Fanless photovoltaic–thermal night heat rejection and greywater-fed hybrid cooling for host-sited AI computing: sub-model validation, uncertainty and a Los Angeles case study

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Highlights

- A 60 °C GPU return lets a 24-hour laundromat absorb 45–93% of the module's heat
- Bare-panel PVT night model within −11% to +2% of ISO 9806 certified coefficients
- Night fans stayed off in 17 of 18 cases at measured bare-panel PVT performance
- Cooling water is 93–100% below an evaporative tower, with no mains water at any site
- Typical laundromat roof: 67–94 kW fan-free, or 100–138 kW with a store reserve

Abstract

Evaporative cooling makes data centres large water users; dry coolers run fans near homes. This study models liquid-cooled AI modules at laundromats, with 45 °C supply and 60 °C return water, whose heat serves the host's wash water and dryer intake air, is rejected at night through a fanless photovoltaic–thermal (PVT) roof array and by day through a hybrid dry/adiabatic cooler fed with treated greywater. An hourly model for 58 and 130.5 kW modules in nine North American climates is formulated with its heat exchangers solved for the 45 °C supply, benchmarked against certified and measured data, and propagated over 5,400 sampled years. The bare-panel PVT sub-model agrees with ISO 9806 certified collector coefficients within −11% to +2% at 47.5 °C, a benchmark rather than a measurement of the proposed array. A 24-hour laundromat using off-peak pricing absorbed 93% and 45% of the two modules' heat. Relative to an evaporative tower at equal duty the system used 93–100% less water and 52–59% less cooling electricity, and drew no municipal water. With measured pad, fan and in-situ panel performance, fans stayed off between 22:00 and 07:00 in 17 of 18 site–module cases, the exception running one hour a year. In Los Angeles the system avoids 599–1,347 $m^3\ yr^{-1}$ of municipal water and, on a screening estimate, the 10–18 m night setback a dry cooler would need. A typical 2,400 ft^2 laundromat roof supports 67–94 kW of compute with fan-free nights, or 100–138 kW with the thermal store held in reserve.

Keywords: data centre cooling; photovoltaic–thermal collector; night radiative cooling; greywater reuse; water–energy nexus; noise regulation

Nomenclature

Symbol	Meaning	Unit
A	PVT array area	m^2
b_1, b_2, b_u, η_0	ISO 9806 unglazed-collector coefficients	various
E	thermal store energy	kWh
f_T, f_d	shares of wash-water and dryer heat the loop can supply	–
h_c	convective heat-transfer coefficient	W m^{-2} K^{-1}
L_{WA}, L_{pA}	A-weighted sound power and pressure level	dB(A)
q	areal heat rejection	W m^{-2}
Q_{35}	dry-coil rating with 35 °C air and 45 °C entering fluid	kW
R_h	hourly residual heat to be rejected	kWh
r	source–receiver distance	m
s	fan speed fraction	–
C_a, C_w	air and water capacity rates	kW K^{-1}
ε, NTU	heat-exchanger effectiveness, number of transfer units	–
t	PVT capacity position between measurement-anchored floor (0) and full model (1)	–
T_{db}, T_{wb}, T_{sky}	dry-bulb, wet-bulb and effective sky temperature	°C
UA	dry-coil conductance	kW K^{-1}
ε	long-wave emissivity	–
η_{pad}	adiabatic pad saturation effectiveness	–
COC	cycles of concentration	–
GHI	global horizontal irradiance	W m^{-2}
LAMC	Los Angeles Municipal Code	
PVT	photovoltaic–thermal	
RO	reverse osmosis	
CWEC	Canadian Weather year for Energy Calculation	
TMY3	typical meteorological year, version 3	
WUE	water usage effectiveness (cooling water per kWh of IT electricity)	L kWh^{-1}

1 Introduction

Data centres convert almost all of their electrical input into low-grade heat, and the way that heat leaves the site decides most of the facility's direct water use. Open evaporative towers reject heat at low electrical cost by evaporating municipal water; dry coolers avoid on-site water use but need larger coils and fans that run day and night (Shehabi et al. 2024; Mytton 2021; Siddik et al. 2021; Lei et al. 2025). Karimi et al. (2022) showed for hot-arid sites that the choice between them is a water–energy trade-off rather than a free improvement. Two further pressures now act on the same decision. Accelerator racks are moving to warm-water liquid cooling: current rack-scale GPU systems accept coolant at up to 45 °C at the inlet (Uptime Institute 2026), so heat

leaves the racks at 55–60 °C, a temperature at which it can be reused or rejected without chillers (ASHRAE 2014; Ebrahimi et al. 2014). And computing is being placed in smaller units nearer to users, where the plant sits close to homes and local water and noise rules, rather than industrial zoning, set the limits.

Three bodies of work bear on a small, distributed alternative. First, waste heat from liquid-cooled computing can serve nearby thermal demands (Wahlroos et al. 2018; Ebrahimi et al. 2014); laundromats have large low-temperature loads in both wash water and tumble-dryer intake air, and many now open 24 hours with off-peak pricing that moves customers into the night. Second, unglazed photovoltaic–thermal (PVT) collectors can reject heat at night by long-wave radiation and convection. In situ measurements have been made for building cooling, where the collector fluid is only a few kelvin above ambient: Eicker and Dalibard (2011) measured 60–65 W m^{-2} when cooling a storage tank and 40–45 W m^{-2} when cooling a ceiling in Madrid; Péan et al. (2015) measured 20–75 W m^{-2} in Denmark, with little difference between PVT and unglazed collectors; and Bilbao and Sproul (2015) recorded about 750 Wh m^{-2} per summer night in Sydney. Spectrally selective emitters can increase radiative cooling further (Raman et al. 2014; Zhai et al. 2017). Third, laundry and car-wash effluent can be treated by membrane processes to a quality suitable for non-potable reuse (Ciabatti et al. 2009; Šostar-Turk et al. 2005; Zaneti et al. 2011; Li et al. 2009), and wetted-media pre-cooling of dry coils is an established way to extend dry-cooler capacity on hot afternoons (He et al. 2015).

These strands have not been combined, and the regime that matters for heat rejection differs from that of the published night-cooling studies. Rejecting computer heat at a 45–60 °C fluid temperature places the panel 25–40 K above the night air, where convection rather than radiation dominates and where areal fluxes are several times those measured in building cooling. No study has quantified, hour by hour and across climates, how much of a computing module's residual heat such an array can carry, how much water and electricity the combination uses relative to conventional towers and dry coolers, whether a host's own greywater suffices, and what this means for night-time fan noise under a real municipal rule.

This paper addresses that gap with a simulation study that is explicit about what has and has not been measured. Its contributions are: (i) a complete hourly formulation of host heat reuse, PVT night rejection, a hybrid dry/adiabatic cooler, greywater treatment and fan noise, with all parameters stated (Section 2); (ii) tests of each sub-model against measured or certified third-party data, which identify the PVT enhancement mechanisms as the one unvalidated component (Section 3); (iii) a measurement-anchored scenario and a Latin-hypercube uncertainty analysis,

from which the conclusions are stated with the conditions on which they depend, including a break-even PVT flux that a single-panel test can confirm or refute (Section 4); and (iv) a Los Angeles case study that converts the results into consequences for the municipal water supply and for compliance with the city's noise ordinance, with the host-site requirements expressed as design variables (Section 5). Section 6 positions the findings against the state of the art.

2 System description and model formulation

2.1 System and dispatch

A liquid-cooled accelerator module captures a fraction $c = 0.85$ of its electrical input in a water loop that supplies the cold plates at 45 °C and returns at 60 °C (Fig. 1). Captured heat first serves the host's temperature-compatible demand for wash water and dryer intake air, then charges a 110 kWh thermal store (3.8 m^3 over a 25 K swing); the remainder is rejected, and every rejection device must return the water at 45 °C or below. At night, defined as global horizontal irradiance (GHI) below 5 W m^{-2}, the dispatch uses, in order, the passive PVT array, the array's recirculating evaporative film, the hybrid cooler's dry coil and finally its wetted adiabatic pads. By day the PVT array generates electricity and can preheat wash water, and the hybrid cooler carries the residual. A quiet-hours variant keeps the PVT array on duty in hours between 22:00 and 07:00 that are already light (summer dawn), so that the cooler's fans need not start inside the regulatory night; it is the central dispatch in this paper. Heat that no device can reject is recorded as unmet, i.e. as a compute power cap.

Store reserve option. Where the module's heat exceeds the host's demand in most hours, as at a laundromat that closes at night, the store in the standard dispatch is full when the night begins and cannot help in the hours the array falls short. In the store-reserve dispatch, from 15:00 the host is served from the store first and the store takes no routine surplus, so it enters the night nearly empty. Between 22:00 and 07:00 it absorbs only the heat the array cannot reject, before any fan starts, and it is drained into spare passive array capacity earlier in the night. The cost is that the heat the host would have taken directly in the evening goes to the hybrid cooler instead, so the option trades heat reuse, daytime fan energy and daytime noise for silent nights; Section 4.4 quantifies the trade.

Wash effluent is captured (fraction 0.9), pretreated, desalted by reverse osmosis (RO, recovery 0.60, capacity 0.3 $m^3\ h^{-1}$) and disinfected by ultraviolet light before a 2 m^3 unheated buffer that feeds the film and pads; RO concentrate goes to the sewer and any deficit is drawn from the municipal supply. When the pads have operated in a week, the pad circuit receives a 0.2 m^3

thermal flush at 70 °C. This flush is a modelling assumption, not evidence of microbiological control: a water-management plan with Legionella control and aerosol assessment is a precondition of any installation (Bartram et al. 2007).

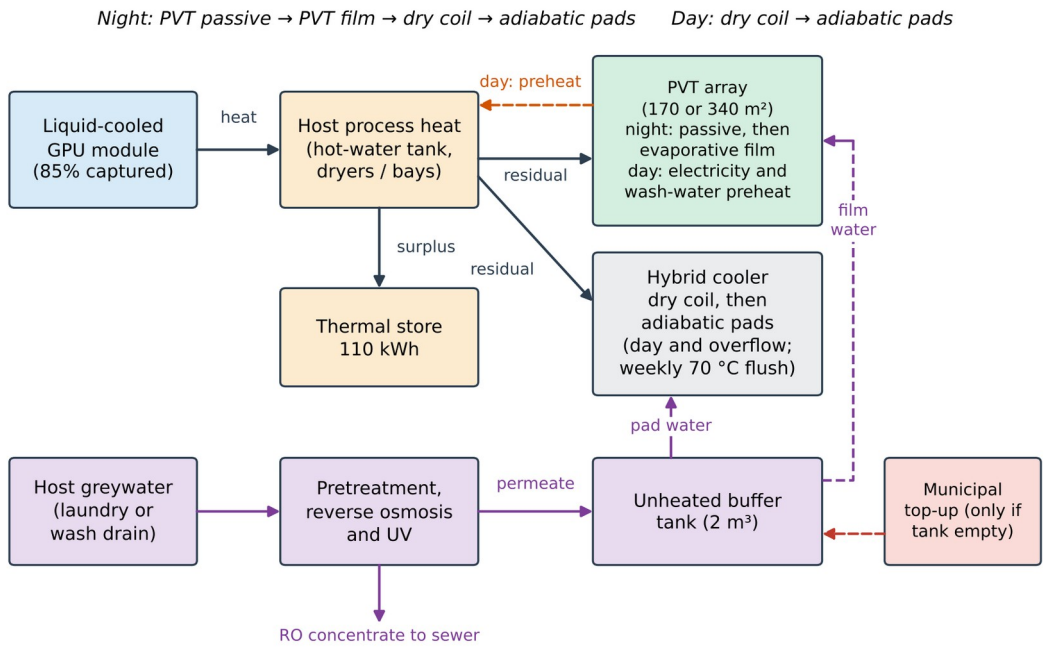

Fig. 1. System arrangement and dispatch order. Water leaves the GPU cold plates at 60 °C and is returned to them at 45 °C or below. Dark arrows carry heat from the computing module, the orange dashed arrow daytime PVT preheat of wash water and purple arrows treated water; RO concentrate goes to the sewer and municipal water is drawn only when the buffer is empty.

2.2 Climates, modules and host

Hourly weather is taken from typical-year files for nine North American airports chosen to span hot-dry (Phoenix, Las Vegas, Fresno), warm coastal (Los Angeles), marine (San Francisco, Vancouver), hot-humid (Miami), cold-humid (Chicago) and very cold continental (Winnipeg) conditions: TMY3 for the seven US sites (Wilcox and Marion 2008) and CWEC 2020 for Vancouver International and Winnipeg Richardson International (Environment and Climate Change Canada 2020), both in EnergyPlus weather format. Vancouver (49.2° N, 4C marine) and Winnipeg (49.9° N, climate zone 7, design heating temperature −32 °C) extend the set to the latitudes at which the winter sun is lowest and nights longest. Case A is a 58.0 kW module (32 accelerator-equivalents of 1.8125 kW) with 100 panels (170 m^2) and a 30 kW hybrid coil; Case B is a 130.5 kW module with 200 panels (340 m^2) and a 40 kW coil. Both run at a flat 80% of rated power. The host is a 24-hour laundromat. Its base profile is 110 wash loads on weekdays and 160 at weekends between 07:00 and 22:00; off-peak (time-of-day) pricing, which operators use to build late-night trade (Planet Laundry n.d.; American Coin-Op n.d.), is represented by moving 25% of loads into 22:00–07:00 and raising volume by 15%. Each load needs 4.96 kWh

of water heating, of which a 60 °C supply can meet $f_T = 0.85$ (mains water at 15 °C raised to 55 °C of a 60 °C set point), and 5.0 kWh of dryer heat, about the 0.17 therm a commercial gas dryer burns per cycle (Multi-housing Laundry Association n.d.), drawn one hour after the wash. Preheating the dryer intake air from about 25 °C to 55 °C, against a supply-air temperature near 85 °C, gives $f_d = 0.50$; air-to-air inlet preheating of dryers is an established measure that saves 20–26% of drying energy even at lower preheat temperatures (Lawrence Berkeley Laboratory n.d.). Each load yields 60 L of greywater. The 07:00–22:00 laundromat without dryer heat used in an earlier version of this work, car-wash and commercial-laundry hosts are reported in Online Resource 1, and the effect of opening hours, off-peak pricing and dryer heat is quantified in Section 4.4. Whether a host roof can carry these arrays is examined in Section 5.4. Table 1 lists the parameters, their central values and the ranges used in the uncertainty analysis.

Table 1. Model parameters: central values, measurement-anchored values (Section 3) and ranges sampled in the uncertainty analysis (Section 4).

Parameter	Central	Anchored	Sampled range	Basis
IT utilisation	0.80	0.80	0.60–0.95	assumption
Liquid capture fraction c	0.85	0.85	0.80–0.90	ASHRAE (2014)
Wash-water heat share f_T	0.85	0.85	0.60–1.00	Section 2.2
Dryer heat per load (kWh) / share f_d	5.0 / 0.50	5.0 / 0.50	– / 0.30–0.60	MLA (n.d.)
Share of loads 22:00–07:00 / volume factor	0.25 / 1.15	0.25 / 1.15	0–0.35 / 0.85–1.30	assumption
Loop supply / return (°C)	45 / 60	45 / 60	fixed (45 °C return in Section 4.4)	Uptime Institute (2026)
PVT surface temperature (°C)	47.5	47.5	fixed	mean fluid less 5 K
PVT capacity position t	1 (full model)	0 (bare × 0.74)	0–1	Section 3.1
Pad effectiveness η_{pad}	0.85	0.75	0.65–0.85	Evapco (n.d.)
Fan power at full speed (kW per kW of Q_{35})	0.030	0.053	0.030–0.053	ICS Cool Energy (n.d.)
Coil air rise / fluid range at rating (K)	6 / 5	6 / 5	5–8 / 3–7	Section 3.2
Coil part-speed conductance exponent	0.6	0.6	0.5–0.8	Section 2.5
Greywater per load (L)	60	60	15–60	assumption
RO recovery	0.60	0.60	0.50–0.75	Ciabatti et al. (2009)
Tower COC	3	3	2–6	Hensley (2009)
Tower fan power (kW per kW rejected)	0.008	0.008	0.004–0.016	Hensley (2009)
Reference sound power offset (dB)	0	0	−3 to +3	datasheet tolerance
Fan-law exponent k (dB per decade)	50	50	40–60	ASHRAE (2019)
Pad noise increment (dB)	2	2	0–4	assumption

2.3 Heat allocation and storage

With Q_c the captured heat, W_h the host's hourly wash-water heat demand and Y_h its dryer heat demand, the compatible demand is $D_h = f_T W_h + f_d Y_{h-1}$, and the heat used directly, the store balance and the residual to be rejected in hour h are

$$U_h = \min(Q_c, D_h) + \min(E_h, D_h - \min(Q_c, D_h)) \quad (1)$$

$$R_h = Q_c - \min(Q_c, D_h) - \Delta E_h^{+}, \; 0 \le E_h \le E_{max} \quad (2)$$

where ΔE_h^{+} is the store charge in that hour. Solar preheat from the PVT array by day serves only the compatible demand that remains after waste heat and the store, so solar heat never displaces waste heat that would then need rejecting; its useful heat follows the certified ISO 9806 unglazed-collector equation (Eq. 8) with the coefficients of a commercial unglazed PVT collector (ICC-SRCC n.d.).

2.4 PVT night rejection

Each night hour the array's capacity is obtained from a quasi-steady surface energy balance at a fixed surface temperature T_s = 47.5 °C, the mean of the 60 °C return and 45 °C supply less 5 K:

$$q = u\,[\, h_c(T_s - T_{db}) + \varepsilon\sigma(T_s^4 - T_{sky}^4) + w\,d\,q_{evap}\,] \quad (3)$$

The effective sky temperature follows from the Berdahl–Martin clear-sky emissivity, corrected towards overcast with the hourly opaque cloud cover (Berdahl and Martin 1984). The convective coefficient combines a Churchill–Chu natural-convection term for an inclined plate (Churchill and Chu 1975) with laminar or turbulent flat-plate forced-convection terms (Incropera et al. 2011) for the wind and, where fitted, the stack-induced flow, as $h_c = (h_{nc}^2 + \Sigma h_{f,i}^2)^{1/2}$. Evaporation from the film uses the Chilton–Colburn analogy, $q_{evap} = h_c(c_p \mathrm{Le}^{2/3})^{-1}(w_s - w_a)h_{fg}$, applied to a wetted fraction w = 0.7 for a duty fraction d chosen hourly as the smallest that meets the residual; the film is locked out below 2 °C. The surface utilisation u is 0.75 for a bare panel. The full model adds four mechanisms: a skirted sub-panel plenum beneath the array, which forms a single continuous sloped plane with the base of a 5 m draft stack at its highest edge (stack velocity $v = [2gH\Delta T/(TK)]^{1/2}$), edge shrouds that raise the local wind speed by a factor of 1.6, a high-emissivity backsheet emitter (ε = 0.95 with an additional sky depression of up to 12 K under clear sky) and a two-phase thermosiphon that raises u to 0.97. In the quiet-hours dispatch, capacity in light hours before 07:00 is reduced by the absorbed solar flux not converted to electricity, 0.70 of plane-of-array irradiance.

Night misting option. An overhead mist array (50 nozzles per 100 panels, 4.5 L h^{-1} each, Sauter mean diameter 45 µm, 1.2 m above the array) can be added as a final wetting stage. The dispatch uses it only in hours that the film at full duty cannot meet, interpolating its duty linearly up to full spray. Droplet evaporation in flight follows the d^2 law for an isolated sphere (Sherwood number 2, Spalding mass-transfer number B_M) over the fall time, giving a kinetic evaporated

fraction f_{kin}. The water evaporated in flight is further capped by the moisture capacity of the air renewed through the spray layer, and it pre-cools the air that then crosses the array:

$$\dot{m}_{fl} = \min[f_{kin}\dot{m}_{mist},\ \dot{m}_s(w_s(T_{wb}) - w_a)], \quad \dot{m}_s = \rho u_{ex} H_s W_p \tag{3a}$$

$$\Delta T_{mist} = \min[\dot{m}_{fl} h_{fg}/(\dot{m}_s c_p),\ T_{db} - T_{wb}] \tag{3b}$$

Here $u_{ex} = \max(u_w, 0.3\ \mathrm{m\ s^{-1}})$ is the exchange velocity, H_s the spray height and W_p the plan width of the array. Eq. 3 is then evaluated with $T_{db} - \Delta T_{mist}$ and the humidified air. Of the spray reaching the array, 82% is intercepted, which raises the wetted fraction to $\min(\dot{m}_{surf}/\dot{m}_{dem}, 0.92)$ compared with 0.70 for the film; 90% of runoff is recovered, and the remaining runoff and the 18% carry-over are counted as water use. The mist loop runs at 6 cycles of concentration.

2.5 Hybrid dry/adiabatic cooler

The rejection devices sit in parallel branches of the 60 °C return, and each branch must return its water at no more than the 45 °C supply temperature, so that the mixed supply to the GPUs is at or below 45 °C. The host heat exchangers meet this with counter-flow approaches of 5 K at the hot end (60 °C loop to 55 °C wash water; 60 °C loop to dryer inlet air heated from 25 to 55 °C). The dry coil is therefore solved as a heat exchanger with both outlet temperatures determined, not as a fixed conductance times the entering temperature difference. It is specified by a catalogue rating Q_{35}, the duty at 35 °C entering air and 45 °C entering fluid with a fluid range $\Delta T_{w,r} = 5$ K and an air rise $\Delta T_{air} = 6$ K at full fan speed. The rating fixes the full-speed air capacity rate $C_a = Q_{35}/\Delta T_{air}$ and, through the effectiveness $\varepsilon = Q_{35}/[C_{min}(45 - 35)]$ and the cross-flow (both fluids unmixed) relation (Incropera et al. 2011), the physical conductance UA:

$$\varepsilon = 1 - \exp\{(1/C_r)\ NTU^{0.22}\ [\exp(-C_r\ NTU^{0.78}) - 1]\} \tag{4}$$

with $NTU = UA/C_{min}$ and $C_r = C_{min}/C_{max}$. In operation the water flow in the coil branch is set by its duty, $C_w = Q/(T_{hot} - T_{sup})$ with $T_{hot} = 60$ °C and $T_{sup} = 45$ °C, so a duty Q is feasible at fan fraction s only if

$$\varepsilon(UA\ s^{0.6},\ C_w,\ C_a s) \cdot C_{min}\ (T_{hot} - T_{a,in}) \geq Q \tag{5}$$

where the conductance follows the air flow with exponent 0.6 (air-side resistance dominant). The coil capacity $Q_{max}(T_{a,in}, s)$ is the largest Q satisfying Eq. 5; it falls to zero as the entering air approaches 45 °C. In each hour the fan runs at the smallest speed meeting the residual (minimum 20%), and fan power follows the cube law, $P_{fan} = P_{full} s^3$. When the coil cannot meet the residual at full speed, the pads cool the entering air by just enough, up to their saturation limit, and the water evaporated is the sensible cooling of the air stream divided by the latent heat:

$$Q_{max}(T_{db} - \Delta T_{pre},\ 1) = R_h, \quad \Delta T_{pre} \le \eta_{pad}(T_{db} - T_{wb}) \tag{6}$$

$$m_{evap} = C_a \Delta T_{pre} / h_{fg} \tag{7}$$

Pad bleed is $m_{evap}/(\mathrm{COC} - 1)$, with 10 cycles on RO permeate and 3 on municipal water. Heat remaining after the pads saturate is unmet.

2.6 Water treatment and electricity

Permeate in hour h is $P_h = \min(0.9\, r\, G_h,\ 0.3,\ V_{max} - V_h)$, where G_h is the host's greywater and r the RO recovery. Demand W_h (film make-up, pad evaporation and bleed, flushes) is drawn from the buffer, and only the shortfall $M_h = \max(W_h - V_h - P_h,\ 0)$ is municipal. Cooling-system electricity comprises fans, loop pumping at 0.008 kW per kW of heat moved, RO and pretreatment at 2.0 kWh m^{-3}, UV at 0.05 kWh m^{-3}, film distribution at 0.15 kWh m^{-3}, and an electric booster that lifts the flush water from 55 °C (loop preheat) to 70 °C. When misting is enabled, the mist-loop make-up is also ozonated in the supply header at an applied dose of 2 g m^{-3} by an air-fed corona-discharge generator using 16.5 Wh g^{-1} including feed-gas drying (ProMinent n.d.); the ultraviolet stage and weekly thermal flush are retained. The daytime electricity generated by the PVT array is accounted for separately. For daytime preheat the certified collector equation is

$$q_u = \eta_0(1 - b_u u_w)G'' - (b_1 + b_2 u_w)(T_m - T_{db}), \quad G'' = G + (E_L - \sigma T_{db}^4) \tag{8}$$

with $\eta_0 = 0.337$, $b_u = 0.0755$ s m^{-1}, $b_1 = 10.1$ W m^{-2} K^{-1} and $b_2 = 0.24$ J m^{-3} K^{-1} (ICC-SRCC n.d.), wind speed u_w and long-wave irradiance E_L.

2.7 Reference plants

Two reference plants reject the same hourly residual R_h (equal duty), which isolates the rejection technology from heat reuse. The open evaporative tower receives 60 °C water with a 15 K range; exit air is saturated at a temperature 60% of the way from the ambient wet-bulb to the mean water temperature, and the latent fraction of the heat follows from the psychrometric change, $f_L = \Delta w\, h_{fg,0}/\Delta h$. Blowdown is $m_{evap}/(\mathrm{COC} - 1)$ less drift (0.005% of circulation), and electricity is 0.020 kW per kW rejected for fans, spray lift and pumping (Hensley 2009). The tower is also applied to all captured heat (whole-module duty) to show the combined effect of reuse and rejection technology. The dry-cooler reference has no pads, PVT or water; it is sized with the same coil specification and outlet constraint (Eqs. 4–5) to reject the largest hourly residual with 45 °C water leaving at the site's 0.4% design dry-bulb temperature, and uses the same fan coefficient as the proposed system.

2.8 Fan noise

Sound power is scaled from a datasheet dry cooler (62.3 dB(A) at 10 m, i.e. $L_{WA,ref}$ = 90.3 dB(A) for hemispherical radiation, at UA_{ref} = 16.7 kW K^{-1}; ICS Cool Energy n.d.) by coil size and fan speed, and propagated hemispherically without ground, barrier or air absorption (ISO 2024):

$$L_{WA} = L_{WA,ref} + 10\log_{10}(UA/UA_{ref}) + k\log_{10}s + \Delta L_{pad}, \quad L_{pA} = L_{WA} - 20\log_{10}r - 8 \qquad (9)$$

with $s = (P_{fan}/P_{full})^{1/3}$, k = 50 dB per decade (ASHRAE 2019) and ΔL_{pad} = 2 dB while the pads are wetted. Hourly levels are energy-averaged into day (07:00–22:00) and night (22:00–07:00) equivalent continuous levels, with fan-off hours silent. Pumps, film water, spectra, directivity, barriers and tonal penalties are not modelled; the calculation is a source–receiver screen, not a compliance assessment.

2.9 Implementation

All calculations are deterministic Python code run on the nine weather files, supplied with every script needed to regenerate each table and figure (Online Resource 3). Model development, analysis and drafting were assisted by generative AI tools as stated in the declaration at the end of the paper; the author checked the code, results and text and is accountable for them.

3 Sub-model validation

No instrumented installation of the combined system exists, so each sub-model is tested separately against measured or certified third-party data. Where the comparison shows a bias, the measured value is carried into a measurement-anchored scenario (Table 1) rather than left as a caveat. Table 2 summarises the outcome.

3.1 PVT array

Operating regime. The relevant reference for heat rejection at 47.5 °C is the ISO 9806 thermal performance test, in which the collector's heat loss is measured over a range of fluid temperatures and wind speeds and summarised by certified coefficients. Evaluating Eq. 8 at night (G = 0) with the certified coefficients of a commercial unglazed PVT collector (ICC-SRCC n.d.) gives the measured-data prediction of night heat loss for a mean fluid temperature T_m. Over all 38,942 night hours of the nine weather files, with wind reduced to collector height (0.67 u_{10}), the bare-panel sub-model at a 47.5 °C surface gave median fluxes of 253–529 W m^{-2} across the nine sites, against 264–529 W m^{-2} from the certified model at T_m = 47.5 °C; the per-site median ratio was 0.89–1.02 (Fig. 2a). Against T_m = 52.5 °C, the mean fluid temperature, the ratio was 0.78–0.88. The bare-panel sub-model is therefore consistent with, and slightly conservative relative to, certified measurements in the regime in which the system operates.

Along-flow coupling. The array is evaluated at a uniform 47.5 °C surface. Solving instead for the fluid cooling from 60 to 45 °C along the absorber, with the surface below the local fluid temperature by the local flux divided by a fluid-to-surface conductance U_{fs}, gave an area-mean bare-panel flux of 0.99–1.07 times the lumped value for U_{fs} = 100 W m^{-2} K^{-1}, 1.04–1.12 for 200 and 0.90–0.98 for 50 W m^{-2} K^{-1} (median over a sample of night hours; 0.77–0.87 for the full model at 50). The lumped evaluation is therefore close to neutral for a well-bonded absorber and overstates capacity by up to 10% for a poorly bonded one; the break-even capacities of Section 4.3 can be read against this range.

Low-temperature regime. At the conditions of the in situ Madrid measurements (fluid 3–8 K above 16–22 °C air), the bare-panel sub-model gave a median of 95 W m^{-2} without, and 85 W m^{-2} with, a coolant-to-surface resistance, and the certified model gave 80 W m^{-2}, against 60–65 W m^{-2} measured when cooling a tank (Eicker and Dalibard 2011) (Fig. 2b). Both the sub-model and the certified test model thus exceed the in situ measurement by a similar margin, 25–50%, which is consistent with flow maldistribution, edge losses and conditions in a building installation that a laboratory test does not reproduce. The ratio 1/1.35 = 0.74 is therefore applied to the bare-panel capacity in the measurement-anchored scenario.

Enhancement mechanisms. The plenum stack, shrouds, emitter and thermosiphon together raise the modelled median capacity to 1.69–2.02 times the bare panel. None of them has been measured, alone or in combination, and no test is available against which to validate them; the same applies to the optional night mist. Rather than assert their performance, Section 4.3 treats the capacity between the measurement-anchored floor (t = 0: bare panel × 0.74) and the full model (t = 1) as an explicit uncertainty and derives the capacity at which the noise conclusion holds.

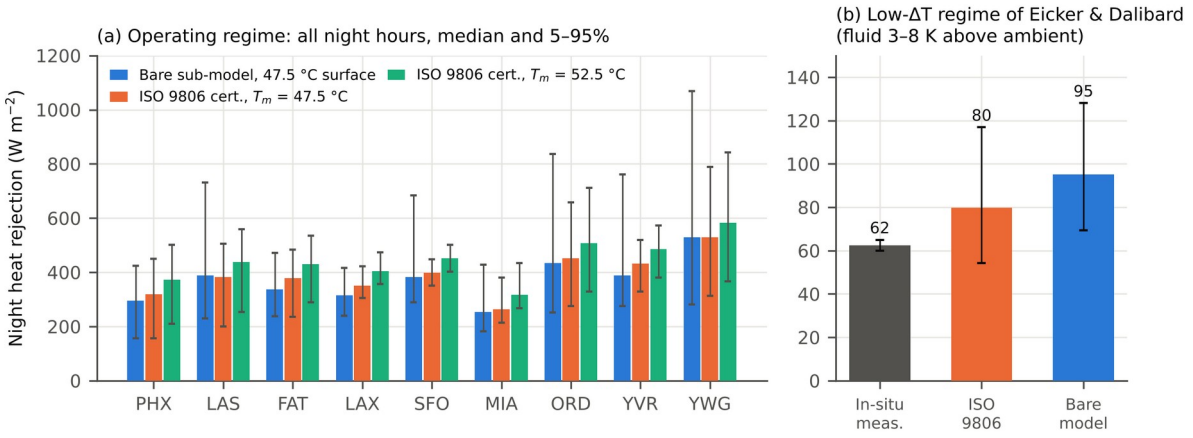

Fig. 2. PVT sub-model validation. (a) Night heat rejection at the operating point: bare-panel sub-model at a 47.5 °C surface and ISO 9806 certified collector model (ICC-SRCC n.d.) at mean fluid temperatures of 47.5 and 52.5 °C,

over every night hour of the weather file at each site (bars: median; whiskers: 5th–95th percentiles). (b) At the low fluid-to-air temperature difference of the in situ measurements of Eicker and Dalibard (2011): measured tank-cooling flux, certified model and bare-panel sub-model over the same envelope of Madrid summer-night conditions (whiskers: range for measurement, 5th–95th percentiles for models).

3.2 Hybrid cooler

Pad effectiveness. Full-scale chamber tests of adiabatic fluid-cooler pads under ASHRAE and CTI procedures found maximum saturation effectiveness of 78% at 2.0 m s^{-1} falling to 72% at 3.6 m s^{-1}, with most commercial pads lower and none reaching the 80–95% claimed in supplier data (Evapco n.d.). The central value of 0.85 is therefore optimistic; 0.75 is used in the anchored scenario and 0.65–0.85 is sampled.

Fan power. The reference dry cooler draws 8.9 kW at 250 kW with a 15 K entering temperature difference, i.e. 0.53 kW per kW K^{-1} of conductance, against 0.30 kW per kW K^{-1} implied by the central coefficient (ICS Cool Energy n.d.). The datasheet value is used in the anchored scenario for both the proposed system and the dry-cooler reference.

Air flow and rating. An air rise equal to the 10 K entering temperature difference at the rating would require an air-side effectiveness of one; the earlier formulation, with a 10 K rise and capacity proportional to the entering temperature difference, was therefore not physically consistent. Catalogue dry coolers are rated with fluid ranges of about 5 K and air rises of 5–8 K, so $\Delta T_{w,r}$ = 5 K and ΔT_{air} = 6 K are used, and 3–7 K, 5–8 K and a part-speed exponent of 0.5–0.8 are sampled.

Outlet temperature. Because Eq. 5 requires the coil water to leave at 45 °C, its capacity falls faster with air temperature than a fixed conductance implies: for the 30 kW coil it is 90 kW at 30 °C air, as before, but 48 kW at 40 °C and 25 kW at 44 °C, against 60 and 48 kW from the fixed-conductance form; with the pads at 0.75 effectiveness and a 22 °C wet bulb it remains above 101 kW at 44 °C. Over the year the coupled model raised pad hours and water only at the hottest sites (Phoenix, Case B: 30 pad hours and 1.8 m^3, against 0 h and 0.2 m^3), and no heat went unmet at any site, in either case, for any coil specification sampled (Online Resource 1). Every result in this paper uses the coupled model.

3.3 Evaporative tower and acoustics

The tower model's annual mean latent fraction was 0.67–0.85 across sites (lowest in Winnipeg), below the near-unity fraction implied by the common rule that evaporation is about 1% of circulation for each 5.6 K of range (Hensley 2009), as expected where sensible transfer is significant. Its whole-module water use, 1.29–1.60 L per kWh of IT electricity, lies just below

the US average on-site value of about 1.8 L kWh^{-1} reported by Mytton (2021), which also includes less efficient chilled-water plants. The acoustic model is calibrated to a single datasheet level, which is not an independent validation; its uncertainty is represented by sampling a ±3 dB offset, the fan-law exponent and the pad increment.

Table 2. Validation summary: reference data, outcome and treatment in the analysis.

Sub-model	Reference data	Outcome	Treatment
PVT bare panel, 47.5 °C	ISO 9806 certified heat-loss coefficients (ICC-SRCC n.d.), 38,942 night hours	Median ratio 0.89–1.02 (T_m 47.5 °C); 0.78–0.88 (52.5 °C)	Accepted
PVT bare panel, low ΔT	In situ, Madrid (Eicker and Dalibard 2011)	Model and certified test both 25–50% above measurement	In-situ derate 0.74 (anchored)
PVT enhancements	None available	Not validated (1.69–2.02 × bare)	Uncertain $t \in [0, 1]$; break-even
Night mist (option)	None for this geometry	Not validated	Reported as an option (Section 4.4)
Pad effectiveness	Full-scale tests (Evapco n.d.)	Measured ≤ 0.78; model 0.85	0.75 anchored; 0.65–0.85 sampled
Coil fan power	Datasheet (ICS Cool Energy n.d.)	0.53 vs 0.30 kW per kW K^{-1}	Datasheet anchored; sampled
Coil air flow and outlet	Energy balance at rating; ε–NTU with 45 °C outlet	Earlier fixed-conductance form inconsistent	Coupled model throughout; rating range, air rise and part-speed exponent sampled
PVT along-flow coupling	Fluid 60 → 45 °C with fluid-to-surface conductance 50–200 W m^{-2} K^{-1}	Lumped 47.5 °C surface within −8% to +12% (bare)	Covered by the 0.74 anchor and break-even
Evaporative tower	Hensley (2009); Mytton (2021)	f_L 0.67–0.85; WUE 1.29–1.60 L kWh^{-1}	Accepted; COC and fan sampled
Fan noise	Datasheet level (calibration only)	Not independently validated	±3 dB, k, pad increment sampled

4 Results and uncertainty analysis

4.1 Heat allocation, water and electricity

The 24-hour laundromat absorbed 320.9 MWh yr^{-1} of the 345.5 MWh yr^{-1} captured from the 58 kW module (93%) and 350.2 MWh yr^{-1} of the 777.3 MWh yr^{-1} captured from the 130.5 kW module (45%), leaving 24.6 and 427.1 MWh yr^{-1} to reject at every site. Heat demand, not climate, sets this split; Case A is close to the point at which the host absorbs everything. In the central model the PVT array carried 64–73% (Case A) and 59–63% (Case B) of the rejected heat passively, and the dry coil carried the daytime residual with almost no pad operation. Table 3 compares the central and measurement-anchored scenarios with the tower at equal duty, and Fig. 3 shows water and electricity per kWh of IT electricity.

Table 3. Annual results: central model / measurement-anchored scenario. Water and electricity reductions are relative to an open evaporative tower rejecting the same residual heat.

Site, case	Tower water (m^3)	Proposed, central (m^3)	Proposed, anchored (m^3)	Water reduction (%)	Electricity reduction (%)	Fan h, 22–07 h	Municipal, anchored (m^3)
Phoenix, A	44	0.0	0.5	100.0 / 98.9	58.4 / 57.2	0 / 0	0.0
Phoenix, B	796	1.8	41.9	99.8 / 94.7	55.9 / 52.1	0 / 0	0.0
Las Vegas, A	43	0.0	1.2	100.0 / 97.3	58.3 / 56.8	0 / 0	0.0
Las Vegas, B	773	0.7	22.3	99.9 / 97.1	57.5 / 55.2	0 / 1	0.0
Fresno, A	42	0.0	0.1	100.0 / 99.8	58.3 / 57.3	0 / 0	0.0
Fresno, B	746	0.0	11.9	100.0 / 98.4	58.7 / 57.4	0 / 0	0.0
Los Angeles, A	42	0.0	0.1	100.0 / 99.8	58.3 / 57.3	0 / 0	0.0
Los Angeles, B	737	0.0	21.7	100.0 / 97.1	59.4 / 58.4	0 / 0	0.0
San Francisco, A	41	0.0	0.0	100.0 / 100.0	58.3 / 57.3	0 / 0	0.0
San Francisco, B	718	0.0	3.2	100.0 / 99.5	59.4 / 58.9	0 / 0	0.0
Miami, A	44	0.0	0.3	100.0 / 99.3	58.3 / 57.2	0 / 0	0.0
Miami, B	778	0.0	52.7	100.0 / 93.2	58.8 / 56.5	0 / 0	0.0
Chicago, A	39	0.0	0.2	100.0 / 99.6	58.3 / 57.2	0 / 0	0.0
Chicago, B	690	0.0	11.8	100.0 / 98.3	59.3 / 58.5	0 / 0	0.0
Vancouver, A	40	0.0	0.0	100.0 / 99.9	58.5 / 57.7	0 / 0	0.0
Vancouver, B	697	0.0	4.5	100.0 / 99.4	59.5 / 59.0	0 / 0	0.0
Winnipeg, A	37	0.0	0.0	100.0 / 100.0	58.5 / 57.7	0 / 0	0.0
Winnipeg, B	646	0.0	3.9	100.0 / 99.4	59.4 / 58.9	0 / 0	0.0

All water is treated greywater: municipal make-up was zero in every central and anchored case, and no heat was unmet.

In the central model the system used 100% (Case A) and 99.8–100.0% (Case B) less water than the tower at equal duty, and 100% and 99.9–100.0% less than a tower serving the whole module, which would draw 526–651 and 1,184–1,464 $m^3\ yr^{-1}$. Cooling electricity was 58.3–58.5% and 55.9–59.5% lower at equal duty. With measured pad effectiveness, datasheet fan power and the in-situ derating of the bare panel, the water reductions were 97.3–100.0% and 93.2–99.5%, and the electricity reductions 56.8–57.7% and 52.1–59.0%. All demand was met from treated greywater. These totals measure evaporation and bleed even where the source was recycled, so zero municipal withdrawal does not mean zero consumptive use.

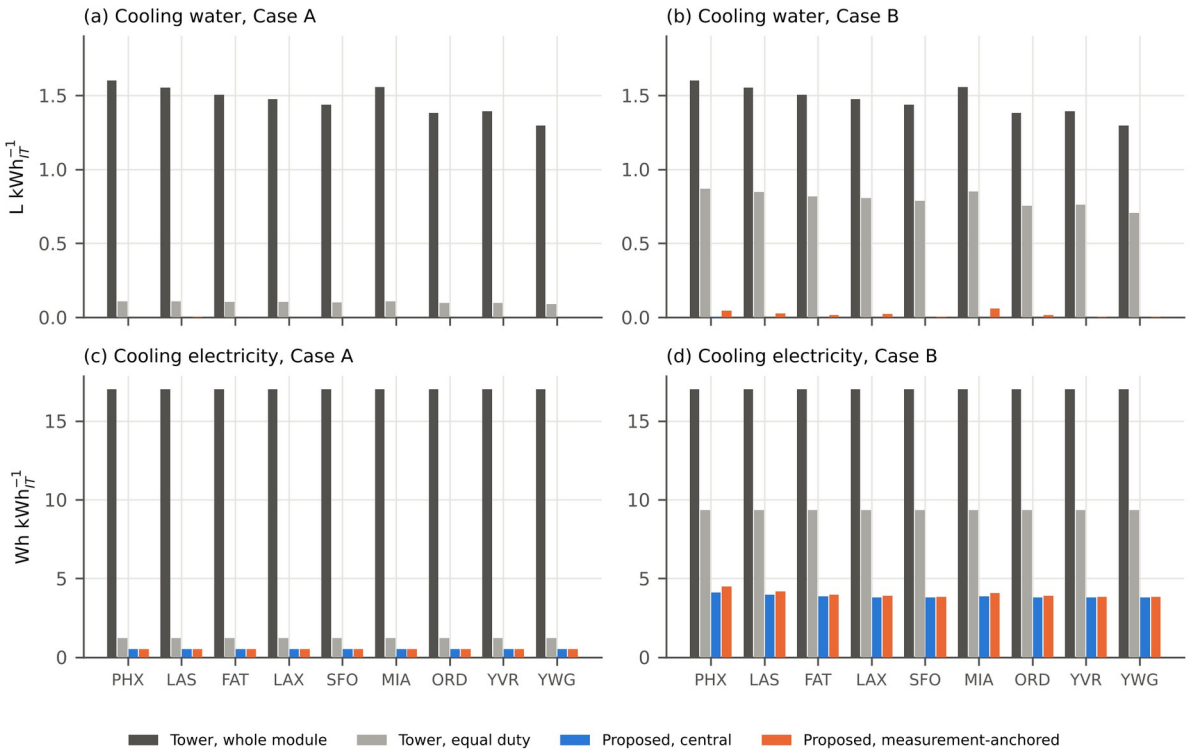

Fig. 3. Cooling water (a, b) and cooling-system electricity at equal duty (c, d) per kWh of IT electricity: evaporative tower for the whole module and for the same residual heat, and the proposed system in the central and measurement-anchored scenarios.

4.2 Comparison with a dry cooler and fan noise

A conventionally sized dry cooler uses no water. Compared on the electricity of rejection (fans, pumps, treatment), the proposed system used 0.3–9.4% less than a design-sized dry cooler in Case A (1.3–15.4% anchored). In Case B it used 1.8–12.4% less at the other seven sites but 3.7% more at Phoenix and 0.8% more at Las Vegas (anchored: 7.8% and 2.2% more). At those sites a dry coil that returns 45 °C water at the 0.4% design temperature of 42–43 °C must be rated at 69 and 63 kW, 1.7 times the hybrid coil, and its fans then turn slowly; the hybrid coil instead runs at full speed with its pads on the hottest afternoons. Pumping, common to both, dominates the electricity of either plant, so the differences are small in absolute terms (under 1 MWh yr^{-1}). Photovoltaic output from the array is not counted as a cooling saving.

The dry cooler ran every night and produced night-time equivalent levels of 15–41 dB(A) at 10 m. Against a presumed 40 dB(A) residential night ambient it raised the total level by more than 5 dB in up to 618 h yr^{-1} (Case B). The proposed cooler's fans stayed off from 22:00 to 07:00 at every site in both cases with the central inputs (Fig. 4). By day the picture depends on climate. In Case B the proposed system's day-time equivalent level was 6.6–13.0 dB below the dry cooler's at six sites but 9.8 dB above it at Phoenix, 7.6 dB above it at Las Vegas, 1.4 dB above it at

Fresno, for the same reason as the electricity difference; in Case A the two were within −4.9 to +2.4 dB.

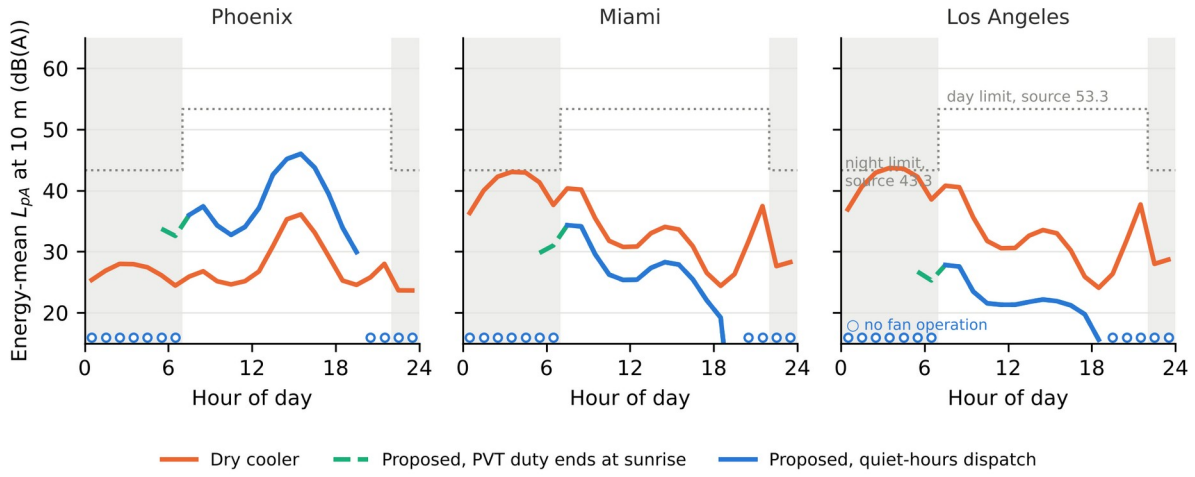

Fig. 4. Modelled hourly-mean fan noise at 10 m in three climates, Case B: dry cooler, proposed system with PVT duty ending at sunrise, and proposed system with the quiet-hours dispatch. Dotted lines mark the cooler level at which the total exceeds the presumed ambient by 5 dB; circles mark hours with no fan operation.

When the fans do run, the two coolers can also be compared hour by hour (Online Resource 2, Table S26). The dry-cooler reference is sized, as is usual, to carry the whole residual unaided at the 0.4% design temperature, whereas the hybrid coil has a fixed rating (30 or 40 kW) and can fall back on its pads. In the 24-hour configuration the hybrid cooler ran only by day, for 721–887 h yr^{-1} in Case A and 3,919–4,210 h yr^{-1} in Case B; it used its pads only at Phoenix (30 h) and Las Vegas (14 h) in Case B. The dry cooler ran 1,931 and 8,760 h yr^{-1}, of which 1,044 and 3,285 at night. While running, the hybrid cooler's energy-mean level at 10 m was 20 dB(A) in Case A, at minimum fan speed, and 22–41 dB(A) in Case B (hourly maximum 31–58 dB(A), reached at full speed with the pads wetted), against 19–32 and 28–38 dB(A) (maximum 45–58 dB(A)) for the dry cooler. In hours when both ran, the hybrid minus dry difference was −1.1 to +2.8 dB in Case A and −8.2 to +3.8 dB in Case B, positive at Phoenix, Las Vegas and Fresno. Where the daytime level is binding in a hot-dry climate, the hybrid coil should be sized up towards the dry cooler's rating; at Phoenix and Las Vegas a 70 kW coil lowered the daytime level by about 10 dB and the rejection electricity by 3–6% (Online Resource 2, Table S31).

4.3 Uncertainty propagation and the condition for fan-free nights

Nineteen inputs (Table 1) were sampled by Latin-hypercube sampling (McKay et al. 1979), 300 samples for each site and case (5,400 annual simulations), and the influence of each input was measured by the Spearman rank correlation with each output (Helton and Davis 2003). The PVT capacity is sampled through *t*, which interpolates every hourly capacity and film-water

table linearly between the anchored floor (t = 0) and the full model (t = 1); the host inputs include the night share of loads (0–35%) and customer volume (85–130% of the base profile).

Both conclusions are robust (Fig. 5). The 5th percentile of the water reduction relative to the tower was 92.5–99.6% across sites in Case A and 89.7–99.0% in Case B, and municipal make-up was zero in every sample except two samples (at most 0.07 $m^3\ yr^{-1}$). The 5th percentile of the electricity reduction was 46.3–47.4% and 41.6–51.0%; its spread comes almost entirely from the tower's own fan coefficient (Spearman ρ = 0.97), i.e. from the reference rather than from the proposed system. The water reduction depended mainly on the PVT capacity position t (ρ = 0.79), the night share of loads (0.26) and IT utilisation (-0.37). Fans ran between 22:00 and 07:00 in no sample at most sites; the share of fan-free samples was 96–100% in Case A and 89–100% in Case B, with at most 13 h yr^{-1}. No heat was unmet in any sample.

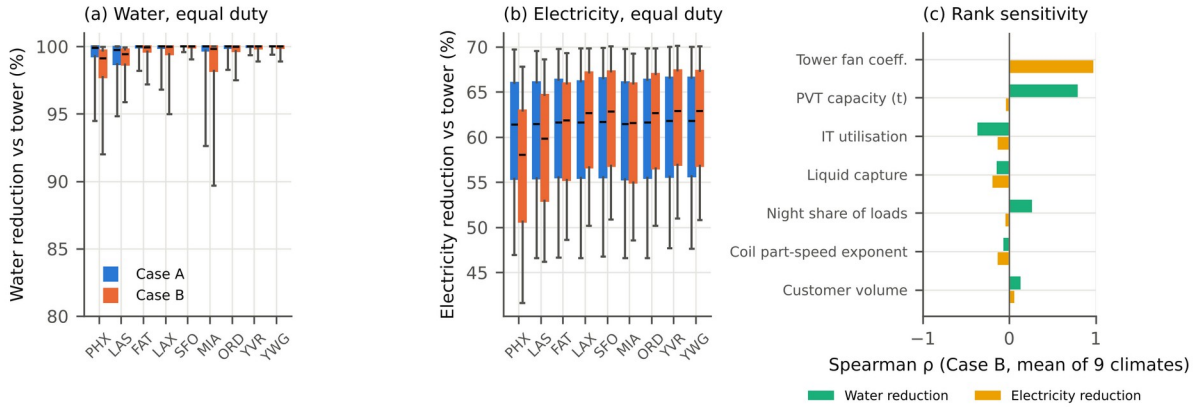

Fig. 5. Latin-hypercube uncertainty analysis (300 samples per site and case). (a) Water reduction and (b) electricity reduction relative to an evaporative tower at equal duty. Boxes: interquartile range and median; whiskers: 5th–95th percentiles. (c) Spearman rank correlation of the seven most influential inputs with the two outputs, Case B, averaged over sites.

The night result can be stated as a measurable condition. For each site and case, the PVT capacity was bisected, as a multiple of the validated bare-panel capacity, to find the smallest capacity giving zero fan hours between 22:00 and 07:00 with all other inputs central; below the anchored floor the bare-panel tables were scaled directly, so the search was not bounded at 0.74. The corresponding median night flux at a 47.5 °C surface was recorded (Table 4). The minimum was 0.31–0.50 times the bare-panel flux in Case A and 0.49–0.76 in Case B, identical with the measurement-anchored cooler. It exceeded the 0.74 anchor only for Las Vegas, Case B (0.757), which is the one fan hour a year in Table 3. Elsewhere the requirement was at most 0.50 (Case A) and 0.66 (Case B) of the bare-panel flux, so fan-free nights follow from bare-panel performance with margin, and do not depend on the unmeasured enhancements. In the earlier configuration of this work (45 °C return, washing heat only, 07:00–22:00 opening), the

anchored capacity left 14–613 fan hours a year at the seven US sites; loop temperature and host operation, not panel enhancement, are what remove them.

Table 4. Minimum PVT capacity for zero fan operation between 22:00 and 07:00 (central inputs): median night flux at a 47.5 °C surface, W m^{-2}, with the ratio to the validated bare-panel flux in brackets.

Site	Bare panel	Full model	Required, Case A	Required, Case B
Phoenix	294	503	123 (0.42)	193 (0.66)
Las Vegas	389	701	195 (0.50)	294 (0.76)
Fresno	337	583	148 (0.44)	207 (0.61)
Los Angeles	314	544	121 (0.38)	182 (0.58)
San Francisco	383	718	117 (0.31)	186 (0.49)
Miami	253	477	105 (0.41)	147 (0.58)
Chicago	434	796	200 (0.46)	280 (0.64)
Vancouver	389	695	181 (0.47)	258 (0.66)
Winnipeg	529	1,007	228 (0.43)	319 (0.60)

Ratios below 0.74 mean the measurement-anchored floor (bare panel × 0.74) exceeds the requirement. A single instrumented panel at 47.5 °C over representative nights can test the requirement at a given site.

4.4 Host operation, loop temperature and night options

Table 5 shows how the host's operation sets the result. Adding dryer heat to a laundromat open 07:00–22:00 raised the host's share of the captured heat from 55% to 73% (Case A) and from 25% to 39% (Case B); opening 24 hours with off-peak pricing raised it to 93% and 45%. Because the laundromat then uses heat through the night, the array has less to reject when it is least effective, and anchored water use fell by roughly two-thirds to an order of magnitude. Night fan operation at the anchored capacity was at most 13 h yr^{-1} even for the laundromat that closes at 22:00, and zero in every case once the thermal store was held in reserve for the night (Section 2.1).

Table 5. Effect of host operation at a 60 °C return, measurement-anchored scenario. Host share and residual heat are climate-independent; water is the range over the nine sites; fan hours are the maximum over sites between 22:00 and 07:00.

Host operation, case	Host share of captured heat	Residual (MWh yr^{-1})	Water, anchored (m^3 yr^{-1})	Fan h, anchored	Fan h, anchored + store reserve
Washing only, 07:00–22:00, A	55%	155	5–49	5	0
Washing only, 07:00–22:00, B	25%	586	24–178	14	0
+ dryers, 07:00–22:00, A	73%	93	4–34	5	0
+ dryers, 07:00–22:00, B	39%	473	22–159	13	0
+ dryers, 24 h, 10% at night, A	80%	68	1–14	1	0
+ dryers, 24 h, 10% at night, B	39%	473	13–113	6	0
+ dryers, 24 h, off-peak pricing: 25% at night, +15% volume (central), A	93%	25	0–1	0	0
+ dryers, 24 h, off-peak pricing: 25% at night, +15% volume (central), B	45%	427	3–53	1	0
+ dryers, 24 h, 35% at night, +30% volume, A	100%	0	0	0	0

Host operation, case	Host share of captured heat	Residual (MWh yr^{-1})	Water, anchored (m^3 yr^{-1})	Fan h, anchored	Fan h, anchored + store reserve
+ dryers, 24 h, 35% at night, +30% volume, B	51%	381	1–24	0	0

For a laundromat that closes at 22:00, the anchored array left at most 13 night fan hours a year at any site. Ozonated night misting (Section 2.4), run only when the film at full duty is insufficient, removed them at Phoenix and Miami and cut them from 9 to 1 h yr^{-1} at Las Vegas, but did not help at Vancouver (13 h), Chicago (8 h), Fresno and Winnipeg, where the remaining hours were below the 2 °C freeze lock-out, when no wetting is allowed. The store reserve removed them at every site without misting, for 5–159 m^3 yr^{-1} of treated water. Misting is therefore not needed in the configurations studied here and is retained only as a design option; in cold climates it cannot address the hours that remain.

The reserve is not free (Online Resource 1, Table S22). Because the store no longer serves the host in the evening, 8–13% of the heat reused with the 58 kW module goes to the hybrid cooler instead, and with the 130.5 kW module, whose host is already saturated, the cooler works harder by day: cooling electricity rose by 1–10% and the day-time equivalent level by 5–15 dB. It is therefore an option for hosts that close at night or sites where night silence is binding, not an improvement in general.

Loop temperature is the other lever. At a 45 °C return, with the panel surface at 40 °C and the host taking washing heat only between 07:00 and 22:00, the same anchored array left fans running 14–613 h yr^{-1}, and the misting option alone did not remove them. Raising the return to 60 °C increases the array's night flux by about a third, the dry coil's capacity by 50% or more, and the share of the host's demand that the loop can meet.

5 Case study: a laundromat host in Los Angeles

Los Angeles is used as the real setting because both constraints are binding there and both are written down. The Los Angeles Department of Water and Power reports per-capita use of about 101 gallons (382 L) per day after a 33% reduction over 15 years (LADWP 2025a) and a median single-dwelling customer use of 7.5 hundred cubic feet (21.2 m^3) per month (LADWP 2024), and a 2025 bill that would have required data centres in California to report their water use to their supplier was vetoed (Papan 2025). The city's noise ordinance prohibits air-conditioning and similar equipment from raising the level on another occupied property by more than 5 dB above the ambient, with presumed residential ambients of 50 dB(A) by day and 40 dB(A) at

night (City of Los Angeles 2026, Sections 111.03 and 112.02). Los Angeles County's alternate water source guidelines list cooling-tower make-up among the non-potable uses permitted for treated onsite sources and set the treatment standards and plan-check route through the Department of Public Health (Los Angeles County Department of Public Health 2016).

5.1 Consequences for the water supply

A tower serving the whole module would draw 599 $m^3\ yr^{-1}$ (Case A) or 1,347 $m^3\ yr^{-1}$ (Case B) of municipal water, the annual use of 2.3 or 5.3 median single-dwelling customers, and would add 19% or 43% to the laundromat's own consumption (about 3,129 $m^3\ yr^{-1}$ at 60 L per load). The proposed system drew no municipal water in any central, anchored or sampled Los Angeles case. Its treated-greywater demand was 0.0 and 0.0 $m^3\ yr^{-1}$ in the central model and 0.1 and 21.7 $m^3\ yr^{-1}$ in the anchored scenario. In the anchored scenario, zero municipal make-up needed 1.5 L of greywater per load in Case B (71 $m^3\ yr^{-1}$), against 60 L available; the hottest site, Phoenix, needed 5.7 L per load.

5.2 Consequences for noise regulation

Applying the ordinance test to every hour, a cooler level of 43.3 dB(A) at night and 53.3 dB(A) by day at the receiver raises the total by exactly 5 dB over the presumed ambient. Inverting Eq. 9 gives the minimum receiver distance at which every hour complies (Fig. 6b). The dry cooler needs 9.9 m (Case A) and 17.5 m (Case B) at night; the proposed system needs none in this screening, in both the central and anchored scenarios, because its fans do not run at night. A typical urban laundromat lot places a roof-top cooler within 10–20 m of the nearest dwelling, so the difference decides whether a site is usable at night without barriers. If the measured night ambient is 45 or 50 dB(A) rather than the presumed 40 dB(A), the dry cooler's Case B setback falls to 9.9 or 5.5 m. By day the proposed system needs 0.2 m and 1.0 m, against 0.9 m and 4.5 m for the dry cooler. These distances are a screening estimate, not a compliance finding: the calculation omits pump and film-circuit noise, which continue at night, as well as tonal character, directivity, ground and barrier effects and the measured ambient at a real site, and the ordinance is applied by the city to measured levels. They show which configuration is likely to need mitigation, and by how much, before a site-specific acoustical assessment.

5.3 Host-site requirements as design variables

Fig. 6a shows night fan hours as a function of array area. Fan-free nights in Los Angeles need about 43 m^2 of array for Case A and 85 m^2 for Case B with the full model, and 85 m^2 and 255 m^2 at the anchored capacity, well below the 170 and 340 m^2 assumed in Cases A and B. Because

the hybrid coil now carries only a small daytime residual, the 30 and 40 kW coils already keep the daytime setback below 2 m; raising the Case B coil to 70 kW reduces it to 0.3 m. Table 6 collects the host-site requirements as design variables for a project developer or a permitting authority.

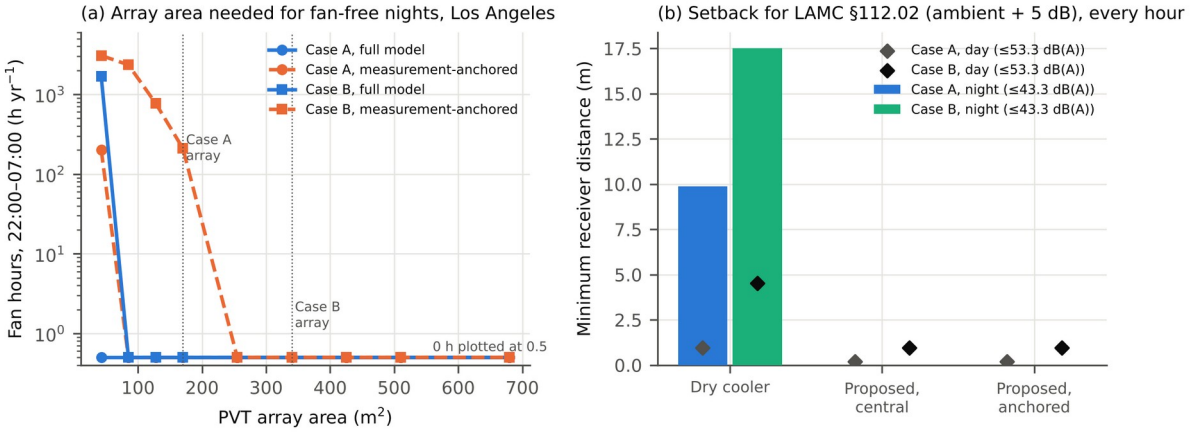

Fig. 6. Los Angeles case study. (a) Fan hours between 22:00 and 07:00 against PVT array area for both module sizes, full model and measurement-anchored capacity (zero plotted at 0.5 h). (b) Minimum receiver distance at which every hour meets the LAMC Section 112.02 test with presumed ambients of 40 dB(A) at night (bars) and 50 dB(A) by day (diamonds).

Table 6. Host-site requirements for the Los Angeles laundromat host expressed as design variables. Ranges span the central model and the measurement-anchored scenario.

Design variable	Case A (58 kW)	Case B (130.5 kW)	Governs
Loop supply / return (°C)	45 / 60	45 / 60	all
Host operation	24 h, dryers, off-peak pricing	as Case A	heat reuse, water
PVT array area for fan-free nights (m^2)	43–85	85–255	night noise
Night PVT flux at 47.5 °C to be demonstrated (W m^{-2}, median)	121	182	night noise
Thermal store dispatch if the host closes at night	store reserve	store reserve	night noise
Hybrid coil rating (kW) / daytime setback (m)	30 / 0.2	40 / 1.0	day noise
Night setback to nearest dwelling, proposed vs dry cooler (m)	0 vs 9.9	0 vs 17.5	LAMC §112.02
Treated greywater for zero municipal make-up (m^3 d^{-1})	≈ 0	≤ 0.2	water supply
Greywater available at 60 L per load (m^3 d^{-1})	7.7	7.7	water supply
Municipal water avoided vs whole-module tower (m^3 yr^{-1})	599	1,347	water supply
Permits	DPH alternate water source plan check; Legionella management plan	as Case A	public health

5.4 Roof fit and module size

To test whether the arrays fit, the usable roof of a flat-roofed building was taken as its area less a 4 ft (1.22 m) clear perimeter, which the California Fire Code requires around rooftop

photovoltaic arrays where neither building axis exceeds 250 ft, and one 4 ft central access pathway (California Building Standards Commission 2025; local fire officials may require more), less a further 15% for air-handling units, dryer exhausts, flues and hatches. The panels form one continuous plane at 30° tilt, not rows, because the skirted plenum beneath them must stay closed for the draft stack at its upper edge to draw air under the panels; gaps would open the plenum and defeat the chimney. Each 1.7 m^2 panel therefore occupies its horizontal projection, 1.47 m^2, at every latitude, and the stack is taken to fit within the 15% allowance. A single plane does not shade itself. On that basis the 170 m^2 array of Case A needs a building of 2,914 ft^2 and the 340 m^2 array of Case B one of 5,157 ft^2. US laundromats occupy 1,000–5,000 ft^2 of retail space, averaging about 2,200 ft^2 (Coin Laundry Association n.d.), and express car-wash tunnels are usually 90–140 ft long (Professional Carwashing & Detailing n.d.), giving buildings of roughly 4,000–6,000 ft^2. The arrays of Cases A and B are larger than fan-free nights require (Section 5.3), so the binding question is how much compute a given roof supports.

Table 7 answers it for three representative roofs: the number of panels that fit and the largest module that keeps the fans off between 22:00 and 07:00 with the 24-hour host, found by bisection on the number of accelerators with the coil scaled in proportion. A typical 2,400 ft^2 laundromat roof holds 78 panels (133 m^2) and supports 67–94 kW at the anchored capacity and 100–138 kW with the store reserve, so the 58 kW module of Case A fits an average laundromat at every site without the reserve, and the 130.5 kW module of Case B fits it with the reserve in San Francisco and Los Angeles. A large laundromat (192 panels) or a car-wash tunnel roof (229 panels) supports 123–228 kW at the anchored capacity and 178–304 kW with the reserve, more than Case B at every site. Because the plane has no rows to space apart, roof fit does not depend on latitude: Vancouver and Winnipeg support 67 and 71 kW on the typical roof at the anchored capacity (100 and 107 kW with the reserve), at the low end of the range with Chicago. No case needed municipal water or curtailment at these limits.

Table 7. Roof fit: panels that fit on representative host roofs (range over the nine sites) and the largest module (IT kW) with no fan operation between 22:00 and 07:00, 24-hour host.

Roof	Roof / usable (m^2)	Panels	Array (m^2)	Full model	Anchored	Anchored + reserve	Anchored + reserve + mist
Typical laundromat (2,400 ft^2)	223 / 115	78	133	125–212	67–94	100–138	100–152
Large laundromat (5,000 ft^2)	465 / 284	192	326	279–489	123–198	178–265	178–303
Express car-wash tunnel (127 x 48 ft)	566 / 338	229	389	330–576	136–228	201–305	201–352

Fig. 7 sizes the module to the host from the other direction: the number of whole NVIDIA B300 GPUs whose captured heat would replace all, or half, of the water- and drying-heat demand of three car-wash and three laundromat sizes, alone and with small, medium and large solar preheating arrays (author's sizing calculation, independent of the hourly model). Full replacement scales almost linearly with throughput, at about 0.21 GPU per daily wash load for laundromats and 0.12 GPU per vehicle a day for car washes. The hourly model's laundromat (143 loads a day on average, with the off-peak uplift) needs 32 GPUs for full replacement on a daily-energy basis, 0.23 per daily load, consistent with Fig. 7 and close to the 32 GPUs of Case A; its car wash implies 0.08 GPU per vehicle a day, below the sizing chart because the model credits only the share of wash-water heat that a 60 °C loop can supply. On the chart's ratio, Case B (72 GPUs) is full replacement for a laundromat of about 348 loads a day, between the small and medium laundromats, whereas the Case B results above use the 143-load host and therefore represent a module oversized for its host by about a factor of two; at a matched host the residual heat, and with it the night array, coil duty, cooling water and roof requirement, would approach those of Case A scaled by throughput. Daily-energy matching overstates the share absorbed hour by hour, because demand and output do not coincide (93%, not 100%, for Case A in Table 5), and solar preheating competes with GPU heat for the same demand, which is why Fig. 7 needs fewer GPUs as the solar array grows.

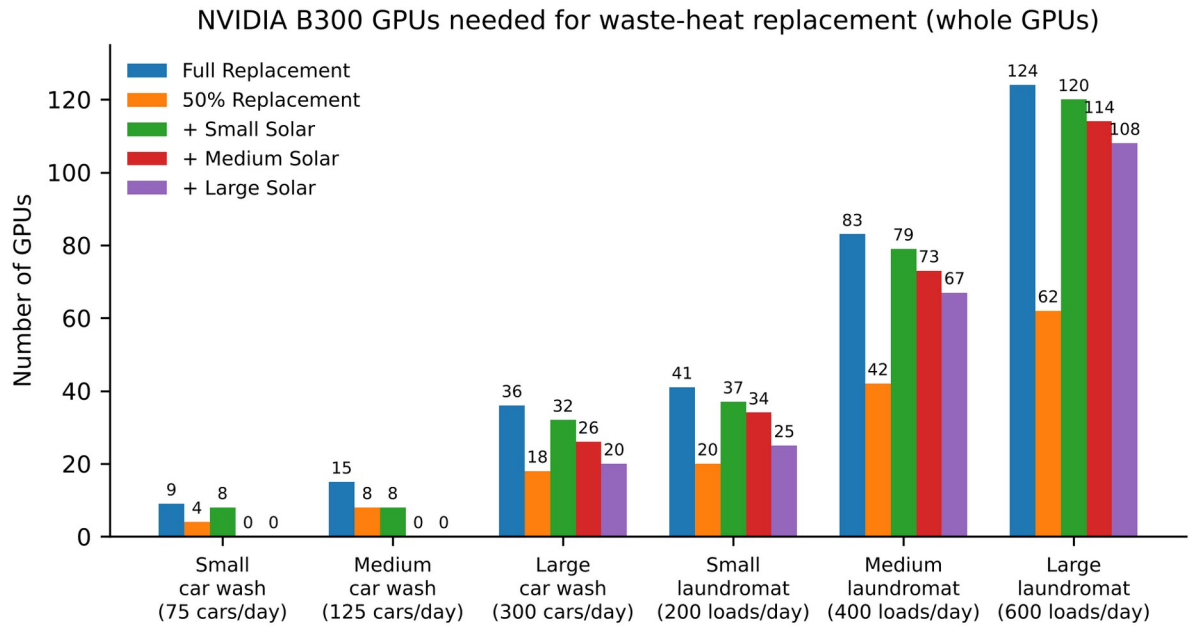

Fig. 7. NVIDIA B300 GPUs needed for host waste-heat replacement (whole GPUs) by host type and size: full replacement, 50% replacement, and full replacement with small, medium and large solar preheating (author's sizing calculation). Car washes: 75, 125 and 300 vehicles a day; laundromats: 200, 400 and 600 loads a day.

5.5 Seasonal tilt setting

The plane is hinged at its lower edge, so its tilt can be changed with the seasons while the stack base stays at the highest point. Because the footprint grows as the plane flattens, the roof layout of Section 5.4 is kept valid only if the plane is never set flatter than its 30° layout angle; an unrestricted 0–60° range, adjusted hourly, raised the irradiance on each panel by 5–7% but forced a larger footprint and reduced the irradiance collected per roof by 7–9%. Within 30–60°, a two-position setting was evaluated: 30° from April to September and a steeper winter angle from October to March, chosen for the most winter irradiance subject to no loss of night rejection. Irradiance was transposed from the hourly beam and diffuse data with an isotropic sky, and night rejection of the bare panel at 47.5 °C was recomputed with the sky view factor $(1 + \cos \beta)/2$ of the tilted plane and a stack height increased by the rise of an 8 m plane (Table 8).

Table 8. Two-position seasonal tilt (April–September / October–March) against a fixed 30° plane: change in annual plane-of-array irradiance and in bare-panel night rejection at 47.5 °C.

Site	Summer / winter tilt	Irradiance (%)	Night rejection (%)	Irradiance, three positions (%)
Phoenix	30° / 50°	+2.2	+0.6	+2.3
Las Vegas	30° / 50°	+2.7	+0.2	+2.8
Fresno	30° / 45°	+1.1	+1.0	+1.0
Los Angeles	30° / 45°	+1.4	+0.6	+1.6
San Francisco	30° / 50°	+1.6	+0.6	+1.6
Miami	30° / 40°	+0.4	+0.3	+0.6
Chicago	30° / 50°	+1.8	+1.0	+1.8
Vancouver	30° / 55°	+1.9	+1.3	+2.1
Winnipeg	30° / 60°	+4.4	+1.4	+4.3

The winter angle was 40–60°, steepest at Vancouver and Winnipeg. Annual irradiance rose by 0.4–4.4% and night rejection by 0.2–1.4%, because a steeper plane gains natural convection and stack draft that slightly outweigh its smaller view of the sky. A third, spring-and-autumn position added no more than 0.2 percentage points. The gains are small; the seasonal setting is retained because steep winter angles shed snow, which would otherwise block night rejection at cold sites, and because it costs nothing in roof area. All other results in this paper use the fixed 30° plane and are therefore slightly conservative. At 60° the upper edge of an 8 m plane stands 6.9 m above the roof, which the structure and wind design must accommodate.

6 Discussion

6.1 Position relative to the state of the art

Studies of data-centre water use have treated the choice of rejection plant as a trade-off between water and electricity at the facility scale (Karimi et al. 2022; Lei et al. 2025; Siddik et al. 2021).

Warm-water GPU cooling changes that trade-off: with water returning at 60 °C, a host with matching thermal demand can absorb most of the heat, and the rest can be rejected without evaporation for most of the year. The present results add night-time noise as a third axis, which matters once computing is placed in small units near homes, and show that the proposed system used less water and less electricity than a tower at every site, and less electricity than a dry cooler sized for the same supply temperature at all but the two hottest sites with the larger module.

For PVT night cooling, previous work measured 20–75 W m^{-2} with fluid close to ambient for building cooling (Eicker and Dalibard 2011; Péan et al. 2015; Bilbao and Sproul 2015). The validation here shows that the same class of collector, held at 47.5 °C for heat rejection, operates in a different, convection-dominated regime in which certified test data and the bare-panel model agree at about 250–530 W m^{-2}. That is why a roof array can carry most of a module's residual heat on bare-panel performance alone; the enhancement mechanisms add margin but are not required for fan-free nights in the configuration studied.

For greywater reuse, published laundry and car-wash systems have targeted reuse in washing (Ciabatti et al. 2009; Zaneti et al. 2011). Cooling make-up is a much smaller end use here: the greywater needed for zero municipal make-up in the anchored scenario was under 3% of the laundromat's supply in Los Angeles and about 9% in Phoenix.

6.2 Limitations

The results are model predictions, and the configuration rests on assumptions that are not yet measured. The host's heat use depends on the 24-hour profile with 25% of loads at night and on the compatible shares of wash-water and dryer heat (f_T = 0.85, f_d = 0.50), which have not been metered at a real laundromat; Table 5 and the sampled ranges show how the results change when they differ. A 60 °C return requires a cold-plate flow sized for a 15 K rise at a 45 °C inlet, within published GPU limits but at the upper end of current practice, and some operators prefer cooler water for resilience (Uptime Institute 2026). The PVT enhancements and the evaporative film have not been measured, and the film's biological behaviour on treated wash-water permeate has not been tested. The surface temperature is fixed rather than solved from a fluid-side model; the coil and tower models are rating-based; the acoustic model rests on one datasheet level and omits pumps, directivity, tonal components and barriers; and a typical-year weather file does not resolve multi-day heatwaves or future warming. Host heat demand per load is taken as climate-independent, with mains water at 15 °C; colder mains at the northern sites would raise the host's demand and its share of the heat, so this is conservative there. At Winnipeg, and on the

coldest Chicago and Vancouver nights, the outdoor circuit needs a glycol mixture or a drain-down arrangement; the lower heat capacity and higher viscosity of glycol, which reduce coil and panel heat transfer by a few percent, are not modelled. No Legionella compliance, water-reuse permit or noise-compliance finding follows from the simulations.

The misting option raises a specific public-health question because it deliberately creates fine droplets from treated wash water above a roof. Ozone is a strong biocide for recirculating cooling water (Pope 1992), but it decays within minutes and leaves little residual downstream of the injection point or in the droplets, and evidence that it prevents Legionella colonisation in real systems is mixed (Edelstein et al. 1982; CDC 2024). The model therefore treats ozonation as one barrier within a water-management programme, alongside ultraviolet disinfection, thermal flushing, automated monitoring of the disinfectant residual, biofilm control in the spray lines, ozone off-gas destruction and drift containment, rather than as a guarantee. Because the store reserve achieves the same result without spray, misting is not part of the recommended configuration.

The evidence path is therefore specific: (i) metering of hourly wash-water and dryer heat demand at a 24-hour laundromat with off-peak pricing; (ii) single-panel trials at 47.5 °C to confirm the anchored flux of Table 4; (iii) a summer greywater treatment, scaling and Legionella-monitoring trial for the wetted pathways; and (iv) a site pilot measuring heat flows, water and day/night sound at the actual boundary.

7 Conclusions

An hourly model of host-sited AI computing that reuses heat in a laundromat, rejects heat through a fanless PVT roof array at night and through a greywater-fed hybrid cooler by day was formulated, with every heat exchanger solved for the 45 °C supply; its sub-models were benchmarked against certified or measured data and its uncertainty was propagated. The conclusions, with the conditions on which each depends, are:

- With 45 °C supply and 60 °C return water, a 24-hour laundromat with dryer heat recovery and off-peak pricing absorbed 93% of a 58 kW module’s heat and 45% of a 130.5 kW module’s. Opening hours, night customers and dryer heat are the main determinants; a laundromat open 07:00–22:00 for washing heat only absorbed 55% and 25%.
- Relative to an evaporative tower rejecting equal heat, the system used 93–100% less water and 52–59% less cooling electricity, and no municipal water, including with measured pad effectiveness, datasheet fan power and an in-situ derating of the bare

panel; the 5th percentile over 5,400 sampled years was at least 89% for water and 41% for electricity. Against a dry cooler sized for the same 45 °C supply, the rejection electricity was between 12% lower and 4% higher; it was higher only at Phoenix and Las Vegas with the larger module, where that dry cooler must be large.

- On bare-panel performance consistent with ISO 9806 certified coefficients and derated by 0.74 for in-situ effects, fans stayed off between 22:00 and 07:00 in 17 of the 18 site–module cases; Las Vegas with the larger module ran them for one hour a year, needing 0.76 rather than 0.74 of the bare-panel capacity. The minimum capacity elsewhere was 0.31–0.66 of the bare panel, so the unmeasured enhancements were not required. This is a benchmark against certified test coefficients, not a measurement of the proposed array at 47.5 °C, which remains the first test to make. If the host closes at night, fans ran at most 13 h yr^{-1}; holding the thermal store in reserve removed them at the cost of 8–13% of the heat reused with the small module and up to 10% more cooling electricity and a 5–15 dB louder day with the large one. These results depend on the 60 °C return: at 45 °C the same array left 14–613 fan hours a year at the seven US sites.
- In Los Angeles the system avoids 599–1,347 m^3 yr^{-1} of municipal water relative to a whole-module tower. On a screening estimate that omits pumps, tonality and propagation effects, it needs no night setback under the city's noise rule, against 10–18 m for a dry cooler; a site-specific acoustical assessment is still required.
- With the panels forming one continuous plane over the chimney plenum, a typical 2,400 ft^2 laundromat roof supports 67–94 kW of fan-free compute at anchored panel performance and 100–138 kW with the store reserve; a large laundromat or car-wash roof supports 178–304 kW with the reserve at every site.

The design is therefore supported as a water-saving and quiet option for distributed computing, conditional on warm-water GPU cooling, on host heat demand that should now be measured at a real 24-hour laundromat, and on a field test of the PVT array at its operating temperature.

CRediT authorship contribution statement

Jonathan Karl Bily: Conceptualisation, Methodology, Software, Formal analysis, Investigation, Visualisation, Writing – original draft, Writing – review and editing.

Declaration of competing interest

The author is the founder of Cloud Wash Laundry Corporation and the named inventor on pending patent applications, held for Cloud Wash Laundry Corporation, concerning co-location of computing equipment with heat-consuming facilities, photovoltaic–thermal heat rejection and wash-water recycling for evaporative cooling.

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Data availability

Code, weather files and all model outputs, including the scripts for the validation, coupled heat-exchanger, uncertainty and case-study analyses, are at https://github.com/bcjohnjr/Greywater-Hybrid-PVT-Cooling-System; material currently in that repository describes earlier configurations. The code is released under the MIT licence; weather files retain their providers' terms.

Declaration of generative AI and AI-assisted technologies in the manuscript preparation process

During the preparation of this work the author used Artificial Inventor/Synthetic Scientist V63 to assist with model development, analysis and drafting, and to assist with the validation, uncertainty and case-study analyses and with revising the text. The author reviewed and edited the code, results and text and takes full responsibility for the content of the publication.

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Supplementary material

Online Resource 1: extended methods and results for the configuration of this paper (climate summary, coupled coil and PVT along-flow checks, heat allocation, water by end use, tower and dry-cooler comparisons, host types, element contributions, one-at-a-time and PVT-capacity sensitivity, break-even capacity, store-reserve trade-off). Online Resource 2: acoustic levels, levels while running and acoustic sensitivities. Online Resource 3: complete Python code, weather files and model outputs, with every script needed to regenerate the tables and figures.