



TECHNICAL ARTICLE

Experimental Investigation and Component-Level Modelling of a Self-Regulating Small-Scale Vapour-Compression Atmospheric Water Generator

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ABSTRACT

Atmospheric water harvesting using small-scale vapour-compression systems offers a decentralised water source, but performance varies strongly with ambient psychrometric conditions. This study experimentally evaluates a compact propane (R290) atmospheric water generator with series-coupled evaporator and condenser airflow and develops a component-level model of its partially wet fin-and-tube evaporator. Three extended tests were used to characterise operation. Model predictions were compared with cumulative collected water using a dry-wet coil formulation and refrigerant-side capacity matching. The simulated cumulative water production differed from the measured value by 6%. In the two 24 h tests, the unit produced 7.0 L outdoors and 4.5 L indoors using 7.2 kWh in each case, corresponding to specific energy consumptions of 1.03 and 1.60 kWh L⁻¹, respectively. Water production was higher under the more humid inlet conditions. The model predicted higher water production at the lower nominal fan setting, reaching approximately 0.50–0.54 L h⁻¹. Predicted compressor power also increased. The analysed outdoor profile showed its highest production during the nocturnal high-humidity period; this is a profile-specific result rather than an annual operating window. The model provides a basis for comparing operating trends under different inlet conditions

KEYWORDS Atmospheric water harvesting, Dehumidifier, Vapour compression cycle, Wet coil, Propane

1. Introduction

Water availability is closely linked to climate, energy use and water infrastructure. Climate change is intensifying the water cycle, leading to frequent droughts, while rapid urbanization in hot-arid regions like the Middle East continues to strain conventional freshwater supplies (Caretta *et al.* 2023). In parallel, water scarcity is not only a matter of quantity but also of water quality and accessibility and institutional definitions emphasise that scarcity arises when available water resources and/or their quality are insufficient to meet demand. This framing is particularly relevant for hot-arid regions, where the energy cost of delivering safe water can be substantial. In regions such as the Middle East and North Africa, water insecurity is often compounded by governance challenges, climate vulnerability, and infrastructure dependencies, reinforcing the need for distributed, resilient alternatives that can complement conventional supply pathways (World Bank 2018). Atmospheric Water Harvesting (AWH) provides an option for distributed water production without a conventional water-supply network (Tu and Hwang 2020). AWH technologies extract water vapour from ambient air and convert it into liquid water. The required energy input depends on the process used. AWH technologies are generally classified into passive condensation (e.g., radiative cooling), sorption-based systems using porous materials, and active condensation via electrically driven cooling cycles

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(Tu and Hwang 2020), (Tashtoush and Alshoubaki 2023). Passive radiative approaches have demonstrated the potential for near-continuous operation under favourable conditions, while sorption-driven concepts can exploit solar heat for regeneration and reduce direct electrical demand (Nishad *et al.* 2025). However, sorption systems can face mass-transfer limitations and often operate cyclically, whereas active condensation systems are more straightforward to control and can achieve high water productivity in humid climates but remain sensitive to ambient conditions and electricity intensity (Salehi *et al.*, 2020).

Among active condensation methods, VCC-based systems use mature refrigeration components and are already commercially available (Tu and Hwang 2020). In practice, many buildings already operate VCC cooling equipment that produces condensate as a by-product, highlighting the conceptual feasibility of using condensation as a local water source (Al-Farayedhi *et al.* 2014). However, the efficiency of VCC-based AWH is highly climate-dependent. In arid regions, the produced water amount can reduce as relative humidity drop (Gido *et al.* 2016), (Patel *et al.*, 2020). Controlled chamber testing and field studies in arid regions have reported large variability in specific energy consumption and daily yield, indicating that catalogue values alone are insufficient for robust techno-economic or environmental assessment (Bagheri 2018), (Khalaf *et al.*, 2024). For VCC-based AWH, water production should be reported together with electricity consumption. Local temperature and humidity also determine when operation is practical (Salehi *et al.*, 2020), (Patel *et al.*, 2020). As a simulation, Habeebullah (Habeebullah 2009) carried out a numerical investigation into the feasibility of harvesting atmospheric water in hot, humid climates using evaporator coils. The simulated coil had a heat transfer surface area of 47 m² and a frontal area of 1.17 m². For a representative humid condition (34.6 °C and 61% RH), the predicted water yield reached 24.8 kg m⁻² day⁻¹ at a moist-air velocity of 1 m s⁻¹. The study also reported that WHR increases with increasing ambient humidity ratio (and decreases when the humidity ratio is lower). In addition, a performance chart was proposed to estimate the maximum water production for ambient conditions spanning 25–40 °C and 30–100% RH. In the study by Al-Farayedhi *et al.* (Al-Farayedhi *et al.* 2014), analytical and experimental investigations were conducted on such systems under Dhahran, Saudi Arabia conditions. In particular, summer ambient conditions (25–50 °C and 15–90% RH) were examined. The experimental results showed that, for a split air conditioner, the rated input power and rated cooling capacity were 2.075 kW and 4.747 kW, respectively. At an air volumetric flow rate of 0.135 m³ s⁻¹, the daily yields in June, July, August, and September were 34.5, 38.1, 70.1, and 59.4 kg·day⁻¹, respectively. Patel *et al.* (Patel *et al.*, 2020) investigated a VCC-based atmospheric water generator across seven laboratory-replicated climatic conditions, reporting that specific energy consumption varies significantly, ranging from 1.15 to 10.74 kWh/L, depending on the ambient psychrometric state as they observed sharp decline in condensation rates under dry conditions.

Recent studies demonstrate why water yield and energy use must be reported together. In year-round field testing under hot and humid conditions, (Faraz Ahmad *et al.*, 2022) reported strong seasonal variation in both production and specific energy consumption, with favourable periods reaching approximately 0.84 kWh L⁻¹ and less favourable periods approaching 2.1 kWh L⁻¹. A mobile atmospheric water generator designed for arid climates achieved water-harvesting rates of 0.23–1.45 kg h⁻¹ with a specific energy consumption of 1.00–4.65 kWh kg⁻¹ showing the sensitivity of refrigeration-based harvesting to the processed-air state and system configuration (Pokorny *et al.* 2022). (Raveesh *et al.* 2023) similarly found that climate and operating conditions strongly determine system viability. Long-term testing of a larger vapour-compression system in an arid climate reported an average specific energy consumption of 0.26 kWh L⁻¹, illustrating the performance advantage that may be achieved through scale and design optimisation but also the limited comparability of catalogue values across device capacities and climates (Khalaf *et al.*, 2024).

Water-production rate alone is not sufficient for comparing AWG systems. Water production rate alone favours high-throughput operation but does not reveal the associated electrical penalty. Specific energy consumption (kWh L⁻¹) and its reciprocal, water productivity (L kWh⁻¹), provide

direct water–energy indicators, while system boundaries must be stated because compressor-only, refrigeration-cycle and whole-device metrics are not equivalent (Cattani *et al.*, 2024). Techno-economic studies further show that climate, utilisation, electricity source, capital recovery and water treatment materially affect levelised water cost. Condensation-based AWH is therefore most practical when local conditions allow sufficiently low SEC (Gayoso *et al.*, 2024).

Despite the expanding AWH literature, gaps remain when compact, off-the-shelf dehumidifiers are repurposed for water harvesting. Although system-level COP energy consumption is often reported, fewer works provide an evaporator-centred, physics-based model that captures partially wet coil behaviour and links it to water production in a reproducible manner. In dehumidifying coils, the heat and mass transfer process typically transitions from a dry region to a wet region along the coil, and this partial wetting can significantly affect both sensible and latent performance. Effectiveness–NTU modelling frameworks for cooling and dehumidifying coils provide a well-established route to represent this coupled behaviour using psychrometric relationships. Moreover, climate-based feasibility is often discussed using coarse averages, whereas hourly weather datasets enable more actionable identification of day/night harvesting potential and seasonal variability, which is essential for sustainability-aligned operation strategies in arid climates.

The literature establishes the climate sensitivity of vapour-compression atmospheric water generation, but fewer studies combine long-duration measurements from a compact off-the-shelf unit with a component-level model that explicitly resolves partial wetting of the evaporator and links coil behaviour to water yield and energy use. The contribution of this study is therefore applied rather than a new thermodynamic theory. A compact R290 unit with series-coupled evaporator–condenser airflow was tested over extended periods. A dry–wet evaporator model was assessed against cumulative water production. The model was then used to examine the effects of psychrometric conditions and nominal airflow setting on water yield and compressor power.

2. Method

The methodology begins with experimental trials to characterize the VCC parameters required for model validation and subsequent simulations. The VCC unit features a fin-and-tube evaporator and condenser arranged in series and ambient air sequentially passes through the evaporator, where temperature and humidity are reduced, and then through the condenser coil. In this configuration, the air leaving the evaporator subsequently passes through the condenser, providing a direct thermal coupling between the two heat exchangers. Optimizing pressure ratio is critical, as compressor performance is the primary driver of overall system efficiency. Marques *et al.* (Marques *et al.*, 2014) reported that increasing compressor displacement from 4 to 8 cm³ increased the modelled isentropic efficiency by about 50%. Despite these insights, a gap remains in characterizing small-scale VCC units with coupled evaporator–condenser airflows. Furthermore, while instantaneous condensate mass (on a per-second or per-minute basis) is difficult to measure precisely in small-scale systems, the model is validated using the cumulative collected mass, providing detailed insights into the condensation rate through experimental data analysis

2.1. Experimental Setup and Procedures

The water harvesting experiments were conducted using a small-scale vapour-compression cycle unit specifically designed for dehumidification. The system uses propane (R290) as a natural refrigerant, known for its low global warming potential. The core of the unit is a finned-tube heat exchanger assembly where the evaporator and condenser are arranged in a coupled series airflow configuration. In this setup, moist ambient air is driven across the evaporator coil by a forced fan, lowering its temperature and humidity. Subsequently, the pre-cooled air passes through the condenser coil. This series-airflow arrangement thermally couples the evaporator outlet air to the condenser inlet condition. The core component, a finned-tube evaporator, was geometrically characterized for the physics-based model. The coil comprises 20 copper tubes arranged in 10 rows

and 2 columns. Each tube has an outer diameter of 7.45 mm and a wall thickness of 0.5 mm. The heat exchanger has a face area of 0.22 m (height) $\times$ 0.27 m (width), with a flow depth of 0.050 m. The fin pack features a fin thickness of 0.14 mm, a fin pitch of 1.0 mm, and a vertical tube pitch of 13 mm. A photograph of the commercial dehumidifier's heat exchanger assembly is provided in Figure 1.

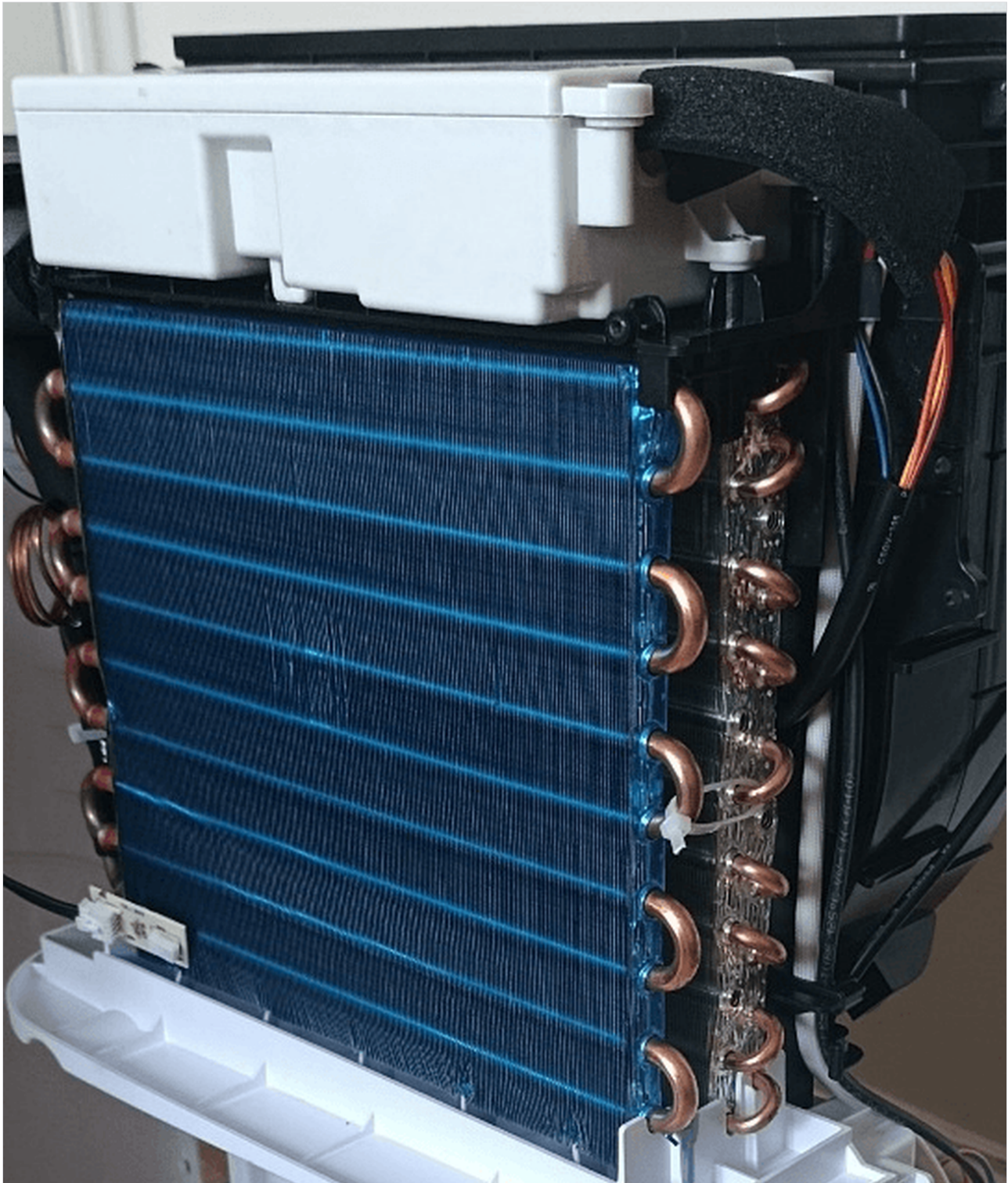


Figure 1. Test unit (condenser outer heat exchanger, evaporator inner heat exchanger).

Test 1 (Indoor - Long Duration): A 36-hour continuous run conducted between June 20 and June 22, 2025. This trial aimed to observe the long-term stability of the condensate rate, yielding a total of 11.9 Litres and total energy consumption of 10.44 kWh. **Test 2 (Outdoor - 24h):** Conducted in outdoor

climate over a 24-hour cycle to capture the impact of diurnal humidity fluctuations. This test resulted in a significant yield of 7.0 Litres with a total energy consumption of 7.2 kWh. **Test 3 (Indoor - 24h):** A controlled indoor environment test for comparison, yielding 4.5 Litres over 24 hours with a similar energy footprint of 7.2 kWh. Figure 2 shows operating conditions of the three tests.

Evaporator and condenser surface temperatures were measured using Type-T thermocouples connected to a four-channel HOBO UX120-014M thermocouple data logger (Onset Computer Corporation, USA). For Type-T thermocouple inputs, the manufacturer specifies a logger accuracy of ± 0.6 °C and a resolution of 0.02 °C. The stated accuracy represents the contribution of the logger electronics and excludes the additional uncertainty associated with the individual thermocouple probes. Ambient air temperature was recorded using the UX120-014M logger's internal 10 k Ω thermistor, which has a manufacturer-specified accuracy of ± 0.21 °C over 0–50 °C and a resolution of 0.024 °C at 25 °C. Inlet and outlet air temperature and relative humidity were measured using HOBO UX100-023A external temperature/RH data loggers (Onset Computer Corporation, USA). Their specified temperature accuracy is ± 0.21 °C over 0–50 °C, with a resolution of 0.024 °C at 25 °C. The specified relative-humidity accuracy is typically $\pm 2.5\%$ over 10–90% RH, increasing to a maximum of $\pm 3.5\%$ when hysteresis is included at 25 °C, with a resolution of 0.01% RH. All temperature and relative-humidity measurements were recorded at one-minute intervals.

The performance of the AWH unit was evaluated under three distinct psychrometric envelopes, as summarized by the inlet air temperature and relative humidity profiles. Test 1 represented a long-duration indoor trial where temperatures fluctuated between 16.1°C and 30.1°C, with humidity levels peaking at 75.9%, providing a baseline for extended stable operation. Test 2 (Outdoor) captured the characteristic diurnal cycle of a hot-arid climate with temperatures reaching up to 31.2°C and relative humidity rising to 72.1% during the cooler nocturnal hours. Relative humidity increased during the cooler nocturnal period. This increased the inlet moisture content and the potential for condensate production. In contrast, Test 3 (Indoor) maintained a higher average temperature profile (25.7°C to 31.2°C) but suffered from lower humidity levels, which frequently dropped below 40%. As noted by Patel et al. (Patel *et al.*, 2020), such variations in the psychrometric state can lead to order-of-magnitude changes in specific energy consumption. Test 2 produced 7.0 L, while Test 3 produced 4.5 L with the same electricity use. The difference is consistent with the higher moisture availability during Test 2. The comparison indicates that dew-point conditions are more informative than dry-bulb temperature alone when interpreting water production. Table 1 provides the summary of the performances.

Table 1. Test performance summary

Test	Duration (h)	Water (L)	Electricity (kWh)	SEC (kWh L ⁻¹)	WP (L kWh ⁻¹)
1 Indoor	36	11.9	10.44	0.88	1.14
2 Outdoor	24	7.0	7.2	1.03	0.972
3 Indoor	24	4.5	7.2	1.60	0.625

The measured specific energy consumption and water productivity were calculated as $SEC = E_{el}/V_w$ and $WP = V_w/E_{el}$, respectively, where E_{el} is total device electricity consumption over the test and V_w is collected water volume. SEC is reported in kWh L⁻¹ and WP in L kWh⁻¹.

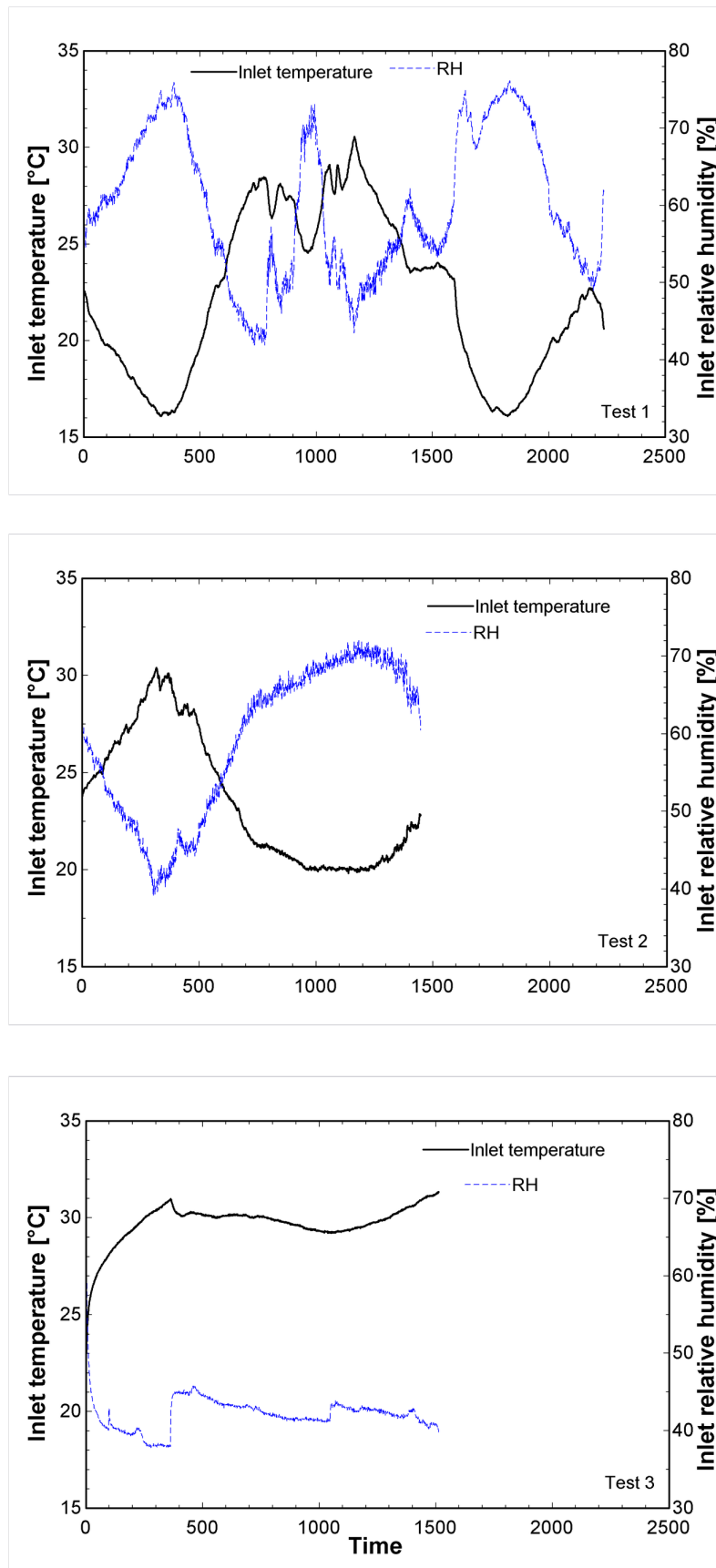


Figure 2. Measured inlet temperature and relative humidity variation of the three tests

2.2. System modelling

The system is a standard VCC system but the key component, the dehumidifying coil, is modelled using a dry-wet coil decomposition. Inlet moist-air properties are evaluated from psychrometric relations, and the dry-coil section cools the air sensibly to the inlet dew-point temperature at constant humidity ratio. The heat transfer and thermodynamic relations were modelled based on the textbooks by Nellis and Klein (Nellis and Klein 2008), (Klein and Nellis 2011). The model couples a partially wet fin-and-tube evaporator to a steady-state propane vapour-compression cycle. Figure 3 summarises the solution sequence. Inlet moist-air properties were calculated from dry-bulb temperature, relative humidity and pressure. The air mass-flow rate used in the simulations was prescribed as 0.0605 kg s^{-1} . Coil geometry and air-side conductance were then calculated. The evaporator was divided into dry and wet sections. The inner loop solved the coupling between heat flux and the refrigerant-side boiling coefficient. The outer loop adjusted the evaporation temperature using the implemented 5 W capacity criterion.

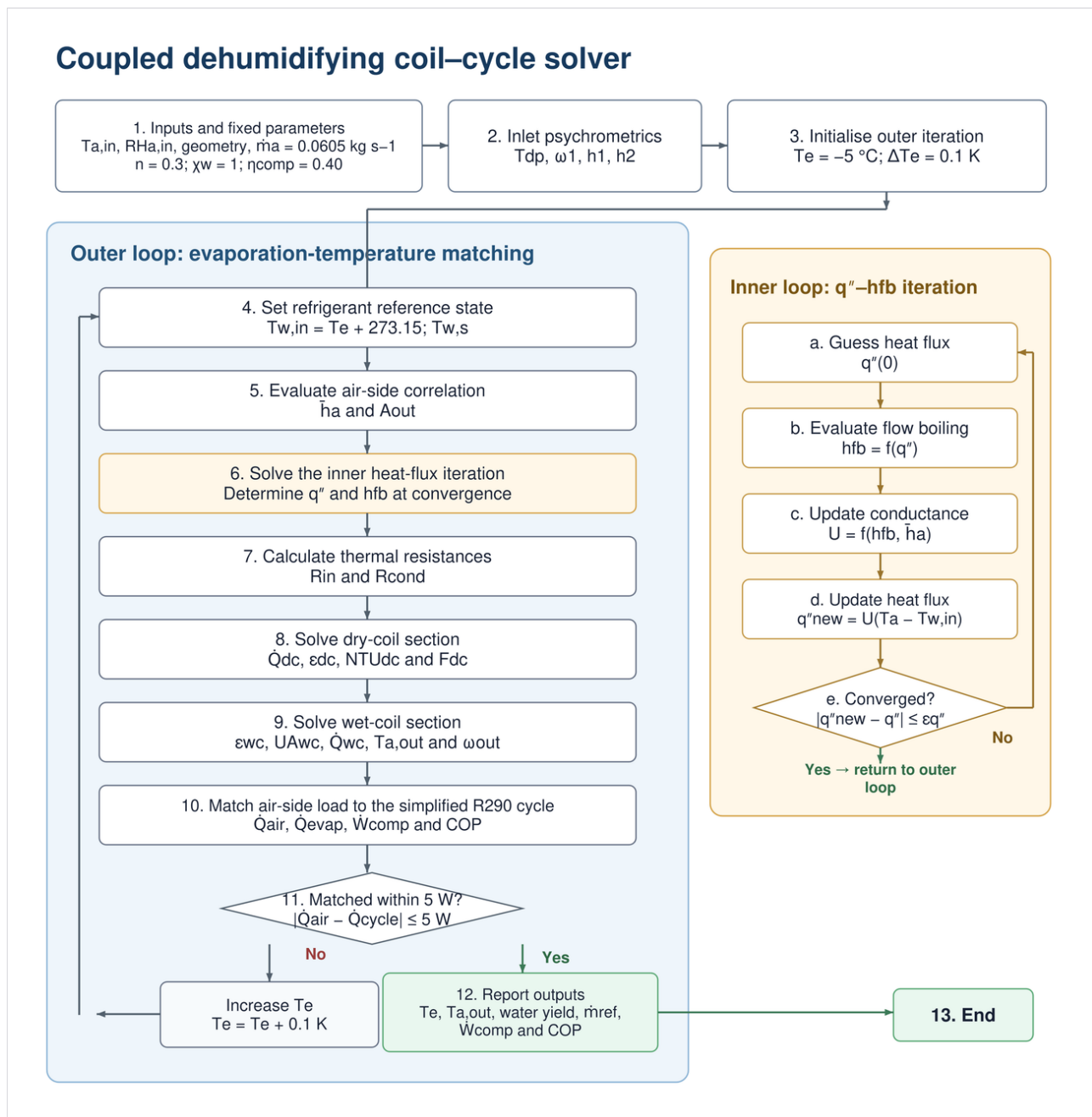


Figure 3. Flowchart of the coupled dehumidifying coil-cycle solver

The formulation assumes quasi-steady operation, negligible pressure drop on the air side, no moisture transfer across the condenser, uniform face velocity within each airflow case, negligible external pipe heat gain and thermodynamic equilibrium properties from EES software. These assumptions are considered when interpreting the validation and energy balance. The airflow comparison therefore uses relative nominal fan settings rather than calibrated absolute flow rates. Compressor-specific outputs are component-model estimates. No manufacturer compressor map was available. These values are therefore used only for within-model trend comparisons.

The solution uses two nested iterations:

1. Outer loop (evaporation temperature matching): The evaporation temperature was initialised at -5°C . If the air-side evaporator load exceeded the refrigerant-side evaporator capacity by more than 5 W, the evaporation temperature was increased by 0.1 K and the complete calculation was repeated:

$$|\dot{Q}_{\text{air}} - \dot{Q}_{\text{cycle}}| \leq 5\text{W} \quad (1)$$

2. Inner loop (implicit coupling in boiling HTC): The boiling correlation requires the heat flux q'' , while q'' depends on the overall coefficient U , which depends on the boiling HTC h_w . Therefore, q'' and h_w are solved by fixed-point iteration:

$$q'' \rightarrow h_w(q'') \rightarrow U(h_w) \rightarrow q'' \quad (2)$$

The air mass flow rate is obtained from volumetric flow rate $\dot{V}_a$ and inlet density $\rho_{a,in}$:

$$\dot{m}_a = \dot{V}_a \rho_{a,in} \quad (3)$$

The air-side heat-transfer coefficient is computed using the EES compact heat exchanger correlation (Klein 2020) :

$$\bar{h}_a = f_{HX}(\dot{m}_a, A_{face}, T_{avg}, P_a) \quad (4)$$

The EES predefined surface `fc_tubes_s80-38T` was used for the air-side heat-transfer correlation. Total external surface area (fin + unfinned tube) is:

$$A_{tot} = A_{fin} + A_{unfin} \quad (5)$$

$$A_{fin} = 2 \left(\frac{W}{p_{fin}} \right) \left(HL - N_{row} N_{col} \frac{\pi D_{out}^2}{4} \right) \quad (6)$$

$$A_{unfin} = \pi D_{out} L_{tube} \left(1 - \frac{t_{fin}}{p_{fin}} \right) \quad (7)$$

With refrigerant mass flow $\dot{m}_{ref}$ and tube inner diameter D_{in} :

$$G = \frac{\dot{m}_{ref}}{\pi D_{in}^2/4} \quad (8)$$

The average refrigerant-side flow-boiling heat-transfer coefficient was calculated using the EES average flow-boiling function for propane in a horizontal tube:

$$\bar{h}_w = f_{boil}(T_{w,in}, G, D_{in}, x_{in}, x_{out}, q'') \quad (9)$$

Implemented heat-flux iteration

An apparent coefficient was used in the implemented heat-flux iteration:

$$U = \left(\frac{1}{\bar{h}_w} + \frac{1}{\bar{h}_a} \right)^{-1} \quad (10)$$

Heat flux is defined as:

$$q'' = U(T_{avg} - T_{w,in}) \quad (11)$$

where

$$T_{avg} = \frac{T_{a,in} + T_{w,in}}{2} \quad (12)$$

Start with an initial guess $q''^{(0)}$. At iteration k :

$$h_w^{(k)} = f(q''^{(k)}) \quad (13)$$

$$U^{(k)} = \left(\frac{1}{h_w^{(k)}} + \frac{1}{\bar{h}_a} \right)^{-1} \quad (14)$$

$$q''^{(k+1)} = U^{(k)}(T_{avg} - T_{w,in}) \quad (15)$$

Convergence criterion:

$$| q''^{(k+1)} - q''^{(k)} | \leq \varepsilon_{q''} \quad (16)$$

Thermal resistances (tube side + wall)

Tube and wall sides thermal resistance using definitions excluding fouling:

$$R_{in} = \frac{1}{\pi D_{in} L_{tube} \bar{h}_w} \quad (17)$$

$$R_{cond} = \frac{\ln(D_{out}/D_{in})}{2\pi k_m L_{tube}} \quad (18)$$

Here, R_{in} is the refrigerant-side convective resistance and R_{cond} is the conductive resistance of the copper tube wall. These appear in both dry and wet coil section resistance formulations.

Dry-coil section (sensible cooling) with NTU_{dc} solved explicitly and air heat capacity rate in the dry section.

Effectiveness is defined as:

$$\varepsilon_{dc} = \frac{\dot{Q}_{dc}}{\dot{Q}_{dc,max}} \quad (19)$$

To keep NTU_{dc} as the single unknown, invert the relation:

$$NTU_{dc} = -\ln(1 - \varepsilon_{dc}) \quad (20)$$

$$UA_{dc} = NTU_{dc} C_{a,dc} \quad (21)$$

The resistance form for the dry fraction is:

$$R_{dc} = \frac{(R_{in} + R_{cond})}{F_{dc}} + \frac{1}{\bar{h}_a A_{tot} F_{dc}} \quad (22)$$

Since $R_{dc} = \frac{1}{UA_{dc}}$ rearranging Eq. (22) gives the dry-coil fraction as:

$$F_{dc} = UA_{dc} \left[R_{in} + R_{cond} + \frac{1}{\bar{h}_a A_{tot}} \right] \quad (23)$$

Wet-coil section (dehumidification)

Wet fraction:

$$F_{wc} = 1 - F_{dc} \quad (24)$$

In the implemented EES model, the reference outlet-air temperature used to evaluate the effective saturated-air specific heat was fixed at 290.5 K. The model defines a saturation effective specific heat using saturated enthalpy difference:

$$c_{p,a,sat} = \frac{h_{sat}(T_{dp}) - h_{sat}(T_{a,out,ref}^*)}{T_{dp} - T_{a,out,ref}^*} \quad (25)$$

The wet-section resistance form:

$$R_{wc} = \frac{(R_{in} + R_{cond})}{F_{wc}} + \frac{c_{p,a}}{A_{tot} \bar{h}_a c_{p,a,sat} F_{wc}} \quad (26)$$

$$R_{wc,0} = (R_{in} + R_{cond}) + \frac{c_{p,a}}{A_{tot} \bar{h}_a c_{p,a,sat}} \quad (27)$$

Then:

$$R_{wc} = \frac{R_{wc,0}}{F_{wc}} \Rightarrow UA_{wc} = \frac{1}{R_{wc}} = \frac{F_{wc}}{R_{wc,0}} \quad (28)$$

Wet-section NTU:

$$NTU_{wc} = \frac{UA_{wc}}{C_{a,wc}} \quad (29)$$

Wet-section effectiveness is computed by EES function:

$$\varepsilon_{wc} = hx(\text{crossflow_one_unmixed}, NTU_{wc}, C_{a,wc}, C_w) \quad (30)$$

Wet-section heat transfer:

$$\dot{Q}_{wc} = \varepsilon_{wc} C_{a,wc} (T_{dp} - T_{w,in}) \quad (31)$$

The simulation process flow rate is summarized in Figure 3.

3. Results and discussion

3.1. Validation of the model

Using the unit dimensions and inlet air conditions as modelling inputs, water production was simulated and compared with the experimental dataset (Figure 4). Instantaneous condensate flow was not measured directly. Instead, a time-resolved experimental estimate was reconstructed from the dry-air mass flow and the inlet-to-evaporator-exit humidity-ratio difference. The evaporator-exit humidity ratio was inferred from the downstream condenser-exit state by assuming no moisture transfer across the condenser. This estimate is sensitive to temperature, relative-humidity and airflow errors. Spatial non-uniformity and condensate drainage lag for temporal-trend comparison. Cumulative collected water is the primary validation quantity. Over the test duration, the simulated cumulative yield was 6% lower than the collected volume. Because the available records do not support formal uncertainty propagation, this 6% difference is reported as a test-specific model discrepancy rather than evidence of agreement within a quantified uncertainty interval.

An independent condenser air-side heat rate was not available. Therefore, formal experimental closure could not be quantified. The model neglects external pipe/casing heat gains, fan heat added to the airstream and transient refrigerant inventory. These omissions are a source of discrepancy and limit the interpretation of the model as a complete whole-device energy balance.

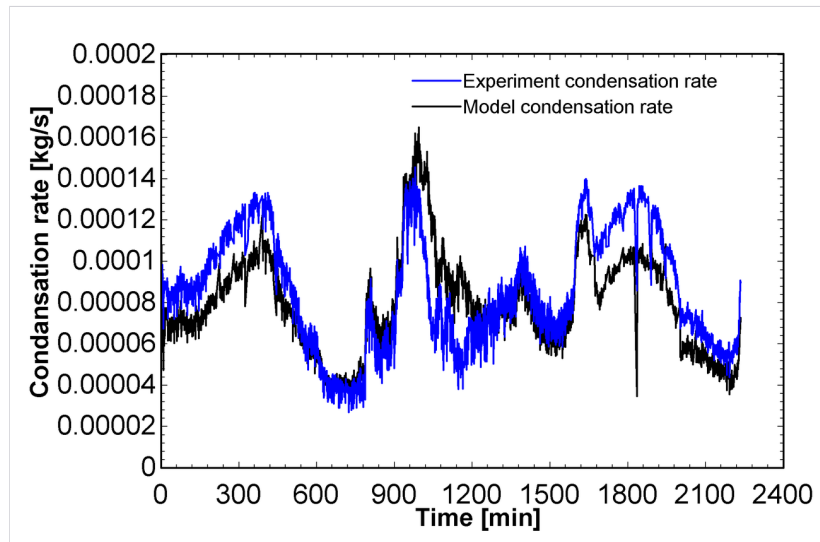


Figure 4. Comparison of condensation rates of the model and experiment

3.2. Impact of inlet conditions

In this section, the impact of varying inlet psychrometric states on the performance of the atmospheric water harvesting unit is examined through the physics-based model. These figures illustrate how ambient temperature and relative humidity dictate the required refrigeration state and the resulting water yield. Figure 5 shows how the simulated evaporating temperature varies with the inlet air temperature for three inlet relative humidity levels (40%, 50%, and 60%). The predicted evaporation temperature increases almost linearly with inlet-air temperature. This trend follows the capacity-matching behaviour of the implemented model. At a given inlet temperature, the model predicts a higher evaporation temperature at higher RH. Higher RH raises the inlet dew-point temperature and changes the air-side load. The matched evaporation temperature therefore shifts upward. At a fixed inlet air temperature, higher RH consistently results in a higher evaporating temperature. In other words, for the same dry-bulb temperature, more humid inlet air allows the evaporator to run at a higher temperature due to the constant/limited cooling capacity. The separation between the curves also becomes more noticeable at higher inlet temperatures, suggesting that humidity effects on the evaporator operating point are amplified under warmer conditions (i.e., when the air's moisture-carrying capacity and latent load are higher). Higher RH also raises the inlet dew-point temperature. This shifts the predicted evaporation temperature upward.

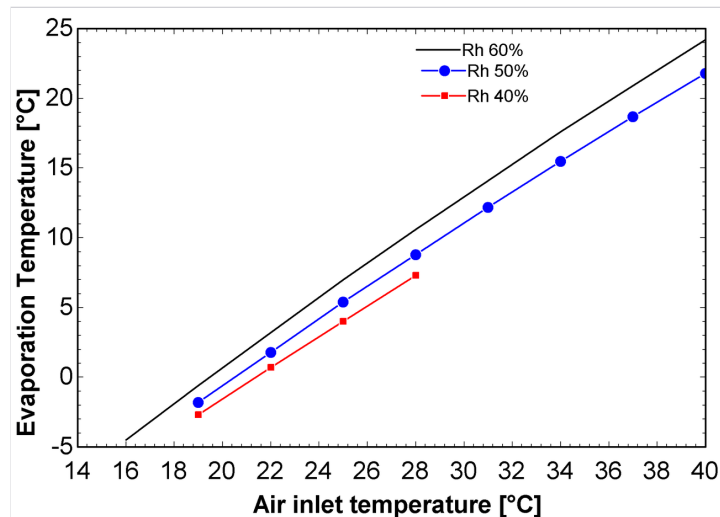


Figure 5. Impact of relative humidity and air inlet temperature of evaporation temperature

Figure 6 illustrates the combined effect of inlet air temperature and relative humidity on the water collection rate. At each inlet temperature, the predicted water-production rate increases with RH.

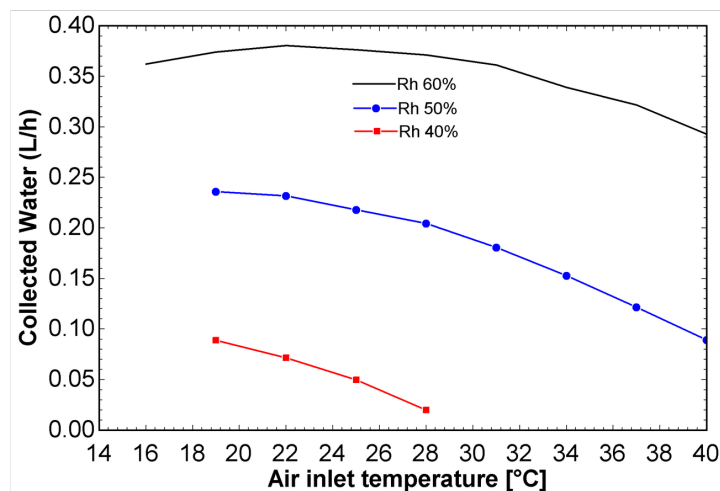


Figure 6. Impact of relative humidity and air inlet temperature on collected water

At RH 40% and RH 50%, predicted water production decreases as inlet temperature increases over the simulated range. At RH 60%, the prediction first increases slightly and reaches a shallow maximum near the lower part of the temperature range. It then decreases as inlet temperature rises further. At a given temperature, higher RH gives a higher predicted water-production rate. Figure 5 shows that the matched evaporation temperature also rises with inlet temperature. The resulting change in the dew-point-to-evaporation-temperature difference limits the fraction of inlet moisture that condenses. This effect is strongest in the lower-RH cases. The temperature effect is strongest at RH 40% and RH 50%. At RH 40%, predicted production approaches zero at the upper end of the simulated range.

Figure 7 compares how the water collection rate changes with inlet air temperature for two airflow settings. It was observed that reducing the airflow to 50% leads to substantially higher collected water across the tested temperature range, while at 100% airflow the collected water is much lower and decreases steadily as the inlet temperature increases. The higher yield at 50%

airflow is consistent with a longer air residence/contact time on the cold surface, which improves dehumidification effectiveness as air temperature can be reduced more and allows a larger fraction of the incoming moisture to condense. In contrast, at 100% airflow the shorter contact time (and potentially higher bypass factor) reduces the extent of cooling below the dew point, so a smaller fraction of water vapour is converted to liquid, and this limitation becomes more pronounced as the inlet temperature rises because warmer air can hold more moisture and requires stronger cooling to reach saturation/condensation conditions. Overall, the figure indicates that air-side control is a major lever for water harvesting, with lower airflow favouring condensation yield.

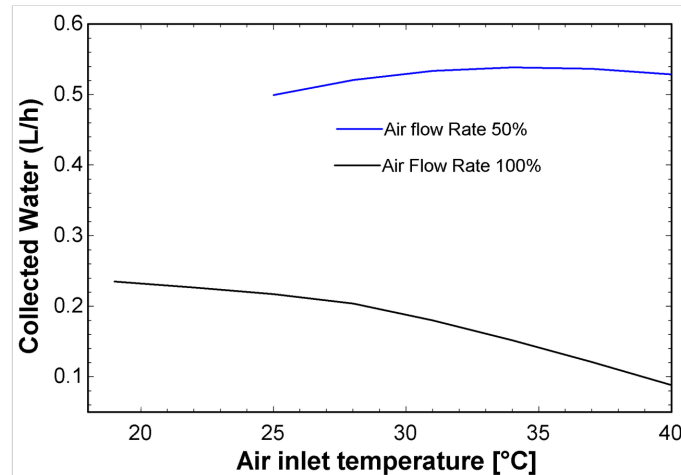


Figure 7. Comparison of collected water by changing the air flow rate

Figure 8 shows that compressor power decreases with increasing inlet air temperature for both airflow settings. At 100% airflow, compressor consumption drops roughly from 226 W to 174 W across the range, while at 50% airflow it decreases from 240 W to 188 W. The consistent downward trend suggests that, as the inlet air becomes warmer, the system operates with a higher evaporating temperature (less severe evaporator condition), which typically reduces compressor pressure ratio and therefore compressor work. A second key observation is that compressor consumption is consistently higher at 50% airflow than at 100% airflow at comparable inlet temperatures. This behaviour is consistent with reduced airflow causing weaker heat transfer on the air side, which can force the evaporator to operate at a lower saturation temperature to achieve the required cooling/dehumidification. A lower evaporating temperature increases the compressor pressure lift and specific work, raising electrical consumption even if the air mass flow is lower. Taken together with the previous figure, these results highlight the practical compromise: lower airflow improves water collection but tends to increase compressor power, so an optimal operating point should balance water yield against energy use.

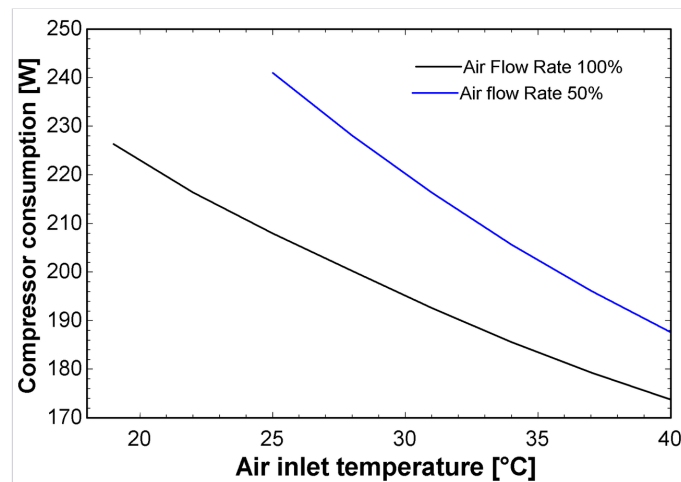


Figure 8. Comparison of compressor consumption by changing the air flow rate

The lower fan setting therefore gives more predicted water but also higher predicted compressor power. Total-device electricity use is required to identify an optimum. Based on the plotted values, the 50% airflow case gives a compressor-specific energy consumption of approximately 0.36–0.48 kWh L⁻¹ across 25–40 °C, whereas the 100% airflow case increases from approximately 0.98 to 1.9 kWh L⁻¹ as inlet temperature rises and water production falls. These values include compressor power only. Fan and auxiliary loads are excluded. They should not be compared directly with the measured total-device SEC.1. Within the model, the lower nominal fan setting gives a lower compressor-specific SEC over the analysed range. Within the model, the lower nominal fan setting gives a lower compressor-specific SEC over the analysed range. This metric excludes fan and auxiliary loads. A whole-device optimum cannot be identified from compressor power alone.

3.3. Screening-level energy-cost context

A screening energy-only water-production cost can be calculated as $C_{\text{energy}} = \text{SEC} \times p_e$, where p_e is the electricity tariff. Using the measured total-device SEC values and an illustrative tariff range of US\$0.10–0.30 kWh⁻¹ gives US\$0.103–0.309 L⁻¹ for the outdoor 24 h test and US\$0.160–0.480 L⁻¹ for the indoor 24 h test. These values exclude capital recovery, maintenance, water treatment or remineralisation, storage, component replacement and distribution. These values are not a levelised cost of water. They represent only the electricity cost associated with the measured SEC. A deployment-specific assessment also requires equipment cost, service life, maintenance, annual duty cycle and annual water yield. Climate is another important factor in AWH economics (Gayoso *et al.* 2024).

4. Conclusions

This study experimentally evaluated and modelled a compact propane vapour-compression atmospheric water generator with series-coupled evaporator and condenser airflow. The component-level dry-wet coil model reproduced cumulative water production within 6% of the measured, supporting its use for parametric analysis within the investigated range.

- The two 24 h tests demonstrate the controlling influence of psychrometric conditions. The outdoor test produced 7.0 L using 7.2 kWh (1.03 kWh L⁻¹; 0.97 L kWh⁻¹), whereas the drier indoor test produced 4.5 L using the same electricity (1.60 kWh L⁻¹; 0.625 L kWh⁻¹). The favourable outdoor profile therefore reduced SEC by 35.7%. The difference between the two tests is consistent with the higher moisture availability during the outdoor test.

- Airflow produced a coupled yield–power trade-off. Reducing airflow to 50% increased water production to approximately 0.50–0.54 L h⁻¹, although compressor power was also higher. When the two outputs were combined, the lower-airflow case provided a lower compressor-specific energy consumption over the analysed range. The optimum must nevertheless be established from total-device electricity because fan and auxiliary power were excluded from the parametric metric.
- For the representative hot-arid outdoor profile analysed in this study, AWH performance was strongly time-dependent. The highest-yield interval occurred at approximately 04:00–07:00, when the predicted yield reached 0.53 kg h⁻¹; this is a profile-specific result and not an annual or seasonal operating window. Establishing a transferable operating schedule requires simulation and validation using a full meteorological year.

Future work will include calibrated airflow measurements, direct condensate-rate measurement and independent compressor-power measurement. Longer tests under different weather conditions are also needed. These data would support a more reliable assessment of fan and compressor control strategies.

5. Nomenclature

Symbol	Definition	Unit
A	Heat-transfer area	m ²
A _{face}	Coil face area	m ²
C	Heat-capacity rate	W K ⁻¹
c _p	Specific heat capacity	J kg ⁻¹ K ⁻¹
D _{in} , D _{out}	Tube inner and outer diameters	m
E _{el}	Total measured electrical energy	kWh
F _{dc} , F _{wc}	Dry- and wet-coil fractions	—
G	Refrigerant mass flux	kg m ⁻² s ⁻¹
h	Specific enthalpy	J kg ⁻¹
$\bar{h}_a$, $\bar{h}_w$	Mean air-side and refrigerant-side heat-transfer coefficients	W m ⁻² K ⁻¹
$\dot{m}$	Mass flow rate	kg s ⁻¹
NTU	Number of transfer units	—
P _{comp}	Compressor electrical power	W
q''	Heat flux	W m ⁻²
Q	Heat-transfer rate	W
R	Thermal resistance	K W ⁻¹
SEC	Measured total-system specific energy consumption	kWh L ⁻¹
SEC _{comp}	Compressor-specific energy consumption	kWh L ⁻¹
T	Temperature	°C or K
U	Overall heat-transfer coefficient	W m ⁻² K ⁻¹
V _w	Collected water volume	L
V _a	Air volumetric flow rate	m ³ s ⁻¹
WP	Water productivity	L kWh ⁻¹
ω	Humidity ratio	kg _w kg _{da} ⁻¹
ε	Heat-exchanger effectiveness	—
η _{is}	Compressor isentropic efficiency	—

Declarations

Author contributions. Cagri Kutlu: Conceptualisation; Methodology; Modelling; Data analysis; Visualisation; Writing – Original Draft; Writing – Review & Editing. Mehmet Tahir Erdinc: Methodology; Modelling; Writing – Review & Editing. Emmanuel Tapia-Brito: Investigation (experimental work); Data curation; Writing – Review & Editing. All authors have read and approved the final version of the manuscript.

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