Potential Impact of Evaporative Cooling Technologies on Australian Office Buildings

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ABSTRACT

This paper presents the results of a preliminary simulation-based study of the potential energy-efficiency benefits of a range of evaporative cooling technologies for Australian office buildings across the full range of Australian climates.

It is found that dewpoint coolers offer the most promising savings potential (13–55 per cent) across climate zones 2–7 (i.e., all climate zones other than Darwin and Thredbo). In climate zone 8 (Thredbo) it is shown that a direct/indirect evaporative cooling arrangement can wholly supplant the need for a chiller. Desiccant wheel systems were found to generate electricity savings in climate zones 1–4 but these were counteracted by the significant amount of energy required for desiccant reactivation.

Overall, the results indicate that there is significant potential for the application of evaporative cooling technologies in Australian office buildings outside the tropics.

INTRODUCTION

Evaporative cooling technologies do not currently play a significant role in commercial office building air conditioning, other than in cooling towers. In this paper, simulation modelling is used to test a range of direct and indirect evaporative cooling technologies applied to the airside of a conventional VAV system serving a medium-sized commercial office building.

METHODOLOGY

The office building was modelled in a thermal simulation package, with the following characteristics:

- Eight-storey building with underground carpark.
- 50 per cent window-wall-ratio (WWR). This is a typical WWR for the office building in Australia.
- 25m by 25m floorplate, four perimeter and one centre zone per floor, the total area is 5,000m^2. The square floor plate was used to ensure the building orientation has no impact on the simulation result.
- Floor-to-ceiling height is 2.7m. This is the typical floor-to-ceiling height for offices.
- Plenum height is 0.9m. This is the typical plenum height for office buildings in Australia.
- Building façade and HVAC systems were modelled as compliant with NCC 2019 Section J provisions [ABCB, 1].
- Chiller was modelled as a water-cooled chiller of IPLV8.7, COP 6, using IES default chiller part-load characteristics, with a chilled water temperature rest based on outside air temperature.
- Boiler was modelled as a condensing boiler of nominal 90 per cent efficiency using detailed part load curves and a hot water temperature reset based on outside air temperature.

Full details of the model are available in Zhang et al [2]. The building form is shown in Figure 1.

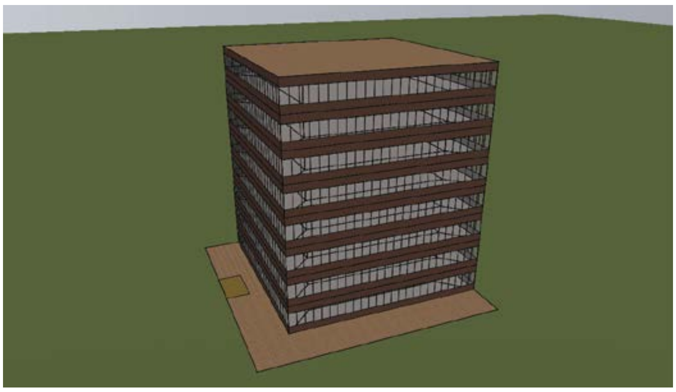

Figure 1. Building form as modelled.

The building simulation was run in each NCC Volume 1 [ABCB, 1] climate zone using IWEC weather data from ASHRAE for the following locations: CZ1 Darwin; CZ2 Brisbane; CZ3 Alice Springs; CZ4 Wagga Wagga; CZ5 Sydney; CZ 6 Melbourne; CZ 7 Canberra; and CZ8 Thredbo.

HVAC CONFIGURATIONS

The HVAC configurations modelled are as follows:

- **SC 1:** A conventional VAV system as shown in Figure 2. This is the baseline against which the evaporative cooling scenarios are tested. Note that by default, the design parameters of the evaporative cooling scenarios are the same as this scenario unless required to change to match the revised configuration.
- **SC 2a:** A VAV system with direct and indirect evaporative cooling as shown in Figure 3.
In this system, the indirect evaporative cooling works by saturating the exhaust air using water sprays or similar and then operating a heat exchanger between this evaporatively cooled airflow and the outside air intake. Direct evaporative cooling is also available to the supply air. The design pressure drop across the heat exchanger was modelled as 100Pa, and 52Pa for the direct evaporative cooler, based on supplier data.
- **SC 2b:** This is a VAV system with a dew-point cooler added to the outside air intake, plus direct evaporative cooling to the supply air as shown in Figure 4. In a dewpoint cooler, notionally two volumes of air are drawn through the cooler's heat exchanger, with one volume being passed into the building and the other volume circulated through the other side of the cooler's heat exchanger while being saturated with water, and then rejected to atmosphere. At the theoretical limit, a device of this nature could generate a supply airstream that has a dry-bulb temperature equal to the dewpoint of the outside air, hence the device's name. The pressure drop through the dew-point cooler was modelled as 150Pa, based on supplier data.
- **SC 2c:** This the same system as SC 2b except with no direct evaporative cooling.
- **SC 3:** This is a VAV system with direct and indirect evaporative cooling and a desiccant wheel, as shown in Figure 5. The desiccant wheel removes moisture from the outside air prior to this entering the indirect evaporative cooler heat exchanger.
The cooled and dried air is then better suited to direct evaporative cooling (being pre-dehumidified) and furthermore, when the direct evaporative cooler is not operating, will have a greater indirect evaporative cooling potential due to the reduced humidity. This, however, comes at a cost: a significant amount of heat is used to reactivate the desiccant wheel in the exhaust air path.
The desiccant wheel was modelled with a pressure drop of 180Pa, the heat recovery wheel at 135Pa (both based on supplier data) and the additional heating coil at 50Pa (based on a two-row coil NCC2019 Table 5.4d).

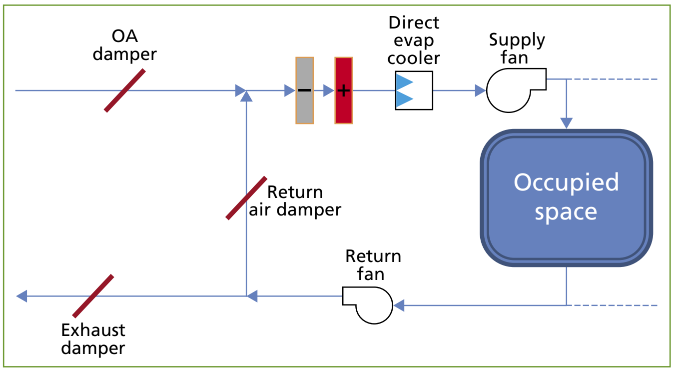

Figure 2. HVAC Configuration SC 1: a conventional VAV system, used as the baseline scenario.

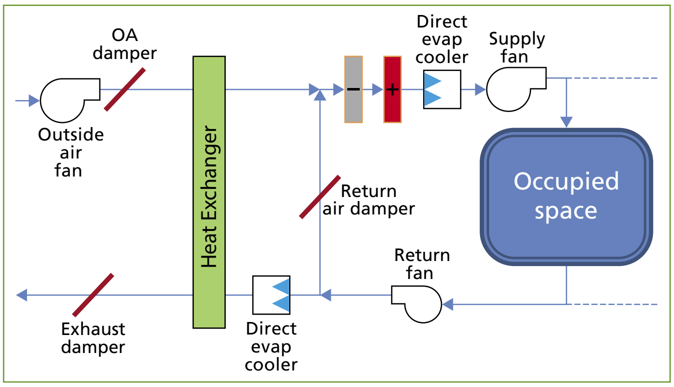

Figure 3. HVAC Configuration SC 2a: VAV system with direct and indirect evaporative cooling.

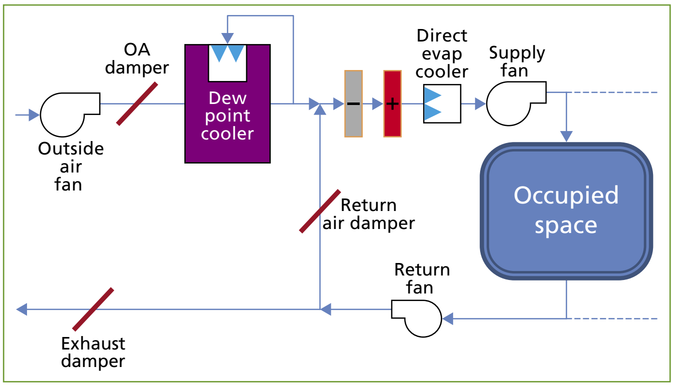

Figure 4. HVAC Configuration SC 2b: VAV system with dew-point cooler and direct evaporative cooling.

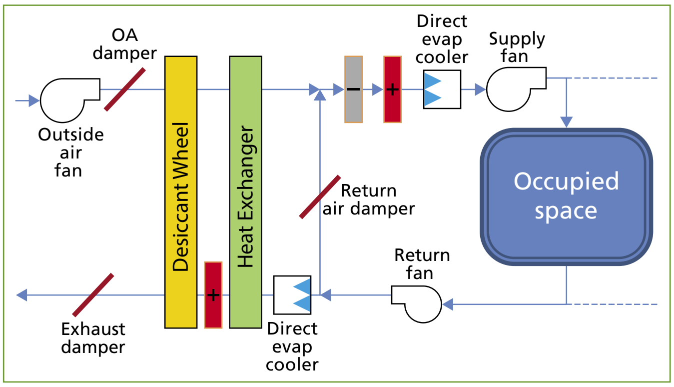

Figure 5. HVAC Configuration SC 3: VAV system with direct/ indirect evaporative cooling and a desiccant wheel.

COMPONENT MODELS

Dew-point cooler

IES does not have a dedicated component for dew-point cooler, so it was necessary to create custom modelling for this. The process of modelling for this component was as follows:

- A data table was obtained from a supplier.
- A multivariable regression equation was derived to calculate the outlet dry bulb temperature based on the inlet dry bulb temperature and moisture content, based on the data table.
- An additional cooling coil controlled to achieve the outlet temperature based on the derived multivariable regression equation; the chiller energy associated with this cooling coil was then excluded from the results.
- The simulated outlet temperature from this arrangement matched the table data well: 98.9 per cent of the data points are within ±5 per cent of the difference.

Desiccant wheel

Similarly, IES does not have a dedicated component model for a desiccant wheel, so a custom model was developed. The process of modelling for this component was as follows:

- A combination of a cooling coil and a heating coil was used to mimic the performance of the desiccant wheel.
- Based on manufacturer's advice, the desiccant wheel has 3g/kg moisture content removal at 55°C reactivation temperature and every g/kg moisture drop equates to a 3.3°C temperature rise of the supply air.
- The cooling coil was used to overcool the incoming air to achieve 3g/kg moisture content removal. The required temperature was calculated based on a regression equation derived from psychrometric formulas.
- The heating coil was used to model the temperature rise after the desiccant wheel. The energy consumed by the cooling and heating coil was excluded from the result.
- Another heating coil was used on the relief air duct to heat the relief air to 55°C, which is the reactivation temperature for the desiccant wheel. The energy consumption of this heating coil was included in the result.

The simulated performance matched the manufacturer's guidance well when the outlet moisture content was greater than 4g/kg. However, below this figure, the simulation underpredicts the moisture removal significantly, which will cause the simulation to underestimate the effectiveness of the wheel under these conditions. This issue is caused by limitations in the use of regression formulae within IES.

HVAC CONTROL

The sequencing of the various components in the test configurations is critical to the energy efficiency of the systems. The baseline control configurations for the HVAC systems are described below. In all cases these were subject to a degree of basic optimisation before being adopted.

SC 1 Standard VAV

The following control was used:

- The zone set-point was set to be 22.5°C with 2°C deadband and 0.5°C proportional band either side. The minimum VAV turndown was set as 50 per cent for centre zones and 30 per cent for perimeter zones.
- The supply air temperature of the AHUs was modelled as follows. The heating supply air temperature was reset from 30°C to 22.5°C for the average zone temperature from 21°C to 21.5°C. The cooling supply air temperature was reset from 22.5°C to 12°C for the average zone temperature from 23°C to 23.5°C. No heating or cooling is provided when the average temperature of the zones is between 21.5°C to 23.5°C.
- The economy cycle was modelled to achieve the target outlet temperature reset from 22.5°C to 12°C for the average zone temperature from 21.5°C to 22°C. The economy cycle is available when the outside air dew point is below 15°C, the outside air-dry bulb temperature is below 24°C and the outside air-dry bulb temperature is less than the return air-dry bulb temperature. This control strategy is to ensure the economy cycle is operating before the chilled water comes into play when the conditions are appropriate.
- An efficient fan curve was used with an x2.7 turndown to 30 per cent flow when no further decrease was assumed. This represents a variable pressure and variable volume fan control. The overall fan efficiency was modelled as 60 per cent
- The minimum outside air was modelled to be modulated between 30 per cent and 100 per cent when the high select zone CO_2 concentration changes from 800ppm to 1,000ppm.

SC 2a Direct and indirect evaporative cooling

The control for this configuration is as follows:

- The direct evaporative cooling is available when the average zone relative humidity (RH) is less than 60 per cent and the post-economy-cycle wet bulb temperature is less than 14°C. The direct evaporative cooler is controlled proportionally to achieve an outlet RH ranging from 0 per cent to 95 per cent as the zone temperature ranges from 22.5°C to 23°C.
- The indirect evaporative cooling was modelled to be operating when the average zone temperature is greater than 22°C. The outside air goes through the dedicated fan and heat exchanger when the outside air temperature is 4°C less than the post-indirect cooler air temperature and average zone temperature is greater than 22°C, or when the outside air temperature is 4°C greater than the return air temperature and the average zone temperature is less than 21.5°C. Otherwise the outside air is bypassed, and the dedicated fan does not run. The heat exchanger efficiency was set to be 70 per cent.
- The economy cycle control is based on the post heat exchanger condition when the heat exchanger is in operation

or otherwise based on the outside air temperature. Other than this, the economy control for SC 2a is the same as that for SC-1.

- The above control strategy gives the sequence of the HVAC component in cooling mode as follows: Outside Air Economy Cycle → Indirect Evaporative Cooling → Direct Evaporative Cooling → Chilled Water Cooling
- Other controls for SC 2a are the same as those for SC 1.

SC 2b Dew-point cooler plus direct evaporative cooling

The controls for this configuration are as follows:

- The direct evaporative cooling is available when the average zone RH is less than 60 per cent and the post dew-point cooler wet bulb temperature is less than 14°C. The direct evaporative cooler is controlled proportionally to achieve an outlet RH ranging from 0 per cent to 95 per cent as the zone temperature ranges from 22.5°C to 23°C.
- The dew-point cooler was modelled to be operating when the average zone temperature is greater than 22°C, the outside air dewpoint is less than 21°C[1] and the outside air-dry bulb temperature is greater than 12°C. The outside air goes through the dedicated fan and the dew-point cooler if the above conditions are satisfied. Otherwise, the outside air is bypassed, and the dedicated fan and dew-point cooler do not run.
- The economy cycle control is based on the post dew-point cooler condition when the dew-point cooler is in operation or otherwise it is based on the outside air temperature. Other than this, the economy control for SC 2b is the same as that for SC 1.
- The above control strategy gives the sequence of the HVAC component in cooling mode as follows:
  Outside Air Economy Cycle → Dew-point cooler →
  Direct Evaporative Cooling → Chilled Water Cooling
- Other controls are the same as those for SC 1.

SC 2c Dew-point cooler only

Controls for this configuration are identical to SC 2b but without a direct evaporative cooler.

SC 3: Direct/indirect evaporative cooling plus desiccant wheel

This configuration is controlled as follows:

- The direct evaporative cooling is available when the average zone RH is less than 60 per cent.
- The indirect evaporative cooling was modelled to be operating when the average zone temperature is greater than 22°C.
- The heat recovery wheel is operating when the desiccant wheel or the indirect evaporative cooler is in operation. The efficiency of the heat recovery wheel was set as 70 per cent.
- The economy cycle control is based on the post heat exchanger condition when the desiccant wheel/indirect evaporative cooling is in operation or otherwise based on the outside air temperature. Other than this, the economy control for SC 3 is the same as that for SC 1.
- The above control strategy gives the sequence of the HVAC component in cooling mode as follows: Outside Air Economy Cycle → Indirect Evaporative Cooling → Direct Evaporative Cooling → Desiccant Wheel → Chilled Water Cooling.
- Other controls are the same as those for SC 1.

INITIAL SIMULATION RESULTS

The basic results of the simulations, showing electricity consumption for all HVAC components (electricity – fans, pumps, chillers; gas – boiler) are presented in Figure 6 to Figure 10. Note that for greenhouse gas emissions calculations, 2020 national average figures of 0.77kg/kWh for electricity and 0.21kg/kWh for gas have been used.

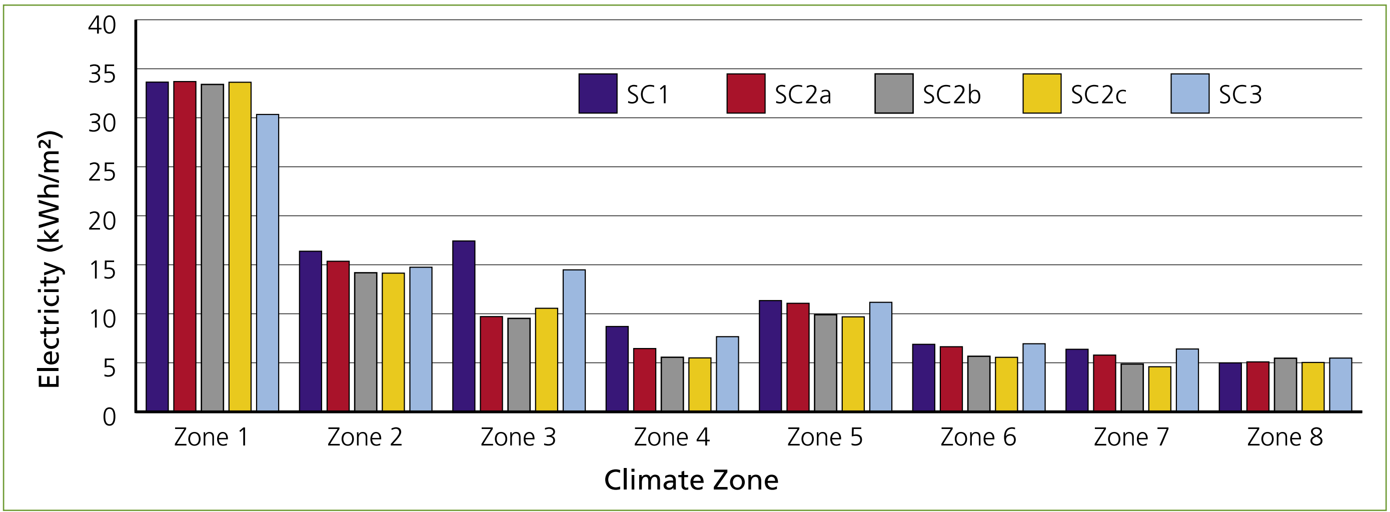

Figure 6. Simulated electricity use.

1 This figure was selected as notionally optimal, within the constraints of the modelling, after some testing of different figures.

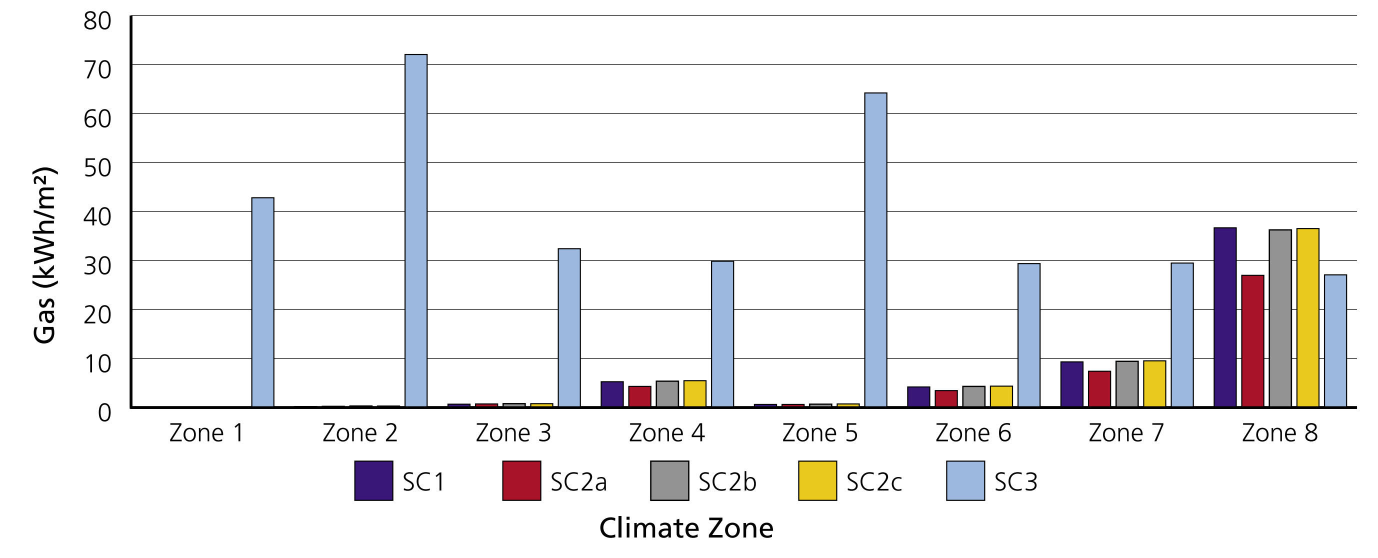

Figure 7. Simulated gas use.

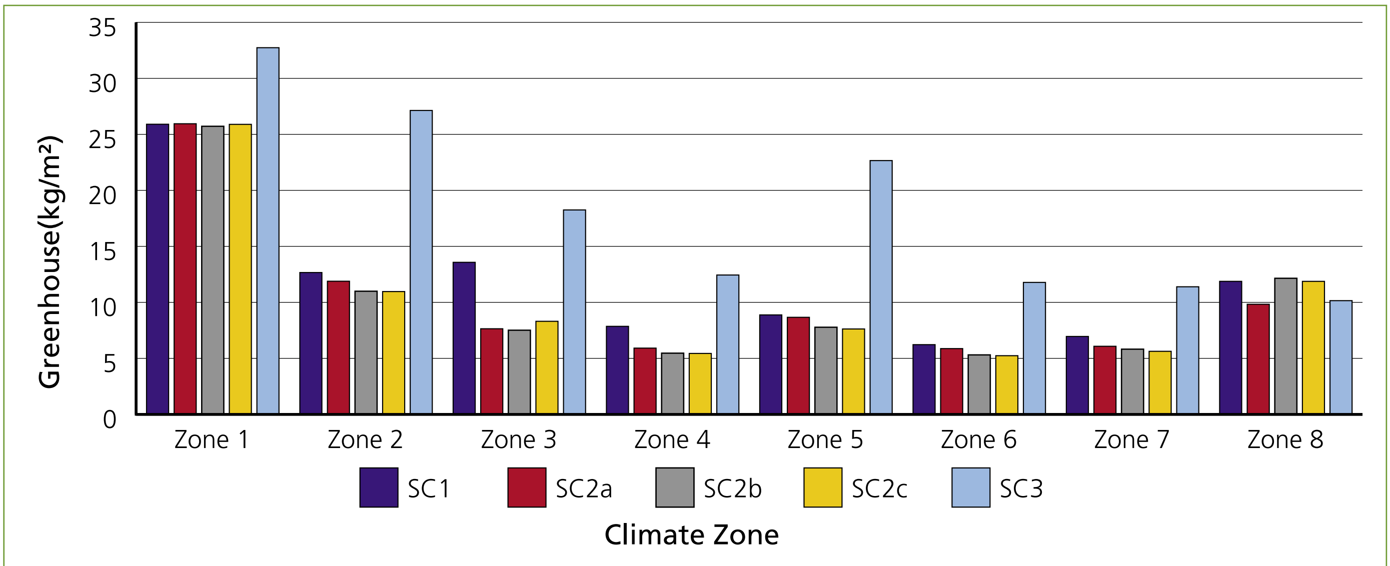

Figure 8. Simulated greenhouse emissions.

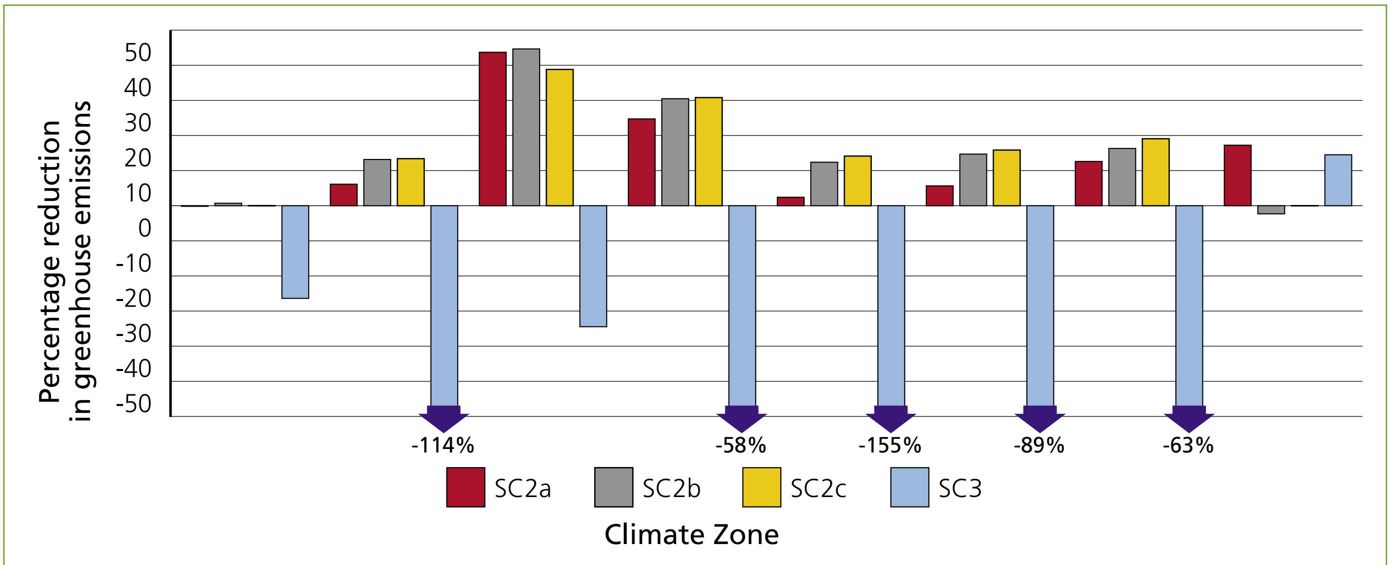

Figure 9. Simulated reduction in greenhouse emissions relative to SC 1.

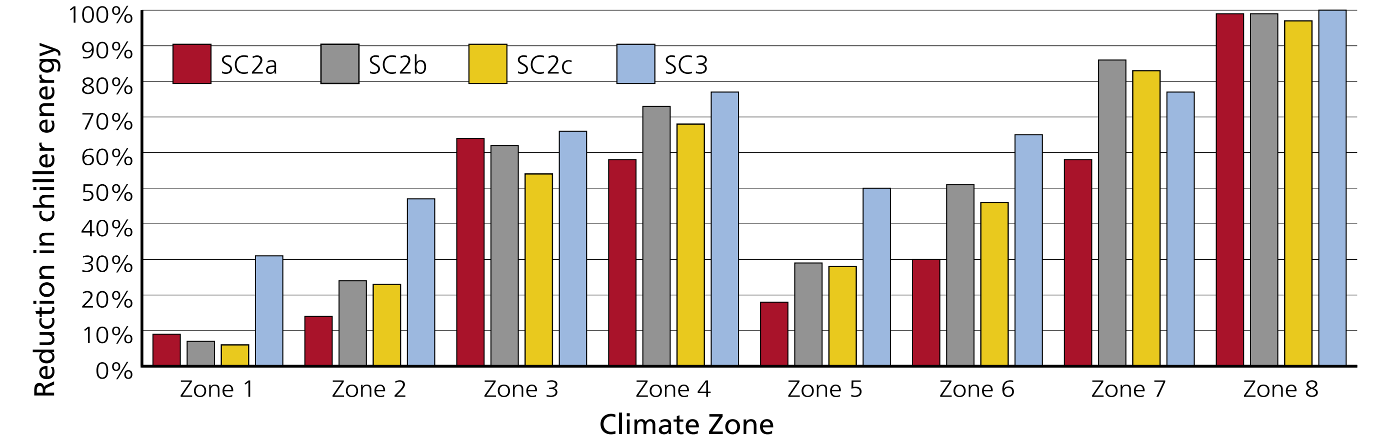

Figure 10. Simulated reduction in chiller energy relative to SC 1.

It can be seen from the figures that:

- None of the technologies is effective in Climate zone 1 (Darwin).
- Configuration SC 2a (direct/indirect evaporative cooling) presents significant benefits in dry climate zones (CZ 3 Alice Springs, CZ4 Wagga Wagga, CZ7 Canberra and CZ8 Thredbo). Total greenhouse savings are in the region of 12–44 per cent, driven by chiller energy use reductions of 54–99 per cent. In CZ8 (Thredbo) SC 2a essentially removed the need for a chiller.
- Configurations SC 2b and SC 2c (dew-point cooler with/without direct evaporative cooling) present significant benefits in all climate zones other than CZ1 Darwin and CZ8 Thredbo. Except in Climate Zone 3 (Alice Springs) the direct evaporative cooling component offers no benefit beyond the dewpoint cooler. SC 2c (Dew-point cooler only) provides greenhouse savings of 13–38 per cent driven by chiller energy use reductions of 23–83 per cent.
- Configuration SC 3 (Direct/indirect evaporative cooling with desiccant wheel offers no benefit except in Climate Zone 8, where a greenhouse emissions reduction of 14.5 per cent is achieved and the evaporative technologies completely replace all cooling, obviating the need for a chiller. In the situation where "free" heat is available for desiccant reactivation, SC 3 achieves a modest electricity saving in Climate zones 1–4; however, other than in Climate Zone 1, this system is still outperformed by other options.

Note that differences between the reductions in chiller energy and the total change in electricity consumption are caused primarily by increases in fan energy caused by higher air flows (driven by higher supply air temperatures) and by the pressure drops across additional components. There may therefore be options to improve the outcomes achieved with evaporative cooling by re-examining duct and coil sizes to minimise the impact of generally higher air volumes. This was not considered in this study.

The overall conclusion from these results is that the most robust technology tested was the dew-point cooler, without additional direct evaporative cooling, which provides positive results in all climate zones bar CZ1 and CZ8 and is furthermore a relatively simple modular addition to a design or even as a retrofit. The major constraint in the application of dewpoint coolers appears to be the limited maximum size of individual units and the associated space requirements.[2] It is further noted that systems with a higher minimum supply air temperature, such as underfloor systems, would be expected to yield greater savings than reported in this paper.

FURTHER CONTROL OPTIMISATION OF SC 2C

In SC 2c the dew-point cooler is operated when conditions permit at a zone temperature of 22°C. This supplements the economy cycle, which is enabled when conditions permit at a zone temperature of 21.5°C to 22°C. However, the chilled water cooling comes into operation between 23°C and 23.5°C, meaning that the chiller is called in to bring the supply air temperature down to 12°C before the VAV starts increasing air volume (from 23.5°C to 24°C). This reduces the extent to which the system can operate solely on the dewpoint cooler. To examine the potential for greater use of the dewpoint cooler, a series of additional scenarios was run as follows:

- **SC 2c-2:** When the dew-point cooler is operating, the supply air temperature set-point controlling the chilled water valve drops from 22.5°C to 12°C as the zone temperature rises from 23.5°C to 24°C.
- **SC 2c-3:** When the dew-point cooler is operating, the supply air temperature set-point controlling the chilled water valve drops from 22.5° to 12°C as the zone temperature rises from 23.25°C to 23.75°C.
- **SC 2c-4:** When the dew-point cooler is operating, the supply air temperature set-point controlling the chilled water valve drops from 22.5° to 12°C as the zone temperature rises from 23.75°C to 24.25°C.

2 The largest dewpoint cooler in the Seeley Climate Wizard range has a supply air volume of 12,800l/s and a plant footprint of approximately 15m² [Seeley International, 3]

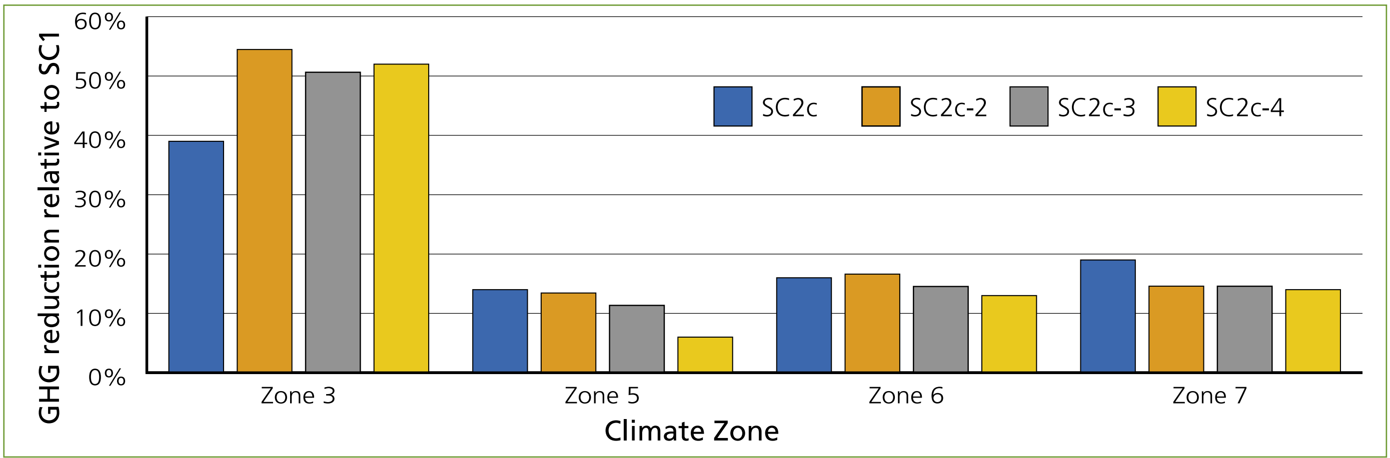

Figure 11. Impact of further control optimisation of SC 2c

In each of the above scenarios, when the dew-point cooler is not operating, the supply air temperature set-point controlling the chilled water valve control operates as per SC 2c. The results in terms of greenhouse emissions are shown in Figure 11 for a subset of climate zones.

The achieved savings show some sensitivity to the detail of control, dependent on climate, most significantly for the arid Climate Zone 3 (Alice Springs). In other climate zones, the impacts are somewhat more marginal. In all cases, the different control scenarios cause only minor modulation of the chiller and fan energy and do not fundamentally change the operation of the system.

CONCLUSION

A simulation model in IES-VE of a 5,000m^2, NCC 2019 Section J compliant office building with a VAV system has been used to test a range of evaporative cooling technologies across the major Australian climate zones.

Overall, the results indicate that the dew-point cooler is the most robust evaporative cooling system, with strong applications in arid climate zones 3 and 4 (greenhouse gas savings of 13–55 per cent) and smaller but significant savings in climate zones 2, 5, 6 and 7 (13–19 per cent) The modular nature of dew-point coolers means that they are a potential retrofit option for building with sufficient roof space although that there are limits on the size of available units that may restrict applicability in larger buildings.

In Climate Zone 8, it has been demonstrated that a direct/indirect evaporative cooling combination (SC 2a) can effectively supplant the need for a chiller, making this system potentially viable.

Evaporative cooling is rarely applied in office buildings in Australia. Given the substantial savings possible, and the pressure to drive office buildings towards net zero emissions, there appears to be a good argument for the use of evaporative cooling to drive HVAC efficiency beyond the current boundaries of best practice VAV systems.

Furthermore, the potential savings may be higher for HVAC systems with higher minimum supply air temperatures, such as underfloor systems. Overall, there is a strong case for greater use of evaporative cooling in Australian office buildings.

Work reported in this paper was undertaken for the PRIME Net Zero Energy HVAC Technology Road Map project.

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