Modelling and Seasonal Optimization of a PVT System for North Sydney Olympic Pool

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Abstract

The North Sydney Olympic Pool (NSOP) is heated by three water source heat pumps whose cold side source is pre-warmed by seawater drawn from Sydney Harbour and by a rooftop photovoltaic-thermal (PVT) collector array. This work develops an independent, first-principles model of that plant in MATLAB, computing pool heat demand, collector performance, source-loop temperature, and heat-pump electricity across a full year at hourly resolution. Every constant is held in a single parameters file, making the model reproducible and free of parameter drift. The photovoltaic-thermal stage the only subsystem for which measured field data is available is validated against nineteen days of live plant measurements, reproducing the measured thermal energy through a single fitted field-correction factor and, independently of that factor, matching the collector outlet temperature to a root-mean-square error of 1.58 °C. The source-loop flow, initially taken from the vendor design maximum, was corrected to the operator-confirmed value of approximately 35 kg/s; at this flow the model reproduces the measured March source temperature to within about one and a half per cent of the vendor's design figure. The validated model gives an annual pool demand of 4,665 MWh met by 838 MWh of heat-pump electricity at a seasonal coefficient of performance of 5.57. A seasonal optimization of the loop flow identifies a zero-capital control strategy reducing the loop flow from the current 35 kg/s to approximately 15.5 kg/s in summer and 23 kg/s in winter that saves on the order of 90 MWh of electricity per year, with the collector split shown to be immaterial. A sensitivity analysis establishes that annual electricity is governed by the seawater subsystem and the hot-loop temperature and is nearly insensitive to the PVT correction factor. All assumptions, data sources, and residual gaps are reported transparently.

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Nomenclature

Symbol	Description	Value / Unit
η_0	Collector optical efficiency (ISO 9806)	0.4112
a_1	Collector first-order loss coefficient	12.106 W/m²K
k	PVT field-correction factor (fitted)	0.68
A_{pvt}	Collector aperture area (207 panels)	534.73 m²
G	Global irradiance on the collector plane	W/m²
ε_{hex}	Seawater exchanger effectiveness (calibrated)	0.60
T_{sea}	Monthly sea temperature (calibrated)	18–27 °C
$\dot{m}_{loop}$	Cold / source loop flow (operator-confirmed)	~35 kg/s
pvt_{split}	Fraction of loop flow through the PVT branch	0.25
c_p	Specific heat of water	4180 J/kg·K
T_{hot}	Hot-loop supply temperature (schematic)	45 °C
η_c	Heat-pump second-law efficiency	0.45
COP	Coefficient of performance (capped)	≤ 6
ε_w	Pool-water emissivity	0.95
σ	Stefan–Boltzmann constant	5.67×10^{-8} W/m²K⁴
U_{gnd}	Ground conduction coefficient	0.5 W/m²K
tariff	Electricity price	0.30 AUD/kWh

1 Introduction

1.1 Motivation

Heating large public swimming pools is among the most energy-intensive activities in municipal aquatic infrastructure. A heated pool is, in thermodynamic terms, a large warm surface held above ambient temperature and left permanently exposed to the air and sky. It loses heat continuously and simultaneously through four mechanisms evaporation, convection, radiation, and conduction and because the water is held at a fixed set-point, every watt lost must be continuously replaced. For a facility with several pools of differing temperatures, some indoors and some outdoors, the aggregate standing load is substantial and runs day and night, year-round.

The conventional means of supplying this heat gas fired boilers or electric resistance heating are either carbon intensive or expensive to run. Air-source heat pumps improve on but are limited by the temperature of the ambient air they draw from, which is coldest precisely when heating demand is highest. The North Sydney Olympic Pool adopts a more sophisticated approach: it uses water-source heat pumps whose cold-side source is deliberately warmed above ambient by two free heat sources seawater drawn from Sydney Harbour, and a rooftop photovoltaic-thermal array that harvests the waste heat of its solar panels. Because a heat pump's efficiency is governed by the temperature difference it must bridge, warming the source directly compresses the electrical demand of the compressors.

This creates an engineering question that is both interesting and practically valuable: given a plant that couples two intermittent, weather-dependent heat sources to a bank of heat pumps serving several pools, how does the system perform across a full year, and can its operation be tuned to reduce electricity without new capital expenditure? Answering this requires a model that is faithful to the plant's physics, validated against its real behavior, and transparent enough to be interrogated.

1.2 Problem statement and objectives

The objective of this work is to build an independent, transparent model of the NSOP heating plant. Three specific goals follow. First, the model must reproduce the plant's real, measured behavior wherever field data exists, so that its predictions can be trusted. Second, it must quantify the annual energy demand, the seasonal efficiency, and the electricity consumption of the system. Third, it must be usable to identify operating settings principally the loop flow rate that minimize electricity use, and to establish how sensitive those conclusions are to the model's underlying assumptions.

A first-principles model is deliberately chosen over a proprietary transient simulation. The value of building the model from physical equations is that every stage can be inspected, each assumption named and tested, and the whole validated against measured data and stress-tested through sensitivity analysis. A black-box simulation may be more detailed, but it cannot be interrogated in the same way, and for a study whose purpose is understanding, transparency is the higher priority.

1.3 Guiding principle: measured reality over design values

A principle runs through the entire methodology: where they disagree, live plant data and operator knowledge take precedence over vendor design figures. This ordering is applied consistently. The collector output is calibrated to the plant's own dashboard rather than to the panels' certified rating; the exchanger effectiveness is calibrated to field data; and, as Section 6 describes, the source-loop flow was corrected from the vendor design maximum to the operator's confirmed operating value. Vendor figures are retained as a secondary cross-reference, valuable for sanity-checking, but they are never used to calibrate the model.

1.4 Scope and boundaries

The model represents the plant's indirect operating mode, in which the PVT and seawater warm the heat-pump source; this is the mode in which the plant operates for the overwhelming majority of the year. The summer direct-heating mode is analyzed and quantified but not fully modelled with switch-over control logic. Field validation is confined to the PVT collector the only subsystem with measured data. The heat-pump COP, the seawater exchange, and the annual electricity are physically grounded and cross-checked against the vendor's design figures, but they are not independently field-validated, and this distinction is maintained explicitly throughout. The plant was under commissioning during the study, so certain operating values notably the loop flows are not yet finalized even at the plant, and this is reflected in the way results are reported.

2. Related Work

2.1 Prior modelling approaches

The modelling of solar-assisted heat-pump systems for aquatic facilities draws on three established bodies of work: solar-thermal collector characterisation, pool thermodynamic load modelling, and heat-pump performance modelling. Each is well developed individually, and the contribution of the present study lies in integrating them for a specific, real, hybrid plant and validating the result against field data.
Solar-thermal collector performance is standardised internationally under ISO 9806 [1], which characterises a collector by an optical efficiency and one or two temperature-dependent loss coefficients determined by steady-state testing. This forms the basis of the collector stage here. For the unglazed collectors used at NSOP, the loss coefficients are dominated by wind-driven convection and long-wave sky radiation, making their performance strongly weather-dependent.

The most directly relevant prior work is the CoolSheet / UNSW [4] techno-economic and dynamic-thermal modelling framework developed for this class of system. That framework describes a comprehensive computational approach: a TRNSYS Type 850 [5] collector model based on the Akhtar–Mullick empirical relations [3], which resolves the top-loss coefficient and cover temperature semi-analytically; a mean-plate collector temperature rather than an inlet temperature; a part-load heat-pump COP curve fitted to manufacturer data; and a detailed treatment of Australian time-of-use tariffs and environmental-certificate economics.

It also documents the two-mode NSOP control strategy and notes practical features such as the buffer tank that stabilises the heat-pump unit against collector fluctuations. The present work aligns with that framework in its underlying physics but adopts deliberately simpler forms of several components, justified in Section 8.3 by the accuracy achieved against measured data. Where the reference framework describes the most sophisticated available methods and design intent, this study reports field-validated behaviour and states its residual gaps openly.

Pool thermodynamic load modelling follows the ASHRAE [2] tradition, decomposing surface heat loss into evaporation, convection, radiation, and conduction, with evaporation typically responsible for the majority of the total. Heat-pump performance modelling rests on the Carnot limit, with real performance expressed as a fraction of the Carnot COP through a second-law efficiency. All three traditions are combined in the model that follows; the contribution is their integration, field-validation, sensitivity characterisation, and optimization for a specific plant.

2.2 Research gap

Existing work on this class of system the CoolSheet/UNSW modelling framework and the vendor's design study characterises expected performance using design-stage models and manufacturer specifications. What is absent is an independent, field-validated account of how the plant performs once installed and commissioned: one that tests the design assumptions against the plant's own measured data, quantifies the shortfall between certified and installed collector output, and identifies which modelling assumptions actually govern the annual result. This study addresses that gap. It validates the collector stage against nineteen days of live field data, corrects a design-stage flow assumption against operator-confirmed operating values, and uses a one-at-a-time sensitivity analysis to locate the model's true uncertainties in the seawater subsystem. In doing so it provides what the design-stage literature does not: a measured-reality account of the plant's efficiency and a concrete, capital-free operating recommendation.

3. System Description

The plant is organized around a closed source loop the cold side of the heat pumps that connects the two heat sources to the pumps. Seawater is drawn from Sydney Harbour at roughly 52 kg/s and passed through a titanium plate heat exchanger, warming the circulating loop fluid without mixing with it. The cold loop, running at approximately 35 kg/s, then splits: about a quarter of the flow is routed through the rooftop PVT array, where it absorbs the thermal energy harvested from the solar panels, before recombining with the bypass. The combined, warmed loop fluid feeds the cold side of three water-source heat pumps, which lift it to a 45 °C hot-water loop that distributes heat to the five pools. A buffer tank on the loop stabilises the heat-pump unit against sub-hourly fluctuations in the collector output. The model reproduces this topology as an hourly energy balance around the source loop. Figure 1 shows the overall plant, in which seawater and the PVT array warm the closed source loop that feeds the heat pumps.

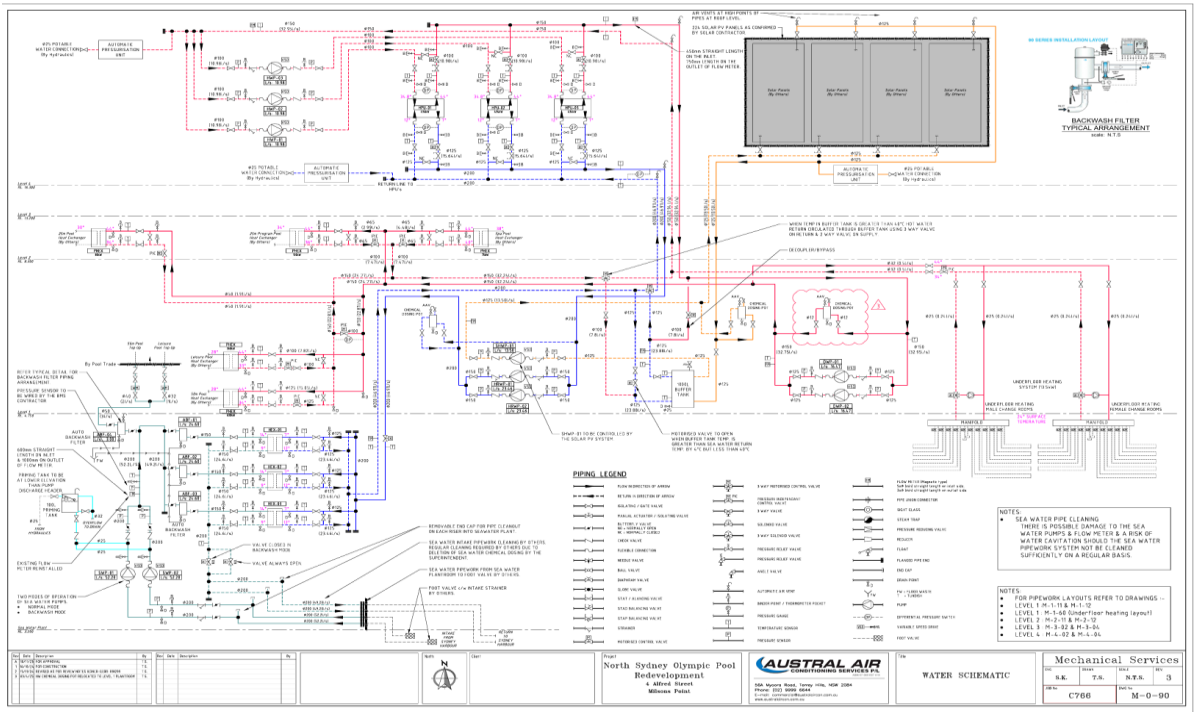


Figure 1. Plant schematic

The facility comprises five distinct bodies of water, each held at its own set-point. The large 50 m pool is outdoor and, by virtue of both its area and its exposure to wind, dominates the total heat demand. The remaining pools are indoor, where near-still air greatly reduces convective and evaporative losses. Their set-points and modelled surface areas, taken from the vendor's measured design data, are given in Table 1.

Table 1. Pool set-points and modelled surface areas.

Pool	Set point	Surface area	Environment
50 m	26 °C	1063 m²	Outdoor
Leisure	26 °C	300 m²	Indoor
Program	34 °C	280 m²	Indoor
25 m	27 °C	394 m²	Indoor
Spa	37 °C	26 m²	Indoor

One entry warrants a note. The 25 m pool is listed with a surface area of 394 m² against a volume of 164 m³, implying an average depth below half a meter, which is physically implausible for a lap pool. The figure is adopted as the vendor's measured value, and its effect on demand is flagged as a point for confirmation with the operator (Section 9).

4. Methodology

4.1 Model architecture

The model is implemented as a set of MATLAB [7] modules that share a single parameters file. This is a deliberate architectural decision. When physical constants are duplicated across scripts, they inevitably drift out of agreement as a model evolves, producing results that silently depend on which script last defined a value. By holding every constant in one authoritative file that every stage reads from, the model guarantees internal consistency and full reproducibility; any change to an assumption is made in one place and propagates uniformly.

Conceptually the model is a chain: weather drives the pool demand; the PVT and seawater warm the source loop; the loop temperature sets the heat-pump COP; and the COP and demand together give the electricity and cost. This computational sequence is summarized in Figure 3 In the code, the demand, source-loop, and heat-pump stages are consolidated within a single dynamic-loop module, because the source temperature must be solved self-consistently each hour. The collector stage is validated separately against field data before being embedded, and the optimizer wraps the whole loop in a parameter sweep. Figure 2 shows the system as represented in the model, reducing the plant to the loops and flows that govern its energy behavior.

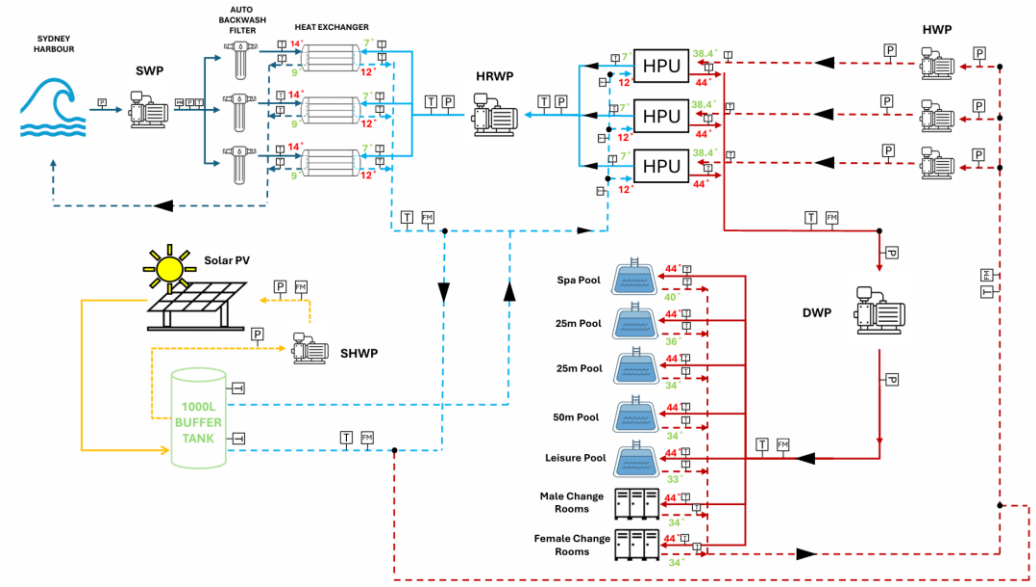


Figure 2. Simplified representation of the system as modelled, showing the three loops and the flows used in the model

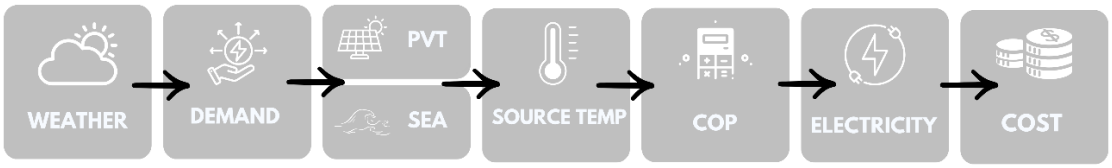


Figure 3. Model pipeline: weather drives demand and (with PVT and seawater) the source temperature; source temperature sets the COP, which with demand gives electricity and cost.

4.2 Ground-truth hierarchy

Data sources are ranked explicitly, and the ranking governs their use. The live plant measurements are the highest authority and the sole basis for validation. The plant schematic is authoritative for topology and for design values such as the hot-loop temperature. The vendor's design figures are secondary cross-reference only, never used to calibrate the model. Where the model and the vendor disagree as on the COP before the flow correction the discrepancy is reported and its origin diagnosed, not removed by tuning.

4.3 Governing equations

4.3.1 Collector efficiency (ISO 9806)

The collector converts incident irradiance into heat with an efficiency that declines as it runs hotter relative to ambient, because a hotter absorber loses more to convection and radiation. The ISO 9806 characteristic expresses this, scaled by a field-correction factor k:

$$\eta = (\eta_0 - a_1 \cdot \frac{T_{in} - T_{amb}}{G}) \cdot k \qquad (1)$$

Here $\eta_0 = 0.4112$ is the optical efficiency and $a_1 = 12.106$ W/m²K the first-order loss coefficient. The ratio $\frac{T_{in} - T_{amb}}{G}$ is the reduced temperature difference; as it grows, efficiency falls. The factor k, discussed in Section 5.1, scales the certified characteristic to the installed array's real output. Power and outlet temperature follow from an energy balance on the flow:

$$\dot{Q}_{pvt} = \eta \cdot A_{pvt} \cdot G \quad , \quad T_{out} = T_{in} + \frac{\dot{Q}_{pvt}}{(\dot{m} \cdot c_p)} \qquad (2, 3)$$

4.3.2 Seawater exchanger (effectiveness method)

The seawater exchanger transfers heat from the Harbour water to the loop across a titanium plate. Its performance is captured by the effectiveness method, in which the heat transferred is a fixed fraction the effectiveness ε_{hex} of the maximum thermodynamically possible, bounded by the minimum-capacity stream:

$$Q_{sea} = \dot{m}_{loop} \cdot c_p \cdot \varepsilon_{hex} \cdot (T_{sea} - T_{return}) \qquad (4)$$

with $\varepsilon_{hex} = 0.60$. The seawater flow (~52 kg/s) exceeds the loop flow (~35 kg/s), so the loop is the minimum-capacity stream and governs the transfer; the seawater acts as a near-constant-temperature reservoir at T_{sea} , and its flow does not appear explicitly.

4.3.3 Heat-pump coefficient of performance

A heat pump delivers more heat than the work it consumes by moving heat from source to load; the ratio is the COP, limited by the temperature lift. Real machines achieve a fraction of the Carnot ceiling through second-law efficiency η_c :

$$COP = \frac{\eta_c \cdot T_{hot,K}}{max(T_{hot,K} - T_{src,K}, 1)}, \quad (5)$$

with η_c = 0.45 and temperatures in kelvin. The hot *side* T_{hot} = 45 °C is the schematic design value. The coefficient of performance is computed directly from this Carnot-limited expression without an upper cap, so the model reports the efficiency implied by the physics rather than an imposed ceiling. A minimal 1 K floor is retained on the lift purely to prevent the mathematical singularity as the lift approaches zero; since the source never exceeds about 29 °C against the 45 °C supply, the true lift stays above 16 K all year and this floor never binds in practice. It is a numerical safeguard, not a performance limit. This equation is the mechanism through which the free source heat pays off: every degree by which seawater and solar warm the source reduces the lift, raising the COP. Electricity is demand divided by COP:

$$\dot{W}_{elec} = \frac{Q_{demand}}{COP} \quad (6)$$

4.3.4 Pool demand (four loss mechanisms)

Each pool's heat loss is the sum of four surface mechanisms, computed from its area and the weather, capped at its exchanger rating:

$$Q = A \cdot [\,(25 + 19V)(P_w - P_a) \cdot c_f + (3.1 + 4.1V)(T_p - T_a) + \varepsilon_w \sigma (T_p^4 - T_{sky}^4) + U(T_p - T_g)\,] \quad (7)$$

representing evaporation (driven by the vapour-pressure difference and scaled by wind, reduced at night by the cover factor c_f), convection, radiation (Stefan–Boltzmann with ε_w = 0.95), and conduction. This four-mechanism decomposition follows the established pool heat-loss framework [8], [9], with the wind-dependent evaporation coefficient (25 + 19V) originating in the pool-evaporation measurements of Smith et al. [10], who calibrated it against the ASHRAE Applications Handbook correlation. Saturation vapour pressure uses the Tetens equation [11], [12], P_{sat}(T) = 610.78·exp (17.27·T/(T+237.3)). For indoor pools, V is a low fixed indoor air speed, sharply reducing the convective and evaporative losses [8]

4.4 The source-loop solution and inlet-temperature provenance

The central numerical challenge is the source temperature feeding the heat pumps, because it depends on itself: the loop is cooled by heat-pump extraction, warmed by the seawater exchanger, and warmed again by the collectors, each depending on the temperature being sought. Time-stepping at hourly resolution is unstable, because the seawater exchange moves many times the loop's hourly heat capacity. The resolution rests on a physical observation: the loop fluid turns over in a few minutes, far faster than the one-hour step, and a buffer tank further stabilises the unit against sub-hourly fluctuations. The loop therefore reaches steady state well within each hour, and the model solves that steady balance directly by fast fixed-point iteration applying the heat-pump extraction, seawater warming, and PVT warming in turn until the temperature converges.

A methodological point concerns the collector inlet temperature, which differs deliberately between validation and the annual run. In the PVT validation (Section 5.1) the collector inlet

is the measured dashboard value, so the comparison isolates the collector physics. In the annual model the inlet is not a measured input it is the loop temperature emerging from the seawater exchanger, computed within the fixed-point solve and determined self-consistently each hour. The annual model therefore generates its own inlet conditions from physics rather than borrowing measurements, which is the more rigorous arrangement and is stated explicitly because the distinction is material to interpretation.

The split of the loop between the PVT branch and the bypass is represented by a single parameter, and the branch and bypass are recombined by a flow-weighted temperature blend. This makes the collector split an explicit, adjustable model input, set to the plant's operating value of 0.25.

4.5 Assumptions and their justification

Table 2. Principal assumptions, their sources, and how each was tested.

Assumption	Value	Source	How tested
Collector efficiency form	ISO 9806	Standard	Field-validated (1.58 °C)
Field-correction factor k	0.68	Fitted to measured energy	Sensitivity (<1%)
Exchanger effectiveness	0.60	Calibrated to March	Sensitivity (~8%/0.1)
Sea temperature	18–27 °C	Calibrated	Sensitivity (~8%/3 °C)
Hot-loop temperature	45 °C	Vendor schematic	Sensitivity (~7%/3 °C)
Cold/source loop flow	~35 kg/s	Operator-confirmed	Corrected + sensitivity
PVT split	0.25	Operator (installer default)	Sensitivity (<0.5%)
Second-law efficiency	0.45	Vendor-consistent	Cross-checked vs COP
Steady-state loop	per-hour	Turnover + buffer tank	Physical justification

5. Validation

5.1 PVT field validation

The collector model was validated against nineteen days of measured field data (2–20 March 2026, 456 hourly records) of inlet temperature, flow, and thermal power from the plant's dashboard, with independent NASA POWER [6] irradiance and air-temperature data. The validation proceeds in three steps. First, the ISO 9806 model is run uncorrected (k = 1), which over-predicts the energy because the certified characteristic assumes ideal installation. Second, a single factor k is chosen so the modelled nineteen-day energy equals the measured total, yielding k = 0.68. Third, the corrected model is compared across all metrics.
The interpretation of k is physical: 0.68 means the installed array delivers about 68 % of its certified output, attributable to thermal contact resistance, soiling, and rooftop conditions the gap between specification and installed reality that a design-based study would not see. The essential point is that k scales energy only; it cannot alter the shape of the modelled temperature curve. Consequently, the agreement in outlet temperature and efficiency are independent checks the fitted factor cannot influence. These are the true validations of physics.

Table 3. PVT validation against nineteen days of field data.

Quantity	Model	Measured	Error
Total thermal energy	5,888 kWh	5,888 kWh	0.0 % (fitted)
Outlet-temperature RMSE	—	—	1.58 °C (independent)
Mean outlet temperature	32.5 °C	32.4 °C	+0.1 °C (independent)
Median efficiency	0.194	0.196	−1.1 % (independent)

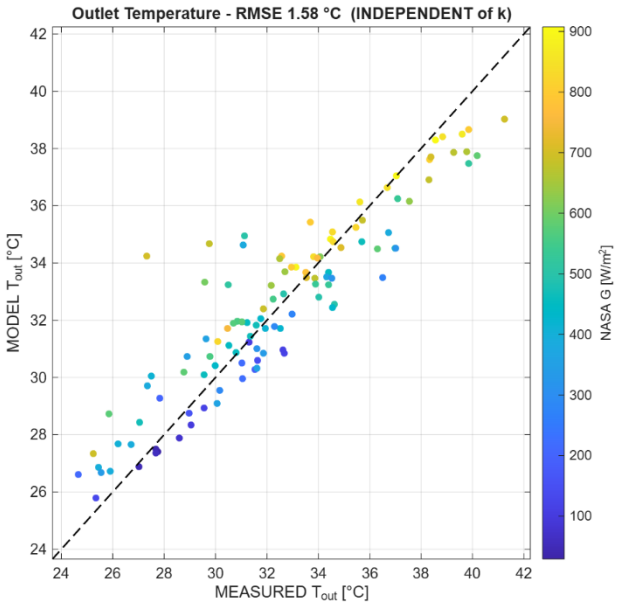


Figure 4. Modelled versus measured collector outlet temperature. Points cluster along the diagonal at an RMSE of 1.58 °C independent validation, since the fitted factor scales energy only.

The independent metrics are strong: an outlet-temperature RMSE of 1.58 °C and a median-efficiency agreement within 1.1 %. Because these cannot be affected by the energy calibration, they confirm the collector physics captures the array's real thermal behavior. Daily agreement is close on clear days and diverges only on overcast days, where the satellite irradiance cannot resolve local clouds, a data-resolution limitation, not a model error.

5.2 The flow correction and cross-check against the vendor

the source-loop flow was initially taken as the vendor design maximum of 13.5 L/s. The operator subsequently confirmed that the cold loop the cold side of the heat pumps actually runs at approximately 35 kg/s (with a commissioning target near 17 kg/s), consistent with the guiding principle that operator knowledge supersedes design figures. The base flow was corrected to 35 kg/s. This correction substantially improved agreement with both the field measurement and the vendor: the March source-temperature gap fell from 3.5 °C to 1.6 °C, and the seasonal COP rose to 5.57, within about one and a half per cent of the vendor's design 5.49 a marked improvement on the discrepancy at the original flow. The earlier gaps were therefore largely attributable to the assumed flow. The cross-check below is a sanity test, not a calibration; the vendor figures were not used to tune the model.

Table 4. Independent cross-check against vendor design figures (at the corrected 35 kg/s flow).

Quantity	This model	Vendor	Agreement
Solar loop flow	13.5 L/s (design max)	13.5 L/s	design ref
Seawater flow	52.2 L/s	52.2 L/s	exact
Heat-pump COP	5.57	5.49	close (+1.5 %)
Source temperature (March)	25.8 °C	~31 °C	consistent

6. Results

6.1 Annual demand and electricity

Running the validated, flow-corrected model across all 8,760 hours gives the annual energy picture. Total pool demand is 4,665 MWh, dominated by the outdoor 50 m pool through both its area and its wind exposure. The heat pumps meet this .at a seasonal coefficient of performance of 5.57, consuming 838 MWh of electricity, with the solar-loop pump running 45 % of hours.. The annual demand breaks down by pool as shown in Figure 5, confirming the outdoor 50 m pool as the dominant load. Over the year, seawater supplies the majority of the source heat, with the PVT contributing a smaller seasonal share (Figure 6)

Table 5. Annual results from the validated, flow-corrected model.

Metric	**Value**
Annual pool heat demand	4,665 MWh
Annual heat-pump electricity	838 MWh
Seasonal COP	5.57
Source temperature range	13.9 – 28.9 °C (mean 20.5)
Solar pump run fraction	45 %

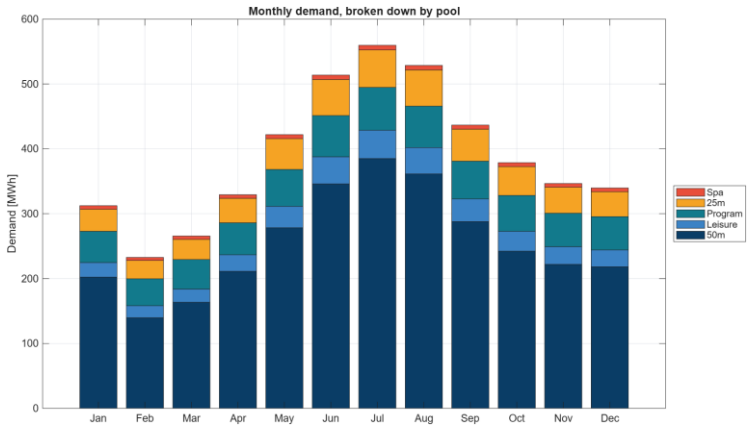


Figure 5. Monthly pool heat demand broken down by pool. The outdoor 50 m pool dominates, and total demand peaks in winter.

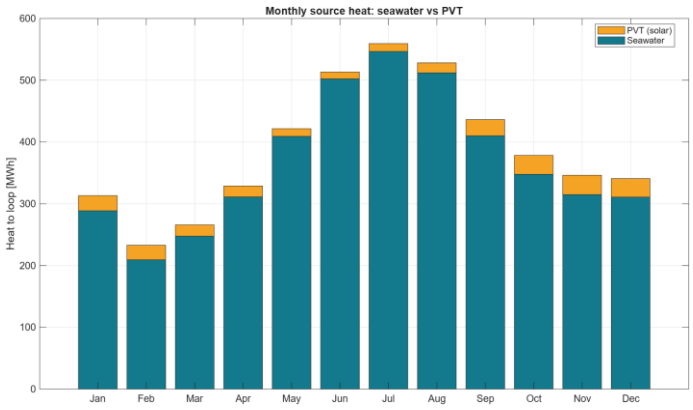


Figure 6. Monthly source heat delivered to the loop, seawater versus PVT. Seawater dominates the source heat year-round which is why the annual electricity is nearly insensitive to the PVT correction factor (Section 7).

6.2 Source temperature and the residual gap

The source temperature follows the expected seasonal cycle, ranging from about 14 °C in winter to about 29 °C in summer, with an annual mean of 20.5 °C. At the corrected flow, the modelled March source is 25.8 °C against a measured 27.4 °C, a residual gap of 1.6 °C, much reduced from the 3.5 °C at the original flow. Section 7 shows the remaining gap originates in the seawater characterisation. It is reported as it stands rather than tuned away. The predicted source temperature across the year is shown in Figure 7 against the measured March value. The resulting COP (Figure 8) peaks in summer and falls in winter as the source cools.

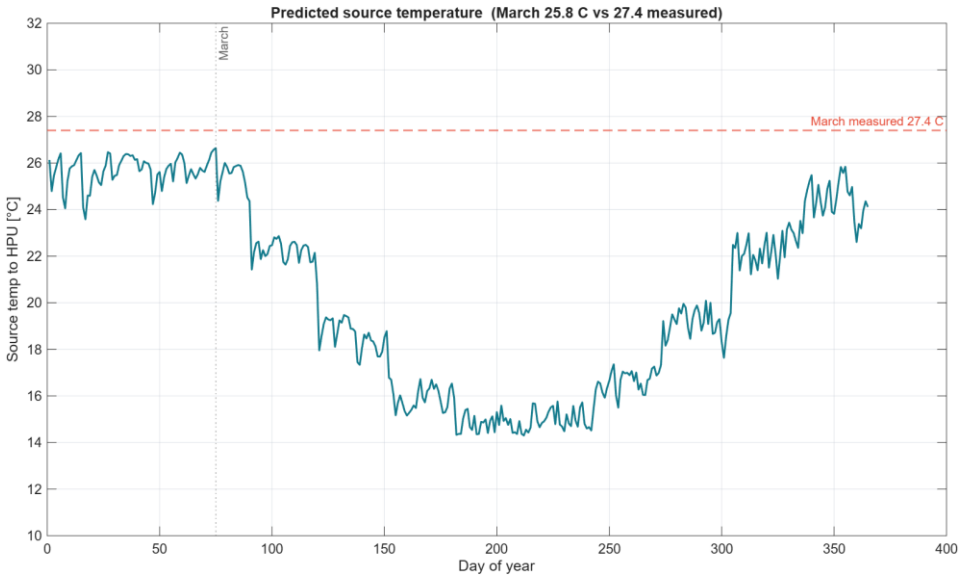


Figure 7. Predicted source temperature across the year with the measured March value marked. At the corrected flow the model runs 1.6 °C below the March measurement.

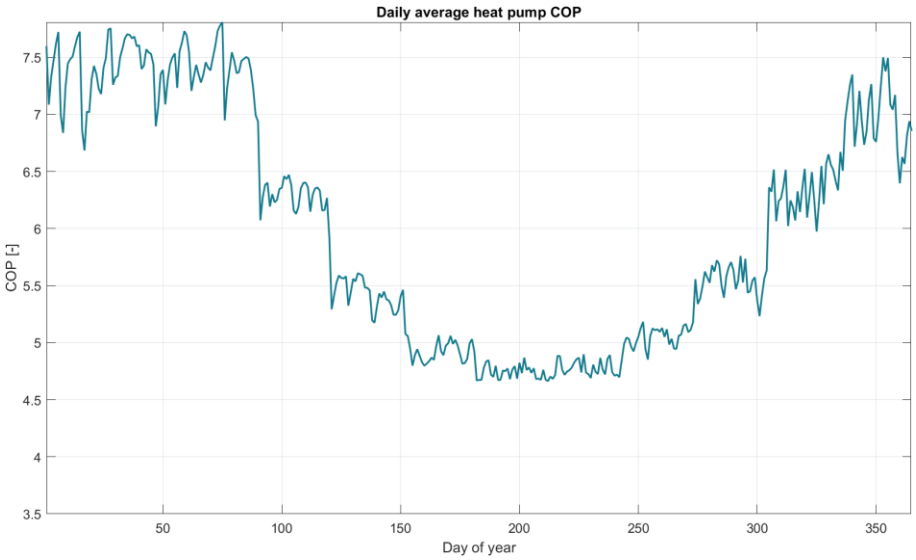


Figure 8. Daily average heat pump COP. The warm summer source yields a high COP; the cold winter source depresses it, precisely when demand is highest.

6.3 Seasonal optimization

With the model validated, it becomes a tool for asking how the plant should be operated. Two settings are controllable: the loop flow and the collector split. These were swept across a grid, with the full-season model evaluated at each combination and the electricity minimizing point identified, using a coarse-then-fine strategy that maps the landscape and then locates the optimum precisely. The baseline for comparison is the plant's current operation 35 kg/s at a 25% split. Because 100% split is not achievable (a pipe restriction confirmed by the operator), the split sweep is capped at the operator's own upper target of 50%.

The optimum differs by season, driven by a physical trade-off: raising the flow keeps the collectors cooler and warms the source, improving the COP, but the pump's own power grows with the cube of flow. Crucially the plant's current 35 kg/s is above the optimum in both seasons, and the two seasons show opposite trends (Figure 9): summer electricity falls to a low-flow optimum, while winter has an interior minimum at a higher flow.

Table 6. Seasonal optimization results versus the current 35 kg/s, 25 % operation.

Season	Optimal flow	Current	Optimized	Saving
Summer (Dec–Feb)	~15.5 kg/s	208.0 MWh	145.3 MWh	62.7 MWh (30 %)
Winter (Jun–Aug)	~23 kg/s	397.7 MWh	370.4 MWh	27.4 MWh (6.9 %)

In summer, with a warm sea and low demand, the source is already ample, so reducing the flow saves pump power with little COP penalty. The optimum is a low flow near 15.5 kg/s, saving about 63 MWh. In winter, with a cold sea and high demand, more flow is needed to extract source heat, so the optimum is an interior minimum near 23 kg/s, saving about 27 MWh. Operating each season at its optimum rather than the flat 35 kg/s saves on the order of 90 MWh per year at no capital cost a change of pump setting only. This independently corroborates the operator's own intention to reduce the loop flow. The practical rule is therefore to run a low flow in summer and a moderate flow in winter, in each case not below the heat pumps' minimum-flow limit. The resulting seasonal saving against current operation is shown in Figure 10.

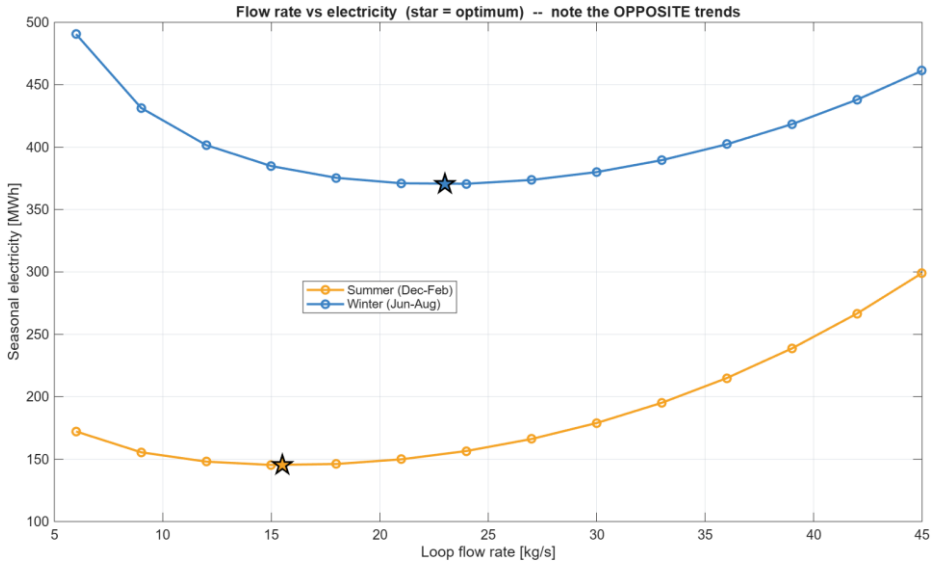


Figure 9. Seasonal electricity versus loop flow. Summer falls to a low-flow optimum near 15.5 kg/s; winter has an interior optimum near 23 kg/s. Both rise steeply toward the current 35 kg/s the plant runs above optimum in both seasons.

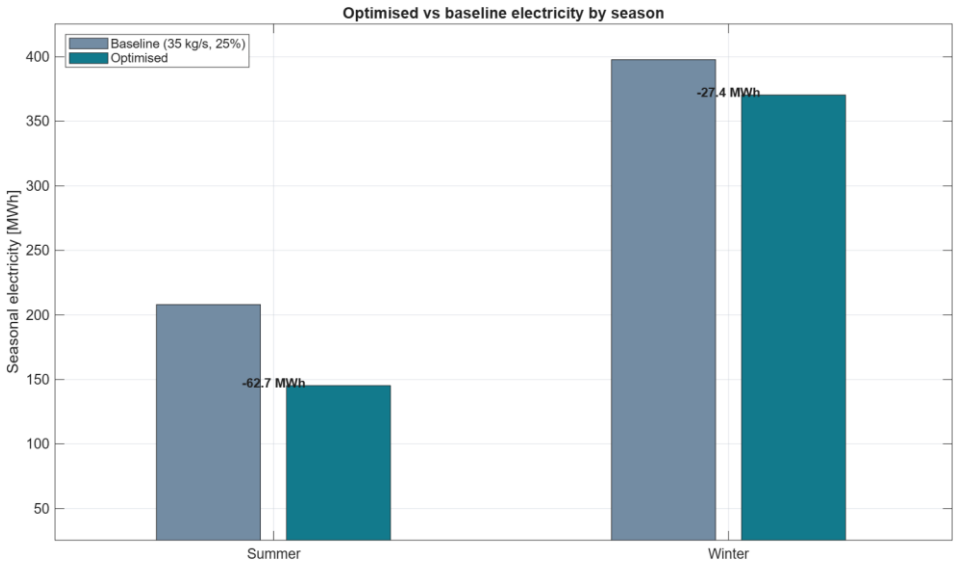


Figure 10. Optimized versus current-operation electricity by season. Reducing the loop flow to each season's optimum saves ~62.7 MWh in summer and ~27 MWh in winter.

The collector split, by contrast, alters electricity by less than 0.5 % across its entire achievable range. The operator confirms the 25 % split is an installer commissioning default rather than an engineered value, with a planned move toward 50 %; the model shows this choice carries no meaningful energy penalty, so the split can be set for operational convenience and the planned change is unobjectionable on energy grounds.

6.4 Direct-mode (Mode 2) analysis

The potential for direct solar heating the collectors warming the cooler pools directly, bypassing the heat pumps was quantified as a post-processing analysis that does not disturb the validated indirect-mode model, using the same 25 % split as the rest of the model. Across the year the collectors reach the required threshold for only a handful of hours in peak summer, with a negligible saving. This confirms the indirect mode dominates year-round, justifies modelling it in full, and means the annual electricity figure is a slight, quantified over-estimate of true consumption.

7. Sensitivity Analysis

A model's predictions are only as trustworthy as the assumptions beneath them, and the purpose of a sensitivity analysis is to establish which assumptions matter. Each principal parameter was varied about its base value with all others fixed, and the resulting annual electricity and seasonal COP recorded. The pool demand was confirmed unchanged across every run, isolating each parameter's effect on the source and heat-pump side. Each parameter was varied about its base value at the final model configuration (35 kg/s, uncapped COP), with all others fixed, and the resulting annual electricity and seasonal COP recorded. The loop flow itself is included as a parameter. The pool demand was confirmed unchanged across every run, isolating each parameter's effect on the source and heat-pump side

Table 7. Sensitivity of annual electricity and seasonal COP to the principal parameters.

Parameter	Variation	COP	Electricity (MWh)	Sensitivity
Exchanger effectiveness ε_{hex}	0.50 / 0.60 / 0.70	5.31 / 5.57 / 5.78	879 / 838 / 808	High
Sea temperature T_{sea}	−3 / base / +3 °C	5.00 / 5.57 / 6.29	933 / 838 / 742	High
Hot-loop temperature T_{hot}	42 / 45 / 48 °C	6.25 / 5.57 / 5.03	747 / 838 / 927	High
Cold-loop flow $\dot{m}$	17 / 35 kg/s	5.09 / 5.57	917 / 838	High
Field factor k	0.60 / 0.68 / 0.75	5.56 / 5.57 / 5.58	839 / 838 / 837	Negligible
PVT split	0.15 / 0.25 / 0.50	5.57 / 5.57 / 5.57	838 / 838 / 838	Negligible

7.1 Which assumptions matter

Annual electricity is highly sensitive to the exchanger effectiveness (about 4 % per 0.1), the sea temperature (about 11 % per 3 °C), the hot-loop temperature (about 10 % per 3 °C), and the loop flow.. It is almost entirely insensitive to the PVT field-correction factor (under 1 %) and to the collector split (under 0.5 %). This last pair of results is significant: the two parameters that received the most attention the fitted collector factor and the much-discussed split turn out to have negligible influence on annual electricity, because the seawater dominates the source heat over the year (Figure 4). The precise value of k, the one parameter that had to be fitted, does not materially affect the headline conclusions.

The pattern carries a clear message about where the model's uncertainty lies. The most influential parameters, the exchanger effectiveness and the sea temperature, are precisely the two that are calibrated rather than measured. The model's largest uncertainties coincide with its two unmeasured inputs, which makes them the clearest targets for future measurement.

The hot-loop temperature, though influential, is fixed by the schematic; its sensitivity represents operating variation rather than model uncertainty.

7.2 Diagnosing the source-temperature gap

The sensitivity runs also diagnose the residual source-temperature gap. It can be closed from either direction within the seawater subsystem, a warmer sea or a higher exchanger effectiveness but is unchanged by the collector factor or the demand. The gap therefore originates in the seawater characterisation, both parameters of which are calibrated. Because neither is measured, closing the gap by adjusting one to match the single measured March value would be tuning to a single data point; the conservative values were retained and the residual gap reported honestly, its origin located. Measured Harbour temperature and an exchanger specification would resolve it properly.

8. Discussion

8.1 Validation and credibility

The validation supports the model's core physics with a specific claim: the collector reproduces measured outlet temperature to within 1.58 °C, independently of the energy calibration. The finding that the installed array delivers only 68 % of certified output is a useful engineering result in itself, quantifying the shortfall between specification and installed reality.

8.2 The flow correction

The single most consequential result of interrogating the model was the flow correction. Adopting the operator's confirmed 35 kg/s in place of the vendor design maximum reduced the March gap from 3.5 to 1.6 °C and brought the COP from thirteen to three per cent of the vendor figure. Two independent discrepancies against measurement and against the vendor were thus resolved by a single correction, and both moved in the right direction, which is strong evidence that the corrected flow is closer to reality. It also illustrates the guiding principle in action: the operator's knowledge of the real plant outperformed the vendor's design value.

8.3 Simplifications relative to the reference framework

The model adopts three deliberate simplifications relative to the CoolSheet / UNSW framework, each defensible on the evidence. The collector efficiency is evaluated at the inlet rather than mean-plate temperature; the ISO 9806 linear characteristic is used rather than the TRNSYS Type 850 (Akhtar–Mullick) formulation; and the COP uses a fixed second-law efficiency rather than a part-load curve. The simpler collector forms reproduced the measured outlet temperature to within 1.58 °C, and the sensitivity analysis showed the collector correction to be nearly irrelevant to annual electricity, so the additional sophistication would not change the conclusions. Each is identified in Section 9 as a natural refinement.

8.4 The optimization result in context

The optimization gives a result that is physically grounded and immediately actionable, and its per-season structure is more informative than a single recommendation. Because the plant currently runs a flat 35 kg/s, above the optimum in both seasons, the opportunity is real and sits on both sides of the year: .a low summer flow and a moderate winter flow together save on the order of 90 MWh annually at no capital cost. The negligible influence of the split removes an entire control variable from consideration and vindicates the operator's freedom to set it as convenient.

9. Limitations and Future Work

The study's limitations are stated plainly, each paired with the refinement it motivates. Field validation limited to the PVT collector. Only the collector has measured data; the COP, seawater exchange, and annual electricity are physically grounded and vendor-cross-checked but not independently validated. Live heat-pump or whole-plant electricity data would allow the annual figures to be validated directly.

Calibrated seawater inputs. The sea temperature and exchanger effectiveness are calibrated rather than measured, and the sensitivity analysis identifies them as the dominant uncertainties and the source of the residual gap. Measured Harbour temperature and an exchanger specification would resolve it.

Commissioning-stage flows. The plant was under commissioning; the cold-loop flow is a design ~35 kg/s with a target near 17 kg/s, and the PVT-branch flow is not yet metered even by the operator. The base flow and its sensitivity are reported to reflect this, but a settled, metered flow would remove the residual uncertainty.

Collector and COP formulations. A mean-plate collector temperature, the TRNSYS Type 850 (Akhtar–Mullick) formulation, and a part-load COP curve are the natural refinements, each described in the reference framework.

Independent second model. A parallel model for instance, a mean-plate collector formulation would permit a formal cross-model comparison and further test the simplifications adopted here.

Minimum-flow limit and pool-area anomaly. The summer low-flow recommendation requires the heat pumps' minimum safe flow as a floor, to be confirmed with the operator; and the 25 m pool's implausible area-to-volume ratio in the vendor data warrants confirmation, as it raises the computed demand.

10. Conclusion

This work delivered an independent, first-principles MATLAB model of the North Sydney Olympic Pool's solar-assisted seawater heat-pump plant, built on a single-source parameter architecture, validated against the plant's own field data, and cross-checked against the vendor's design study. The collector stage reproduces measured outlet temperature to within 1.58 °C independently of any energy calibration. Correcting the source-loop flow to the operator-confirmed value reduced the March source-temperature gap to 1.6 °C and brought the seasonal COP to 5.57, within about one and a half per cent of the vendor figure; the full model gives an annual demand of 4,665 MWh met by 838 MWh of electricity. A sensitivity analysis established that the model is governed by the seawater subsystem and the hot-loop temperature and is insensitive to the fitted collector factor and the collector split, and it diagnosed the residual gap as originating in the calibrated seawater inputs. The central practical result is a zero-capital seasonal control strategy: because the plant currently runs above its optimum flow in both seasons, reducing the loop flow to roughly 15.5 kg/s in summer and 23 kg/s in winter saves on the order of 90 MWh of electricity per year, a finding that independently corroborates the operator's own intention to lower the flow. Throughout, the model's assumptions, data sources, and residual gaps have been stated transparently and, where possible, diagnosed rather than concealed including the flow correction that resolved two independent discrepancies at once. The outcome is a validated, inspectable, and stress-tested tool for understanding and improving the plant, together with a concrete, capital-free efficiency recommendation and a clear map of where the next increment of measured data would most improve the model.

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