

Performance simulation of a Solar-boosted Ocean Thermal Energy Conversion plant in Singapore

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Abstract. Working fluid selection is critical to the efficiency of Ocean Thermal Energy Conversion (OTEC) systems, where the inherently small temperature difference between warm surface and cold deep seawater limits achievable efficiency. Solar-boosted OTEC (SOTEC) addresses this where solar thermal energy raises the warm seawater source temperature. However, established working fluid selection criteria—dry/wet fluid classification, boiling temperature, latent heat of vaporization, and critical pressure—have only been validated above SOTEC-relevant heat source temperatures (50–60°C). This study evaluates these criteria at the pertinent temperature and identifies suitable working fluids for SOTEC deployment in Singapore across a three-stage thermodynamic screening methodology: (1) design-point efficiency comparison of 21 pure fluids at evaporator/condenser boundary temperatures (333.15K/283.15K), tested through Spearman rank correlation and Mann-Whitney U tests; (2) seasonal analysis of top-performing fluids across Singapore's monthly daytime and nighttime seawater temperatures; and (3) crossover analysis across a 10-50K solar boost sweep. Latent heat of vaporization was the only criterion convincingly corroborated at SOTEC-scale temperatures ($r_s = +0.760$, $p < 0.001$), with critical pressure showing moderate correlation ($r_s = +0.545$, $p = 0.011$). However, wet/dry classification and boiling temperature showed no significant relationship. Ammonia performed best throughout all conditions due to its shallow saturation curve slope, which raises its turbine enthalpy recovery, combined with ammonia's large latent heat of vaporization, which dilutes the relative cost of pump work. In the absence of solar-boosting, the efficiency spread among candidates significantly narrowed, emphasizing the significance of working fluid selection in solar boosting contexts. Additionally, solar boosting was found to raise ammonia's efficiency by 2.68-3.16 times, with the efficiency margin with its closest competitor widening.

Keywords: Ocean Thermal Energy Conversion, solar thermal energy, organic rankine cycle, working fluid selection

1. Introduction

Especially with the ubiquitous use of artificial intelligence, and the 2026 oil crisis¹, global energy demand

¹ Market Report - February 2026 – Analysis'. IEA, 12 Feb. 2026, <https://www.iea.org/reports/oil-market-report-february-2026>.

continues to surge while conventional fossil fuel deposits remain scarce². Thus, persistent research into reliable renewable alternatives capable of supplying consistent, dispatchable power is critical. Among ocean-based renewable technologies, this consistency advantage is especially prominent for Singapore. The city-state is a tropical island on the equator, so its waters sustain a relatively consistent temperature gradient between the solar-heated surface seawater and the cooler, deep water throughout the year. This makes Singapore a location where these ocean-based energy technologies are more viable than in many other geographical settings. One such technology is Ocean Thermal Energy Conversion (OTEC).

OTEC is a renewable energy technology that generates power by exploiting the naturally occurring temperature gradient between the warm surface water and the colder deep water to drive a closed thermodynamic power cycle. After being first proposed in 1881 by D'Arsonval, OTEC was largely unexplored [1]. However, in recent times OTEC has garnered considerable research attention, with most of the work being focused on ammonia as the working fluid of choice for OTEC. Uehara et al. [2, 3] conducted extensive research on OTEC as an energy system and adopts ammonia as the pure working fluid studied for closed-cycle OTEC.

However, OTEC has a significant disadvantage that has limited its commercial competitiveness and thus, its ability to be implemented: The inherently small temperature difference that OTEC has to work with, typically 15K-25K, between surface and deep seawater [4]. Since the thermodynamic efficiency of any heat engine is due to the Carnot efficiency — the upper limit of the efficiency that a heat cycle can have because it represents the efficiency of a purely reversible cycle, with no energy dissipation—which scales directly with temperature difference, OTEC systems are inherently limited to low efficiencies.

Numerous strategies have sought to mitigate this limitation. Alternative cycle architectures have been proposed, like the Kalina cycle, which uses a zeotropic ammonia-water mixture rather than employing a pure working fluid [5]. Zeotropic mixtures' temperature glide brings the potential for efficiency gains over conventional closed-cycle OTEC systems that employ Rankine cycles operating across the same temperature span [6]. However, these approaches only partially address the underlying problem. Yamada et al. [7] proposed the concept of solar boosting OTEC, forming the system called SOTEC, to increase the temperature of the hot reservoir, thereby providing a more direct improvement to efficiency as it directly addresses OTEC's inefficiency problem. Additionally, the concept of supplementing OTEC with solar thermal energy is fitting for Singapore since it receives abundant solar irradiance year-round³. SOTEC has been demonstrated to be a viable approach by Yamada et al. [7], in which a small-scale SOTEC plant was simulated to have low-cost solar-thermal collectors, which reported efficiency improvements over conventional OTEC.

SOTEC represents a promising renewable energy technology for Singapore, however the choice of which pure working fluid should be used in SOTEC still remains unclear. While ammonia dominates the broad field of OTEC research [3, 5, 7], which reflects its accepted suitability for conventional OTEC-scale temperatures, it is not certain that ammonia still performs optimally once a solar boost is introduced and the operating temperature range expands as working fluid selection for SOTEC has received comparatively less attention. This gap inspired this study's focus on recognising alternative pure working fluid candidates specifically suited to SOTEC operation in Singapore's conditions, primarily through an evaluation of existing pure working fluid selection criteria.

² 'Executive Summary – Key Questions on Energy and AI – Analysis'. IEA, <https://www.iea.org/reports/key-questions-on-energy-and-ai/executive-summary>. Accessed 21 Sept. 2026.

³ Sunny Days Ahead for Solar Adoption in Singapore. <https://www.ema.gov.sg/news-events/news/feature-stories/2023/sunny-days-ahead-for-solar-adoption-in-singapore>. Accessed 21 Sept. 2026.

Working fluid selection criteria have been validated using data calibrated at OTEC-scale temperatures and significantly higher temperature Organic Rankine Cycle (ORC) systems [8-11]. Critically, however, no substantial validation exists for these same criteria at hot source temperatures characteristic of solar boosted OTEC systems in tropical settings, which *Yamada et al.* showed to be usually in the 50-70 °C range.

This study therefore addresses two research questions. First, do established working fluid selection criteria remain valid predictors of cycle efficiency at true SOTEC-scale operating temperatures (50–60°C)? Second, once the top-performing working fluids for this temperature range are identified under Singapore's realistic seasonal operating conditions, does an efficiency crossover occur between them across the solar boost range achievable in Singapore's seasonal cycle?

2. Methodology

2.1. Model of SOTEC plant

The solar collector will be implemented in the warm seawater line, as shown in Figure 1. The warm seawater is pumped from the ocean surface and is heated by the solar collector. An alternative method of installing the solar collector could have been in the working fluid line to directly heat the working fluid after receiving some heat from the warm seawater, through the heat exchanger. However, Yamada et al. determined that the latter was not as preferable because of several reasons including those that would potentially increase the operating cost of the system. However, the most pressing reasons for this study is the safety and environmental risks posed from the increased likelihood of the working fluid leaking into the environment over long-term operation, due to the extension of the working fluid line [7]. Additionally, this plant reverts back to OTEC operation, whose cycle is shown in Figure 2, during nighttime when solar radiation is inadequate for SOTEC operation.

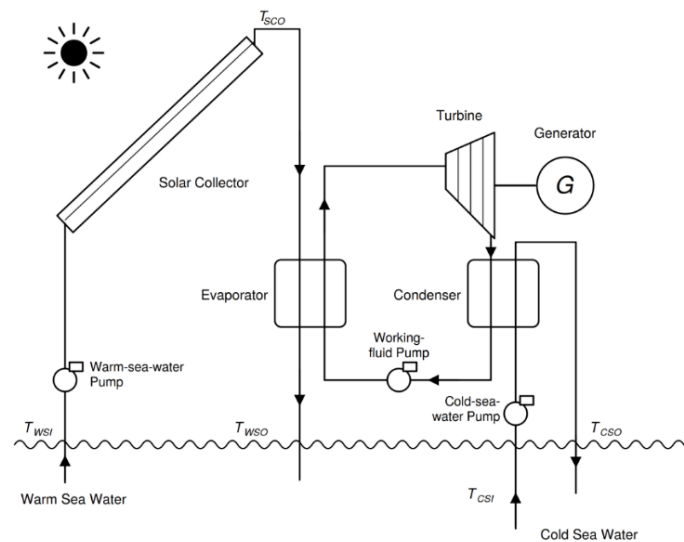


Figure 1. Schematic of SOTEC operation, wherein the solar collector is installed in the warm seawater line. The symbols representing various pumps, heat exchangers, turbine and the generator have also been included in the cycle schematic diagram. Adapted from Yamada et al. [7]

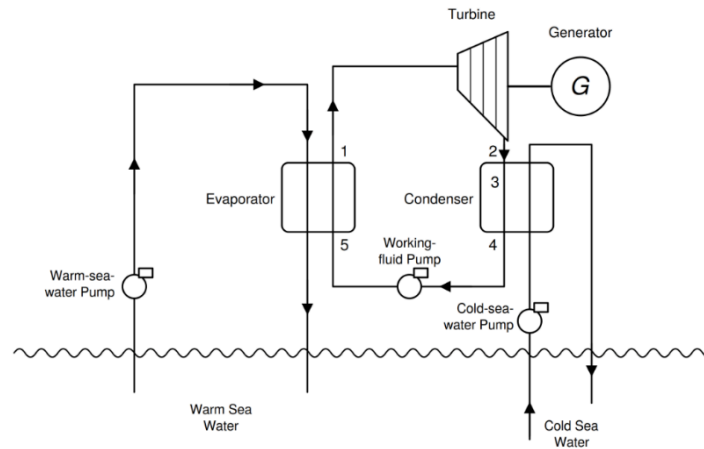


Figure 2. Schematic of conventional closed-cycle OTEC operation. Adapted from Yamada et al. [7]

In the simulation, a saturated Rankine cycle was assumed and an isentropic state-point analysis will be conducted in order to determine the cycle efficiency of the SOTEC system. Figure 3 and 4 show the system's temperature-entropy diagram, which is the same as that of fundamental ORC. At point 1, the working fluid enters the turbine as saturated vapour and undergoes isentropic expansion to the condenser pressure. Dry fluids exit the turbine as superheated, while wet fluids' state will already be within the two-phase dome. Nevertheless, this turbine exit state is displayed by point 2 for both fluid types. At point 3, the condenser rejects latent heat from the working fluid at a constant temperature and pressure as the vapour becomes a saturated liquid. At point 4, the pump compresses the working fluid isentropically, increasing its pressure to that of the evaporator. Finally, heat is added in the evaporator as sensible and latent heat to complete the organic Rankine cycle. Additionally, points 2 and 5 represent theoretical isentropic states for the sole purpose of calculating ideal enthalpy changes, from which turbine and pump work are derived using their respective isentropic efficiencies.

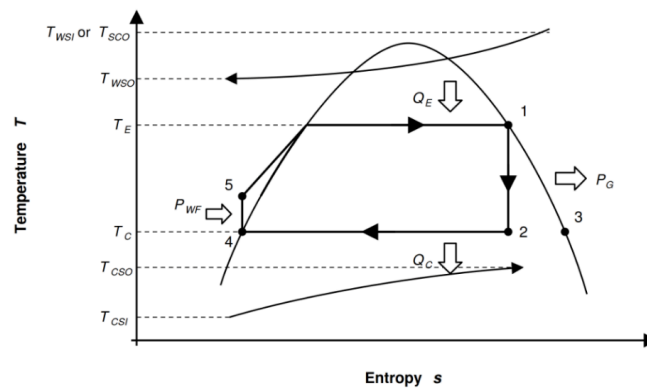


Figure 3. T-s diagram of the closed organic Rankine cycle studied for wet fluids; the numbers and variables correspond to those in Figures 1 and 2. Adapted from Yamada et al. [7]

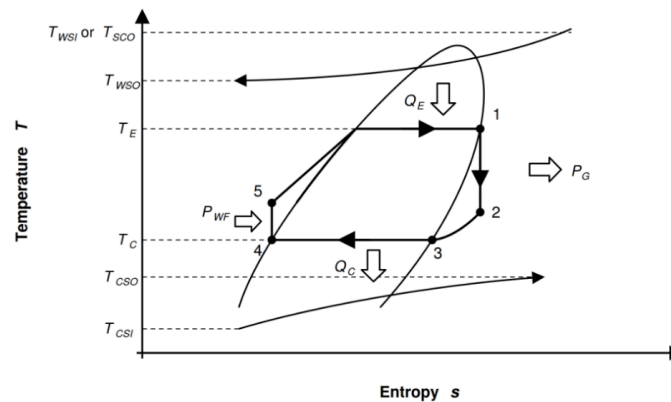


Figure 4. T-s diagram of the closed organic Rankine cycle studied for dry fluids; the numbers and variables correspond to those in Figures 1 and 2

2.2. Working fluid selection criteria

The following criteria will be evaluated throughout the subsequent sections of the research, where each assessed fluid's thermophysical data can be found in Table 1 below.

Critical Pressure

Wang et al. [8] suggests that fluids that have high critical pressure tend to have high cycle efficiencies because of this, using R32 as an illustration.

H1: Critical pressure will correlate positively with calculated cycle efficiency across the set of candidate working fluids.

Latent Heat of Vaporization

Chen et al. [9] identifies fluids' latent heat of vaporization as a determinant of thermodynamic performance as higher cycle efficiencies were attained while using working fluids with higher latent heats were employed.

H2: Latent heat of vaporization, assessed at the evaporator boundary temperature, will correlate positively with calculated cycle efficiency.

Slope of Vapor Saturation Line and Dry/Wet/Isentropic Classification

Dai et al. [10] conveys the idea that a working fluid's wet behaviour leads to increased cycle efficiency.

H3: Fluids with saturation curve slopes, on temperature entropy diagrams, that are negative, and thus exhibit wet fluid behaviour at the boundary temperatures, will exhibit higher calculated cycle efficiencies

Boiling Temperature

Hung et al. [11] suggests that for higher efficiencies working fluids should have higher boiling temperatures and the highest performing fluids had the highest boiling temperatures.

H4: The boiling temperature of the selected working fluids will correlate positively with calculated cycle efficiency.

The 21 pure fluids to be analysed were selected based on a few stages. Firstly, a literature review of a greater number of potential pure working fluids for subcritical ORC was conducted [12, 13]. Then, there needed to be an approximately equal number of fluids that exhibited dry behaviour and wet behaviour at the boundary conditions in order to properly evaluate existing literature on working fluid criteria. Finally, they were selected based on their critical temperatures, because as Xu et al. [14] suggests, critical temperature can be the sole criterion for fluid selection based on optimising thermal efficiency. Their findings suggest that for a

waste heat inlet temperature of 343K, the critical temperatures of optimal working fluids would fall in the range of 313K and 443K. However, since their finding is recommended for heat source temperatures of approximately 373K-573K, it was not applied as a strict constraint, but rather as a general approximation for the fluids that should be studied.

Since the aim of the research is to not just find the most efficient working fluid that must also therefore be practically applicable, but to test the validity of existing working fluid criteria at this untested temperature range for ORC, certain working fluids have not been excluded due to particular environmental, flammability or toxicity issues.

Table 1. The fluids assessed in the study as well as their respective thermophysical properties that are of interest to this study due to their impact on working fluid selection criteria

Working Fluid	Critical Temperature (K)	Critical Pressure (MPa)	Wet/Dry Behaviour	Boiling Point (K)	Latent Heat of Vaporization (kJ/kg)	Slope of Vapour Saturation Curve (K/(kJ·kg ⁻¹ ·K ⁻¹))
Ammonia	405.560	11.363	Wet	239.834	997.798	-96.385
Cyclopropane	398.692	5.605	Wet	241.671	342.885	-670.096
Hydrogen Sulfide	373.101	8.999	Wet	212.855	324.355	-194.016
Propylene	364.211	4.555	Wet	225.531	249.693	-417.849
Propyne	402.701	5.658	Wet	248.015	406.354	-522.441
R114	420.608	3.352	Dry	276.741	112.727	1,587.111
R115	353.102	3.129	Wet	233.932	63.909	-3,307.890
R12	385.120	4.136	Wet	243.398	114.465	-2,775.474
R1234yf	367.850	3.382	Wet	243.665	110.404	-10,712.763
R1234ze(E)	382.513	3.635	Dry	254.177	135.482	15,459.699
R124	395.428	3.624	Dry	261.187	123.124	4,641.066
R134a	374.212	4.059	Wet	247.067	139.125	-1,781.150
R152a	386.411	4.517	Wet	249.128	229.343	-822.544
R161	375.250	5.010	Wet	235.597	266.755	-414.295
R22	369.295	4.990	Wet	232.340	139.937	-632.033
R227ea	374.900	2.925	Dry	256.809	87.846	1,612.304
R236fa	398.070	3.191	Dry	271.662	122.699	1,624.780
R245fa	427.010	3.651	Dry	288.198	169.119	1,734.490
R32	351.255	5.783	Wet	221.499	175.514	-175.344
R601a	460.350	3.378	Dry	300.967	316.415	552.540
RC318	388.371	2.778	Dry	267.175	85.994	1,012.176

^aThe following thermophysical properties were collected from CoolProp software [15].

2.3. Computational methodology

Instead of net power output, cycle efficiency was chosen as the primary metric for comparison across the chosen working fluids and thus, is the primary data to be collected. Cycle efficiency is a ratio of work output to heat input. Since the heat input is fixed at 200 kW for every fluid, cycle efficiency isolates what fraction of

a given heat input each working fluid's thermodynamic properties enables conversion to useful work, independent of the mass flow rate chosen. Meanwhile, net power output depends directly on mass flow rate, which has been let to vary for each fluid in order for them to absorb the same heat input.

Thermodynamic state properties at each of the five cycle points were calculated using the open source thermophysical property library, CoolProp [15].

Working fluid mass flow rate ($\dot{m}_{WF}$), enthalpy (h), and entropy (s) at each state point were determined as described below. Working fluid mass flow rate is fixed by a specified evaporator heat flow rate, Q_e :

$$\dot{m}_{WF} = \frac{Q_e}{h_1 - h_5} \quad (1)$$

Turbine Work Output (TW) can be defined as the product of the working fluid mass flow rate, the adiabatic enthalpy difference between the evaporator and the condenser ($h_1 - h_2$) and the turbine isentropic efficiency (η_T).

$$TW = \dot{m}_{WF} \times (h_1 - h_2) \times \eta_T \quad (2)$$

Pump Work Input (PW) can be calculated as follows:

$$PW = \frac{\dot{m}_{WF} \times (h_5 - h_4)}{\eta_P} \quad (3)$$

Where η_P is isentropic pump efficiency and h_5 and h_4 represent the working fluid's enthalpy at points 5 and 4, respectively.

The cycles net Shaft Output Power (SOP) is computed by:

$$SOP = TW - PW \quad (4)$$

Meanwhile, the total Heat Input (HI) to the evaporator is defined by:

$$HI = \dot{m}_{WF} \times (h_1 - h_5) \quad (5)$$

Equation (5), by construction of Equation (1), equals Q_e for every working fluid evaluated — thereby allowing direct efficiency comparison across fluids independent of the arbitrary reference heat flow rate chosen.

Thus, cycle efficiency (η) is calculated by:

$$\eta = \frac{SOP}{HI} \quad (6)$$

The evaporating and condensing temperatures of working fluids (T_1 and T_4 , respectively) were fixed relative to assumed external seawater inlet temperatures via terminal pinch point temperature differences:

$$T_{WSI} = T_1 + \Delta T_{pinch,hot} \quad (7)$$

$$T_{CSI} = T_4 - \Delta T_{pinch,cold} \quad (8)$$

T_{WSI} represents the warm seawater inlet temperature, and T_{CSI} the cold seawater inlet temperature. The values of $\Delta T_{pinch,hot}$ and $\Delta T_{pinch,cold}$ were chosen to have the following values:

$$\Delta T_{pinch,hot} = 10K \quad (9)$$

$$\Delta T_{pinch,cold} = 5K. \quad (10)$$

The following values for pinch point temperature differentials are assumed for all subsequent data processing. These values have larger magnitudes relative to existing work on OTEC [16] primarily to be conservative regarding calculated cycle efficiencies, thus calculated data would represent lower bounds rather than optimistic estimates.

Additionally, for analysis in subsequent stages, the following equations will be used, where Δs represents fluids' entropy separation in kJ/kgK, s_g represents the specific entropy of saturated vapour and s_1 , the fluids' entropy at point 1.

$$\Delta s = s_g - s_1 \quad (11)$$

$$\text{Turbine Term} = \frac{TW}{HI} \times 100\% \quad (12)$$

$$\text{Pump Term} = \frac{PW}{HI} \times 100\% \quad (13)$$

The following assumptions have been applied to this work:

- Subcritical saturated cycle, resulting in no superheating or subcooling
- Cycle lacks a recuperator
- Wetness degradation effect of fluids on turbine blades is not accounted for

Moreover, standard steady-state, steady-flow simplifications are applied throughout: isobaric heat addition and rejection, without any pressure drops; negligible external heat loss anywhere in the cycle, including piping and auxiliary components; and negligible kinetic and potential energy changes across components.

Further, the following design parameters, as shown in Table 2, are held constant throughout this work:

Table 2. Design parameters of the thermodynamic cycle that are held constant

Parameters	Values
Evaporator Pinch Point Temperature Difference	10K
Condenser Pinch Point Temperature Difference	5K
Evaporator heat flow rate, Q_e	200kW
Working Fluid Pump Efficiency	0.75
Turbine Efficiency	0.80
Generator Efficiency	0.90
Evaporator Exit Vapor Quality	$Q = 1$
Condenser Exit Vapor Quality	$Q = 0$

Thermophysical properties for all working fluids — including enthalpy, entropy, critical temperature and pressure, latent heat of vaporization, boiling temperature and slope of vapor saturation lines at boundary temperatures — were obtained from CoolProp [15].

Monthly mean sea surface temperature data for Singapore's waters were obtained from Ombak and its MESN-SJI buoy, specifically, at a monthly-mean resolution and derived from in-situ buoy measurements⁴. Data for 2025's months were used. These values define the warm-water source temperature entering the evaporator, from which the working fluid's evaporating temperature is set via the assumed pinch point. The monthly mean is taken as the baseline unboosted (night-time) source temperature, while the daytime operation temperature comes from the modelled solar boost applied additively to the monthly mean temperature.

The analysis will therefore proceed in three sequential stages, each building on the previous.

⁴ Water Temperature and Weather Data - Marine Environmental Sensing Network (MESN). National University of Singapore, 2025, https://data.gov.sg/datasets/d_db2cd76f4349bfe4b40db22d8b295e54/view.

Stage 1 — Design-point screening. All 21 candidate fluids are evaluated at the fixed boundary temperatures of 333.15K in the evaporator (an external warm seawater inlet temperature of 30°C) and 283.15K in the condenser (an external cold seawater inlet temperature of 5°C). Calculated cycle efficiencies are to be tested against the criteria H1–H4, which are derived from existing literature, to assess their validity at SOTEC-scale temperatures, and the top-performing fluids are identified empirically from their calculated cycle efficiencies.

For stage 1, the external warm seawater inlet temperature was held constant at the specific temperature of 30°C which is a representative value of the annual mean surface seawater temperature for 2025, which was 29.47°C. Monthly data for surface seawater temperature will be exploited in Stages 2 and 3.

Stage 2 — Seasonal Robustness Check. The shortlisted fluids are to be firstly screened for their practical operability. Then, they will be re-evaluated across Singapore's monthly surface seawater temperatures under both boosted (daytime) and unboosted (night-time) temperatures, determining if the Stage 1 ranking holds or if there will be any crossover under realistic seasonal variation.

A 20-40K solar boost was identified by Yamada et al. [7] as preferable since too high a solar boost would result in difficulties maintaining high thermal efficiency simultaneously in two disparate temperature ranges as the plant alternates between OTEC and SOTEC operation. Additionally, too high a solar boost would require a large solar collector area, which would be impractical given Singapore's limited space. However, too low a solar boost would result in minimal increases in efficiency. Hence, the solar boost of 40K was chosen for the first 2 screening stages. Furthermore, concerning the collector area and irradiance conditions required to physically achieve this 40K solar boost, they are treated as outside the scope of this working-fluid-focused analysis.

Stage 3 — Crossover analysis. The confirmed top fluid and its closest competitor will be evaluated across a wide range of solar boosts for a representative monthly surface seawater temperature, identifying whether an efficiency crossover occurs within the modelled boost range or whether one fluid dominates throughout.

3. Stage 1 design-point screening

3.1. Design point definition

Stage 1 of the screening evaluates the 21 candidate working fluids at fixed boundary temperatures to isolate the influence of fluid properties from the effects of seasonal and operational variation. Applying the modelled 40K solar boost to the baseline warm surface seawater temperature of 30°C yields an inlet warm seawater temperature of 343.15K at the evaporator. However, the assumed pinch point temperature difference on the hot side results in the working fluid evaporating temperature being 333.15K. Meanwhile at the condenser, the cold, deep seawater inlet temperature is 278.15K (5°C), which with the assumed 5K pinch point applied results in a condensing temperature of 283.15K, therefore requiring the working fluid to expand across a 50K temperature difference in the cycle.

Additionally, all 21 fluids are evaluated under identical conditions – according to the design parameters and assumptions outlined previously – so any differences in calculated cycle efficiency can be attributed solely to differences in fluids' thermophysical properties.

3.2. Cycle efficiency results

The 21 candidate working fluids' calculated cycle efficiencies range from 9.020% (R115) to 10.793% (ammonia). The range of the fluids' efficiencies is 1.773%, while the median value is 10.068%. To contextualize the magnitude of this range, there is approximately a 20% difference in efficiency between the

best and the worst performing working fluids under identical boundary conditions and heat input. Thus, reiterating the implication of selecting unsuitable working fluids for ORC and SOTEC, alike. The majority of working fluids studied have efficiencies between 9.6% and 10.2%, with ammonia and propyne (10.607%) separating themselves from the next most efficient fluid by 0.196%, while R115, the lowest performing fluid, is 0.363% below the next lowest fluid. Alongside ammonia and propyne, cyclopropane (10.411%), R601a (10.396%) and R245fa (10.369%) form the group of fluids with the highest calculated cycle efficiencies.

It is also worth noting that ammonia, which Table 3 shows has the highest reported efficiency result, is the pure working fluid most conventionally used in standard, closed-cycle OTEC [17].

Table 3. The cycle efficiencies of the fluids studied, in addition to their entropy separation, turbine term and pump term values

Working Fluid	Cycle Efficiency (%)	Entropy Separation	Turbine Term	Pump Term
Ammonia	10.793	0.546	11.137	0.344
Cyclopropane	10.411	0.093	10.928	0.546
Hydrogen Sulfide	10.194	0.230	11.311	1.117
Propylene	9.687	0.084	10.820	1.132
Propyne	10.607	0.099	11.034	0.427
R114	10.101	-0.030	10.347	0.246
R115	9.020	-0.013	10.204	1.184
R12	10.083	0.0150	10.601	0.056
R1234yf	9.577	-0.010	10.348	0.771
R1234ze(E)	9.917	-0.009	10.434	0.517
R124	10.078	-0.010	10.478	0.400
R134a	9.890	0.020	10.521	0.631
R152a	10.246	0.061	10.746	0.499
R161	10.068	0.107	10.802	0.734
R22	9.965	0.066	10.878	0.913
R227ea	9.428	-0.037	9.983	0.556
R236fa	9.936	-0.030	10.243	0.307
R245fa	10.369	-0.021	10.496	0.156
R32	9.627	0.202	10.957	1.330
R601a	10.396	-0.068	10.504	0.108
RC318	9.383	-0.052	9.774	0.390

3.3. Criteria evaluation and hypothesis testing

To test the hypotheses H1-H4, non-parametric tests – Spearman rank correlation and Mann-Whitney U tests – were conducted based on the fluids' cycle efficiency results in Table 3, with Table 4 showing the results of these non-parametric tests.

Table 4. Values from the statistical evaluation of working fluid selection criteria

Thermophysical Property	Value	Value	Value
Critical Pressure	$r_s = + 0.545$	$p = 0.011$	
Latent Heat of Vaporization	$r_s = + 0.760$	$p = 6.461 \times 10^{-5}$	
Wet/Dry Behaviour	$U = 47$	$p = 0.717$	$z = - 0.362$
Boiling Temperature	$r_s = + 0.160$	$p = 0.489$	
Slope of Vapor Saturation Curve	$r_s = - 0.386$	$p = 0.084$	

To test Dai et al.'s claim [10], a Mann-Whitney U test was conducted because the wet/dry classification is categorical rather than continuous, so the U test compares the efficiency distributions of wet and dry fluid groups, rather than by rank correlation. The Mann-Whitney U statistic was computed from pooled ranks and converted to a z-score using the normal approximation, with two-tailed p -values obtained from the standard normal distribution.

Also, for the U test, the fluids' wet dry classification at the hot boundary temperature was used because the isentropic expansion originates at the fluids' state at the evaporating temperature and the slope of the vapor saturation curve here determines where the expansion process terminates relative to the two-phase saturation dome, therefore classification is assigned based on the slope at this point.

H1. Wang et al.'s claim [8] that there is a positive relationship between fluids' critical pressure and their cycle efficiency was tested by Spearman rank correlation ($r_s = + 0.545$, $p = 0.011$) across all 21 candidates, following the traditional Spearman Rank correlation method proposed by Spearman [18]. The results indicate a moderate-positive correlation that is unlikely to have arisen by chance, thus corroborating this hypothesis that working fluids with higher critical pressures were consistently associated with higher cycle efficiencies at SOTEC-scale boundary temperatures.

H2: Chen et al.'s findings [9] that showed that latent heat of vaporization was a determinant of thermodynamic performance in working fluids was tested by Spearman rank correlation ($r_s = + 0.760$, $p = 6.461 \times 10^{-5}$) across the select working fluids, evaluated at the evaporating temperature of 333.15K. Thus, revealing a strong, positive correlation that is the most statistically robust relationship observed among the criteria and is extremely unlikely to have arisen from random variation. Hence, significantly buttressing the claim that pure fluids' latent heat of vaporization substantially contributes to their thermodynamic efficiency.

H3: To test Dai et al.'s claim [10] that a fluid's wet behaviour leads to increased efficiency, a Mann-Whitney U test ($U = 47$, $z = -0.362$, $p = 0.717$) was conducted, following the traditional method proposed by Mann and Whitney [19]. The results show that no statistically significant difference in cycle efficiency between wet and dry fluids was found, since the p value is significantly greater than 0.05⁵. Exemplifying this, the median efficiency of wet fluids was 10.068%, while that of the dry fluids was just slightly behind at 10.007%. Critically, the wet fluid group contained both the most and least thermodynamically efficient fluids signifying that the wet/dry fluid classification is too coarse to be the sole predictor of efficiency for working fluids.

H4. Hung et al.'s suggestion [11] of a positive relationship between fluids' boiling temperature and their efficiency was tested by Spearman rank correlation ($r_s = + 0.160$, $p = 0.489$) across the selected working fluids. Therefore, indicating a weak, positive association that cannot be distinguished from random variation.

⁵ Flabs. Mann-Whitney U Test Explained – Non-Parametric Guide. <https://flabslis.com/blogs/interpreting-mann-whitney-u-test>. Accessed 21 Sept. 2026.

Hence, despite the two working fluids with the highest boiling temperatures (R245fa and R601a) having two of the highest efficiencies (10.369% and 10.396%, respectively), the hypothesis is not corroborated.

The wet/dry classification that is included in H3 and is common in literature surrounding working fluids used in ORC and OTEC is derived from the sign of the vapour saturation curve slope. To identify whether the fluids' slope's magnitude could contribute to determining its cycle efficiency, an inquiry that the conventional categorical classification discards, a Spearman rank correlation was conducted between fluids' $|dT/ds|$ at the evaporating and condensing temperature (333.15K and 283.15K, respectively) and their cycle efficiency. However, the slope's magnitude at the evaporating temperature was the primary test of the Spearman rank correlation ($r_s = -0.395$, $p = 0.077$). While the correlation is weak, negative and it does not reach the conventional statistical significance threshold of $p < 0.05$, the direction indicates that fluids with smaller magnitudes of their slopes, shallower slopes, were associated with higher efficiency.

3.4. Mechanistic explanation

Pure ammonia is typically the most studied working fluid for conventional OTEC cycles and now is also shown to have the highest cycle efficiency of a set of 21 select working fluids for a SOTEC cycle (40K solar boost). Hence, it remains critical to analyse the mechanism underlying its continued advantage at this higher evaporating temperature and the margin by which it leads to identify the specific criteria that engender the most thermodynamically efficient pure working fluids.

3.4.1. Sign of the slope of the vapor saturation curve

Ammonia is classified as a wet fluid. In ORC and OTEC systems, there can be a reluctance for employing wet fluids due to not a thermodynamic issue, but rather a mechanical one. Wet fluids have the tendency to cause turbine blade erosion from liquid droplets forming during expansion [20]. However, this practicality constraint is not accounted for by the cycle and thus, does not penalise wet fluids here. Conversely, this cycle configuration captures an efficiency cost that affects dry fluids, but not wet fluids: non-recuperation. Dry fluids exit the turbine in a superheated state, which increases the load of heat exchange on the condenser and effectively wastes energy that isn't converted into work [21]. The presence of a recuperator enables the enthalpy retained at the turbine exit to be recovered, therefore dry fluids in this study experience a direct reduction in their shaft output in comparison with fluids that expand further into the two-phase region. Additionally, and a stronger explanation for why wet behaviour can confer efficiency advantages for their fluids, for wet fluids, their isentropic expansion line terminates inside the two-phase dome upon reaching the condenser temperature. This advantage is materialized by fluids with shallow vapor saturation slopes, and particularly so in ammonia's case, as will be discussed below. However, as depicted by the Mann-Whitney U test, wetness alone confers no efficiency advantage.

3.4.2. Magnitude of the slope of the vapor saturation curve

While the sign of the slope of a fluid's vapor saturation curve at the evaporating temperature (dT/ds at 333.15K), on a Temperature-entropy graph, determines whether isentropic expansion terminates inside the two-phase dome or as superheated vapor, the slope's magnitude determines the extent of the separation between this constant-entropy line and the saturated vapor boundary across the 50K expansion. Smaller magnitudes for the slope mean that for the same temperature change, the saturation curve spans a relatively larger entropy range. However, the isentropic expansion line, by definition, keeps entropy constant, at s_1 . Therefore, this entropy separation grows as the slope's magnitude decreases.

The relevance in terms of efficiency follows from how the fluid distributes its energy release in the cycle between work extracted in the turbine and waste heat rejected in the condenser. Since condensation is an isothermal process, from point 2 to 4 in Figure 3, the heat the condenser rejects is directly set by the entropy

that the fluid will cover. Thus, the entropy separation at the condenser boundary point represents energy released through the turbines' blades in the form of useful, extracted work, rather than as rejected waste heat in the condenser. Hence, since the fluid's full entropy of vaporization represents the collective of the remaining entropy and the separation between the isentropic entropy line and the saturated vapor boundary at the turbine exit, a shallower slope widens this separation and transfers a larger proportion of this total energy output to the turbine.

Out of all 21 fluids examined, ammonia's saturation slope is the shallowest, with a magnitude of just 96.385 K/(kJ·kg⁻¹·K⁻¹), against a dataset median of 1012.176. Critically, ammonia correspondingly has the greatest value for entropy separation at the condenser boundary. Ammonia's 0.546 kJ/kg·K is significantly greater than the next two highest values: 0.230 kJ/kg·K (hydrogen sulfide) and 0.202 kJ/kg·K (R32). This places ammonia as the second-best performer on the turbine side with a turbine recovery fraction of 11.137%, against a range spanning 9.774% (RC318) to 11.310% (hydrogen sulfide). However, despite ammonia's shortcoming to hydrogen sulfide on the turbine side, ammonia's efficiency remains supreme because its pump term is significantly lower: 0.344%, as opposed to hydrogen sulfide's 1.117%.

Thus, two considerations follow from here. First, the mechanism operates only where expansion terminates within the two-phase region. Among fluids whose exit lies in superheated vapour, dry fluids, there is no observed comparable relationship between the fluids' slope magnitude and its turbine recovery, since no condensation occurs at the turbine for there to be any separation to represent this condensation. The wet behaviour of a fluid is thus a prerequisite for the slope mechanism rather than an advantage on its own, consistent with the absence of any correlation between the wet and dry categories and efficiency, as established above. Second, the enthalpy drop is not recovered as a proportion of heat input in the same ratio across all fluids. Because cycle efficiency is expressed relative to the heat inputted in the evaporator, a fluid requiring a large heat input converts its enthalpy drop into a smaller fractional return. Exemplifying this, ammonia's turbine recovery term is only 11.137% and as a result, is marginally exceeded by hydrogen sulfide's 11.311%, despite ammonia's separation being more than twice as wide. Like the wet/dry classification, slope magnitude alone hence does not account for ammonia's superior ranking, and the remaining margin is determined by the pump work term, as will be examined below.

3.4.3. Magnitude of latent heat

As outlined above, ammonia's primacy in the efficiency rankings does not come from the turbine side, but rather the pump side, and the thermodynamic property liable is the same one that hindered ammonia's turbine recovery fraction: its large magnitude of latent heat of vaporization (997.80 kJ/kg).

The enthalpy difference between point 1 and point 5 ($h_1 - h_5$) represents the total heat supplied to the working fluid in the evaporator, which consists of sensible heating of the compressed liquid and the latent heat of vaporization. The latter is the larger contributor of the two to the heat input supplied in the evaporator. Thus, ammonia's significant latent heat magnitude consequently gives it a specific heat input ($h_1 - h_5$) that is much larger than that of most other candidate working fluids studied. Hence, Ammonia's greater specific heat input dilutes ammonia's turbine recovery fraction, as shown in equation 10, due to a larger denominator as opposed to that of other fluids.

Meanwhile, pump work ($h_5 - h_4$) is governed by the compression of the liquid from the condenser pressure up to the evaporator pressure, which is a quantity that is determined by this pressure differential and the density of the saturated liquid entering the pump at the evaporator temperature. In fact, ammonia's pump work is not the smallest, but instead is mid-range (3.200 kJ/kg) among the candidates examined, which range from 2.760 to 3.670 kJ/kg.

Like the turbine term, Ammonia's pump term is also significantly reduced by ammonia's large latent heat. However, this reduction is asymmetric in comparison with the reduction in the turbine term. A large latent heat diminishes the turbine recovery by a marginal amount because the enthalpy drop rises alongside the rising specific heat input. The pump penalty, however, is substantially diminished because pump work does not rise with it. Ammonia's pump term represents 0.344% of heat input, in contrast to 1.117% for hydrogen sulfide and 1.330% for R32. Hydrogen sulfide and R32 both similarly possess a shallow vapor saturation slope, but have smaller latent heats: 324.400 kJ/kg and 175.500 kJ/kg, respectively. Therefore, ammonia's smaller pump penalty more than offsets its turbine recovery fraction, which is what ultimately decides the efficiency rankings, with ammonia at the apex. This is corroborated by the results of the Spearman rank correlation, which showed strong positive correlation between a fluid's latent heat of vaporization and its cycle efficiency.

Thus, ammonia's exceptional efficiency values stem from these three factors working in conjunction. Ammonia, having wet behaviour, has the benefit of a shallow vapor saturation slope having an impact on cycle efficiency. However, a shallow saturation slope, for fluids with smaller latent heats, can generate a strong turbine recovery fraction, but ultimately suffers due to a larger pump work, as R32 demonstrates. Meanwhile, a large latent heat, in isolation, would contain the pump penalty to a relatively smaller magnitude, but would not be able to deliver a competitive turbine recovery. But a cooperation among these factors results in the fluid with the highest efficiency. Ammonia ranks first in terms of latent heat and vapor saturation slope, while ranking second in terms of turbine fraction and sixth on that of the pump, however every fluid surpassing it on one term proves to be significantly worse on the other.

3.5. Operability screening

For the subsequent stages evaluating working fluids' performance under seasonal conditions and under a variety of solar boosts, only the highest-ranked fluids are carried forward. This is due to the fact that the efficiency margin separating them from the rest of the studied fluids is substantially larger than any plausible increases in efficiency that the remaining fluids could possibly have with the change in boundary temperatures, in the subsequent stages, such that a sufficient enough reordering among these lower-ranked fluids to displace the top ranked ones could not occur.

As elaborated above, cycle efficiency is largely a result of the amalgamation of several factors, therefore the fluids' rankings in any one factor will not determine their progression to the subsequent cycle simulations, but rather their ranking in cycle efficiency will.

The five highest-ranked fluids by cycle efficiency are ammonia, followed empirically by propyne, cyclopropane, R601a and R245fa, which were all carried forward to the seasonal analysis. R152a was the most efficient fluid that was excluded because it is 0.123% less efficient than the least efficient fluid in this group (R245fa), which is deemed relatively significant.

Table 5 reports the safety classifications and considerations for the 5 shortlisted fluids. The following ASHRAE classifications were derived from the Standard 34 Classifications⁶. No formal ASHRAE Standard 34 data was found for propyne and cyclopropane, so they were instead assessed against flammability and toxicity

⁶ ASHRAE. Designation and Safety Classification of Refrigerants. ANSI/ASHRAE Standard 34-2019, Addendum f, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc, 12 Dec. 2019, https://www.ashrae.org/file%20library/technical%20resources/standards%20and%20guidelines/standards%20addenda/34_2019_f_20191213.pdf.

standards reported in propyne's⁷ and cyclopropane's⁸ Safety Data Sheets (SDS). None of the fluids bring hazards that would stymie practical deployment. Despite certain fluids posing flammable and toxicity risks, these fluids that are both flammable and toxic are used in commercial ORC practice, therefore these classifications are recorded more as practical considerations instead of acting as screening criteria to prevent the progression of certain fluids.

Table 5. Practical deployment classifications for shortlisted working fluids

Working Fluid	ASHRAE Standard 34 Classifications	SDS Toxicity/Flammability Notes
Ammonia	B2L	
Propyne		Specific target organ toxicity and Category 1 flammable gas
Cyclopropane		Specific target organ toxicity (Category 3) and Category 1 flammable gas
R601a	A3	
R245fa	B1	

One operational consideration warrants note. The normal boiling temperatures of R601a and R245fa, at 300.98K and 288.20K respectively, exceed the condensing temperature of 283.15K. Hence, both would require sub-atmospheric pressures during operation, potentially posing air ingress risks. However, no other shortlisted fluid is affected in neither this 5 fluid set nor in the wider 21 fluid collection. The implications are that a practical plant would require additional measures to be taken to facilitate these pressures, which are discussed further during latter stages of the study.

4. Stage 2. seasonal robustness check

4.1. Seasonal operating conditions

Stage 2 assesses the five shortlisted fluids under Singapore's measured seasonal conditions, replacing the fixed 30°C evaporator temperature baseline in Stage 1 with monthly mean Ocean Surface Temperature (OST) data across 2025, obtained from Ombak from in-situ measurement⁹. Two boost conditions were simulated each month: solar-boosted daytime operation, in which the warm seawater inlet temperature represents the 40K solar boost applied to the monthly OST before the 10K hot-side pinch point temperature difference is applied; and unboosted night-time operation, where it is taken directly as the monthly OST before a 5K pinch point is applied. Here, a 5K pinch point was applied, instead of 10K, because for unboosted operation, which is essentially OTEC, there is an inherently small accessible temperature difference, so too large a magnitude for

⁷ Chemical Info for Propyne. <https://chemicalsafety.com/sds1/sdsviewer.php?id=35378684&name=Propyne>. Accessed 22 Sept. 2026.

⁸ Chemical Info for CYCLOPROPANE. <https://chemicalsafety.com/sds1/sdsviewer.php?id=30253745&name=CYCLOPROPANE>. Accessed 24 Sept. 2026.

⁹ Water Temperature and Weather Data - Marine Environmental Sensing Network (MESN). National University of Singapore, 2025, https://data.gov.sg/datasets/d_db2cd76f4349bfe4b40db22d8b295e54/view.

a pinch point would render an OTEC plant unsuitable for operation. Meanwhile, like in stage 1, the condensing temperature remains fixed at 283.15K throughout with a 5K cold-side pinch point applied, due to the fact that deep seawater temperatures are substantially more stable on a seasonal basis than that of the surface.

In this cycle, the solar boost is applied uniformly across all 12 months, regardless of variations in monthly irradiance, so seasonal variation is derived solely from OST fluctuations, which is constrained to a relatively limited annual range due to Singapore's equatorial waters. Hence, this stage functions to ascertain whether the Stage 1 ranking is preserved under realistic operating conditions, and to empirically identify ammonia's closest competitor(s) across the range of warm seawater inlet temperatures.

4.2. Seasonal cycle efficiency results

Ammonia retains the highest cycle efficiency in every month under both daytime and nighttime operation, without exception, as shown in Figures 5 and 6. Propyne and R601a similarly hold their positions throughout: second and third position, respectively. The sole crossover that occurs is between cyclopropane and R245fa, which exchange fourth and fifth place between daytime and night-time operation consistently in all twelve months. For daytime operation in all 12 months, cyclopropane is more efficient, while the contrary occurs for nighttime operation across all months.

Across all fluids, boosted efficiencies range from 9.972% to 10.920% and unboosted efficiencies from 3.235% to 4.068%, with solar augmentation raising ammonia's efficiency by a factor of between 2.68 and 3.16 depending on the month. The unboosted condition is of particular interest. Firstly, when the solar boost is removed, the efficiency spread separating the five fluids narrows from between 0.422 and 0.462 percentage points to between 0.050 and 0.071 points. Correspondingly, ammonia's efficiency advantage over propyne contracts from between 0.172 and 0.190 points to between 0.021 and 0.030 points. Secondly, unboosted cycle efficiency is significantly more impressionable to surface temperature changes in relative terms. Using ammonia as an example, its unboosted cycle efficiency varies by 23.8% between the warmest and coolest months, while its solar-boosted cycle efficiency deviates by only 5.1% between the same two months.

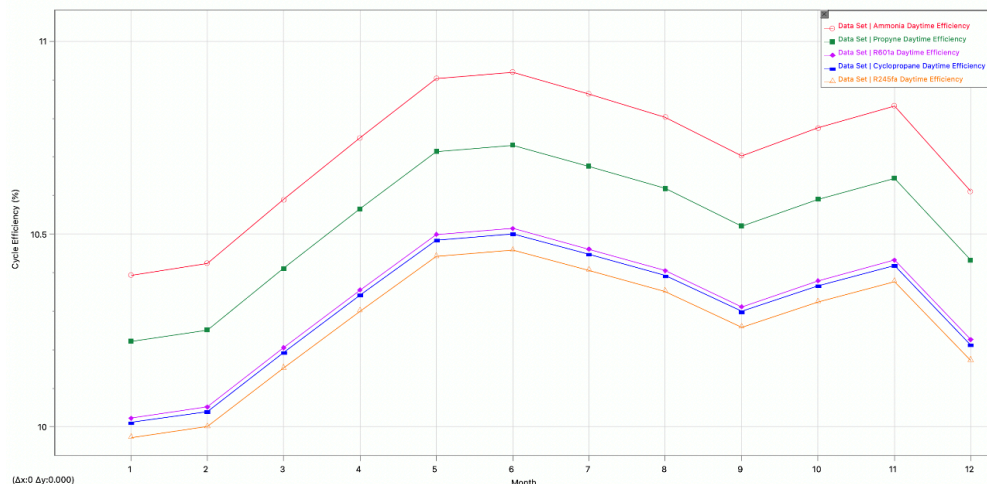


Figure 5. Monthly variation of daytime cycle efficiency for the 5 selected working fluids across the months of 2025

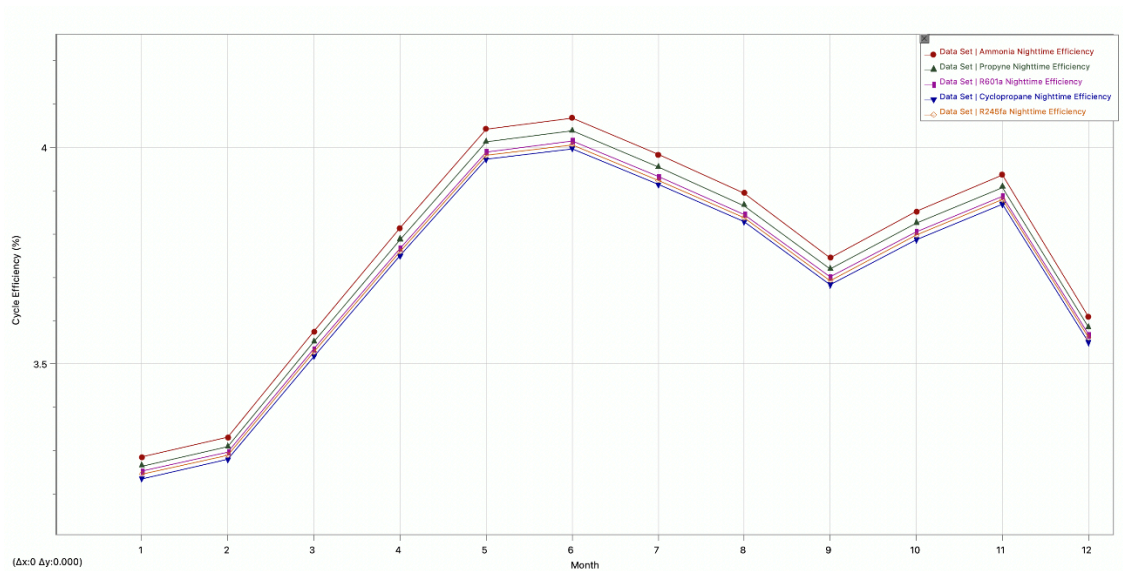


Figure 6. Monthly variation of nighttime cycle efficiency for the 5 selected working fluids across the months of 2025

4.3. Boosted versus unboosted performance

The two operating conditions directly represent Solar-Boosted OTEC (SOTEC) and conventional OTEC operation, the unboosted night-time case representing the cycle that a conventional OTEC plant would run continuously, even during daytime. Hence, the comparison quantifies the contribution of solar augmentation to the efficiency of the cycle, in addition to its influence on the ranking of the 5 most efficient fluids selected for Stage 2.

Using the leading candidate working fluid as example, ammonia, solar boosting results in cycle efficiencies of between 10.393% and 10.920% across the year, as opposed to between 3.285% to 4.068%, when unboosted. Across the 12 months, this increase in cycle efficiency ranges from 168.43% (June) to 216.36% (January). Evidently, this increase is attributable to the enlarged temperature difference that SOTEC provides the cycle with (between 47.35K and 50.61K) as opposed to OTEC (between 12.35K and 15.61K).

Table 6. Cycle efficiencies of the selected 5 working fluids during daytime operation across the 12 months of 2025

Months	Ammonia Cycle Efficiency (%)	Propyne Cycle Efficiency (%)	R601a Cycle Efficiency (%)	Cyclopropane Cycle Efficiency (%)	R245fa Cycle Efficiency (%)
January	10.393	10.222	10.022	10.011	9.972
February	10.424	10.251	10.051	10.039	10.000
March	10.589	10.410	10.205	10.192	10.152
April	10.749	10.565	10.355	10.342	10.300
May	10.903	10.714	10.499	10.484	10.442
June	10.920	10.730	10.514	10.500	10.458
July	10.863	10.675	10.461	10.447	10.406
August	10.803	10.618	10.405	10.392	10.351
September	10.703	10.521	10.311	10.299	10.258

Table 6. Continued

October	10.775	10.590	10.379	10.366	10.324
November	10.832	10.645	10.432	10.418	10.377
December	10.611	10.432	10.226	10.213	10.172

Table 7. Cycle efficiencies of the selected 5 working fluids during nighttime operation across the 12 months of 2025

Months	Ammonia	Propyne	R601a	Cyclopropane	R245fa
January	3.285	3.264	3.252	3.235	3.246
February	3.331	3.309	3.297	3.280	3.290
March	3.575	3.551	3.535	3.518	3.528
April	3.814	3.787	3.767	3.750	3.760
May	4.042	4.012	3.990	3.972	3.981
June	4.068	4.038	4.015	3.997	4.006
July	3.983	3.954	3.932	3.914	3.924
August	3.895	3.866	3.846	3.828	3.838
September	3.745	3.719	3.701	3.683	3.693
October	3.852	3.825	3.805	3.787	3.797
November	3.937	3.908	3.887	3.869	3.879
December	3.609	3.584	3.568	3.550	3.561

As Tables 6 and 7 show, the solar boost has a significant effect on the importance of working fluid selection itself. Working fluids' efficiencies in January show that, under boosted operation, the five candidates are separated by 0.422%, while in absence of this boost, they are separated by just 0.050%. Thus, demonstrating a rather limited consequence for working fluid selection of pure fluids in conventional OTEC. However in SOTEC, the value of pure working fluid selection becomes more tangibly significant as the solar boost widens the temperature difference that the cycle operates across. Additionally, this corresponds with the mechanisms outlined previously in Stage 1: both the entropy separation determining the turbine recovery fraction and the pressure differential against which the pump operates governing the pump work penalty depend on a wide temperature difference and compress as that range decreases.

Despite this convergence of efficiency values, no change in the most efficient fluid occurs. The same holds true for the second and third ranked fluids, propyne and R601, respectively. The only rearrangement in the overall ranking arises between cyclopropane and R245fa, in which exchange positions between day and nighttime operation are separated by no more than 0.041% at any operating condition. Moreover, propyne is confirmed as ammonia's closest competitor under practical operating conditions in Singapore.

Furthermore, the hot-side Pinch Point Temperature Differential (PPTD) remained at 10K for SOTEC. However, it was changed to 5K for unboosted operation because a 10K pinch point for OTEC operation with an inherently small temperature differential accessible is unequivocally unrealistic and no plant would operate on such conditions. Therefore, since the choice in PPTD for solar boosted operation was made to be conservative and the choice for that of unboosted operation was less so, it can be inferred that the difference between actual performance of the two is likely larger than indicated by the presented data.

4.4. Competitor identification for stage 3

Propyne is clearly ammonia's closest competitor with regards to efficiency and therefore, these two will be assessed at a variety of solar boosts. The remaining 3 fluids will not be due to the relatively larger gap between propyne and the third ranked fluid (R601a) as opposed to propyne and ammonia.

5. Stage 3. solar boost sweep analysis

5.1. Sweep definition and rationale

Stage 3 examines ammonia and propyne across a sweeping solar boost range from 10K to 50K in a 2K increment. This range extends to either side of the preferable 20–40K interval identified by Yamada et al. [7], so that each of the fluids' performance can be observed both within the idealised optimum solar boost range and past its boundaries.

The sweep was carried out for two representative months: the coolest and warmest months of 2025, which are January (mean OST of 300.504K) and June (mean OST of 303.756K), respectively. For each trial, the evaporating temperature reverts back to the inclusion of the 10K pinch point, which is applied to the monthly OST augmented with the respective solar boosts. Meanwhile, the condensing temperature remains fixed at 283.15K.

Stage 3 therefore isolates the magnitude of the solar boost as the independent variable, with the two months' OST providing the variation in baseline evaporating temperature, so the two influences on relative performance between the working fluids can be illustrated.

5.2. Solar boost sweep results

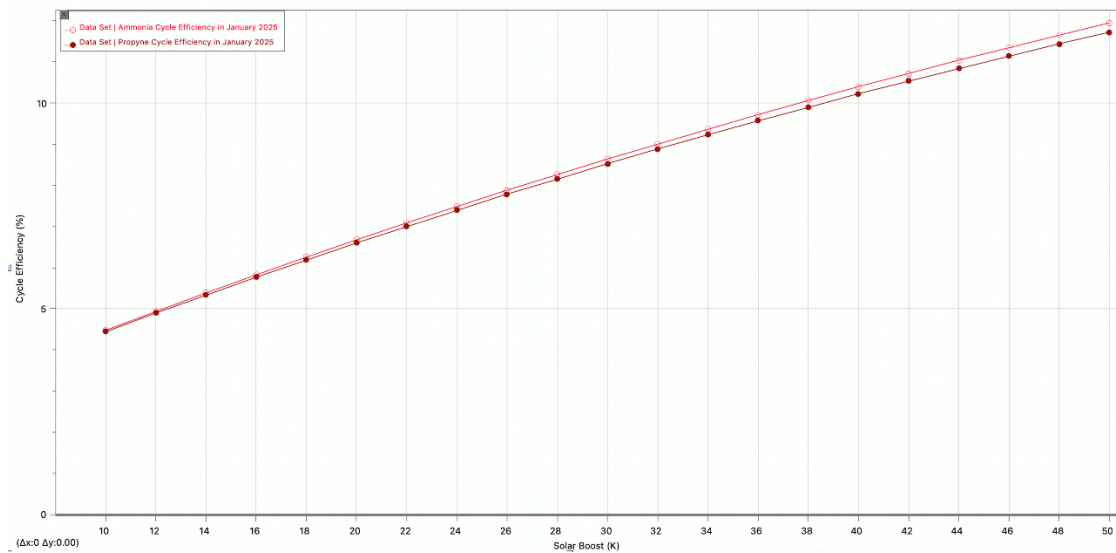


Figure 7. Cycle efficiency for ammonia and propyne across the solar boost sweep of 10K to 50K for January 2025

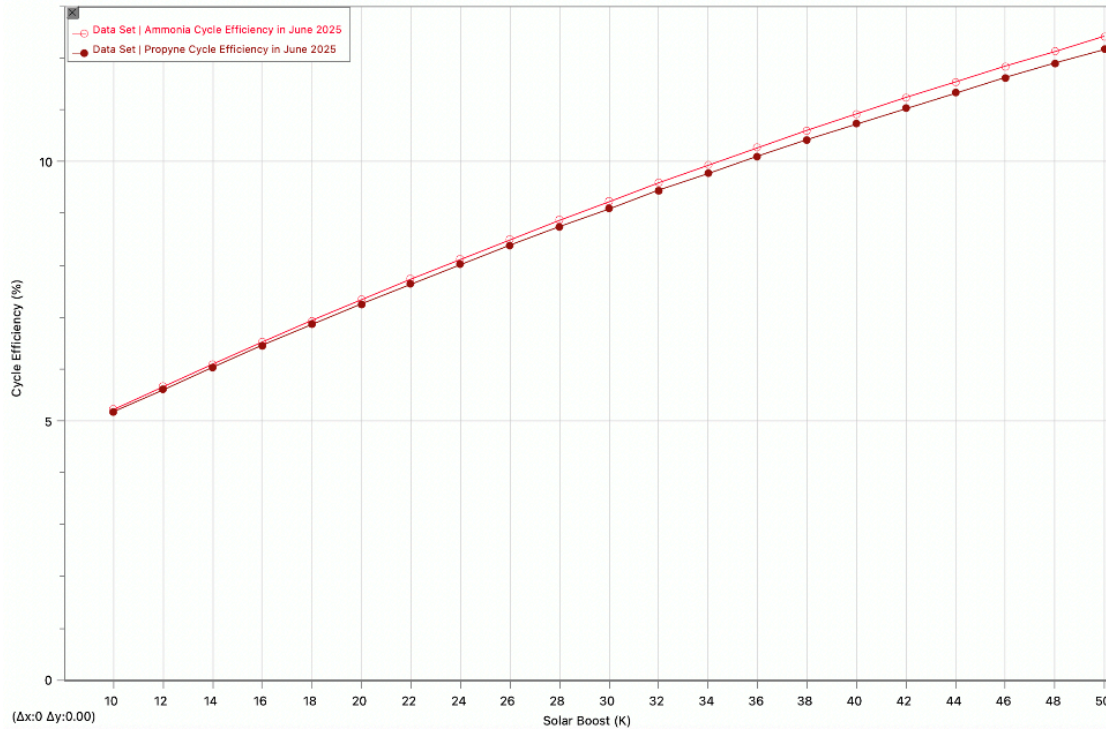


Figure 8. Cycle efficiency for ammonia and propyne across the solar boost sweep of 10K to 50K for June 2025

Like in Stage 1 and 2, ammonia remains the most efficient fluid, as opposed to propyne. Here, in stage 3, ammonia's cycle efficiency is higher at each and every solar boost increment, across both representative months examined, as shown in Figures 7 and 8. While Stage 2 shows that the efficiency values compressed as the evaporating temperature decreased, as made evident in the case of nighttime operation, the following results in Table 8 show that as the solar boost is incrementally increased, ammonia's efficiency advantage grows monotonically over propyne, which is its closest competitor. The difference between ammonia's and propyne's efficiency increases from 0.0470%, at a 10K boost, to 0.2478%, at a 50K boost, in June; and from 0.036% to 0.229%, over the same temperature range in January. Therefore, not only does a crossover fail to exist between the fluids' efficiencies, over a considerably large solar boost range for SOTEC, but instead they incrementally diverge, thus outlining ammonia as the clear choice of pure working fluid for SOTEC operation with a greater magnitude of solar augmentation.

Table 8. Cycle efficiencies of 2 most efficient fluids across a solar boost sweep from 10K – 50K, during the coolest and warmest months

Solar Boost (K)	January 2025		June 2025	
	Ammonia	Propyne	Ammonia	Propyne
10	4.478	4.442	5.219	5.172
12	4.937	4.894	5.662	5.607
14	5.386	5.336	6.095	6.033
16	5.825	5.768	6.519	6.448
18	6.255	6.189	6.933	6.854

Table 8. Continued

20	6.675	6.601	7.338	7.251
22	7.086	7.004	7.734	7.638
24	7.487	7.396	8.122	8.016
26	7.88	7.78	8.501	8.385
28	8.265	8.155	8.871	8.745
30	8.64	8.521	9.233	9.097
32	9.007	8.878	9.586	9.44
34	9.366	9.226	9.932	9.775
36	9.716	9.566	10.269	10.101
38	10.059	9.898	10.599	10.42
40	10.393	10.222	10.92	10.73
42	10.72	10.537	11.234	11.033
44	11.039	10.845	11.54	11.328
46	11.35	11.144	11.839	11.615
48	11.653	11.436	12.13	11.894
50	11.949	11.72	12.414	12.166

5.3. Mechanistic explanation

The reasoning for why ammonia's cycle efficiency preferentially increases over propyne's derives from the mechanisms established earlier, in Stage 1. Both relevant mechanisms depend on the solar boost temperature range and therefore, enhance ammonia's efficiency with a greater magnitude of solar boost.

Firstly, ammonia's advantage on the turbine side is due to its entropy separation at the condenser boundary, which increases as the saturation curve covers a more expansive temperature range. Since the fluids' slope of vapour saturation curve governs the increase in entropy separation, and ammonia's is shallower than propyne's, each increment of solar boost increases ammonia's entropy separation relatively more than it does propyne's. Their entropy separations at the endpoint of this temperature sweep substantiates this. In June, across the 40K increase in solar boost, ammonia's separation grows from 0.23683 to 0.65716 kJ/kg·K, while propyne's grows from 0.04383 to 0.12022. While both fluids' separation increases by approximately the same factor, in absolute terms, ammonia obtains an entropy separation growth of approximately 5 times that of propyne. Since the entropy separation at this condenser boundary reflects the energy released as useful work in the turbine instead of rejected heat in the condenser, the enthalpy ammonia recovers as useful work, through this route, increases as opposed to propyne's across the 40K solar boost sweep.

Secondly, as established, ammonia's advantage on the pump side arises from ammonia's large latent heat, which significantly contributes to a substantial specific heat input, which renders ammonia's pump penalty fraction insignificant as the pump work is relatively fixed, in comparison. However, the advantage is only measurable when the pump work becomes appreciable.

The pressure differential that the pump must surmount is the difference in saturation pressures at the evaporating and condensing temperatures. Given that the condensing temperature remains constant at all trials at 283.15K, the condenser pressure will similarly remain fixed. Meanwhile, the saturation pressure rises sharply and non-linearly as temperature increases, along the vapor pressure curve. At low solar boosts and thus, low evaporating temperatures, the pressure differential between the saturation pressures at the evaporating and condensing temperatures is negligible. As a result, the pump work for both fluids is

inconsequential and so, the effect that ammonia's large latent heat has on further reducing the pump penalty fraction is limited. However, as increased solar boosts raises the evaporating temperature and thus, pressure, the pressure differential grows, and pump work becomes a measurable component. In June, ammonia's pump term rises from 0.09911% to 0.4746%, at a 10K and 50K solar boost, respectively, while propyne's increases from 0.1310% to 0.5773%, across the same solar boost change. Thus, ammonia's latent heat recovers its effect.

Thus, solar boosting strengthens the effect of the two properties from which ammonia derives its advantage in the first place. Therefore, the results in Stage 2, where the unboosted, nighttime operation caused a compression of the fluids' efficiencies where the mechanism operated in the opposite direction to Stage 3. Hence, the divergence between the fluids' efficiencies is not due to a different phenomenon, but rather the same one, perceived from the opposite end of the operating range.

5.4. Boost-driven versus baseline-driven

The two influences on ammonia's efficiency margin over propyne can be isolated. Across the 40K solar boost increment, the margin in June increases by 0.2008%, from 0.0470% to 0.2478% at 10K and 50K solar boosts, respectively. Meanwhile, the margin between the two months at any solar boost ranges from 0.0112% to 0.0192%. Thus, the solar boost governs the divergence between ammonia and propyne's efficiency significantly more than the seasonal variation does.

This is due to the relative scale of the two variables. The applied solar boost spans a range of 40K, however, by comparison the difference in mean ocean surface temperature of the warmest and coolest months of 2025 is only 3.252K, as the seasonal variation is only reflected in the temperature variation of the two months. Thus, the divergence between the cycle efficiencies is primarily a result of solar boosting rather than the depending on the month in which the cycle is operating. This finding can be generalized across the annual range of ocean surface temperatures given that the two months considered had mean temperatures on either end of this annual range.

5.5. Relation to the literature-cited optimal range

Yamada et al. [7] claim that the most preferable range for solar boost magnitude to be 20-40K for a variety of reasons, but partly due to it being a cost-effective method when employing a single working fluid. This argument rests on reasoning rather than demonstration across the solar boost range. Meanwhile, the results from Stage 3 empirically test the validity of this claim. As elaborated upon earlier, ammonia's cycle efficiency is superior to propyne's among all evaporating temperatures considered, from a 10K boost to 50K. The fact that the difference between the fluids' efficiency actually increases over the solar boost range, which is 10K more extreme than either boundary of the range suggested, demonstrates that no crossover between fluids occurs and therefore substantiating the proposition of having single-fluid operation for SOTEC and that the fluid should be ammonia. In fact, this proposition is bolstered more significantly at the higher end of the range because the advantage of using ammonia as a working fluid increases with solar boost.

5.6. Stage 3 output

Hence, ammonia is the robustly selected pure working fluid across the extended solar boost range examined and the full seasonal cycle in Singapore, with no fluid exceeding ammonia's cycle efficiency at any point. The case for ammonia instead of propyne strengthens as the solar boost applied to the SOTEC system increases. Thus, no basis for dual-fluid operation arises for a SOTEC plant installed in Singapore.

6. Discussion and limitations

6.1. Discussion

Most of the established working fluid selection criteria do not sufficiently hold true in this analysis, which bears implications extending beyond simply research on the application of SOTEC. Stage 1 showed that the only relationship in any of the literature hypotheses examined that was convincingly substantiated for SOTEC operation was that fluids with higher latent heats of vaporization typically had higher cycle efficiencies (H2). The relationship in H1 had correlation, however it was moderate, while those in H3 and H4 could not be distinguished from random variation. This highlights the caution not only with making heuristics that condense underlying thermodynamic behaviour into forms convenient for rapid working fluid selection, but in general as well. Criteria that are verified at one operating temperature range cannot be extrapolated to another without further confirmation. However, whether these criteria regain their predictive value in recuperated SOTEC cycles, where the mechanisms handicapping dry fluids no longer occur, this investigation cannot acknowledge.

The findings presented earlier identifies ammonia as the most suitable pure fluid working fluid, in terms of thermodynamic performance, as it has the greatest cycle efficiency for both nighttime and daytime SOTEC operation. However, this does not directly translate to it being the most appropriate working fluid for a pure OTEC cycle, although it is often considered so. This is because ammonia's efficiency advantage over other pure fluids in nighttime operation (which represents a typical OTEC cycle) is much smaller than with the solar boost, thus giving rise to increased weighting for other factors that influence working fluid selection: safety classification, environmental impact, and operational costs, which could distinguish more sharply between the pure working fluid candidates examined here than cycle efficiency would. Thus, the choice of solar boosting an OTEC cycle does not solely have implications on the achievable efficiency of the cycle, but also on fluid selection. Pure working fluid selection and the boost design of SOTEC should be considered together rather than independently.

6.2. Limitations

With regards to the cycle's modelling, there are the following limitations. Firstly, a non-recuperated cycle was assumed meaning that at the turbine exit, the superheat retained by dry fluids is not recovered, thus resulting in lower cycle efficiencies for dry fluids, rendering the comparison between dry and wet behaviour specific to non-recuperated designs. Additionally, wet fluids do not suffer from the risk of their two-phase expansion causing turbine blade erosion as it is not accounted for in the calculated cycle efficiency. Instead, turbine and pump efficiencies are held constant, irrespective of exit vapour quality or the operating condition. Next, any pressure drops, external heat losses and non-equilibrium condensation that would occur in a physical SOTEC plant have not been modelled in this study. Also, the analysis has been restricted to pure fluids, excluding zeotropic mixtures and the potential benefits they may convey, such as their temperature glides.

Additionally, while the working fluid pump efficiency has been accounted for, the cold and warm seawater pump efficiencies have not, as cycle efficiency has only been assessed across the working fluid loop. These inefficiencies consist of a significant burden that would reduce the thermodynamic performance of fluids in ORC.

The boundary condition of the condenser has been assumed to have a condensing temperature of 283.15K throughout, which assumes stable deep-water temperature. However, the actual temperature of cold-water intake depends on the depth of the seawater pipe, which has not been accounted for, and the temperature increase of the working fluid as it rises up the pump. These would influence the calculated cycle efficiencies

and thus, significant pinch points have been set accordingly. However, these pinch points were set without reference to the heat transfer area of the heat exchanger, thus not establishing the heat exchanger size of these values. Additionally, fluids' heat transfer performance has not been identified, so fluids with poor coefficients would require greater exchanger areas for the same duty, which would result in greater costs. Both of these unidentified factors impact equipment sizing and thus, total costs incurred by a prospective SOTEC plant, which falls outside of the scope of the analysis which focuses on the ranking of fluids' thermodynamic efficiencies. Since identical pinch points were imposed to all fluids, neither asymmetrically favours one fluid over another.

Ocean seawater temperatures are monthly averages for 2025, which masks any interannual variability. Additionally, the solar boost, particularly the 40K boost used in Stage 1 and 2, was applied as a fixed input, instead of being derived from solar irradiance, because the solar collector area necessary to provide the solar boost falls outside the scope of the study.

Moreover, the comparison between wet and dry fluids, with only 13 and 8 fluids respectively, lacks sufficient potency to identify minimal differences, so the absence of a significant difference between the two classifications is more due to an inability to detect one, rather than it not existing.

Finally, due to the focus of the analysis, fluids were ranked solely on thermodynamic efficiency, although considering the minimum requirements for fluids' toxicity, flammability and global warming potential for the fluids to be further examined. As such, factors contributing to working fluid operational costs, chemical and thermal stability, and other environmental criteria, such as ozone depletion potential, were not considered. For example, additional measures would have needed to be taken to facilitate the employment of R245fa and R601a because they involve sub atmospheric pressures. However, these factors were excluded because these properties do not affect the thermodynamic performance of each fluid, so each would require a separate basis of comparison. So, the fluid shortlist identified should be considered as identifying the most thermodynamically suited fluids for the SOTEC cycle, instead of the fluids best suited for application.

7. Conclusion

A Solar-Boosted Ocean Thermal Energy Conversion (SOTEC) system for Singapore's waters was modelled, and working fluid selection criteria, from existing literature, for organic Rankine cycles was examined across twenty-one candidate working fluids, at SOTEC-scale temperatures. Most of the examined criteria were found to not reliably predict cycle efficiency at such temperatures, whereas the latent heat of vaporization emerged as the most potent sole predictor. Ammonia was verified as the most thermodynamically efficient pure working fluid out of those tested due to its combination of its wet behaviour, shallow vapour saturation curve slope, and its significant latent heat of vaporization.

Using Singapore's 2025 seasonal cycle as representative, solar boosting across these months was shown to raise ammonia's cycle efficiency by factors of between 2.68 and 3.16, for the coolest and warmest months, respectively. Additionally, ammonia retained its position as the most efficient fluid at every solar boost increment, from 10K to 50K, in both the warmest and coolest months, with its margin over propyne widening from 0.035% to 0.228% as the solar boost magnitude increased. The study therefore identifies single-fluid operation across both daytime and nighttime conditions as ideal, from a purely efficiency perspective. Moreover, the efficiency difference between the top five candidates narrows significantly in the absence of solar augmentation, such as from 0.422%, with the solar boost, to 0.050%, without it, for January. Thus, signifying that working fluid selection, on a purely efficiency basis, becomes more consequential only as the solar boost magnitude increases.

Future studies should account for the seawater pumping power to enable comparison with thermal efficiencies recorded in other literature and should quantify external factors that affect working fluid selection to result in a comprehensive fluid selection criteria that does not treat efficiency as the sole metric. Additionally, they should determine the achievable solar boost from solar irradiance data and solar collector area, instead of treating it as fixed across the annual range and extend the examination to include recuperated cycles.

Global energy demand has risen over the last few years and especially at the start of 2026, therefore viable renewable energy technologies are needed more than ever. Ocean thermal systems, particularly in equatorial locations like Singapore, offer a renewable resource whose stability outmatches intermittent sources. Thus, the development of more advanced OTEC-based systems will become more necessary to match this increased energy demand, particularly in countries like Singapore.

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