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Article

A Proposal to Use Power from Marine Solar-Thermal Injection Power System to Drive a Seawater Desalination-Electrolysis Plant

Marijo Miljkovic

Independent Researcher, Belgrade 11000, Serbia; marijo.miljkovic@ymail.com

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Abstract: To enable the efficient and sustainable use of thermal energy from natural fluids, such as seas, lakes, and the atmosphere, a novel power cycle was developed to harness low-temperature heat sources. This innovation, termed the Injection Power Cycle (IPC), incorporates an injector (ejector or thermal compressor) into the power cycle. Initial attempts using a conventional Thermal Compressor (TC) demonstrated insufficient efficiency for practical application. To address this limitation, a cooling stream was introduced into the TC, enhancing the compression of the Working Fluid (WF) and improving the system's overall performance. Nevertheless, despite these enhancements, the IPC initially remained slightly less efficient than the Rankine Power Cycle (RPC). To further increase the IPC's efficiency, the external turbine was integrated into the TC, enabling the utilization of the full heat potential of the total WF mass flow. As a result, the IPC emerged as the most efficient power cycle under conditions of a small temperature difference between the evaporating and condensing temperatures. The primary objective of this research paper is to demonstrate the potential of a novel power-generating technology, with the expectation that subsequent research efforts will validate the concept and optimize the performance of the Injection Power System (IPS) through the development and testing of a prototype. Finally, from a technical perspective, the author proposes utilizing power generated by a marine solar-thermal system based on the Injection Power Cycle (IPC) to drive a combined seawater desalination and electrolysis plant, thereby producing both potable water and hydrogen as sustainable outputs.

Keywords: Power Cycle; Renewable Energy; Injector; Solar-Thermal System; Desalination; Hydrogen

1. Introduction

1.1. The History of the Invention

A century ago, steam ejector refrigeration systems were commonly used to air condition large industrial and commercial buildings, as well as passenger trains. However, due to the low efficiency of conventional ejectors (also referred to as injectors or thermal compressors), these systems were gradually replaced by refrigeration systems utilizing mechanical compressors. Over time, the efficiency of thermal compressors (TCs) improved significantly through the introduction of a cooling stream into conventional TCs, which enhanced the compression of the Working Fluid (WF).

The concept of an "Ejector Refrigeration Cycle with the injection of a high-density fluid into a diffuser (a mixing chamber)" was first presented at the World Engineers Summit (WES) on Climate Change 2015 in Singapore [1,2]. A revised version of this research was subsequently published in the American Journal of Engineering Research (AJER) in 2018 [3].

Following this development, the TC with internal cooling was applied in a power cycle to extend the expansion of the WF [4]. This modification increased the temperature range within which the power cycle operates, surpassing the temperature difference between the heat source and the heat sink. As a result, the technical work produced by the WF increased, thereby improving the overall efficiency of the Injection Power Cycle (IPC) [4]. The first presentation of the IPC took place at the WES 2017 in Singapore [4].

Despite these advancements, the IPC's efficiency remained slightly lower than that of the Rankine Power Cycle (RPC). To further enhance its performance, the external turbine was integrated into the TC, allowing the full heat potential of the total WF mass flow through the TC to be utilized. This innovation made the IPC the most efficient power cycle, suitable for applications such as Ocean Thermal Energy Conversion (OTEC) power plants. Furthermore, OTEC power can be utilized for seawater desalination and seawater electrolysis to produce hydrogen [5].

The first presentation of the IPC with a turbine integrated into the TC with internal cooling was delivered in the form of a poster at the virtual International Conference on Ocean Energy (ICOE) 2021, hosted by the United States.

1.2. Thermal Compressor with an Integrated Turbine and Internal Cooling

To fully harness the thermal potential of the working fluid (WF) within the thermal compressor (TC), the external turbine is relocated from the bypass to the interior of the TC. This integration enables simultaneous compression of the WF (the motive stream) in the mixing chamber, mixing with a cooling stream (a high-density fluid), and mechanical work generation through the turbine's rotation.

The mixing process in the chamber generates significant heat due to friction. To mitigate this, an internal cooling mechanism is implemented within the TC. The secondary cooling stream introduces a liquid that evaporates within the chamber, thereby cooling the motive stream. This evaporation process enhances the compression of the working medium, optimizing the TC's performance (refer to **figure 1**).

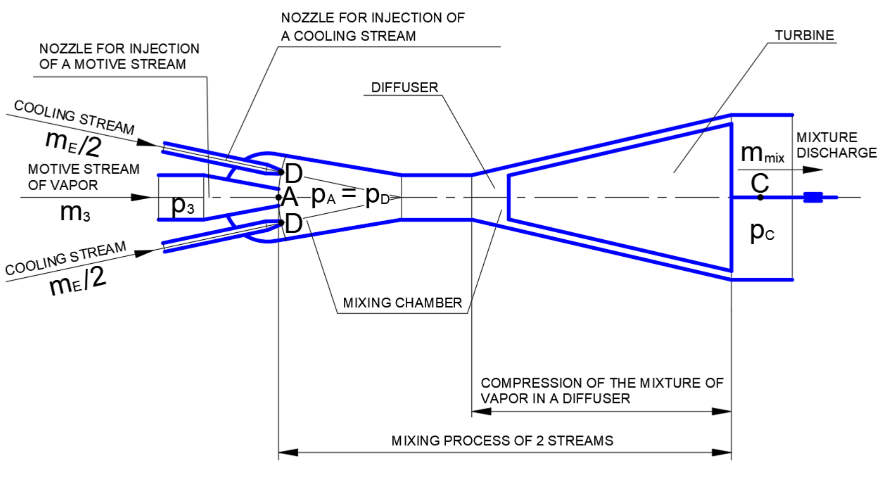

Figure 1. Thermal compressor (injector) with an integrated turbine and internal cooling.

1.3. The Purpose and Contribution of This Research

The oceans convert solar energy into thermal energy. They function as natural solar thermal collectors and represent the largest such collectors on Earth. However, the temperature difference between the surface warm-water layer and the deep cold-water layer remains relatively small. Consequently, a power cycle intended for application in Ocean Thermal Energy Conversion (OTEC) systems must achieve superior performance compared to existing power cycles to operate effectively under these conditions.

The IPC emerged as the most efficient power cycle under conditions of a small temperature difference between the evaporating and condensing temperatures. The primary objective of this research paper is to demonstrate the potential of a novel power-generating technology, with the expectation that subsequent research efforts will validate the concept and optimize the performance of the Injection Power System (IPS) through the development and testing of a prototype

2. Methods

The mathematical model for calculating the Injection Power System (IPS) is presented in Section 3: System Description, Calculation, and Discussion. The thermophysical properties of specific states of the R32 Working Fluid (WF) were determined using the SOLKANE Refrigerants 7.0 calculation program. Additionally, newer versions of the software, such as SOLKANE Refrigerants 8.0 or SOLKANE Refrigerants 9.0, which are available online, can also be utilized for similar calculations.

3. System Description, Calculation and Discussion

3.1. System Description

Figure 2 presents a schematic of the injection power cycle (IPC) as applied in an ocean thermal energy conversion (OTEC) power system. After the liquid WF $[(1 + k) \cdot \dot{m}]$ exits the condenser at state 1, it splits into two streams. The first stream $(k \cdot \dot{m})$ enters the TC nozzle as the cooling stream, expanding to state D in the mixing chamber. The second stream $(1.00\ \dot{m})$ flows into the pump, where its pressure is elevated to 14.74 bar. The pressurized liquid WF then enters the evaporator (state 2), where it undergoes phase change into vapor. The resulting hot, high-pressure vapor (state 3) enters the TC nozzle as the motive stream, expanding to state A and subsequently mixing with the cooling stream in the mixing chamber. During the mixing process, the combined stream undergoes cooling and subsequently enters the diffuser. In the diffuser, part of the mixture's kinetic energy $(\dot{m}3 \cdot vA2/2)$ is converted into mechanical work (wT) by driving a turbine. As the mixture's velocity and kinetic energy decrease within the diffuser, its pressure rises to match the medium (condensing) pressure, accompanied by an increase in enthalpy. Finally, the WF enters the condenser (state C), where it is fully condensed, completing the cycle.

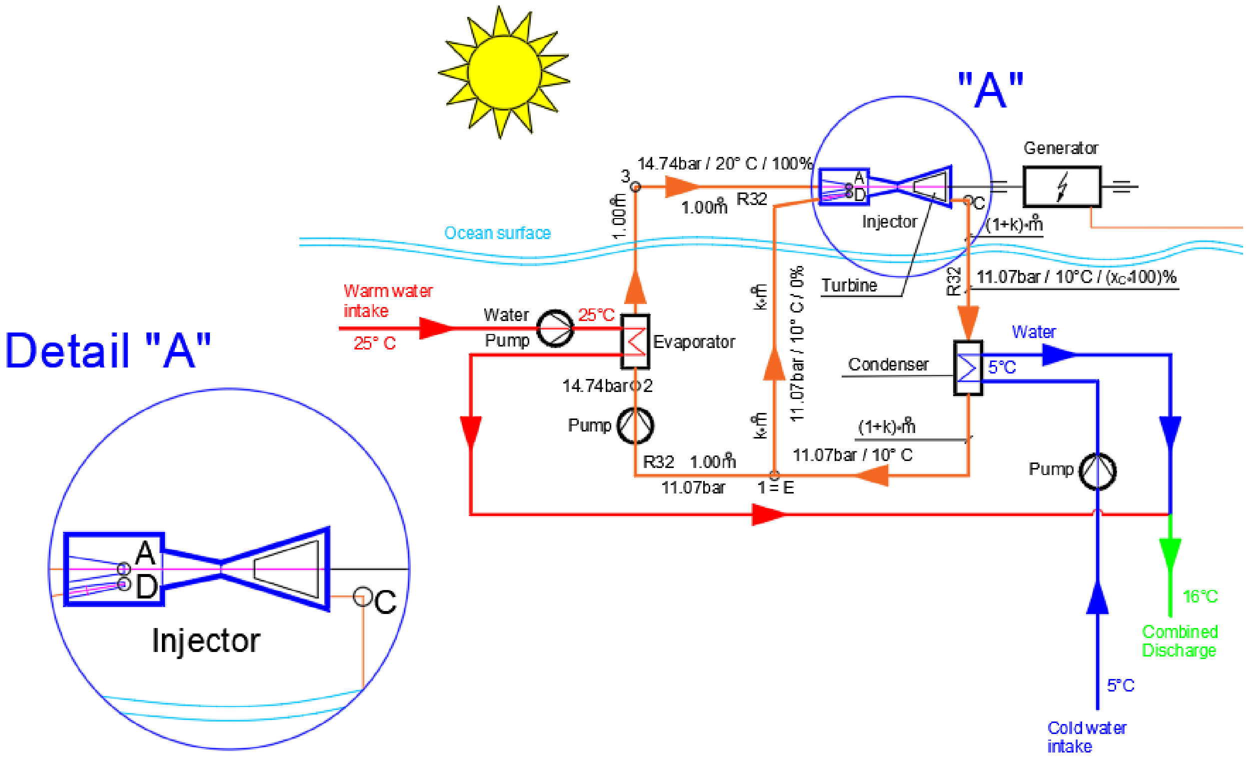

Figure 2. Schematic diagram of the IPC applied in an OTEC power system.

3.2. Calculation and Discussion

The initial conditions for the calculation of the IPC are identical to those used for the calculation of the RPC applied in the OTEC power system [4].

Thus, the initial conditions for the calculation of the IPC:

- Working fluid (WF): R32 Refrigerant
- Heat source temperature: tSO=25°C
- Heat sink temperature: tSI=5°C
- Evaporating temperature in the evaporator: t3=20°C=293.15 K
- Condensing temperature in the condenser: t1 =10°C=283.15 K

The 1st Law of Thermodynamics, as outlined in reference [6], is applied to the processes occurring within the thermal compressor and is given in equation (1):

$$q_{in} + w_{in} + \sum[\dot{m}_{in} \cdot (h_{in} + v_{in}^2/2 + g \cdot z_{in})] = q_{out} + w_{out} + \sum[\dot{m}_{out} \cdot (h_{out} + v_{out}^2/2 + g \cdot z_{out})] \tag{1}$$

The equation that determines the enthalpy of the mixture exiting the thermal TC (injector), based on the First Law of Thermodynamics, is given in equation (2):

$$h_C = [(k \cdot h_1) + h_3] / (1+k) - w_T = 478.98 \text{ kJ/kg} \tag{2}$$

where:

$w_T = 9.288$ kJ/kg – the specific technical work output of the turbine per unit mass of the Working Fluid (WF)

$k = \dot{m}_E / \dot{m}_3 = \dot{m}_1 / \dot{m}_3 = 0.10473$

$q_{in} = 0$, $q_{out} = 0$, $z_{in} = 0$, $z_{out} = 0$, $w_{in} = 0$

The technical work output of the Injection Power System (IPS) is determined using equation (3):

$$W_{IPS} = (1 + k) \cdot w_T + w_{P1\text{-}2} = 9.751 \text{ kJ/kg} \tag{3}$$

The heat intake of the IPS is calculated as shown in equation (4):

$$Q_{IPSin} = q_{2\text{-}3} = 298.16 \text{ kJ/kg} \tag{4}$$

The heat rejected by the IPS is determined using equation (5):

$$Q_{IPSout} = (1 + k) \cdot q_{C\text{-}1} = 288.41 \text{ kJ/kg} \tag{5}$$

Efficiency of the IPS is given by equation (6) :

$$\eta_{IPS} = w_{IPC} / q_{IPCin} = 0.032703 \tag{6}$$

The temperature range in which the IPS operates is expressed in equation (7):

$$_{\Delta}t_{IPSmax} = t_3 - t_A = 67.5\text{°C} = 67.5 \text{ K} \tag{7}$$

The efficiency of the process in a diffuser combining compression, work generation and internal cooling, is determined using equation (8):

$$\eta_{comb} = [h_C - h_A + w_T \cdot (1 + k)] / (h_3 - h_A) = 0.6000 = 60.00\ \% \tag{8}$$

The states and properties of the WF R32 for the IPC are presented in Table 1. Fluid states denoted with the letter 'r' (such as 2r, Ar, and Dr) correspond to the states achieved at the end of reversible (frictionless) processes. Additionally, fluid states denoted by the letter 's' (e.g., 2s) correspond to the state of saturated fluid (e.g., liquid).

The validity of the calculation is confirmed by the complete adherence of the Injection Power System (IPS) to the Law of Conservation of Energy, as demonstrated in equation (9):

$$Q_{IPSin} - Q_{IPSout} - W_{IPS} = 298.16 - 288.41 - 9.75 = 0 \tag{9}$$

This would not be achievable if any errors of any kind were present.

Table 1. States and properties of the R32 WF for the IPC

State	**T** **˚C**	**p** **bar**	**h** **kJ/kg**	**s** **kJ/(kg·K)**	**x** **-**	**ρ** **kg/m³**	**η** **-**	**wt** **kJ/kg**	**q** **kJ/kg**
1=E	10.00	11.07	217.91	1.0629	0.0000	1,020.00			
2r	10.20	14.74	218.27	1.0642				-0.36	
2	10.29	14.74	218.42	1.0648			0.71	-0.51	
2s	20.00	14.74	236.16	1.1249	0.0000	981.00			
3	20.00	14.74	516.58	2.0815	1.0000	40.83			298.16
Ar	-47.5	1.25	436.17	2.0815	0.8335	202.98		80.41	
A	-47.5	1.25	448.23	2.1349	0.8654	164.76	0.85	68.35	
Dr	-47.5	1.25	206.32	1.0629				11.59	
D	-47.5	1.25	208.06	1.0706	0.2298	925.84	0.85	9.85	
C	10.00	11.07	478.98	1.9849	0.8742	154.71	0.60		261.07

The Author has reviewed other research papers on similar power-generating systems. However, no paper was found that provides a detailed determination of all characteristic states of the working fluid. Specifically, for each state, comprehensive information on temperature, pressure, enthalpy, and entropy should be provided. Such thermodynamic data would allow other researchers to verify both the energy balance (according to the Law of Energy Conservation) and the entropy balance.

Additionally, when evaluating the efficiency of two power-generating systems, the temperature difference between the evaporating and condensing stages must be identical to ensure a valid comparison.

In this regard, the following research papers can be discussed conditionally:

Lee et al. [8]: The efficiency of the OTEC power cycle utilizing an ejector (thermal compressor/injector) is 2.47%, which is 15% higher than that of the basic OTEC power cycle (2.2%). The working fluid used in this analysis is R32. Although the temperature difference between the evaporating and condensing stages is 11.76°C, which is 17.6% larger than that observed in the IPS, the overall efficiencies of both OTEC power cycles remain significantly lower—24.5% and 32.7%, respectively—compared to the overall efficiency of the IPS presented in the present study (3.27%).

Yoon et al. [9]: This study demonstrated that the maximum thermal efficiency of the basic Ocean Thermal Energy Conversion (OTEC) power cycle is 3.009%, which remains 8% lower than that of the Injection Power Cycle (IPC) under identical temperature differences of 10°C between the evaporation and condensation stages and assuming a turbine efficiency of 0.95. Notably, this result was obtained using Ammonia (R717) as the Working Fluid (WF).

Faizal et al. [10]: In this experimental study of a closed-cycle demonstration OTEC plant, a maximum overall thermal efficiency of approximately 1.5% was achieved with a larger temperature difference between the warm and cold water (25°C), compared to the 20°C temperature difference in the present study. The working fluid used was R134-a.

Cerezo-Acevedo et al. [11]: This study details the development of the design for a closed-cycle OTEC prototype plant. For a temperature difference of 20°C between surface and deep seawater, and using R152a as the working fluid, the expected thermal efficiency of the system is 2.4%. This represents a 26.6% reduction compared to the overall efficiency of the Injection Power System (IPS) presented in the present study, which is 3.27%.

Ma et al. [12]: In this study, the thermal efficiency of the OTEC power-generation sub-cycle (PGC) is 2.19%, which is 33% lower than the overall efficiency of the Injection Power System (IPS) presented in the present study (3.27%), despite the temperature difference between the evaporating and condensing stages being 14.5°C—45% larger than that observed in the IPS. The PGC is essentially a Rankine Power Cycle (RPC) utilizing ammonia (R717) as its working fluid.

Samsuri et al. [13]: In this study, the authors reported that the thermal efficiencies of two OTEC power systems were 3.43% and 7.98% for the Basic OTEC Rankine Cycle and the proposed OTEC Rankine Cycle (with two-stage turbines), respectively. The working fluid (WF) in both power cycles was ammonia (R717). However, the temperature difference between the evaporation and condensation stages was 20°C, which is twice the temperature difference observed in the present study (10°C). Additionally, the authors made errors in equations (2)–(4), where they incorrectly summed energy parameters (kW) with mass flow parameters (kg/s), and they also violated the law of conservation of energy, as their study shows that $Qin - Qout + Wp(wf) - Wt \neq 0$.

Yuan et al. [14]: The maximum thermal efficiency of the ejector power cycle experimental test facility was 0.84%, which is 74% lower than that observed in the present study, for a temperature difference between the warm and hot water of 25°C (25% larger than that in the present study). The working fluid used in this study was an ammonia-water mixture (solution).

Ahmad et al. [15]: The thermal efficiency of the Kalina Cycle System 11 (KCS 11) is reported to be only 1.6721%, with a working fluid (WF) outlet temperature of 130°C at the vapor generator (turbine inlet), corresponding to a temperature difference of 103°C between the evaporation and condensation stages. Notably, the efficiency of both the pump and the turbine was assumed to be 100%. Despite these idealized assumptions, the reported thermal efficiency of the Kalina Cycle is 48.9% lower than the efficiency achieved in the present study. The working fluid used in the KCS 11 system was a binary ammonia-water mixture.

Ikegami et al. [16]: The thermal efficiency of the double-stage Rankine Cycle reported in this study is 3.25%, which is slightly lower than that observed in the present study. However, the temperature difference between the evaporation and condensation stages is 11.5°C, which is 15% higher than that in the present study. The working fluid used in this study was an ammonia (R717).

Uehara et al. [17]: In this study, the authors compared two power cycles: the Kalina Cycle and the Uehara Cycle, reporting thermal efficiencies of 5.22% and 5.30%, respectively. However, both cycles—particularly the Uehara Cycle—are more complex than the Injection Power Cycle (IPC) and operate with temperature differences between the evaporation and condensation stages of 19.5°C and 20.24°C, respectively, which are 95% and 102.4% higher than those observed in the present study. Moreover, the authors calculated the performance of individual processes, including the expansion of the working fluid (WF) in the turbine and the pressurization of the WF in the pump, under idealized conditions that assume the absence of friction losses. These calculations assumed 100% efficiency for the turbines and pumps, a critical detail that was not acknowledged by many subsequent studies citing these results. Presenting data in this way introduces confusion into the scientific literature, as it overestimates the achievable thermal efficiencies of these cycles under real-world conditions.

Yi et al. [18]: The authors reported thermal efficiencies of 1.82% and 2.28% for the ejector absorption double-stage power cycle and the two-stage absorption power cycle, respectively, with a temperature difference between the warm and cold seawater of 23°C. The working fluid (WF) selected for both cycles was an ammonia-water mixture. A significant limitation of the mathematical model used in their analysis is the assumption that all processes within the ejector are ideal, meaning frictional losses are entirely neglected. Despite this idealized assumption, the reported thermal efficiencies are considerably lower than those achieved in the present study.

As oceans convert solar energy into thermal energy, they function as natural solar thermal collectors and represent the largest such collectors on Earth. However, the temperature difference between the surface warm-water layer and the deep cold-water layer remains relatively small. Consequently, a power cycle intended for application in Ocean Thermal Energy Conversion (OTEC) systems must achieve superior performance compared to existing power cycles to operate effectively under these conditions.

Furthermore, OTEC power, or power derived from another low-temperature heat source, can be utilized to produce the following (refer to **Figure 3**):

potable water through seawater desalination and

Hydrogen through electrolysis of water (Khan et al. [5]).

Hydrogen can be further stored in energy carriers such as formic acid, an organic liquid hydrogen storage medium (Eppinger et al. [7]).

Formic acid can be transported to regions that are not in proximity to OTEC resources, providing a versatile fuel option for various applications. It can serve as a fuel for electric vehicles (EVs) equipped with fuel cells, internal combustion vehicles (e.g., those using reciprocating engines or gas turbines), and stationary power plants. The ability to store and transport hydrogen within a liquid carrier at ambient conditions—room temperature and atmospheric pressure—offers several advantages over conventional hydrogen storage methods. These include the elimination of large, heavy storage tanks and the avoidance of significant changes to the existing global fuelling infrastructure. Additionally, formic acid is a stable and safe energy storage medium with low flammability, further enhancing its practicality for widespread use.

According to Khan et al.[5], the authors focused on a process involving Seawater Reverse Osmosis (SRO) coupled with Proton Exchange Membrane (PEM) electrolysis. Their findings indicate that the capital and operating costs of SRO are negligible, as commercial water electrolysis consumes significantly more energy compared to the SRO process.

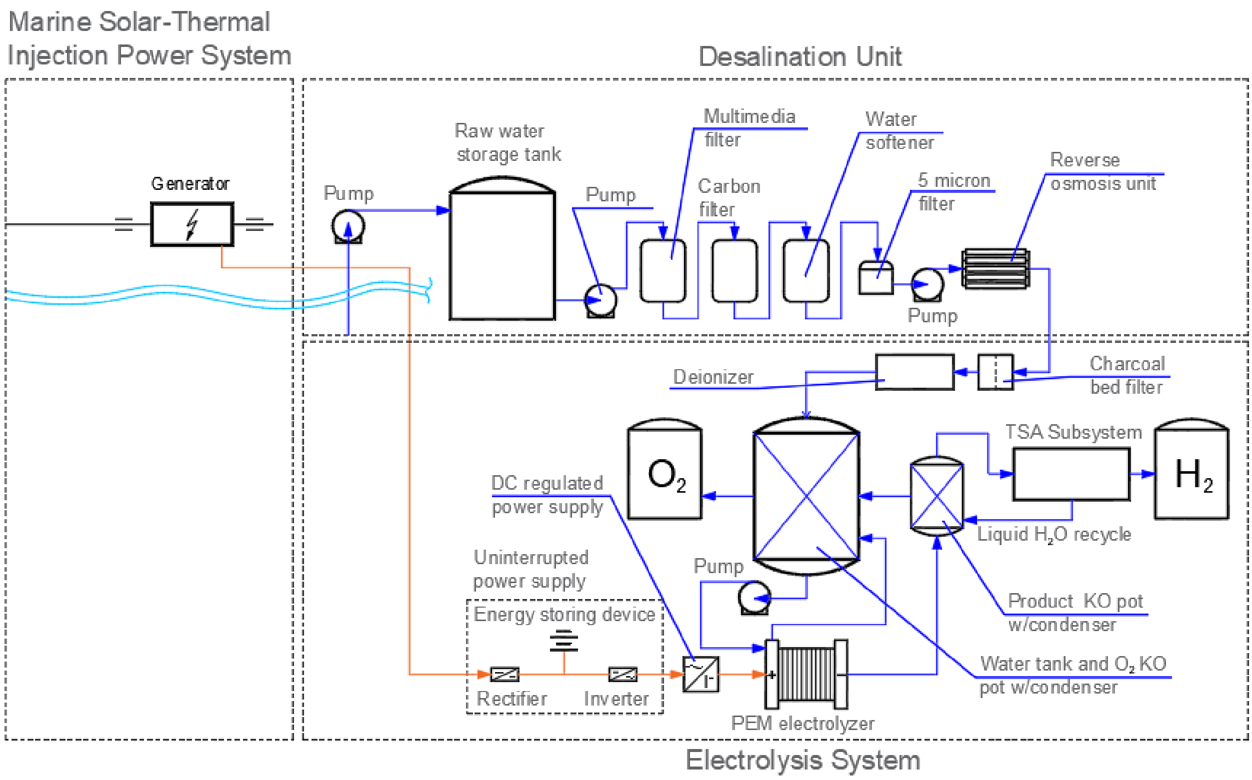

Figure 3. Schematic diagram of a seawater desalination-electrolysis plant powered by a marine solar-thermal Injection Power System (IPS).

4. Conclusions

Firstly, the application of a Thermal Compressor (TC)—also referred to as an injector or ejector—in a power cycle enables an increase in the temperature range within which the cycle operates. In this study, the temperature range increased from ΔtRPC=10K to ΔtIPCmax=67.5K, leading to an improvement in the efficiency of the real power cycle (accounting for friction) from ηRPC=0.03099 to ηIPC=0.032703, representing a 5.53% increase.

In comparison with findings from other studies, the distinctive feature of the Injection Power Cycle (IPC)—its ability to operate across a temperature range exceeding the temperature difference between the heat source and the heat sink—yields an efficiency improvement of at least 8.68 % in the real power cycle.

Therefore, this study demonstrates the potential of the proposed thermodynamic cycle. While the efficiency of individual processes in the Injection Power Cycle (with an efficiency of 0.85 for the expansion of the motive stream, 0.85 for the expansion of the cooling stream and 0.60 for the combined process in the diffuser) is lower than those in the Rankine Power Cycle (with an efficiency of 0.95 for the expansion of the working fluid in the turbine), the IPC achieves higher overall efficiency compared to the RPC. The primary objective of this research paper is to demonstrate the potential of a novel power-generating technology, with the expectation that subsequent research efforts will validate the concept and optimize the performance of the Injection Power System (IPS) through the development and testing of a prototype.

Next, OTEC power, or power derived from another low-temperature heat source, can be utilized to drive a seawater desalination-electrolysis plant (refer to Figure 3) to produce potable water and hydrogen. The generated hydrogen can subsequently be stored in organic liquid hydrogen carriers, such as formic acid.

Furthermore, formic acid can be transported to regions distant from OTEC resources. It can serve as a fuel for electric vehicles (EVs) equipped with fuel cells, internal combustion vehicles (e.g., those powered by reciprocating engines or gas turbines), and stationary power plants.

Thus, the transition from traditional "black" oil fields to sustainable "blue" hydrogen fields may become a reality in the near future.

Additionally, for larger differences between the evaporating and the condensing temperatures, the IPC can be integrated into combined power cycles, serving as a secondary power cycle with the primary cycle being, for example, the RPC.

In conclusion, under conditions of a small temperature difference between the evaporating and condensing temperatures, the Injection Power Cycle (IPC) achieves the highest overall efficiency compared to the Rankine Power Cycle RPC) analyzed in this study.

The discovery of these novel energy technologies—the Injection Power System (IPS) and the Thermal Compressor (TC) with internal cooling—represents a transformative milestone, initiating a new paradigm in the advancement of thermodynamic science and engineering.

Additionally, there is potential to further improve the IPC's efficiency by increasing the effectiveness of the diffuser process, which combines compression, work performance, and cooling. In the present analysis, the combined efficiency of this process is relatively low, with a value $\eta comb = 0.6000 = 60.00$ %. This indicates that, upon the successful development of a fully functional prototype of the Thermal Compressor (TC), integrating internal cooling and a turbine, a significant enhancement in the TC's efficiency is expected. Consequently, this improvement is anticipated to contribute to a substantial increase in the overall efficiency of the Injection Power System (IPS). Furthermore, the overall performance of the IPS could be further enhanced by employing ammonia (R717) or an ammonia-water mixture as the working fluid.

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Institutional Review Board Statement

Not applicable.

Informed Consent Statement

Not applicable.

Data Availability Statement

The calculation method and data collection process are described in sufficient detail in Section 3, „Methods."

Conflicts of Interest

The author declares no conflict of interest.

Appendix A

Table 2. Nomenclature

t	temperature (°C)
p	pressure (bar)
h	enthalpy (kJ/kg)
s	entropy (kJ/kg·K)
$\dot{m}$	mass flow (kg/s)
v	velocity (m/s)
z	height (m)
g	gravity of Earth (9.81 m/s2)
x	vapor quality (-)
ρ	density (kg/m3)
q	amount of heat per unit mass exchanged with its surroundings (kJ/kg)
w	amount of technical work per unit mass done by the system on its surroundings (kJ/kg)
Q	The heat exchanged between the system and its surroundings (kJ/kg)
W	technical work output of the system (kJ/kg)
η	efficiency (%)

Appendix B

Table 3. List of Acronyms

IPC	Injection Power Cycle
IPS	Injection Power System
RPC	Rankine Power Cycle
ERC	Ejector Refrigeration Cycle
OTEC	Ocean Thermal Energy Conversion
TC	Thermal Compressor
WF	Working Fluid
PEM	Proton Exchange Membrane
SRO	Seawater Reverse Osmosis
TSA	Temperature Swing Adsorption
in	inlet
out	outlet
r	reversible (process)
s	saturation (state)
WES	World Engineer's Summit

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