler.

11.4 Proposed Measures

It is proposed that the treatment of HVAC zoning in NCC 2025 is strengthened by expanding J6D3 as follows:

1. Variable airflow delivery is included as the primary method for modulation of servicing to diverse zones on a common AHU.
2. Exclusion of situations where an air-conditioning system serves zones that can be simultaneously in heating and cooling.
3. Limitations are placed on the combination of zones in line with normal industry best practice.
4. Improvements to the identification of zoning requirements in Specification 34.

Three additional items were addressed in the proposed code text:

1. Recooling has been added to mirror the reheating requirement in J6D3. This is relevant to the rare possibility that a system might use terminal cooling;
2. The phrasing of 1 b) iv and v has been expressed such that it also covers the situation where FCUs are in potential conflict with a dedicated outside air system
3. An additional provision has been added to Specification 34 6(b) to cover the common situation where a simulator integrated multiple similar zones (e.g. all the VAV zones on single façade of a single floor) into a single zone.

11.4.1 Proposed Code text

J6D3 Air-conditioning Control

(1) An air-conditioning system –
  a) …{no change}
  b) Must -
    a. not use a single air-handler to serve any two air-conditioning zones where there is possibility of one zone requiring heating at the same time as the other requires cooling; and
    b. Where when serving more than one air-conditioning zone or area with different heating or cooling needes —
      i. thermostatically control the temperature of each zone or area; and

ii. not control the temperature by mixing actively heated air and actively cooled air; and
iii. Service the diverse loads firstly through the modulation of airflow; and
iv. Limit zone-level reheating of air cooled by equipment upstream of the zone to not more than 7.5K at any intended zone airflow
v. Limit recooling of air heated by equipment upstream of the zone to not more than 7.5K at any intended zone airflow

c) …{as per c) in NCC 2022}

Specification 34 C3

(6) For the purposes of (1)(i), services must include—
- a) …{no change}
- b) The same air-conditioning zoning including:
  - i. assumptions and means of calculating the temperature difference across air-conditioning zone boundaries; and
  - ii. assumptions relating to the combination of zones of similar heating and cooling requirements for the purpose of simulation simplification; and
  - iii. separation of zones no less than,
    1. separate floors
    2. for spaces larger than $20m^2$
       - a. perimeter zones, being spaces up to 4m deep from a wall with windows, separately for each 90° cardinal aspect; and
       - b. centre zones, being spaces that are further than 4m from a wall with windows that comprise more than 20% of the floor area of a storey after perimeter zones have been removed;
- c) …{no change}

APPENDICES

12 Appendix: Chillers

12.1 Chiller Part Load Curves

The chiller part load curves presented in Appendix sections 12.1.1 and 12.1.2 depict those of a representative and average chiller of the respective technology type. Data acquired for real chillers were screened for completeness ($COP_{25\%}$, $COP_{50\%}$, $COP_{75\%}$, $COP_{100\%}$) and subsequently collated to derive average, desensitised values for each chiller type at the corresponding part loads and operating conditions. This data was then correlated to standardised chiller equation formats used in IES.

12.1.1 Air cooled chillers

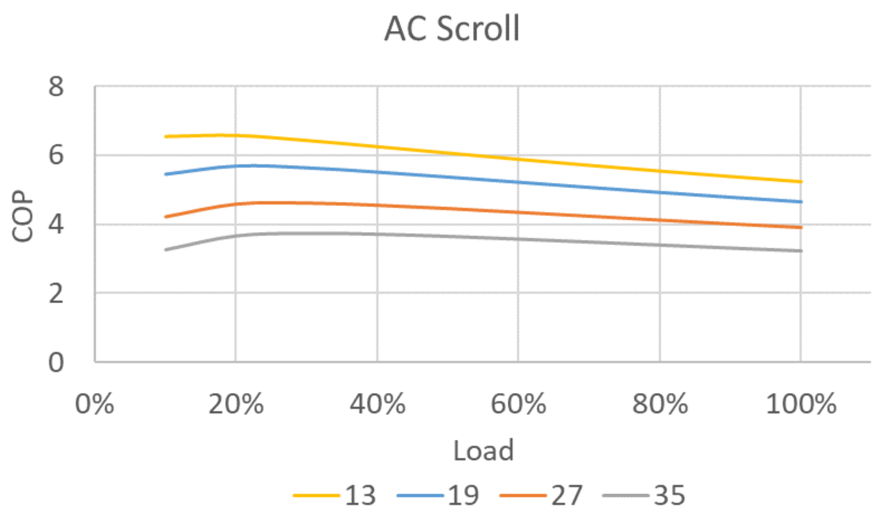


Figure 111. Air cooled scroll chiller part load performance at different outside air temperatures.

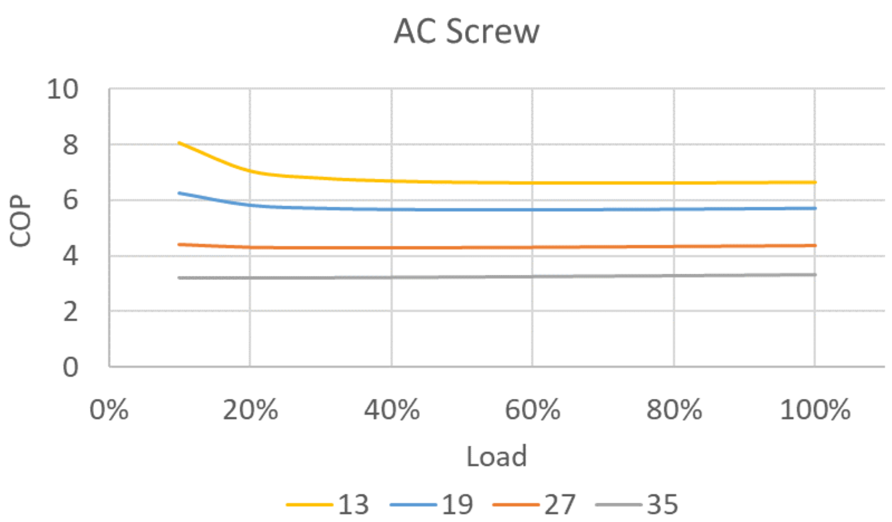


Figure 112. Air cooled screw chiller part load performance at different outside air temperatures.

12.1.2 Water cooled chillers.

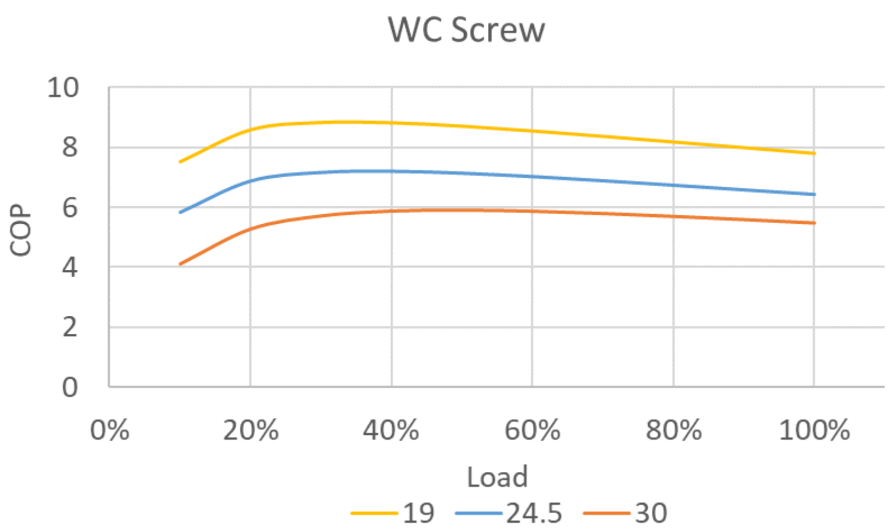


Figure 113. Water cooled screw chiller part load performance at different condensing temperatures.

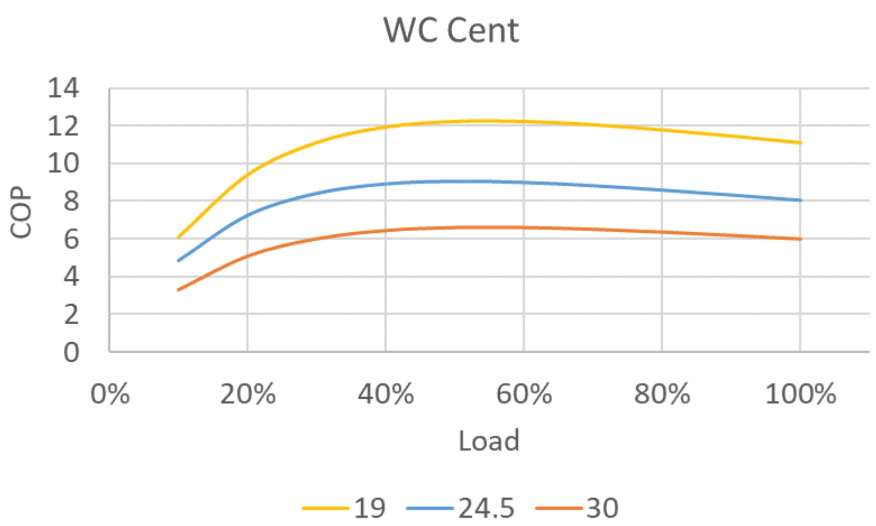


Figure 114. Water-cooled centrifugal chiller part load performance at different condensing temperatures.

12.2 Chiller data

Table 60 to Table 66 catalogue chiller nominal capacity, EER, IPLV, compressor technology and list prices (excluding GST but including delivery to East Coast sites) for all chillers assessed in the study, in ascending order of chiller nominal capacity. Each table collates chiller data which fall into the technology-capacity bins currently adopted in NCC 2019.

Table 60 Air-cooled chiller data for nominal capacities 0 – 528 kW.

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
88.3	2.859	3.509	AC Screw	$75,400
93.2	3.37	6.0154	AC Scroll	$50,150
100.2	3.191	5.77	AC Scroll	$53,500
114	3.24	5.556	AC Scroll	$53,590
129.3	2.763	5.02	AC Scroll	$51,500
154	3.355	5.32	AC Scroll	$63,500
158.1	2.885	4.7	AC Scroll	$60,500
158.6	2.706	3.95	AC Scroll	$63,500
200	3.204	4.823	AC Scroll	$121,100
200	3.243	5.307	AC Scroll	$124,100
224.1	3.243	5.131	AC Scroll	$76,500
233	3.17	4.592	AC Screw	$117,700
397.6	3.16	4.602	AC Screw	$151,850
400	3.463	5.56	AC Scroll	$97,500
411.7	3.12	5.24	AC Scroll	$102,240
417.8	3.095	5.05	AC Scroll	$101,500
439.6	3.6	5.39	AC Screw	$140,500
500	3.28	5.74	AC Screw	$161,500
500	2.92	5.2	AC Screw	$165,500

Table 61 Air-cooled chiller data for nominal capacities > 528 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
550	3.108	5.591	AC Scroll	$182,000
550	2.802	5.369	AC Screw	$224,100
550	3.152	5.832	AC Screw	$281,100
700	3.083	5.548	AC Scroll	$211,100
700	3.125	5.742	AC Screw	$282,300
700	3.167	5.564	AC Screw	$324,700
702.9	3.23	5.002	AC Scroll	$158,600
755.6	3.167	5.03	AC Scroll	$155,500
756.1	3.491	5.31	AC Screw	$90,500
773.2	3.083	5.01	AC Screw	$179,500
900	3.091	4.235	AC Screw	$392,650
959.4	3.117	5.77	AC Screw	$325,500
959.9	2.99	5.04	AC Screw	$255,500
972.7	2.726	5.33	AC Screw	$295,500
987.3	3.486	5.4	AC Screw	$275,500
1026.8	3.11	4.981	AC Scroll	$222,100
1100	2.964	5.673	AC Screw	$332,100
1100	3.251	5.954	AC Screw	$398,100
1350	2.83	5.47	AC Screw	$375,500
1350	3.26	5.75	AC Screw	$405,500
1365	3.387	5.34	AC Screw	$345,500
1390	2.916	5.08	AC Screw	$320,500
1500	2.983	4.13	AC Screw	$335,100
1500	3.117	4.969	AC Screw	$351,100

Table 62 Water-cooled chiller data for nominal capacities 0 - 264 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
84.24	4.621	5.254	WC Screw	$61,500
139.1	4.95	6.14	WC Screw	$60,500
142.1	4.659	6.16	WC Screw	$58,500
160.6	4.689	5.364	WC Screw	$87,150
216.8	5.19	8.8494	WC Screw	$107,600

Table 63 Water-cooled chiller data for nominal capacities 264 - 528 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
355.9	5.151	9.83	WC Centrif	$137,500
393.9	5.044	8.44	WC Screw	$150,500
400	5.56	10.2	WC Centrif	$152,500
400	5.486	8.928	WC Screw	$112,000
400	5.62	9.216	WC Screw	$182,000
404	5.25	8.8	WC Screw	$135,500
407	4.89	9.4647	WC Screw	$144,700
441	5.526	6.59	WC Screw	$97,500
459	5.51	7.09	WC Screw	$107,500
469.8	5.268	9.357	WC Centrif	$171,100
492.9	6.16	9.0919	WC Centrif	$210,700
500	5.298	9.56	WC Screw	$145,500

Table 64 Water-cooled chiller data for nominal capacities 528 - 1055 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
600	5.628	8.425	WC Centrif	$340,500
600	5.8	10.8	WC Centrif	$215,500
600	5.6	10.7	WC Centrif	$235,500
739.8	5.5	7.48	WC Screw	$132,500
742.9	6.05	10.3	WC Centrif	$213,500
750	5.72	10.3	WC Centrif	$280,500
800	5.337	9.349	WC Screw	Unavailable
800	5.494	9.557	WC Screw	$132,000
800	5.666	9.359	WC Screw	$301,100
830	5.576	8.825	WC Centrif	$329,500
933	5.33	8.4	WC Screw	$245,500
989.2	5.32	8.0812	WC Screw	$261,100
1000	6.07	10.6	WC Centrif	$285,500
1000	5.63	9.93	WC Centrif	$419,500
1000	6.066	9.814	WC Screw	$175,500
1000	5.49	8.48	WC Screw	$207,500
1010	6.14	10.42	WC Centrif	$363,500
1049.3	5.34	8.5593	WC Centrif	$331,000
1050	5.812	9.693	WC Centrif	$333,500

Table 65 Water-cooled chiller data for nominal capacities 1055 - 1407 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
1200	5.9	10.9	WC Centrif	$305,500
1200	5.8	10.9	WC Centrif	$345,500
1300	6.14	10.5	WC Centrif	$363,500
1300	5.86	10.6	WC Centrif	$353,500
1300	5.91	10.6	WC Centrif	$558,500
1341	6.013	9.66	WC Centrif	$315,500
1342.8	6.31	10.2106	WC Centrif	$528,400
1350	5.89	9.684	WC Screw	$210,500
1400	5.773	9.322	WC Screw	$188,100
1400	5.6	9.094	WC Screw	$284,400

Table 66 Water-cooled chiller data for nominal capacities > 1407 kW

Capacity (kW)	EER	IPLV	Technology	Cost (excl GST)
1500	4.98	7.1172	WC Screw	$255,500
1600	6	11	WC Centrif	$425,500
1765.9	5.23	7.9633	WC Screw	$422,550
1800	6	11.1	WC Centrif	$385,500
1958.5	5.78	9.1565	WC Centrif	$528,400
2000	6.4	11.5	WC Centrif	$465,500
2400	6.4	11.5	WC Centrif	$495,500
4275	6.195	10.67	WC Centrif	$1,084,000

12.3 BCR Results

The graphs in the following sub-sections document the results from step 10 of the analysis process described in Section 2.4.3. The parameters which form the underlying basis of economic analysis in the Chillers BCR are as follows:

- Assessment Timeframe: 20 years
- Equipment Replacement Time: 20 years for both air-cooled and water-cooled chillers
- Discount Rate: 5%
- Electricity costs were obtained for Year 1 to 27 from the CIE[36] model. Costs beyond Year 27 were assumed to be constant due to the extent of available modelling at the time of writing. A graphical representation of the assumed price path for electricity is shown in Figure 115.

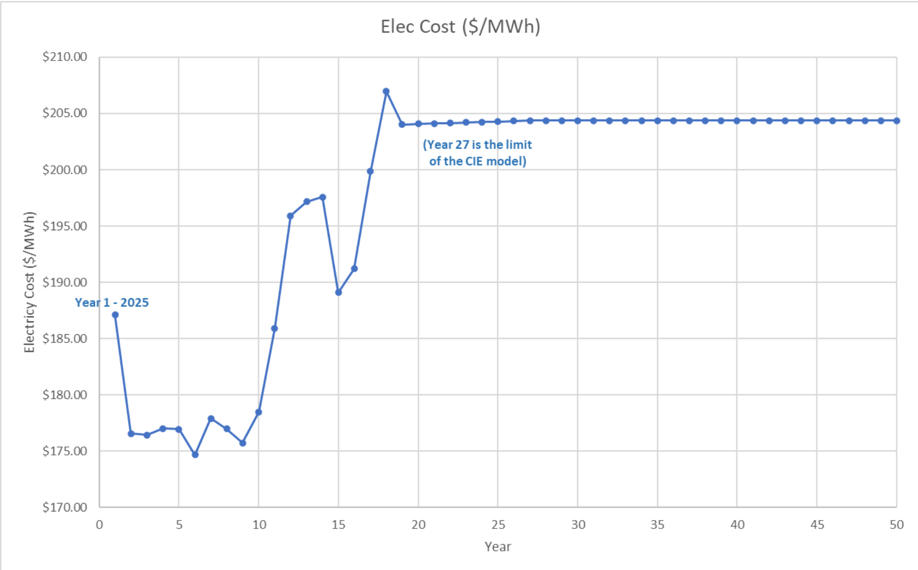


Figure 115 Electricity Costs ($/MWh) from CIE model used in the economic analysis

12.3.1 C9A (Overnight) Air-cooled Chillers (0-528 kW)

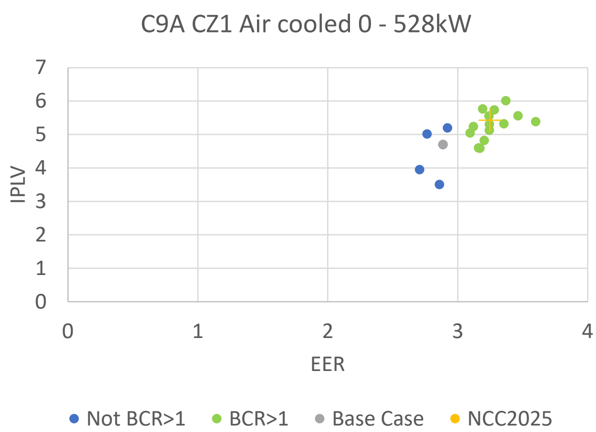


Figure 116 C9A CZ1 Air cooled 0-528 KW

[36] The Centre for International Economics

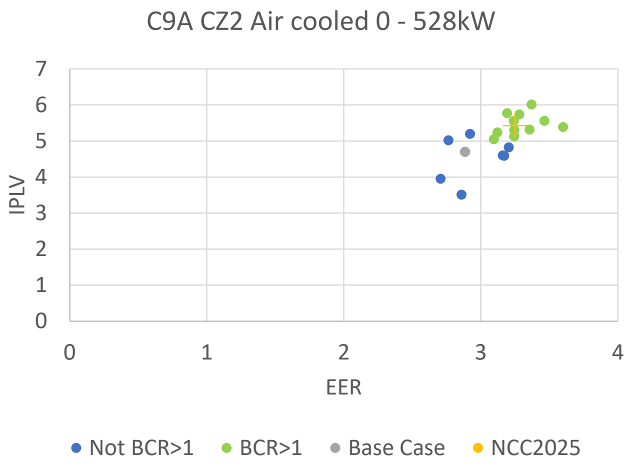


Figure 117 C9A CZ2 Air cooled 0-528 KW.

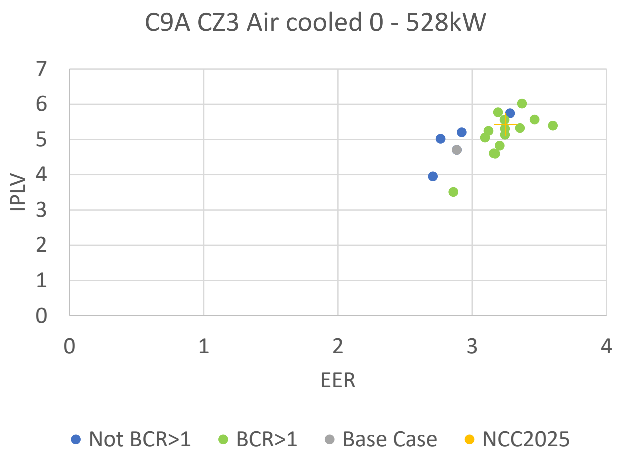


Figure 118 C9A CZ3 Air cooled 0-528 KW.

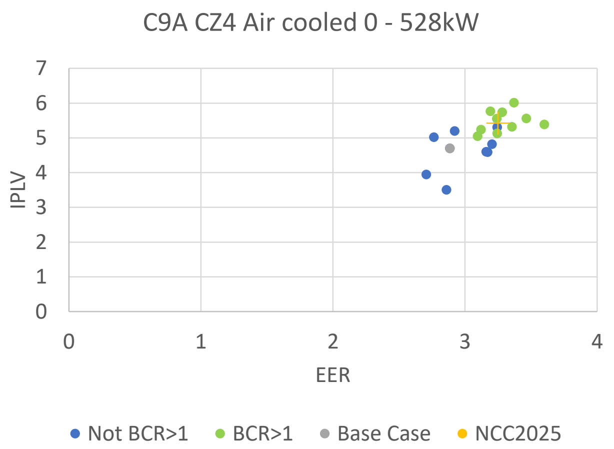


Figure 119 C9A CZ4 Air cooled 0-528 KW.

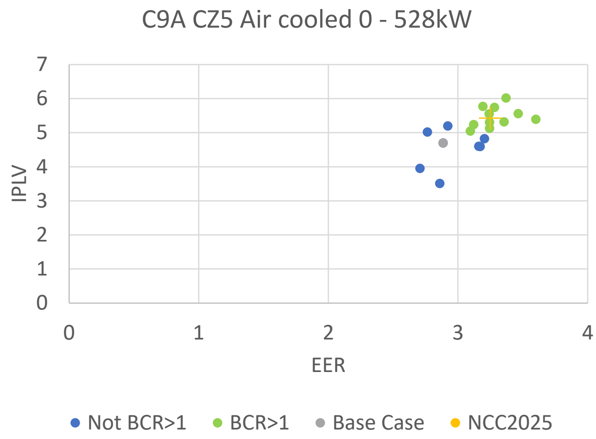


Figure 120 C9A CZ5 Air cooled 0-528 KW.

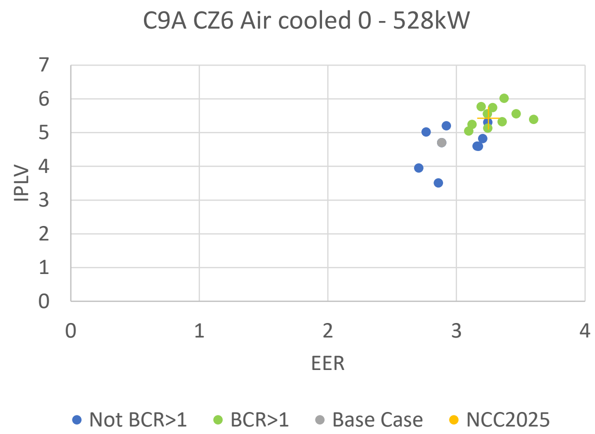


Figure 121 C9A CZ6 Air cooled 0-528 KW.

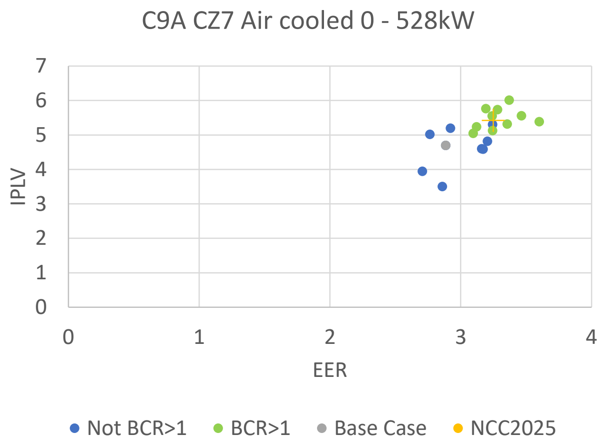


Figure 122 C9A CZ7 Air cooled 0-528 KW.

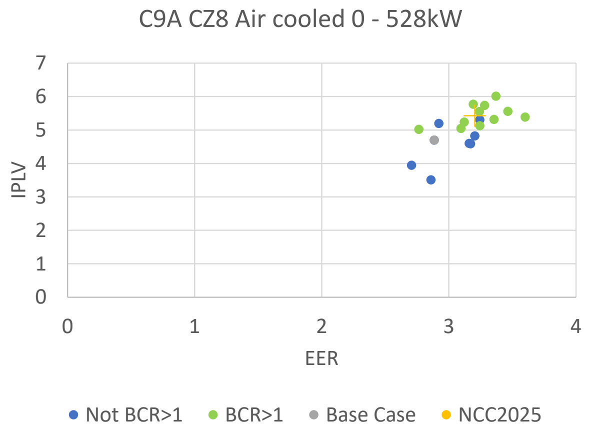


Figure 123 C9A CZ8 Air cooled 0-528 KW.

12.3.2 C9A Air cooled >528KW

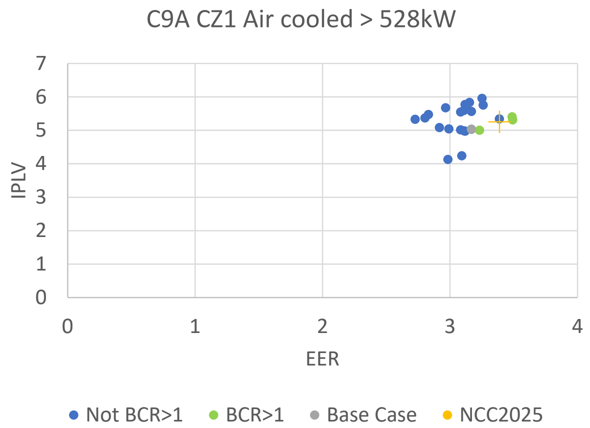


Figure 124 C9A CZ1 Air cooled >528KW.

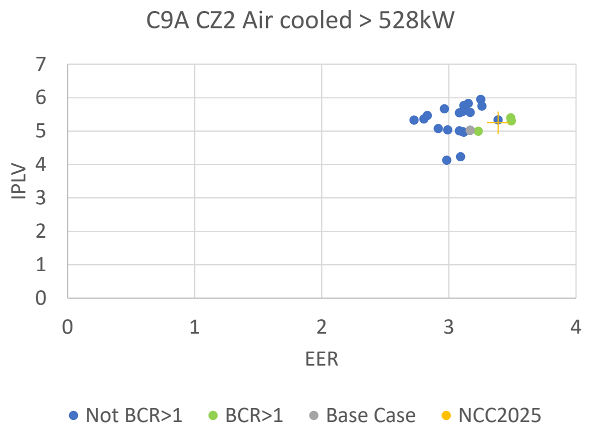


Figure 125 C9A CZ2 Air cooled >528KW.

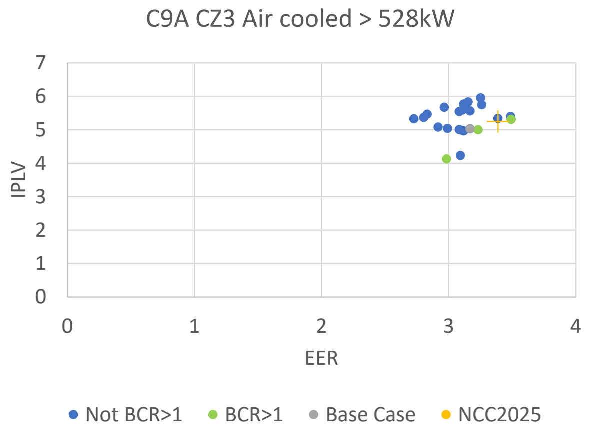


Figure 126 C9A CZ3 Air cooled >528KW.

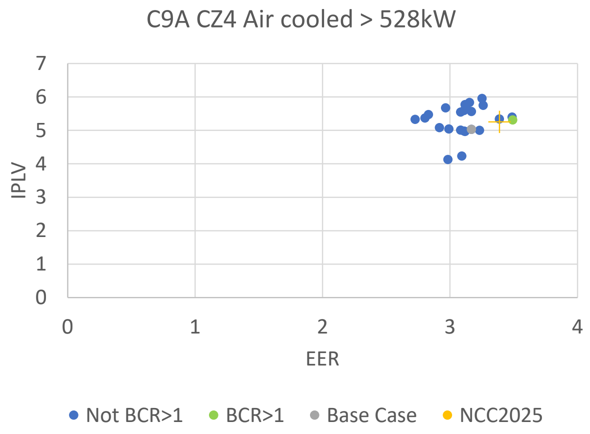


Figure 127 C9A CZ4 Air cooled >528KW.

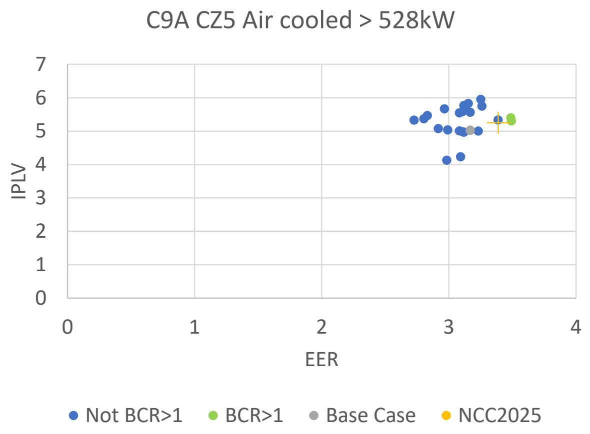


Figure 128 C9A CZ5 Air cooled >528KW.

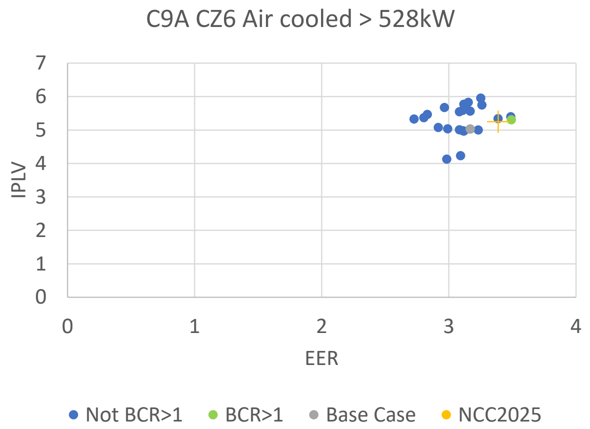


Figure 129 C9A CZ6 Air cooled >528KW.

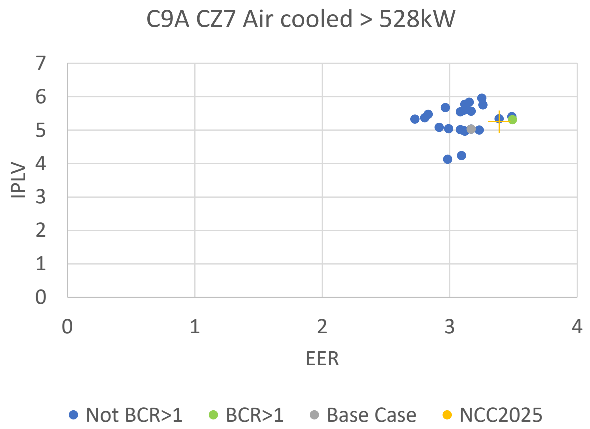


Figure 130 C9A CZ7 Air cooled >528KW.

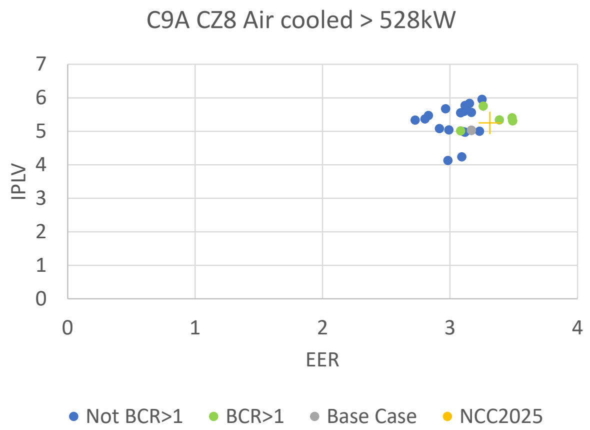


Figure 131 C9A CZ8 Air cooled >528KW.

12.3.3 C9A Water cooled 0-264KW

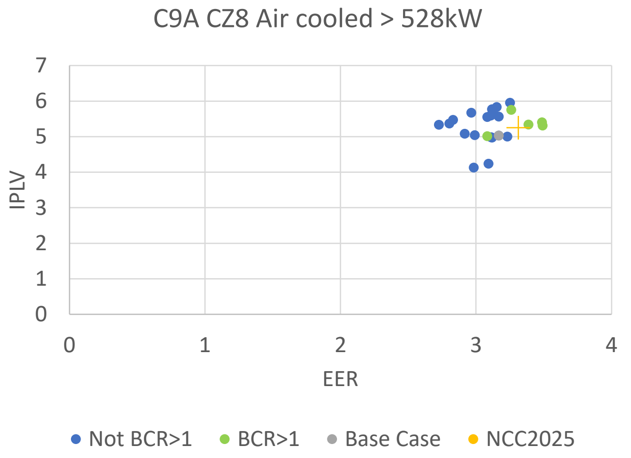


Figure 132 C9A CZ1 Water cooled 0-264KW.

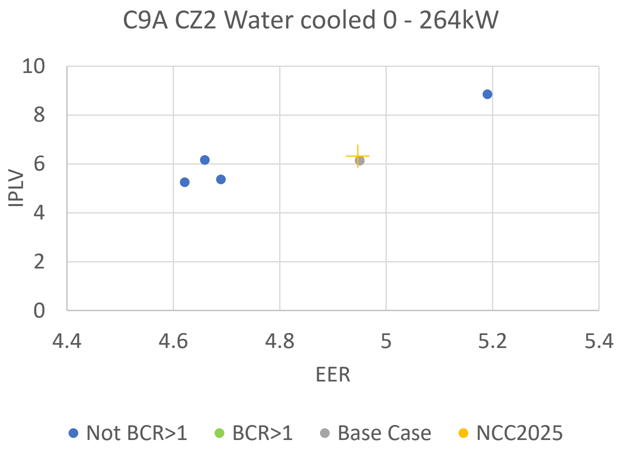


Figure 133C9A CZ2 Water cooled 0-264KW.

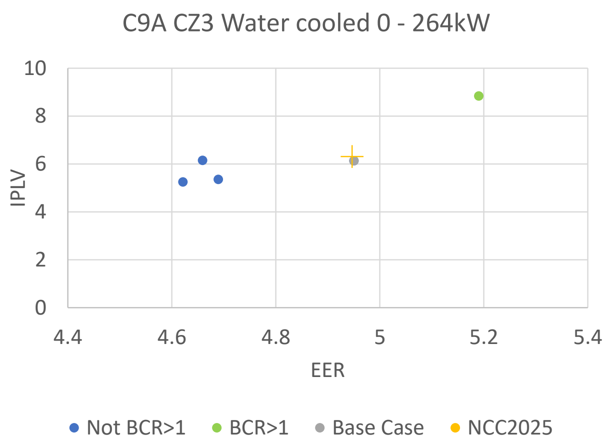


Figure 134 C9A CZ3 Water cooled 0-264KW.

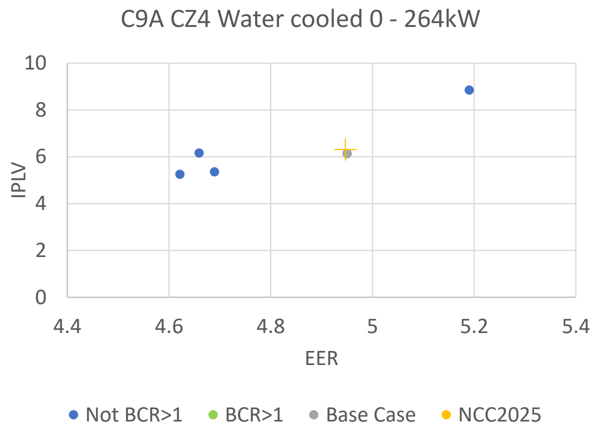


Figure 135 C9A CZ4 Water cooled 0-264KW.

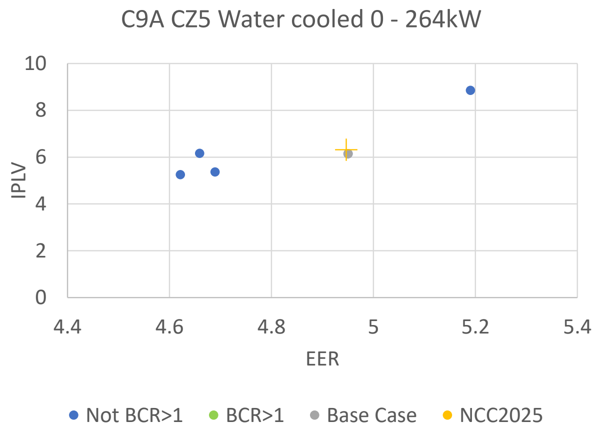


Figure 136 C9A CZ5 Water cooled 0-264KW.

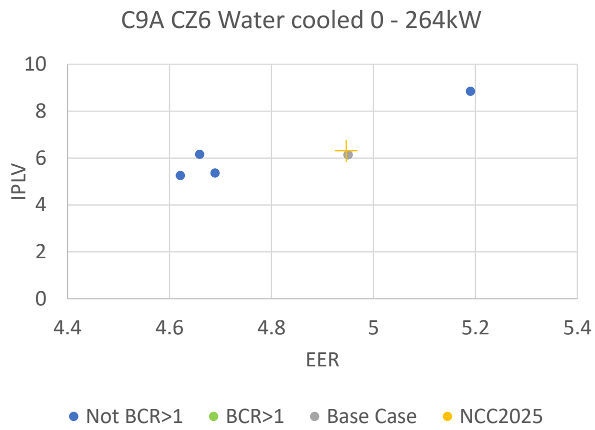


Figure 137 C9A CZ6 Water cooled 0-264KW.

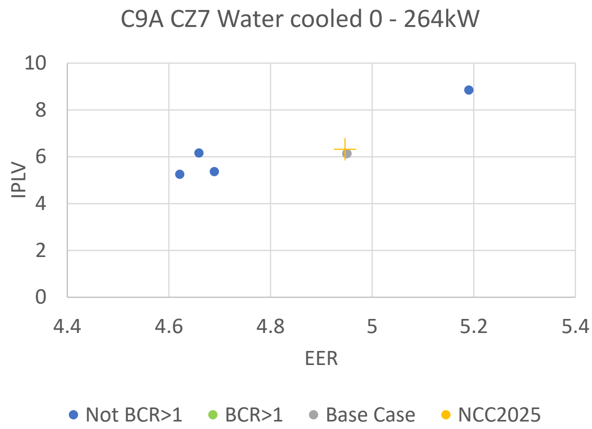


Figure 138 C9A CZ7 Water cooled 0-264KW.

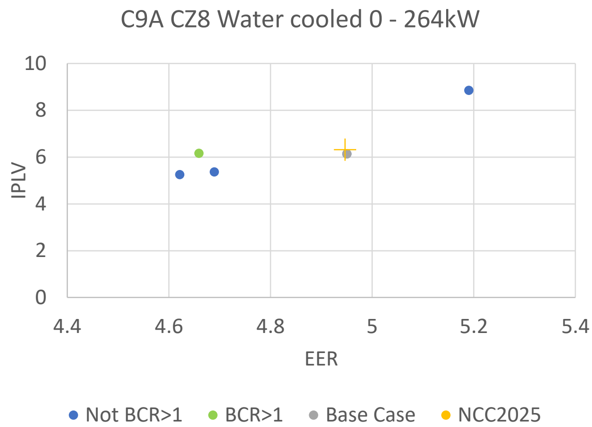


Figure 139 C9A CZ8 Water cooled 0-264KW.

12.3.4 C9A Water cooled 264-528KW

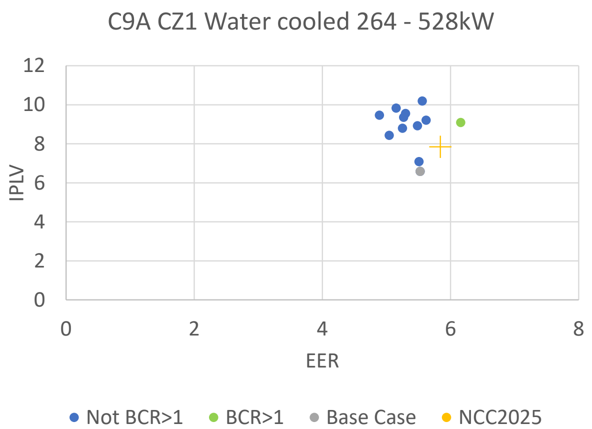


Figure 140 C9A CZ1 Water cooled 264-528KW.

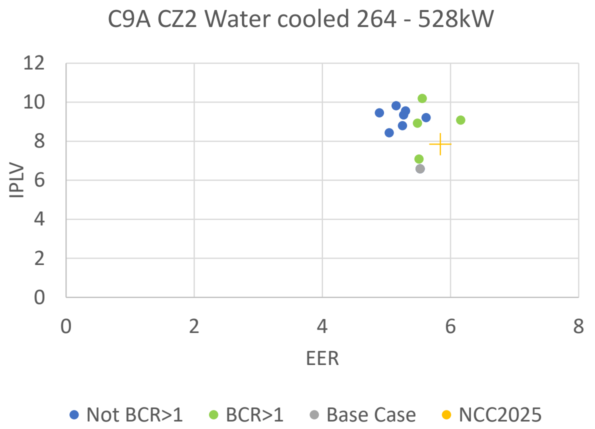


Figure 141 C9A CZ2 Water cooled 264-528KW.

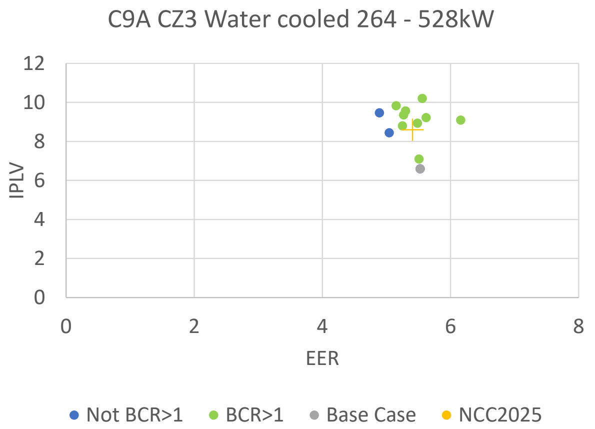


Figure 142 C9A CZ3 Water cooled 264-528KW.

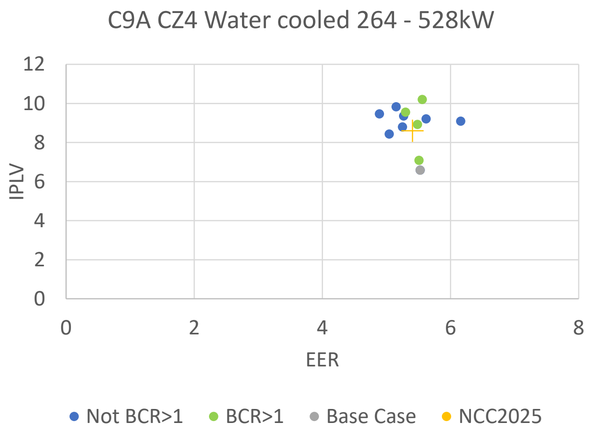


Figure 143 C9A CZ4 Water cooled 264-528KW.

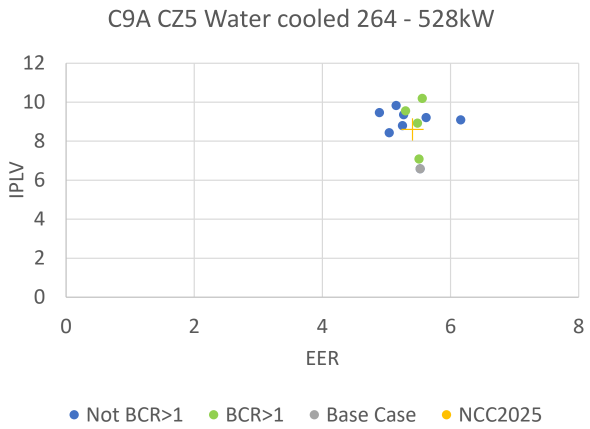


Figure 144 C9A CZ5 Water cooled 264-528KW.

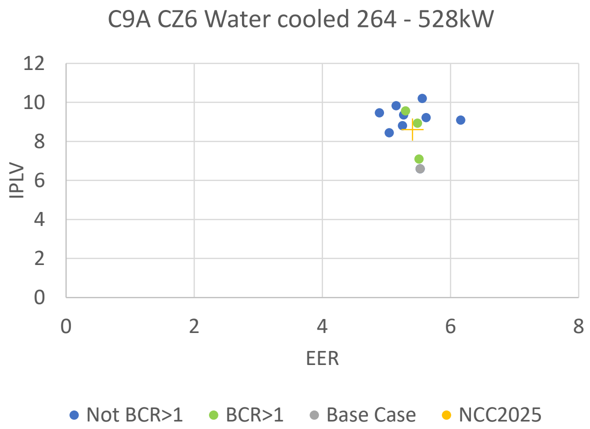


Figure 145 C9A CZ6 Water cooled 264-528KW.

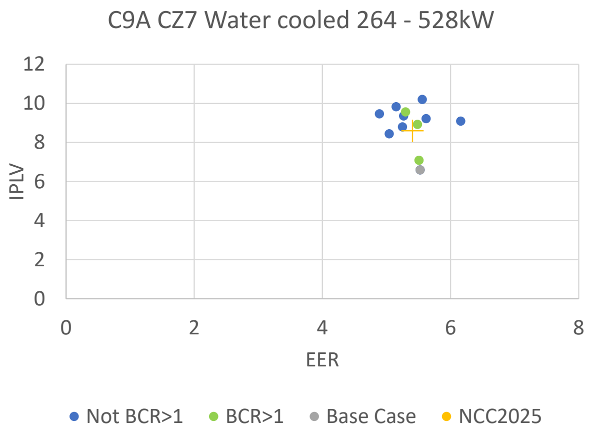


Figure 146 C9A CZ7 Water cooled 264-528KW.

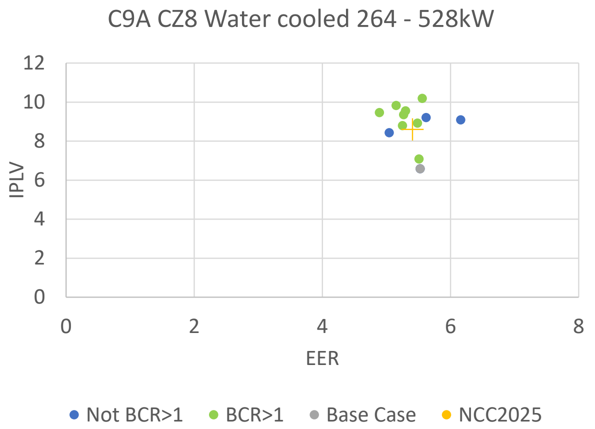


Figure 147 C9A CZ8 Water cooled 264-528KW.

12.3.5 C9A Water cooled 528-1055 KW

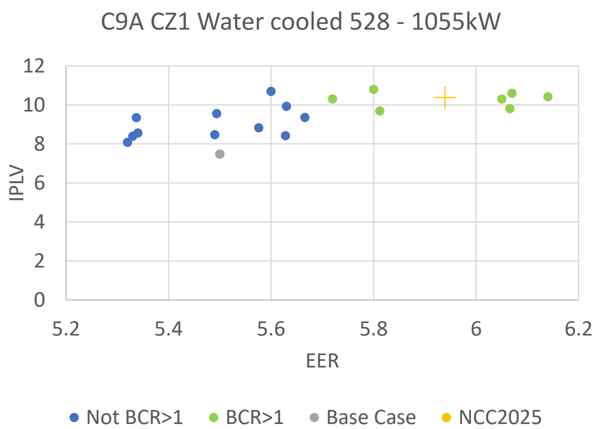


Figure 148 C9A CZ1 Water cooled 528-1055 KW.

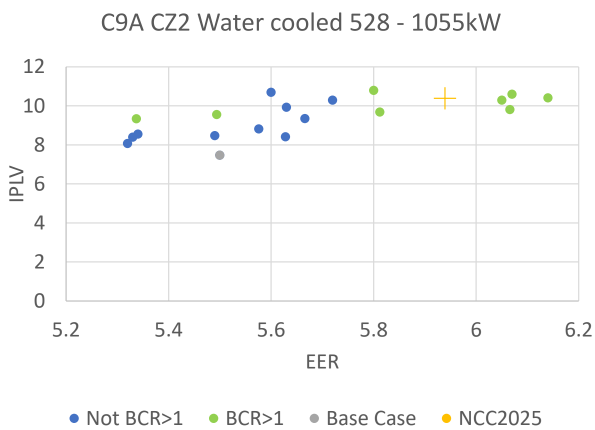


Figure 149 C9A CZ2 Water cooled 528-1055 KW.

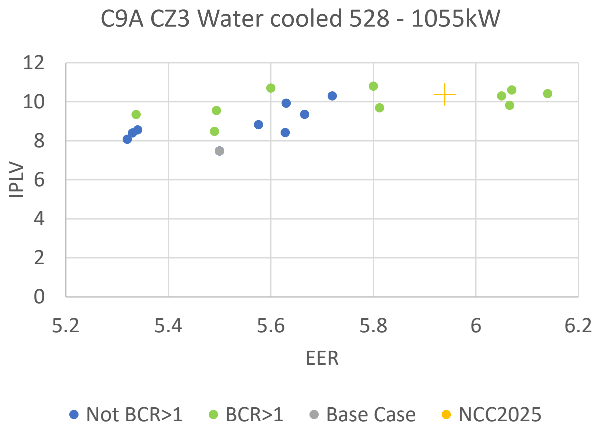


Figure 150 C9A CZ3 Water cooled 528-1055 KW.

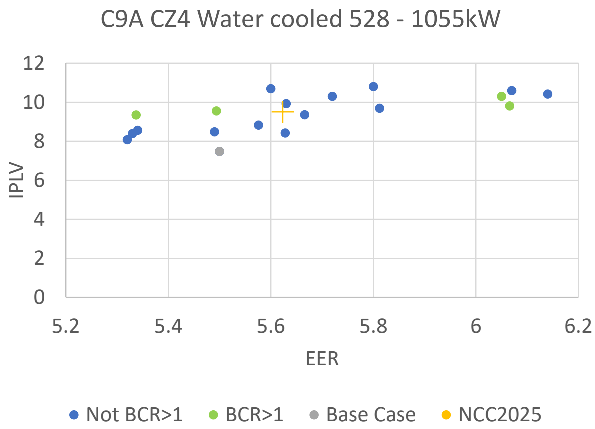


Figure 151 C9A CZ4 Water cooled 528-1055 KW.

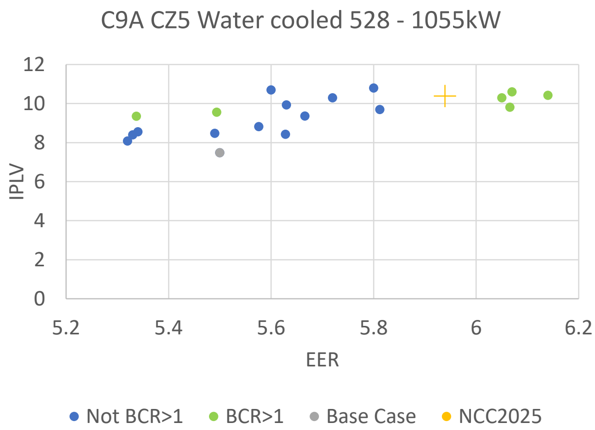


Figure 152 C9A CZ5 Water cooled 528-1055 KW.

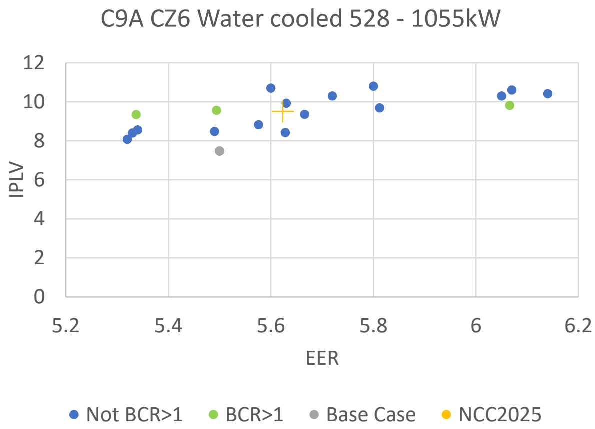


Figure 153 C9A CZ6 Water cooled 528-1055 KW.

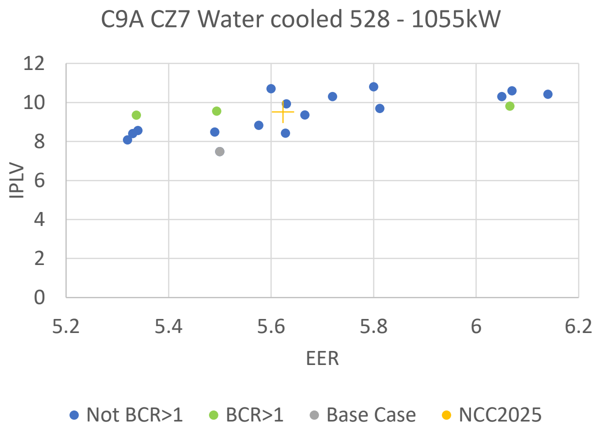


Figure 154 C9A CZ7 Water cooled 528-1055 KW.

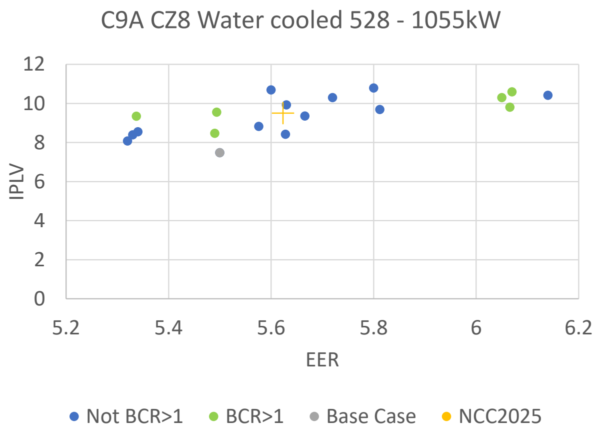


Figure 155 C9A CZ8 Water cooled 528-1055 KW.

12.3.6 C9A Water cooled 1055-1407 KW

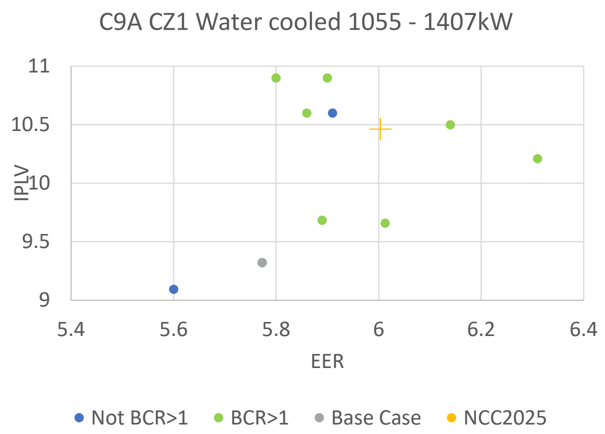


Figure 156 C9A CZ1 Water cooled 1055-1407 KW.

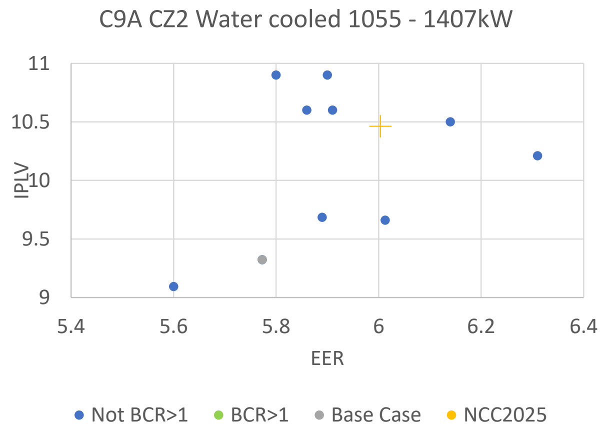


Figure 157 C9A CZ2 Water cooled 1055-1407 KW.

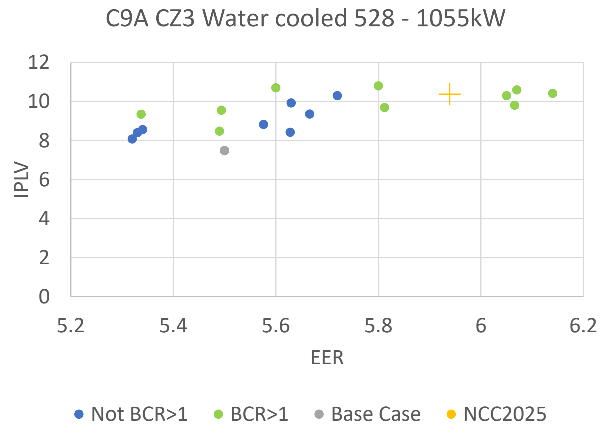


Figure 158 C9A CZ3 Water cooled 1055-1407 KW.

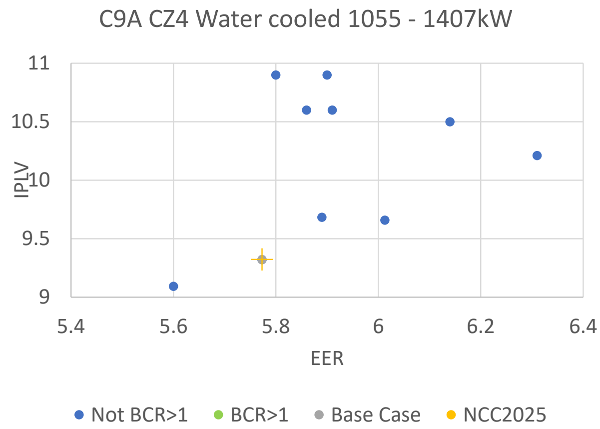


Figure 159 C9A CZ4 Water cooled 1055-1407 KW.

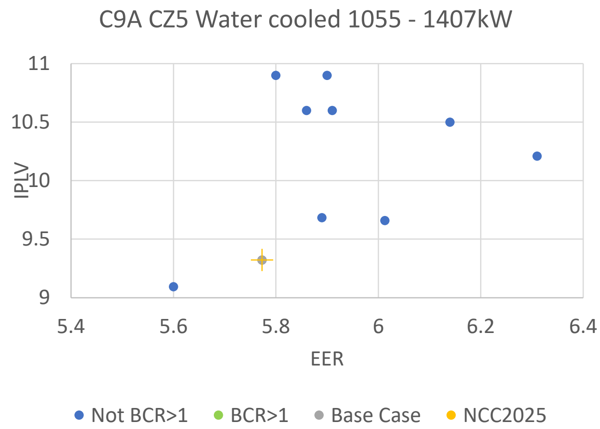


Figure 160 C9A CZ5 Water cooled 1055-1407 KW.

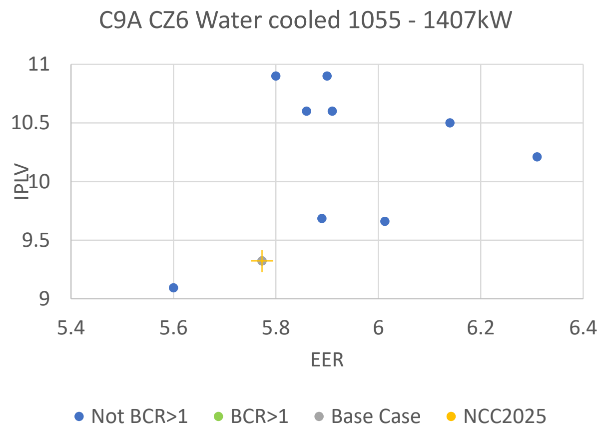


Figure 161 C9A CZ6 Water cooled 1055-1407 KW.

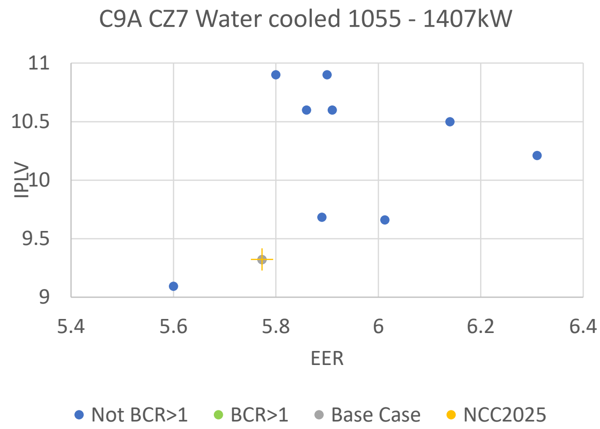


Figure 162 C9A CZ7 Water cooled 1055-1407 KW.

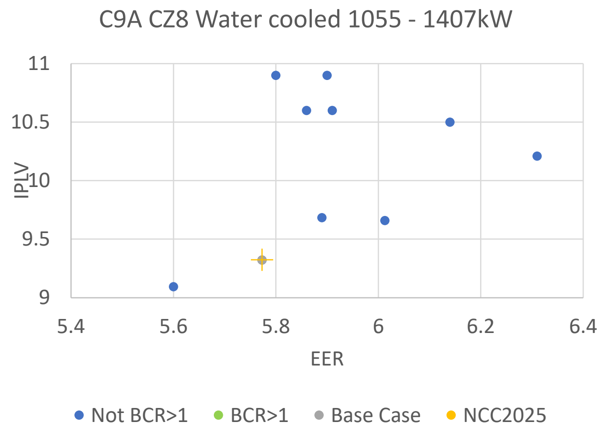


Figure 163 C9A CZ8 Water cooled 1055-1407 KW.

12.3.7 C9A Water cooled >1407 KW

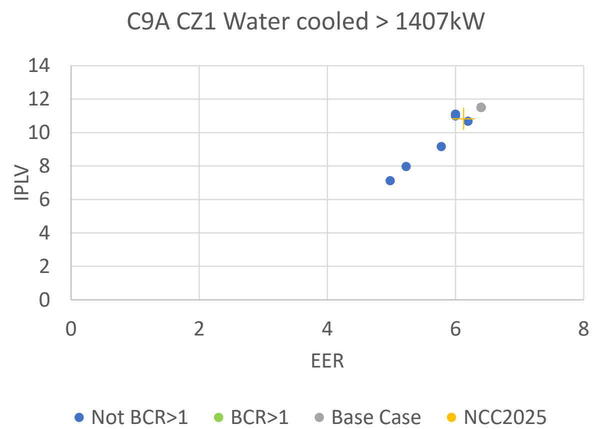


Figure 164 C9A CZ1 Water cooled >1407 KW.

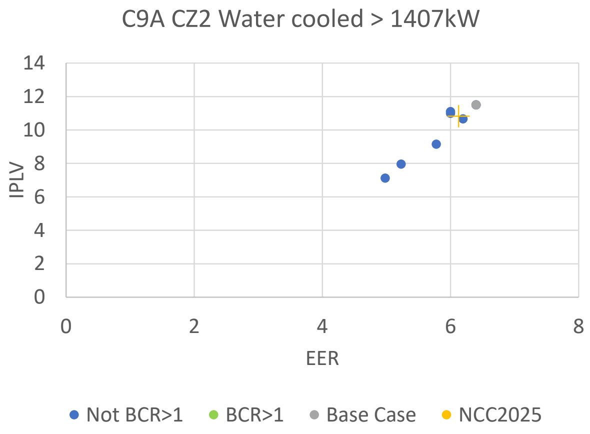


Figure 165 C9A CZ2 Water cooled >1407 KW.

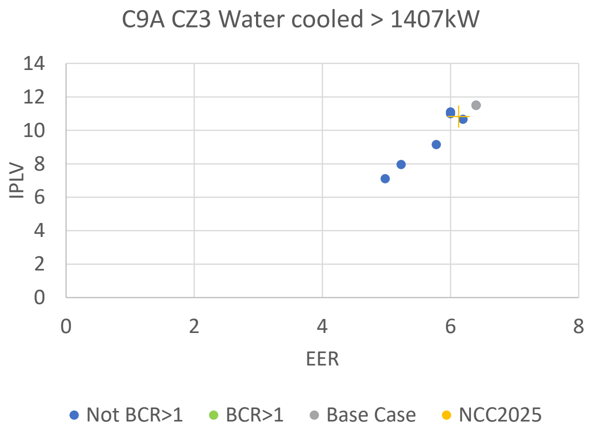


Figure 166 C9A CZ3 Water cooled >1407 KW.

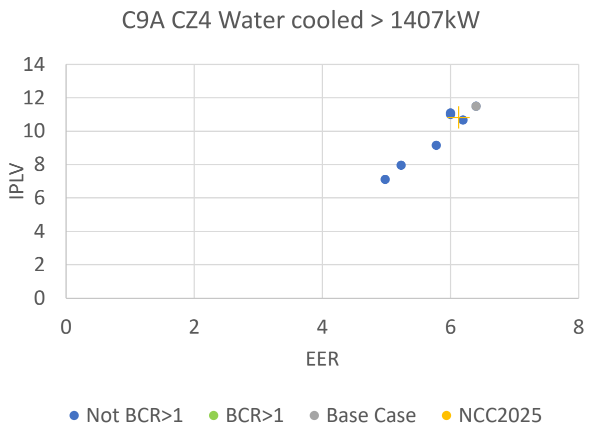


Figure 167 C9A CZ4 Water cooled >1407 KW.

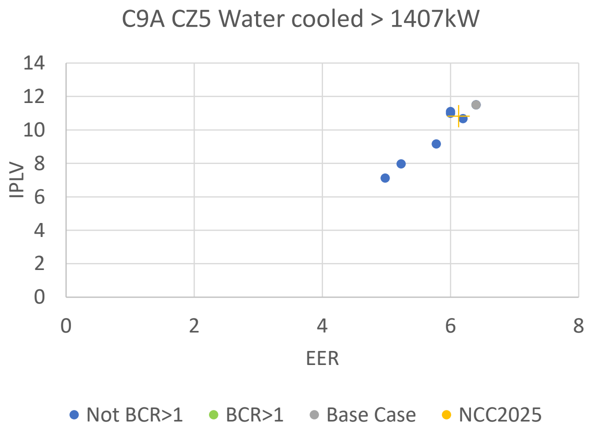


Figure 168 C9A CZ5 Water cooled >1407 KW.

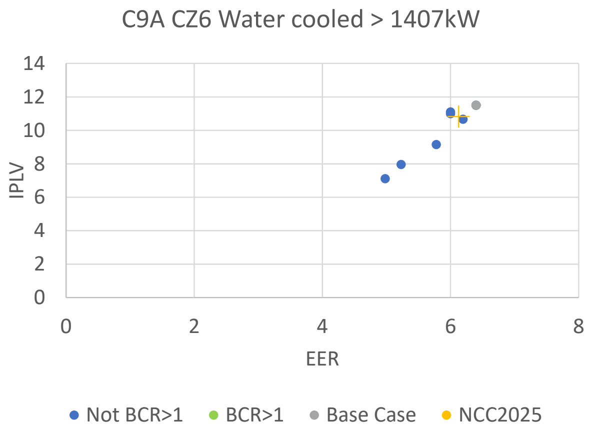


Figure 169 C9A CZ6 Water cooled >1407 KW.

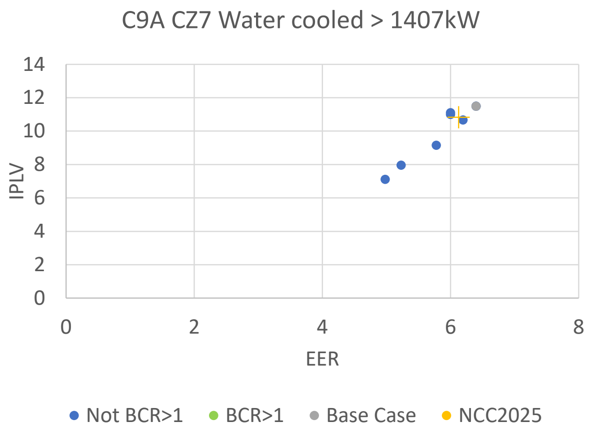


Figure 170 C9A CZ7 Water cooled >1407 KW.

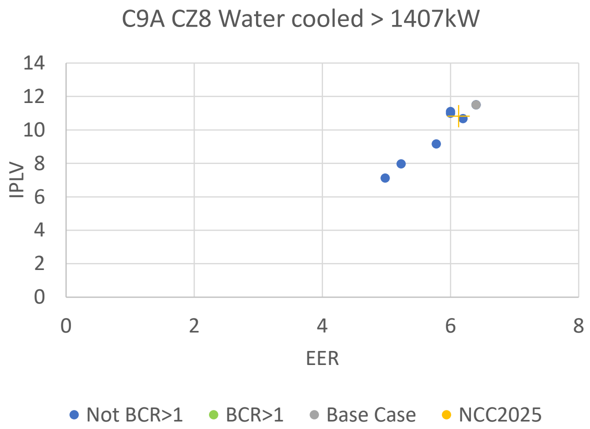


Figure 171 C9A CZ8 Water cooled >1407 KW.

12.3.8 C50L Air cooled 0-528 KW

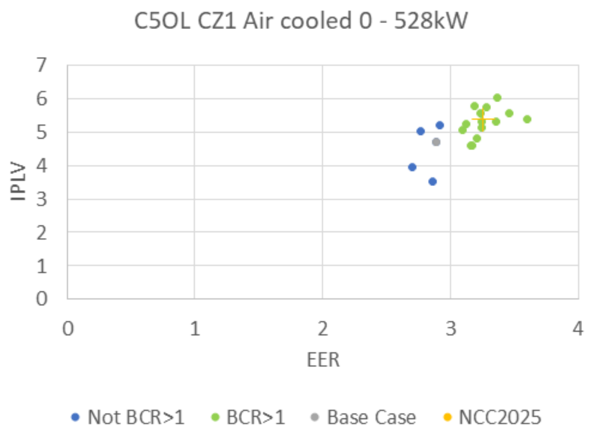


Figure 172 C50L CZ1 Air cooled 0-528 KW.

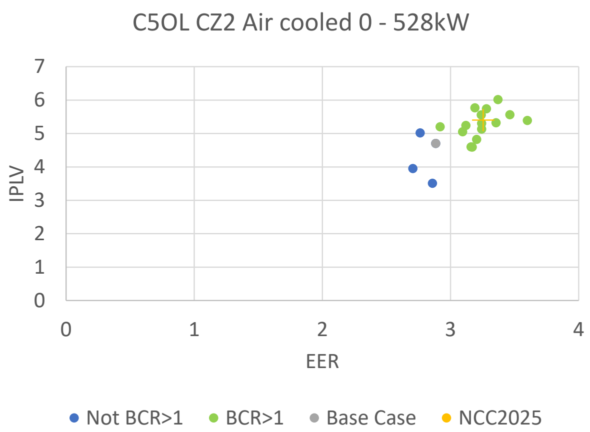


Figure 173 C50L CZ2 Air cooled 0-528 KW.

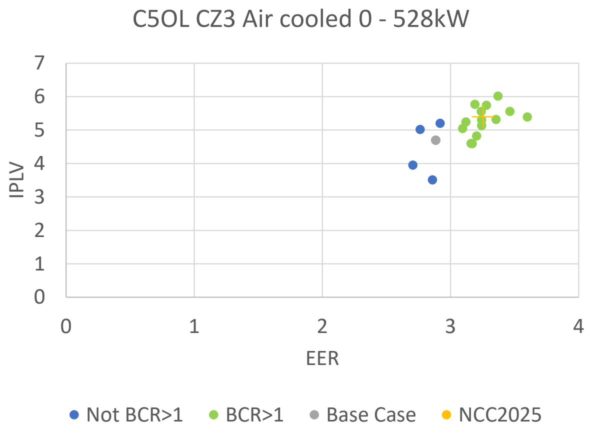


Figure 174 C50L CZ3 Air cooled 0-528 KW.

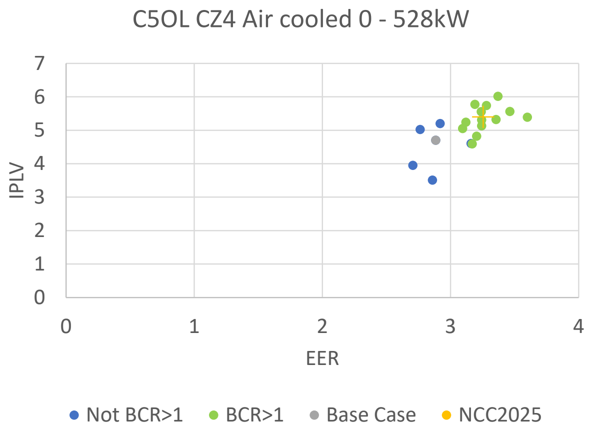


Figure 175 C50L CZ4 Air cooled 0-528 KW.

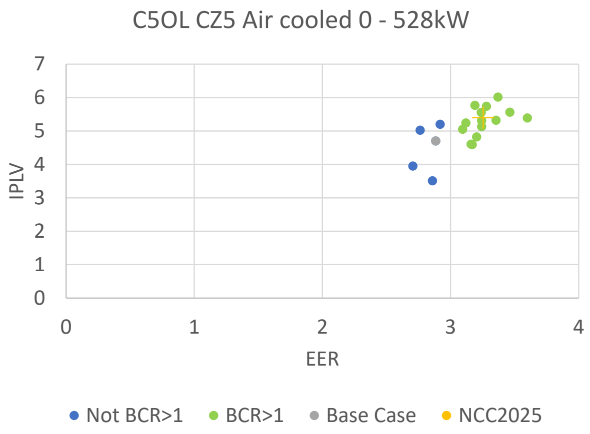


Figure 176 C50L CZ5 Air cooled 0-528 KW.

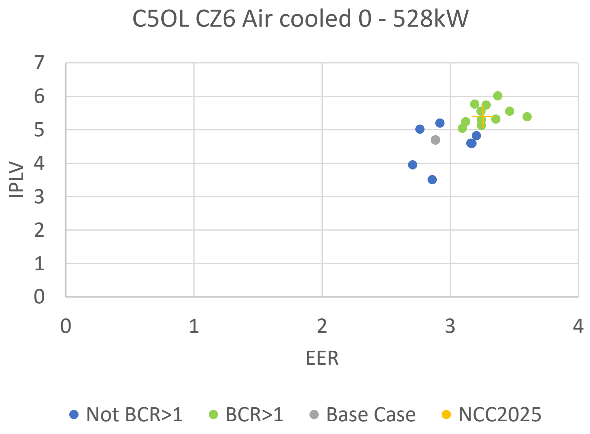


Figure 177 C50L CZ6 Air cooled 0-528 KW.

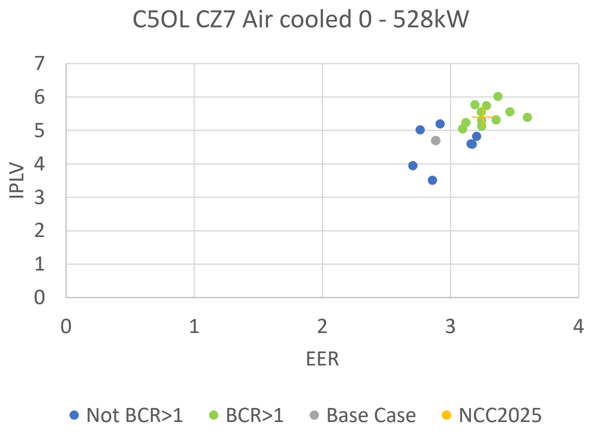


Figure 178 C50L CZ7 Air cooled 0-528 KW.

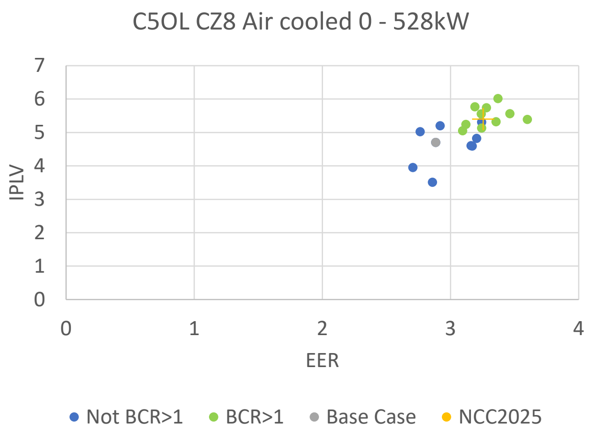


Figure 179 C50L CZ8 Air cooled 0-528 KW.

12.3.9 C50L Air cooled >528 KW

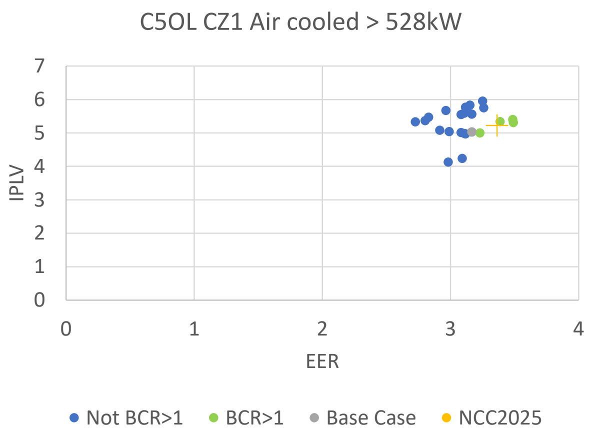


Figure 180 C50L CZ1 Air cooled >528 KW.

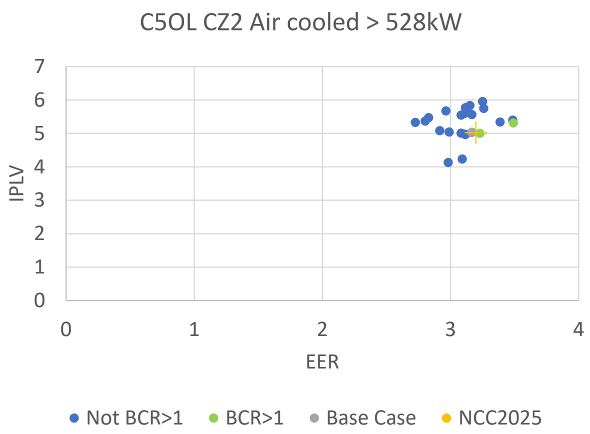


Figure 181 C50L CZ2 Air cooled >528 KW.

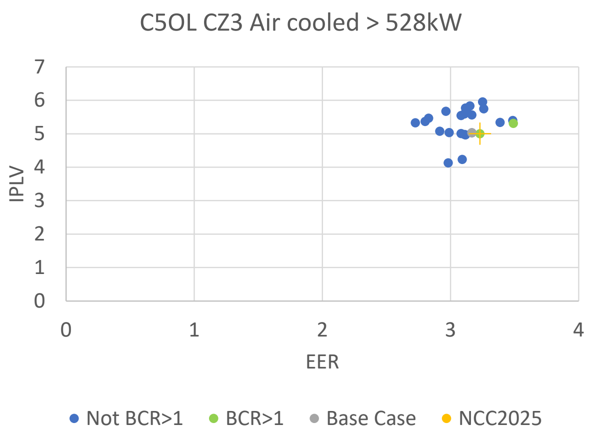


Figure 182 C50L CZ3 Air cooled >528 KW.

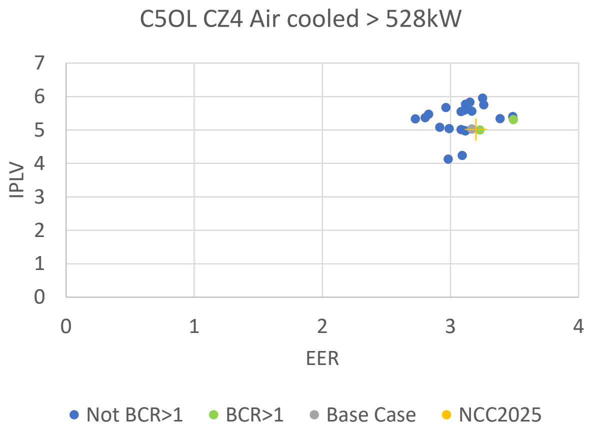


Figure 183 C50L CZ4 Air cooled >528 KW.

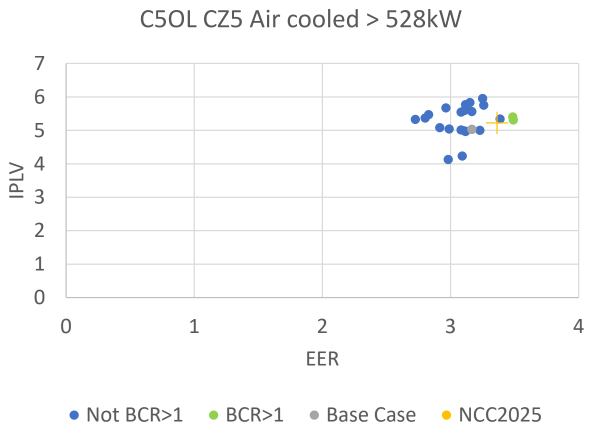


Figure 184 C50L CZ5 Air cooled >528 KW.

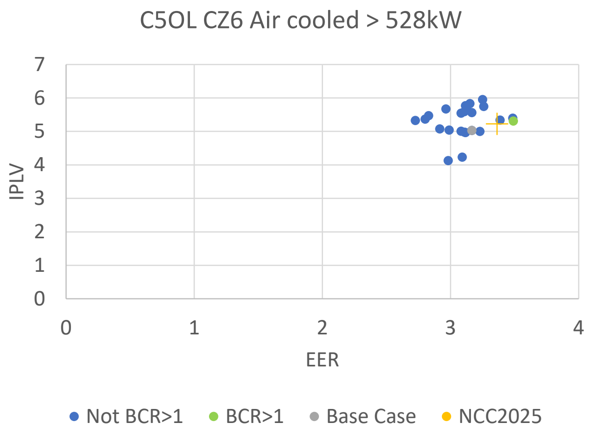


Figure 185 C50L CZ6 Air cooled >528 KW.

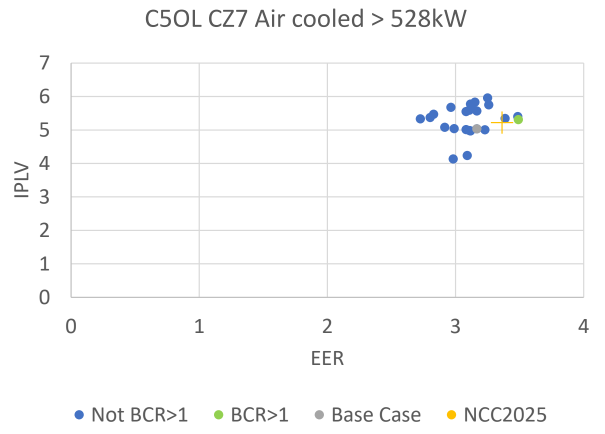


Figure 186 C50L CZ7 Air cooled >528 KW.

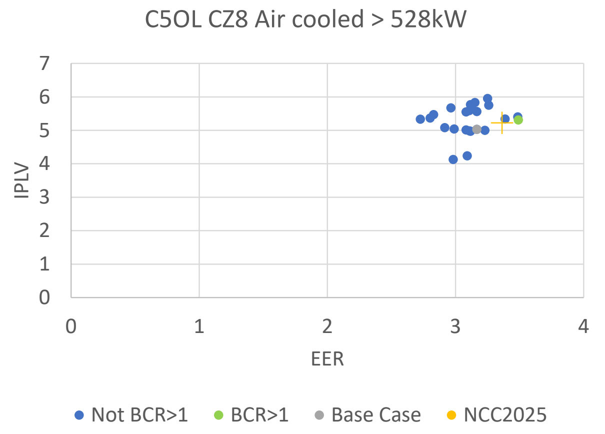


Figure 187 C50L CZ8 Air cooled >528 KW.

12.3.10 C50L Water cooled 0-264 KW

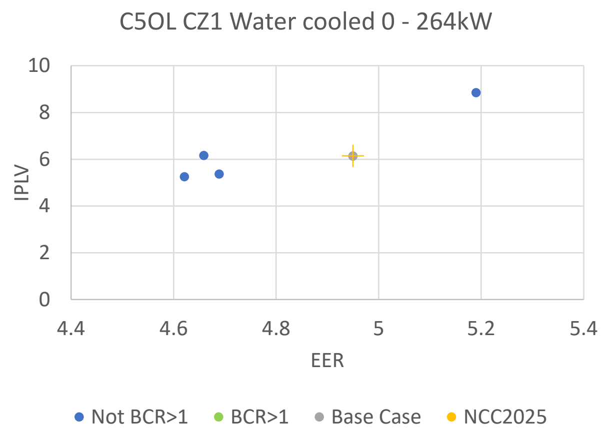


Figure 188 C50L CZ1 Water cooled 0-264 KW.

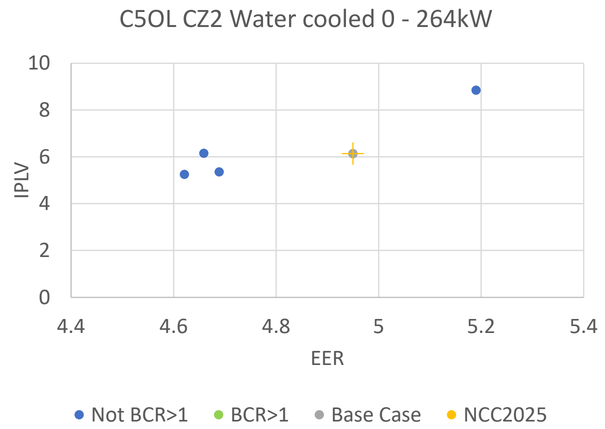


Figure 189 C50L CZ2 Water cooled 0-264 KW.

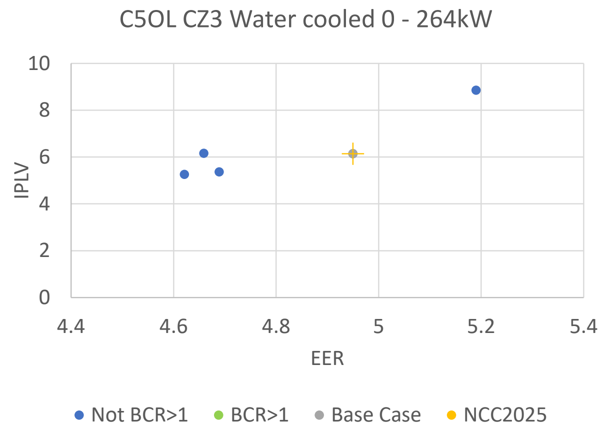


Figure 190 C50L CZ3 Water cooled 0-264 KW.

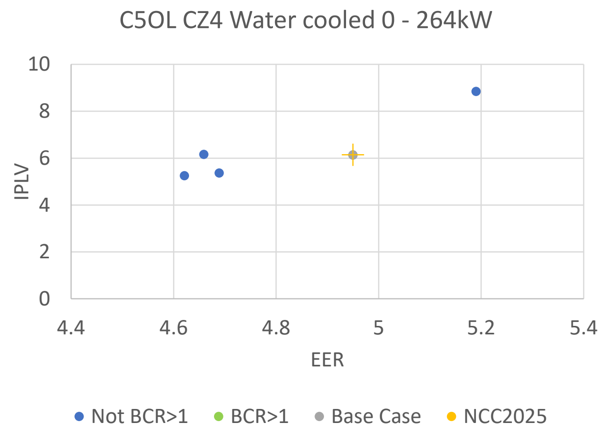


Figure 191 C50L CZ4 Water cooled 0-264 KW.

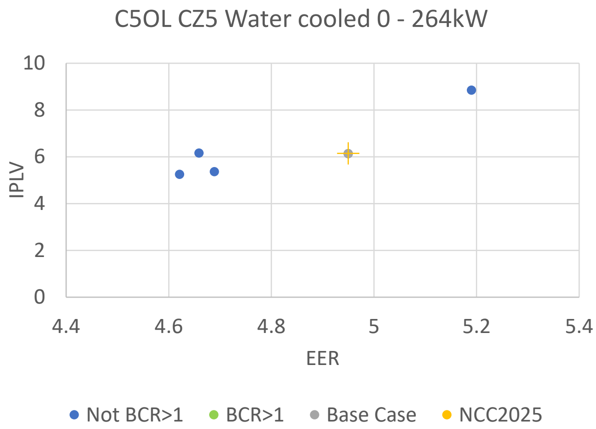


Figure 192 C50L CZ5 Water cooled 0-264 KW.

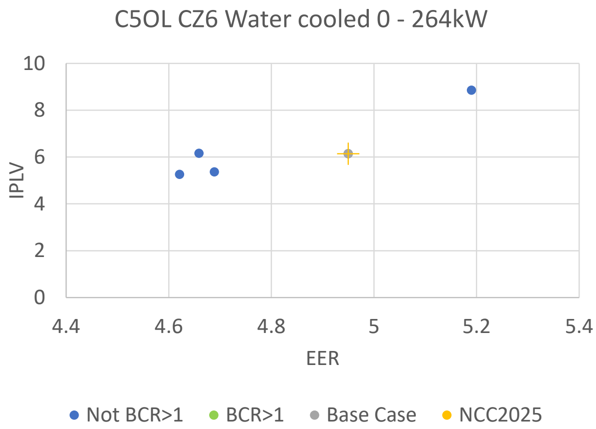


Figure 193 C50L CZ6 Water cooled 0-264 KW.

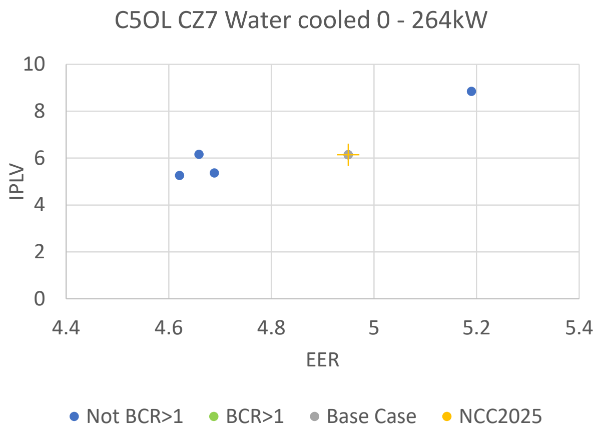


Figure 194 C50L CZ7 Water cooled 0-264 KW.

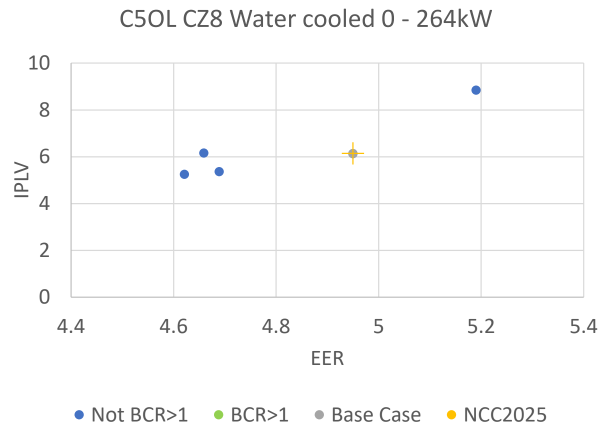


Figure 195 C50L CZ8 Water cooled 0-264 KW.

12.3.11 C50L Water cooled 264-528 KW

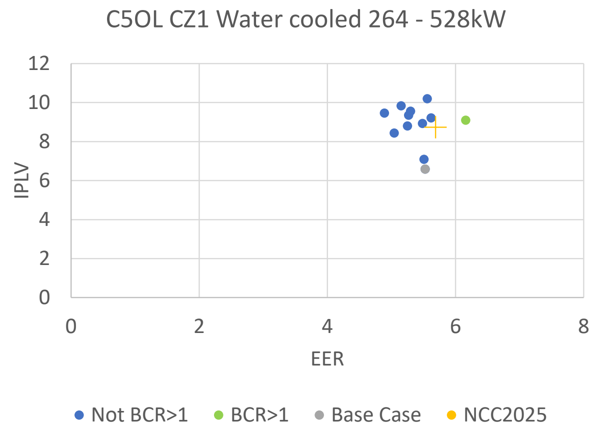


Figure 196 C50L CZ1 Water cooled 264-528 KW.

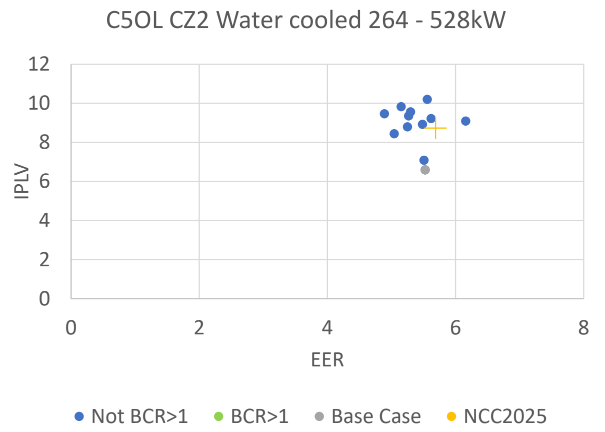


Figure 197 C50L CZ2 Water cooled 264-528 KW.

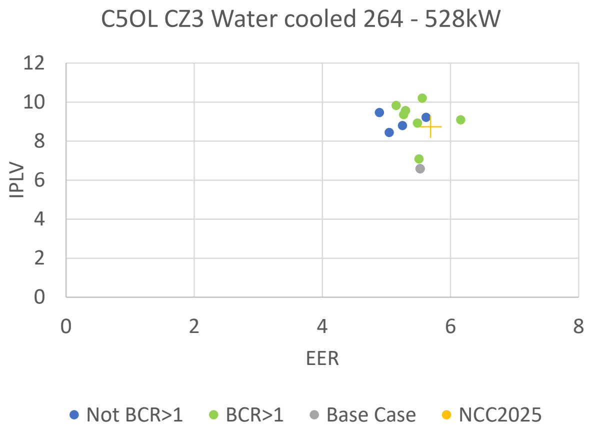


Figure 198 C50L CZ3 Water cooled 264-528 KW.

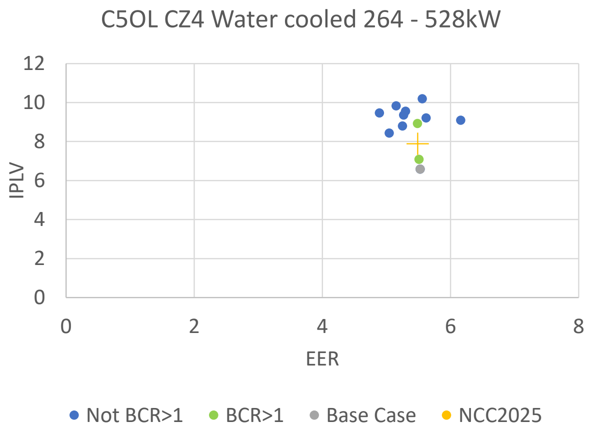


Figure 199 C50L CZ4 Water cooled 264-528 KW.

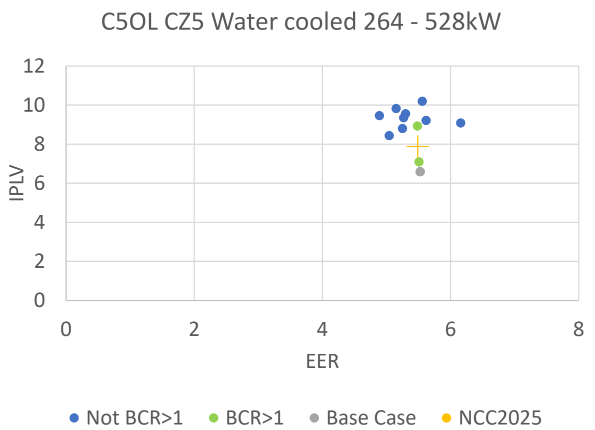


Figure 200 C50L CZ5 Water cooled 264-528 KW.

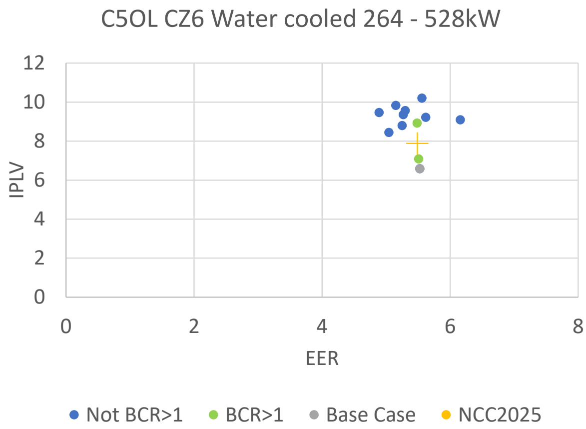


Figure 201 C50L CZ6 Water cooled 264-528 KW.

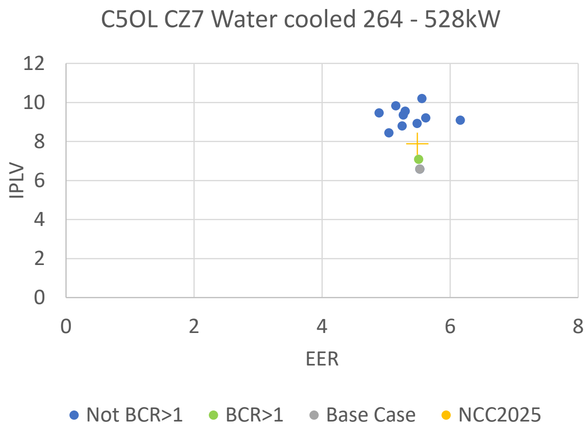


Figure 202 C50L CZ7 Water cooled 264-528 KW.

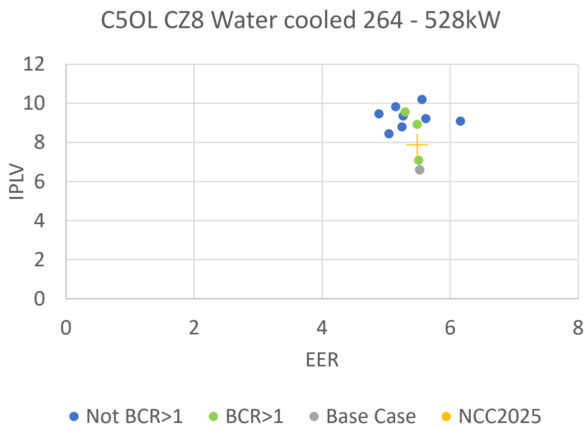


Figure 203 C50L CZ8 Water cooled 264-528 KW.

12.3.12 C50L Water cooled 528-1055 KW

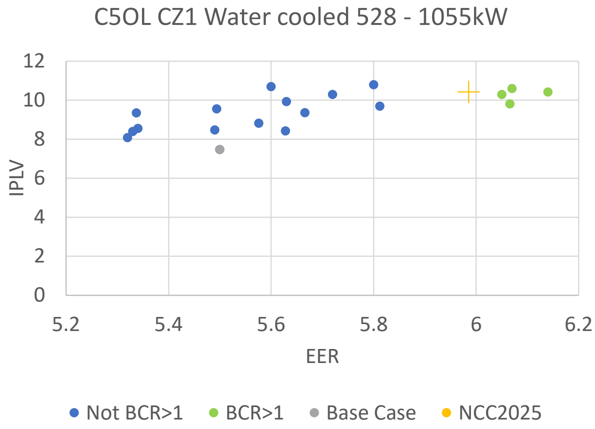


Figure 204 C50L CZ1 Water cooled 528-1055 KW.

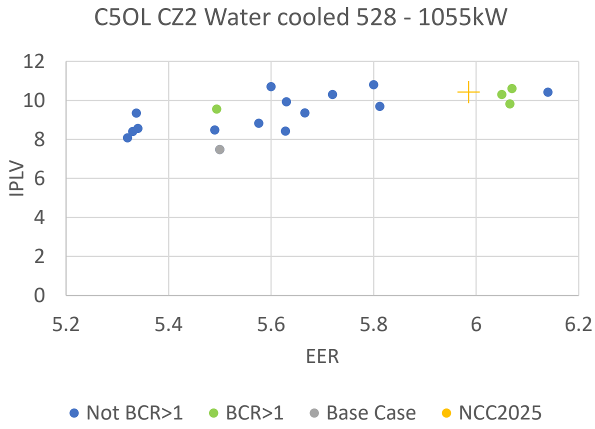


Figure 205 C50L CZ2 Water cooled 528-1055 KW.

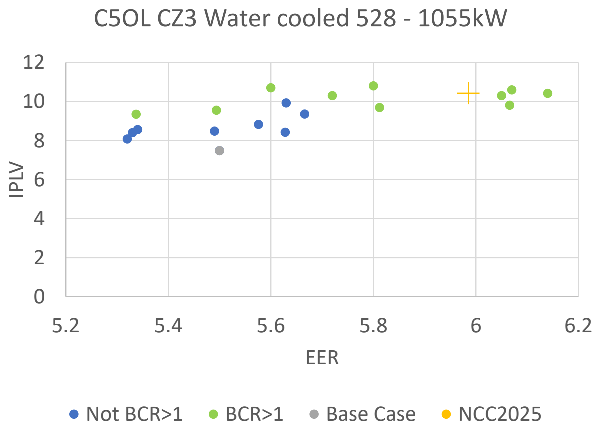


Figure 206 C50L CZ3 Water cooled 528-1055 KW.

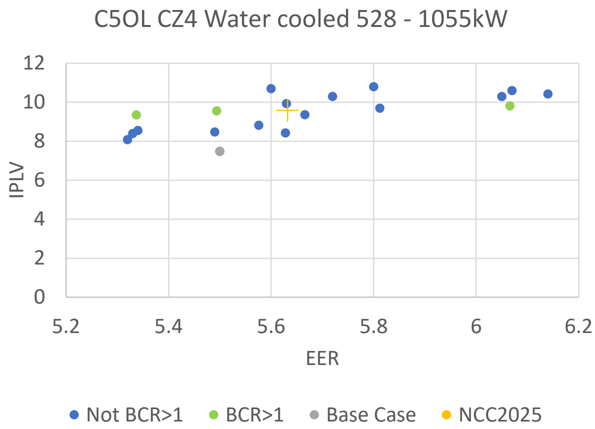


Figure 207 C50L CZ4 Water cooled 528-1055 KW.

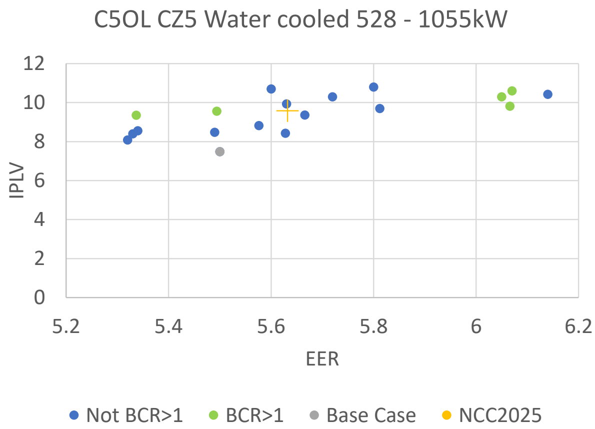


Figure 208 C50L CZ5 Water cooled 528-1055 KW.

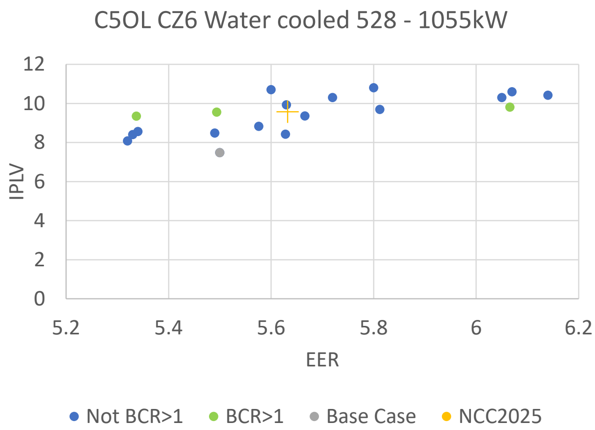


Figure 209 C50L CZ6 Water cooled 528-1055 KW.

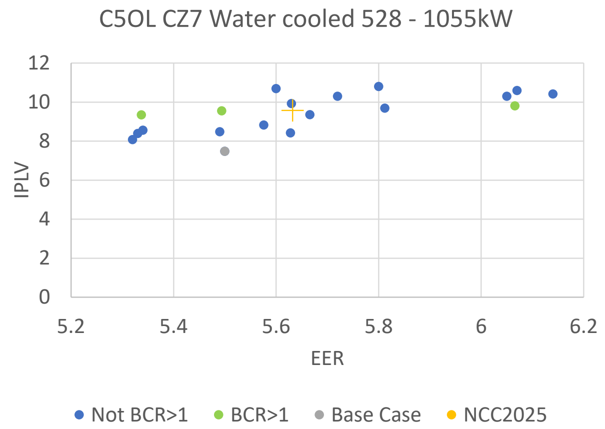


Figure 210 C50L CZ7 Water cooled 528-1055 KW.

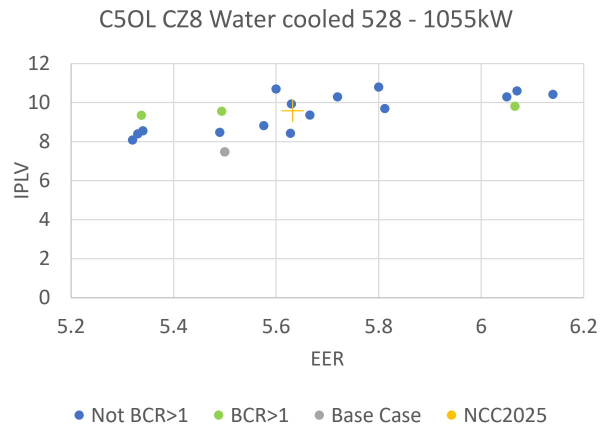


Figure 211 C50L CZ8 Water cooled 528-1055 KW.

12.3.13 C50L Water cooled 1055-1407 KW

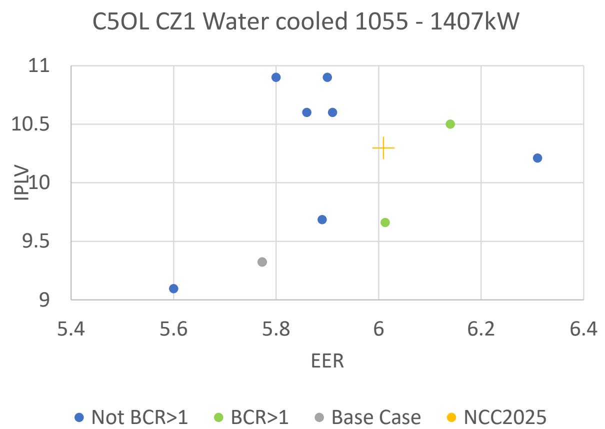


Figure 212 C50L CZ1 Water cooled 1055-1407 KW.

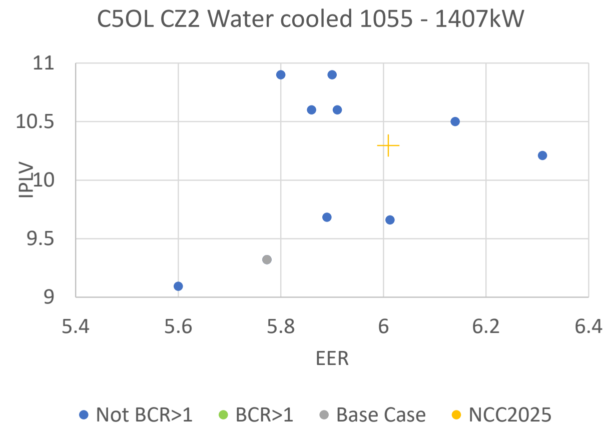


Figure 213 C50L CZ2 Water cooled 1055-1407 KW.

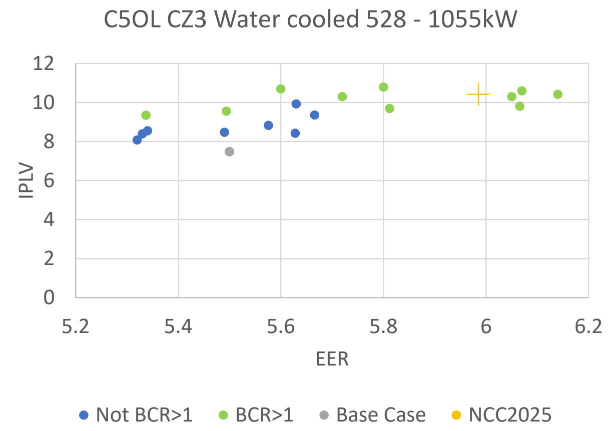


Figure 214 C50L CZ3 Water cooled 1055-1407 KW.

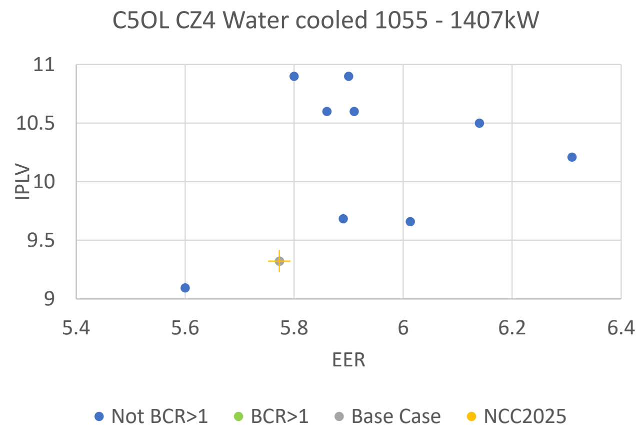


Figure 215 C50L CZ4 Water cooled 1055-1407 KW.

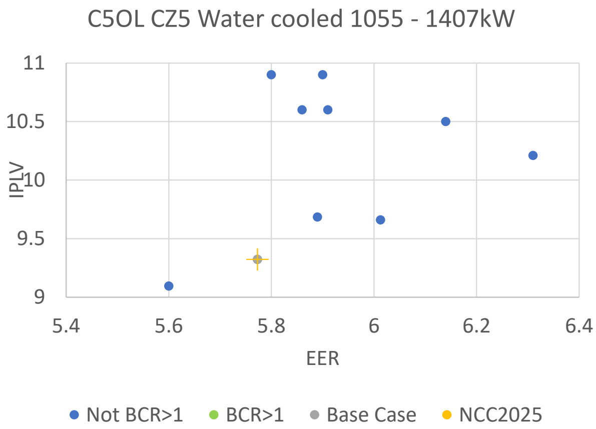


Figure 216 C50L CZ5 Water cooled 1055-1407 KW.

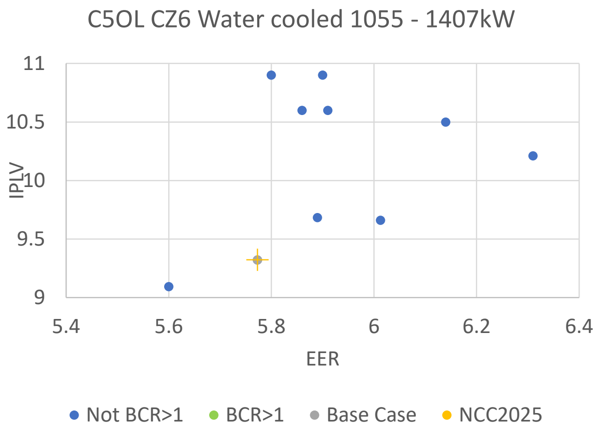


Figure 217 C50L CZ6 Water cooled 1055-1407 KW.

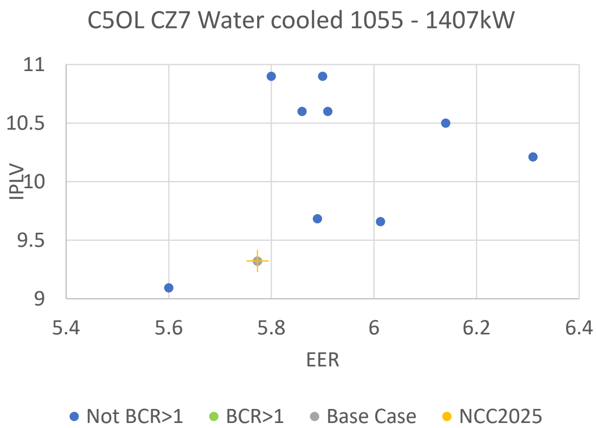


Figure 218 C50L CZ7 Water cooled 1055-1407 KW.

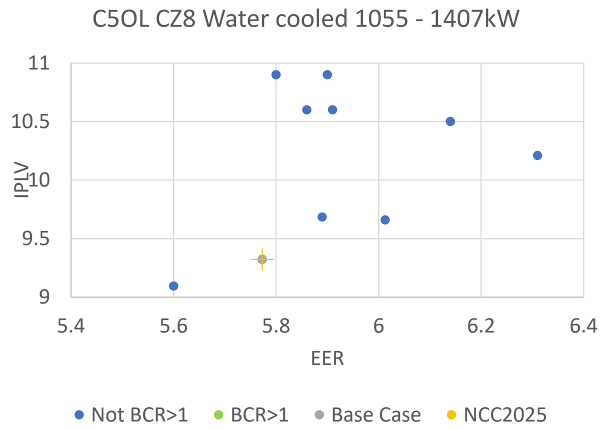


Figure 219 C50L CZ8 Water cooled 1055-1407 KW.

12.3.14 C50L Water cooled >1407 KW

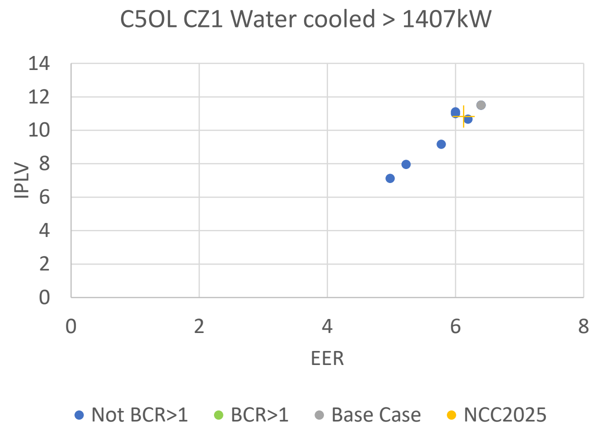


Figure 220 C50L CZ1 Water cooled >1407 KW.

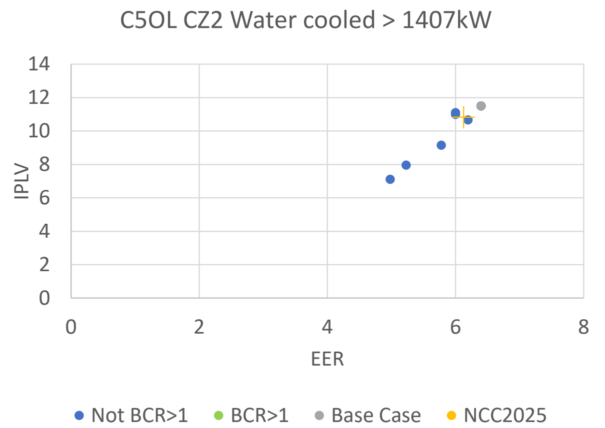


Figure 221 C50L CZ2 Water cooled >1407 KW.

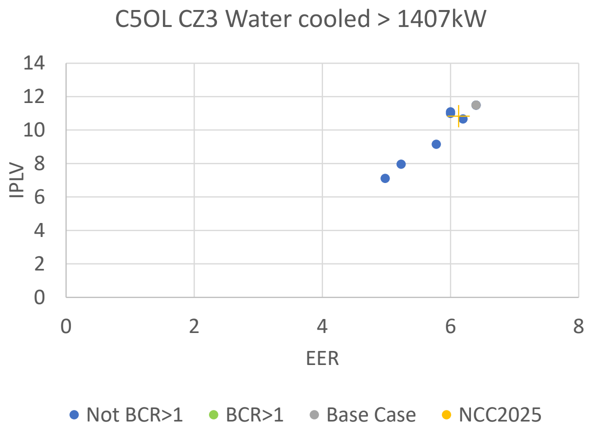


Figure 222 C50L CZ3 Water cooled >1407 KW.

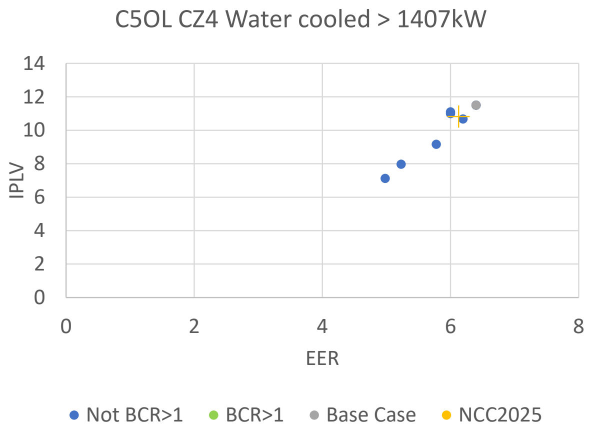


Figure 223 C50L CZ4 Water cooled >1407 KW.

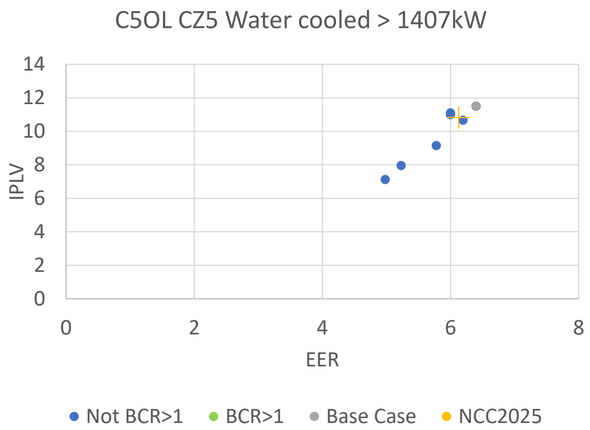


Figure 224 C50L CZ5 Water cooled >1407 KW.

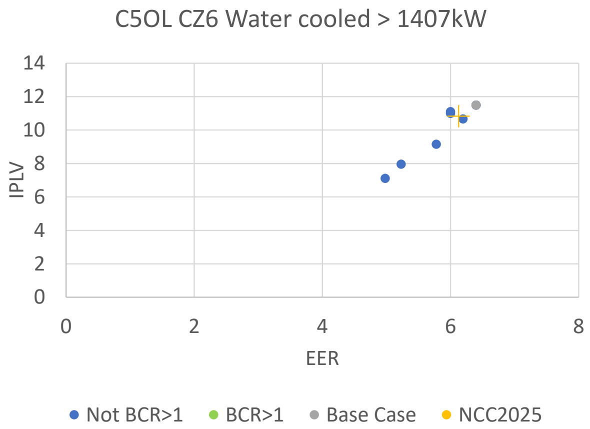


Figure 225 C50L CZ6 Water cooled >1407 KW.

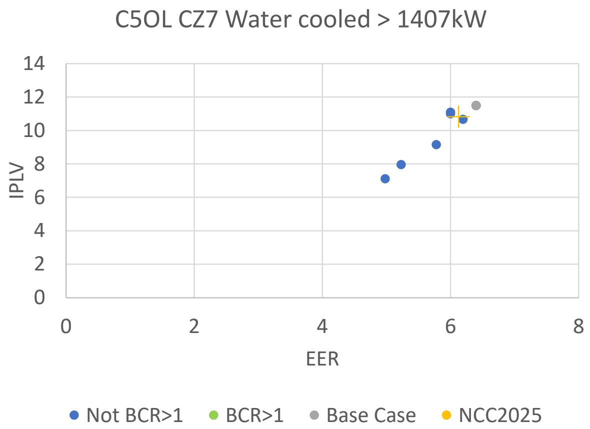


Figure 226 C50L CZ7 Water cooled >1407 KW.

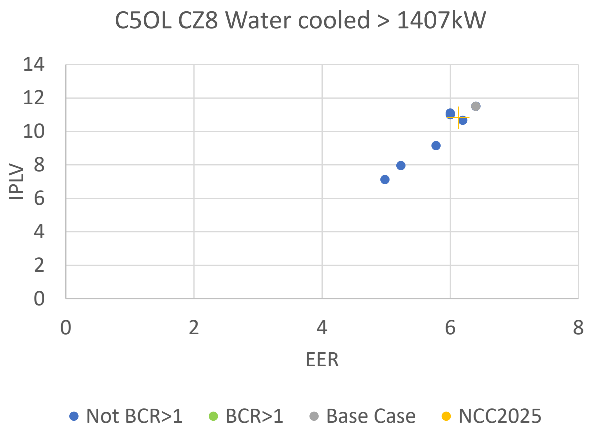


Figure 227 C50L CZ8 Water cooled >1407 KW.

12.4 Refinement of notional EER/IPLV values

The following graphs illustrate the process form step 11 of the simulation methodology in Section 2.4.3, in which results from the step 10 individual climate zone analyses (as shown in Section 12.3) are rationalised to remove minor variances and inconsistencies.

12.4.1 C50L Air cooled 0-528 KW

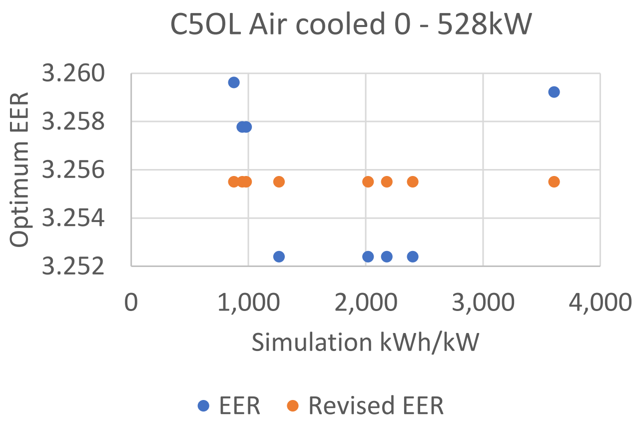


Figure 228 EER C50L Air cooled 0-528 KW.

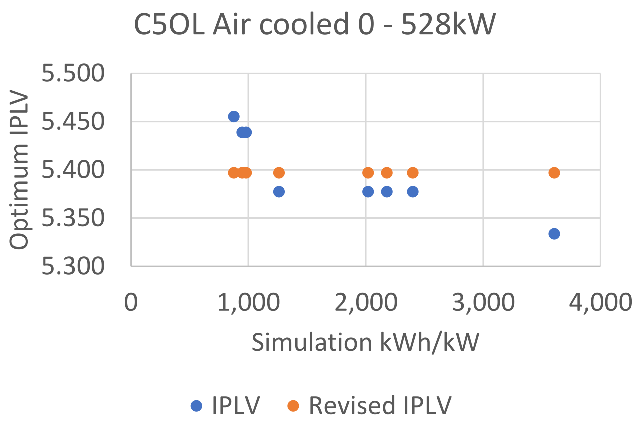


Figure 229 IPLV C50L Air cooled 0-528 KW.

12.4.2 C50L Air cooled >528 KW

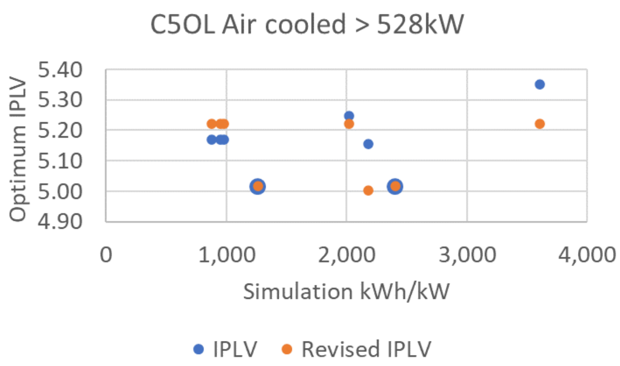


Figure 230 EER C50L Air cooled >528 KW.

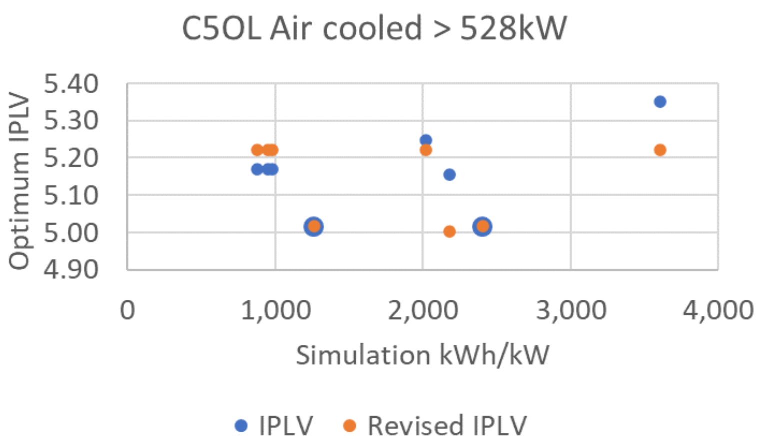


Figure 231 IPLV C50L Air cooled >528 KW.

12.4.3 C50L Water cooled 0-264 KW

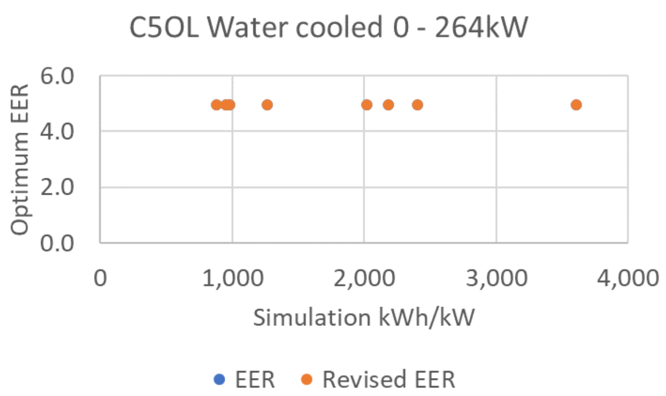


Figure 232 EER C50L Water cooled 0-264 KW.

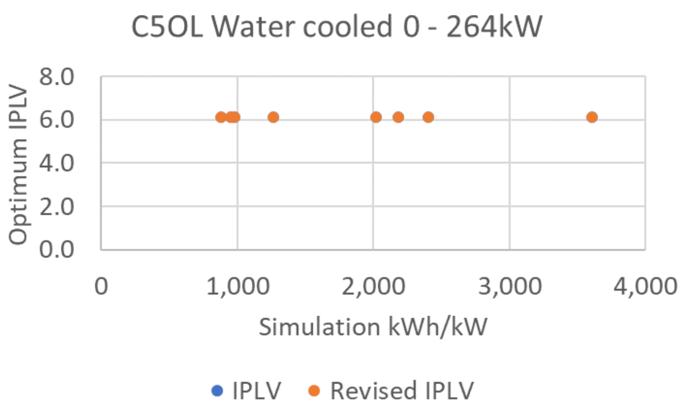


Figure 233 IPLV C50L Water cooled 0-264 KW.

12.4.4 C5OL Water cooled 264-528 KW

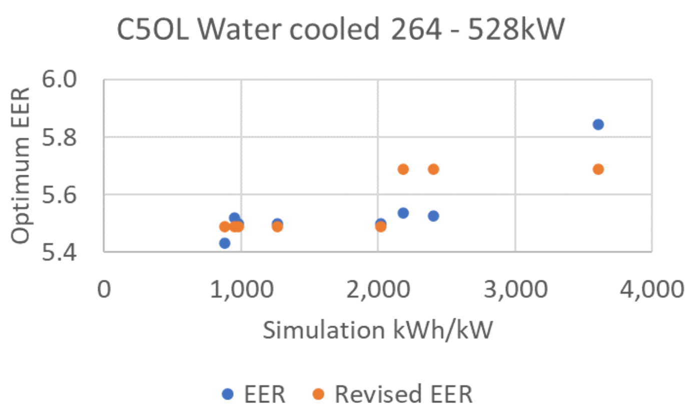


Figure 234 EER C5OL Water cooled 264-528 KW.

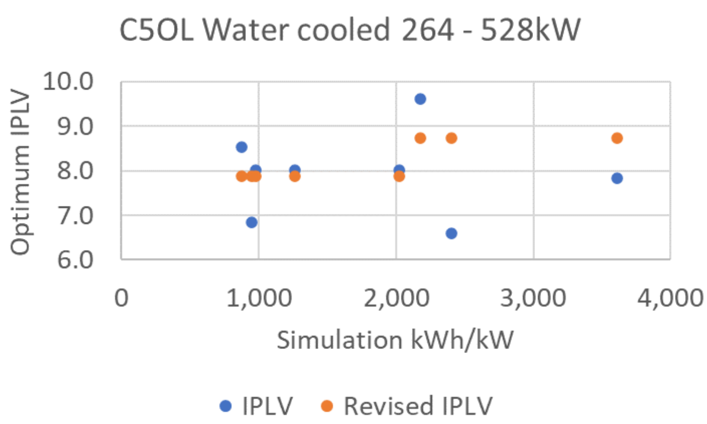


Figure 235 IPLV C5OL Water cooled 264-528 KW.

12.4.5 C5OL Water cooled 528-1055 KW

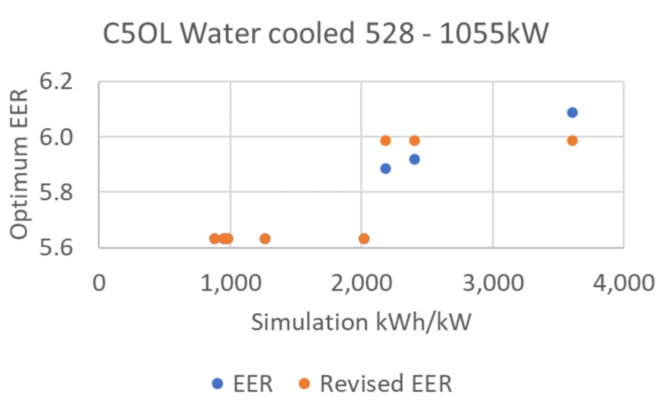


Figure 236 EER C5OL Water cooled 528-1055 KW.

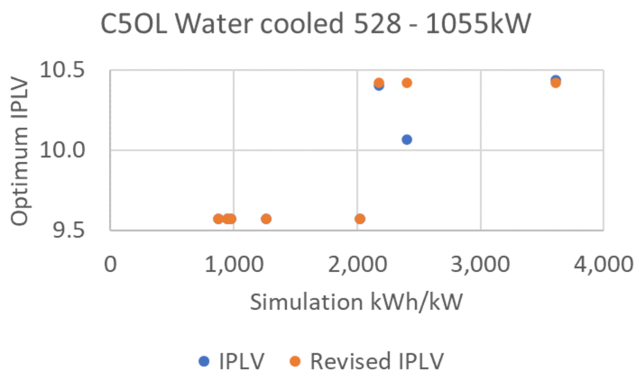


Figure 237 IPLV C5OL Water cooled 528-1055 KW.

12.4.6 C5OL Water cooled 1055-1407 KW

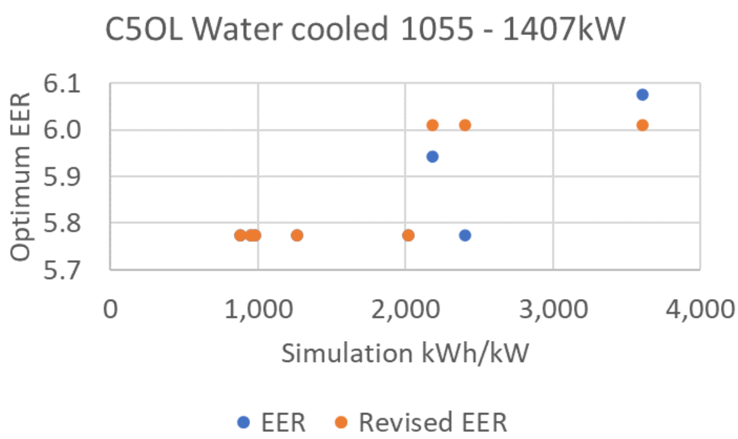


Figure 238 EER C5OL Water cooled 1055-1407 KW.

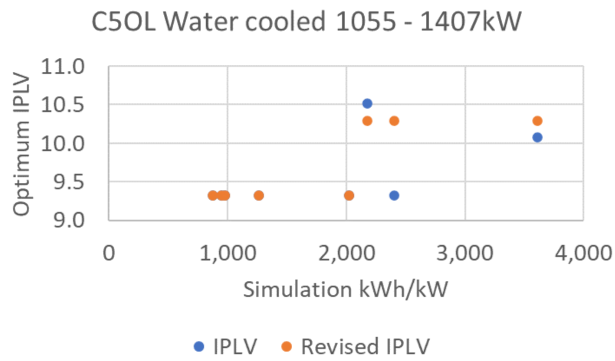


Figure 239 IPLV C5OL Water cooled 1055-1407 KW.

12.4.7 C5OL Water cooled >1407 KW

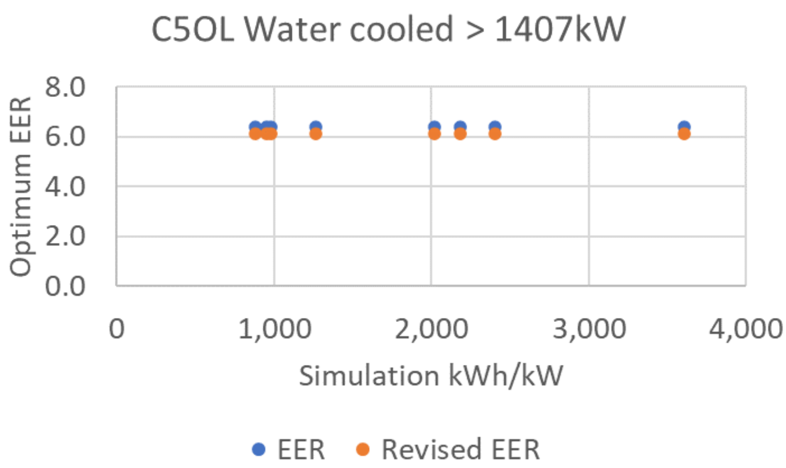


Figure 240 EER C5OL Water cooled >1407 KW.

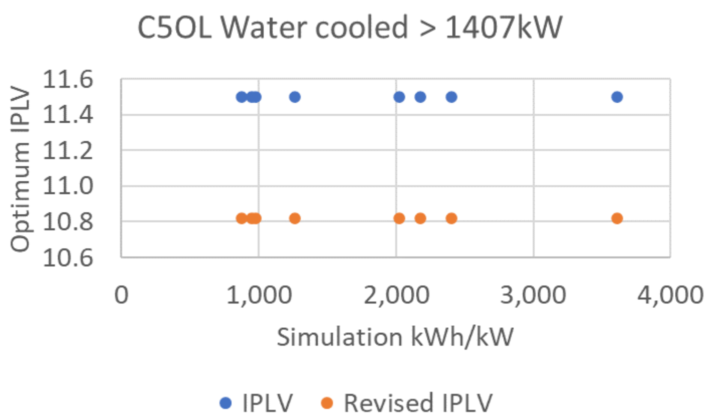


Figure 241 IPLV C5OL Water cooled >1407 KW.

12.4.8 C9A Air cooled 0-528 KW

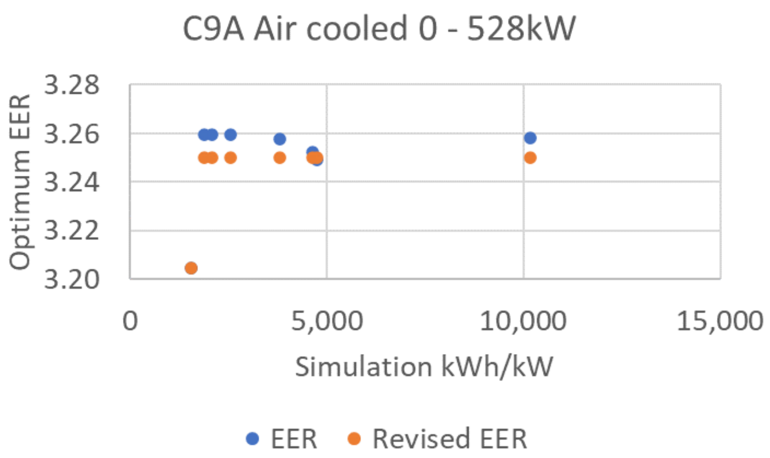


Figure 242 EER C9A Air cooled 0-528 KW.

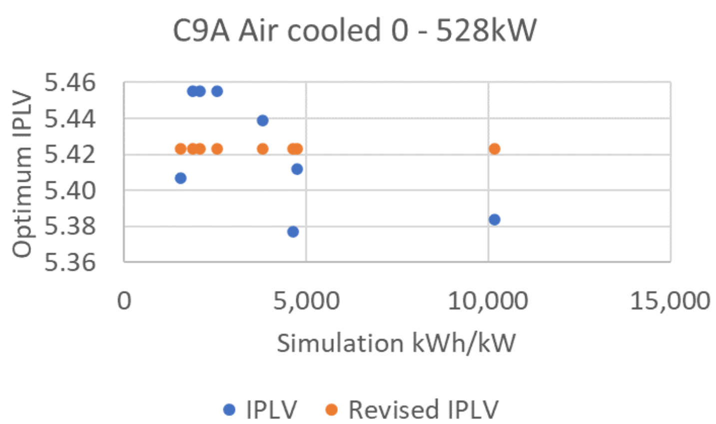


Figure 243 IPLV C9A Air cooled 0-528 KW.

12.4.9 C9A Air cooled >528 KW

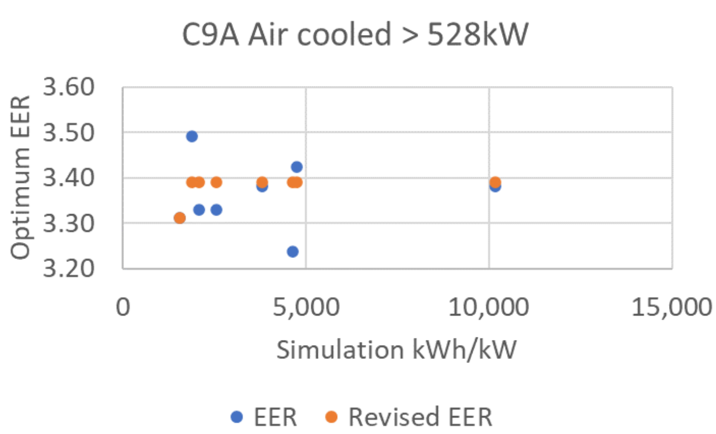


Figure 244 EER C9A Air cooled >528 KW.

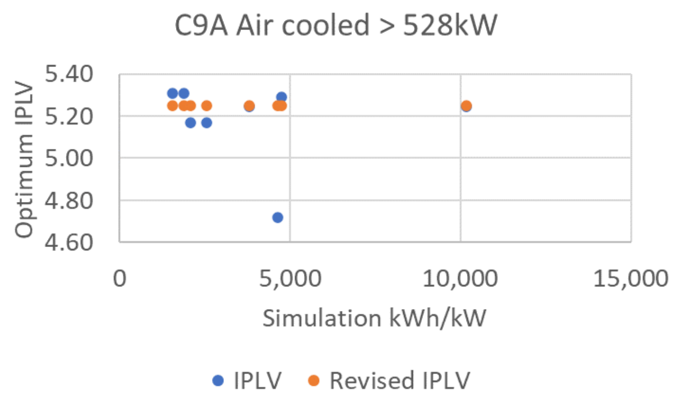


Figure 245 IPLV C9A Air cooled >528 KW.

12.4.10 C9A Water cooled 0-264 KW

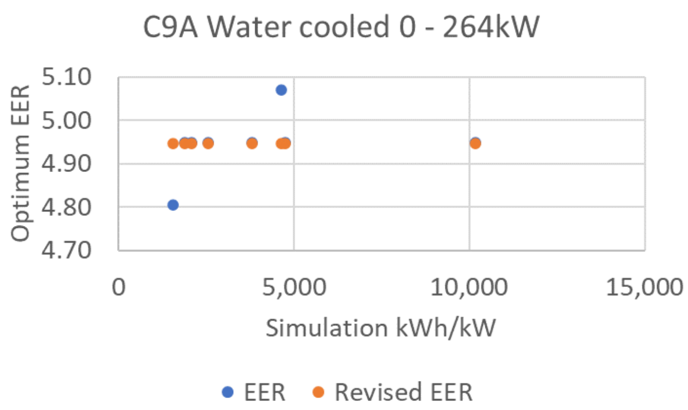


Figure 246 EER C9A Water cooled 0-264 KW.

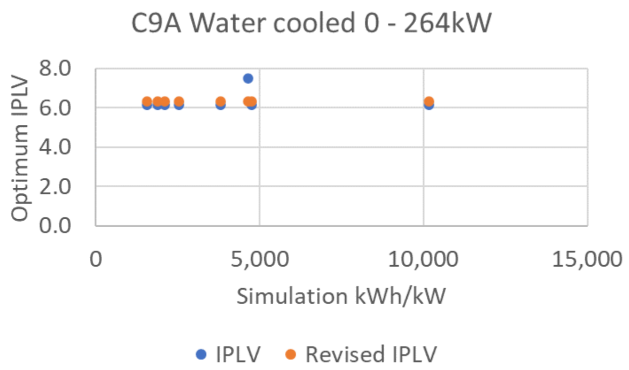


Figure 247 IPLV C9A Water cooled 0-264 KW.

12.4.11 C9A Water cooled 264-528 KW

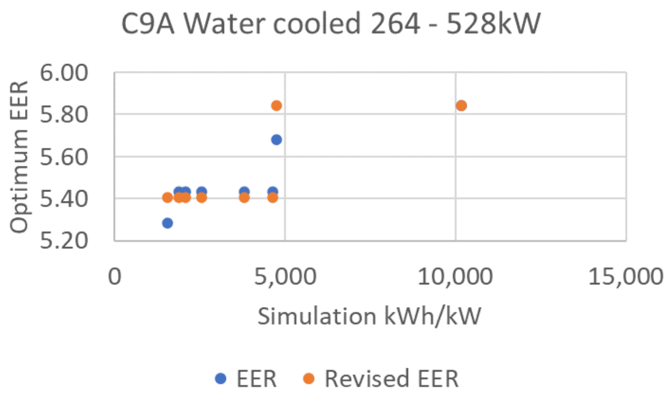


Figure 248 EER C9A Water cooled 264-528 KW.

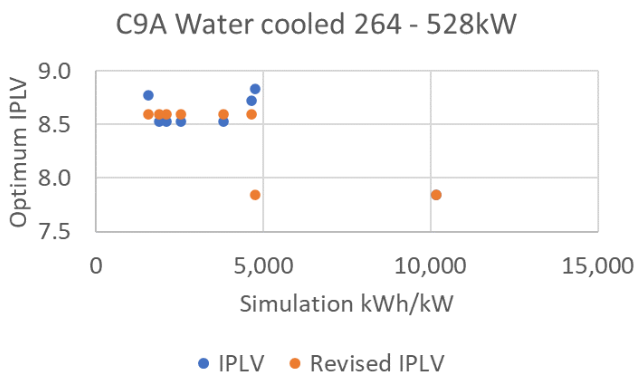


Figure 249 IPLV C9A Water cooled 264-528 KW.

12.4.12 C9A Water cooled 528-1055 KW

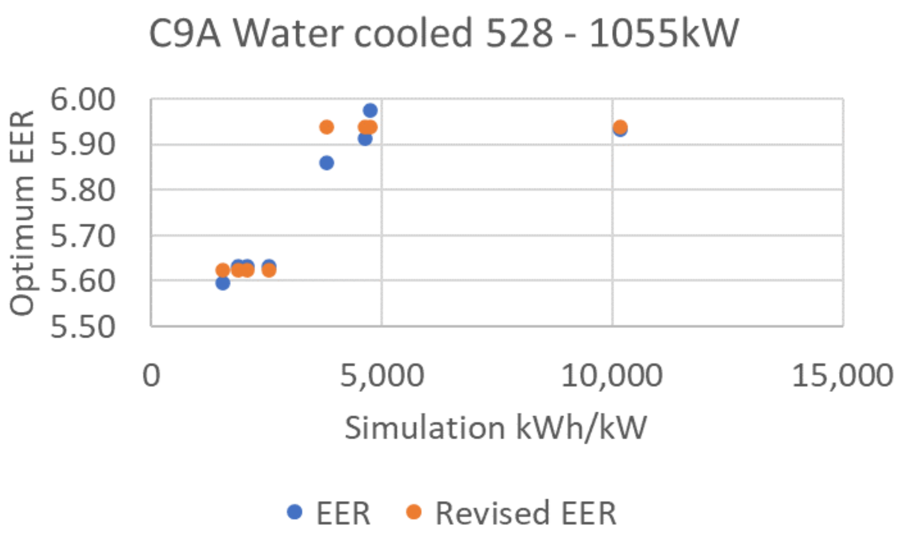


Figure 250 EER C9A Water cooled 528-1055 KW.

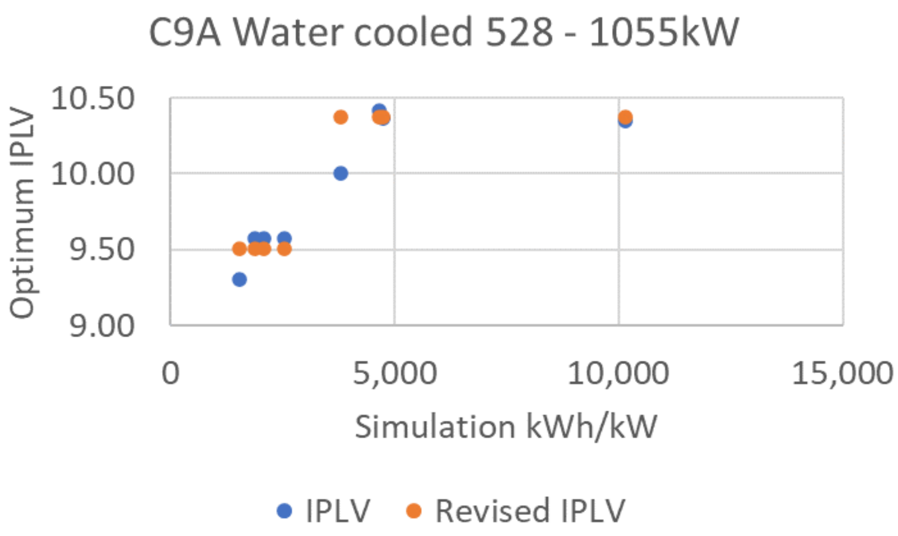


Figure 251 IPLV C9A Water cooled 528-1055 KW.

12.4.13 C9A Water cooled 1055-1407 KW

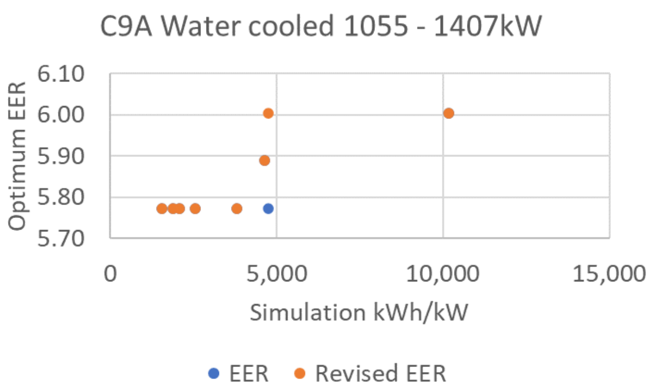


Figure 252 EER C9A Water cooled 1055-1407 KW.

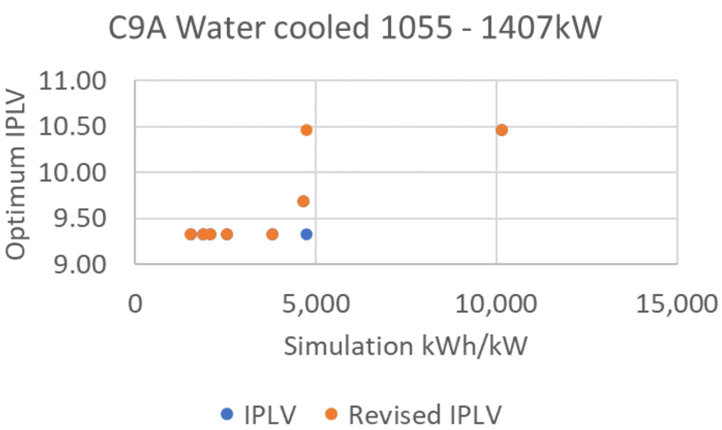


Figure 253 IPLV C9A Water cooled 1055-1407 KW.

12.4.14 C9A Water cooled >1407 KW

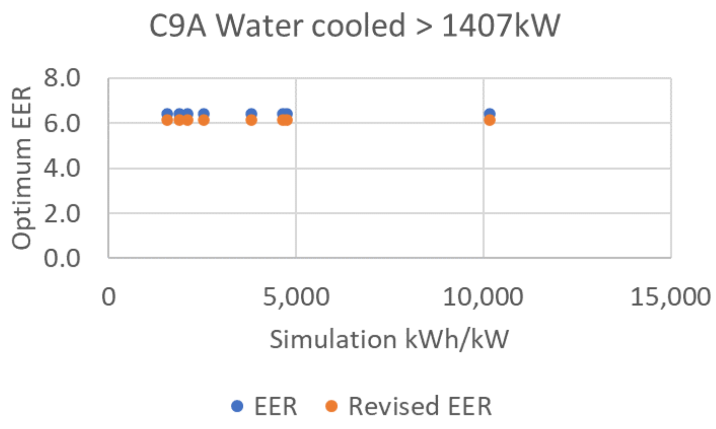


Figure 254 EER C9A Water cooled >1407 KW.

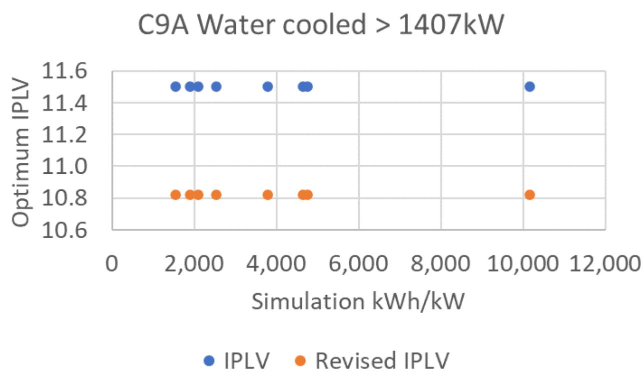


Figure 255 IPLV C9A Water cooled >1407 KW.

13 Appendix: Unitary Air-Conditioning Units

13.1 PAC unit part load curves

The cooling and heat performance of the PAC units under various operating conditions are shown below, for units with a nominal cooling EER or heating COP of 3. Data in these curves was derived from available data from actual units, as supplied by manufacturers. The data was correlated against the standard equations representing PAC unit performance used by IES.

13.1.1 Fixed speed compressor

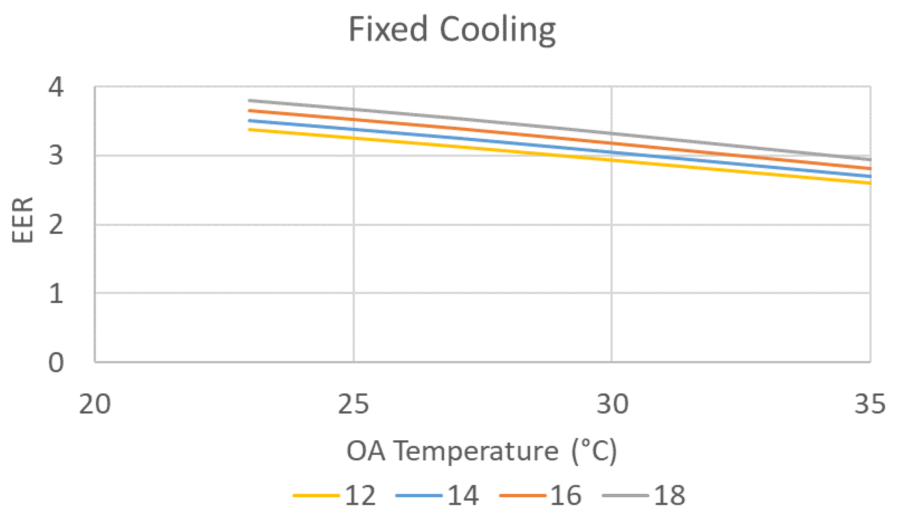


Figure 256. Cooling EER vs OAT at 100% load at different on-coil WB temperatures for fixed speed PAC unit with EER=3

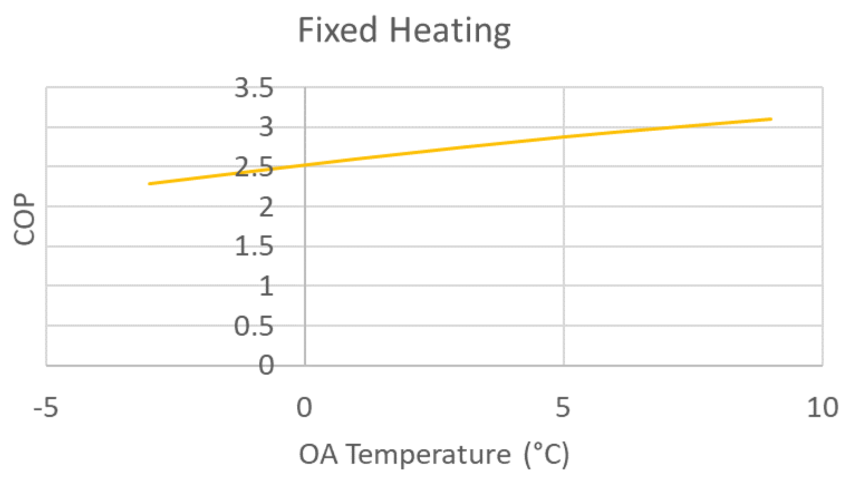


Figure 257. Heating COP vs OAT at 100% load at 20°C DB on-coil temperature for fixed speed PAC unit with COP=3.

13.1.2 Variable speed compressor

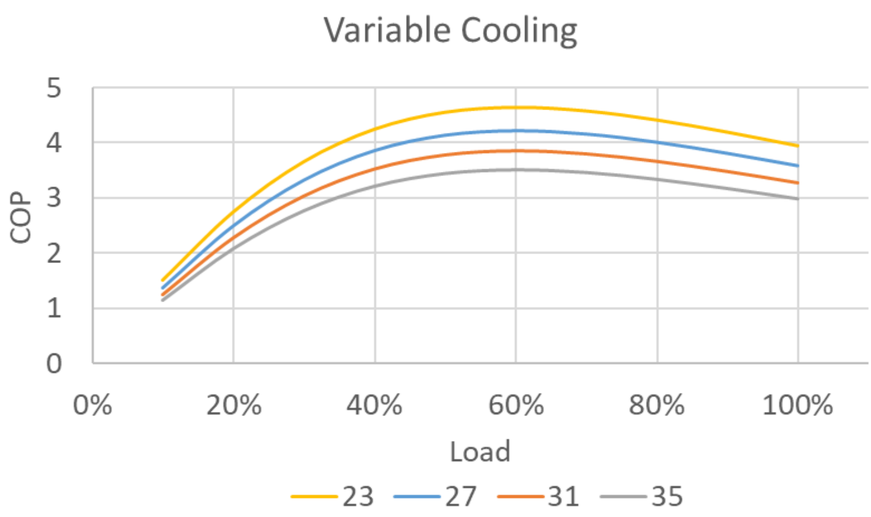


Figure 258. Cooling EER vs part load at different OAT for variable speed PAC unit with EER=3.

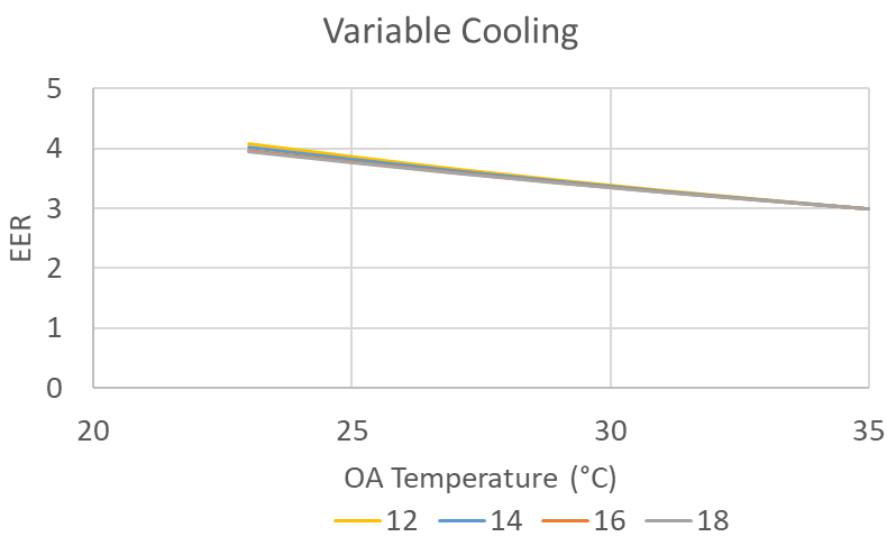


Figure 259. Cooling EER vs OAT at 100% load at different on-coil WB temperatures for variable speed PAC unit with EER=3.

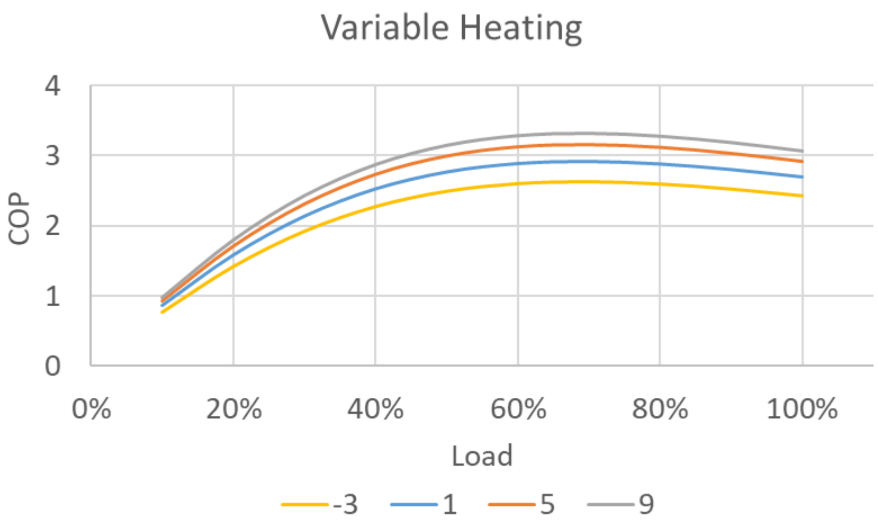


Figure 260. Heating COP vs part load at different outside air temperatures for variable speed PAC unit with EER=3.

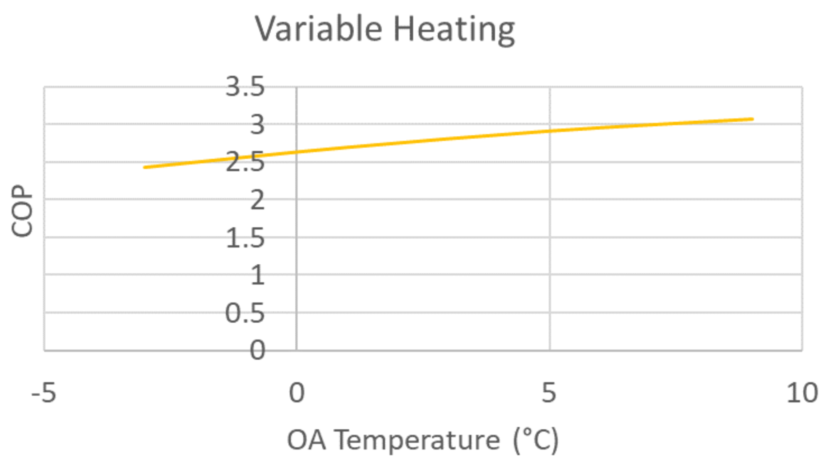


Figure 261. Heating COP vs OAT at 100% at different20°C DB on-coil DB temperatures for variable speed PAC unit with COP=3.

13.2 PAC Unit Data

Table 67 Fixed PACS

Cooling Nominal Capacity (kWr)	EER	Heating Nominal Capacity (kWr)	COP	Number of Compressors
11.6	3.35	10.8	3.58	1
16	3.34	15.24	3.25	1
16.1	3.24	14.4	3.52	1
18.6	3.3	16.7	3.52	1
19.06	3.25	19.9	3.53	1
20	3.2	19	3.55	1
22.35	3.39	23	3.74	1
27	3.24	25.68	3.5	1
28.3	3.35	28.6	3.59	1
28.3	3.35	28.6	3.59	1
32	3.35	33	3.57	1
32.5	3.27	32.9	3.25	1
32.5	3.3	35	3.52	2
33	3.31	32.9	3.35	2
34	3.14	35.78	3.3	2
39.2	3.07	39.5	3.3	2
39.2	3.07	39.5	3.3	2
43	3.14	43.1	3.35	2
49.1	3.2	51.8	3.37	2
53	3.1	56.1	3.61	2
60.8	2.97	62.47	3.1	2
64	3.1	63.95	3.44	2
69.2	2.98	71.5	3.04	2
80.04	3.02	81.95	3.09	3
97	2.98	95.5	3.51	3

Table 68 Digital PACs

Cooling Nominal Capacity (kWr)	EER	Heating Nominal Capacity (kWr)	COP	Number of Compressors
18.2	3.17	16.2	3.44	1
20	3.14	18.1	3.33	1
29.2	3.34	29.7	3.49	1
29.9	3.56	29	3.82	1
32.4	3.24	33.5	3.35	1
43.9	3.23	41.1	3.02	2
52.9	2.92	53.4	2.95	2
67.9	3.3	67.5	3.28	2
79.4	3.1	78	3.05	2

Table 69 Variable PACS

Cooling Nominal Capacity (kWr)	EER	Heating Nominal Capacity (kWr)	COP	Number of Compressors
14.65	3.35	16	3.51	1
17	3.34	18.4	3.5	1
19	3.33	20	3.64	1
21	3.44	23	3.5	1
25.4	3.3	24.5	2.27	1
33	3.28	32.5	3.43	1
35.9	3.23	37	3.22	1
44.9	3.1	44.4	3.13	1
55.6	3.11	57	2.98	1
94.3	3.16	95.8	3.34	2
135	3.01	134	3.35	2
155	3.37	145	3.36	4
187.5	3.04	180.45	3.03	2
200	3.03	203	3.54	4

13.2.1 VRF data

Table 70 VRF above 39kW

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
39.20	8.68	4.5	44.10	9.72	4.5
40.00	11.79	3.4	45.00	13.23	3.4
40.00	11.56	3.5	40.00	9.76	4.1
40.00	11.20	3.6	45.00	12.80	3.5
40.00	10.98	3.6	45.00	10.23	4.4
40.00	10.96	3.7	45.00	10.69	4.2
40.00	10.70	3.7	45.00	11.00	4.1
44.80	10.89	4.1	50.40	12.39	4.1
44.80	10.30	4.4	50.00	11.30	4.4
45.00	14.47	3.1	45.00	11.39	4.0
45.00	13.98	3.2	50.00	12.50	4.0
45.00	13.98	3.2	50.00	12.50	4.0
45.00	12.96	3.5	50.00	12.98	3.9
45.00	12.90	3.5	50.00	13.60	3.7
45.00	12.90	3.5	50.00	12.60	4.0
45.00	12.26	3.7	50.00	11.83	4.2
45.00	11.52	3.9	45.00	10.54	4.3
47.50	14.84	3.2	47.50	11.67	4.1
47.50	13.98	3.4	53.00	13.00	4.1
47.50	13.97	3.4	53.00	12.99	4.1
50.00	15.30	3.3	56.00	14.90	3.8
50.00	15.20	3.3	50.00	12.69	3.9
50.00	14.74	3.4	56.00	15.05	3.7
50.00	14.40	3.5	56.00	14.50	3.9
50.00	14.01	3.6	56.00	13.56	4.1
50.00	13.97	3.6	56.00	13.49	4.2
50.00	13.15	3.8	50.00	12.13	4.1
50.40	14.59	3.5	56.00	14.04	4.0
50.40	12.00	4.2	56.50	12.90	4.4
50.40	10.91	4.6	56.70	11.94	4.7
55.90	13.90	4.0	62.50	14.60	4.3
56.00	17.70	3.2	63.00	17.10	3.7
56.00	17.50	3.2	63.00	17.20	3.7
56.00	17.50	3.2	63.00	16.15	3.9
56.00	16.91	3.3	63.00	17.54	3.6
56.00	16.62	3.4	63.00	15.95	4.0
56.00	16.00	3.5	63.00	16.57	3.8

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
56.00	14.78	3.8	56.00	13.72	4.1
56.00	14.51	3.9	63.00	14.82	4.3
56.00	12.77	4.4	63.00	14.69	4.3
60.80	13.70	4.4	68.00	15.10	4.5
61.50	19.25	3.2	69.00	20.25	3.4
61.50	18.60	3.3	69.00	19.60	3.5
61.50	17.04	3.6	61.50	15.30	4.0
61.50	16.24	3.8	69.00	16.44	4.2
61.50	16.20	3.8	69.00	16.32	4.2
61.50	15.50	4.0	69.00	16.10	4.3
61.60	15.70	3.9	69.30	16.76	4.1
63.00	18.91	3.3	69.00	19.22	3.6
67.00	22.64	3.0	69.00	18.93	3.6
67.00	21.30	3.1	75.00	22.20	3.4
67.00	19.30	3.5	67.00	16.88	4.0
67.00	17.96	3.7	75.00	18.06	4.2
67.00	17.92	3.7	75.00	18.08	4.2
67.00	17.40	3.9	75.00	17.80	4.2
67.20	17.40	3.9	74.30	18.80	4.0
67.20	15.50	4.3	75.00	17.00	4.4
69.00	21.16	3.3	76.50	22.43	3.4
72.80	20.20	3.6	74.30	19.15	3.9
72.80	17.20	4.2	81.50	18.60	4.4
73.00	22.25	3.3	81.50	23.90	3.4
73.50	21.21	3.5	73.50	18.20	4.0
73.50	19.96	3.7	82.50	19.26	4.3
73.50	19.92	3.7	82.50	19.73	4.2
73.50	19.90	3.7	82.50	22.60	3.7
73.50	19.40	3.8	82.50	19.90	4.1
73.50	18.91	3.9	73.50	17.40	4.2
76.80	18.47	4.2	88.20	20.65	4.3
78.30	19.00	4.1	87.50	20.30	4.3
78.50	21.60	3.6	87.50	23.40	3.7
78.50	21.60	3.6	87.50	21.50	4.1
80.00	24.84	3.2	88.00	27.24	3.2
80.00	23.12	3.5	80.00	19.52	4.1
80.00	21.96	3.6	90.00	20.45	4.4
80.00	21.92	3.7	90.00	21.38	4.2
80.00	20.54	3.9	80.00	18.99	4.2
81.50	18.49	4.4	94.50	20.20	4.7

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
83.50	24.00	3.5	93.50	23.80	3.9
83.50	23.10	3.6	93.50	24.30	3.8
83.90	20.70	4.1	94.00	21.80	4.3
84.00	21.76	3.9	94.50	22.23	4.3
85.00	27.68	3.1	95.00	29.68	3.2
85.00	26.03	3.3	85.00	21.15	4.0
85.00	24.94	3.4	95.00	23.19	4.1
85.00	24.96	3.4	95.00	22.73	4.2
85.00	22.17	3.8	85.00	20.58	4.1
89.40	22.60	4.0	100.00	23.50	4.3
89.50	26.20	3.4	101.00	27.30	3.7
89.50	23.49	3.8	100.50	23.85	4.2
89.60	20.35	4.4	100.80	22.95	4.4
90.00	29.50	3.1	100.00	31.54	3.2
90.00	28.94	3.1	90.00	22.78	4.0
90.00	27.96	3.2	100.00	25.00	4.0
90.00	27.95	3.2	100.00	25.00	4.0
90.00	25.80	3.5	100.00	25.20	4.0
90.00	24.43	3.7	90.00	22.16	4.1
95.00	29.68	3.2	95.00	23.34	4.1
95.00	28.20	3.4	106.00	27.50	3.9
95.00	27.96	3.4	106.00	26.00	4.1
95.00	27.94	3.4	106.00	25.98	4.1
95.00	26.69	3.6	95.00	23.74	4.0
95.00	25.22	3.8	106.50	25.47	4.2
95.00	24.20	3.9	107.00	25.10	4.3
95.20	23.28	4.1	107.10	25.02	4.3
96.00	33.10	2.9	108.00	34.28	3.2
96.00	28.70	3.3	108.00	30.00	3.6
100.00	30.40	3.3	100.00	25.38	3.9
100.00	28.02	3.6	112.00	27.12	4.1
100.00	27.94	3.6	112.00	26.98	4.2
100.50	26.94	3.7	112.00	27.09	4.1
100.80	24.98	4.0	112.10	27.06	4.1
101.00	35.06	2.9	113.00	36.21	3.1
101.00	30.60	3.3	113.00	29.70	3.8
101.00	30.40	3.3	113.00	30.80	3.7
101.00	26.10	3.9	113.00	26.70	4.2
106.00	31.90	3.3	119.00	31.70	3.8
106.00	31.51	3.4	119.00	29.71	4.0

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
106.00	30.59	3.5	119.00	29.44	4.0
106.40	26.08	4.1	118.40	28.52	4.2
107.00	28.10	3.8	120.00	28.80	4.2
112.00	35.00	3.2	126.00	34.40	3.7
112.00	35.00	3.2	126.00	32.31	3.9
112.00	33.24	3.4	126.00	31.90	4.0
112.00	30.30	3.7	125.00	30.40	4.1
112.00	28.29	4.0	124.70	31.19	4.0
117.00	35.70	3.3	131.00	36.70	3.6
117.60	28.31	4.2	131.00	30.74	4.3
118.00	32.60	3.6	132.00	32.40	4.1
120.00	34.68	3.5	120.00	29.28	4.1
120.00	32.94	3.6	135.00	30.68	4.4
120.00	32.88	3.7	135.00	32.07	4.2
123.00	38.80	3.2	138.00	39.40	3.5
123.20	30.17	4.1	137.30	33.49	4.1
124.00	34.50	3.6	138.00	34.10	4.0
125.00	37.59	3.3	125.00	30.91	4.0
125.00	35.90	3.5	140.00	33.88	4.1
125.00	35.94	3.5	140.00	32.95	4.3
128.80	33.10	3.9	143.60	35.56	4.0
129.00	40.20	3.2	144.00	41.80	3.4
130.00	40.50	3.2	130.00	32.54	4.0
130.00	38.92	3.3	145.00	35.69	4.1
130.00	38.93	3.3	145.00	35.23	4.1
130.00	36.50	3.6	145.00	36.20	4.0
134.00	42.60	3.1	150.00	44.40	3.4
134.40	34.80	3.9	148.50	37.60	3.9
135.00	43.41	3.1	135.00	34.17	4.0
135.00	41.90	3.2	150.00	37.50	4.0
135.00	41.93	3.2	150.00	37.50	4.0
135.00	38.70	3.5	150.00	37.80	4.0
140.00	41.10	3.4	156.00	40.10	3.9
140.00	40.90	3.4	157.00	41.80	3.8
140.00	33.66	4.2	156.20	36.78	4.2
142.50	44.52	3.2	142.50	35.01	4.1
142.50	41.94	3.4	159.00	39.00	4.1
142.50	41.91	3.4	159.00	38.97	4.1
145.00	44.88	3.2	145.00	36.03	4.0
145.00	43.50	3.3	162.00	42.40	3.8

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
145.00	41.93	3.5	162.00	39.49	4.1
145.00	41.95	3.5	162.00	39.54	4.1
145.60	35.87	4.1	162.50	39.45	4.1
146.00	44.00	3.3	164.00	44.50	3.7
150.00	45.90	3.3	168.00	44.70	3.8
150.00	45.60	3.3	150.00	38.07	3.9
150.00	42.03	3.6	168.00	40.68	4.1
150.00	41.91	3.6	168.00	40.47	4.2
151.20	35.89	4.2	168.80	39.00	4.3
152.00	46.20	3.3	171.00	47.20	3.6
156.00	48.30	3.2	175.00	46.90	3.7
156.00	45.52	3.4	175.00	43.27	4.0
156.00	44.56	3.5	175.00	42.93	4.1
156.80	37.75	4.2	175.10	41.75	4.2
157.00	47.90	3.3	176.00	48.00	3.7
162.00	50.70	3.2	182.00	49.10	3.7
162.00	49.40	3.3	182.00	48.90	3.7
162.00	49.01	3.3	182.00	45.87	4.0
162.00	47.21	3.4	182.00	45.39	4.0
168.00	53.10	3.2	189.00	51.30	3.7
168.00	52.50	3.2	189.00	51.60	3.7
168.00	52.50	3.2	189.00	48.46	3.9
168.00	49.86	3.4	189.00	47.85	4.0

Table 71 VRF below 39kW

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
11.20	2.88	3.9	12.50	2.60	4.8
11.20	2.48	4.5	12.50	2.51	5.0
14.00	3.83	3.7	14.00	3.04	4.6
14.00	3.36	4.2	16.00	3.28	4.9
16.00	4.51	3.5	16.00	3.59	4.5
16.00	3.95	4.1	18.00	3.90	4.6
16.00	3.38	4.7	18.00	3.73	4.8
20.00	5.46	3.7	22.40	5.10	4.4
22.40	6.61	3.4	25.00	5.92	4.2
22.40	6.12	3.7	25.00	6.15	4.1

Total Cooling Capacity (kW)	Input Power (Cooling) (kW)	EER	Total Heating Capacity (kW)	Input Power (Heating) (kW)	COP
22.40	5.94	3.8	25.00	6.25	4.0
22.40	5.76	3.9	22.40	5.27	4.3
22.40	5.60	4.0	22.40	4.80	4.7
22.40	5.17	4.3	25.00	5.68	4.4
22.40	5.17	4.3	25.00	5.67	4.4
22.40	4.49	5.0	25.20	4.78	5.3
24.00	6.88	3.5	26.00	6.82	3.8
28.00	8.09	3.5	31.50	8.33	3.8
28.00	7.87	3.6	28.00	6.47	4.3
28.00	7.39	3.8	28.00	6.86	4.1
28.00	7.25	3.9	31.50	7.41	4.3
28.00	7.24	3.9	31.50	7.28	4.3
28.00	6.84	4.1	31.50	7.23	4.4
28.00	6.80	4.1	31.50	7.29	4.3
28.00	5.80	4.8	31.50	5.92	5.3
32.00	6.76	4.7	36.00	7.46	4.8
33.50	9.65	3.5	33.50	8.44	4.0
33.50	9.64	3.5	38.11	10.11	~~3.8~~
33.50	9.49	3.5	37.50	9.89	3.8
33.50	8.98	3.7	37.50	9.03	4.2
33.50	8.96	3.7	37.50	9.04	4.2
33.50	8.71	3.8	37.50	9.81	3.8
33.50	8.70	3.9	37.50	8.91	4.2
33.60	7.58	4.4	37.80	8.26	4.6
38.40	8.55	4.5	43.00	9.40	4.6

14 Appendix: Heat Pumps

14.1 Heat Pump data

The heating and cooling performance data for 49 individual heat pumps was gathered along with costs for each. As part of the exploration of available seasonal performance, equivalent performance metrics were not available across the full range. The most commonly available seasonal performance data has been compiled in to Table 72. This data availability is uniform with the 4 pipe chiller equipment in the next section, as individual manufacturers generally supply both types of equipment.

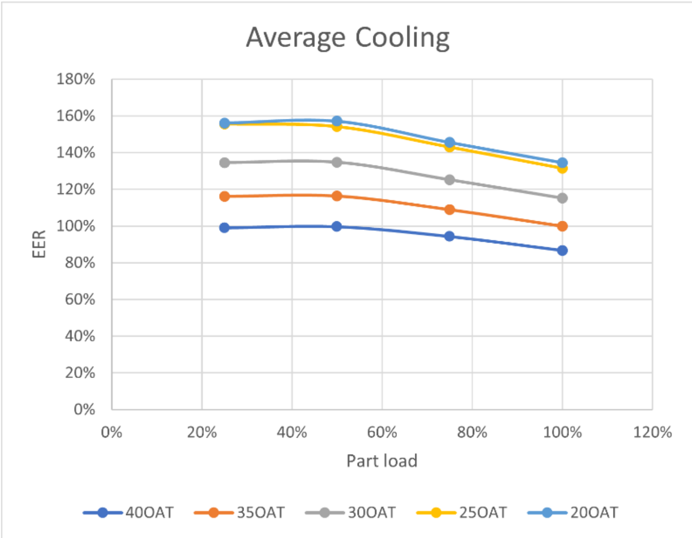


Figure 262: Average heat pump cooling performance at given load points and ambient temperatures.

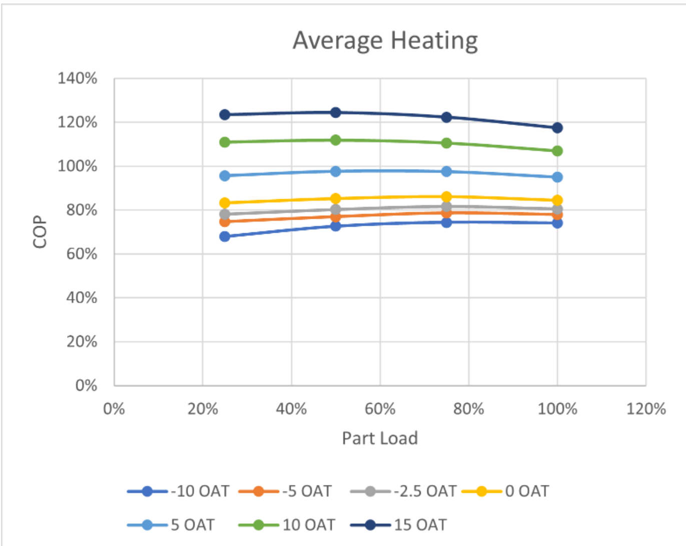


Figure 263: Average heat pump heating performance at given load points and ambient temperatures.

Table 72: Heat pump heating/cooling performance data

Cooling - Nominal Cap (kW)	Cooling - EER	Cooling - IPLV	Heating - Nominal Cap (kW)	Heating - COP	Heating - SCOP @ 35° SWT
120.8	3.14	5.42	133.9	3.3	Not Available
196.1	2.78	4.51	214.2	3.27	3.98
336.5	2.92	5.05	364.1	3.26	3.89
604.1	2.96	4.6	661.5	3.28	4.13
966.5	2.92	4.92	982.2	3.15	3.73
200.0	2.98	6.4	200.0	3.39	4.26
168.0	2.87	Not Available	191.0	3.05	3.21
424.0	2.9	Not Available	456.0	2.97	3.35
315.0	3.13	Not Available	311.0	3.17	Not Available
647.0	3.14	Not Available	621.0	3.17	Not Available
201.2	2.85	4.72	215.0	3.43	3.66
118.3	2.69	4.32	132.0	2.9	3.36
360.1	2.99	4.96	400.0	3.41	Not Available
265.0	2.65	4.2	300.0	2.88	3.27
579.3	2.88	5.05	634.0	3.31	3.76
558.5	2.59	4.53	627.0	2.93	3.37
1235.8	3.12	Not Available	1274.0	3.12	Not Available
116.0	2.71	3.48	134.9	3.4	3.23
127.0	2.76	5	138.0	3.2	3.97
130.0	2.42	4.5	150.1	3.08	3.51
445.0	2.64	4.08	483.4	3.05	3.19
447.0	3.04	5.49	495.1	3.4	4.09
433.5	2.87	4.66	492.5	3.21	3.855
656.0	2.82	4.62	705.5	3.22	Not Available
756.0	2.96	5.38	740.2	3.44	Not Available
703.0	2.91	4.5	766.4	3.2	Not Available
676.4	2.98	4.89	752.7	3.3	Not Available
1135.0	2.87	4.54	1207.0	3.26	Not Available
1003.0	2.9	4.65	1062.0	3.39	Not Available
1103.0	2.91	5.47	1090.0	3.5	Not Available
158.0	3.22	5.48	213.3	3.96	4.05
405.0	2.98	4.53	Not Available	Not Available	3.77
810.0	2.99	5.48	Not Available	Not Available	3.68
786.0	2.94	5.55	Not Available	Not Available	3.76
1140.0	2.79	4.86	Not Available	Not Available	4.08
279.1	3.39	5.06	275.1	3.33	3.82
377.0	3.01	Not Available	379.2	3.27	Not Available
620.5	2.93	Not Available	626.3	3.29	Not Available
753.0	3.28	Not Available	766.0	3.31	Not Available
122.0	3	Not Available	131.0	3	Not Available

Cooling - Nominal Cap (kW)	Cooling - EER	Cooling - IPLV	Heating - Nominal Cap (kW)	Heating - COP	Heating - SCOP @ 35° SWT
255.0	3.06	Not Available	255.0	3.06	Not Available
359.0	3.26	Not Available	369.0	3.27	Not Available
342.0	3.08	Not Available	351.0	3.11	Not Available
400.0	3.57	Not Available	384.0	3.66	Not Available
647.0	3.02	Not Available	666.0	3.23	Not Available
636.0	2.97	Not Available	666.0	3.1	Not Available
516.0	3.39	Not Available	471.0	3.18	Not Available
1139.0	3.39	Not Available	1045.0	3.22	Not Available
1139.0	3.47	Not Available	1101.0	3.51	Not Available

15 Appendix: 4 pipe Chillers

15.1 4- Pipe chiller data

The heating and cooling performance data for 66 individual 4 pipe chillers was gathered along with costs for each. As part of the exploration of available seasonal performance, equivalent performance metrics was not available across the full range. The most commonly available seasonal performance data has been compiled in to Table 73.

Table 73: 4 pipe Chiller heating/cooling performance data

Cooling - Nominal Cap (kW)	Cooling - EER	Cooling - IPLV	Heating - Nominal Cap (kW)	Heating - COP	Heating - SCOP @ 35° SWT	TER
206.7	2.98	4.84	209.9	3.14	Not Available	7.67
230.6	3.02	4.91	246	3.08	Not Available	7.48
259.2	3.01	4.89	272.7	3.19	Not Available	7.58
299.6	3.01	4.78	306.2	3.2	Not Available	7.66
332.2	2.86	4.92	340.6	3.15	Not Available	7.60
386.3	3.02	5.05	396.2	3.16	Not Available	7.56
426.2	2.91	5.11	437.6	3.19	Not Available	7.68
490.5	2.96	5.14	504.9	3.16	Not Available	7.58
544.3	2.87	5.23	562.7	3.11	Not Available	7.53
598.2	2.96	5.19	618.7	3.1	Not Available	7.40
638.8	2.9	5.15	660.8	3.15	Not Available	7.58
699.7	2.97	5.06	723.7	3.14	Not Available	7.52
743.3	2.89	5.06	772.6	3.11	Not Available	7.41
810.1	2.96	5.08	829.5	3.12	Not Available	7.57
853.8	2.89	5.06	888.9	3.09	Not Available	7.46
919.4	2.95	5.08	940.2	3.09	Not Available	7.41
963	2.89	5.14	988.2	3.07	Not Available	7.35
41.4	3.16	4.46	44.4	3.31	3.40	7.51
62.1	3.15	4.41	65.9	3.3	3.39	7.61
104	3.19	4.38	105	3.33	4.41	7.90
193	3.21	4.79	198	3.38	3.71	7.46
272	3.25	4.75	286	3.35	3.72	7.58
403	3.2	4.84	400	3.33	3.76	7.54
643	2.96	4.76	663	3.27	3.63	7.21
856	2.96	4.83	876	3.2	3.75	6.94
42.8	3.06	5.52	46.2	3.32	3.39	7.41
64.1	3.34	4.62	67.9	3.4	3.45	7.76
109	3.21	4.20	111	3.46	3.42	7.83
187	3.16	5.48	193	3.32	3.74	7.57
264	3.29	5.30	270	3.32	3.72	7.62
372	3.13	5.30	384	3.34	3.77	7.69
599	3.01	5.07	637	3.3	3.65	7.23

Cooling - Nominal Cap (kW)	Cooling - EER	Cooling - IPLV	Heating - Nominal Cap (kW)	Heating - COP	Heating - SCOP @ 35° SWT	TER
805	3.01	5.30	839	3.33	4.15	7.32
406	2.99	5.26	443	3.49	3.94	7.40
502	2.84	4.77	562	3.43	3.90	7.08
602	2.84	4.30	665	3.43	3.88	7.08
798	2.93	4.94	849	3.38	3.88	7.43
1020	3.04	4.70	1100	3.59	3.74	7.65
438.4	3.15	4.48	439.7	3.41	Not Available	Not Available
653.7	3.19	4.90	654.7	3.38	Not Available	Not Available
279.8	3.39	Not Available	290.3	3.57	Not Available	Not Available
423.4	2.98	Not Available	437.7	3.51	Not Available	Not Available
632	2.97	Not Available	658.3	3.5	Not Available	Not Available
278	3.43	Not Available	288.7	3.64	Not Available	Not Available
426	3.02	Not Available	440.6	3.5	Not Available	Not Available
635.9	3.01	Not Available	662.6	3.49	Not Available	Not Available
130	2.87	Not Available	132	3.3	Not Available	Not Available
126	2.55	Not Available	138	2.89	2.96	6.71
122	2.85	Not Available	132	3.04	3.20	6.95
189	2.98	Not Available	196	3.36	3.59	8.31
318	2.77	Not Available	341	3.39	3.62	8.03
191	3.02	Not Available	197	3.34	3.52	8.29
321	2.82	Not Available	344	3.38	3.56	8.02
106.2	3.47	4.7	109.5	3.55	Not Available	8.27
105.2	2.85	4.74	112.8	3.27	Not Available	7.21
101	3.156	5.11	106.9	3.32	Not Available	7.35
345.3	3.01	4.91	374.4	3.2	Not Available	7.36
310.8	2.57	4.33	340.5	2.85	3.25	6.76
298.3	2.7	4.53	320	2.98	Not Available	6.94
701.4	3.15	4.54	704.2	3.3	Not Available	7.90
712.2	3.09	Not Available	671.3	3.3	Not Available	Not Available
567.4	2.89	4.52	606.2	3.06	Not Available	7.38
762	3.06	4.92	817.1	3.17	Not Available	7.47
1048	3.05	5.18	969.1	3.15	Not Available	7.70
1039	2.8	5.13	997.7	3.35	Not Available	7.77
1125	2.73	5.08	1060	3.1	Not Available	7.38

16 Appendix: Dewpoint cooler and indirect evaporative cooling

16.1 Equipment Tables

Dewpoint coolers data was gathered from one supplier across three models.

Due to the customised nature of indirect evaporative coolers, the individual components (air-handling heat exchanger unit and evaporative pads) were all sourced from different suppliers. The heat exchanger performance data was gathered from two suppliers and the cost data was gathered for three suppliers. The evaporative pad performance and cost data were gathered from two suppliers. To achieve the indirect evaporative cooler functionality this equipment was costed by a Mechanical Contractor (having National presence) using concept design scenarios.

Table 74: Dewpoint Cooler Models and Nominal Performance Data

Model	Nominal Airflow (L/s)	Climate Zone	Example Ambient/Inlet Conditions (°CDB/°CWB)	Leaving Air Temperature	Total Input Power (kW)
Model 1	1,100	1	35.4 / 28.2	27.5	1.8
Model 1	1,100	2	30.7 / 25.8	25.3	1.8
Model 1	1,100	3	41.8 / 24.2	22.6	1.8
Model 1	1,100	4	39.0 / 22.5	20.9	1.8
Model 1	1,100	5	35.8 / 22.0	20.6	1.8
Model 1	1,100	6	35.5 / 21.5	20.1	1.8
Model 1	1,100	7	36.2 / 21.7	20.5	1.8
Model 1	1,100	8	21.7 / 15.0	15.5	1.8
Model 2	4,700	1	35.4 / 28.2	28	3.75
Model 2	(reduced output)	2	30.7 / 25.8	25.8	3.75
Model 2	4,700	3	41.8 / 24.2	23.1	3.75
Model 2	(reduced output)	4	39.0 / 22.5	22.4	3.75
Model 2	4,700	5	35.8 / 22.0	21.1	3.75
Model 2	(reduced output)	6	35.5 / 21.5	20.6	3.75
Model 2	4,700	7	36.2 / 21.7	21	3.75
Model 2	(reduced output)	8	21.7 / 15.0	16	3.75
Model 2	7,400	1	35.4 / 28.2	28	12.5
Model 2	7,400	2	30.7 / 25.8	25.8	12.5
Model 2	7,400	3	41.8 / 24.2	23.1	12.5

Model	Nominal Airflow (L/s)	Climate Zone	Example Ambient/Inlet Conditions (°CDB/°CWB)	Leaving Air Temperature	Total Input Power (kW)
Model 2	7,400	4	39.0 / 22.5	22.4	12.5
Model 2	7,400	5	35.8 / 22.0	21.1	12.5
Model 2	7,400	6	35.5 / 21.5	20.6	12.5
Model 2	7,400	7	36.2 / 21.7	21	12.5
Model 2	7,400	8	21.7 / 15.0	16	12.5
Model 3	8,500	1	35.4 / 28.2	28	14.2
Model 3	8,500	2	30.7 / 25.8	25.8	14.2
Model 3	8,500	3	41.8 / 24.2	23.1	14.2
Model 3	8,500	4	39.0 / 22.5	22.4	14.2
Model 3	8,500	5	35.8 / 22.0	21.1	14.2
Model 3	8,500	6	35.5 / 21.5	20.6	14.2
Model 3	8,500	7	36.2 / 21.7	21	14.2
Model 3	8,500	8	21.7 / 15.0	16	14.2

16.2 Cost Data

Table 75: Energy recovery unit average equipment costs with itemised installation costs - 500 L/s

Equipment	Unit Average Cost	Unit Install	Total
HRV: Nominal Airflow - 500L/s			
Install Breakup			
Ductwork supply (OA system)		$ -	
Ductwork supply (R/A system)		$285.00	
Ductwork supply (E/A system)		$855.00	
Ductwork supply (S/A system)		$1,825.00	
Ductwork installation		$1,568.00	
Filters & Frames		$850.00	
Unit installation		$1,868.00	
Cranage		$ -	
Controls		$4,295.00	
Electrical		$1,850.00	
Other (witnessing/commissioning etc.)		$2,563.00	
TOTALS	**$13,940**	**$15,959**	**$29,899.00**

Table 76: Evaporative pad average equipment costs with itemised installation costs - 500 L/s

Equipment	Unit Average Cost	Unit Install	Total
Evaporative Pad: Nominal Airflow - 500L/s			
Install Breakup			
Ductwork supply (OA system: N/A)		$ -	
Ductwork supply (R/A system)		$370.00	
Ductwork supply (E/A system included above)		$ -	
Ductwork supply (S/A system: N/A)		$ -	
Housing box		$2,000.00	
Unit installation		$1,520.00	
Cranage		$ -	
Controls		$850.00	
Electrical		$850.00	
Water supply (evaporative pads)		$1,650.00	
Other (witnessing/commissioning)		$1,281.00	
TOTALS	**$7,050.00**	**$8,521.00**	**$15,571.00**

Table 77: Energy recovery unit average equipment costs with itemised installation costs - 2,000 L/s

Equipment	Unit Average Cost	Unit Install	Total
HRV: Nominal Airflow - 2,000L/s			
Install Breakup			
Ductwork supply (OA system)		$ -	
Ductwork supply (R/A system)		$570.00	
Ductwork supply (E/A system)		$1,995.00	
Ductwork supply (S/A system)		$4,380.00	
Ductwork installation		$3,136.00	
Filters & Frames		$1,809.00	
Unit installation		$1,868.00	
Cranage		$ -	
Controls		$4,295.00	
Electrical		$1,950.00	
Other (witnessing/commissioning)		$2,563.00	
TOTALS	**$29,839.00**	**$22,566.00**	**$52,405.00**

Table 78: Evaporative pad average equipment costs with itemised installation costs - 2,000 L/s

Equipment	Unit Average Cost	Unit Install	Total
Evaporative Pad: Nominal Airflow - 2,000L/s			
Install Breakup			
Ductwork supply (OA system: N/A)		$ -	
Ductwork supply (R/A system)		$570.00	
Ductwork supply (E/A system included above)		$ -	
Ductwork supply (S/A system: N/A)		$ -	
Housing box		$3,500.00	
Unit installation		$3,136.00	
Cranage		$ -	
Controls		$2,000.00	
Electrical		$2,850.00	
Water supply (evaporative pads)		$1,650.00	
Other (witnessing/commissioning)		$2,563.00	
TOTALS	**$11,915.00**	**$16,269.00**	**$28,184.00**

Table 79: Energy recovery unit average equipment costs with itemised installation costs - 4,000 L/s

Equipment	Unit Average Cost	Unit Install	Total
HRV: Nominal Airflow - 4,000L/s			
Install Breakup			
Ductwork supply (OA system)		$ -	
Ductwork supply (R/A system)		$855.00	
Ductwork supply (E/A system)		$2,850.00	
Ductwork supply (S/A system)		$6,205.00	
Ductwork installation		$3,136.00	
Filters & Frames		$2,072.00	
Unit installation		$1,868.00	
Cranage		$ -	
Controls		$4,295.00	
Electrical		$1,950.00	
Other (witnessing/commissioning)		$2,563.00	
TOTALS	**$45,850.00**	**$25,794.00**	**$71,644.00**

Table 80: Evaporative pad average equipment costs with itemised installation costs - 4,000 L/s

Equipment	Unit Average Cost	Unit Install	Total
Evaporative Pad: Nominal Airflow - 4,000L/s			
Install Breakup			
Ductwork supply (OA system: N/A)		$ -	
Ductwork supply (R/A system)		$855.00	
Ductwork supply (E/A system included above)		$ -	
Ductwork supply (S/A system: N/A)		$ -	
Housing box		$4,000.00	
Unit installation		$3,136.00	
Cranage		$ -	
Controls		$2,000.00	
Electrical		$2,850.00	
Water supply (evaporative pads)		$1,650.00	
Other (witnessing/commissioning)		$2,563.00	
TOTALS	**$13,925.00**	**$17,054.00**	**$30,979.00**

16.3 Simulation Results

Table 81: C5OL initial simulation results for indirect evaporative cooling

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	EHC heating energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Gas (MWh)
	CZ3								
DM-No CO2-C5OL-2019-CZ3		**Base, no CO2**	42.7044	2.4491	215.873	90.9477	54.3932	75.1414	42.7044
DM-C5OL-2019-CZ3		**Base, CO2**	37.3335	2.1939	211.232	90.2086	53.7936	74.3207	37.3335
HX Only-No CO2-Constant Fan-C5OL-CZ3		**HX 2ΔT**	38.4836	2.3199	208.723	103.929	53.3595	73.5476	38.4836
IE-No CO2-Constant Fan-C5OL-CZ3		**IE 2ΔT**	38.4146	2.3155	195.965	113.563	50.6607	69.578	38.4146
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ3		**HX 8ΔT**	38.3758	2.3208	210.956	97.5016	53.9204	74.2496	38.3758
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ3		**IE 8ΔT**	38.466	2.3211	198.378	109.033	51.3066	70.4427	38.466
	CZ5								
DM-No CO2-C5OL-2019-CZ5		**Base, no CO2**	19.7565	1.1634	160.017	59.916	37.8949	52.2863	19.7565
DM-C5OL-2019-CZ5		**Base, CO2**	17.5914	1.0799	155.439	59.4802	37.4933	51.7463	17.5914
HX Only-No CO2-Constant Fan-C5OL-CZ5		**HX 2ΔT**	18.2037	1.1664	158.64	64.006	37.8474	52.1407	18.2037
IE-No CO2-Constant Fan-C5OL-CZ5		**IE 2ΔT**	18.2111	1.1663	150.285	78.9993	36.6975	50.0869	18.2111
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ5		**HX 8ΔT**	18.1973	1.1659	159.562	60.9724	37.903	52.2597	18.1973
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ5		**IE 8ΔT**	18.2049	1.166	155.049	65.4134	37.6577	51.6614	18.2049
	CZ7								
DM-No CO2-C5OL-2019-CZ7		**Base, no CO2**	209.687	0	68.1898	57.8267	23.8968	22.9268	209.687
DM-C5OL-2019-CZ7		**Base, CO2**	182.2	0	67.3054	55.7581	23.4936	22.7567	182.2
HX Only-No CO2-Constant Fan-C5OL-CZ7		**HX 2ΔT**	185.74	0	67.2015	64.7846	23.7345	22.7281	185.74

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	EHC heating energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Gas (MWh)
IE-No CO2-Constant Fan-C5OL-CZ7		**IE 2ΔT**	185.846	0	62.4207	73.8042	22.813	21.3059	185.846
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ7		**HX 8ΔT**	185.676	0	68.089	62.2202	23.9195	22.9461	185.676
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ7		**IE 8ΔT**	185.716	0	65.1418	66.5211	23.3968	22.2246	185.716
	CZ8								
DM-No CO2-C5OL-2019-CZ8		**Base, no CO2**	379.154	0	22.737	86.0281	11.6213	5.6179	379.154
DM-C5OL-2019-CZ8		**Base, CO2**	330.343	0	23.0772	78.9333	11.3119	5.6758	330.343
HX Only-No CO2-Constant Fan-C5OL-CZ8		**HX 2ΔT**	326.288	0	22.8609	87.2098	11.5271	5.6476	326.288
IE-No CO2-Constant Fan-C5OL-CZ8		**IE 2ΔT**	326.475	0	20.4818	93.0115	11.1949	5.258	326.475
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ8		**HX 8ΔT**	326.331	0	22.8589	87.2594	11.5241	5.6468	326.331
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ8		**IE 8ΔT**	326.422	0	22.1734	87.8894	11.4945	5.5895	326.422

Table 82: C5OL expanded simulation results for indirect evaporative cooling

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	EHC heating energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Gas (MWh)
	CZ1								
DM-No CO2-C5OL-2019-CZ1		Base, no CO2	0.0259	0	519.127	123.368	92.8527	139.998	0.0259
DM-C5OL-2019-CZ1		Base, CO2	0.0269	0	477.364	119.195	90.4365	136.257	0.0269
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ1		HX 8ΔT	0.0259	0	513.314	127.35	92.7012	139.57	0.0259
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ1		IE 8ΔT	0.0269	0	483.961	142.902	90.5765	136.229	0.0269
	CZ2								
DM-No CO2-C5OL-2019-CZ2		Base, no CO2	7.2091	0.2695	243.3	72.4167	55.3997	81.7782	7.2091
DM-C5OL-2019-CZ2		Base, CO2	7.2091	0.2695	243.3	72.4167	55.3997	81.7782	7.2091
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ2		HX 8ΔT	6.6862	0.2707	243.263	72.5505	55.4263	81.7667	6.6862
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ2		IE 8ΔT	6.6858	0.2707	232.017	82.5991	54.3154	79.8985	6.6858
	CZ3								
DM-No CO2-C5OL-2019-CZ3		Base, no CO2	42.7044	2.4491	215.873	90.9477	54.3932	75.1414	42.7044
DM-C5OL-2019-CZ3		Base, CO2	37.3335	2.1939	211.232	90.2086	53.7936	74.3207	37.3335
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ3		HX 8ΔT	38.3758	2.3208	210.956	97.5016	53.9204	74.2496	38.3758
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ3		IE 8ΔT	38.466	2.3211	198.378	109.033	51.3066	70.4427	38.466
	CZ4								
DM-No CO2-C5OL-2019-CZ4		Base, no CO2	142.785	6.9925	101.255	65.0322	31.7105	37.3045	142.785
DM-C5OL-2019-CZ4		Base, CO2	121.063	6.3928	99.1664	64.0084	31.1999	36.8763	121.063
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ4		HX 8ΔT	126.334	6.7399	100.438	69.5411	31.6949	37.2317	126.334
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ4		IE 8ΔT	126.414	6.7386	95.0798	76.7119	30.4079	35.5083	126.414

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	EHC heating energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Gas (MWh)
	CZ5								
DM-No CO2-C5OL-2019-CZ5		Base, no CO2	19.7565	1.1634	160.017	59.916	37.8949	52.2863	19.7565
DM-C5OL-2019-CZ5		Base, CO2	17.5914	1.0799	155.439	59.4802	37.4933	51.7463	17.5914
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ5		HX 8ΔT	18.1973	1.1659	159.562	60.9724	37.903	52.2597	18.1973
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ5		IE 8ΔT	18.2049	1.166	155.049	65.4134	37.6577	51.6614	18.2049
	CZ6								
DM-No CO2-C5OL-2019-CZ6		Base, no CO2	85.291	0	70.3319	49.3991	20.7741	23.2044	85.291
DM-C5OL-2019-CZ6		Base, CO2	70.8179	0	69.4773	48.8101	20.5275	23.1411	70.8179
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ6		HX 8ΔT	76.7357	0	69.7911	52.5408	20.7971	23.183	76.7357
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ6		IE 8ΔT	76.7688	0	67.6521	55.1711	20.5333	22.7709	76.7688
	CZ7								
DM-No CO2-C5OL-2019-CZ7		Base, no CO2	209.687	0	68.1898	57.8267	23.8968	22.9268	209.687
DM-C5OL-2019-CZ7		Base, CO2	182.2	0	67.3054	55.7581	23.4936	22.7567	182.2
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ7		HX 8ΔT	185.676	0	68.089	62.2202	23.9195	22.9461	185.676
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ7		IE 8ΔT	185.716	0	65.1418	66.5211	23.3968	22.2246	185.716
	CZ8								
DM-No CO2-C5OL-2019-CZ8		Base, no CO2	379.154	0	22.737	86.0281	11.6213	5.6179	379.154
DM-C5OL-2019-CZ8		Base, CO2	330.343	0	23.0772	78.9333	11.3119	5.6758	330.343
HX Only-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ8		HX 8ΔT	326.331	0	22.8589	87.2594	11.5241	5.6468	326.331
IE-Latent-8 dT-No CO2-Constant Fan-C5OL-CZ8		IE 8ΔT	326.422	0	22.1734	87.8894	11.4945	5.5895	326.422

Table 83: C9A indirect evaporative cooling results

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Annual electricity use (MWh)	Gas (MWh)
	CZ1								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ1		HX 8ΔT	0	493.4288	113.0215	101.1342	138.9193	846.5038	0
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ1		IE 8ΔT	0	464.8428	127.1459	98.6147	135.2975	825.9009	0
	CZ2								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ2		HX 8ΔT	12.0217	187.5573	106.4324	46.1258	58.7812	410.9184	12.0217
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ2		IE 8ΔT	12.1404	179.7856	112.0398	44.9504	57.2408	406.157	12.1404
	CZ3								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ3		HX 8ΔT	41.7999	159.0588	123.7269	42.7107	53.2544	420.5507	41.7999
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ3		IE 8ΔT	42.5801	135.5838	137.0973	38.8399	46.5222	400.6233	42.5801
	CZ4								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ4		HX 8ΔT	122.4481	72.5189	124.8614	24.9051	24.6134	369.3469	122.4481
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ4		IE 8ΔT	122.7673	63.4812	132.1794	23.0923	22.0061	363.5263	122.7673
	CZ5								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ5		HX 8ΔT	27.6308	110.0211	113.6623	29.5547	34.7397	315.6086	27.6308
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ5		IE 8ΔT	27.6807	107.1628	116.2157	29.0903	34.0851	314.2346	27.6807
	CZ6								
HX Only-Latent-8 dT-No CO2-		HX 8ΔT	66.376	43.9341	114.3998	15.8398	14.4522	255.0019	66.376

Simulation Lookup	Climate Zone	Case	Boilers energy (MWh)	Chillers energy (MWh)	Distr fans energy (MWh)	Distr pumps energy (MWh)	Heat rej fans/pumps energy (MWh)	Annual electricity use (MWh)	Gas (MWh)
Constant Fan-C9A-CZ6									
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ6		**IE 8ΔT**	66.3864	40.7687	117.4384	15.3297	13.6996	253.6228	66.3864
	CZ7								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ7		**HX 8ΔT**	183.2644	41.4303	131.1603	18.4248	13.552	387.8318	183.2644
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ7		**IE 8ΔT**	183.6225	37.1064	135.4882	17.5118	12.3841	386.113	183.6225
	CZ8								
HX Only-Latent-8 dT-No CO2-Constant Fan-C9A-CZ8		**HX 8ΔT**	338.7415	7.523	139.9184	12.632	2.4915	501.3064	338.7415
IE-Latent-8 dT-No CO2-Constant Fan-C9A-CZ8		**IE 8ΔT**	339.1124	7.2088	142.3346	12.5827	2.4179	503.6564	339.1124

17 Appendix: VSD applications - pumps and fans

17.1 Cost data

The costing for VSDs was based on prices from two manufacturers across the full range of sizes assessed. The final pricing curve was derived from the average $/kW from both manufacturers.

Table 84. Price information of VSDs based on motor size from ABB and Danfoss

Motor size	ABB Prices	Danfoss Prices
0.75	$539.00	$547.04
1.50	$664.00	$675.22
3.00	$748.00	$902.72
7.50	$1,209.00	$1,236.82
15.00	$1,819.00	$1,764.36
22.00	$2,406.00	$2,592.46
30.00	$2,759.00	$2,976.48
37.00	$3,357.00	$3,582.80
45.00	$4,219.00	$4,322.76
55.00	$4,769.00	$5,236.40

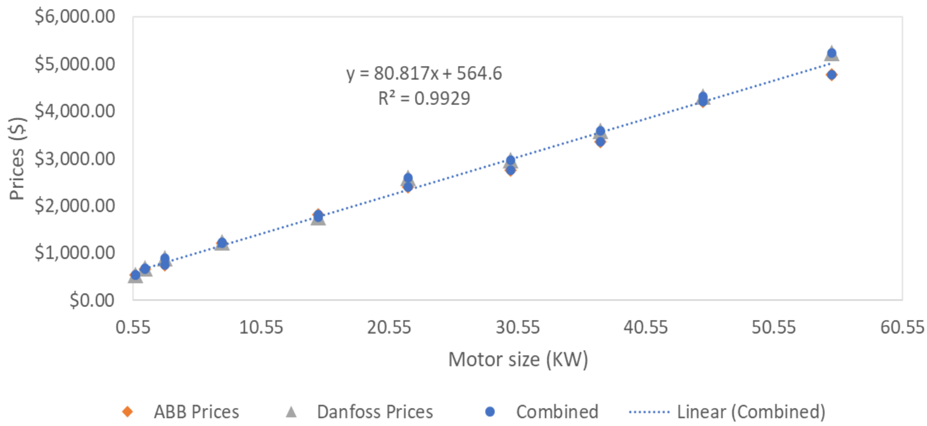


Figure 264. Cost curve for VSD pricing

17.2 Cooling tower simulation results

Cooling tower operation was simulated using a C5OL model, with operation and performance as described in

Table 85:

Table 85. Cooling Tower simulation results based on C5OL model

Speed Type	Configuration	Climate Zone	Cooling Tower Fan Operation Hours	Condenser Water Pump Operation Hours	Annual CT Fan Consumption (kWh)	Annual CDWP Consumption (kWh)
One Speed	Interlocked	1	2860	2860	87313	43939
One Speed	Parallel	1	2860	2860	92296	43961
VSD	Interlocked	1	2860	2860	72502	43964
VSD	Parallel	1	2860	2860	64912	43969
One Speed	Interlocked	2	2726	2766	50987	26276
One Speed	Parallel	2	2689	2766	55556	26222
VSD	Interlocked	2	2719	2766	41353	26265
VSD	Parallel	2	2685	2766	30866	26217
One Speed	Interlocked	3	2443	2578	45627	24572
One Speed	Parallel	3	2333	2578	49767	24554
VSD	Interlocked	3	2432	2578	35726	24571
VSD	Parallel	3	2325	2578	28187	24533
One Speed	Interlocked	4	1737	2214	22309	12665
One Speed	Parallel	4	1613	2214	24378	12639
VSD	Interlocked	4	1719	2214	15587	12661
VSD	Parallel	4	1596	2214	10319	12637
One Speed	Interlocked	5	2415	2703	31873	17388
One Speed	Parallel	5	2327	2703	34372	17374
VSD	Interlocked	5	2393	2703	23360	17390
VSD	Parallel	5	2318	2703	17304	17370
One Speed	Interlocked	6	1667	2493	13596	8243
One Speed	Parallel	6	1512	2493	14925	8216
VSD	Interlocked	6	1636	2493	9258	8243
VSD	Parallel	6	1484	2493	5418	8215
One Speed	Interlocked	7	1484	2032	13376	7992
One Speed	Parallel	7	1380	2032	14785	7971
VSD	Interlocked	7	1463	2032	8772	7992
VSD	Parallel	7	1361	2032	4990	7971
One Speed	Interlocked	8	1021	1593	3608	1732
One Speed	Parallel	8	863	1593	3739	1937
VSD	Interlocked	8	972	1593	2033	1732
VSD	Parallel	8	841	1593	1751	1937

18 Appendix: Economy Cycle

18.1 Costing build-up

18.1.1 Overview

The costing for built-up economy cycle units was based on specific designs for economy cycles in the situations described in Table 86

Table 86. Economy cycle cases used to assess costs.

Archetype	Operation profile	HVAC type	Supply Air Size
School	Business hours	FCU	500 L/s
Small Hotel	24/7	FCU	500 L/s
Medium Office	Business hours	FCU	1,000 L/s
Aged Care	24/7	FCU	1,000 L/S
Small retail	Business hours	AHU	2,000 L/s
Hospital	24/7	AHU	2,000 L/s
Large Retail	Business hours	AHU	5,000 L/s
Large Office	Business hours	AHU	5,000 L/s

The general allowances in the build-ups are as follows:

1. For small FCU systems (500L/s, 1000 L/s):
    a. Allow for 3m rigid ducts and a radius bend for each of Relief Air, Outside Air and Return Air ductwork.
    b. Include labour costs for installation of additional ductwork and controller
2. For larger AHU systems (2000 L/s, 5000L/s):
    a. Differences in design are only within plant room.
    b. It can be assumed that the plant room is designed to be the outside air plenum and is sized such that the air handler is adjacent to the walls and requires a minimal length of relief air ductwork of 3m.
    c. Ductwork for the outside and relief air are to be sized based on louvre dimensions given.

18.1.2 Component Lists

Table 87. Economy cycle component list - School 500 l/s

Item	Economy	Without Economy
Outside air louvre	750 x 300	300 x 250
Outside air damper	500 x 250	Not present
Outside air ductwork	500 x 250	250 x 250
Non-return damper	Not required	Required
Relief air damper	500 x 250	Not required
Relief air ductwork	500 x 250	250 x 250
Relief air louvre	750 x 300	300 x 250
Return air damper	500 x 250	Not required

Return air ductwork	500 x 250	350 x 250
PLC controller	Required	Not required

Table 88. Economy cycle component list – Small hotel 500 l/s

Item	Economy	Without Economy
Outside air louvre	750 x 300	150 x 150
Outside air damper	500 x 250	Not present
Non-return damper	Not required	150 x 150
Outside air ductwork	500 x 250	150 x 150
Relief air damper	500 x 250	Not required
Relief air ductwork	500 x 250	150 x 150
Relief air louvre	750 x 300	150 x 150
Relief air on/off motorised damper	200 diameter	Not required
Return air damper	500 x 250	Not required
Return air ductwork	500 x 250	400 x 250
Return air plenum	900 x 450 x 1000	350 x 250 x 1000
PLC controller	Required – additional points and engineering	Required

Table 89. Economy cycle component list – Medium Office 1000 l/s

Item	Economy	Without Economy
Outside air louvre	900 x 450	300 x 300
Outside air damper	900 x 300	Not required
Non-return damper	Not required	300 x 150
Outside air ductwork	900 x 300	300 x 150
Relief air egg crate grill	1200 x 600	Not required
Relief air damper	900 x 300	Not required
Relief air ductwork	900 x 300	Not required
Relief air louvre	1200 x 300	Not required
Relief air on/off motorised damper	250 x 250	Not required
Return air damper	600 x 300	Not required
PLC controller	Required – additional points and engineering	Required

Table 90. Economy cycle component list – Aged Care 1000 l/s

Item	Economy	Without Economy
Outside air louvre	900 x 450	450 x 300
Outside air damper	900 x 300	Not required
Non-return damper	Not required	300 x 250
Outside air ductwork	900 x 300	300 x 250
Relief air egg crate grill	1200 x 600	600 x 300
Relief air damper	900 x 300	Not required
Relief air ductwork	900 x 300	300 x 250
Relief air louvre	1200 x 300	450 x 300

Return air damper	600 x 300	Not required
PLC controller	Required – additional points and engineering	Required

Table 91. Economy cycle component list – Hospital and small retail 2000 l/s. Although different designs were assessed for these cases, the component list was identical.

Item	Economy	Without Economy
Outside air louvre	1200x1000	500x400
Outside air ductwork	As required	As required
Economy cycle damper	900x900	Not present
Return air damper	600x600	Not present
Relief air damper	600x600	Not present
Relief air louvre	1200x750	Not present
Relief air ductwork	As required	Not present

Table 92. Economy cycle component list – Large Office and large retail 5000 l/s. Although different designs were assessed for these cases, the component list was identical.

Item	Economy	Without Economy
Outside air louvre	1750 x 1750	750 x 750
Outside air ductwork	As required	As required
Economy cycle damper	1200 x 1200	Not present
Return air damper	1100 x 1000	Not present
Relief air damper	1100 x 1000	Not present
Relief air louvre	1500 x 1500	Not present
Relief air ductwork	As required	Not present

18.2 Pricing

The compiled pricing for each case is listed in Table 93.

Table 93. Compiled pricing for economy cycle cases.

Description	Cost With Economy Cycle	Cost Without Economy Cycle	Price Difference (Ex GST)
School (500 L/s)	$23,940	$20,190	$3,750
Small Hotel (500 L/s)	$23,940	$20,190	$3,750
Office (1000 L/s)	$28,120	$23,290	$4,830
Aged Care (1000 L/s)	$28,120	$23,090	$5,030
Hospital (2,000 L/s)	$95,630	$90,280	$5,355
Small retail (2,000 L/s)	$95,630	$90,280	$5,355
Large Office (5,000 L/s)	$111,090	$101,770	$9,320
Large Retail (5,000 L/s)	$111,090	$101,770	$9,320

18.3 Simulation Results

18.3.1 AHU Results (C5OL/C9AS)

Table 94. Simulation results - Economy cycle energy use, FCU/AHU systems based on C5OL and C9AS archetypes.

Archetype	Scenario	Zone	Annual electricity use (kWh)	Normalised Electricity use (kWh/ L/s)
C5OL	Eco	CZ1	822,394	17.76
C5OL	Eco	CZ2	450,110	10.43
C5OL	Eco	CZ3	429,253	7.89
C5OL	Eco	CZ4	237,644	4.90
C5OL	Eco	CZ5	305,239	7.65
C5OL	Eco	CZ6	161,956	3.88
C5OL	Eco	CZ7	169,314	3.73
C5OL	Eco	CZ8	118,998	2.43
C5OL	No Eco	CZ1	823,253	17.78
C5OL	No Eco	CZ2	455,690	10.56
C5OL	No Eco	CZ3	443,537	8.15
C5OL	No Eco	CZ4	252,756	5.22
C5OL	No Eco	CZ5	317,778	7.97
C5OL	No Eco	CZ6	186,994	4.49
C5OL	No Eco	CZ7	186,024	4.11
C5OL	No Eco	CZ8	147,499	3.01
C9AS	Eco	CZ1	208,804	38.94
C9AS	Eco	CZ2	72,394	11.60
C9AS	Eco	CZ3	92,938	14.06
C9AS	Eco	CZ4	72,925	8.66
C9AS	Eco	CZ5	46,857	6.42
C9AS	Eco	CZ6	35,713	5.02
C9AS	Eco	CZ7	83,786	8.81
C9AS	Eco	CZ8	129,674	13.72
C9AS	No Eco	CZ1	209,147	39.01
C9AS	No Eco	CZ2	73,752	11.82
C9AS	No Eco	CZ3	95,724	14.48
C9AS	No Eco	CZ4	75,905	9.02
C9AS	No Eco	CZ5	48,772	6.68
C9AS	No Eco	CZ6	38,596	5.43
C9AS	No Eco	CZ7	85,654	9.01
C9AS	No Eco	CZ8	130,037	13.76

18.3.2 PAC Results (C5OM/C9C)

Table 95. Simulation results - Economy cycle energy use, PAC unit systems based on C5OM and C9C archetypes.

Archetype	Scenario	Zone	Annual electricity use (kWh)	Normalised Electricity use (kWh / L/s)
C5OM	Eco	CZ1	394,098	164.48
C5OM	Eco	CZ2	286,816	90.02
C5OM	Eco	CZ3	286,108	80.50
C5OM	Eco	CZ4	231,726	44.96
C5OM	Eco	CZ5	232,603	62.43
C5OM	Eco	CZ6	195,837	32.39
C5OM	Eco	CZ7	213,340	33.86
C5OM	Eco	CZ8	214,597	23.80
C5OM	No Eco	CZ1	394,541	164.65
C5OM	No Eco	CZ2	294,969	91.14
C5OM	No Eco	CZ3	300,159	82.85
C5OM	No Eco	CZ4	241,739	47.22
C5OM	No Eco	CZ5	243,987	65.25
C5OM	No Eco	CZ6	211,995	37.40
C5OM	No Eco	CZ7	224,020	37.20
C5OM	No Eco	CZ8	227,117	29.50
C9C	Eco	CZ1	623,022	28.37
C9C	Eco	CZ2	316,147	10.40
C9C	Eco	CZ3	338,436	15.05
C9C	Eco	CZ4	260,998	12.59
C9C	Eco	CZ5	248,405	7.23
C9C	Eco	CZ6	187,249	6.44
C9C	Eco	CZ7	245,768	14.17
C9C	Eco	CZ8	251,643	19.71
C9C	No Eco	CZ1	630,264	28.50
C9C	No Eco	CZ2	328,270	10.60
C9C	No Eco	CZ3	366,348	15.63
C9C	No Eco	CZ4	283,675	13.07
C9C	No Eco	CZ5	260,219	7.49
C9C	No Eco	CZ6	201,396	6.78
C9C	No Eco	CZ7	254,968	14.41
C9C	No Eco	CZ8	254,696	19.73

18.4 BCR results

18.4.1 FCU/AHU Economy Cycle

Table 96. BCR results for economy cycle on FCU and AHU systems

	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
Day-500 l/s	0.01	0.06	0.13	0.15	0.15	0.29	0.18	0.28
Day-1000 l/s	0.01	0.10	0.20	0.23	0.24	0.46	0.28	0.44
Day-2000 l/s	0.03	0.18	0.36	0.42	0.43	0.82	0.51	0.80
Day-5000 l/s	0.04	0.26	0.52	0.61	0.62	1.18	0.73	1.15

Night-500 l/s	0.03	0.11	0.21	0.17	0.13	0.20	0.10	0.02
Night-1000 l/s	0.05	0.16	0.31	0.26	0.19	0.30	0.14	0.03
Night-2000 l/s	0.09	0.30	0.58	0.49	0.36	0.56	0.27	0.05
Night-5000 l/s	0.13	0.43	0.83	0.70	0.52	0.80	0.39	0.08

18.4.2 PAC Unit Economy Cycle

Table 97. BCR results for economy cycle on PAC unit systems

	CZ1	CZ2	CZ3	CZ4	CZ5	CZ6	CZ7	CZ8
Day-500 l/s	0.01	0.09	0.19	0.23	0.23	0.44	0.27	0.43
Day-1000 l/s	0.02	0.17	0.34	0.40	0.41	0.78	0.48	0.76
Day-2000 l/s	0.04	0.27	0.55	0.65	0.67	1.27	0.78	1.23
Day-5000 l/s	0.06	0.44	0.89	1.05	1.07	2.04	1.25	1.97
Night-500 l/s	0.05	0.16	0.31	0.26	0.19	0.30	0.14	0.03
Night-1000 l/s	0.08	0.28	0.55	0.46	0.34	0.53	0.26	0.05
Night-2000 l/s	0.14	0.46	0.89	0.75	0.56	0.86	0.42	0.08
Night-5000 l/s	0.22	0.74	1.43	1.20	0.89	1.38	0.67	0.13

19 Appendix: Simulation Models

19.1 Large Office C5OL

The Large Office archetype was used in the assessment of the following measures:

- Chiller
- Cooling tower
- Dewpoint cooler
- Indirect evaporative cooling + Series 2
- Central plant heat pump
- HVAC zoning

19.1.1 General layout

The Large Office model represents a 10 storey, 12,250 m² office building with a square footprint as shown in Figure 265. The conditioned area of the building is 11,040m². The building has two levels of underground carpark and 1 rooftop plant room.

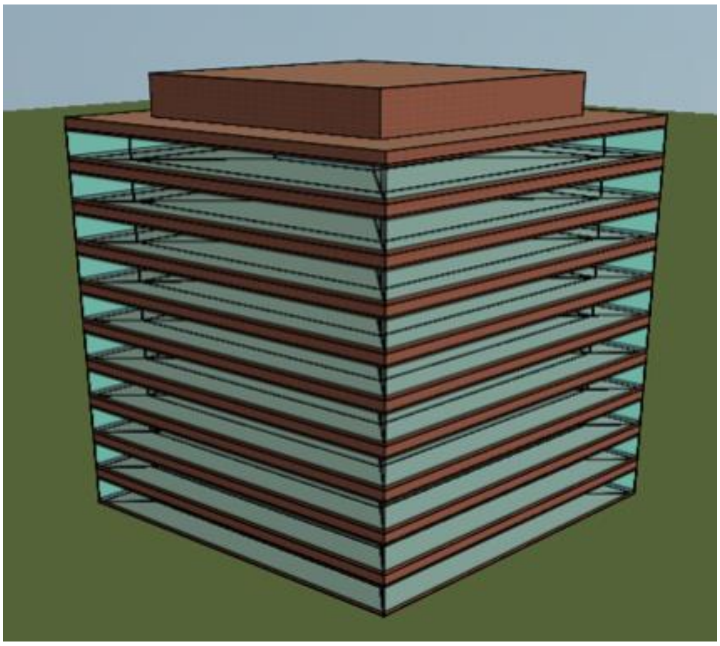


Figure 265. C5OL Modelled Geometry View.

The Floorplate is divided in façade and centre zones as shown in Figure 266.

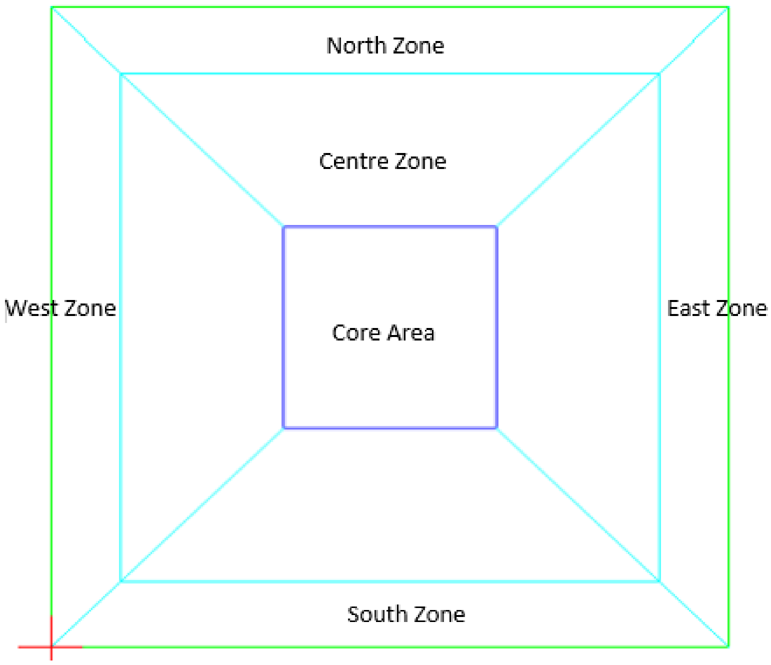


Figure 266. C5OL Modelled Zoning.

19.1.2 HVAC

Major plant

Air-conditioning for this archetype is provided by a water-cooled chiller system with fixed primary pumping and gas heating hot water.

Chilled water plant consists of two identical chillers each selected at 60% of design load. Chiller efficiencies are applied based on the base case efficiencies listed in Table 5.

Heating hot water plant consists of two identical boilers each selected at 60% of design load. Boiler efficiency is set to be 90%.

Air-conditioning is provided by centre, north, south, east and west air-handlers each serving all 10 floors. VAV turndowns are to 30% and 50% of maximum flow in perimeter and centre zones respectively. Fan efficiencies are calculated as per Section J6D5 in NCC 2022.

Air handler cooling and heating coils are sized with an oversizing factor of 1.1.

Control

The VAV zone temperature control was modelled with a 2°C dead band from 21.5°C to 23.5°C with 0.5°C proportional band on either side.

The perimeter zone AHU supply air temperature was controlled by high-select. The centre zone AHU supply air temperature was controlled by the average zone temperature.

The dry bulb economy cycle and the CO_2 control to the minimum outside air was modelled when required as per NCC 2022.

Chilled water temperature and heating hot water temperature was modelled to be reset based on the outside air temperature. The chillers and boilers were staged up and down at 50% of the design load.

19.1.3 Schedules and internal loads

The schedules and internal loads for this archetype are modelled as per Table S35C2c, Table S35C2d, Table S35C2l and Table S35C2n in NCC 2022.

19.2 Medium Office C5OM

The Medium Office archetype was used in the assessment of the following measures:

- PAC
- VRF
- Economy cycle
- Roof

19.2.1 General layout

The Medium Office model represents a 2 storey, 2,304 m²office building with a rectangle footprint as shown in Figure 267. The conditioned area of the building is 2,080m².

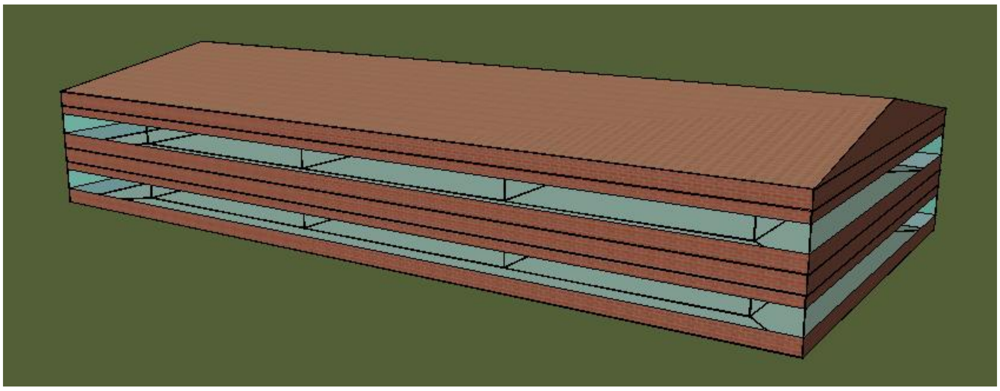


Figure 267. C5OM Modelled Geometry View.

The Floorplate is divided in façade and centre zones as shown in Figure 268.

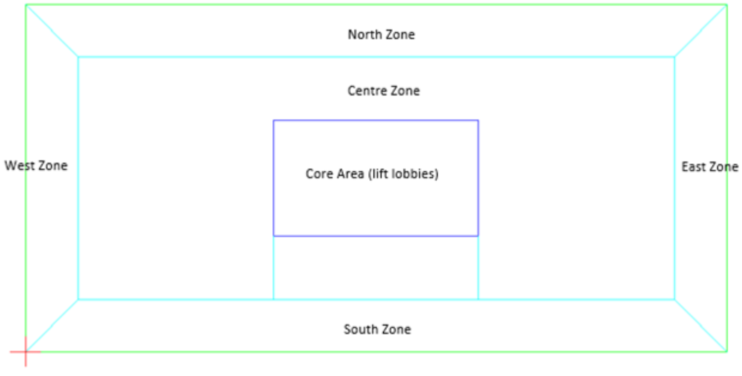


Figure 268. C5OM Modelled Zoning.

19.2.2 HVAC

Major plant

Air-conditioning for this archetype is provided by PAC systems. The supply airflow was modelled as constant flow system. The PAC units are sized with an oversizing factor of 1.2.

Control

The zone temperature control was modelled with a 2°C dead band from 21.5°C to 23.5°C with 0.5°C proportional band on either side.

The dry bulb economy cycle and the CO_2 control to the minimum outside air was modelled when required as per NCC 2022.

19.2.3 Schedules and internal loads

The schedules and internal loads for this archetype are modelled as per Table S35C2c, Table S35C2d, Table S35C2l and Table S35C2n in NCC 2022.

19.3 Large Hospital C9A

The Large Hospital archetype was used in the assessment of the following measures:

- Chiller
- Cooling tower
- Indirect evaporative cooling + Series 2
- Central plant heat pump
- HVAC zoning

19.3.1 General layout

The Large Hospital model represents an 8 storey hospital building. It has been assumed that L3 to L7 are wardroom floors. Only these floors are included in the energy simulation. The total area of these floors is 6,480m² and the conditioned area is 5,710m². The building has two levels of underground carpark and 1 rooftop plant room.

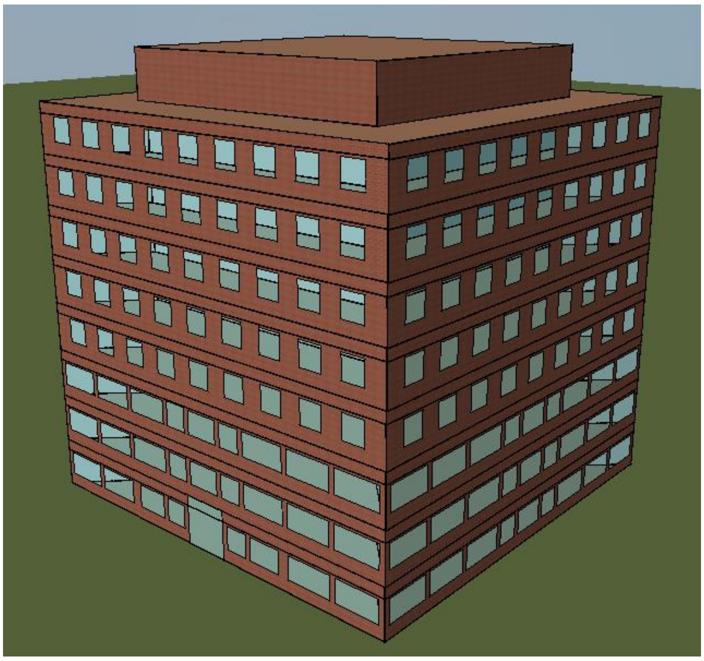


Figure 269. C9AL Modelled Geometry View.

The Floorplate is divided in wardrooms and corridors as shown in Figure 270.

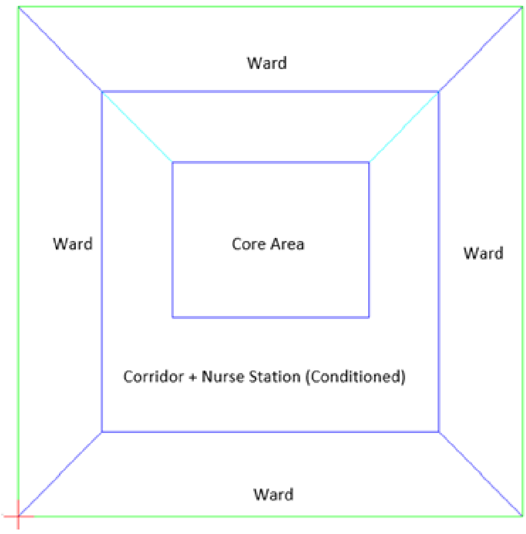


Figure 270. C9A Modelled Zoning.

19.3.2 HVAC

Major plant

Air-conditioning for this archetype is provided by a water-cooled chiller system with fixed primary pumping and gas heating hot water.

Chilled water plant consists of two identical chillers each selected at 60% of design load. Chiller efficiencies are applied based on the base case efficiencies listed in Table 5.

Heating hot water plant consists of two identical boilers each selected at 60% of design load. Boiler efficiency is set to be 90%.

Air-conditioning is provided FCUs serving the wardrooms and one constant flow AHU serving all corridors. Fan efficiencies are calculated as per Section J6D5 in NCC 2022.

FCU and AHU cooling and heating coils are sized with an oversizing factor of 1.1.

Control

The zone temperature control was modelled with a 2°C deadband from 21.5°C to 23.5°C with 0.5°C proportional band on either side.

The corridor zone AHU supply air temperature was controlled by the average zone temperature.

The heat exchanger used to precondition the minimum outside air was modelled when required as per NCC 2022.

Chilled water temperature and heating hot water temperature was modelled to be reset based on the outside air temperature. The chillers and boilers were staged up and down at 50% of the design load.

19.3.3 Schedules and internal loads

The schedules and internal loads for this archetype are modelled as per Table S35C2g, Table S35C2l and Table S35C2n in NCC 2022.

19.4 Aged Care/Small Hospital C9C/C9AS

The Aged Care/Small Hospital archetype was used in the assessment of the following measures:

- PAC (C9C)
- VRF (C9C)
- Economy cycle (C9AS)
- Roof (C9AS)

19.4.1 General layout

The Aged Care/Small Hospital model represents a single storey, 2,048m² building with a donut shape footprint as shown in Figure 271.

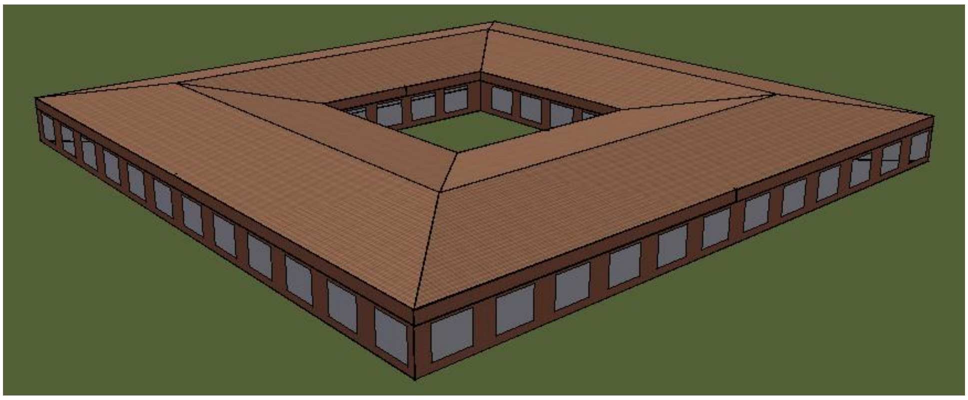


Figure 271. C9A/C9AS Modelled Geometry View.

The Floorplate is divided in bedroom/wardroom and corridor as shown in Figure 272.

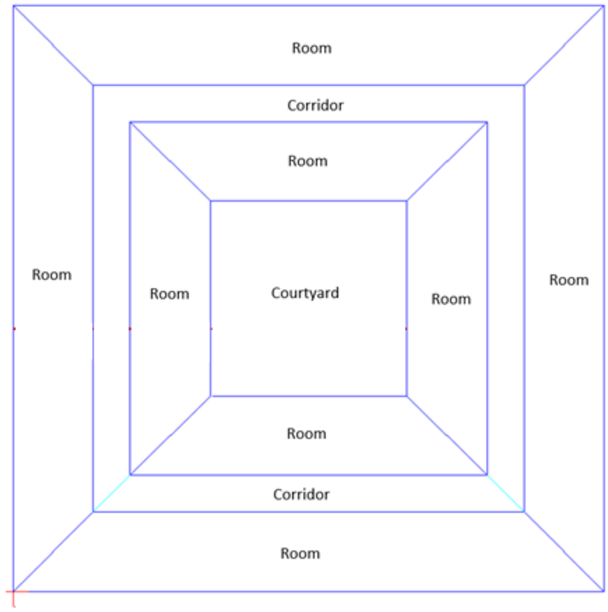


Figure 272. C9C/C9AS Modelled Zoning.

19.4.2 HVAC

Major plant

Air-conditioning for this archetype is provided by PAC systems except when testing VRF measure. The supply airflow was modelled as constant flow system. The PAC units are sized with an oversizing factor of 1.2.

When this archetype was used to test VRF, the air-conditioning system was converted to VRF systems. The supply air was delivered to the zones by constant flow FCUs. The FCU cooling coils and heating coils were served by outdoor VRF unit. The coils and VRF unit are sized with an oversizing factor of 1.1.

Control

The zone temperature control was modelled with a 2°C deadband from 21.5°C to 23.5°C with 0.5°C proportional band on either side.

The heat exchanger used to precondition the minimum outside air was modelled when required as per NCC 2022.

19.4.3 Schedules and internal loads

The schedules and internal loads for this archetype are modelled as per Table S35C2g/ Table S35C2k, Table S35C2l and Table S35C2n in NCC 2022. Note that the only differences between the C9C and C9AS versions of this archetype are in the schedules and internal loads. The change to C9AS (which has continuous HVAC operation) from C9C (which has no HVAC operation from 10am-4pm) arose as it was found that the lack of daytime HVAC operation was both unrealistic for the archetype and distortionary. Time and resources did not permit the rerunning of earlier analyses using the C9AS model; however, the whole building analysis will be conducted using this model and will therefore draw out any effects.