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FINAL REPORT

An electrified residential thermal system using heat pumps and thermal storage

July 2026



Final report

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Acknowledgement of Country

The authors of this report would like to respectfully acknowledge the Traditional Owners of the ancestral lands throughout Australia and their connection to land, sea and community. We recognise their continuing connection to the land, waters, and culture and pay our respects to them, their cultures and to their Elders past, present, and emerging.

What is RACE for 2030?

RACE for 2030 is a 10-year cooperative research centre (CRC) with $350 million of resources to fund and support research towards a reliable, affordable, and clean energy future.

Disclaimer

The authors have used all due care and skill to ensure the material is accurate as at the date of this report. The authors do not accept any responsibility for any loss that may arise by anyone relying upon its contents.

Executive Summary

This report presents the design, development, prototyping, and performance evaluation of a novel 3-in-1 heat pump system for residential applications. The system integrates space heating, space cooling, and domestic hot water (DHW) production into a single reversible heat pump coupled with hot and cold thermal energy storage. The central design philosophy is to decouple energy generation from energy use, enabling thermal energy to be produced during periods of low-cost or low-emissions electricity (particularly rooftop PV surplus and low daytime grid prices) and stored for later use.

The motivation for this work is driven by several converging trends in the Australian residential energy sector. These include high and growing rooftop PV penetration, declining feed-in tariffs that reduce the value of exported solar electricity, increasing interest in distributed energy storage, the emergence of time-of-use electricity pricing with low or zero-cost daytime energy, and the ongoing electrification of residential buildings as gas connections are phased out. Together, these trends create a strong opportunity for thermal storage-based electrified heating and cooling systems that maximise self-consumption of renewable electricity, while helping to stabilise the grid.

A full-scale prototype was designed and constructed for a representative detached dwelling (230 m², 5-star NatHERS, Melbourne climate zone 6, five occupants). The system comprises an 8-kW R32 reversible heat pump, 1,000-L hot water storage, 2,000-L cold water storage, a high-capacity DHW heat exchanger, circulation pumps, motorised valves, and purpose-designed fan coil units (FCUs). The prototype was assembled and tested in a controlled laboratory environment at ActronAir facilities.

A key innovation of the project is the development of a hybrid digital twin framework, combining TRNSYS system simulation with detailed first-principles Python models, particularly for the FCU, where standard simulation tools were insufficient. The FCU model captures coupled heat transfer and fluid dynamics, including pressure drop, outlet air conditions, and water-side thermal behaviour. This modelling framework enabled system sizing, operational optimisation, and validation against experimental data.

The TRNSYS model was verified against laboratory experiments, demonstrating good agreement in both thermal storage stratification behaviour and dynamic charging/discharging performance. This validation confirms that the model is suitable for system-level analysis, design optimisation, and future scenario evaluation.

Component-level testing of the water-to-air coil showed strong agreement between simulated and measured performance, with minor deviations attributed primarily to sensor response delays and measurement phase shifts. Overall, the modelling approach provides a reliable basis for design iteration and system optimisation.

Economic analyses show that this 3-in-1 system can significantly reduce operating costs compared with a conventional baseline system (central HVAC plus separate DHW heat pump). Two operating cases were evaluated: grid-only operation under time-of-use (ToU) tariffs and hybrid operation with rooftop PV. While ToU-based operation provides moderate savings in regions with strong tariff

differentials (notably Melbourne and Sydney), the integration of rooftop PV delivers substantially higher energy cost savings, and exceeding 45% in all cases. A bottom-up cost assessment indicates that the system can be delivered within ±25% of a typical current baseline system cost, with estimated simple payback periods of approximately 4.5 years (Melbourne and Sydney) and 6 years (Brisbane).

From an environmental perspective, the system provides substantial emissions reductions by shifting electricity consumption from high-emissions evening periods to low-emissions daytime periods aligned with solar PV generation. Compared to a baseline system with PV, the 3-in-1 heat pump achieves emissions reductions of approximately 52–56% in Melbourne, 64–68% in Sydney, and 67–69% in Brisbane over the 2025–2035 period. When scaled across residential stock, cumulative emissions reductions could reach hundreds of millions of kilograms of CO_2-e across Australian states, even under conservative adoption assumptions.

Importantly, the results presented are conservative, as the system is not yet fully optimised in terms of control strategy or configuration. Further improvements in scheduling, storage utilisation, and predictive control are expected to deliver additional gains in both economic and environmental performance.

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1 Project journey

The project aimed to develop a cost-optimised residential thermal energy system that integrates a reversible heat pump with hot and cold thermal storage to provide space heating, space cooling, and domestic hot water (DHW) from a single appliance. The core design philosophy was to incorporate thermal storage into the system architecture, enabling the use of rooftop photovoltaic (PV) electricity or low-emissions grid electricity to generate and store heating and cooling energy during periods of high renewable electricity availability. The stored thermal energy can then be delivered to the dwelling on demand, independent of when the energy was generated.

By shifting energy consumption to periods of low-cost and low-emissions electricity, the system has the potential to reduce both capital and operating costs (CAPEX and OPEX) compared with conventional residential heating, cooling, and hot water solutions. Throughout this report, this concept is referred to as the **3-in-1 heat pump system.**

Several market and policy developments have created a strong opportunity for the deployment of such a product in Australia:

- **High penetration of rooftop PV systems** – Australia has one of the highest rates of rooftop PV adoption in the world. According to the Clean Energy Council [1], by 2025 more than 4.3 million Australian homes, equivalent to approximately one in three households, had installed rooftop PV systems. In addition, the average installed system size has continued to increase, reaching approximately 10.6 kW by the end of 2025.
- **Declining solar PV feed-in tariffs** – The financial value of exporting surplus solar electricity to the grid has steadily decreased, reducing the economic benefit of rooftop PV systems for households that are unable to utilise a significant proportion of their solar generation during daytime hours. This creates an incentive for technologies that can increase on-site utilisation of solar electricity.
- **High cost of electrochemical battery storage** – While residential battery systems have become increasingly popular, their installed cost remains relatively high. As of May 2026, typical residential battery systems cost approximately $1,000 per kWh of usable storage capacity [2] , with adoption often relying on government incentives and rebate programs.
- **Growth of time-of-use and solar soak tariffs** – Electricity retailers are increasingly offering low-cost, and in some cases free, electricity during the middle of the day. These tariffs are driven by the growing contribution of both utility-scale and distributed solar generation, which can create periods of excess electricity supply. Thermal energy storage provides an effective means of capturing value from these low-cost electricity periods.
- **Electrification of residential buildings and the phase-out of natural gas** – Policy frameworks across several Australian jurisdictions are accelerating the transition away from residential gas consumption. For example, regulations in Victoria now require new dwellings to be fully electric and prohibit new connections to the gas network. This trend is expected to increase the demand for efficient electric technologies capable of delivering multiple thermal services from a single integrated system.

1.1 Concept design

The system incorporates a single reversible heat pump capable of producing both hot water (up to 65°C) and chilled water (down to 1°C), together with thermal energy storage. The concept was developed around two possible storage configurations:

1- **Dedicated hot and cold storage tanks** – one cold-water storage tank for space cooling and one hot-water storage tank for space heating and DHW.
2- **Interchangeable hot/cold storage tank** – one thermal storage tank that can operate in either heating or cooling mode, combined with a dedicated DHW storage tank.

The schematic presented in Figure 1 represents the first configuration. While the second configuration has the potential to reduce system footprint and further lower capital costs, the dedicated hot and cold storage arrangement was selected for the current prototype development phase for several reasons:

1- **Evaluation of thermal storage performance** – The selected configuration enabled the testing of thermal performance and stratification characteristics in two storage tanks of different sizes (a 2,000-L cold water tank and a 1,000-L hot water tank). This provided valuable insights into storage behaviour and informed the design of future system configurations.
2- **Operational flexibility between heating and cooling modes** – The use of separate hot and cold storage tanks allows the system to rapidly switch between heating and cooling operation, making it suitable for climates and seasons characterised by variable weather conditions and fluctuating thermal loads.
3- **Increased cooling storage capacity** – Thermal energy storage for cooling is inherently more challenging to achieve in a compact form because chilled water stores relatively little sensible cooling energy per unit volume. The large cold-water tank provided sufficient cooling storage capacity for maximum operational flexibility during system evaluation.

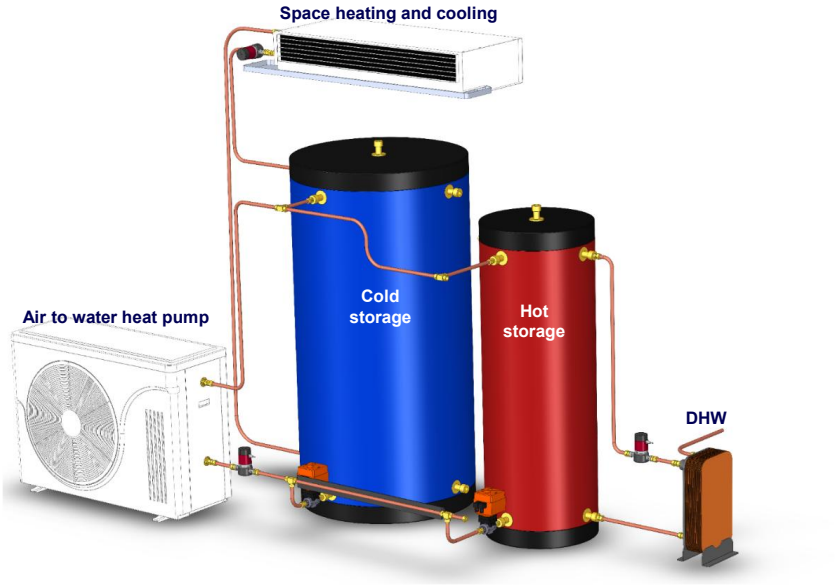


Figure 1. The conceptual configuration of the 3-in-1 heat pump system with one heat pump unit, two storage tanks which can deliver space heating and cooling via a fan coil unit and domestic hot water (DHW) through a plate heat exchanger.

The findings from this development phase provide the technical knowledge required to assess and optimise alternative storage configurations in future iterations. Accordingly, the primary objective of the current prototype was to validate the performance and operability of the two-tank configuration.

1.2 What makes this project distinctive

The 3-in-1 heat pump system incorporates several innovative features that differentiate it from conventional residential heating, cooling, and hot water solutions:

- Integrated delivery of three thermal services – The system provides space heating, space cooling, and DHW through a single heat pump and thermal energy storage (HP + TES) architecture. This integrated approach enables electrical load shifting and optimisation of energy consumption in response to rooftop PV generation and time-of-use electricity tariffs.
- Digital twin-based design optimisation – A detailed digital twin model has been developed to simultaneously optimise system component sizing and operational strategies. The model incorporates building-specific heating, cooling, and hot water demand profiles together with time-of-use electricity tariffs to maximise economic and energy performance.
- Reduced refrigerant charge – By using a single reversible heat pump to generate both heating and cooling, the system requires significantly less refrigerant than conventional solutions that rely on multiple independent heating and cooling units. This reduces both system complexity and environmental impact.
- Integration with rooftop PV systems – The system can be coordinated with onsite solar PV generation to maximise self-consumption of renewable electricity, reduce electricity imports from the grid, and improve overall energy efficiency.

Collectively, these features enable the 3-in-1 heat pump system to deliver a highly integrated, flexible, and future-ready solution for residential thermal comfort and hot water production.

1.3 The project structure

The project was designed to develop and validate a full-scale working prototype of the 3-in-1 heat pump system. The primary objective was to assess whether an appropriately selected combination of heat pump, thermal storage, and heat delivery units could provide sufficient heating, cooling, and DHW capacity with a system footprint suitable for medium- and high-density residential applications.

To achieve this objective, the project followed the development pathway outlined in Figure 2 over an 18-month period. The work progressed through a series of stages encompassing system modelling, component selection, prototype design, construction, commissioning, and performance evaluation. This structured approach enabled key technical assumptions to be validated and provided the data required to assess the technical feasibility and future commercial potential of the 3-in-1 heat pump concept.

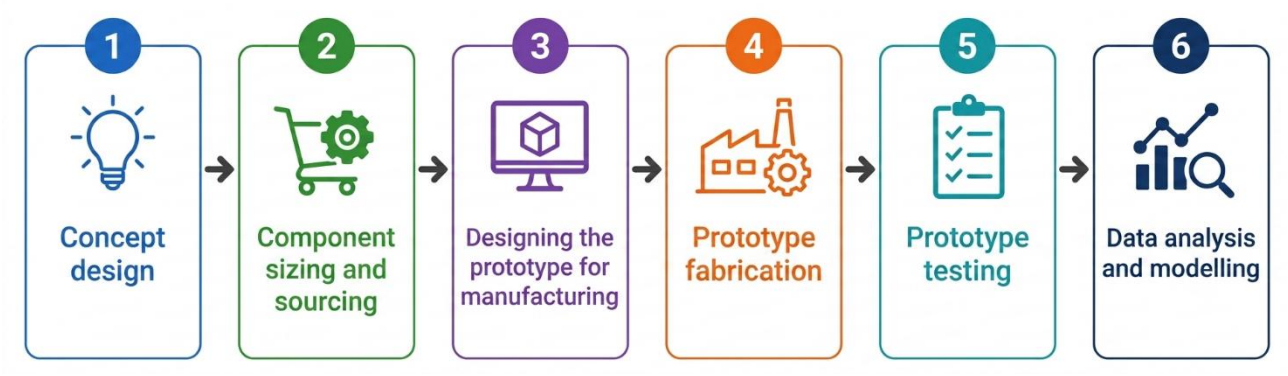


Figure 2. The project lifecycle.

1.4 Early risks and mitigations

Several technical and project delivery risks were identified during the early stages of the project, and mitigation measures were implemented to minimise their impact on project outcomes.

At the outset, an R290 (propane) heat pump was considered the preferred technology for the prototype. This preference was based on two key advantages:

- Higher hot water temperatures – R290 heat pumps are capable of delivering hot water temperatures in the range of 70–80°C, which can reduce the required hot water storage volume and improve overall system compactness.
- Low environmental impact – R290 is a natural refrigerant with an extremely low global warming potential (GWP), making it well aligned with industry and regulatory trends towards the phase-out of high-GWP refrigerants.

However, the adoption of R290 introduced project risks related to equipment availability, lead times, and safety requirements associated with handling a flammable refrigerant within the industry partner's testing facilities. To mitigate these risks and maintain the project schedule, an R32 heat pump was selected for the initial prototype development, integration, and testing activities, while retaining R290 as a preferred option for future system development and commercialisation.

A second challenge was the availability of suitable thermal storage tanks within the Australian market. Storage tanks of the required size and configuration were difficult to source locally, creating a risk to project timeline and prototype performance objectives. The project team therefore evaluated both custom-fabricated solutions and international suppliers before ultimately procuring the tanks from an overseas manufacturer that met the technical requirements.

Another key project risk related to the performance of commercially available fan FCUs. Existing FCUs were not capable of delivering the required heat transfer performance. As a result, the design and development of a high-performance FCU was established as a major project activity.

1.5 Digital twin, component engineering, and procurement

To support system design and optimise prototype performance, the project team developed a hybrid digital twin environment that combined Python-based component models with a TRNSYS system simulation model. This modelling framework was used to simulate annual energy flows, optimise

component sizing, evaluate operational strategies, and compare the performance of R290 and R32 heat pump technologies under realistic operating conditions.

One of the key components that could not be modelled with sufficient detail using standard TRNSYS libraries was the FCU. The thermal and hydraulic performance of the FCU is fundamental to the success of the 3-in-1 heat pump concept.

To address this limitation, the project team developed a detailed first-principles model of the FCU in Python. The model incorporates both heat transfer and fluid mechanics and is capable of simulating:

- air-side pressure drop,
- water-side pressure drop,
- heat transfer between the air and water streams,
- supply and return water temperatures,
- outlet air temperature, and
- airflow and thermal performance under varying operating conditions.

This modelling capability enabled the FCU design to be optimised prior to fabrication and provided a platform for assessing how the unit responds to control signals and interacts with the hot and cold thermal storage tanks.

The outputs of the digital twin and component-level modelling activities were subsequently used to guide equipment procurement and prototype development, ensuring that the selected components aligned with the system's performance objectives and design constraints.

1.6 Thermal load analysis

The sizing of the major system components for the full-scale prototype required the development of representative heating, cooling, and domestic hot water load profiles for a specific residential building. For this purpose, a detached 230-m² dwelling comprising four bedrooms and a living area was selected as the case study. The dwelling was assumed to achieve a 5-star NatHERS energy rating and to be located in Climate Zone 6 (Melbourne, Victoria).

This building was intentionally chosen because Melbourne's climate is characterised by relatively high space-heating demands, which present a challenging application for residential heat pump systems from both sizing and economic perspectives. Designing the system for this operating environment therefore provided a robust basis for assessing the technical feasibility and commercial viability of the 3-in-1 heat pump concept.

The floor plan of the house is presented in Figure 3. By selecting a relatively large dwelling with significant heating requirements, the project established a conservative design case to evaluate whether a compact 3-in-1 heat pump system could successfully provide space heating, space cooling, and domestic hot water while maintaining practical storage volumes and system costs.

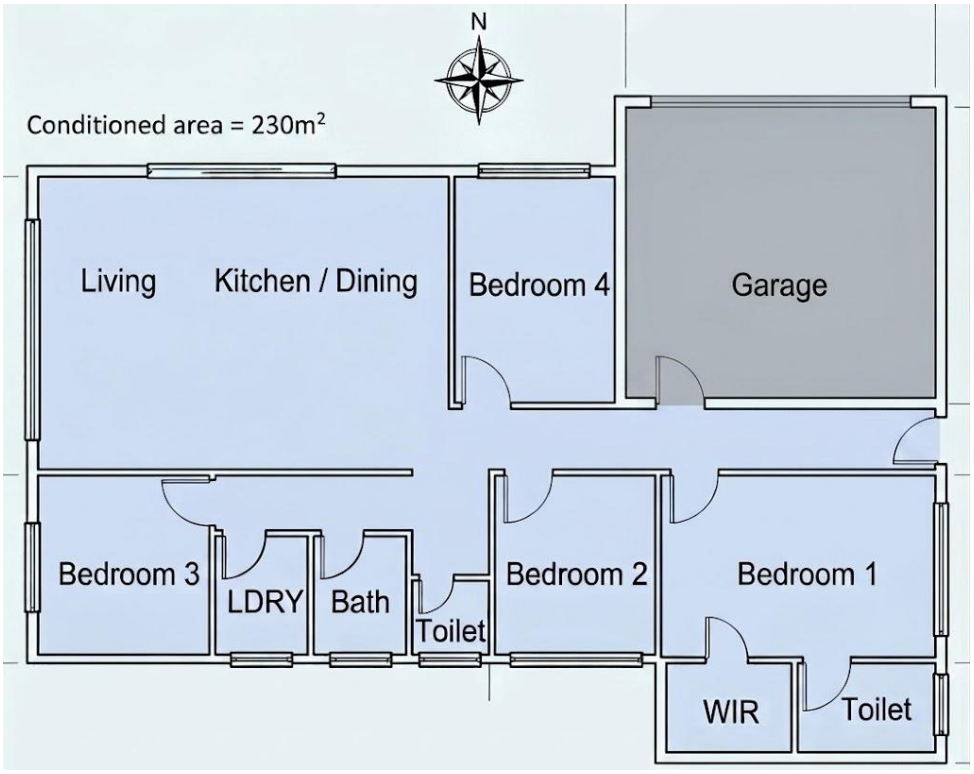


Figure 3. The floor plan of the case study house; the blue areas are 24/7 air-conditioned.

The geometry of the reference building used to generate the thermal load profiles was developed within the TRNSYS modelling environment and is shown in Figure 4. This building model enables heating and cooling loads to be calculated dynamically using location-specific weather data, providing a realistic representation of the building's thermal behaviour throughout the year.

The modelling approach also provides the flexibility to readily modify key building parameters, including floor area, layout, construction characteristics, orientation, and geographic location. This capability allows the impact of different dwelling types and climatic conditions on system performance to be evaluated efficiently.

The heating and cooling loads generated by the building model are directly integrated into the TRNSYS simulation of the 3-in-1 heat pump system, enabling the interaction between the building, thermal storage, heat pump, and heat delivery systems to be assessed under realistic operating conditions.

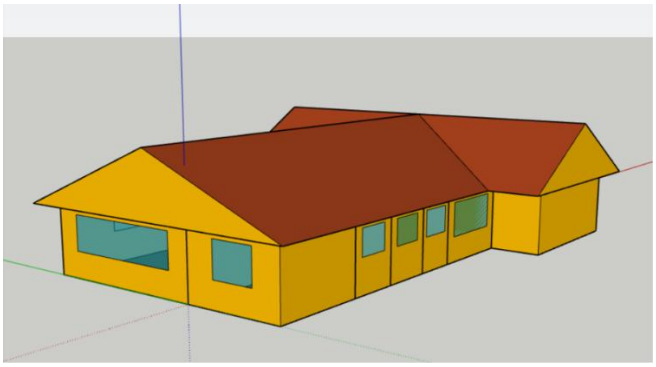


Figure 4. The three-dimensional model of the building created in TRNSYS for calculating the thermal load profile.

2 Project outcomes

Figure 5 and Figure 6 presents the daily thermal demand of the reference dwelling over a typical year, separated into heating and cooling loads. The DHW demand is included within the total heating load and provides a relatively constant baseload throughout the year, whereas the space heating and cooling demands exhibit strong seasonal variation. The DHW load profiles were generated in accordance with Australian standards for residential hot water consumption.

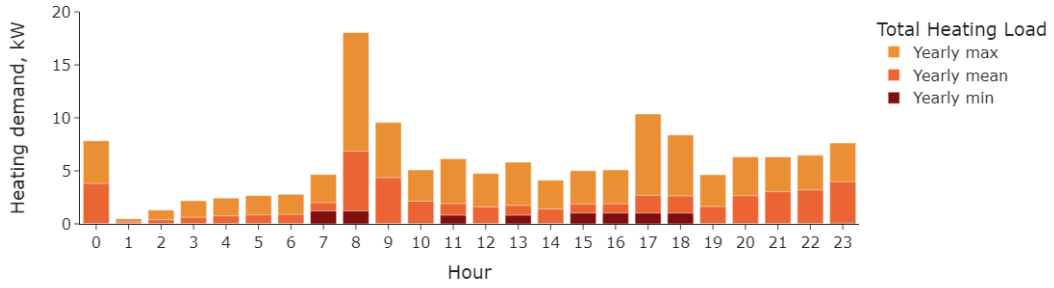


Figure 5. Hourly heating demand profiles for a representative day with the highest annual heating load (Yearly Maximum), a day with the average annual heating load (Yearly Mean), and a day with the lowest annual heating load (Yearly Minimum). The hourly heating demand includes both space heating and domestic hot water (DHW) loads.

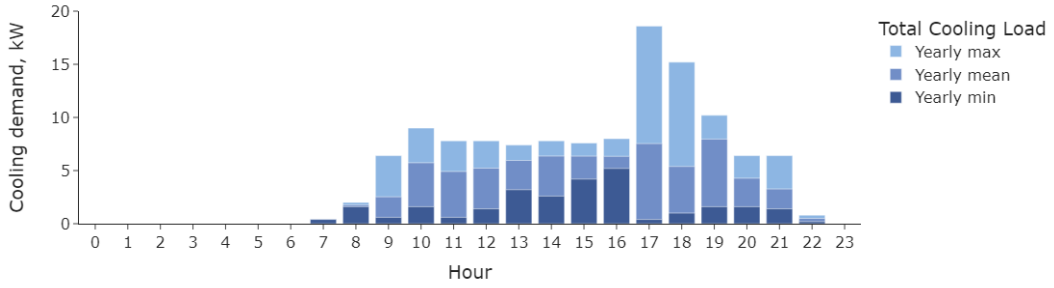


Figure 6. Hourly cooling demand profiles for a representative day with the highest annual cooling load (Yearly Maximum), a day with the average annual cooling load (Yearly Mean), and a day with the lowest annual cooling load (Yearly Minimum).

To determine appropriate thermal storage capacities, it was necessary to analyse the hourly distribution of thermal demand throughout the day. The annual maximum, minimum, and mean demand values were evaluated for each hour. The mean values were calculated using only those days with non-zero thermal demand to ensure representative operating conditions.

The analysis showed a substantial difference between the annual maximum and mean hourly heating demands. As a result, designing the system to satisfy the annual peak load while restricting heat pump operation exclusively to daytime hours would require impractically large storage capacities and heat pump sizes. Therefore, the first sizing scenario considered the annual peak heating demand profile with continuous heat pump operation (24 hours per day), as illustrated in Figure 7. Under these conditions, a hot thermal storage capacity of approximately 23 kWh, combined with a heat pump thermal capacity of 6 kW, would be sufficient to bridge overnight demand and satisfy the morning and evening heating peaks.

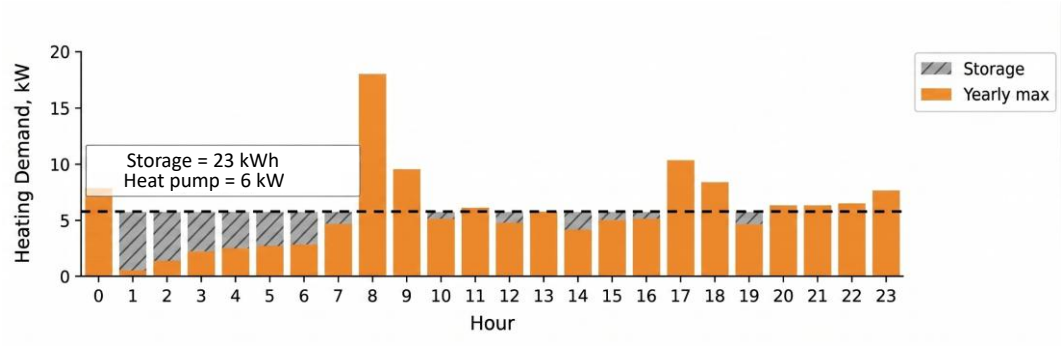


Figure 7. The sizing of hot tank for overnight storage and peak shaving in the morning; the hashed area shows the energy flowing into the storage.

A second scenario was then evaluated using the annual mean heating demand profile. Under these more typical operating conditions, it becomes feasible to restrict heat pump operation to selected periods of the day to maximise the utilisation of rooftop PV generation and/or avoid high electricity tariffs. As shown in Figure 8 and Figure 9, achieving this level of load shifting requires approximately 42 kWh of hot thermal storage together with a heat pump thermal capacity of 8–9 kW.

Since the requirements of the second scenario also encompass those of the first scenario, a heat pump with a thermal capacity of approximately 9 kW combined with 42 kWh of hot thermal storage was selected as the preferred design point. This configuration provides sufficient storage capacity to shift the majority of annual heating demand to periods of low-cost or renewable electricity availability, while still maintaining the capability to meet extreme winter heating loads by allowing heat pump operation outside the preferred operating window whenever the stored thermal energy falls below a predefined threshold.

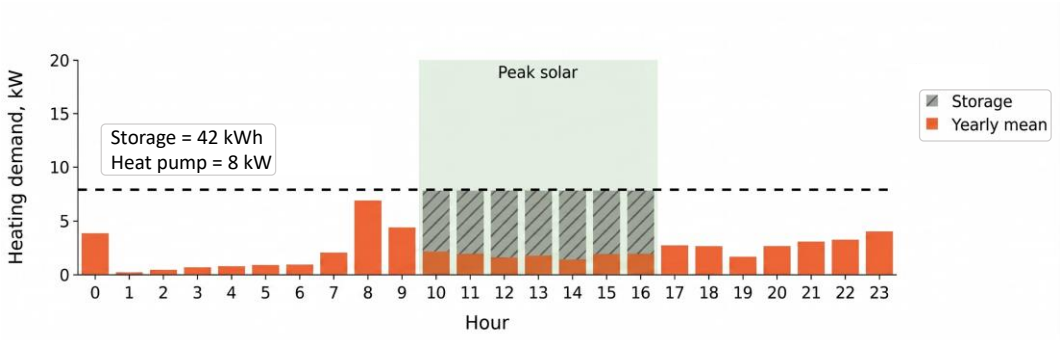


Figure 8. Heat pump operation restricted to peak solar periods; the hashed area shows the energy flowing into the storage tank.

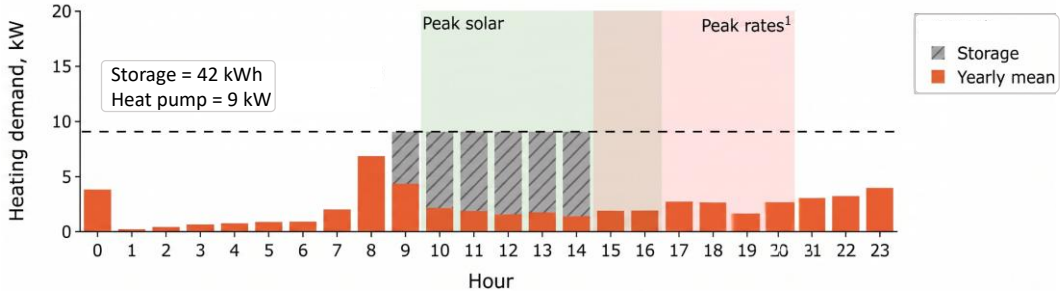


Figure 9. Heat pump operation restricted based on multiple criteria (e.g., to maximise peak solar but avoid peak grid rates).

A similar methodology was applied to the sizing of the cooling system, including the heat pump and cold thermal storage tank. The resulting sizing recommendations for both heating and cooling applications are summarised in Table 1.

Table 1. The resulting component sizes

Heat pump heating capacity [kW]	Heat pump cooling capacity [kW]	Hot tank capacity [kWh]	Cold tank capacity [kWh]
8-9	7	42	35

2.1 Water-to-air coil

The water-to-air coil unit was jointly designed and developed by the research team and ActronAir's engineering group. The manufactured prototype is shown in Figure 10. The design process was guided by several key performance constraints. Air velocity through the coil was limited to less than 2.5 m/s to prevent condensate carryover during cooling operation, while the air-side pressure drop was targeted to remain below approximately 120 Pa. This limit was established to constrain fan power consumption to around 150 W, equivalent to approximately 1% of the unit's thermal output.

The first-principles numerical model developed as part of the project was used extensively throughout the design process to predict heat transfer performance, air-side pressure drop, and water-side pressure drop for successive design iterations. This modelling capability was particularly important because excessive air-side pressure losses can significantly increase fan energy consumption and adversely affect the overall system efficiency and operating costs.



Figure 10. The manufactured coil.

The performance of the final coil design was assessed by comparing model predictions against experimental measurements. Under heating operation, the model slightly overestimates the heat transfer rate, as shown in Figure 11, resulting in corresponding overestimation of the air-side and water-side temperature changes. Despite this small bias, the simulated and measured results demonstrate good overall agreement, providing confidence in the model's ability to represent the thermal performance of the coil.

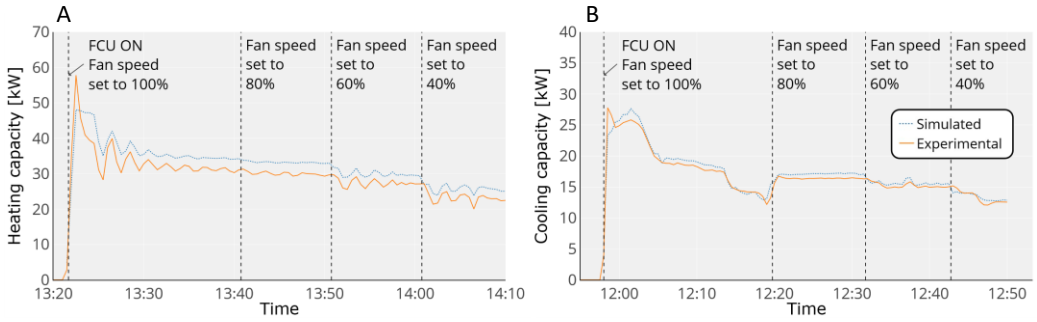


Figure 11. Heat tranfer rate of the coil under heating mode (A) and cooling mode (B).

Similarly, the predicted and measured air-side and water-side pressure drops are in close agreement, as presented in Figure 12. A slight phase shift can be observed in the simulated water-side pressure drop results, which is also reflected in the outlet temperature measurements. This discrepancy is likely attributable to the physical placement of sensors within the experimental setup and the finite response time of the measurement instruments, resulting in a time delay between changes in inlet conditions and their detection at the outlet measurement locations.

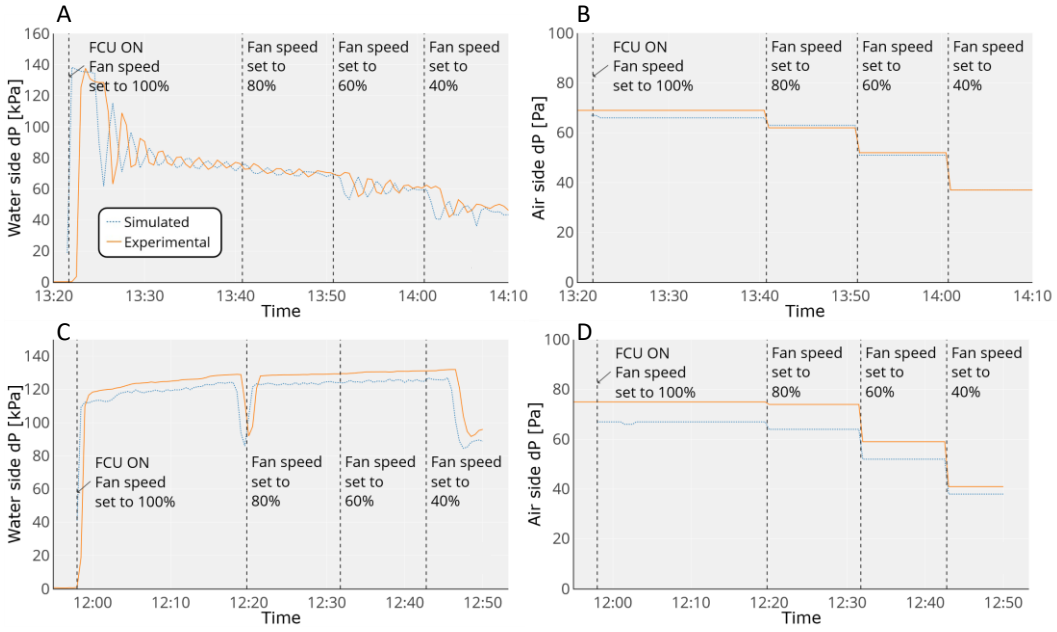


Figure 12. The pressure drop under heating mode at the water (A) and air side (B); Pressure drop under cooling mode at the water side (C) and air side (D) under cooling mode.

Overall, the experimental results demonstrate that the developed modelling framework is capable of accurately predicting both the thermal and hydraulic performance of the coil, providing a reliable tool for future design optimisation and system development activities.

The heat transfer rate to/from the air was calculated based on the inlet and outlet temperature of water in the fan coil to ensure that the latent heat of condensation at the air side of the coil is also accounted for, however minor discrepancies in heat transfer prediction may be related to condensation effects not fully captured in the model.

2.2 Thermal storage tanks

For the required thermal storage capacities mentioned earlier, the tank volumes needed were determined using the expected operating temperature ranges for both hot and cold thermal storage (see Table 2). The selected storage tanks, sourced from international suppliers, are shown in Figure 13.

Table 2. Calculated and selected thermal storage tank volumes. The *calculated volume* represents the effective storage volume required to achieve the target thermal storage capacity. The *selected nominal volume* corresponds to the manufacturer's rated tank capacity. The *effective tank volume* represents the usable water volume that can be extracted and circulated during operation.

	Required capacity [kWh]	Expected ΔT [°C]	Calculated volume [L]	Selected tank (nominal volume) [L]	Effective tank volume [L]
Hot tank	42	35	1028	1000	910
Cold tank	35	18	1670	2000	1600

From a commercialisation perspective, a more compact system configuration is considered feasible. For the thermal loads evaluated in this case study, it is envisaged that a single thermal storage tank with a volume between 1,000 L and 2,000 L, combined with a dedicated 300 L DHW tank, would be sufficient. In this arrangement, the larger storage tank would operate as either a hot or cold thermal storage vessel depending on the season, while the dedicated DHW tank would provide year-round hot water service. Given the limited overlap between peak heating and cooling seasons, this approach has the potential to significantly reduce system footprint and capital cost.

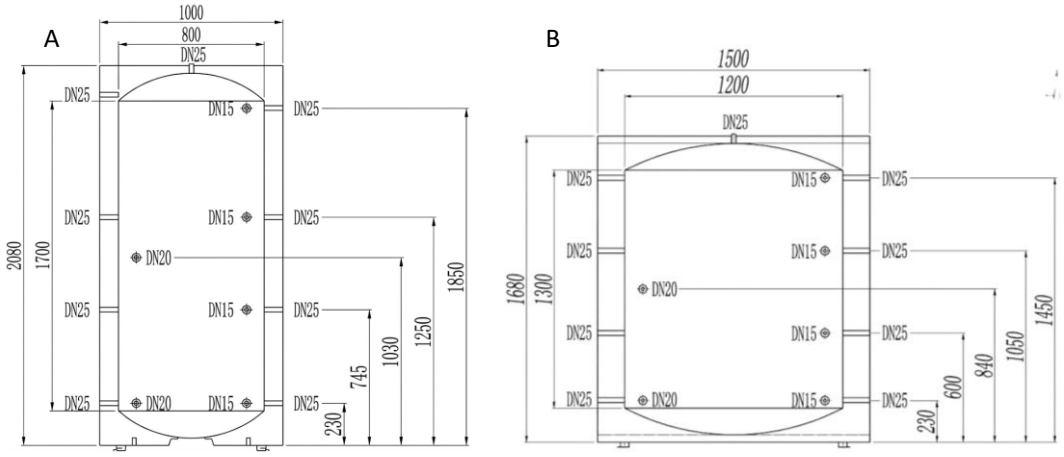


Figure 13. The configuration of the hot 1000L tank (A) and cold 2000L tank (B); dimension are in mm.

2.3 Integration and test readiness: assembly, controls, and lab setup

Following component procurement and fabrication, the prototype system was assembled and commissioned within ActronAir's climate chamber laboratories. The fully assembled prototype is shown in Figure 14. The prototype incorporated the major system components identified during the design phase, including an 8 kW R32 heat pump, a 1,000 L hot thermal storage tank, a 2,000 L cold thermal storage tank, a DHW plate heat exchanger with a water-to-water heating capacity of approximately 52–57 kW, three variable-speed circulation pumps, and specially designed FCU, with two alternative FCU designs selected for final performance testing.

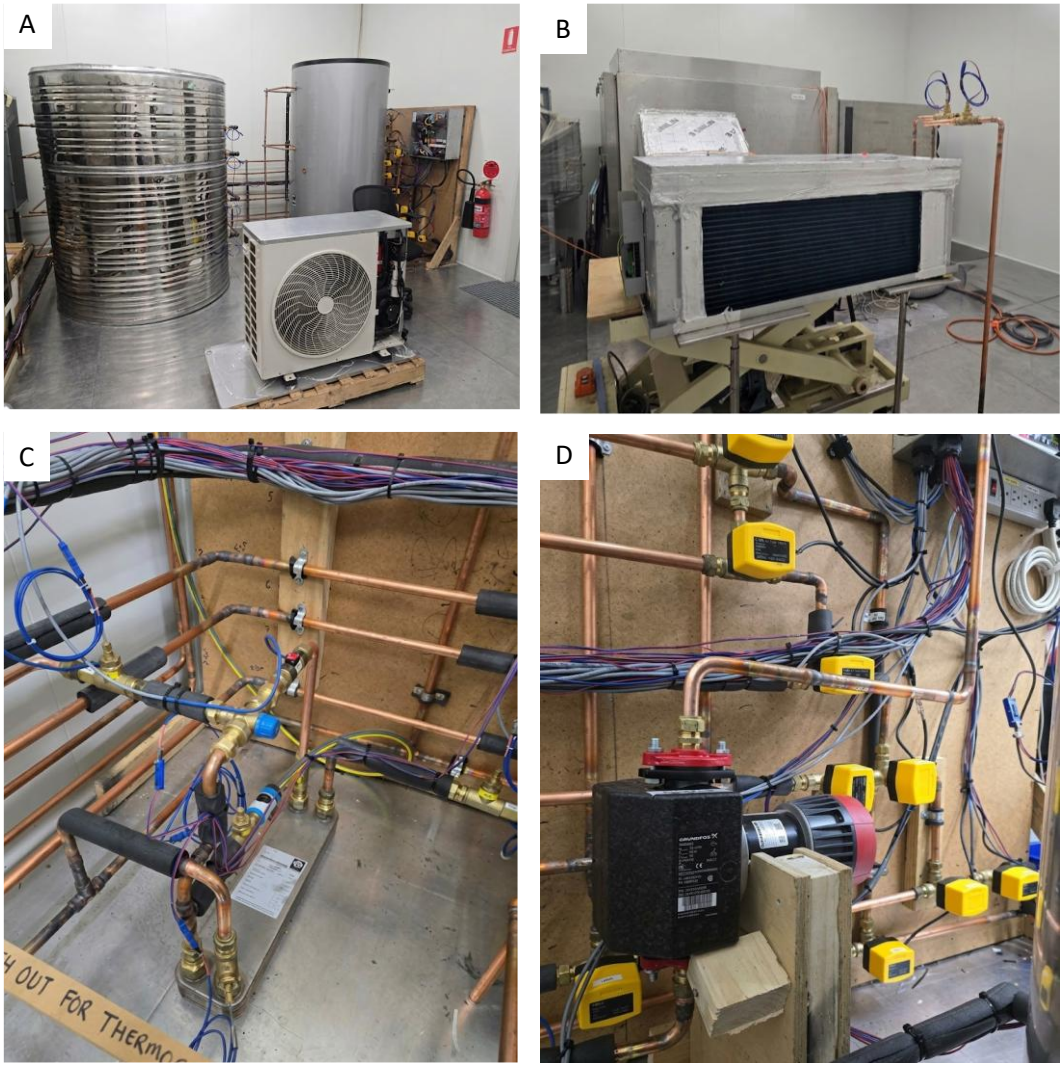


Figure 14. Fully assembled 3-in-1 heat pump system prototype: (A) thermal storage tanks and heat pump; (B) fan coil unit (FCU); (C) domestic hot water (DHW) plate heat exchanger; and (D) circulation pumps and motorised valve network used for flow routing. The photographs were taken prior to the installation of the 9 mm pipe insulation to improve visibility of the hydraulic components and pipework.

The heat pump was integrated with both thermal storage tanks through a network of motorised three-way valves. This hydraulic arrangement enables the system to direct water flow according to the selected operating mode and provides the flexibility required to support multiple thermal services. The valve network allows the system to:

- Charge the hot thermal storage tank;
- Charge the cold thermal storage tank;
- Supply the fan coil unit in heating mode;
- Supply the fan coil unit in cooling mode;
- Deliver thermal energy to the DHW heat exchanger; and
- Operate in combined modes involving simultaneous charging and discharging of thermal storage.

A dedicated control system was developed using LabVIEW hardware and software to coordinate the operation of all major system components. The control platform included a graphical user interface for system monitoring and operation, together with multiple predefined operating modes, including

hot tank charging, cold tank charging, space heating, space cooling, DHW production, standby operation, and simultaneous charging/discharging modes.

Data acquisition and control functions were implemented using National Instruments hardware, which managed valve actuation, circulation pump control, sensor monitoring, and data logging. This integrated control architecture provided the flexibility required for prototype testing while also establishing a platform for the development and evaluation of advanced control strategies in future project phases.

The successful assembly, commissioning, and control integration of the prototype established the foundation for the experimental testing and performance validation activities described in the following sections.

2.4 System testing

Following installation and assembly, all subsystems and components underwent individual commissioning and verification tests. These activities included measurement of pressure drops throughout the hydraulic circuits, validation of flow rates, and verification of all temperature sensors to ensure accurate operation and data acquisition.

To eliminate unintended interactions between air streams during testing, the FCU was installed in a separate controlled-environment room. This arrangement ensured that the heating or cooling output of the FCU did not influence the ambient conditions surrounding the heat pump, thereby allowing the performance of each subsystem to be evaluated independently. The air temperature and humidity conditions for the FCU tests were controlled using ActronAir's climate chamber facilities.

The heat pump itself was operated under a constant ambient temperature of approximately 25°C throughout the testing program. Maintaining a fixed ambient condition did not compromise the validity of the experimental results, as the primary objective of the testing was to evaluate the thermal storage, heat delivery, control functionality, and system integration aspects of the prototype rather than the variation in heat pump performance with outdoor conditions. This approach also improved test repeatability and enabled more consistent comparison between operating modes and system configurations. In contrast, the performance of the heat delivery/extraction subsystem (fan coil and hydronic circuit) was tested under airside conditions of approximately 20 °C for heating tests and 25 °C for cooling tests, representing scenarios expected in real-world operation. These conditions were kept within a band of ±2°C[1].

2.4.1 Heat pump

The charging and discharging power flow from the heat pump into the hot tank and from the hot tank to the fan coil is presented in Figure 15. The maximum temperature reached by the heat pump was ~64 °C which was 1°C below the target of 65°C expected from the R32 heat pump. During five hours of charging period, a total of 47 kWh was produced by the heat pump in heating mode. Subsequently, ~37 kWh of heat was extracted from the tank during the discharge period.

[1] In one case, the heating test airside temperature was ~17 which was lower than the desirable set point of 20°C.

In cooling mode, the minimum temperature reached ~3.8 °C which was 2.8 °C above the target chilled water temperature. The safety feature in the existing heat pump limits the water temperature preventing it from reaching 1 °C. Approximately 53 kWh of cooling was produced over 6.25 hours of operation of the heat pump confirming that the heat pump delivered ~8.5 kW of cooling on average while it reduced the temperature of the chilled water to 3.8°C.

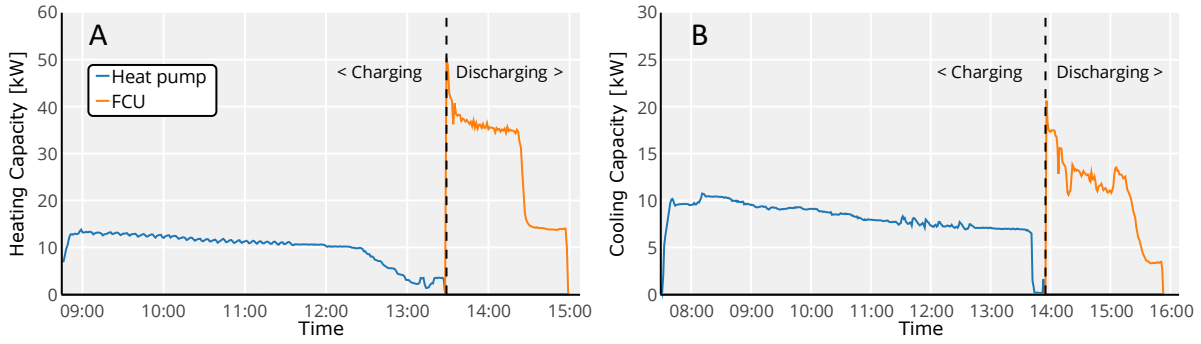


Figure 15. The thermal energy generation, storage, and use; the thermal power of the heat pump during charging and thermal power of the fan coil during discharge in (A) heating and (B) cooling mode.

2.4.2 Performance of the fan coil unit and its controller

The primary objective of this test was to assess whether the fan coil design with its PID controller could effectively maintain the return-water temperature to the thermal storage tank at the desired set point, while delivering the cooling/heating air at the target temperature.

Under heating mode when the fan coil was connected to hot tank, the coil was fed by 64°C and delivered air at ~50°C steadily. Under cooling mode, it delivered cooling air at 12-13 °C when it was fed by ~3.8°C chilled water (see Figure 16).

In heating Mode

- The fan coil delivered approximately 31 kW, 30 kW, 28 kW, and 22 kW of heating capacity at fan speeds of 100%, 80%, 60%, and 40%, respectively. The fan coil exceeded the designed heating power capacity of 15 kW by 100% at peak power. This excess capacity is a result of the coil needing to serve both the design heating and cooling loads. The coil sized for cooling becomes 'oversized' for heating.
- Across all operating conditions, a stable return-water temperature was achieved, consistent with the design target.
- The fan coil maintained a stable supply-air temperature between approximately 50°C and 52°C while operating with an inlet-air temperature of around 18°C, thereby meeting the target supply-air temperature requirement. The fan coil met the design target air supply temperature.
- The measured air-side pressure drops across the fan coil was 69 Pa, 62 Pa, 52 Pa, and 37 Pa at fan speeds of 100%, 80%, 60%, and 40%, respectively. The fan coil successfully kept the pressure drop at < 58% of the design target.

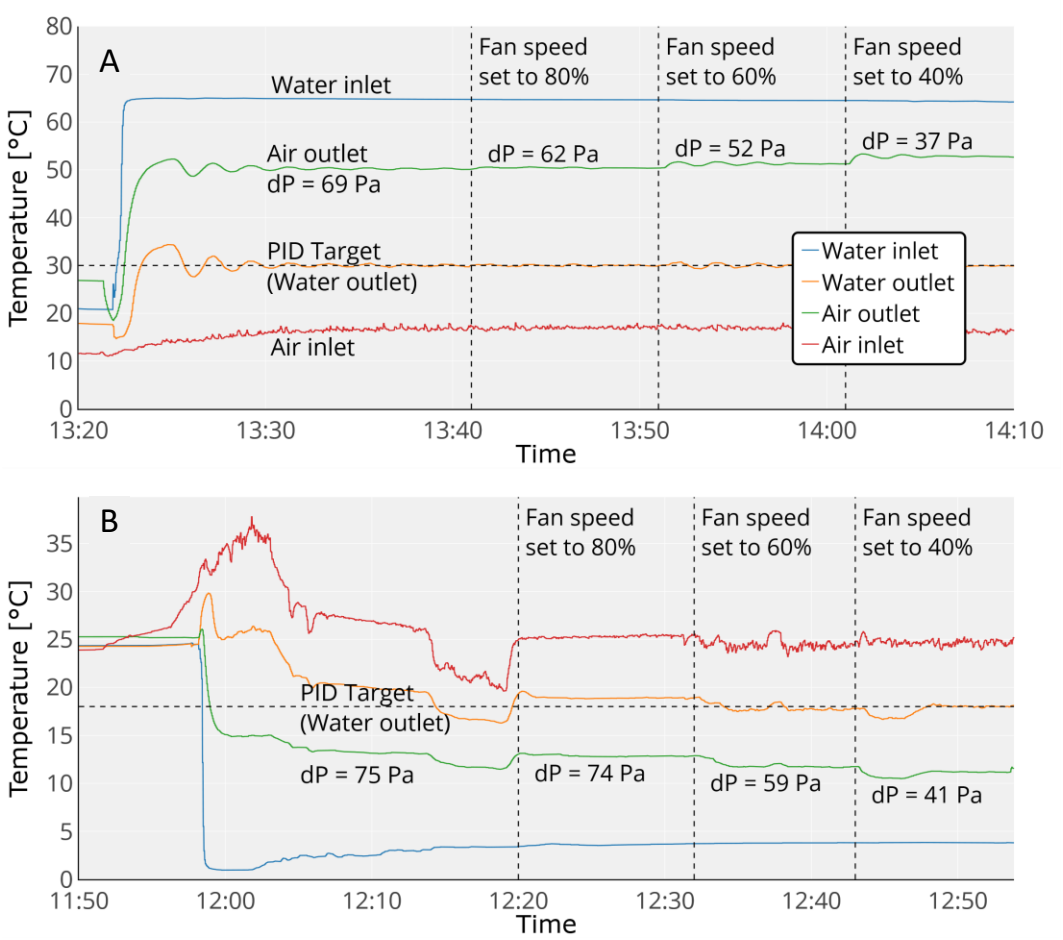


Figure 16. The thermal performance of the fan coil unit under (A) heating mode when it is fed by ~58°C water, delivering air at 45-50°C and (B) cooling mode when it is fed by 6°C water delivering air at 12-13°C

In cooling Mode

- The fan coil delivered approximately 17–18 kW, 16 kW, 15 kW, and 13 kW of cooling capacity at fan speeds of 100%, 80%, 60%, and 40%, respectively. It exceeded the design target of 15 kW by 13% at peak cooling power.
- A stable return-water temperature of 18°C was maintained at all fan speeds, in accordance with the design target.
- The fan coil maintained stable supply-air temperatures of approximately 14°C, 13°C, 12°C, and 11°C, at fan speeds of 100%, 80%, 60%, and 40%, respectively, while operating with an inlet-air temperature of 24–25°C. These results met the target supply-air temperature requirements.
- The measured air-side pressure drop across the fan coil was 75 Pa, 74 Pa, 59 Pa, and 41 Pa at fan speeds of 100%, 80%, 60%, and 40%, respectively. This mean that the fan coil kept the peak pressure drop at 63% of the peak allowed threshold of 120 Pa.

2.4.3 Storage tank performance

In heating mode

- The return water temperature achieved by the fan coil provided approximately 37 kWh of usable thermal storage capacity from the 1000 L thermal storage tank, corresponding to an

effective working volume of 910 L. The theoretical required capacity of the tank was 42 kWh (see Table 2). This means that that hot tank achieved 88% of the design target.

- Consistent stratification was maintained throughout discharge cycles. As it is seen more clearly in Figure 17 during discharging period, the temperature difference between the top and bottom of the tank show a difference of ~30°C, which is accepted as sufficient level of stratification in hot water tanks when not charging.

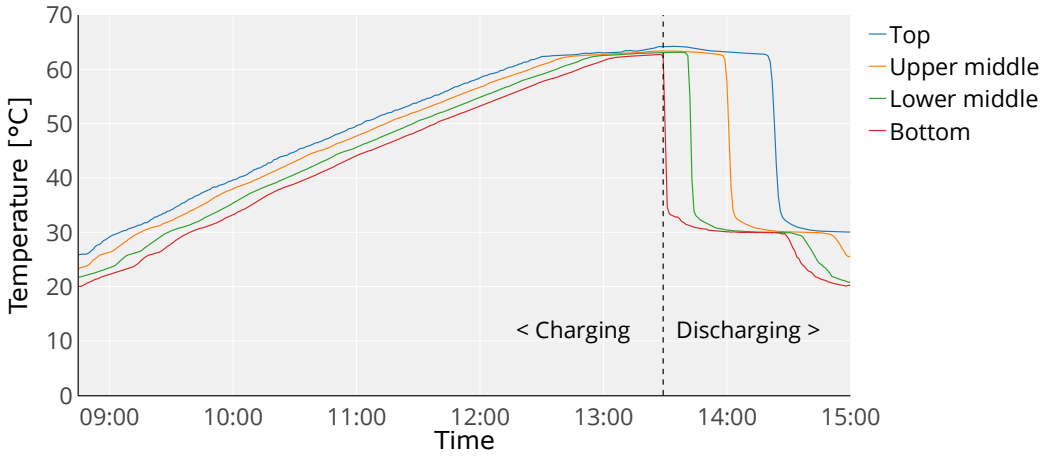


Figure 17. Temperature gradient inside the hot tank during charging and discharging.

In cooling mode

- This operating temperature range provided approximately 26 kWh of usable thermal storage capacity from the 2000 L thermal storage tank, corresponding to an effective working volume of 1600 L. Hence, the cold storage tank achieved ~75% of the required storage capacity of 35 kWh (see Table 2 and Figure 18).

Consistent stratification was maintained throughout discharge cycle. The stored chilled water showed an acceptable level of thermal stratification by a 13°C temperature difference between the top and bottom of the tank once charging stopped.

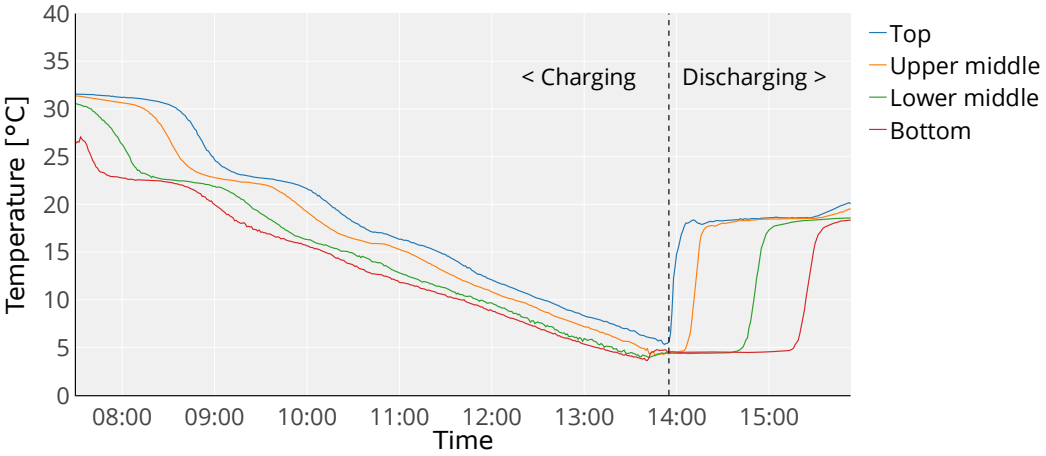


Figure 18. The vertical temperature distribution inside the cold tank during charging and discharging.

2.4.4 Domestic hot water subsystem

The DHW delivery unit extracts hot water from the hot tank to heat up mains water and deliver it at a set temperature of 45°C through a tempering valve. During testing, this unit achieved this set temperature with an accuracy of ±2 °C. The response time from pump activation to hot water delivery was approximately 25 seconds. The return water temperature to the storage tank reached the design set point of 30 °C, consistent with the fan coil return temperature. The heating power during operation was approximately 51 kW and reached 98% of the design target. Under variable load conditions, i.e. turning the hot water valve on and off, the system responded reliably.

2.4.5 Model verification

Under heating operation, with the heat pump connected to the hot thermal storage tank, the simulated charging and discharging cycles closely reproduced the experimental behaviour in terms of both timing and thermal stratification, as shown in Figure 19. Similarly, good agreement was obtained between the simulated and measured temperature distributions within the storage tanks during cooling operation. These results demonstrate that the model can accurately represent the dynamic thermal behaviour of the system and predict the performance of the key components with sufficient accuracy for design and assessment purposes.

The verified TRNSYS model forms the basis of the performance assessments and impact analysis presented in the subsequent sections of this report.

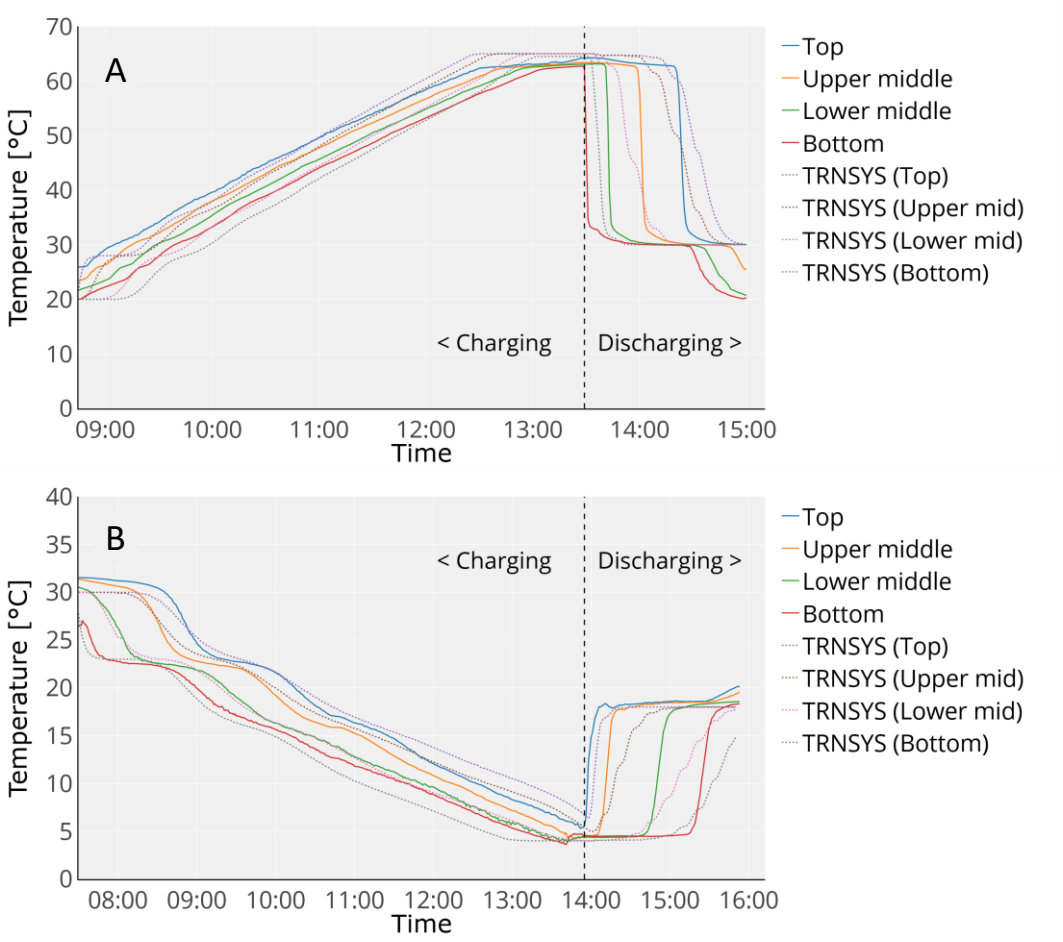


Figure 19. Temperature distribution within the thermal storage tanks during charging and discharging operation in heating mode (A) and cooling mode (B). Comparison between TRNSYS and experimental measurements.

3 Impact

This section presents an analysis of the potential cost and greenhouse gas emissions impacts of the 3-in-1 heat pump system, including operational energy costs and achievable savings relative to conventional residential thermal systems. The projected emissions reduction associated with deploying the system in residential applications until 2035 is also examined.

It is important to note that, at this stage of development, neither the system configuration nor the control strategy has been fully optimised. The primary objective of this project was to design, construct, and validate a functional prototype to demonstrate the operation of the integrated system and its individual components under realistic conditions. As such, further optimisation work is required to refine both system sizing and control logic, particularly to ensure that thermal energy is generated and stored at the most economically and environmentally advantageous times.

Although a semi-optimised control strategy was implemented during prototype testing, there remains significant potential for improvement through more advanced optimisation of operational scheduling, storage utilisation, and demand response strategies.

Accordingly, the cost savings and emissions reductions presented in this section should be interpreted as conservative estimates. They are expected to represent a lower bound of the system's potential performance. Further optimisation of both system design and control strategies is likely to yield substantially greater economic and environmental benefits.

3.1 Cost of 3-in-1 heat pump system

The operating cost of the 3-in-1 heat pump system is evaluated for two scenarios:

- Case 1: Operation using grid electricity only under ToU tariffs.
- Case 2: Operation using a combination of behind-the-meter rooftop PV electricity and grid electricity.

For comparison, the system performance is benchmarked against a baseline configuration consisting of a central HVAC system with a total heating and cooling capacity of 15 kW, without thermal storage, combined with a separate heat pump-based domestic hot water system sized to meet the demand of a 5–6 person household.

Three different locations as listed in Table 3 were selected for the analysis.

Table 3. Locations selected for cost and emissions analysis

Location	Description
Melbourne	location in climate zone 6 with significant heating demand and moderate cooling demand
Sydney	location in climate zone 5 with moderate heating and cooling demand
Brisbane	location in in climate zone 2 with high cooling load

For the Melbourne case, the rooftop PV system size was assumed to be 9 kW, reflecting the higher heating demand and reduced winter solar availability characteristic of this climate. This assumption is also consistent with observed national trends, where average residential PV system sizes are approaching 10–11 kW, making a 9 kW system a realistic and conservative choice. For Sydney and

Brisbane, a PV system size of 6 kW was assumed, reflecting their higher solar availability and different demand profiles. The applicable ToU electricity tariffs for each location are presented in Table 4.

Table 4. Grid electricity tariffs used in the running cost analysis of this report; in all locations it has been assumed that grid electricity is free between 11:00 and 14:00.

Location	*Peak*	*Shoulder*	*Off-peak*
Melbourne	15:00-21:00 ($0.40 per kWh)	-	All other times ($0.20 per kWh)
Sydney	15:00-21:00 ($0.69 per kWh)	-	All other times ($0.38 per kWh)
Brisbane	16:00-21:00 ($0.50 per kWh)	12:00 am -11:00 & 21:00-23:59 ($0.40 per kWh)	All other times ($0.38 per kWh)

The resulting annual operating costs for both the baseline system and the 3-in-1 heat pump system are summarised in Table 5. Under grid-only operation, Brisbane exhibits limited cost-saving potential due to relatively small differences between peak and off-peak tariff rates. In contrast, Melbourne and Sydney show more pronounced savings due to larger tariff differentials, enabling more effective load shifting through thermal storage. When rooftop PV is included, operating cost savings increase significantly across all locations, exceeding 45% in all cases.

Table 5. Yearly running cost of 3-in-1 heat pump system vs the baseline; the savings from the use of the 3-in-1 heat pump option have been presented in brackets for each case.

	YEARLY RUNNING COST USING GRID ONLY WITH TOU TARIFFS		YEARLY RUNNING COST USING GRID PLUS ROOFTOP PV	
	Baseline	3-in-1 heat pump	Baseline	3-in-1 heat pump
MELBOURNE	$1,723	$1,182 (31%)	$1,128	$612 (46%)
SYDNEY	$1,614	$1,169 (28%)	$939	$457 (51%)
BRISBANE	$1,174	$976 (17%)	$607	$333 (45%)

These operational savings should be considered in conjunction with the capital cost of the 3-in-1 heat pump system. A bottom-up cost assessment conducted by the research team indicates that the system can be manufactured and delivered to end users at a capital cost within ±25% of the baseline system cost.

Assuming the upper bound of this cost range, the estimated simple payback period for a user adopting the 3-in-1 heat pump system in place of the baseline system is approximately 4.5 years in Melbourne and Sydney, and approximately 6 years in Brisbane.

3.2 Emissions savings

The assessment of emissions savings associated with the 3-in-1 heat pump system requires the use of time-resolved grid emissions intensity profiles. At the time of writing, hourly emissions intensity data, including forward projections, were obtained from the Australian Energy Regulator [3–5]. These profiles are presented in Figure 20 and show a clear reduction in emissions intensity during the middle of the day, particularly within the 09:00–15:00 period, driven by high penetration of PV generation.

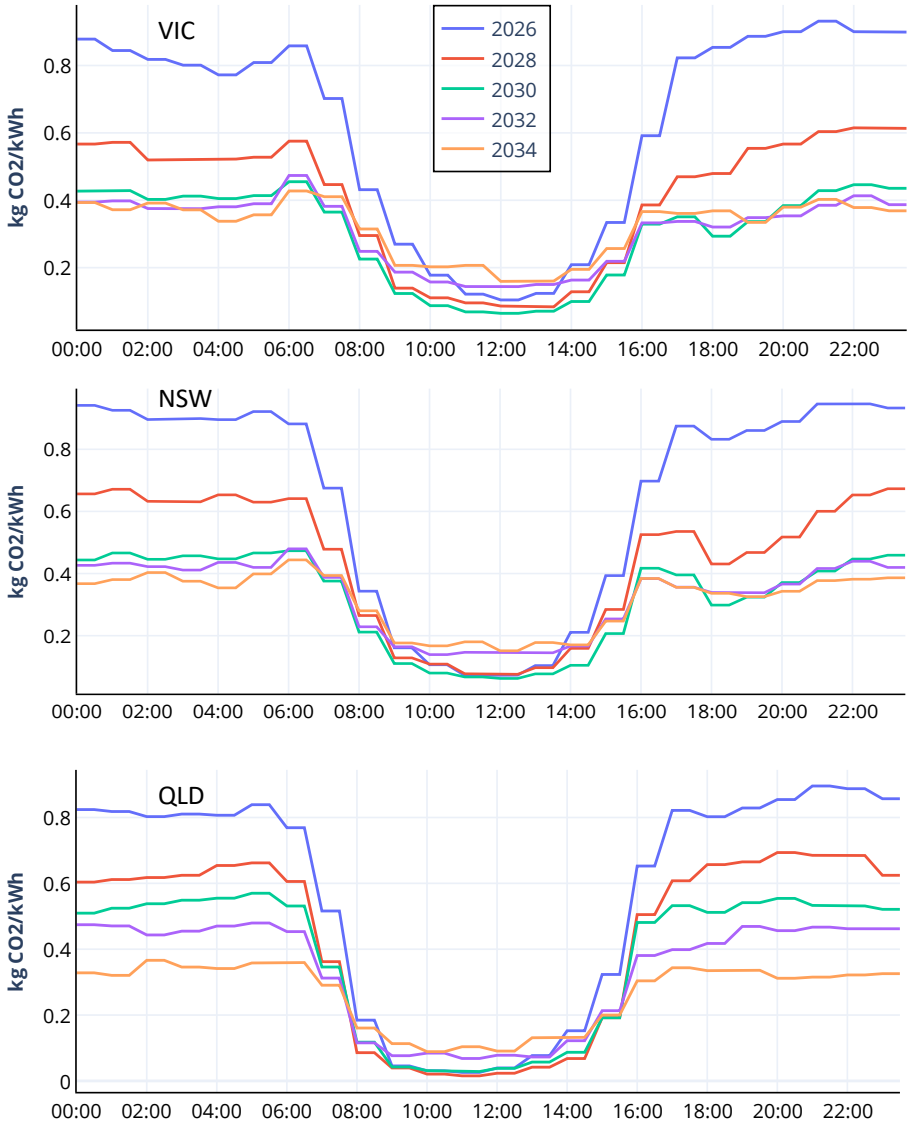


Figure 20. The mean hourly emissions profile of the grid projected into the future; for the sake of clarity, only every second year has been presented.

The operational emissions of the 3-in-1 heat pump system are generally lower than those of the baseline system due to its ability to shift electricity consumption from evening and night-time periods to midday periods when grid emissions intensity is lower. This temporal shifting is enabled by thermal storage and aligns electricity demand with periods of higher renewable energy availability. The resulting shifted electricity consumption profiles for each location are also shown in Figure 21.

Overall, the analysis indicates that electricity consumption outside daytime hours is reduced by approximately 46%, 47%, and 36% in Melbourne, Sydney, and Brisbane, respectively. This load shifting not only contributes to reduced operating costs, but also has a direct impact on system-wide greenhouse gas emissions, as described below.

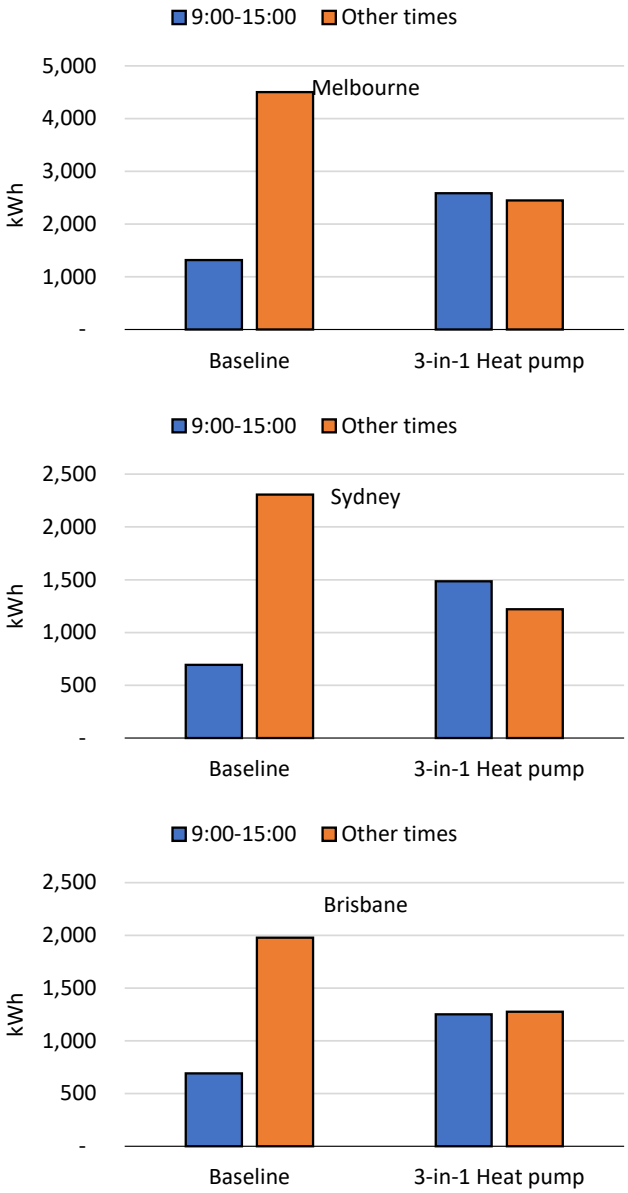


Figure 21. The annual grid electricity load in daytime period for baseline compared to 3-in-1 Heat pump.

Figure 22 presents the resulting emissions trajectories for a single residential installation over the period 2025–2035. As shown in the baseline case, annual emissions decline progressively between 2025 and 2035, reflecting the ongoing decarbonisation of the Australian electricity grid through increased penetration of renewable generation. However, even under these projected conditions, the baseline heat pump system does not achieve zero operational emissions in any of the locations considered by 2035.

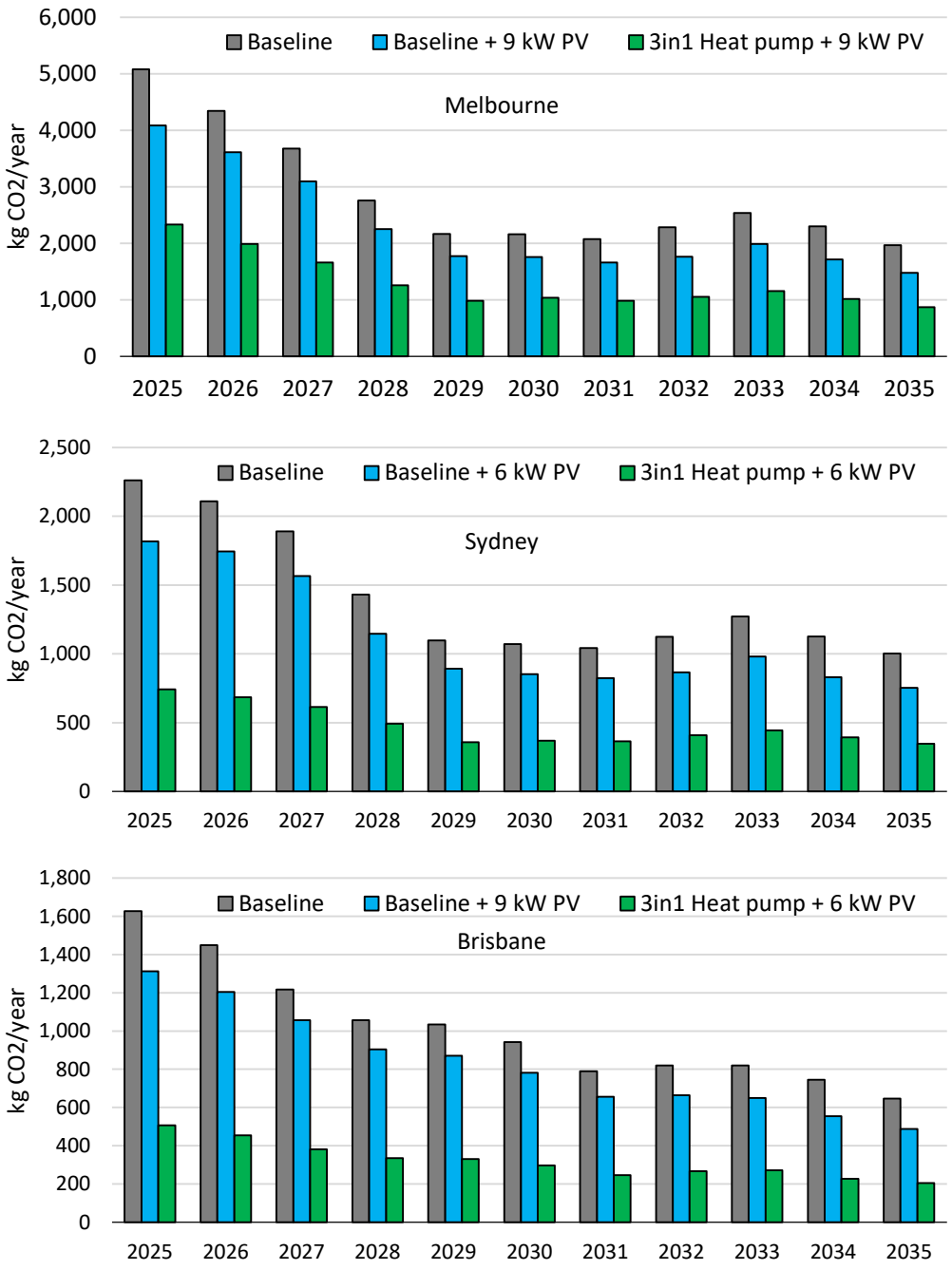


Figure 22. The projected annual emissions of the 3-in-1 heat pump system in comparison to the baseline with and without rooftop PV.

When rooftop PV is added to the baseline system, annual emissions are reduced by approximately 16–25% in Melbourne, 17–26% in Sydney, and 13–26% in Brisbane, relative to the baseline system. In contrast, the 3-in-1 heat pump system delivers substantially greater emissions reductions, achieving approximately 52–56% in Melbourne, 64–68% in Sydney, and 67–69% in Brisbane over the same period.

These results demonstrate that combining thermal storage with load shifting towards low-emissions periods of grid operation provides a significantly more effective emissions reduction pathway than PV integration alone.

3.3 Aggregated emissions reduction

To provide a conservative estimate of the potential emissions savings associated with large-scale deployment of the 3-in-1 heat pump system, the following assumptions were adopted:

- Total Addressable Market (TAM): The TAM was defined as all detached residential dwellings within each Australian state. The number of eligible dwellings is provided in Table 5.
- Adoption rate: It was assumed that the penetration of the 3-in-1 heat pump system begins in 2027, with an initial adoption rate of 1% of the detached housing stock per year. Under this linear uptake assumption, cumulative adoption reaches approximately 9% of the housing stock by 2035 in each state.
- Emissions comparison basis: Emissions savings were calculated by comparing the 3-in-1 heat pump system combined with rooftop PV against a baseline configuration consisting of the conventional system also paired with rooftop PV in each state. This ensures that the results isolate the impact of the thermal storage and load-shifting capability of the proposed system.

Table 6. Assumed grid load and relative size of population for Australian major cities used for impact calculations

	No. of detached homes
VIC	1,825,000
NSW	1,950,000
QLD	1,500,000

The resulting annual and cumulative emissions savings are presented in Figure 23. Under these assumptions, cumulative emissions reductions over the analysis period reach approximately 606 million kg CO_2-e in Victoria, 424 million kg CO_2-e in New South Wales, and 264 million kg CO_2-e in Queensland.

These results represent a conservative estimate of potential impact, as they do not account for accelerated technology adoption, further grid decarbonisation beyond current projections, or additional optimisation of system operation and control strategies.

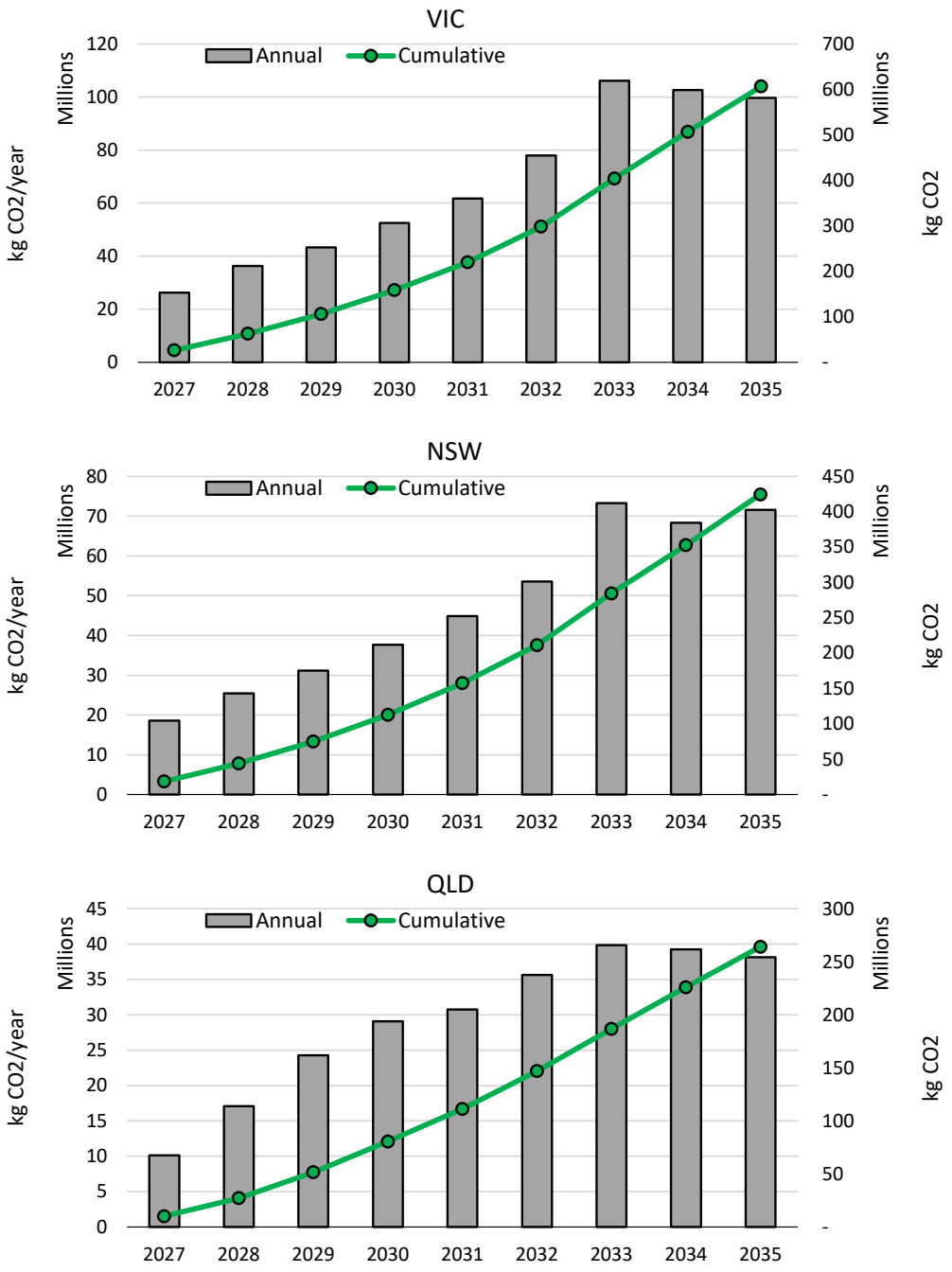


Figure 23. Annual emissions savings (left axis) and cumulative emissions savings (right axis) associated with deployment of the 3-in-1 heat pump system in each state from 2027 onwards. Savings are calculated relative to a baseline configuration comprising the conventional system combined with rooftop PV in the corresponding state.

4 Outcome of IRG and stakeholder workshops

4.1 Research and knowledge sharing

The key findings of this work was shared with the broader community in the energy productivity field through a seminar on 14 May 2026, organised by the Australian Alliance for Energy Productivity (A2EP). The recordings of the event are available to the public through the A2EP's website.

A journal publication containing the key findings on the lifecycle cost and emissions savings of the 3-in-1 heat pumps system is being prepared with publication expected in the first quarter of 2027. This publication will outline in more details how this system can be optimised for saving emissions and/or levelised cost of thermal energy in various climate zones in Australia.

4.2 Outcome of IRG and stakeholder engagements

For improving the commercial readiness of the 3-in-1 heat pump system, the key feedback from the IRG is as the following:

1- The cold and hot storage size needs to be reduced to optimise the footprint of the system more suitable for high-density built environment.
2- The system needs to be suitable both for new builds as well as retrofit in existing buildings.
3- The response time of the system needs to be high enough to ensure that the users receive hot water as well as space conditioning with minimal delay on par with existing products in the market.

Conclusion

The development, integration, and testing of the 3-in-1 heat-pump-with-thermal-storage system have demonstrated that the prototype is both technically robust and capable of delivering the performance outcomes envisioned during the concept-design stage. Across all major subsystems, including the heat pump, hydronic circuit, storage tanks, fan-coil units, control logic, and the domestic-hot-water module, the system either met or exceeded the original design targets established at the outset of the project. These results provide strong validation of the fundamental design principles and confirm that the integrated architecture is viable for residential applications requiring space heating, space cooling, and domestic hot water delivered from a single, optimised thermal-energy system.

Performance testing highlighted several strengths that directly support the case for commercialisation. The fan-coil units exceeded the design heating and cooling capacities, enabling compact storage-tank sizing. The current hot and cold tanks were 1000L and 2000L for the specific case study home and offered 88% and 75% of the required hot and cold storage capacities. Several major modifications have been envisaged to reduce the storage tank footprints while increasing their capacity to 100% of the required level. These techniques include increasing the hot tank temperature to >70°C, lowering the chilled water tank to <-3°C as well as combining the hot and cold tank into a

single interchangeable tank for space heating and cooling, with an add-on small 300L tank for hot water. It is estimated that with this method, the required storage footprint reduces from the current 3000L to a combination of one 850L and one 300L tank.

The 3-in-1 heat pump operated reliably across all operating modes, achieving temperatures close to design targets for both hot- and cold-water production. The thermal energy storage tanks demonstrated stable and effective stratification, while domestic hot water delivery remained consistent and within the desired performance range. The control system also performed reliably across the full range of operating conditions. Collectively, these results provide strong evidence that the system can meet the performance requirements expected of a future commercial product.

The CAPEX of the 3-in-1 heat pump system is expected to be within ±25% of the lowest-cost alternative solutions providing the same thermal services. Combined with operating cost reductions of approximately 40–50% when paired with rooftop PV, the system is projected to achieve a simple payback period of around 4–6 years for end users. This would enable households to electrify heating and cooling services in a cost-effective manner while delivering value comparable to, or greater than, residential electrochemical battery systems, with no degradation of the storage over time and an expected tank life in excess of 20 years.

The integration of TES enables substantial load shifting towards daytime periods. When supplied by onsite PV generation or low-emissions grid electricity during midday hours, the system has the potential to reduce household operational emissions by approximately 50–70%. Even under conservative market uptake assumptions, deployment of the technology across only 1% of its target market annually would deliver significant cumulative emissions reductions. These outcomes provide a

strong rationale for supportive policy measures, including rebates and low-emissions heating upgrade programs, which could accelerate adoption, improve project economics for end users, and contribute to the decarbonisation of the built environment.

Given this strong technical foundation, the project team believes that the system has substantial commercialisation potential, contingent primarily on design refinement and optimisation for manufacturability, cost, and installation footprint. The concept, integrating a reversible heat pump with both hot and cold stratified storage, has now been validated at prototype scale, and the achieved performance metrics support the pathway toward a market-ready solution.

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