Exergoeconomic analysis of cascaded organic power plant for the Port Harcourt climatic zone, Nigeria

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Abstract: This paper presents the exergoeconomic analysis of a 100 kW solar driven organic Rankine cycle (ORC) power plant for the Port Harcourt climatic zone, latitude 4.5–5.5ºN and longitude 6.5–7.5ºE, at an ambient temperature range of 23–31°C. A cascade cycle of R134a and R290 working fluids was considered for the proposed plant. The relationships between thermodynamic properties and characteristics were formulated and numerical solutions obtained in the Microsoft Excel and MATLAB environments for the assessment of the performance of the plant. The size and mass flow rate of water through the flat plate solar collector, mass flow rates, efficiencies, and other relevant parameters of the cycles were determined. The energy and exergy efficiencies of the proposed plant, at the optimal collector operation, are 18.92 and 21.61%, respectively. The total capital investment, levelized cost of energy, payback time and the earning power of the investment were estimated to be 352 US$/kW, 0.0072 US$/kWh, 2 years 7 months, and 14.3%, respectively. The unit cost of electricity obtained did not consider energy storage, which would have significantly increase the unit cost of electricity. The component exergoeconomic factors, the relative cost difference and the average specific cost of revenue of the plant were also determined. These results might be particularly useful to researchers and energy engineers who might wish to optimize the system for effective electricity generation. From simulations performed, it is viable to install an ORC power plant in the climatic zone considered in this study.

PUBLIC INTEREST STATEMENT

This research work was carried out to configure and conduct a performance and economic assessment of a low-temperature solar-operated system that would supply electric power of 100 kW capable of meeting the load demand of about 30–50 families in the rural areas around Port Harcourt city, which is a hot and humid riverine area in Southern Nigeria. Free, renewable and environment-friendly solar energy and low-temperature boiling fluids were used to generate the power. The performance and economic indicators were found to be exceedingly good; the time it would take an investor to recoup their money and start making profit was 2 years and 7 months, and one kilowatt-hour of electric energy would cost 0.72cent. However, these costs would significantly increase, if energy storage facilities (batteries) were incorporated into the proposed power system to ensure 24/7 power supply as opposed to the one proposed here, which depended on the availability of sunshine.
1. Introduction

Many developing countries such as Nigeria with an increasing population, expanding economy and rise in energy demands are faced with shortages in electric power supply. For example, the electricity supply by the Nigerian Energy Supply Industry is totally insufficient and unreliable. The acute shortage of energy supply has seen many businesses in Nigeria to own independent diesel power generators, with serious consequences on the environment (Oko, Diemuodeke, Omunakwe, & Nnamdi, 2012). Analysis also suggests that small scale concentrating solar power systems with organic Rankine cycle (ORC) power plant could be economically competitive with photovoltaic and diesel generators based on the levelized cost of electricity (LCOE), in the range $0.30–$0.50 per kWh, for off-grid duty (Matthew, Amy, Sylvain, & Harold, 2010). They argue that optimized feedback control architecture of the ORC is necessary to obtain the balance between fixed characteristics of the fluid machinery and the variable input from the solar field. There is the need to seek and adopt renewable energy sources as well as technologies which improve the conversion efficiency and diminish the negative impacts of fossil engines (Galanis, Cayer, Roy, Denis, & Désilets, 2009). Therefore, the utilization of renewable energy can help reduce fossil fuel consumption and alleviate environmental problems (Song, Wang, Dai, & Zhou, 2012). The ORC power plant is one of such technology which has the advantage of utilizing low-grade heat sources such as biomass, geothermal heat, waste heat and solar radiation compared to the conventional Rankine cycle (Mago, Chamra, & Somayaji, 2006). The ORC is a thermodynamic vapour and liquid cycle which uses an organic working fluid to generate electricity.

According to Nag (2012), solar thermal cycles are classified in respect of the flat-plate temperature as: low for temperatures not greater than 100°C; medium for temperatures between 150 and 300°C; and high for temperatures above 300°C. Prabhu (2006) investigated the optimization and economics of solar trough ORC electricity systems and concludes that small units can be easily built to meet local retail demand. The working fluids (refrigerants) suitable for the ORCs have been studied by Mago et al. (2006); and Saleh, Koglbauer, Wendland, and Fischer (2007) perform a thermodynamic analysis of 31 pure working fluids for ORC cycles operating between 30 and 100°C and find that dry fluids yield the highest thermal efficiencies in subcritical cycles with regenerator. Gu and Sato (2002) analysed the supercritical cycles using R290, R125 and R134a as the working fluids for a particular geothermal resource (saturated liquid brine at 229.4°C, 2,763 kPa with a mass flow rate of 230.2 kg/s), and found that propane and R134a produce greater power output compared to R125; and that they are appropriate working fluids of supercritical cycles for geothermal binary design. Zhang, Yamaguchi, Fujima, Enomoto, and Sawada (2007) analysed a solar powered CO2 transcritical plant requiring 0.3–10 kW of electricity and 1–50 kW of heat, and establish that the electric power output increases with solar radiation, air temperature, collector area and water inlet temperature, but decreases with increasing heat exchanger area. Kane, Larrain, Favrat, and Allani (2003) study the coupling of linear Fresnel collectors with a cascaded 9kWe ORC, using R123 and R134a as working fluids. They conclude that an overall efficiency of 7.74% is obtainable, with a collector efficiency of 57%. Jing, Gang, and Jie (2010) developed a model of an ORC cycle using R123 as working fluid and coupled to solar collectors. The efficiency of an ORC is estimated to be between 10 and 20% depending on the temperature levels of the evaporator and condenser (Sami, 2008). According to Galanis et al. (2009), the solar ORC power plant located in Nevada in USA is capable of generating 64 MW of power in 140 hectares of desert to power 40,000 homes and generate electricity at a competitive 9–13 cents per kWh. They, therefore, suggest that plants with 1–10 MW capacity would be economically attractive for isolated communities in Africa.
Quoilin, Orosz, Hemond, and Lemort (2011) suggest that the selection of the optimal evaporating temperature be based on a trade-off between the collector efficiency and cycle efficiency with a maximum overall efficiency close to 8% for a typical mid-season ambient temperature of 15°C. Manolakos, Papadakis, Kyritsis, and Bouzianas (2007) in studying a 2 kWe low-temperature solar ORC with R134a as working fluid and evacuated tube collectors, establish an overall efficiency below 4%. Wang et al. (2010) also studied a 1.6 kWe solar ORC using a rolling piston expander, and recorded an overall efficiency of 4.2% with evacuated tube collectors and 3.2% with flat-plate collectors. They predict that an overall efficiency of about 7.9% for a solar insolation of 800 W/m² and an evaporating temperature of 147°C is obtainable. The 1 MWe Sagaru Solar ORC plant in Arizona uses n-pentane as working fluid and shows an overall efficiency of 12.1%, for a collector efficiency of 59% (Canada, Cohen, Cabel, Brosseau, & Price, 2004). According to McMahan (2006), the advantages of an ORC make its technology more economically attractive when used at small and medium power scales.

Pall and Maciej (2009) prove the usefulness of exergy-aided thermo economics optimisation thermal power plants. According to César et al. (2012), thermo economic analysis through appropriate mathematical relationships, based on the Second law of Thermodynamics and economics concepts, can quantify production cost associated with exergy losses. They conclude that the organic Rankine cogeneration cycles which use alkyl benzenes as the working fluid have the lowest production cost of electric energy, about 157–178 US$10^6/kWh for simple ORC. Paulus (2006), and Audrius and Nick (2011) show two rules for formulating thermo economic equations: the F- and P-principles, which are necessary for finding the specific costs of exergy associated with each plant component. The F-Principle states that the total cost associated with the removal of exergy must be equal to the cost at which the removed exergy was supplied; whereas the P-Principle states that each exergy unit is supplied to any stream associated with the product at the same average cost. Exergoeconomic analysis has been successfully applied to solar thermal power plants to optimize heat exchangers (Audrius & Nick, 2011). The specific exergy costing method (SPECOM) is used to evaluate micro combined heat and power (CHP) systems exergy cost of electricity: micro GT system is 0.107 EUR/kWh and ORC system is 0.207 EUR/kWh (Audrius & Nick, 2011). The specific exergy cost considers the interaction of a system and its environment and the effect of the irreversibilities in the system on cost.

Therefore, this paper seeks to assess the viability of operating an ORC power plant in Port Harcourt climatic zone, Nigeria, which is located at latitude of 4.5–5.5°N and longitude of 6.5–7.5°E, based on exergoeconomic analysis. Findings from the paper would assist in formulating economic and energy policies towards solving the perennial energy crisis in Nigeria.

2. Problem formulation and solution methods

The plant and thermodynamic diagrams of the organic Rankine power plant under consideration are shown in Figures 1 and 2. The flat plate solar collector (FPSC) receives thermal radiation and transfers thermal energy to the collector fluid, water (R718), which the collector fluid pump (CFP) circulates through the thermal storage tank (TST) and the FPSC. The heating fluid (R600a), which is circulated by the heating fluid pump (HFP), absorbs thermal energy from the solar collector fluid in the TST, and uses part of it to evaporate and superheat the working fluid (R134a) at high pressure in the vapour generator (VG) from state 5 to state 1. The superheated R134a then enters the vapour turbine (VT), which expands to produce power, $W_T$. The spent vapour leaves the VT at state 2; desuperheats in the desuperheating regenerator (RG) to state 2; condenses to state 2f and sub-cools to state 3 in the R134a condenser-evaporator (CON-EV). It is then pumped to state 4 by the working fluid pump (WFP), preheated to state 5 and 5f in the RG and condenser-preheater (CON-PH) before entering the VG to repeat the cyclic process. The cascading R290 reversed cycle, processes 6–7–8–9–6, is used to achieve the condensation and preheating of the power cycle. The low-pressure, low-temperature R290 exiting the CON-EV at state 6 is compressed to state 7 by the vapour compressor (VC); condensed to state 8 in the CON–PH; throttled to state 9 in the expansion valve (ExV); and
R134a is selected as the working fluid since is a good working fluid candidate for heat source at low temperatures in ORC applications (Vélez et al., 2012). Moreover, the choice of R134a as the working fluid favours the stratosphere since there is an intense campaign to adopt more environmental-friendly refrigerants (US Environmental Protection Agency, 2006). The choice of R600a as the heating fluid is on it has no adverse effects on materials, and has relatively high critical temperature over the working fluids.

2.1. Energy and exergy analysis
The Rankine cycle and vapour compression cycles, Figure 2 are the bases of this analysis, using the First and Second laws of Thermodynamics.

2.1.1. Energy analysis
The steady flow energy equation is applied to each unit of the plant.

The actual vapour turbine (VT) work $W_T$ [kW] is given as
where \( w_T \) [kJ/kg] is the specific turbine work, \( m \) [kg/s], \( h_1 \) and \( h_2 \) [kJ/kg] are the R134a mass flow rate, and the specific enthalpies at the turbine inlet and outlet, respectively; and \( \eta_T \) [-] is the isentropic turbine efficiency, defined as

\[
\eta_T = \frac{h_1 - h_2}{h'_1 - h'_2}
\]

Energy balance in the CON-EV is given as

\[
\dot{Q}_e = \dot{m}(h_{2g} - h_3) = \dot{m}(h_{6g} - h_{8f})
\]

where \( \dot{m} \) [kg/s], \( \dot{Q}_e \) [kW], \( h_{2g}, h_3, h_{6g} \) and \( h_{8f} \) [kJ/kg] are the mass flow rate of R290, refrigeration capacity and specific enthalpies, respectively.

The WFP power, \( \dot{W}_p \) [kW] is given as

\[
\dot{W}_p = \dot{m}w_P \approx \dot{m}v_3(P_b - P_k)
\]

where \( v_3 \) [m³/kg], \( w_P \) [kJ/kg], \( P_b \), and \( P_k \) [kPa], are the specific volume of the working fluid at pump inlet, specific pump work, vapour generator (boiler) pressure, and the condenser-evaporator pressure, respectively; and \( \eta_p \) [-] is the isentropic pump efficiency, defined as

\[
\eta_p = \frac{h_4 - h_3}{h'_4 - h'_3}
\]

The vapour generator (VG) heat interaction is given as

\[
\dot{Q}_b = \dot{m}(h_1 - h_{5f}) = \dot{m}_{hg}(h_{13} - h_{14})
\]

where \( \dot{m}_{hg} \) [kg/s], \( h_{13} \) and \( h_{14} \) [kJ/kg] are the mass flow rate of the heating fluid (R600a), inlet and exit specific enthalpies, respectively, Figures 1 and 2.

The refrigerant (R290) actual vapour compressor power is given as

\[
\dot{W}_{VC} = \dot{m}|w_{VC}| = \dot{m}\frac{h_7 - h_{6g}}{\eta_{VC}}
\]

where \( w_{VC}, h_7 \) and \( h_{6g} \) [kJ/kg], are the specific vapour compressor work, inlet and exit specific enthalpies, respectively; and \( \eta_{VC} \) [-] is the isentropic vapour compressor efficiency, defined as

\[
\eta_{VC} = \frac{h_7 - h_{6g}}{h'_7 - h'_{6g}}
\]

The net specific work is expressed as

\[
\dot{w}_{net} = \dot{w}_r - |\dot{w}_{VC}| - |\dot{w}_p| - |\dot{w}_{CFP}| - |\dot{w}_{HFP}| \approx \dot{w}_r - 3|\dot{w}_p| - |\dot{w}_{VC}|
\]

where \( \dot{w}_{CFP} \) and \( \dot{w}_{HFP} \) [kW] are the collector feed pump and the heating feed pump, respectively.

The CON-PH heat interaction, \( Q_{con-ph} \) [kW] is given as
Assuming no heat loss from the system, the energy balance is

\[ Q_{\text{con-ph}} = m(h_f - h_s) = m_r(h_r - h_{bf}) \]  

(7)

The rate of heat transfer \( \dot{Q} \) (kW) in the collector is given by

\[ Q = U_L A_c \Delta t_{lgm} = \dot{m}_w c_{pw} \Delta t \]  

(9)

where \( U_L \) [kW/m² K] is the overall heat transfer coefficient, \( \dot{m}_w \) [kg/s] is the mass flow rate of the fluid (R718), \( A_c \) [m²] is the collector area and \( c_{pw} \) [kJ/kgK] is the specific heat capacity of water at constant pressure, respectively.

The logarithmic-mean temperature difference \( \Delta t_{lgm} \) [°C] for heat exchanger according to Oko (2008), can be calculated as

\[ \Delta t_{lgm} = \frac{\Delta t_b - \Delta t_s}{\ln \frac{\Delta t_b}{\Delta t_s}} \]  

(10)

where \( \Delta t_b \) and \( \Delta t_s \) refer to high and low temperature differences, respectively.

A measure of a flat plate collector performance is the collector efficiency, \( \eta_{I, \text{col}} \) [-], defined as the ratio of the useful energy gain per unit area to the incident solar energy over a particular time period:

\[ \eta_{I, \text{col}} = \frac{\dot{Q}/A_c}{q^*_\delta} = \frac{(\dot{m}_w c_{pw} \Delta t)/A_c}{q^*_\delta} \]  

(11)

where \( q^*_\delta \) [MJ/m² day] is the optimal insolation reaching the absorber plate, which can be obtained according to Oko and Nnamchi (2012) mathematical analysis of optimal tilt angle for low latitude climatic regions.

The combined system efficiency of the collector and the power cycle according to Quoilin et al. (2011) is given by

\[ \eta_{I, \text{overall}} = \eta_{I, \text{col}} \times \eta_{I, \text{ORC}} \]  

(12)

where \( \eta_{I, \text{overall}} \) [-] is the overall system efficiency

2.1.2. Exergy analysis

According to Oko (2008), the exergy balance for any control volume, neglecting the potential, chemical and kinetic exergies is expressed as

\[ E_{\text{heat}} + W_{CV} + \sum m e_{m} - \sum m e_{out} = E_D \]  

(13)
where $\dot{E}_D$ [kW] is the exergy destruction rate, $\dot{W}_{cv}$ [kW] is the work rate, $\dot{m}$ [kg/s] is the mass flow rate, $E_{heat}$ [kW] is the heat exergy rate, $e_{in}$ and $e_{out}$ [kJ/kg] are the specific inlet and outlet exergies, respectively.

The heat exergy rate at temperature $T_j$ in each component of the system is expressed as

$$\dot{E}_{heat} = \sum_j \left(1 - \frac{T_0}{T_j}\right) Q_j$$  \hspace{1cm} (14)

Two types of exergy efficiency are the brute-force and functional efficiencies (Arif, 2008):

1. Brute-force exergy efficiency for any system is defined as the ratio of the sum of all output exergy terms to the sum of all input exergy terms. It is a straightforward method but does not differentiate between desired streams and streams of exergy losses dissipated to the environment.

$$\eta_{II,bf} = \frac{\dot{E}_{out}}{\dot{E}_i}$$  \hspace{1cm} (15)

2. Functional exergy efficiency for any system is defined as the ratio of the exergy associated with the desired energy output to the exergy associated with the energy expended to achieve the desired output.

$$\eta_{II,f} = \frac{\dot{W}_{net}}{\dot{E}_i}$$  \hspace{1cm} (16)

The flow specific exergy is given as

$$e_i = (h_i - h_0) - T_0(s_i - s_0)$$  \hspace{1cm} (17)

where $T_0$ [K] is the dead state temperature (temperature of the environment) and $i$ represents the thermodynamic state.

The exergy input to the power cycle is given by

$$\dot{E}_{13} = \dot{m}_{hf}c_{phf}\left[ (T_{13} - T_0) - T_0 \ln \left( \frac{T_{13}}{T_0} \right) \right]$$  \hspace{1cm} (18)

The exergy in and out of the collector is given as

$$\dot{E}_i = \dot{m}_{w}c_{pw}\left[ (t_i - T_0) - T_0 \ln \left( \frac{t_i}{T_0} \right) \right]$$  \hspace{1cm} (19)

where $i = 10, 11$

The Brute-force exergy efficiency of the collector is given as

$$\eta_{II, col} = \frac{\dot{E}_{11}}{\dot{E}_{11} + \dot{E}_{10}}$$  \hspace{1cm} (20)

where exergy from the sun to collector is expressed as (Arif, 2008)

$$\dot{E}_{11} = A_{s}q_{e}F_{rad,max}$$
The useful work-radiation exergy which is the maximum efficiency ratio (or exergy-to-energy ratio for radiation) for determining an exergy of thermal emission at temperature, $T_s$, is given by

$$ E_{rad,\text{max}} = 1 + \frac{1}{3} \left( \frac{T_0}{T_s} \right)^4 - \frac{4}{3} \frac{T_0}{T_s} $$

(21)

where $T_s$ [K] is the apparent black body temperature of the sun (Arif, 2008). The overall exergy efficiency of the combined system of the collector and the power cycle is the product of the two separate efficiencies, given by

$$ \eta_{II,\text{overall}} = \eta_{II,\text{col}} \times \eta_{II} $$

(22)

The total ORC irreversibility rate is given as

$$ I_T = \sum_j I_{Dj} = I_{D,\text{VG}} + I_{D,\text{VT}} + I_{D,\text{RG}} + I_{D,\text{CON-EV}} + I_{D,\text{P}} + I_{D,\text{VC}} + I_{D,\text{CON-PH}} + I_{D,\text{ExV}} $$

(23)

and

$$ I_{Dj} = T_0 S_{\text{gen},j} = E_{Dj} $$

(24)

where $n[-]$ is the number of components $I_j$ (kW) is the total irreversibility rate, and $I_{Dj}$ (kW) is the irreversibility rate and $S_{\text{gen},j}$ [kW/K] is the entropy generation rate; $j = \text{VG, VT, RG, CON-EV, P, VC, CON-PH and ExV}$.

2.2. Economic analysis

The purpose of economic analysis is to maximize profit while supplying electricity at a minimum cost to the consumer. It provides the necessary costs needed to carry out an exergoeconomic analysis. The cost of a power system includes both capital (fixed) cost and the operation and maintenance (variable) cost. The total capital investment $TCI$ [US$] of an ORC power plant is given by Pall and Maciej (2009) as

$$ TCI = C_F + C_S + C_W + C_L + C_A $$

(25)

where $C_F$ [US$] is the fixed capital investment and includes purchased equipment cost (PEC); $C_S$ [US$] is the start-up cost; $C_W$ [US$] is the working capital; $C_L$ [US$] is the cost of licensing research and development; and $C_A$ [US$] is the allowance for funds used during construction.

2.2.1. Capital budgeting rules and financing mechanism

Different methods are usually used to rank projects and to decide whether or not they should be accepted for inclusion in the capital budgeting; only the net present value (NPV) and the payback time (PBT) concepts are given attention in this work. The rationale for the NPV is that if NPV is zero, the project’s cash flow is exactly sufficient to pay back the invested capital and provide the required rate of return on that capital. Also, if NPV is positive, then it is generating more cash than is needed to service debt, provide the return to shareholders, results in a positive economic value added (EVA) as well as a positive market value added (MVA). NPV relies on discounted cash flow (DCF) techniques and is expressed as

$$ NPV = CF_0 + \frac{CF_1}{(1+k)^1} + \frac{CF_2}{(1+k)^2} + \ldots + \frac{CF_n}{(1+k)^n} = \sum_{t=0}^{N} \frac{CF_t}{(1+k)^t} $$

(26)

where $CF_t$ [S] is the accepted net cash flow at period $t$ [years], $k[-]$ is the project’s cost of capital, and $N$ [years] is the plant economic life; $CF_t$ [S] is treated as negative since it represents cash outflows.

2.2.2. Levelized cost of electricity

Levelization is the process of converting a series of non-uniform costs into a uniform series. Both series have the same present value (Black & Veatch, 2001). The LCOE is the price at which electricity must be generated from a specific source to break even over the lifetime of the project. It is an
economic assessment of the cost of the energy generating system including the initial investment, operations and maintenance, cost of fuel and cost of capital investment over its life measured in price per kWh or per MWh. According to Hazlehurst (2009), the LCOE is given as

\[
LCOE = \frac{NPV_{CF}}{NPV_E} \tag{27}
\]

where \(NPV_{CF} \text{[US$]}\) and \(NPV_E \text{[kWh]}\) are the net present value of cash flow and energy, respectively. Annuity is a series of equal cash flows occurring during some period of time, usually on yearly bases (Pall & Maciej, 2009), as given below:

\[
A = CRF \times PV = \frac{i_{eff} \left(1 + i_{eff}\right)^N}{\left(1 + i_{eff}\right)^N - 1} \times PV \tag{28}
\]

where \(i_{eff} \text{[−]}\), \(CRF \text{[−]}\), \(N \text{(year)}\) and \(PV \text{(US$)}\) are the effective rate of return, capital recovery factor, plant economic life and the present value of the invested capital. By the sinking fund method of depreciation, the amount \(A_n \text{(US$)}\) set aside at the end of each year for \(N \text{ years}\) to pay back the invested capital is given as (Rajput, 2012)

\[
A_M = \left(\frac{i}{1 + i}\right)^N \left(TCI - Sv\right) \tag{29}
\]

where \(i \text{[−]}\) is the annual rate of compounded interest on the invested capital, \(Sv \text{(US$)}\) is the salvage value, \(TCI \text{(US$)}\) is the total capital investment and \(N \text{(years)}\) is the plant economic life. Annual depreciation \(D \text{(US$)}\) is given as

\[
D = \frac{PEC - Sv}{N} \tag{30}
\]

2.3. Exergoeconomic analysis of the plant

The aim of exergoeconomic analysis for power plants is to identify and quantify the exergy destruction (César et al., 2012). Exergoeconomics combines exergy and economic analysis to provide more useful information on the performance analysis of a system. Quantification of exergy losses due to the system thermal interaction with its environment is important for determining specific electricity production. Therefore, exergy costing, the interaction of a system and its environment and the effect of the irreversibilities in the system on cost, is investigated on bases of inlet or exit exergy streams for each component. The capital cost rate of the \(n\text{th}\) unit, \(\dot{Z}_n \text{[US$/h]}\) is given as

\[
\dot{Z}_n = \frac{PEC \times \phi_n \times CRF}{\tau_{op}} \tag{31}
\]

where \(PEC \text{(US$)}\) is the purchase equipment cost; \(\phi_n \text{[−]}\) is the maintenance factor for each plant component; \(CRF \text{[−]}\) is the capital recovery factor, and \(\tau_{op} \text{[hr]}\) is the annual operating hours of the plant.

The thermo-economic balance (exergy cost) equation for each component is given by Audrius and Nick (2011), and shows that the sum of cost rates allied with all inlet exergy streams plus the monetary charge connected with owning and operating the \(n\text{th}\) component equals the sum of all outlet streams, is given as:

\[
\sum (c_e \dot{E}_e) + c_w W = c_q \dot{E}_q + \sum c_i \dot{E}_i + \dot{Z}_n \tag{32}
\]

where \(c_e\) and \(c_i \text{[US$/kWh]}\) are the exit and inlet specific costs of the exergy, respectively; \(c_q \text{ and } c_e \text{[US$/kWh]}\) are the average specific costs of the exergy for work and heat transfer, respectively; \(\dot{E}_e\) and \(\dot{E}_i \text{[kW]}\) are the exit and inlet exergy streams, respectively; \(\dot{E}_q\) and \(W \text{[kW]}\) are the exergy streams for heat transfer and work, respectively; \(\dot{Z}_n \text{[US$/h]}\) is the capital cost rate (levelized cost rate) of the \(n\text{th}\) unit.
Auxiliary equations were formulated using the F and P principles in the form of linear equations according to Equation (34) below, since there are more streams than the plant components (Audrius & Nick, 2011).

\[ A \times \dot{C} = \dot{Z} \]  

(33)

where \( A \) is the coefficient matrix obtained from main and auxiliary equations, \( \dot{C} \) is the unknown column vector of cost flow rates and \( \dot{Z} \) is the column vector of capital cost flow rates. Eleven (11) auxiliary, and 12 main equations were formulated for 12 components based on the principles given above with a total of 23 streams.

Other exergoeconomic variables include the specific cost of fuel and product per unit exergy of each component and the cost rate of exergy destruction which are given by Audrius and Nick (2011) as

\[ c_{f,n} = \frac{\dot{C}_{f,n}}{E_{f,n}} \]  

(34)

Fuel exergy refers to that exergy consumed by a component to produce a useful product exergy

\[ c_{p,n} = \frac{\dot{C}_{p,n}}{E_{p,n}} \]  

(35)

where \( c_{f,n} \) [US$/kJ], \( E_{f,n} \) [kW], \( c_{p,n} \) [US$/kJ] and \( E_{p,n} \) [kW] are the specific cost of fuel, exergy of fuel, specific cost of the product of the \( n \)th unit, and the exergy of the product of the \( n \)th unit, respectively.

\[ \dot{C}_{D,n} = c_{f,n} \dot{E}_o \]  

(36)

where \( \dot{C}_{D,n} \) [US$/h] is the cost rate of exergy destroyed in the \( n \)th component; \( \dot{E}_D \) [kW] is the exergy destruction rate.

The measures of exergoeconomic performance are the relative cost difference, \( r_n [-] \) and the exergoeconomic factor, \( f_n [-] \) for the \( n \)th unit. The relative cost difference represents the relative increase in the average cost per unit exergy between the fuel and product, respectively of a component and is given as

\[ r_n = \frac{c_{p,n} - c_{f,n}}{c_{f,n}} \]  

(37)

The exergoeconomic factor, \( f_n [-] \) combines the component capital investment cost, \( Z_n \) [US$/h] with the cost rate of exergy destruction, \( \dot{C}_{D,n} \) [US$/h] and shows the contribution of the capital investment cost of the \( n \)th component to the total cost increase of the system as given below:

\[ f_n = \frac{Z_n}{Z_n + \dot{C}_{D,n}} \]  

(38)

It is a tool used in the iterative optimization of an energy system (Paulus, 2006). A very high exergoeconomic factor is an indication to reduce capital costs, and a very low value indicates that additional capital should be invested in order to reduce exergy destruction.

The specific revenue \( r_{sp} \) [US$/kJ] associated with electric power generated equals the average cost of the power (Paulus).

\[ r_{sp} = \frac{c_{qf} E_{qf} + \sum_n Z_n}{W_{net}} \]  

(39)

where \( c_{qf} \) [US$/kJ] is the specific cost of exergy of the sun; the cost of solar radiation should not be penalized since it is natural and requires no input cost, therefore it is assumed to be zero in this study.
Paulus demonstrated the two rules for formulating the auxiliary equations for the specific revenues associated with exergy flows: the F-Principle for revenue and the P-Principle for revenue, respectively; however, this paper is limited to the average cost of power as given above.

2.4. Computational algorithm

The algorithm presented here is developed based on Equations (1)–(39) inclusive for the computation and simulation of the ORC power plant.

```
start

**Energy and Exergy**
get (∅, W, t, T, P, U, t, T, τ, η, η, η) ** Input Data **
compute: ** Out command **
∅; q; m; m; m; W; Q; A; Q; **parameters to be computed**
η; η; η; η; η; η; η; using the appropriate equation **
get(n), ** Input Data, n is the No. of components **
I := 0; ** Initial sum**
for j := 1 to n **Number of components**
do
compute **Output command **
I(j) ** Irreversibility rate, kW in the jth component**
η(j); ** Exergy efficiency % in the jth component**
I := I + I(j) ** Total irreversibility kW **
j := j + 1
until j := n
end do

**Economic Analysis**
get(i, N, unit cost of O&M, unit cost of components) ** i is the interest rate**
for t := 0 to N ** where N = 1,2,...,6, plant economic life**
do
if t := 0
compute:
TCI **Total capital investment**
else
compute:
A; D; NPV, EP and CCS/D; LCOE ** Economic merit characteristics**
end if
end do

** Exergoeconomic Analysis**
get(PEC, φ, τ);
compute (..) ** Component cost rates **
compute: **using the relevant equation given above**
c; c; C; I; t; t; ** exergoeconomic characteristics**
stop
```
3. Results and discussion
The input data, output data and the key parameter simulation for the solar ORC are presented in this section. The computations were carried out based on the input data in Table 1 on the bases of Equations (1)–(40) inclusive, and the meteorological data presented by Oko and Nnamchi (2012).

3.1. Results of energy and exergy analysis
This section presents the optimal characteristics and simulations of the plant based on the input data in Table 1. Output parameters of the ORC power plant has been described in Table 2.

| Table 1. Input data for computation and simulation |
|--------------------------------------------------|
| **Component** | **Quantity** | **Symbol** | **Units** | **Value** |
| FPSC          | Sunshine duration | $\tau$ | h | 4.25 |
|               | Collector tilt angle | $\theta$ | degree | 3.69 |
|               | Absolute black body temperature | $T_s$ | °C | 5327 |
|               | Collector inlet temperature | $t_{10}$ | °C | 23–31* |
|               | Collector outlet temperature | $t_{11}$ | °C | 60–80* |
| TST           | Inlet temperature cold side | $t_{31}$ | °C | $t_{s9} - 1.0$ |
|               | Outlet temperature cold side | $t_{32}$ | °C | $t_{s9} - 0.5$ |
| VG            | Inlet temperature hot side | $t_{35}$ | °C | $t_{s9} - 5.0$ |
|               | Outlet temperature hot side | $t_{36}$ | °C | $t_{s9} - 2.0$ |
|               | Inlet temperature cold side | $t_{s9}$ | °C | 20–40* |
|               | Outlet temperature cold side | $t_{s10}$ | °C | 20–40* |
| CON-EV        | Inlet temperature hot side | $t_{s11}$ | °C | 8–12* |
|               | Outlet temperature hot side | $t_{s12}$ | °C | 5–10* |
|               | Inlet temperature cold side | $t_{s13}$ | °C | 0–5.0* |
| RG            | Inlet temperature cold side | $t_{s14}$ | °C | 5–10* |
|               | Inlet temperature hot side | $t_{s15}$ | °C | 13–15* |
|               | Outlet temperature cold side | $t_{s16}$ | °C | 11–16* |
|               | Outlet temperature hot side | $t_{s17}$ | °C | 8–12* |
| VT            | Inlet temperature | $t_{s18}$ | °C | 5–10* |
|               | Outlet temperature | $t_{s19}$ | °C | 5–10* |
| WFP           | Inlet temperature | $t_{s20}$ | °C | 5–10* |
|               | Outlet temperature | $t_{s21}$ | °C | 5–10* |
| CON-PH        | Inlet temperature hot side | $t_{s22}$ | °C | 30–35* |
|               | Outlet temperature hot side | $t_{s23}$ | °C | 15–20* |
|               | Inlet temperature cold side | $t_{s24}$ | °C | 11–16* |
|               | Outlet temperature cold side | $t_{s25}$ | °C | 20–40* |
|               | Ambient temperature | $T_0$ | °C | 27 |
|               | Atmospheric pressure | $P_0$ | kPa | 101 |
|               | Turbine isentropic efficiency | $\eta_T$ | % | 80 |
|               | Pump isentropic efficiency | $\eta_P$ | % | 70 |
|               | Vapour compressor isentropic efficiency | $\eta_{vc}$ | % | 70 |
|               | Annual working hours | $r_{op}$ | hr | 8,760 |
|               | Component maintenance factor | $\phi_{m}$ | – | 1.06 |

*The assumed range of values for the simulation.
The result of the exergy inflow stream, exergy outflow stream, exergy loss and the exergetic efficiency of the ORC units are shown in Table 3. The collector has the highest exergy destruction with exergetic efficiency of 23.07%, while the HFP has the least exergy loss with an exergetic efficiency of 99.36%.

Figure 3 shows the Grassmann diagram of the ORC power plant and shows that exergy decreases across each component due to irreversibility. The evaporator has the highest exergy destruction due to the high rate of heat interaction as compared to other components.

The net specific work increases rapidly with the evaporator temperature according to Figure 4. The turning point determines the maximum available work based on the heat source corresponding to the fixed maximum insolation (i.e. optimal net specific work for the current analysis is obtained at generator temperature of 33.85°C, Figure 4).

### Table 2. Output parameters of the ORC power plant

| Quantity                              | Symbol | Units | Value |
|---------------------------------------|--------|-------|-------|
| Mass flow rate of R718                | \( m_w \) | kg/s  | 9.82  |
| Mass flow rate of R600a               | \( m_{hf} \) | kg/s  | 8.52  |
| Mass flow rate of R134a               | \( m_r \)  | kg/s  | 2.60  |
| Mass flow rate of R290                | \( m_r \)  | kg/s  | 1.50  |
| Optimal evaporator temperature       | \( t_e \)  | °C    | 34.14 |
| Optimal collector temperature         | \( t_{10} \) | °C    | 30.11 |
| Optimal collector tilt angle          | \( \theta^* \) | degree | 3.69  |
| Optimal insolation                    | \( q_0 \)  | MJ/m² day | 14.31 |
| Collector area                        | \( A_c \)  | m²    | 2.92  |
| Net power output                      | \( W_{net} \) | kW    | 100.00 |
| ORC (First law) efficiency            | \( \eta_I \) | %     | 15.78 |
| Collector (First law) efficiency      | \( \eta_{I,col} \) | %     | 23.18 |
| Overall thermal efficiency            | \( \eta_{I,overall} \) | %     | 3.66  |
| Exergy (Second law) efficiency of the ORC | \( \eta_{II} \) | %     | 10.13 |
| Exergy efficiency of the collector    | \( \eta_{II,col} \) | %     | 16.15 |
| Overall exergy efficiency of the plant| \( \eta_{II,overall} \) | %     | 2.34  |
| Cooling capacity                      | \( Q_e \)  | kW    | 495.09 |

### Table 3. Exergy characteristics of each component of the plant

| Unit      | \( E_{in} = E_i \) (kW) | \( E_{out} = E_p \) (kW) | \( E_{in} \) (kW) | \( \eta_{II} \) (%) |
|-----------|--------------------------|---------------------------|-------------------|---------------------|
| FPSC      | 3704.37                  | 854.54                    | 2849.84           | 23.07               |
| CFP       | 1348.55                  | 1342.27                   | 6.28              | 99.53               |
| HFP       | 725.52                   | 720.83                    | 0.99              | 99.36               |
| TST       | 2465.83                  | 1575.39                   | 890.44            | 63.89               |
| VG        | 1007.58                  | 119.37                    | 888.21            | 11.85               |
| VT        | 119.37                   | 62.04                     | 57.33             | 51.97               |
| RG        | 62.26                    | 26.84                     | 35.42             | 43.11               |
| CON-EV    | 280.87                   | 41.59                     | 239.28            | 14.81               |
| WFP       | 1.88                     | 0.21                      | 1.67              | 11.29               |
| VC        | 105.62                   | 24.54                     | 81.09             | 23.23               |
| CON-PH    | 22.92                    | 3.06                      | 19.86             | 13.36               |
| ExV       | 255.65                   | 1.03                      | 254.63            | 0.40                |
Figure 5 shows that an increase in the ambient condition of collector inlet air favours the plant performance. An optimal First law thermal and overall efficiency of 18.92% and 4.58%, respectively, are obtained, which agrees with the work of Sami (2008) that the efficiency of an ORC is between 10 and 20% depending on the temperature levels of the evaporator and condenser.

The exergy efficiency of the collector decreases with increasing collector inlet temperature, while the ORC exergy efficiency and overall exergy efficiency increases to a maximum with the collector inlet temperature as shown in Figure 6. At the maximum point, any further increase in the temperature leads to a corresponding decrease in the exergy efficiencies.

An optimal evaporating temperature of about 34.14[°C] is obtained, which gives minimum overall plant irreversibility, Figure 7—same temperature minimizes the loss of exergy in each plant component; though the optimal evaporating temperature is higher than the design evaporating temperature by 3.14[°C].

3.2. Results of economic analysis
Table 4 shows the cost of components and the computed cost of total capital investment. The project cash flow and the necessary assumed economic variables are asterisked as shown in Table 5.
Table 4. Breakdown of the total capital investment of the 100 kW ORC power plant

| Component cost       | Quantity | Total cost (US$) |
|----------------------|----------|------------------|
| Direct cost          |          |                  |
| FPSC                 | 80US$/m² | 3.42 m²          | 273.6 |
| TST                  | 40/kW    | 100 kW           | 4,000 |
| HFP & CFP (290)/pc   | 1 pc     | 290              |
| VG                   | 40/kW    | 100 kW           | 4,000 |
| VT                   | 1,000/pc | 1 pc             | 1,000 |
| RG & CON-PH          | 2 × (40/kW) | 100 kW     | 8,000 |
| CON-EV               | 40/kW    | 100 kW           | 4,000 |
| WFP                  | 300/pc   | 1 pc             | 300   |
| VC                   | 1,000/pc | 1 pc             | 1,000 |
| ExV                  | 5/pc     | 1 pc             | 5     |
| Working fluids R600a, R134a and R290 | 3 × (33US$/kg) | 5 kg | 495 |
| Total PEC            |          |                  | 23,363.60 |
| Piping 7% PEC        |          |                  | 1,635.45 |
| Installation of equipment 6% of PEC |          |                  | 1,401.82 |
| Electrical equipment 4% of PEC |          |                  | 934.54 |
| Civil and structural work 7% of PEC |          |                  | 1,635.45 |
| Total direct cost    |          |                  | 28,970.86 |
| Indirect cost        |          |                  |
| Engineering and supervision 6% of PEC |          |                  | 1,401.82 |
| Construction cost 3% of direct cost |          |                  | 1,738.25 |
| Wages and salaries 3% of Direct cost |          |                  | 869.13 |
| Contingency 3% of direct cost |          |                  | 1,158.84 |
| Total indirect cost  |          |                  | 5,168.03 |
| Fixed capital investment (FCI) |          |                  | 34,138.89 |
| Other outlays        |          |                  |
| Start-up cost 1% of FCI |          |                  | 341.39 |
| Working capital 3% of PEC |          |                  | 700.91 |
| Total capital investment (TCI) |          |                  | 35,181.19 |

Source: Matthew et al. (2010) and Pall and Maciej (2009).
Table 5. ORC power plant project cash flow

| S.no | Item                                      | N (years) |
|------|-------------------------------------------|-----------|
|      |                                            | 0   | 1   | 2   | 3   | 4   | 5   | 6   |
| 1    | Cash in (US$) (eqn.32)                    | 0   | 21,108.71 | 10,051.77 | 6,377.26 | 4,548.31 | 3,457.55 | 2,735.85 |
| 2    | TCI (US$)                                 | 35,181.19 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3    | Non-depreciated capital (US$)             | 35,181.19 | 35,181.19 | 25,890.06 | 21,244.50 | 18,147.46 | 15,824.68 | 13,966.46 |
| 4    | Depreciation (US$) (eqn.33)               | 0 | 9,291.12 | 4,645.56 | 3,097.04 | 2,322.78 | 1,858.23 | 1,548.52 |
| 5    | O&M (US$)                                 | 0 | 7.32 | 14.64 | 21.96 | 29.28 | 36.60 | 43.92 |
| 6    | Taxable income (US$) (1−(4 + 5))         | 11,810.27 | 5,391.57 | 3,258.25 | 2,196.25 | 1,562.73 | 1,143.40 |
| 7    | Income tax *0.5% of (6)                   | 0 | 59.051 | 26.96 | 16.29 | 10.98 | 7.81 | 5.72 |
| 8    | Cumulative cash in (US$)                  | 0 | 21,108.71 | 31,160.48 | 37,537.74 | 42,086.05 | 45,543.60 | 48,279.45 |
| 9    | Annual cash out (2 + 5 + 7)               | 35,181.19 | 66.37 | 41.60 | 38.25 | 40.26 | 44.41 | 49.64 |
| 10   | Cumulative cash out (US$)                 | 35,181.19 | 35,247.56 | 35,289.16 | 35,327.41 | 35,367.67 | 35,412.08 | 35,461.72 |
| 11   | Annual cash surplus/deficit (1−9)         | −35,181.20 | 21,042.34 | 10,051.77 | 6,377.26 | 4,548.31 | 3,457.55 | 2,735.85 |
| 12   | Cumulative cash surplus/deficit (US$)      | −35,181.20 | −14,138.80 | −4,128.68 | 2,210.327 | 6,718.377 | 10,131.52 | 12,817.73 |
| 13   | PV of cash flow (US$) (11 × 14)           | −35,181.20 | 19,129.40 | 8,272.87 | 4,762.587 | 3,079.059 | 2,119.292 | 1,516.29 |
| 14   | DF @ k = i_eff = 10%                      | 1.00 | 0.91 | 0.83 | 0.751315 | 0.683013 | 0.620921 | 0.56 |
| 15   | NPV of cash flow (US$)                    | 3,698.31 |
| 16   | Capacity kW                               | 100 | 200 | 300 | 400 | 500 | 600 |
| 17   | Annual capacity (kWh)                     | 0 | 36,600.00 | 73,200 | 109,800 | 146,400 | 183,000 | 219,600 |
| 18   | DF                                        | 1.00 | 0.91 | 0.83 | 0.75 | 0.68 | 0.62 | 0.56 |
| 19   | PV of energy (kW)                         | 0 | 33,272.73 | 60,495.87 | 82,494.37 | 99,993.17 | 113,628.6 | 123,958.50 |
| 20   | NPV of energy (kW)                        | 513,843.2 |
| 21   | LCOE (US$/kW)                             | 0.0072 |
| 22   | PBT (years)                               | 2.60 |
| 23   | Earning power of investment, EP (%)        | 14.27 |

*Assumed values.

Table 6. Levelized cost comparison of energy systems

| Energy                       | Capital cost (US$/kW) | Capacity factor (%) | Fuel cost (US$/MWh) | O&M cost (US$/MWh) | Life, N (years) | LCOE (US$/kW) |
|------------------------------|-----------------------|--------------------|---------------------|--------------------|-----------------|--------------|
| Solar PV (Crystalline)       | 5,500                 | 18                 | 0                   | 8                  | 25              | 0.14         |
| Fuel cell DG                 | 3,250                 | 90                 | 45                  | 10                 | 20              | 0.08         |
| Solar thermal                | 4,000                 | 25                 | 0                   | 8                  | 20              | 0.09         |
| Solar ORC power plant        | 352                   | 25                 | 0                   | 0.2                | 6               | 0.007*       |
| Nat Gas (CCGT)               | 1,000                 | 85                 | 43                  | 6                  | 40              | 0.04         |
| Coal                         | 2,750                 | 85                 | 30                  | 10                 | 45              | 0.05         |
| Nuclear                      | 6,000                 | 92                 | 7                   | 17                 | 40              | 0.06         |
| Wind                         | 2,500                 | 35                 | 0                   | 10                 | 25              | 0.04         |
| Geothermal                   | 3,500                 | 95                 | 0                   | 25                 | 20              | 0.04         |

Source: Hazlehurst (2009).

*Values from the present work.
The LCOE is estimated to be 0.007197 US$/kWh, and a positive NPV of the cash flow is obtainable. This means that at 10% interest rate, the project will be economically viable and the investment will experience high economic value added on the invested capital. The firm’s wealth would increase by a NPV of 513843.2 US$ and will run clean and free electricity generation for a period of 3 years 5 months.

The levelized cost of different technologies are shown in Table 6, a close comparison shows that the unit cost of electricity and the capital cost per kW is relatively low in the 100 kW solar ORC power plant though the predicted life is low compared to other technologies.
Figure 8 shows the variation of the cumulative cash flow with the plant economic life. For a 6-year predicted life, the PBT is estimated to be 2 years 7 months while in the remaining 3 years and 5 months, the investment starts receiving a huge return on the invested capital.

The NPV profile (Figure 9) shows the relationship between the NPV and plant’s cost of capital. The earning power (EP) of the investment is 14.27%, i.e. the interest rate at which the NPV of the investment equals zero.

### 3.3. Results of exergoeconomic analysis

This section shows the results of the necessary exergoeconomic variables and the annual capital cost rate of each component. It is observed in Table 7 that the exergoeconomic factor is highest in the CFP, whereas VG and VC feature the lowest value, which means that further technical consideration is required in the development of VG and VC to reduce exergy destruction.

The plant specific cost of revenue \( r_{sp} \) is estimated to be 0.0308 (US$/kJ) and decreases with increased plant capacity, as shown in Figure 10. The figure shows the average cost associated with electricity supply.
4. Conclusion

The exergoeconomic analysis of a proposed 100 kW solar driven ORC power plant for the Port Harcourt climatic zone, Nigeria, has been carried out based on the prevailing ambient temperature range of 23–31 °C, and using R134a as the working fluid. The overall energy and exergy efficiencies of 18.92 and 21.61%, respectively, collector surface area of 2.92 m² and collector fluid (water) mass flow rate of 9.817 kg/s were obtained. It was shown that the FPSC suffered the largest exergy destruction rate, followed by the vapour generator and compressor, and the feed water pump had the least exergy destruction rate. The inclusion of the organic fluid regenerator and preheater appreciably improved the efficiency of the plant. The total capital investment, levelized cost of energy (LCOE), payback time and the earning power of the investment were estimated to be 352US$/kW, 0.0072US$/kWh, 2 years 7 months and 14.3%, respectively. This means that at 14.3% interest rate, in the remaining 3 years and 5 months of the plant's life after breaking even, the project will earn a NPV of 513843.2US$. The unit cost of electricity obtained did not consider the energy storage, which would have significantly increase the unit cost of electricity. These results may be particularly useful to researchers and energy engineers who may wish to optimize the system for effective electricity generation and refrigeration. From simulations performed, it is viable to install an ORC power plant in the climatic zone considered in this study.

Nomenclature

| Symbol | Description |
|--------|-------------|
| A      | Annuities, US$ |
| Ac     | Area of collector, m² |
| AM     | Annual sinking fund saving amount, US$ |
| CCS/D  | Cumulative cash surplus/deficit, US$ |
| C      | Specific cost of exergy, US$/kJ |
| Ĉ      | Cost rate of exergy, US$/h |
| ei     | Specific exergy, kJ/kg |
| E      | Exergy rate, kW |
| ED     | Exergy destruction rate, kW |
| EP     | Earning power of the investment, % |
| Eheat  | Heat exergy rate, Kw |
| E_rad, max | Maximum radiation exergy, kJ/kg |
| fn     | Exergoeconomic factor, % |
| h      | Specific enthalpy, kJ/kg |
| ID     | Component Irreversibility rate, kW |
\( I_T \)  Total Irreversibility rate, kW
\( k \)  cost of capital, %
LCOE  Levelized cost of electricity, US$/kWh
\( m \)  Mass flow rate of working fluid R134a, kg/s
\( m_r \)  Mass flow rate of cooling fluid R290, kg/s
\( m_{hf} \)  Mass flow rate of heat transfer fluid R600a, kg/s
\( m_w \)  Mass flow rate of the carrier fluid (R718), kg/s
\( N \)  plant economic life, years
NPV  Net present value, US$
\( \text{PBT} \)  Payback time, years
\( \dot{Q} \)  Rate of heat transfer, kW
\( Q_b \)  Heat added to the boiler, kW
\( Q_e \)  Cooling capacity, kW
\( q_s \)  Solar insolation, MJ/m² day
\( R \)  Gas constant of R600a, kJ/kgK
\( r_n \)  Relative cost difference, %
\( r_{sp} \)  Plant specific cost of revenue, US$/kJ
\( s \)  Specific entropy, kJ/kgK
TCI  Total capital investment, US$
\( T_0 \)  Dead state temperature, °C
\( T_s \)  Apparent black body temperature of the sun, °C
\( W_T \)  Turbine power, kW
\( W_p \)  Power of working fluid pump, kW
\( W_{VC} \)  Power of compressor, kW
\( W_{net} \)  Net power output, kW
\( w_{net} \)  Net specific work, kJ/kg
\( Z_n \)  Capital cost rate of each component, US$/h
\( \Delta t_{lpm} \)  Logarithmic-mean temperature difference, °C
\( \theta \)  Tilt angle, degrees
\( \eta_{I,\text{col}} \)  Collector efficiency, %
\( \eta_{I,\text{ORC}} \)  Thermal (First law) efficiency, %
\( \eta_{I,\text{overall}} \)  Overall system thermal efficiency, %
\( \eta_{II} \)  Exergetic (second law) efficiency, %

Subscripts
\( w \)  Water (R718)
\( hf \)  Heating fluid
\( \text{col} \)  Collector
Abbreviations

CFP  Collector feed water pump
CON-EV  Condenser-evaporator
CON-PH  Condenser organic fluid preheater
EVA  Economic value added
ExV  Expansion valve
FPSC  Flat plate solar collector
HFP  Heating fluid pump
ORC  Organic Rankine cycle
PEC  Purchased equipment cost
RG  Regenerator
TST  Thermal storage tank
VC  Vapour compressor
VG  Vapour generator (boiler)
WFP  Working fluid pump

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