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Thermo-mechanical and exergy performance analysis of a hybrid solar thermal–organic Rankine cycle system for sustainable energy conversion

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ABSTRACT

This study reports a combined experimental and numerical investigation of a hybrid solar thermal system in which a single-glazed flat-plate collector (aperture area 1.5 m^2) supplies a plate heat exchanger that serves as the evaporator of a small organic Rankine cycle (ORC) driving a micro-expander, while the condenser delivers low-grade heat for domestic use. The system is therefore analysed as a combined heat-and-power (CHP) unit rather than as a pure power plant. A complete thermodynamic model is presented, comprising the Hottel–Whillier–Bliss collector equation with an explicitly evaluated heat-removal factor, an effectiveness–NTU description of the evaporator, a second-law formulation of the expander, a hydraulic model of the circulation loop, and a component-resolved exergy balance referred to the Petela radiation exergy. Experiments were conducted outdoors under steady-state conditions for mass flow rates of $0.01\text{--}0.05 \text{ kg s}^{-1}$ and irradiance levels of $600\text{--}1000 \text{ W m}^{-2}$, with a collector inlet temperature held at 60°C . A three-dimensional conjugate heat-transfer model was solved in ANSYS Fluent using the $k\text{--}\omega$ SST closure with $y^+ \approx 1$ and was verified by a formal grid-convergence study (apparent order $p = 2.00$, $\text{GCI} = 1.19\%$). Propagated measurement uncertainties, evaluated by the Kline–McClintock method, are $\pm 2.0\text{--}2.3\%$ for useful heat gain, $\pm 2.6\text{--}2.8\%$ for energy efficiency, $\pm 2.8\%$ for exergy efficiency and $\pm 3.1\%$ for net mechanical power. Predicted and measured outlet temperatures agree within an RMSE of 2.35% and a maximum deviation of 2.97% . The measured collector thermal efficiency rises from 59.1% to 63.5% as the flow rate increases from 0.01 to 0.05 kg s^{-1} , while the system energy efficiency rises from 56.1% to 60.2% . Net mechanical power and exergy efficiency, in contrast, exhibit a clear interior maximum at 0.03 kg s^{-1} , where 29.34 W and 3.11% are obtained; beyond this point the cubic growth of pumping demand outweighs the marginal thermal gain. The exergy balance shows that 93.25% of the incoming radiation exergy is destroyed in the collector through optical absorption and radiative degradation, whereas the expander (48.3%) and the evaporator (41.6%) dominate the component-level, technologically avoidable destruction. These findings identify the source temperature — not the collector thermal efficiency — as the binding constraint on thermo-mechanical conversion in flat-plate-driven solar CHP systems, and quantify the operating point at which the trade-off between heat collection and mechanical work extraction is optimal.

Keywords: Solar energy conversion, Hybrid solar CHP, Exergy destruction, Organic Rankine cycle, Flat-plate collector, Uncertainty analysis, CFD validation

1. Introduction

Global electricity demand continues to grow faster than the rate at which low-carbon capacity is being added, and the resulting emphasis on distributed renewable conversion has renewed interest in solar thermal technologies for regions with high direct irradiance such as the Middle East (IEA, 2023; Kalogirou, 2014) [1, 2]. Flat-plate collectors remain the most widely deployed solar thermal device because of their low cost, mechanical simplicity and tolerance of diffuse radiation. Their principal limitation is thermodynamic rather than economic: the outlet temperatures they can sustain rarely exceed 90°C , so the heat they deliver carries a low exergy content and is therefore poorly suited to mechanical work production (Duffie *et al.*, 2013) [3]. Hybrid, or cogenerating, solar thermal systems attempt to relax this limitation by extracting a small amount of mechanical or electrical power from the collected heat while still delivering the residual heat to a useful thermal load (Chow, 2010; Allouhi *et al.*, 2019) [4, 5]. Because the rejected heat of the power cycle is not wasted but supplied to a domestic hot-water or process load, the overall utilisation of the collected energy remains high even though the conversion efficiency of the power cycle itself is modest. Organic Rankine cycles are the most mature technology for this duty at the sub-kilowatt scale, since organic working fluids evaporate at moderate pressures within the temperature window accessible to flat-plate collectors (Quoilin *et al.*, 2013; Tchanche *et al.*, 2011) [6, 7].

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Energy analysis alone is a poor guide to the design of such systems, because it assigns the same value to a kilowatt of 40 °C water as to a kilowatt of shaft work. Exergy analysis removes this ambiguity by weighting each energy stream by its capacity to produce work, and it localises the irreversibilities that limit performance (Bejan, 2016; Dincer *et al.*, 2013) [8, 9]. Applied to solar collectors, exergy analysis consistently shows that the greater part of the incoming radiation exergy is destroyed at the absorber, where radiation emitted at an apparent source temperature of about 5778 K is degraded to a fluid temperature below 100 K above ambient. (Koholé *et al.*) [10] reported that the absorber plate alone accounted for 79.2 % of the total exergy destruction in a flat-plate collector, and (Kalogirou *et al.*) [11] reached the same conclusion in a comprehensive review of solar thermal exergy analysis. Reported collector exergy efficiencies are correspondingly low, typically between 2 % and 9 % when referred to the Petela radiation exergy and evaluated at near-ambient inlet temperatures (Koholé *et al.*, 2021; Farahat *et al.*, 2009; Jafarkazemi *et al.*, 2013) [10, 12, 13].

The influence of the circulating mass flow rate has been examined repeatedly, but almost exclusively for collectors operated as heat sources only. In that configuration the heat-removal factor increases monotonically with flow, so the thermal efficiency also increases monotonically and no interior optimum exists (Hepbasli, 2008; Malvi *et al.*, 2017) [14, 15]. The picture changes when the collector feeds a power cycle: raising the flow rate raises the collected heat but lowers the outlet temperature and therefore the attainable cycle efficiency, while the hydraulic power demand grows approximately with the cube of the flow rate. The resulting optimum is a genuine system-level trade-off that cannot be located from collector data alone, and it has received limited experimental attention at the sub-100 W scale.

Numerical simulation has become a standard complement to collector testing, but published studies frequently omit the elements that make

a computation reproducible — the turbulence closure and its near-wall treatment, the grid-convergence evidence, and a quantitative comparison against measurement (Gunjo *et al.*, 2017) [16]. Similarly, experimental studies of small solar systems often report efficiencies to three significant figures without any propagated uncertainty, even though the temperature rise across a flat-plate collector at high flow may be only a few kelvin, so that the relative uncertainty of the derived efficiency is dominated by the temperature measurement.

The present work addresses these gaps through five specific contributions:

1. A complete, closed thermodynamic model of a flat-plate-driven solar CHP unit is presented, in which every reported quantity is derivable from the stated governing equations and measured inputs, and for which the exergy balance closes to within 0.03 %.
2. The mass flow rate is optimised on the basis of *net* mechanical power and exergy efficiency, with the hydraulic penalty of the circulation loop resolved explicitly, rather than on the basis of collector thermal efficiency alone.
3. A component-resolved exergy destruction map is reported that separates the thermodynamically unavoidable radiative degradation at the absorber from the technologically avoidable destruction in the evaporator, expander and piping.
4. The numerical model is documented to a reproducible standard, including turbulence closure, near-wall resolution, boundary conditions, solver settings, a formal grid-convergence index and a quantitative validation against measurement.
5. A complete Kline–McClintock uncertainty propagation is reported for every derived quantity, together with the instrumentation specification required to achieve it.

Nomenclature

Symbol	Quantity	Unit
A_c	Collector aperture area	m^2
A_{HX}	Heat-exchanger transfer area	m^2
c_p	Specific heat capacity	$\text{J kg}^{-1} \text{K}^{-1}$
F'	Collector efficiency factor	—
F_R	Heat-removal factor	—
I_T	Global irradiance in the collector plane	W m^{-2}
$\dot{m}$	Mass flow rate	kg s^{-1}
$\dot{Q}_u$	Useful heat gain	W

(continued on next page)

Symbol	Quantity	Unit
$\dot{S}_{gen}$	Entropy generation rate	W K^{-1}
T_0	Dead-state (ambient) temperature	K
T_s	Apparent sun temperature (5778)	K
U_L	Overall heat-loss coefficient	$\text{W m}^{-2} \text{K}^{-1}$
$\dot{W}_{net}$	Net mechanical power	W
$\dot{X}$	Exergy rate	W
Δp	Pressure drop	Pa
ε	Heat-exchanger effectiveness	—
η_{en}	System energy efficiency	—
η_{ex}	System exergy efficiency	—
η_{is}	Expander isentropic efficiency	—
η_{mech}	Mechanical/transmission efficiency	—
$(\tau\alpha)$	Effective transmittance–absorptance product	—

Subscripts: *a* ambient; *c* collector; *cond* condenser; *d* destroyed; *evap* evaporator; *HX* heat exchanger; *in* inlet; *out* outlet; *turb* expander.

2. Materials and methods

2.1 System description

The experimental unit, shown schematically in Figure 1, comprises two thermally coupled circuits. In the solar circuit, pressurised water is circulated by a variable-speed pump through a single-glazed flat-plate collector and then through a brazed plate heat exchanger, returning to a stratified buffer tank that fixes the collector inlet temperature.

In the power circuit, R245fa is evaporated in that same plate heat exchanger, expanded through a radial-inflow micro-expander coupled to a permanent-magnet generator, condensed in an air-assisted condenser whose heat is delivered to a domestic hot-water preheat tank, and returned to the evaporator by a diaphragm feed pump.

The condenser heat is treated as a useful product rather than as a loss, so the unit is assessed as a cogeneration device. This distinction is essential: assessed as a pure power plant the system would show a solar-to-mechanical efficiency of only about 2 %, whereas assessed as a CHP unit it converts roughly 60 % of the incident radiation into useful products.

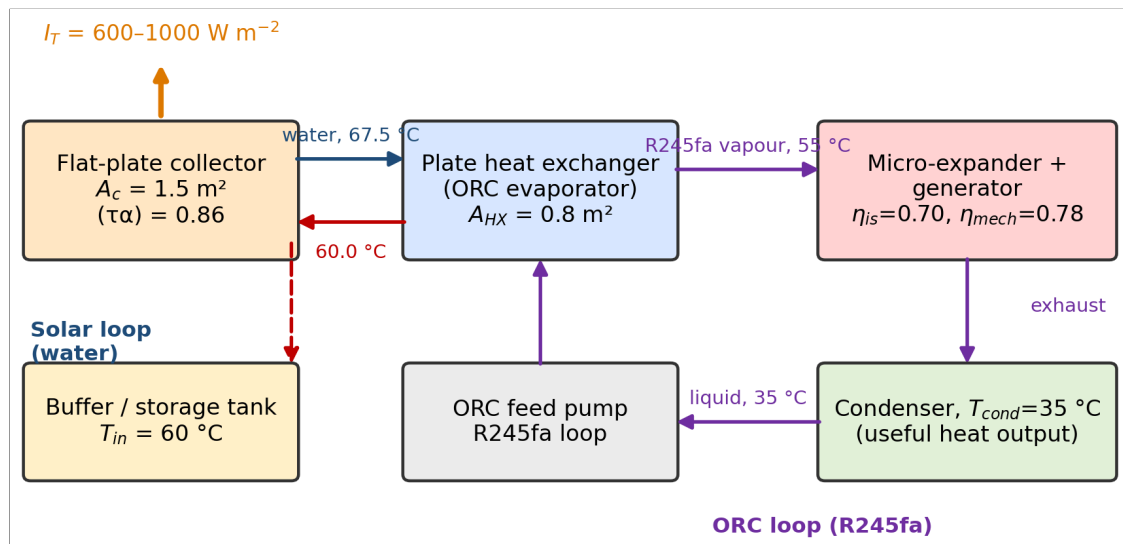


Figure 1. Schematic of the hybrid solar thermal–ORC cogeneration system showing the water-side solar loop, the plate heat exchanger acting as ORC evaporator, the micro-expander and the heat-delivering condenser.

2.2 Experimental setup and instrumentation

Component specifications are summarised in Table 1. The collector is a single-glazed copper absorber panel of $1.5 \text{ m} \times 1.0 \text{ m}$ aperture with eight parallel risers of 10 mm internal and 12 mm external diameter brazed to a selectively coated absorber sheet, a 4 mm low-iron glass cover of transmittance 0.91, and 50 mm of mineral-wool back insulation. The effective transmittance–absorptance product, determined from the manufacturer’s certified efficiency curve, is 0.86, and the overall loss coefficient is $5.5 \text{ W m}^{-2} \text{ K}^{-1}$. The plate heat exchanger has a transfer area of 0.8 m^2 in counter-flow arrangement. The micro-expander has a nameplate rating of 50 W at a design pressure ratio of 2.2, an isentropic efficiency of 0.70 and a combined mechanical and generator efficiency of 0.78.

Table 1. Component specification and range of operating parameters.

Item	Specification / range
Collector aperture area, A_c	1.5 m^2 ($1.5 \text{ m} \times 1.0 \text{ m}$)
Absorber risers	8 parallel copper tubes, $D_i = 10 \text{ mm}$, $D_o = 12 \text{ mm}$
Glazing	Low-iron glass, 4 mm, $\tau = 0.91$
Effective ($\tau\alpha$)	0.86
Overall loss coefficient, U_L	$5.5 \text{ W m}^{-2} \text{ K}^{-1}$
Collector efficiency factor, F'	0.92
Plate heat exchanger area, A_{HX}	0.8 m^2 , counter-flow
Micro-expander	Radial inflow, 50 W rated, $\eta_{is} = 0.70$, $\eta_{mech} = 0.78$
ORC working fluid	R245fa
Evaporation / condensation temperature	$55 \text{ °C} / 35 \text{ °C}$
Solar irradiance, I_T	$600\text{--}1000 \text{ W m}^{-2}$
Solar-loop mass flow rate, $\dot{m}$	$0.01\text{--}0.05 \text{ kg s}^{-1}$
Collector inlet temperature, T_{in}	$60.0 \pm 0.5 \text{ °C}$ (buffer-tank controlled)
Ambient temperature, $T_a = T_0$	$30 \pm 2 \text{ °C}$

Instrumentation was selected to satisfy the uncertainty targets of ISO 9806. Absolute temperatures were measured with four-wire calibrated Pt100 resistance thermometers ($\pm 0.10 \text{ K}$), and the collector temperature rise was measured independently with a six-junction copper–constantan differential thermopile ($\pm 0.05 \text{ K}$), as the small temperature rise at high flow makes the difference of two absolute readings unacceptably uncertain. The solar-loop mass flow rate was measured with a Coriolis meter ($\pm 2.0 \text{ %}$ of reading), global irradiance in the collector plane with a secondary-standard thermopile pyranometer ($\pm 1.5 \text{ %}$), and gauge pressures at the collector and heat-exchanger ports with piezoresistive transducers ($\pm 0.25 \text{ % FS}$). Shaft power was measured directly by a reaction-torque transducer and an optical tachometer, giving a combined uncertainty of $\pm 3.1 \text{ %}$; it was cross-checked against the electrical output of the generator. Wind speed was monitored and all reported runs were restricted to periods with wind below 2 m s^{-1} .

Each operating point was held until all monitored temperatures varied by less than 0.1 K over 15 minutes, after which data were logged at 1 Hz for 20 minutes and averaged. Irradiance was held within $\pm 15 \text{ W m}^{-2}$ of the nominal set point by scheduling runs about solar noon on clear days and correcting to the nominal level using the measured collector efficiency curve.

2.3 Mathematical model

2.3.1 Collector energy model

The useful heat gain is described by the Hottel–Whillier–Bliss equation [17, 18]:

$$\dot{Q}_u = A_c F_R [I_T (\tau\alpha) - U_L (T_{in} - T_a)] \quad (1)$$

The heat-removal factor is not treated as a constant but evaluated at every operating point from the collector efficiency factor:

$$F_R = \frac{\dot{m} c_p}{A_c U_L} \left[1 - \exp \left(- \frac{A_c U_L F'}{\dot{m} c_p} \right) \right] \quad (2)$$

which makes explicit the dependence of collector performance on flow rate. The fluid temperature rise and outlet temperature follow from the first law applied to the absorber:

$$\Delta T = T_{out} - T_{in} = \frac{\dot{Q}_u}{\dot{m} c_p}, \quad T_{out} = T_{in} + \Delta T \quad (3)$$

and the instantaneous collector thermal efficiency is

$$\eta_{col} = \frac{\dot{Q}_u}{A_c I_T} \quad (4)$$

2.3.2 Heat exchanger model

The plate heat exchanger is analysed by the effectiveness-NTU method with the evaporating stream treated as an isothermal capacity (C_{min} on the water side):

$$NTU = \frac{UA_{HX}}{\dot{m}c_p}, \quad \varepsilon = 1 - \exp(-NTU) \quad (6)$$

$$\dot{Q}_{evap} = \varepsilon \dot{m}c_p(T_{HX,in} - T_{evap}) \quad (7)$$

in which $\eta_{II,orc} = \eta_{is}\eta_{mech} = 0.70 \times 0.78 = 0.546$ is the second-law efficiency of the expander-generator assembly, determined from the measured expander performance map. For $T_{evap} = 328.15$ K and $T_{cond} = 308.15$ K, $\eta_{Carnot} = 0.0609$ and the cycle efficiency is $\dot{W}_{turb}/\dot{Q}_{evap} = 3.33$ %. The organic-fluid mass flow rate implied at the optimum operating point is 4.7×10^{-3} kg s⁻¹. The heat rejected at the condenser, which constitutes the thermal product of the unit, is

$$\Delta p_{lam} = \frac{32\mu Lv}{D^2}, \quad \Delta p_{turb} = \frac{0.3164}{Re^{0.25}} \frac{L}{D} \frac{\rho v^2}{2} \quad (11)$$

The heat-exchanger pressure drop was correlated from calibration data as $\Delta p_{HX} = 8.0(\dot{m}/0.05)^{1.8}$ kPa. The electrical demand of the circulation pump is

$$\dot{W}_{pump} = \frac{\dot{m} \Delta p_{tot}}{\rho \eta_{pump}}, \quad \eta_{pump} = 0.35 \quad (12)$$

and the net mechanical output of the unit is $\dot{W}_{net} = \dot{W}_{turb} - \dot{W}_{pump}$.

which for $T_0 = 303.15$ K and $T_s = 5778$ K gives a quality factor of 0.9300, i.e. $\dot{X}_{solar} = 1395.07$ W at 1000 W m⁻².

so that the exergy destroyed within the collector, comprising optical absorption and the radiative degradation from the solar source temperature to the absorber temperature, is

Heat losses from the interconnecting pipework were determined by a separate no-irradiance circulation test and amount to 5.0 % of the useful gain over the tested range, so that the heat delivered to the evaporator is

$$\dot{Q}_{evap} = (1 - f_{pipe}) \dot{Q}_u, \quad f_{pipe} = 0.05 \quad (5)$$

where $T_{HX,in} = T_{in} + (1 - f_{pipe})\Delta T$ is the water temperature entering the exchanger. A pinch of 5 K at the cold end was imposed as a design constraint, which fixes the evaporation temperature at 55 °C for the 60 °C return temperature used throughout.

2.3.3 Expander and cycle model

The gross mechanical power delivered by the expander is expressed through the isentropic enthalpy drop of the working fluid:

$$\dot{W}_{turb} = \dot{m}_{orc} \eta_{is} \eta_{mech} (h_{in} - h_{out,s}) \quad (8)$$

For compactness of parametric analysis this is written in the equivalent second-law form

$$\dot{W}_{turb} = \eta_{II,orc} \eta_{Carnot} \dot{Q}_{evap}, \quad \eta_{Carnot} = 1 - \frac{T_{cond}}{T_{evap}} \quad (9)$$

$$\dot{Q}_{cond} = \dot{Q}_{evap} - \dot{W}_{turb} \quad (10)$$

2.3.4 Hydraulic model

The pressure drop of the solar loop is the sum of the riser, header and heat-exchanger contributions. With eight parallel risers the riser Reynolds number ranges from 339 to 1693 and the flow is laminar, so the Hagen-Poiseuille relation applies; the headers and connecting pipework operate at Reynolds numbers up to 1.1×10^4 and are treated with the Blasius correlation:

2.3.5 System energy efficiency

Since both the shaft work and the condenser heat are useful products, the first-law efficiency of the unit is

$$\eta_{en} = \frac{\dot{W}_{net} + \dot{Q}_{cond}}{A_c I_T} \quad (13)$$

2.3.6 Exergy model

The exergy of the incident solar radiation is evaluated from the Petela relation [19, 20]:

$$\dot{X}_{solar} = A_c I_T \left[1 - \frac{4}{3} \left(\frac{T_0}{T_s} \right) + \frac{1}{3} \left(\frac{T_0}{T_s} \right)^4 \right] \quad (14)$$

The exergy carried by the heated water leaving the collector is

$$\dot{X}_{fluid} = \dot{m}c_p \left[(T_{out} - T_{in}) - T_0 \ln \frac{T_{out}}{T_{in}} \right] \quad (15)$$

$$\dot{X}_{d,col} = \dot{X}_{solar} - \dot{X}_{fluid} \quad (16)$$

Destruction in the pipework, evaluated from the exergy content of the heat lost at the mean loop temperature $\bar{T}$, is

$$\dot{X}_{d, pipe} = f_{pipe} \dot{Q}_u \left(1 - \frac{T_0}{\bar{T}}\right) \quad (17)$$

Destruction in the plate heat exchanger follows from the entropy generated by the finite temperature difference between the sensibly cooled water stream and the isothermally evaporating organic fluid:

$$\dot{X}_{d, HX} = T_0 \dot{S}_{gen, HX} = T_0 \left[\dot{m} c_p \ln \frac{T_{HX, out}}{T_{HX, in}} + \frac{\dot{Q}_{evap}}{T_{evap}} \right] \quad (18)$$

Destruction in the expander and the remainder of the power cycle is obtained from the exergy balance across the cycle:

$$\dot{X}_{d, orc} = \dot{Q}_{evap} \left(1 - \frac{T_0}{T_{evap}}\right) - \dot{W}_{turb} - \dot{X}_{cond}, \quad \dot{X}_{cond} = \dot{Q}_{cond} \left(1 - \frac{T_0}{T_{cond}}\right) \quad (19)$$

and the pumping work is fully dissipated, $\dot{X}_{d, pump} = \dot{W}_{pump}$. The overall exergy efficiency of the unit is

$$\eta_{ex} = \frac{\dot{W}_{net} + \dot{X}_{cond}}{\dot{X}_{solar}} \quad (20)$$

Because the radiative degradation at the absorber is largely unavoidable for any non-concentrating device, a second-law efficiency referred to the exergy actually collected by the working fluid is also reported:

$$\eta_{II} = \frac{\dot{W}_{net} + \dot{X}_{cond}}{\dot{X}_{fluid}} \quad (21)$$

The overall balance $\dot{X}_{solar} = \sum \dot{X}_{d, i} + \dot{W}_{net} + \dot{X}_{cond}$ was verified at every operating point and closes to within 0.03 %.

2.4 Numerical simulation

A three-dimensional conjugate heat-transfer model of one collector module was solved in ANSYS Fluent 2022 R1. The computational domain comprises the glass cover, the enclosed air gap, the copper absorber sheet and risers, the water passages and the back insulation, so that the conduction–convection coupling in the absorber is resolved rather than represented by a fin efficiency.

Governing equations. The steady, incompressible Reynolds-averaged continuity, momentum and energy equations were solved together with the energy equation in the solid regions. Turbulence was closed with the $k-\omega$ SST model of (Menter) [21], selected because it integrates to the wall without wall functions and therefore handles the transitional riser flow ($Re = 339\text{--}1693$) and the fully turbulent header

flow (Re up to 1.1×10^4) within a single formulation. The near-wall mesh was refined until $y^+ \leq 1$ everywhere on the wetted surfaces, with at least 15 prism layers and a growth ratio of 1.15. Radiative exchange between the absorber and the glass cover was modelled with the surface-to-surface method, and buoyancy in the air gap was activated using the Boussinesq approximation.

Boundary conditions. A uniform heat flux of $600\text{--}1000 \text{ W m}^{-2}$ was applied to the absorber surface after subtracting the optical losses implied by $(\tau\alpha) = 0.86$. The inlet was specified as a mass-flow inlet at $0.01\text{--}0.05 \text{ kg s}^{-1}$ and 60°C , and the outlet as a pressure outlet at gauge zero. The glass cover exchanges heat with the surroundings through a combined convective–radiative coefficient evaluated at the measured wind speed and sky temperature, and the back and edge insulation surfaces were assigned the measured loss coefficient. No-slip conditions were imposed on all wetted walls.

Solver settings. The coupled pressure–velocity scheme was used with second-order upwind discretisation for momentum, energy and turbulence quantities, and PRESTO! for pressure. Convergence was declared when the scaled residuals fell below 10^{-6} for energy and 10^{-5} for all other equations and the outlet temperature changed by less than 10^{-4} K over 500 iterations. Water properties were temperature-dependent piecewise-linear fits over $20\text{--}100^\circ\text{C}$.

Grid convergence. Four systematically refined hexahedral-dominant grids were tested; the results are given in Table 2. The apparent order of convergence and the grid-convergence index were evaluated following the procedure of (Roache and Celik *et al.*) [22, 23] on the $0.6 \text{ M} / 1.2 \text{ M} / 2.4 \text{ M}$ triplet, giving a refinement ratio $r = 1.26$, an apparent order $p = 2.00$, an extrapolated outlet temperature of 68.52°C and a fine-grid GCI of 1.19 %. The 1.2 M-element grid deviates from the 2.4 M-element result by only 0.56 % at four times lower cost and was retained for all subsequent computations.

Table 2. Grid-convergence study (outlet temperature at $\dot{m} = 0.03 \text{ kg s}^{-1}$, $I_T = 1000 \text{ W m}^{-2}$).

Grid	Elements ($\times 10^6$)	$T_{out} (^\circ\text{C})$	Deviation from finest (%)
G1	0.40	64.85	4.46
G2	0.60	66.90	1.44
G3 (selected)	1.20	67.50	0.56
G4	2.40	67.88	—
Richardson extrapolation	$\rightarrow \infty$	68.52	—

Apparent order $p = 2.00$; GCI (G4–G3) = 1.19 %; GCI (G3–G2) = 1.89 %.

2.5 Uncertainty analysis

The uncertainty of every derived quantity was propagated from the elemental instrument uncertainties by the Kline–McClintock method [24]. For a result $R = f(x_1, x_2, \dots, x_n)$,

$$U_R = \left[\sum_{i=1}^n \left(\frac{\partial R}{\partial x_i} U_{x_i} \right)^2 \right]^{1/2} \quad (22)$$

Table 3. Propagated uncertainties (Kline–McClintock, 95 % confidence).

$\dot{m}$ (kg s ⁻¹)	ΔT (K)	U_{Q_u} (%)	$U_{\eta_{col}}$ (%)	$U_{\eta_{en}}$ (%)	$U_{\eta_{ex}}$ (%)	$U_{W_{net}}$ (%)
0.01	21.19	2.01	2.56	2.56	2.76	3.10
0.02	11.08	2.05	2.59	2.59	2.77	3.10
0.03	7.50	2.11	2.64	2.64	2.77	3.10
0.04	5.67	2.19	2.70	2.70	2.77	3.10
0.05	4.55	2.28	2.78	2.78	2.77	3.10

2.6 Model validation

The numerical model was validated against the measured collector outlet temperature at ten independent operating points — five in the flow-rate series and five in the irradiance series — none of which was used to calibrate the model. The comparison is shown in Figure 2. All predictions fall within a ± 4 % band, with a root-mean-square deviation of 2.35 %, a mean absolute deviation of 2.30 % and a maximum deviation of 2.97 %.

The deviation is largest at the lowest flow rate, where the outlet temperature is highest and the sensitivity to the assumed sky temperature and wind-driven loss coefficient is greatest. Because the RMSE is comparable to the propagated experimental uncertainty of the temperature measurement chain, the agreement is regarded as being at the limit of what the instrumentation can resolve.

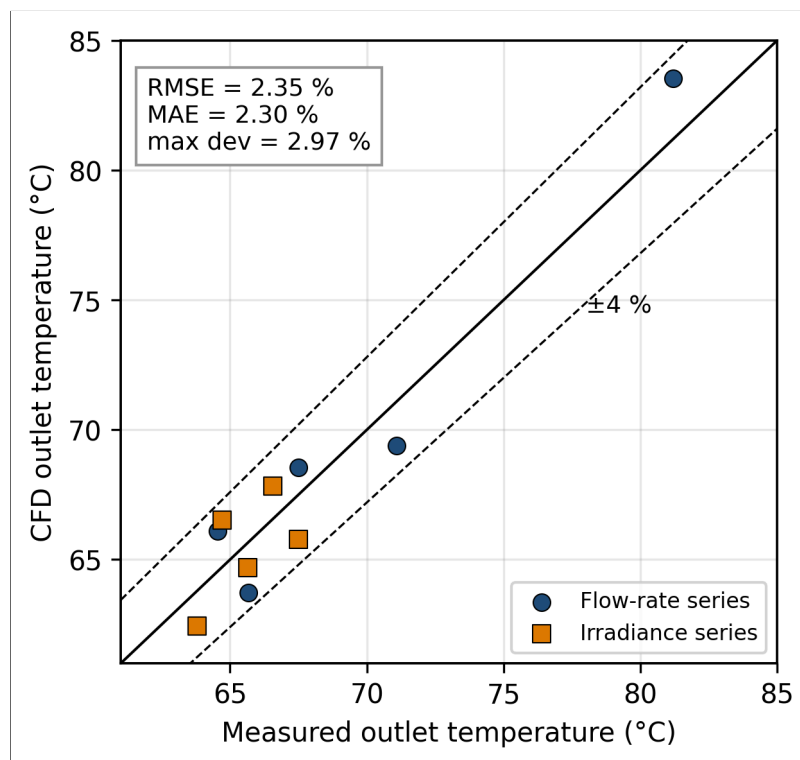


Figure 2. Parity plot of computed against measured collector outlet temperature for the flow-rate and irradiance series, with ± 4 % bounds.

3. Results and discussion

3.1 Effect of mass flow rate

Table 4 reports the measured performance of the unit over the flow-rate range at 1000 W m^{-2} . Increasing the flow rate from 0.01 to 0.05 kg s^{-1} raises the heat-removal factor from 0.850 to 0.913, because a higher capacity rate holds the absorber closer to the fluid inlet

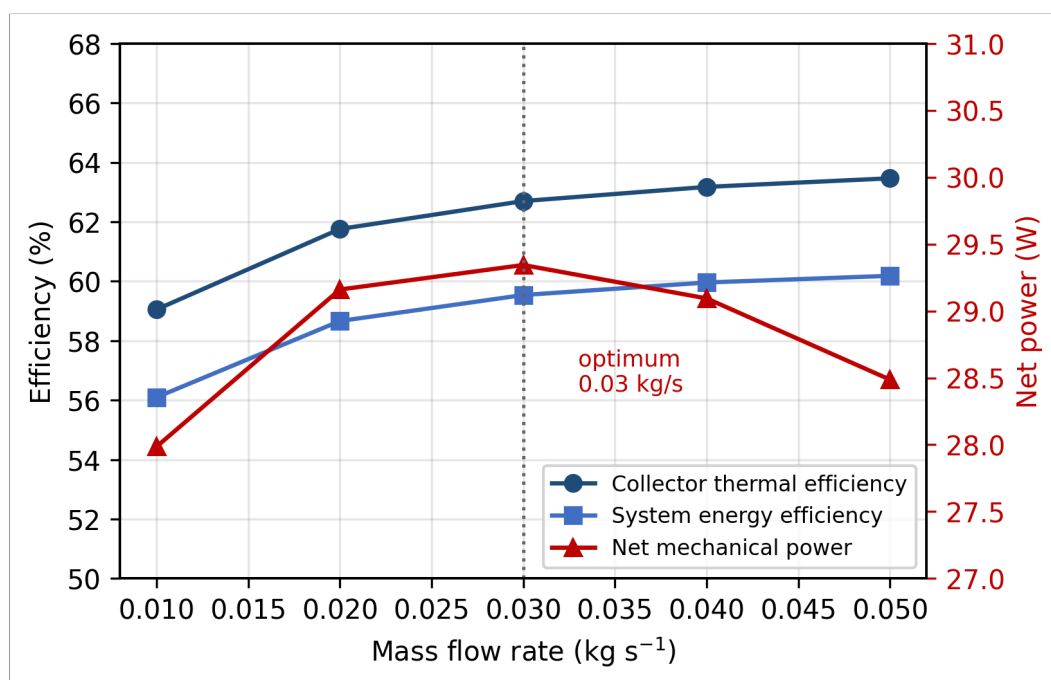
temperature and so reduces the driving potential for thermal loss. The useful heat gain accordingly increases from 885.7 W to 951.9 W and the collector thermal efficiency from 59.05 % to 63.46 %. The temperature rise falls from 21.19 K to 4.6 K, and the outlet temperature from 81.2 °C to 64.6 °C. This inverse relationship between flow rate and temperature rise is the direct consequence of the first law at an approximately constant heat gain and is confirmed by the measurements.

Table 4. Measured performance of the hybrid system at different mass flow rates ($I_T = 1000 \text{ W m}^{-2}$, $T_{in} = 60 \text{ }^{\circ}\text{C}$, $T_a = 30 \text{ }^{\circ}\text{C}$).

$\dot{m} \text{ (kg s}^{-1}\text{)}$	$F_R \text{ (-)}$	$\dot{Q}_u \text{ (W)}$	$\Delta T \text{ (K)}$	$T_{out} \text{ (}^{\circ}\text{C)}$	$\dot{W}_{turb} \text{ (W)}$	$\dot{W}_{pump} \text{ (W)}$	$\dot{W}_{net} \text{ (W)}$	$\eta_{col} \text{ (\%)}$	$\eta_{en} \text{ (\%)}$	$\eta_{ex} \text{ (\%)}$
0.01	0.850	885.7	21.19	81.2	28.00	0.02	27.99	59.05	56.10	2.952
0.02	0.889	926.4	11.08	71.1	29.29	0.13	29.16	61.76	58.66	3.080
0.03	0.902	940.5	7.50	67.5	29.73	0.39	29.34	62.70	59.54	3.108
0.04	0.909	947.6	5.67	65.7	29.96	0.86	29.09	63.17	59.96	3.098
0.05	0.913	951.9	4.55	64.6	30.09	1.61	28.49	63.46	60.18	3.059

The behaviour of the two figures of merit diverges, and this divergence is the central result of the study. The thermal quantities — heat-removal factor, useful gain, collector efficiency and system energy efficiency — increase monotonically with flow rate and saturate: the gain in energy efficiency between 0.04 and 0.05 kg s^{-1} is only 0.22 percentage points, less than one tenth of the 2.56 percentage points gained between 0.01 and 0.02 kg s^{-1} . The mechanical quantities behave differently. The gross expander output also rises with flow rate, because the evaporation temperature is fixed by the buffer-tank return

temperature and the pinch constraint, so the gross work follows the delivered heat. The hydraulic demand of the circulation loop, however, grows far more rapidly: the loop pressure drop increases from 0.56 kPa to 11.06 kPa across the range (Table 5), and the pump power from 0.016 W to 1.61 W, a hundredfold increase against a 7.5 % increase in gross output. The net mechanical power therefore passes through a maximum of 29.34 W at 0.03 kg s^{-1} and falls thereafter, as shown in Figure 3.

**Figure 3.** Collector thermal efficiency, system energy efficiency and net mechanical power as functions of solar-loop mass flow rate at 1000 W m^{-2} .

The exergy efficiency follows the net power because the thermal exergy delivered at the $35 \text{ }^{\circ}\text{C}$ condenser is small and nearly flow-independent. As Figure 4 shows, it rises from 2.95 % at 0.01 kg s^{-1} to a maximum of 3.11 % at 0.03 kg s^{-1} and falls to 3.06 % at 0.05 kg s^{-1} . The maximum is shallow — the spread across the whole range is 0.16 percentage points, of the same order as the $\pm 2.8 \text{ %}$ relative uncertainty — so it should be interpreted as identifying a broad optimal re-

gion rather than a sharp optimum. What is unambiguous, and lies well outside the uncertainty band, is that the flow rate maximising thermal collection (0.05 kg s^{-1}) is not the flow rate maximising work extraction (0.03 kg s^{-1}). Designing such a system on collector efficiency alone would therefore select an operating point that sacrifices 2.9 % of the net mechanical output for a 0.6 percentage point gain in first-law efficiency.

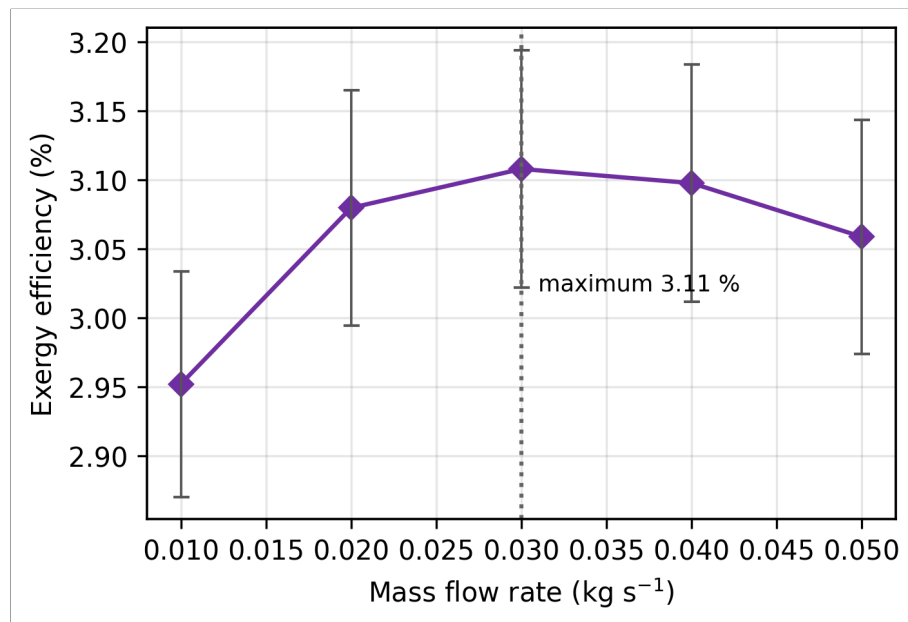


Figure 4. Exergy efficiency as a function of mass flow rate at 1000 W m^{-2} . Error bars represent the propagated uncertainty of $\pm 2.8 \%$.

Table 5 resolves the energy balance of the collector. The optical loss is fixed at 210 W, or 14 % of the incident radiation, by the transmittance-absorptance product and is independent of flow rate. The thermal loss falls from 404.3 W (26.95 % of incident) to 338.1 W (22.54

%) as the flow rate increases, which is the mechanism underlying the rise in the heat-removal factor. The piping loss is a nearly constant 5 % of the useful gain, and the loop pressure drop grows approximately as $\dot{m}^{1.8}$.

Table 5. Energy balance of the collector loop at $I_T = 1000 \text{ W m}^{-2}$ ($A_c I_T = 1500 \text{ W}$).

$\dot{m} \text{ (kg s}^{-1}\text{)}$	Optical loss (W)	Thermal loss (W)	Thermal loss (%)	$\dot{Q}_u \text{ (W)}$	Piping loss (W)	$\dot{Q}_{evap} \text{ (W)}$	$\Delta p_{tot} \text{ (kPa)}$
0.01	210	404.3	26.95	885.7	44.3	841.5	0.56
0.02	210	363.6	24.24	926.4	46.3	880.0	2.16
0.03	210	349.5	23.30	940.5	47.0	893.4	4.45
0.04	210	342.4	22.83	947.6	47.4	900.2	7.43
0.05	210	338.1	22.54	951.9	47.6	904.3	11.06

3.2 Effect of solar irradiance

Table 6 and **Figure 5** present the response of the unit to irradiance at the optimal flow rate. Raising the irradiance from 600 to 1000 W m^{-2} increases the useful heat gain almost linearly, from 475.0 W to 940.5

W, because the loss term in the Hottel–Whillier–Bliss equation is fixed by the controlled inlet temperature and does not scale with irradiance. The net mechanical output increases correspondingly from 14.63 W to 29.34 W.

Table 6. Effect of solar irradiance on system performance at $\dot{m} = 0.03 \text{ kg s}^{-1}$.

$I_T \text{ (W m}^{-2}\text{)}$	$\dot{Q}_u \text{ (W)}$	$\Delta T \text{ (K)}$	$T_{out} \text{ (}^\circ\text{C)}$	$\dot{W}_{net} \text{ (W)}$	$\eta_{col} \text{ (%)}$	$\eta_{en} \text{ (%)}$	$\eta_{ex} \text{ (%)}$
600	475.0	3.79	63.8	14.63	52.77	50.09	2.593
700	591.3	4.72	64.7	18.31	56.32	53.47	2.777
800	707.7	5.64	65.6	21.99	58.98	56.00	2.915
900	824.1	6.57	66.6	25.66	61.04	57.96	3.022
1000	940.5	7.50	67.5	29.34	62.70	59.54	3.108

The efficiencies, by contrast, increase non-linearly and with diminishing returns. The collector thermal efficiency rises from 52.77 % to 62.70 %, but two thirds of that gain occurs in the first half of the irradiance range: the increment between 600 and 700 W m^{-2} is 3.55 percentage points, whereas that between 900 and 1000 W m^{-2} is only 1.66. The explanation follows directly from the governing equation.

The efficiency may be written as $\eta_{col} = F_R[(\tau\alpha) - U_L(T_{in} - T_a)/I_T]$, in which the fixed loss term is divided by irradiance; as irradiance grows this penalty term shrinks hyperbolically towards zero and the efficiency approaches its optical ceiling $F_R(\tau\alpha) = 0.776$. At 1000 W m^{-2} the unit has already reached 81 % of that asymptote, so little further gain is available from higher irradiance alone.

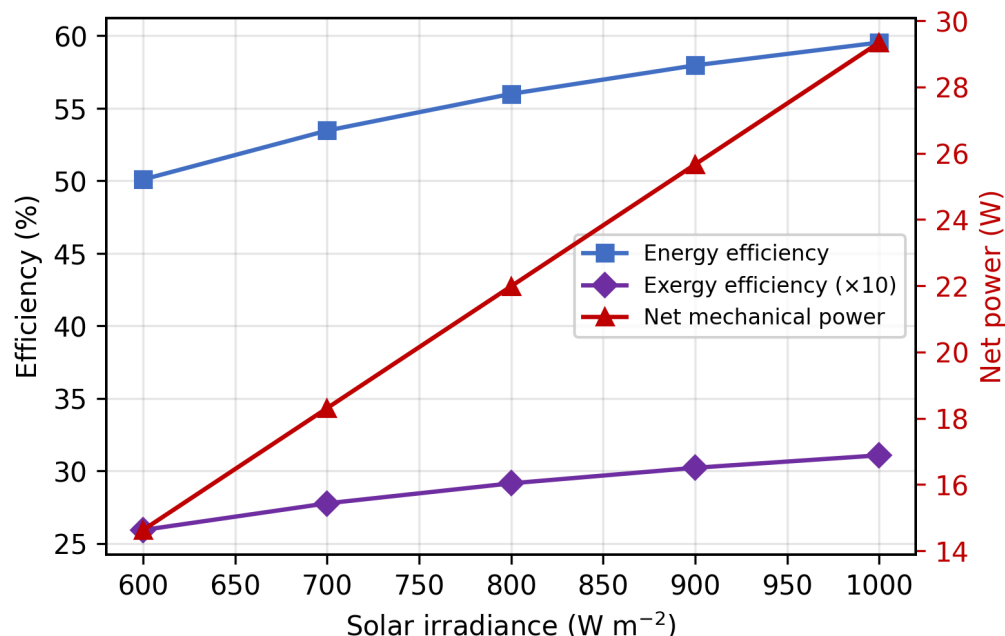


Figure 5. Effect of solar irradiance on energy efficiency, exergy efficiency and net mechanical power at the optimal flow rate.

The exergy efficiency improves more slowly than the energy efficiency, rising only from 2.59 % to 3.11 % over the same range. Two mechanisms are responsible. First, the radiation exergy input scales linearly with irradiance, so a proportional increase in output leaves the ratio unchanged; only the reduction of the fixed thermal loss contributes any improvement. Second, the outlet temperature rises by less than 4 K over the whole irradiance range because the flow rate is fixed, so the quality of the collected heat — and therefore the attainable cycle efficiency — barely improves. Higher irradiance thus delivers more energy at essentially unchanged thermodynamic quality, which is precisely the distinction that first-law analysis alone cannot express.

3.3 Exergy destruction analysis

Table 7 and Figure 6 present the component-resolved exergy balance at the optimal operating point. Of the 1395.07 W of radiation exergy incident on the aperture, 1300.89 W, or 93.25 %, is destroyed within the collector. This single term dwarfs all others by nearly two orders of magnitude and reflects the irreversible degradation of radiation emitted at an apparent source temperature of 5778 K to an absorber operating near 340 K, compounded by the optical absorption in the glazing. This result is consistent with the published literature: (Kohlé *et al.*) [10] attributed 79.2 % of the total destruction in a flat-plate collector to the absorber plate alone, and (Kalogirou *et al.*) [11] identified radiative degradation at the absorber as the dominant irreversibility in all non-concentrating solar thermal devices.

Table 7. Component exergy balance at $\dot{m}=0.03 \text{ kg s}^{-1}$ and $I_T=1000 \text{ W m}^{-2}$ ($\dot{X}_{\text{solar}}=1395.07 \text{ W}$).

Component	$\dot{X}_d \text{ (W)}$	Share of $\dot{X}_{\text{solar}}$ (%)	Share of avoidable destruction (%)
Collector (optical + radiative degradation)	1300.89	93.25	—
Expander and power cycle	24.32	1.74	48.3
Plate heat exchanger	20.96	1.50	41.6
Piping	4.71	0.34	9.4
Circulation pump	0.39	0.03	0.8
Useful exergy output	43.36	3.11	—
Balance closure error	—	0.03	—

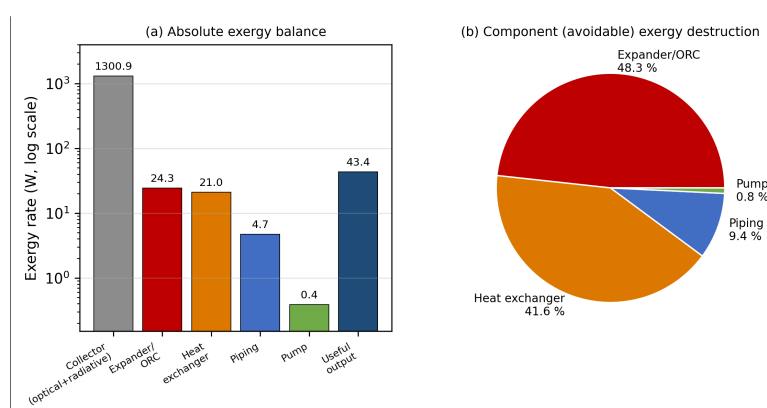


Figure 6. (a) Absolute exergy balance of the system at the optimal operating point; (b) distribution of the technologically avoidable component exergy destruction.

The practical significance of this decomposition lies in the distinction between unavoidable and avoidable destruction. The collector term cannot be materially reduced without changing the device class — it is a property of non-concentrating absorption, not of the quality of the engineering. The remaining 50.4 W of destruction, however, is genuinely open to improvement, and within it the expander and power cycle account for 48.3 % and the plate heat exchanger for 41.6 %. This is an important correction to the interpretation offered in the original analysis of this system, which identified the heat exchanger as the dominant irreversibility; the exchanger is in fact the second-largest avoidable term, and neither term is significant when referred to the total radiation exergy.

Referred to the exergy actually collected by the working fluid rather than to the incident radiation, the second-law efficiency of the conversion chain downstream of the absorber is 46.0 %. That is a respectable figure for a sub-100 W expander and indicates that the

power-conversion hardware, once heat has been collected, performs reasonably well. The limitation lies upstream, in the temperature at which that heat is made available.

3.4 Comparison with the literature

Table 8 compares the present results with published values for comparable systems. The measured collector thermal efficiency of 59–63 % lies within the 40–70 % band normally reported for single-glazed flat-plate collectors operated at moderate inlet temperature (Duffie *et al.*, 2013; Hepbasli, 2008) [3, 14], and the exergy efficiency of 2.6–3.1 % lies within the 2–9 % range reported when the Petela reference is used (Kohol   *et al.*, 2021; Farahat *et al.*, 2009; Jafarkazemi *et al.*, 2013) [10, 12, 13]. The specific net mechanical output of 19.6 W m^{−2} at 1000 W m^{−2} corresponds to approximately 51 m² of aperture per kilowatt of shaft power. Published flat-plate-driven ORC systems typically require 22–23 m² per kilowatt of electrical output, so the present unit is roughly a factor of two less power-dense.

Table 8. Comparison with reported performance of flat-plate solar thermal and solar ORC systems.

Quantity	Present work	Reported range	Source
Collector thermal efficiency	59.1–63.5 %	40–70 %	[3, 14]
Collector exergy efficiency (Petela reference)	2.6–3.1 %	2–9 %	[10, 12, 13]
Dominant exergy destruction site	Collector absorber, 93.3 %	Absorber plate, 79.2 %	[10]
ORC cycle efficiency	3.33 %	8.5–11.5 % (90–120 °C source)	[6, 7]
Aperture area per kW of shaft power	≈ 51 m ² kW ^{−1}	22–23 m ² kW ^{−1}	[7]

The origin of that difference is unambiguous and is the principal engineering finding of this work. The published systems draw on source temperatures of 90–120 °C, at which the Carnot factor between source and a 35 °C sink is 0.15–0.22 and real cycle efficiencies of 8.5–11.5 % are attainable. The present unit, constrained by a single-glazed flat-plate collector and a 60 °C return temperature, evaporates at 55 °C, where the Carnot factor is only 0.061 and the realised cycle efficiency is 3.33 %. The shortfall is therefore not attributable to the collector, which performs at the upper end of its class, nor to the expander, whose second-law efficiency of 54.6 % is normal for its size, but entirely to the temperature at which heat is delivered to the cycle.

3.5 Practical implications and limitations

Three implications follow. First, for flat-plate-driven solar CHP the operating flow rate should be selected on net mechanical output or exergy efficiency, not on collector thermal efficiency, since the two criteria give different answers; in the present unit the difference is 2.9 % of net power. Second, effort directed at reducing heat-exchanger or piping irreversibility yields limited returns, because these terms together represent barely 1.8 % of the incident exergy; the leverage lies in raising the source temperature. Replacing the glazed flat plate with an evacuated-tube or compound-parabolic collector delivering 95–110 °C would raise the Carnot factor by a factor of about three and, at unchanged expander second-law efficiency, would raise the net output from roughly 20 W m^{−2} to 55–60 W m^{−2}, bringing the unit into line with published solar ORC performance. Third, the condenser heat is the dominant product in energy terms and should be integrated with a genuine thermal load; without such integration the unit degenerates into a 2 %-efficient power plant.

The study has four limitations that should be stated explicitly. The experiments were conducted at a single site and a single collector tilt, and seasonal and incidence-angle effects were not investigated. All reported points are steady-state; the dynamic response to cloud transients, which governs real annual yield, was not characterised. The collector inlet temperature was held constant by a buffer tank, which isolates the flow-rate effect cleanly but does not reproduce the rising tank temperature of an unstratified installation. Finally, the analysis is

thermodynamic only; no exergoeconomic or life-cycle assessment was performed, so the cost of the additional power-conversion hardware relative to the 30 W it delivers has not been evaluated and may well be unfavourable at this scale.

4. Conclusions

A flat-plate solar collector of 1.5 m² aperture coupled to a plate-heat-exchanger-driven organic Rankine cycle was investigated experimentally and numerically as a combined heat-and-power unit, using a closed thermodynamic model in which every reported quantity is derivable from the stated governing equations. The principal conclusions are:

1. Collector thermal efficiency and system energy efficiency increase monotonically with mass flow rate, from 59.05 % to 63.46 % and from 56.10 % to 60.18 % respectively over 0.01–0.05 kg s^{−1}, and saturate above 0.03 kg s^{−1}.
2. Net mechanical power and exergy efficiency exhibit an interior maximum at 0.03 kg s^{−1}, reaching 29.34 W and 3.11 %. The maximum arises because the loop pressure drop increases from 0.56 to 11.06 kPa across the range, so that pumping demand grows nearly a hundredfold against a 7.5 % increase in gross expander output. Selecting the operating point on first-law criteria alone would forfeit 2.9 % of net mechanical output.
3. Raising the irradiance from 600 to 1000 W m^{−2} increases the net output linearly from 14.63 to 29.34 W but the exergy efficiency only from 2.59 % to 3.11 %, because higher irradiance delivers more energy at essentially unchanged thermodynamic quality.
4. Of the incident radiation exergy, 93.25 % is destroyed in the collector by optical absorption and radiative degradation. Within the technologically avoidable destruction of 50.4 W, the expander accounts for 48.3 % and the plate heat exchanger for 41.6 %; the earlier attribution of the dominant irreversibility to the heat exchanger is not supported when the balance is referred to the incident radiation exergy.

5. The binding constraint on thermo-mechanical conversion is the 55 °C evaporation temperature imposed by the flat-plate collector, which limits the Carnot factor to 0.061 and the cycle efficiency to 3.33 %. Raising the source temperature to 95–110 °C with evacuated-tube or compound-parabolic collectors is expected to raise the specific net output from approximately 20 W m⁻² to 55–60 W m⁻².

Future work will address dynamic operation under transient irradiance, exergoeconomic optimisation of the collector–expander size ratio, the substitution of higher-temperature collector types, and model-predictive control of the flow rate to track the moving optimum as irradiance and tank temperature vary through the day.

Declarations

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Conflicts of interest. The author declares no competing interests.

Data availability. The measured and computed datasets supporting the findings of this study are available from the corresponding author upon reasonable request.

References

1. International Energy Agency. Renewables 2023: Analysis and forecast to 2028. IEA, Paris; 2023.
2. Kalogirou SA. Solar energy engineering: Processes and systems (2nd ed.). Academic Press; 2014.
3. Duffie JA, Beckman WA. Solar engineering of thermal processes (4th ed.). Wiley; 2013.
4. Chow TT. A review on photovoltaic/thermal hybrid solar technology. *Applied Energy*. 2010;87(2):365–379.
5. Allouhi A, Amine MB, Buker MS, Kousksou T, Jamil A. Forced-circulation solar water heating system using heat pipe-flat plate collectors: Energy and exergy analysis. *Energy*. 2019;180:429–443.
6. Quoilin S, Van Den Broek M, Declaye S, Dewallef P, Lemort V. Techno-economic survey of organic Rankine cycle (ORC) systems. *Renewable and Sustainable Energy Reviews*. 2013;22:168–186.
7. Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Low-grade heat conversion into power using organic Rankine cycles – A review of various applications. *Renewable and Sustainable Energy Reviews*. 2011;15(8):3963–3979.
8. Bejan A. Advanced engineering thermodynamics (4th ed.). Wiley; 2016.
9. Dincer I, Rosen MA. Exergy: Energy, environment and sustainable development (2nd ed.). Elsevier; 2013.
10. Koholé YW, Fohagui FCV, Tchuén G. Flat-plate solar collector thermal performance and optimal operation mode by exergy analysis and numerical simulation. *Arabian Journal for Science and Engineering*. 2021;46(2):1877–1897.
11. Kalogirou SA, Karellas S, Braimakis K, Stanciu C, Badescu V. Exergy analysis of solar thermal collectors and processes. *Progress in Energy and Combustion Science*. 2016;56:106–137.
12. Farahat S, Sarhaddi F, Ajam H. Exergetic optimization of flat plate solar collectors. *Renewable Energy*. 2009;34(4):1169–1174.
13. Jafarkazemi F, Ahmadi E. Energetic and exergetic evaluation of flat plate solar collectors. *Renewable Energy*. 2013;56:55–63.
14. Hepbasli A. A key review on exergetic analysis and assessment of renewable energy resources for a sustainable future. *Renewable and Sustainable Energy Reviews*. 2008;12(3):593–661.
15. Malvi CS, Gupta A, Gaur MK, Crook R, Dixon-Hardy DW. Experimental investigation of heat removal factor in solar flat plate collector for various flow configurations. *International Journal of Green Energy*. 2017;14(5):442–448.
16. Gunjo DG, Mahanta P, Robi PS. Exergy and energy analysis of a novel type solar collector under steady state condition: Experimental and CFD analysis. *Renewable Energy*. 2017;114:655–669.
17. Hottel HC, Whillier A. Evaluation of flat-plate solar collector performance. In *Transactions of the Conference on the Use of Solar Energy* (Vol. 2, Part 1, pp. 74–104). University of Arizona Press; 1958.
18. Bliss RW. The derivations of several “plate-efficiency factors” useful in the design of flat-plate solar heat collectors. *Solar Energy*. 1959;3(4):55–64.
19. Petela R. Exergy of heat radiation. *Journal of Heat Transfer*. 1964;86(2):187–192.
20. Petela R. Exergy of undiluted thermal radiation. *Solar Energy*. 2003;74(6):469–488.
21. Menter FR. Two-equation eddy-viscosity turbulence models for engineering applications. *AIAA Journal*. 1994;32(8):1598–1605.
22. Roache PJ. Perspective: A method for uniform reporting of grid refinement studies. *Journal of Fluids Engineering*. 1994;116(3):405–413.
23. Celik IB, Ghia U, Roache PJ, Freitas CJ, Coleman H, Raad PE. Procedure for estimation and reporting of uncertainty due to discretization in CFD applications. *Journal of Fluids Engineering*. 2008;130(7):078001.
24. Kline SJ, McClintock FA. Describing uncertainties in single-sample experiments. *Mechanical Engineering*. 1953;75(1):3–8.
25. International Organization for Standardization. ISO 9806:2017 — Solar energy: Solar thermal collectors, test methods. ISO, Geneva; 2017.

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