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Thermal Economic Analysis of Thermoelectric Generation System with Different Cycle Flow

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Abstract

The thermoelectric generation cycle system is widely used, and different cycle processes have their own advantages. This study utilized simulation method, through the modeling of Rankine cycle, Kalina cycle and Uehara cycle, the differences in thermal efficiency (η) and leveled cost of energy (LCOE) between different cycle processes were analyzed under different heat source temperature conditions. Meanwhile, the effects of evaporation temperature and turbine inlet pressure on thermal-economic performance in different cycle systems were also explored. The results show that Kalina cycle has high thermal efficiency in each temperature condition of heat source, but its economic performance is poor. In low temperature heat source, the LCOE of Rankine cycle with pure ammonia work flow is the lowest under the same conditions. Rankine cycle should be set in saturated steam state, and designers should as far as possible to increase the turbine inlet pressure and reduce the evaporation superheat. Instead, in high temperature heat source, the LCOE of Uehara cycle is the lowest. Moreover, there is a deviation of the turbine inlet pressure corresponding to the minimum LCOE and the maximum thermal efficiency of Kalina cycle and Uehara cycle. Designed according to the minimum LCOE principle at this point, more power generation and less equipment cost can be obtained. Therefore, thermal-economic performance of different cycles under different heat source temperature conditions should be considered in cycle flow design.

Key words: Thermoelectricity, Economic analysis, Rankine Cycle, Kalina Cycle, Uehara Cycle

1. Introduction

Thermoelectric generation is widely used in the utilization of geothermal energy, industrial waste heat, solar energy, Ocean Thermal Energy Conversion (OTEC) and other thermal resources. These resources have the characteristics of green, clean and renewable. It abounds with thermal energy resources in China, and the temperature range of each heat source is different.

Geothermal energy in China, mainly distributed in the southwest and southeast coastal areas. The total potential of geothermal power generation is 9960 MW. The high temperature geothermal energy above 150℃ is mainly used for power generation, and the medium temperature of 90~150℃ and the low temperature of 25~90℃ are mainly used in heating directly (Wang, et al., 2017).

Waste heat resource account for 17%~67% of total industrial raw material consumption, and the recovery rate can up to 60%. Waste heat resource is divided by temperature into high temperature waste heat (> 600℃), medium temperature waste heat (300~600℃) and low temperature waste heat (<300℃) (Lian, et al., 2011).

Solar thermal energy mainly used in solar hot water. The sales volume of Chinese solar heat collection system is 2.737 million m^2 (18926 MW) in 2020. Solar thermal energy can be divided into high-temperature photothermal energy (>300℃), medium-temperature photothermal energy (100~300℃) and low-temperature photothermal energy (< 100℃) according to the heating temperature level.

Ocean thermal energy conversion. China 's ocean thermal energy resources are mainly distributed in the South China Sea and the East China Sea. The potential installed capacity in the South China Sea is 330.7GW (Shi, et al., 2011). Warm sea water temperature is about 25~30℃, and it can provide about 20℃ temperature difference.

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Increasing the development of thermal energy resources can effectively reduce China's dependence on fossil energy, which is one of the important means to achieve the goal of carbon neutrality. Therefore, thermoelectric power generation technology has received more and more attention, and the performance optimization of thermoelectric power generation cycle system has become a research hotspot of scholars. Rankine Cycle is the most mature technology in thermoelectric generation system. Zhang Kai, 2019 analyzed the thermal economy efficiency of organic Rankine cycle with different non-azeotropic refrigerants, and proposed a selection criterion for mixed refrigerants based on the temperature matching optimization of heat sources in open cycle system. Wang Meng et al., 2018 conducted a multi-objective optimization of OTEC system based on the organic Rankine cycle. Through comparing the results of the six work flows, the R717 work flow was selected to be the best in terms of exergy efficiency and leveled cost of energy (LCOE), which value were 28.17% and 0.341 $/kWh, respectively. Kalina Cycle improves the cycle efficiency by 10%~20% compared with Rankine Cycle under the same conditions. Wang Jiangfeng, et al., 2013 optimized the inlet pressure, inlet temperature and ammonia concentration of Kalina cycle. The optimization results show that under the solar heat source of 122.48℃, the optimal cycle thermal efficiency of Kalina cycle can up to 8.54%. Wu Shuangying, et al., 2017 studied the Kalina cycle under 423.15K (205.85℃) flue gas waste heat power generation. The results show that the net output power of the cycle will reach a maximum value with the change of evaporation pressure, and decrease linearly with the increase of flue gas outlet temperature. At the same time, there is a minimum value of LCOE in the system. The Uehara Cycle was improved on the basis of the Kalina Cycle. Based on the experimental data of the Uehara cycle power generation system on the actual ocean temperature difference, Yoshitaka Matsuda, et al., 2018 used the least square method to derive the expressions of the system power generation on the hot side and the cold side with respect to pressure and work flow flow. Chen Fengyun, 2016 found that the optimal work flow concentration and turbine inlet pressure exist in the Uehara cycle under the specified temperature of cold and heat sources, and the increase of vapor extraction rate within the calculation range will increase the thermal efficiency and reduce the turbine output power.

At present, researchers at home and abroad mostly focus on the analysis and optimization of the cycle efficiency and economic benefits of a single cycle process. The research of the Uehara cycle is only limited to OTEC, and its economic benefits are rarely studied. At the same time, there is a lack of comparison of thermal and economic performance between different cycles. Therefore, in this paper, three heat source temperatures (28℃, 122℃ and 255℃) are used to cover most low and medium temperature thermal power generation scenarios, including ocean thermal energy. The thermal economic

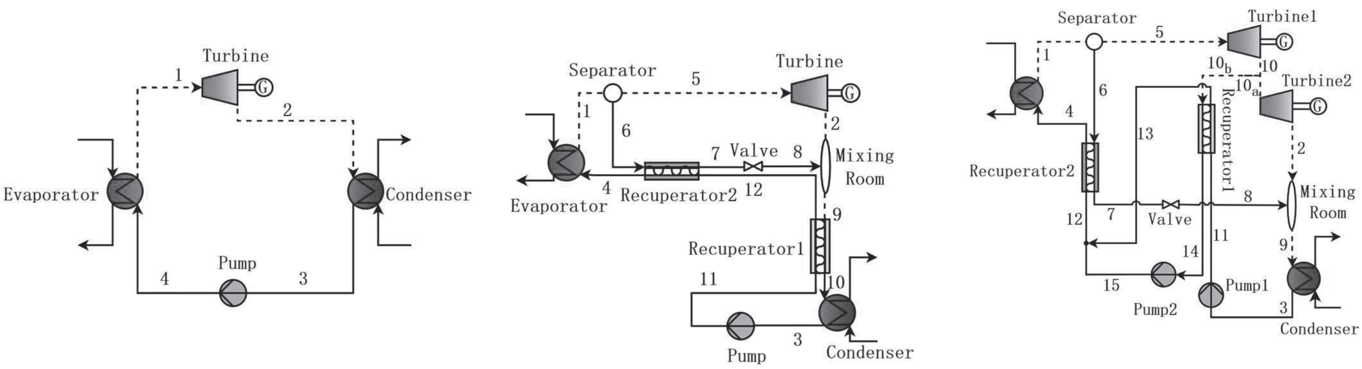
Nomenclature		***Greek symbols***	
		ε	isentropic efficiency (%)
A	area (m^2)	η	thermal efficiency (%)
B	calculation coefficients	ρ	Ammonia concentration
C	cost ($)	ϕ	maintenance factor (%)
CRF	capital recovery factor	ψ	mechanical efficiency (%)
F	modify factory	***Subscripts***	
h	enthalpy (kJ/kg)	all	all devices
i	interest rate (%)	BM	bare module
K	cost calculation factor	cn	condenser
$LCOE$	Leveled cost of electricity ($/kWh)	ep	equipment
$LMTD$	logarithmic mean temperature difference (℃)	ev	evaporator
m	mass flow rare (kg/s)	KC	Kalina Cycle
N	life time (year)	M	materials
n	annual operation time (h)	max	maximum
P	pressure (bar)	min	minimum
Q	heat flow (kW)	net	net work
T	temperature	p	pump
ΔT	temperature difference (℃)	RC	Rankine Cycle
U	heat transfer coefficient (kW/m^{2}K)	rc	recuprator
VF	vapor fraction	t	turbine
W	power (kW)	UC	Uehara Cycle

performance of Rankine cycle, Kalina cycle and the Uehara cycle is compared and analyzed, and the influence of evaporation temperature and turbine inlet pressure on the thermal efficiency and economic performance of the cycle is explored, which provides reference for the design and operation of the thermal power generation cycle system under corresponding working conditions.

2. Cycle flow and mathematics model

2.1 Cycle flow

Figure 1 is the schematic diagram of Rankine cycle, Kalina cycle and Uehara cycle from left to right. In the Rankine cycle, the liquid work flow absorbs the heat of the heat source in the evaporator and then becomes saturated vapor. The saturated vapor expands in the turbine and outputs power. The turbine drives the generator to rotate and generate electricity. The expanded exhaust vapor enters the condenser and is cooled by the cold source to liquid. The liquid work flow is pressurized by the work flow pump and returns to the evaporator to complete the cycle. The Kalina cycle uses non-azeotropic refrigerants (ammonia-water), adding gas-liquid separation process and two recuperation processes to the Rankine cycle. The non-azeotropic work flow (ammonia-water) is heated in the evaporator to form a gas-liquid mixing state, and the vapor work flow separated in the separator enters the turbine expansion to work, driving the generator to generate electricity. The liquid work flow separated in the separator is mixed with the exhaust steam in the turbine after passing through the recuperator 2. The mixed work flow is heated to the liquid work flow after condensation and pressure in recuperator 1. The liquid work flow after heating in recuperator 1 will be heated for the second time by the liquid work flow separated from the separator in recuperator 2, and finally returned to the evaporator to complete the cycle. The Uehara cycle was improved on the basis of Kalina cycle, so non-azeotropic refrigerant (ammonia-water) was also used. Different from the Kalina cycle, the Uehara cycle adopted two turbines, and the extraction point (state point 10) was set in the turbine room. A small part of the exhaust steam was extracted and then it heated for the liquid work flow that was after condensation and pressurization the first time in the recuperator 1, while this small part of the exhaust steam was condensed in the recuperator 1 and mixed with the liquid work flow after pressurization by the work flow pump 2. Other processes of the work flow are consistent with the Kalina cycle.



a. Rankine Cycle b. Kalina Cycle c. Uehara Cycle

Fig.1 Schematic diagrams of the thermal energy conversion systems

2.2 Model assumptions and initial conditions

To reduce the complexity of modeling, the following assumptions are made for the three loop processes,

(1) Ignoring the pressure and heat losses caused by pipelines, separators, absorbers and other components.

(2) the work flow is isentropic process in turbine and pump.

(3) the work flow is saturated liquid after passing through the condenser.

according to References (Bernardoni, et al., 2019, Ogrisick, 2009 and Köse, et al., 2021), three sets of initial data were selected according to different heat source temperatures, as shown in Table 1.

Table 1 Initial parameters of model

Parameters	Ref. Bernardoni	Ref. Ogrisick	Ref. Köse
$T_{hotsource}$ / ℃	28	122	255
$T_{coldsource}$ / ℃	4	5	20
m /kg · s^{-1}	92.9	16.8	2.701
work flow	Ammonia	Ammonia-water	Ammonia-water
ρ	1.00	0.82	0.90
ε_t /%	89	87	82
ψ_t /%	-	98	-
ε_p /%	80	98	80

2.3 Mathematical model

2.3.1 Thermodynamic control equation

The three cycles are mainly composed of key components such as heat exchanger, turbine and work flow pump. The heat exchanger consists of evaporator, condenser and recuperator.

(1) Rankine Cycle

The heat absorbed by work flow in evaporator is

$$Q_{ev}^{RC}=m^{RC}\left(h_1^{RC}-h_4^{RC}\right) \tag{1}$$

The heat released by the work flow in the condenser is

$$Q_{cn}^{RC}=m^{RC}\left(h_2^{RC}-h_3^{RC}\right) \tag{2}$$

The turbine output power is

$$W_t^{RC}=m^{RC}\left(h_1^{RC}-h_2^{RC}\right) \tag{3}$$

Energy consumption of work flow pump is

$$W_p^{RC}=m^{RC}\left(h_4^{RC}-h_3^{RC}\right) \tag{4}$$

The net output power of the system is

$$W_{net}^{RC}=W_t^{RC}-W_p^{RC} \tag{5}$$

The system thermal efficiency is

$$\eta^{RC}=\frac{W_{net}^{RC}}{Q_{ev}^{RC}} \tag{6}$$

(2) Kalina Cycle

The heat absorbed by work flow in evaporator is

$$Q_{ev}^{KC}=m_{ev}^{KC}\left(h_1^{KC}-h_4^{KC}\right) \tag{7}$$

The heat released by the work flow in the condenser is

$$Q_{cn}^{KC}=m_{cn}^{KC}\left(h_{10}^{KC}-h_3^{KC}\right) \tag{8}$$

Heat exchange of recuperator 1 is

$$Q_{rc1}^{KC}=m_9^{KC}\left(h_9^{KC}-h_{10}^{KC}\right)=m_{11}^{KC}\left(h_{12}^{KC}-h_{11}^{KC}\right) \tag{9}$$

Heat exchange of recuperator 2 is

$$Q_{rc2}^{KC}=m_6^{KC}\left(h_6^{KC}-h_7^{KC}\right)=m_{12}^{KC}\left(h_4^{KC}-h_{12}^{KC}\right) \tag{10}$$

The turbine output power is

$$W_t^{KC}=m_t^{KC}\left(h_5^{KC}-h_6^{KC}\right) \tag{11}$$

Energy consumption of work flow pump is

$$W_p^{KC}=m_p^{KC}\left(h_{11}^{KC}-h_3^{KC}\right) \tag{12}$$

The net output power of the system is

$$W_{net}^{KC}=W_t^{KC}-W_p^{KC} \tag{13}$$

The system thermal efficiency is

$$\eta^{KC}=\frac{W_{net}^{KC}}{Q_{ev}^{KC}} \tag{14}$$

(3) Uehara Cycle

The heat absorbed by work flow in evaporator is

$$Q_{ev}^{UC}=m_{ev}^{UC}\left(h_1^{UC}-h_4^{UC}\right) \tag{15}$$

The heat released by the work flow in the condenser is

$$Q_{cn}^{UC}=m_{cn}^{UC}\left(h_9^{UC}-h_3^{UC}\right) \tag{16}$$

Heat exchange of recuperator 1 is

$$Q_{rc1}^{UC}=m_{11}^{UC}\left(h_{13}^{UC}-h_{11}^{UC}\right)=m_{10b}^{UC}\left(h_{10b}^{UC}-h_{14}^{UC}\right) \tag{17}$$

Heat exchange of recuperator 2 is

$$Q_{rc2}^{UC}=m_6^{UC}\left(h_6^{UC}-h_7^{UC}\right)=m_{12}^{UC}\left(h_4^{UC}-h_{12}^{UC}\right) \tag{18}$$

The turbine output power is

$$W_t^{UC}=m_{t1}^{UC}\left(h_5^{UC}-h_{10}^{UC}\right)+m_{t2}^{UC}\left(h_{10a}^{UC}-h_2^{UC}\right) \tag{19}$$

Energy consumption of work flow pump is

$$W_p^{UC}=m_{p1}^{UC}\left(h_{11}^{UC}-h_3^{UC}\right)+m_{p2}^{UC}\left(h_{15}^{UC}-h_{14}^{UC}\right) \tag{20}$$

The net output power of the system is

$$W_{net}^{UC}=W_t^{UC}-W_p^{UC} \tag{21}$$

The system thermal efficiency is

$$\eta^{UC}=\frac{W_{net}^{UC}}{Q_{ev}^{UC}} \tag{22}$$

2.3.2 Economic model

Since the costs of work flows, pipelines, separators, absorbers and expansion valves are far less than those of other equipment, they are not included in cost calculations. Circulating system cost calculation mainly includes heat exchanger, turbine and pump. According to the method proposed by Richard, 2018, the investment cost calculation formula is as follows:

$$log_{10} C_{ep} = K_1+K_2\, log_{10}(X) + K_3\left[log_{10}(X)\right]^2 \tag{23}$$

X is the equipment cost measurement parameter. The heat exchanger uses the heat exchange area A, m^2 as the measurement parameter, while the pump and turbine use the work W_t, W_p, kW as the measurement parameters.

The heat exchange area of the heat exchanger is calculated as follows:

$$A^*=\frac{Q^*}{U^* LMTD^*} \tag{24}$$

$LMTD$ Calculation of logarithmic mean temperature difference in heat exchanger

$$LMTD^*=\frac{\Delta T_{max}^*-\Delta T_{min}^*}{ln\frac{\Delta T_{max}^*}{\Delta T_{min}^*}} \tag{25}$$

The superscript * is the abbreviation of RC, KC and UC. In the simplified calculation, the heat exchangers in the cycle use a unified heat exchanger without considering the influence of the fluid state. The total heat transfer coefficient U^* is taken as a constant value 3000 $W \cdot m^{-2} \cdot K^{-1}$ in Reference (Shun, 2020).

For equipment with different materials and pressure environments, correction coefficients need to be added. The revised investment cost is

$$C_{BM}=C_{ep}F_{BM}=C_{ep}(B_1+B_2F_MF_P) \tag{26}$$

Where F_{BM} is the total correction coefficient, F_M is the material correction coefficient, B_1 and B_2 are the calculation coefficients, and F_P is the pressure correction coefficient. The calculation formula is

$$log_{10} F_P = C_1+C_2\, log_{10}(P) + C_3\left[log_{10}(P)\right]^2 \tag{27}$$

Total investment cost converted to 2016 price is

$$C_{all}=\frac{542}{397}\left(C_{BM}^{ev}+C_{BM}^{cn}+C_{BM}^{rc}+C_{BM}^{t}+C_{BM}^{p}\right) \tag{28}$$

The correlation coefficient of cost calculation is listed in table 2.

2.3.3 Economic indicators

In this paper, the thermal efficiency η is selected as the power generation efficiency index of the circulation system, and the leveled cost of electricity (LCOE) is used as the economic index of the system. The interest rate i of RMB commercial loan above the five-year period is 4.65%, the operation period N is 20 years, the operation and maintenance cost Φ is 6% of the total investment cost, and the annual operation time n is 7800 h.

After calculating investment costs, the capital recovery cost (CRF) could be found by

$$CRF=\frac{i(1+i)^N}{(1+i)^N-1} \tag{29}$$

Leveled cost of electricity (LCOE) is

$$LCOE=\frac{C_{all}\times CRF+\phi}{W_{net}\cdot n} \tag{30}$$

Table 2 Parameters of cost calculation

Parameters	Heat Exchanger	Turbine	Pump
K_1	4.6656	2.7051	3.8696
K_2	-0.1557	1.4398	0.3161
K_3	0.1547	-0.1776	0.1220
B_1	0.96	-	1.89
B_2	1.21	-	1.35
C_1	0	-	-0.2454
C_1	0	-	0.2590
C_1	0	-	-0.0136
F_M	1	-	1.45
F_{BM}	2.17	3.5	-

2.4 Model verification

Aspen plus software was used to model the three circulation processes In this paper. PENG-ROB was selected as the physical property method of thermoelectric generation cycle. The work flow was pure ammonia in Rankine cycle, and ammonia was used in Kalina cycle and the Uehara cycle.

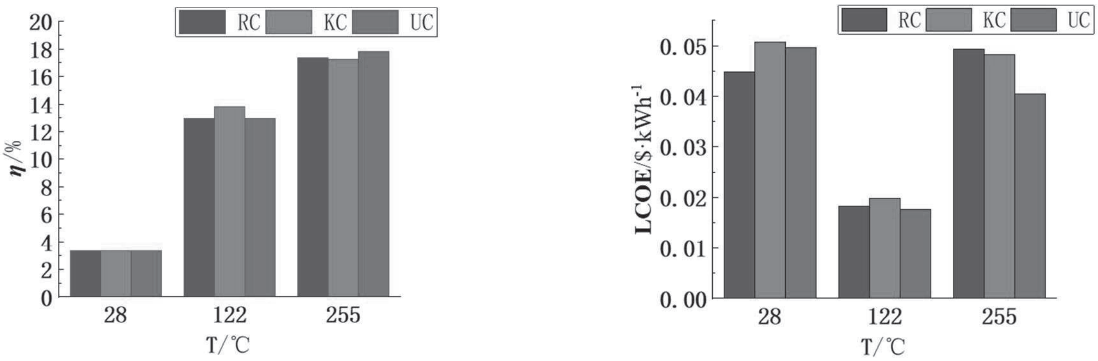
Substituting the data of References (Bernardoni, et al., 2019, Ogrisick, 2009 and Goto, et al., 2011) into the model, the simulation results and the data of each key state point in the literature are listed in Table 3. By comparison, it is found that the simulation results are basically consistent with the parameter values of each key state point in the literature, and the error is within a reasonable range, which proves the accuracy of the model calculation.

Table 3 Parameters of key state point

	State Point	T/ °C		VF		m/kg · s^{-1}	
		Result	Ref.	Result	Ref.	Result	Ref.
Rankine Cycle	1	22.33	22.05	1	1	92.90	92.90
	2	10.39	10.11	0.976	0.972	92.90	92.90
	3	10.39	10.11	0	0	92.90	92.90
	4	10.48	10.20	0	0	92.90	92.90
Kalina Cycle	1	116	116	0.69	0.68	16.8	16.8
	2	41.73	43	0.95	0.94	11.44	11.4
	3	8.29	8	0	0	16.8	16.8
	4	60.47	63	0	0	16.8	16.8
	5	116	116	1	1	11.44	11.4
	6	116	116	0	0	5.36	5.4
	9	43.40	46	0.64	0.64	16.8	16.8
	10	30.50	30	0.57	0.56	16.8	16.8
	11	8.65	8	0	0	16.8	16.8
Uehara Cycle	1	27.26	27.26	0.66	-	2.060	2.060
	2	11.34	11.07	0.99	-	1.355	1.438
	3	12.52	12.32	0	-	2.058	2.055
	4	16.52	15.77	0	-	2.060	2.060
	5	27.26	27.26	1	-	1.357	1.443
	6	27.26	27.26	0	-	0.703	0.617
	9	15.87	15.75	0.65	-	2.058	2.055
	10	20.05	18.40	1	-	1.357	1.443
	11	12.68	12.38	0	-	2.058	2.055
	12	12.69	12.39	0	-	2.060	2.060
	15	18.17	18.40	0	-	0.002	0.005

3. Cycle flow and mathematics model

Bring the initial parameters into the calculation model and the loop results are shown in Figure 2 and Table 4. At the same temperature of the cold and heat source, the flow rate of the work flow, and the outlet and inlet pressure of the turbine, only the cycle flow is changed. Compared with the results of the three cycles, it can be seen that the net power generation of the Rankine cycle with pure ammonia as the work flow is greater than that of the other cycles under the three heat source temperature conditions. The reason is that the pure ammonia work flow can enter into the turbine after the evaporation, and participate in the expansion process. The Kalina cycle and the Uehara cycle use ammonia solution as the work flow. Under the condition of the total work flow flow rate is constant, the work flow through the evaporator is in the gas-liquid mixed state. Only the vapor work flow can enter the turbine and work, and the flow rate of the work flow into the turbine is reduced. Its output power and power generation are correspondingly reduced. In terms of thermal efficiency, when the heat source temperature is 28℃ and 122℃, the thermal efficiency of the Kalina cycle is the highest, and when the heat source temperature is 255℃, the thermal efficiency of the Uehara cycle is the highest. This is because the Kalina cycle and the Uehara cycle add a regenerative process in the process, which reduces the heat absorption Q_{ev} of the evaporator, and the decrease is greater than the decrease of net power generation. Therefore, the thermal efficiency derived from formula (14) (22) is slightly higher than that of Rankine cycle. In terms of LCOE and investment recovery period, at 28℃, although the total heat transfer area demand of Rankine cycle is large, it will lead to higher total investment cost, but due to the large net power generation, the LCOE of Rankine cycle is small during the 20-year operation period. At 122℃ and 255℃, the Uehara cycle greatly reduces the heat transfer area of the heat exchanger required for the project, thereby reducing the investment cost and making it an advantage in leveling the cost of electricity.



a. T-η b. T-LCOE

Fig.2 Result of the thermal efficiency and LCOE of cycles with initial paramerers

Table 4 The simulation results of cycle

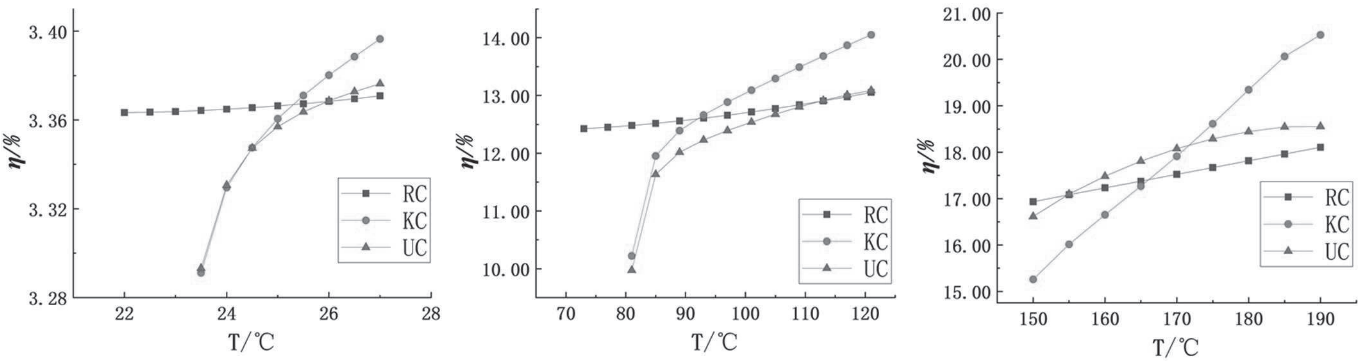
$T_{hotsource}$ / ℃	28			122			255		
Parameters	RC	KC	UC	RC	KC	UC	RC	KC	UC
Q_{eva}/MW	119.34	88.31	88.18	24.23	16.14	16.92	3.57	2.92	2.50
W_t/MW	4.07	3.04	3.02	3.22	2.31	2.27	0.67	0.58	0.58
W_p/kW	52.68	50.32	50.59	75.63	76.88	73.95	50.62	80.02	46.91
W_{net}/MW	4.02	2.98	2.97	3.14	2.23	2.20	0.62	0.50	0.53
η/%	3.368	3.380	3.369	12.960	13.825	12.989	17.377	17.267	17.807
A/m^2	12313	9907	9597	1897	1127	701	553	107	92
LCOE/ $\$\cdot$kWh^{-1}	0.0448	0.0508	0.0497	0.0183	0.0198	0.0177	0.0493	0.0483	0.0405

The temperature and flow rate of the heat source are known based on the data in References (Bernardoni, et al., 2019, Ogrisick, 2009 and Köse Ö., et al., 2021). In the design process, the net power generation and power generation cost of the cycle can be affected by changing the evaporation temperature of the work flow and the inlet pressure of the turbine.

3.1 Effect of evaporation temperature

Under different heat source temperature conditions, when the pressure is constant, increasing the evaporation temperature of the work flow can increase the steam enthalpy, thereby increasing the turbine power. However, the increase of evaporation temperature will increase the heat load of evaporator, thereby increasing the heat transfer cost. Therefore, it is necessary to study its specific impact on the cost. The changes of η and LCOE with evaporation temperature are shown in Fig. 3 and Fig. 4.

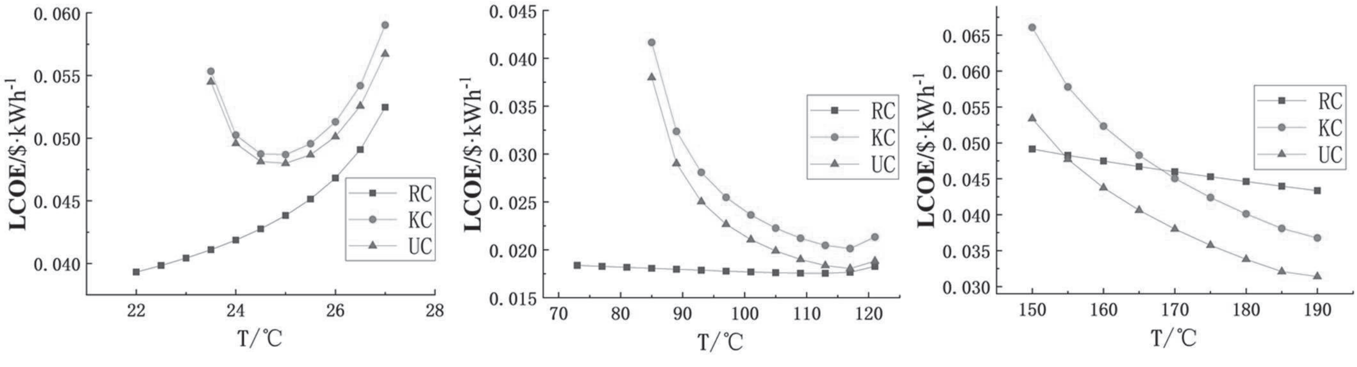
The thermal efficiency of Rankine cycle increases linearly with the increase of heat source temperature under three heat source conditions. The growth rate of thermal efficiency of Kalina cycle and the Uehara cycle has slowed down, because the cycle uses ammonia-water mixed work flow. The boiling point of ammonia is lower than that of water and evaporates first. In the process of increasing evaporation temperature, the mass fraction of water vapor will increase until all work flows become superheated. The ammonia concentration of steam gradually decreases and finally tends to be stable. Under the same pressure and temperature, the enthalpy of ammonia is higher than that of water and works more in the turbine. Therefore, when the ammonia concentration decreases, the growth rate of thermal efficiency will slow down.



a. 28 ℃ T-η b. 122 ℃ T-η c. 255 ℃ T-η

Fig.3 Effect of the evaporation temperature on thermal efficiency of cycles

Economic indicators. It can be seen from Fig.4-a that in OTEC environment with the heat source temperature of 28℃, the LCOE of Rankine cycle increases with evaporation temperature, because the evaporation temperature increases the demand for heat transfer area of evaporator. Therefore, in the ocean temperature difference environment, the Rankine cycle should select the saturated vapor temperature of the work flow as the evaporation temperature, so that the cost is the lowest. The Kalina cycle and the Uehara cycle achieve a minimum value (0.0487$/kWh、0.0476$/kWh) near 25℃. This is because the evaporation temperature continues to rise, which will make the evaporation temperature close to the heat source temperature, greatly increase the heat transfer demand area, and increase the evaporator cost. When the heat source temperature was 122℃, the LCOE of Rankine cycle was basically maintained at about 0.018$/kWh. The Kalina cycle and the Uehara cycle were similar to the heat source at 28℃, and the LCOE reached the minimum at 117℃. When the heat source temperature is 255℃, the evaporation temperature increases, and the LCOE of the Rankine cycle decreases linearly due to the increase of power generation. The Kalina cycle and the Uehara cycle are lower than the Rankine cycle at 170℃ and 155℃, respectively. The LEOC of the Uehara cycle is the lowest, which is 0.031$/kWh. So, the Uehara cycle in high temperature environment has more cost advantage.

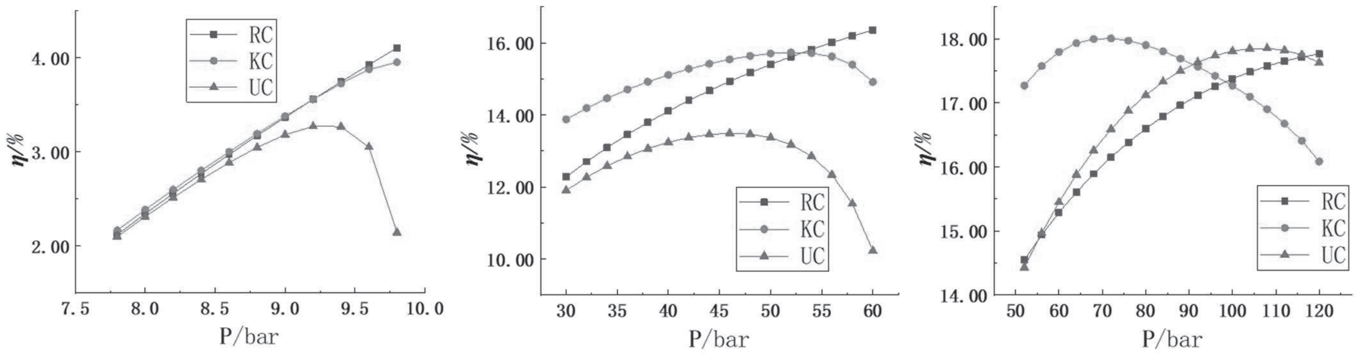


a. 28 ℃ T-LCOE b. 122 ℃ T-LCOE c. 255 ℃ T-LCOE

Fig.4 Effect of the evaporation temperature on leveled cost of energy

3.2 Effect of turbine inlet pressure

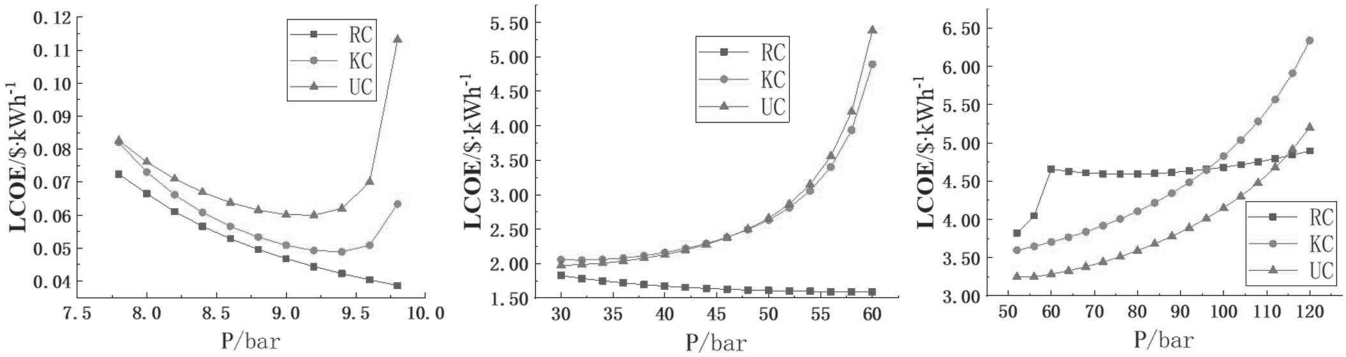
The turbine inlet pressure is an important parameter affecting the work flow state, which determines the enthalpy of the work flow at the turbine inlet. For the ammonia-water mixed work flow, the turbine inlet pressure also affects the amount of air intake. When the given parameters are kept as initial values and only the turbine inlet pressure is changed, the changes of η and LCOE with the turbine inlet pressure are shown in Figure 5 and Figure 6.



a. 28 ℃ P-η b. 122 ℃ P-η c. 255 ℃ P-η

Fig.5 Effect of the turbine input pressure on thermal efficiency of cycles

It can be seen from Figure 5 that when the heat source temperature is 28℃, the thermal efficiency of the Rankine cycle increases linearly with the increase of pressure, and the growth rate of the Kalina cycle slows down after 9.2bar. At this time, the thermal efficiency of the Rankine cycle is higher than that of the Kalina cycle. The reason is that the increase of pressure will lead to the decrease of the vapor fraction of the work flow, and the work flow participating in the turbine will be reduced, which will reduce the power generation and lead to the decrease of thermal efficiency. It is also because of this that the gas work flow of the Uehara cycle is reduced, not only its power generation is reduced, but also its regenerative effect is greatly reduced due to the use of suction regenerative method. After the Uehara cycle reaches the extremum at 9.4bar, the thermal efficiency begins to decline. When the heat source temperature is 122℃, the variation of thermal efficiency is the same as that at 28℃. The maximum thermal efficiency of Kalina cycle and the Uehara cycle is 15.7% and 13.5% at 52bar and 46bar, respectively. At this time, the vapor fraction of work flow evaporation is only 44.6% and 53.9%. In the medium and high temperature environment at 255℃, when the pressure is greater than 92bar, the extraction and heat recovery effect of the Uehara cycle is better than that of the Kalina cycle, which increases the gas phase fraction of the work flow under the same evaporation temperature, thus increasing its power generation and increasing the cycle thermal efficiency. The thermal efficiency of the Kalina cycle reaches an extreme value of 18% at 72bar, and the corresponding evaporation gas fraction is 92.1%. The thermal efficiency of the Uehara cycle is 17.85% at 106bar, and the evaporation gas fraction is 80.3%.



a. 28 ℃ P-LCOE b. 122 ℃ P-LCOE c. 255 ℃ P-LCOE

Fig. 6 Effect of the turbine input pressure on leveled cost of energy

As can be seen from Fig. 6, when the work flow flow rate and evaporation temperature are the same, the gas phase fraction of the work flow out of the evaporator decreases with the increase of pressure in the Kalina cycle and the Uehara cycle, and the flow rate of the vapor work flow entering the turbine decreases, resulting in the decrease of power generation and the increase of LCOE. Rankine cycle has high net power generation, and its LCOE is the smallest when the heat source temperature is 28℃ and 122℃. When the heat source temperature was 255℃, the LCOE of the Uehara cycle was the smallest, and the Rankine cycle had little change, but with the increase of pressure, the LCOE was lower than that of the Kalina cycle and the Uehara cycle at 96bar and 114bar, respectively. It is worth noting that below 60bar, the LCOE of Rankine cycle drops sharply. This is because when the evaporation temperature is the same, the decrease of pressure leads to the increase of superheat degree of pure ammonia work flow. After the expansion of the turbine, the work flow gradually becomes superheated, which makes the temperature of the work flow at the turbine outlet increase and the logarithmic mean temperature difference of the condenser increase. Therefore, the heat transfer area required by the condenser is reduced, and the cost is reduced. However, at the same time, because the pressure difference between the two ends of the turbine is reduced, the net output power of the cycle is reduced. As shown in Fig. 7, the power generation will also be reduced. Therefore, it is not recommended to use the Rankine cycle in this state. At this time, the Kalina cycle and the Uehara cycle have higher net output power and power generation, and the pressure in this state should be used.

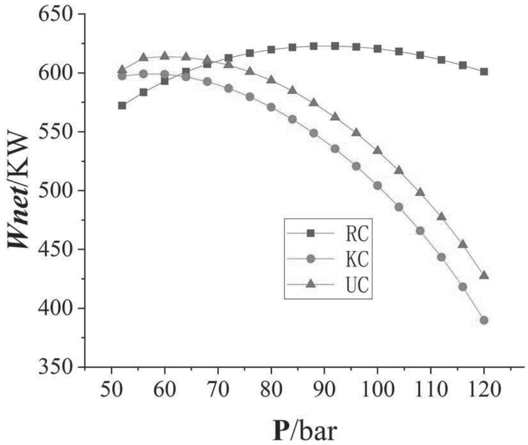


Fig.7 Effect of the turbine input pressure on net work of cycle

4. Conclusions

This paper selects 28℃, 122℃, 255℃ three heat source temperature, covering the ocean energy thermoelectric power generation, industrial thermal power generation, solar thermal power generation and geothermal power generation most of the thermoelectric power generation application scenarios. thermal efficiency and the thermal economic efficiency of the Rankine cycle, the Kalina cycle and the Uehara cycle is compared and analyzed, and the following conclusions are drawn.

(1) From the perspective of economic efficiency, when the heat source temperature is 28℃, the LCOE of pure ammonia work flow is the lowest, when the heat source temperature is 122℃, the Rankine cycle and the Uehara cycle have lower LCOE, but the net power generation and total investment cost of the Rankine cycle are higher than those of the Uehara cycle. When the heat source temperature is 255℃, the Uehara cycle has the lowest LCOE, and its power generation is equivalent to the Rankine cycle, so the Uehara cycle is more suitable for high temperature environment.

(2) Evaporation temperature increases the thermal efficiency and net power generation of the cycle, but it also increases the cost of the evaporator. When the evaporation temperature is close to the heat source temperature, the LCOE of the Kalina cycle and the previous cycle appears a minimum. When the evaporation temperature of the work flow is designed, the temperature range can be reduced to the heat source temperature. For the Rankine cycle, when the heat source is low, the difference between the heat source temperature and the saturated temperature of the work flow in the Rankine cycle is small, and the evaporation superheat of the work flow should be reduced. In the case of high temperature heat source, the heat source temperature is quite different from the saturated temperature of Rankine cycle work flow, and the evaporation superheat can be appropriately increased.

(3) Increasing the turbine inlet pressure improves the thermal efficiency of the Rankine cycle, while the Kalina cycle and the Uehara cycle will have a maximum point. The Rankine cycle pressure should be set as the saturated steam pressure at the specified evaporation temperature, while the maximum thermal efficiency of the Kalina cycle and the

previous cycle has a deviation from the minimum LCOE, and the deviation increases with the improvement of the heat source temperature. In order to balance the power generation and construction cost, the pressure should be set according to the LCOE standard.

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