Thermodynamic Evaluation of PTC based Organic Rankine Cycle for Power & Cooling

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Abstract—The paper focuses on thermodynamic evaluation (first and second law) for PTC based organic Rankine cycle with single ejector (EORC) and organic Rankine cycle with double ejector (DEORC) using refrigerant R141b as working fluid and Therminol VP1 as heat transfer fluid. Thermal energy storage tanks are used to store the PTC heat, which provides the continuous net power output and cooling during insufficient solar radiation. The thermodynamic evaluation is carried out to evaluate the performance of PTC field based EORC & DEORC system which produces power and cooling simultaneously. Parametric analyses of EORC and DEORC show that inlet temperature and pressure of turbine at various extraction ratio has the significant effect on first & second law efficiency and cooling to power ratio of this system. It is also found that, annual average energy output of EORC and DEORC is 4710.57 MWh and 5104.97 MWh respectively while annual average exergy output of EORC and DEORC is 3893.86 MWh and 3436.27 MWh respectively.

Keywords—Solar energy; PTC Field; Ejector; First Law Efficiency; Second Law Efficiency; Entrainment Ratio; EORC; DEORC; Cooling/Power Ratio.

I. INTRODUCTION

As the depletion of conventional energy sources is going on and due to the use of fossil fuel for producing the power cause the emissions of green-house gases like CO₂, CO, NO_x etc. which results in global warming. The conventional methods for the production of refrigeration use chlorofluorocarbons (CFCs) and hydro-fluorocarbon (HCFC) refrigerants which causes Ozone depletion and consequently green-house effect.

The renewable energy i.e. solar thermal can be used instead of fossil fuels to reduce the CO₂ emissions. Concentrated solar power (CSP) technologies (solar power tower, paraboloid dish, parabolic trough collector and linear Fresnel reflectors) are used to produce the thermal energy of high temperature and transfer it to the heat transfer fluid [1]. Concentrated solar power plants are gaining increasing interest, by using the parabolic trough collector system due to their advantages in terms of light weight in construction, lower operating cost and higher efficiency. Parabolic trough collector (PTC) technology is the best technology among all CSP technologies for power generation. Reference [2] discussed on exergy analysis and parametric study of PTC on different mass flow rates and concentration ratios for power generation. Their result shows that the exergy output,

exergetic and thermal efficiencies are enhanced as the solar intensity increases. Reference [3] presented an exergy analysis of parabolic trough solar receiver. Their result shows that inlet temperature of heat transfer fluid and solar irradiance has significant effect on the thermal and exergetic performance of the PTC. As inlet temperature of heat transfer fluid increases the exergy efficiency increases but energy efficiency decreases.

Due to the variation in day-night solar irradiation, it is not possible to run the power plant continuously using CSP technology alone. The working of CSP can be enhanced by the incorporation of thermal energy storage (TES) system. Solar thermal energy can be stored during the peak sunshine hours and utilized in non-availability of sunshine hours as well as night cycle for the production of power. Reference [4] demonstrate the feasibility of two tank molten salt storage for parabolic trough solar plants. Their findings show that the use of thermal storage systems improve continuous operation of solar thermal plant and reduces the specific electricity cost. Reference [5] designed a solar thermal power plant with thermal storage for 1 MWe capacity at DNI of 0.6 kW/m². The plant produces about 0.8 MWh of energy output at a capacity factor of 15.6%. Reference [6] analyzed an organic Rankine cycle coupled with PTC using thermal storage has been optimized financially and energetically. Their findings show that the most suitable working fluid is cyclohexane for producing 1 MWe with PTC field.

In order to explore the solar thermal technology efficiently, the combined power and ejector cooling cycle had been proposed by many researchers. Reference [7] developed a computer programme and studied the effect of variation of performance parameters on combined power and cooling cycle. It is shown that ejector refrigeration system is more economical than absorption refrigeration system. A combined power and ejector cooling cycle was proposed by Reference [8] using R245fa as the working fluid. Their results shown that as generator temperature increases from 335 K to 415K, energy efficiency increase from 15.8% to 38.0% and exergy efficiency increase from 45.2% to 57.2% respectively. The generating temperature cannot be increased beyond a limit due to increase in turbine size. Reference [9] developed a computer program for a combined power and ejector cooling cycle using R123 as the working fluid to determine the effects of various operating parameters on the performance of the cycle. Their results show that the first and second law efficiency increase with the increase in evaporator temperature, and maximum exergy losses occurs in the boiler and ejectors. In addition to this, there is increase in first law efficiency and decrease in second law efficiency with increase in turbine inlet pressure. Reference [10] investigated the performance of a combined

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power and ejector refrigeration cycle from thermodynamic point of view and determine the optimum values of the turbine and pump inlet pressures which minimize the total thermal conductance of the cycle for the working fluid considered. Reference [11] analyses a combined power and ejector cooling system using low grade energy as heat source. The system performance were compared for five working fluid (HFE7100, HFE7000, methanol, ethanol, water) at a source temperature of 120°C , evaporator temperature of 10°C and condenser temperature of 35°C . Methanol shown the highest efficiency (19.50%) followed by ethanol and water (17.30%). Their results also shown that the heat source temperature, condenser temperature and evaporator temperature have significant effect on the power output, ejector entrainment ratio and thermal efficiency of the system. An energy and exergy analyses of combined power and ejector refrigeration cycles was reported by [12]-[15] which shows that the maximum irreversibility/exergy loss occurs in heat addition process followed by the ejector and turbine.

In order to utilize the advantages of solar energy, to reduce fossil fuel consumption and alleviating environmental problems, a PTC field based EORC and DEORC is proposed for improving overall energy conversion efficiency. In EORC cooling is produce at single temperature while in DEORC cooling is produce at two different temperatures. The cycle has been analyses from both First and Second law view points and the effect of operating parameters on the performance is observed. The refrigerant R141b is used as working fluid and Therminol VP1 as heat transfer fluid. Thermal storage tanks are also used to store the thermal energy from the Sun which provides the continuous net power output during insufficient solar radiation.

II. SYSTEM DESCRIPTION

Fig. 1a and Fig. 1b illustrates a PTC field based EORC and DEORC respectively. Fig. 2a and Fig. 2b illustrates the T-s diagram of EORC and DEORC respectively.

Fig. 1a Schematic diagram of Organic Rankine cycle with single ejector (EORC)

Fig. 1b Schematic diagram of Organic Rankine cycle with double ejector (DEORC)

Fig. 2a T-s diagram of Organic Rankine cycle with single ejector (EORC)

The DEORC consists of parabolic trough collector (PTC) solar field, high temperature vessel, low temperature vessel, pumps, heat recovery vapor generator, turbine, ejector, condenser, throttle valve, evaporator and heat exchanger. Solar energy is used to heat the heat transfer fluid Therminol VP1 (1) with the help of PTC field. Heat transfer fluid (HTF) enters the heat recovery vapor generator (G_1) which is used to superheat the high pressure refrigerant and leaves at (2), which again enters into the heat recovery vapor generator (G_2) to vaporize the refrigerant and return back to the PTC solar field. Superheated refrigerant vapor (4) expands in the turbine. After expansion up to the extraction pressure, refrigerant vapors are extracted (5) from turbine at extraction pressure and then it flows into the ejector nozzle, entrains secondary vapor (13) from the evaporator (E_1) and mixes in mixing chamber of the ejector (EJ_1). The ejector exit stream (6) mixes with the turbine exhaust (14) and is cooled (7-8) in the heat exchanger (HE) by rejecting heat to the refrigerant leaving from the pump (P_1) which is preheated (15-3). The refrigerant leaving from heat exchanger (8) is condensed in the condenser (C_1) and converts into saturated liquid (9). The saturated liquid (9) from condenser is divided into two parts. One part (11) enters into throttle valve (TV_1) and other part (10) into pump (P_1). The saturated liquid (10) pumped with the help of pump (P_1) into the heat exchanger (15). The preheated refrigerant (3) leaving from heat exchanger enters into heat recovery vapor generator (G_1) in which heat is absorbed and convert into superheated vapor (4). The saturated liquid (11) expands in the throttle valve (TV_1) to the evaporator pressure (12). The refrigerant (12) enters into the evaporator.

(E₁) and produce the cooling effect by evaporation of refrigerant (12-13). The refrigerant vapour (16) flows into the ejector nozzle and entrains secondary vapor (22) from the evaporator (E₂) mixes in mixing chamber of the ejector. The ejector (EJ₂) exit stream (17) is condensed in the condenser (C₂) and converts into saturated liquid (18). Saturated liquid (18) is divided into two parts. One part (20) enters into throttle valve (TV₂) and other part (19) into pump (P₂). The saturated liquid (19) pumped with the help of pump (P₂) into the heat recovery vapor generator (G₂) and converts into saturated vapor (16). The saturated liquid (20) expands in the throttle valve (TV₂) to the evaporator pressure (21). The refrigerant (21) enters into the evaporator (E₂) and produce the cooling effect by evaporation of refrigerant (21-22). The system description of organic Rankine cycle with single ejector (EORC) is same as DEORC.

Fig. 2b T-s diagram of Organic Rankine cycle with double ejector (DEORC)

For the thermodynamic analysis, the parameters considered for the operation of EORC and DEORC are depicted in “Table I”.

TABLE I: MAIN PARAMETERS CONSIDERED FOR THE ANALYSIS

Atmospheric Temperature (°C)	25
Atmospheric pressure (MPa)	0.10135
Turbine inlet Temperature (°C)	150
Turbine inlet pressure (MPa)	0.6
Turbine extraction pressure (MPa)	0.24
Extraction ratio	0.3
Turbine isentropic efficiency (%)	85
Pump isentropic efficiency (%)	70
Condenser temperature (°C)	30
Evaporator (E ₁) temperature (°C)	-5
Evaporator (E ₂) temperature (°C)	0
Effectiveness of the heat exchanger	0.75
Aperture area (m ²)	10000
Direct normal irradiance (kWm ⁻²)	0.60
HTF temperature inlet to HRVG (°C)	160
HTF temperature exit to HRVG (°C)	90
HRVG efficiency (%)	100
Nozzle efficiency (%)	90
Mixing chamber efficiency (%)	85
Diffuser efficiency (%)	85

III. THERMODYNAMIC ANALYSIS

Based on mass, momentum and energy equations the entrainment ratio is developed by [13].

$$\mu = \sqrt{\eta_n \eta_m \eta_d (h_{pf,n1} - h_{pf,2s}) / (h_{mf,ds} - h_{mf,m})} - 1 \quad (1)$$

The enthalpy and entropy values for R141b at salient state points of the EORC and DEORC are taken from NIST Standard Reference Database 23, REFPROP 6.01(1998) [16].

A. First law analysis for DEORC

Solar energy received from the Sun

$$\dot{Q}_{\text{Solar}} = I A_p \quad (2)$$

where I = Direct normal irradiance (kWm⁻²), A_p = Aperture Area (m²)

Heat gain in the PTC field for DEORC

$$\begin{aligned} \dot{Q}_{\text{gain}} &= \dot{m}(h_1 - h_{2a}) = \eta_{l, \text{PTC}} I A_p \\ &= \dot{m}_4(h_4 - h_3) + \dot{m}_{16}(h_{16} - h_{23}) \end{aligned} \quad (3)$$

where $\eta_{l, \text{PTC}}$ = First law efficiency of PTC field

$$\eta_{l, \text{PTC}} = a - b \left[\frac{T_m - T_0}{I} \right] - c \left[\frac{(T_m - T_0)^2}{I} \right] \quad (4)$$

a = optical efficiency = 0.7, b = first order loss coefficient = 0.1, c = second order loss coefficient = 0, T_m = mean temperature = (T₁ + T_{2a})/2.

The first law efficiency (η_l) of the DEORC can be defined as the ratio of the net power output ($\dot{W}_{\text{net}}$) and cooling output in both the evaporators ($\dot{Q}_{e1}$ & $\dot{Q}_{e2}$) to the solar energy input ($\dot{Q}_{\text{Solar}}$).

$$\eta_l = \text{Energy output} / \text{Solar energy input} = (\dot{W}_{\text{net}} + \dot{Q}_{e1} + \dot{Q}_{e2}) / \dot{Q}_{\text{Solar}} \quad (5)$$

Turbine work:

$$\dot{W}_T = \dot{m}_4(h_4 - h_5) + \dot{m}_4(1 - E_r)(h_5 - h_{14}) \quad (6)$$

$$\text{where extraction ratio: } E_r = \dot{m}_5 / \dot{m}_4 \quad (7)$$

$$\text{Pump work: } \dot{W}_P = \dot{m}_4(h_{15} - h_{10}) \quad (8)$$

$$\text{Net power output: } \dot{W}_{\text{net}} = \dot{W}_T - \dot{W}_P \quad (9)$$

Cooling output in evaporator (E₁)

$$\dot{Q}_{e1} = \dot{m}_5 \mu_1 (h_{13} - h_{12}) \quad (10)$$

Cooling output in evaporator (E₂)

$$\dot{Q}_{e2} = \dot{m}_{16} \mu_2 (h_{22} - h_{21}) \quad (11)$$

Entrainment ratio:

$$\text{Ejector (EJ}_1\text{)} - \mu_1 = \dot{m}_{13} / \dot{m}_5 \quad (12)$$

$$\text{Ejector (EJ}_2\text{)} - \mu_2 = \dot{m}_{22} / \dot{m}_{16} \quad (13)$$

B. Second law analysis for DEORC

Exergy analysis is based on second law approach. Exergy may be defined as the maximum possible reversible work obtainable in bringing the state of the system to the dead state, Mathematically

$$\dot{E} = \dot{m}[(h - h_0) - T_0(s - s_0)] \quad (14)$$

Second law efficiency of PTC field may be reported as

$$\eta_{II, PTC} = \{\dot{m}(h_1 - h_{2a}) - T_0(s_1 - s_{2a})\} / IA_p \psi \quad (15)$$

Maximum useful work obtainable from sun radiation (ψ) is calculated from [17] formula that is given as

$$\psi = 1 - \frac{4}{3} \frac{T_0}{T_{\text{solar}}} + \frac{1}{3} \left(\frac{T_0}{T_{\text{solar}}} \right)^4 \quad (16)$$

The second law efficiency (η_{II}) of DEORC may be reported as

$$\eta_{II} = \text{Exergy output} / \text{Exergy input} \\ = (\dot{W}_{\text{net}} + \dot{E}_{e_1} + \dot{E}_{e_2}) / \dot{E}_{\text{solar}} \quad (17)$$

where, $\dot{E}_{\text{solar}}$ is exergy input associate with solar radiation falling on PTC field, $\dot{E}_{e_1}$ is the exergy associate with cooling output in the evaporator (E_1) and $\dot{E}_{e_2}$ is the exergy associate with cooling output in the evaporator (E_2),

$$\dot{E}_{e_1} = \dot{m}_{13}[(h_{12} - h_{13}) - T_0(s_{12} - s_{13})] \quad (18)$$

$$\dot{E}_{e_2} = \dot{m}_{22}[(h_{21} - h_{22}) - T_0(s_{21} - s_{22})] \quad (19)$$

$$\dot{E}_{\text{solar}} = IA_p \psi \quad (20)$$

where T_{solar} is the Apparent Sun Temperature = 4500K

IV. RESULTS AND DISCUSSION

The first and second law analyses is conducted for EORC and DEORC, their results are compared by varying operating parameters such as turbine inlet temperature (TIT) and turbine inlet pressure (TIP) at different extraction ratio on the performance of the cycle.

Fig. 3 Monthly energy and exergy output of EORC

Fig. 4 Monthly energy and exergy output of DEORC

Fig. 3 and Fig. 4 show the monthly energy and exergy output of EORC and DEORC. It is observed that due to higher DNI in the month of March, April, May and October the energy and exergy output is higher while in the remaining months the energy and exergy output obtained is lower due to low value of DNI. The highest energy output (715.53 MWh) and exergy output (593.8 MWh) obtained in the month of March while lowest energy output (214.74 MWh) and exergy output (176.57 MWh) in the month of January in EORC. The maximum energy output in the month of March is 749.95 MWh and maximum exergy output 544.38 MWh while least energy output 244.07 MWh and exergy output is 142.82 MWh registered in the month of January in DEORC. The annual average energy output of EORC and DEORC is 4710.57 MWh and 5104.97 MWh respectively while annual average exergy output of EORC and DEORC is 3893.86 MWh and 3436.27 MWh respectively. It is concluded from Fig. 3 and Fig. 4 that yearly average energy output increases while average exergy output decreases with the addition of ejector refrigeration system in the EORC.

Fig. 5 shows the energy input/output and exergy input/output in EORC and DEORC at input thermodynamic parameters as shown in Table I. In EORC the total useful energy output of the cycle ($\dot{W}_{\text{net}} + \dot{Q}_{e_1}$) is 686 kW and the exergy output ($\dot{W}_{\text{net}} + \dot{E}_{e_1}$) is 571.22 kW is observed. In DEORC the total energy output ($\dot{W}_{\text{net}} + \dot{Q}_{e_1} + \dot{Q}_{e_2}$) is 711.2 kW and the exergy output ($\dot{W}_{\text{net}} + \dot{E}_{e_1} + \dot{E}_{e_2}$) is 535.26 is observed. It is clear that energy output in DEORC is higher than EORC.

Fig. 5. Energy input/output, Exergy input/output of EORC and DEORC

Fig. 6, 7, and 8 shows the effect of turbine inlet temperature (TIT) on first law efficiency, second law

efficiency and cooling/power ratio at different extraction ratio in EORC and DEORC. As the turbine inlet temperature increases the net power output increases due to increase in enthalpy drop in the turbine and cooling output increases due to increase in the enthalpy of R141b at the inlet of the ejector. This causes the increase in velocity of stream at the exit of the nozzle ejector, results in increase in entrainment ratio hence cooling output in EORC and DEORC increases. The rate of increase of net power output is higher than rate of increase of cooling output, the cooling to power ratio decreases. Due to increase in power output and cooling output the first law efficiency increases with increase in turbine inlet temperature. The second law efficiency increases with increase in turbine inlet temperature due to increase in total exergy output. The result shows that the first law efficiency of DEORC is greater than first law efficiency of EORC. With the increase in extraction ratio at the same turbine inlet temperature, the net power output decreases and cooling output increases. Since the rate of increase in cooling output is higher than the rate of decrease in net power output. The cooling to power ratio and first law efficiency increases while second law efficiency decreases as the extraction ratio increases. Fig. 6, 7, and 8 indicate that first law efficiency and cooling to power ratio is higher and second law of efficiency is lower for DEORC than EORC at the same working conditions.

Fig. 6. Variation of first law efficiency with TIT at different extraction ratio

Fig. 7. Variation of second law efficiency with TIT at different extraction ratio

Fig. 8. Variation of Cooling/Power ratio with TIT at different extraction ratio

Fig. 9. Variation of First law efficiency with TIP at different extraction ratio

Fig. 10. Variation of Second law efficiency with TIP at different extraction ratio

Fig. 9, 10, and 11 show the effect of variation of turbine inlet pressure (TIP) on first law efficiency, second law efficiency and cooling to power ratio at different extraction ratio. As the turbine inlet pressure increases, the net power output increases and cooling output decreases for EORC and DEORC. The rate of increase in net power output is greater than rate of decrease in cooling output results in increase in first law efficiency and decrease in cooling to power ratio for EORC and DEORC with increase in turbine inlet pressure. The same trend for exergy efficiency observed with increase in turbine inlet pressure as shown in Fig. 10. With the increase in extraction ratio at the same turbine inlet pressure, the net power output decreases and cooling output increases. Since the rate of increase in cooling output is higher than the rate of decrease in net power output. The cooling to power ratio and first law efficiency increases while second law efficiency decreases as the extraction ratio increases. Fig. 9, 10, and 11 indicate that first law efficiency and cooling to power ratio is higher and second law efficiency is lower for DEORC than EORC at the same working conditions.

Fig. 11. Variation of Cooling/Power ratio with TIP at different extraction ratio

Table II shows the performance of the EORC and DEORC. It is observed that the cooling and power output in DEORC is 711.23 kW as compared to 686 kW of EORC. The total energy output in DEORC is 11.85% of solar energy input (100%) is observed.

TABLE II: THE PERFORMANCE OF THE EORC AND DEORC.

	Organic Rankine cycle with single ejector (EORC)		Organic Rankine cycle with double ejector (DEORC)	
	Amount	% of solar energy input	Amount	% of solar energy input
Solar energy input (kW)	6000.00	100.00	6000.00	100.00
Turbine work (kW)	572.50	9.54	531.90	8.85
Total pump work (kW)	9.55	0.159	9.37	0.15
Total cooling output (kW)	123.10	2.05	188.70	3.15

Net power output (kW)	562.90	9.38	522.53	8.70
Energy output (kW)	686.00	11.43	711.23	11.85
Energy loss (kW)	5314.00	88.57	5288.77	88.15

Table III shows the distribution of exergy in EORC and DEORC. The total exergy output in DEORC is 535.26 kW as compared to exergy output of 571.22 kW in EORC.

TABLE III: THE DISTRIBUTION OF EXERGY IN EORC AND DEORC.

	EORC		DEORC	
	Amount	% of exergy	Amount	% of exergy
Exergy input (kW)	5470.00	100.00	5470.00	100.00
Exergy output (kW)	571.22	10.44	535.26	9.785
Exergy destruction/losses (kW):				
(i) Generator	218.50	3.99	212.84	3.89
(ii) Turbine	81.46	1.49	75.68	1.38
(iii) Ejector	87.23	1.59	110.19	2.02
(iv) Condenser	60.44	1.10	63.75	1.165
(v) Pump	9.36	0.17	9.19	0.168
(vi) Heat exchanger	10.77	0.20	10.00	0.183
(vii) Throttle valve	1.65	0.03	2.24	0.041
(viii) Evaporator	4.85	0.09	5.92	0.108
(ix) Parabolic trough collector	4424.52	80.90	4444.93	81.26
Total Exergy destruction/loss (kW)	4898.78	89.56	4934.74	90.215

V. CONCLUSION

This paper presents thermodynamic analysis of PTC based EORC & DEORC using R141b as working fluid and Therminol VP1 as heat transfer fluid. The following main conclusions in this study are as follows:

- With the increase in inlet temperature of turbine (110°C - 160°C), the first and second law efficiency of EORC and DEORC increases while cooling /power ratio decreases.
- With the increase in the inlet pressure of turbine (600 kPa - 1100 kPa), the first and second law efficiency of EORC and DEORC increases whereas cooling /power ratio decreases.
- The first law efficiency and cooling to power ratio is higher in DEORC as compared to EORC at the same working condition.
- In EORC the maximum energy output (715.53 MWh) and the maximum exergy output (593.8 MWh) obtained in the month of March while minimum energy output (214.74 MWh) and minimum exergy output (176.57 MWh) in the month of January. The annual average energy output of EORC is 4710.57 MWh while annual average exergy output of EORC 3436.27 MWh.
- In DEORC the highest energy output (749.95 MWh) and exergy output (544.38 MWh) is registered in the month of March while least energy output (244.07 MWh) and exergy output (142.82 MWh) registered in the month of January respectively. The annual average energy output of DEORC is 5104.97 MWh while annual average exergy output of DEORC is 3436.27 MWh respectively.
- With the addition of ejector in EORC the first law efficiency increases from 11.43% to 11.85% while second law efficiency decreases from 10.44% to

9.785% at input thermodynamic parameters as shown in Table I.

APPENDIX

Nomenclature

A_p	Aperture area (m ²)
I	Direct normal irradiance (kWm ⁻²)
$\dot{E}$	Exergy rate (kW)
$\dot{Q}$	Heat transfer rate (kW)
T	Absolute temperature (K), Turbine
P	Pressure (MPa), Pump
$\dot{W}$	Power output/input (kW)
h	Specific enthalpy (kJ kg ⁻¹)
$\dot{m}$	Mass flow rate (kg s ⁻¹)
s	Specific entropy (kJ kg ⁻¹ K ⁻¹)
G	Heat recovery vapor generator (HRVG)
EJ	Ejector
C	Condenser
HE	Heat exchanger
TV	Throttle valve
E	Evaporator
Er	Extraction ratio
HTF	Heat transfer fluid (Therminol VP1)
$EORC$	Organic Rankine cycle with single ejector
$DEORC$	Organic Rankine cycle with double ejector
PTC	Parabolic trough collector

Greek symbols

μ	Entrainment ratio
ε	Cooling to power ratio
η	Efficiency (%)

Subscript

e_1	Evaporator 1
e_2	Evaporator 2
p	Pump
T	Turbine
1, 2, 3,.....a, b, c,.....	state points in Figures 1a, 1b, 2a, 2b.

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