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Simulation and Parametric Study on Kalina Cycle Coupled Absorption Heat Transformer

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Keywords: Kalina cycle; Absorption heat transformer; Ammonia-water mixture

ABSTRACT

A novel Kalina cycle coupled absorption heat transformer (KC-AHT) is proposed in this paper. The temperature of geothermal tail water of Kalina cycle, as an example in Husavik, Iceland, is about 80oC, which can still be used for district heating. But for general regions, 60oC is enough as the heat source of district heating. In the premise of satisfying district heating, some measures can be adopted to decrease the temperature of geothermal tail water for the target to improve the power output. Consequently, the energy contained by geothermal tail water can be recycled using KC-AHT to decrease the thermal pollution caused by geothermal tail water and increase the power output. In order to expedite the calculation and study of KC-AHT under various boundary conditions, the engineering equation solver (EES) software package is used, and the simulation result is validated. The result shows that the generator inlet temperature of basic solution is increased from 64.6oC to 73.2oC, which leads to the net power output of KC-AHT increases by 6.8% and 14.9% than that of Kalina cycle in reference and Kalina power plant in Husavik, Iceland, respectively. Besides, the parametric study shows that the best ammonia concentration of basic solution is 0.9 in general regions, not 0.82 in Iceland, the temperature of final tail water and cooling water has a large effect on the power output, the optimum turbine inlet pressure and ammonia concentration are existent to guarantee the highest net power output under a certain condition.

1. INTRODUCTION

The geothermal resources are rich in China, but most of them belong to low- and medium-temperature geothermal resources (Zhang, et al., 2009). Water is adopted as working medium in the traditional Rankine cycle which shows a great limit during heat exchange process with heat source in the generator because of the constant evaporation temperature. Consequently, it has a large temperature difference which results in irreversible loss and low thermal efficiency. The problem is reduced by Kalina cycle which proposed by Dr. Kalina. Compared with traditional Rankine cycle, the most important difference is that ammonia-water mixture is adopted as working medium in Kalina cycle with the aim of reducing thermal irreversibility during the heat transfer process, especially between the heat source and the evaporating working medium.

The research of Kalina cycle on technology and economic reveals the essence and direction of improvement. El-sayed and Tribus (1985) compared Rankine cycle and Kalina cycle, they pointed out that the thermal efficiency of Kalina cycle increases by 10%~20% than that of Rankine cycle. Marston (1994) pointed out the temperature of separator and the turbine inlet pressures are the key factors in optimizing Kalina cycle. Kalina and Leibowitz (1991) proposed a power cycle with geothermal energy as heat source and pointed out that Kalina cycle had a higher power output than Rankine cycle with isobutene as working medium in the same geothermal resource. Madhawa(2007) pointed out that Kalina cycle was more promising through comparing the thermal efficiency between Kalina cycle and Organic Rankine Cycle (ORC) that applied in low temperature geothermal resource. Bo (1989 and 2007) analyzed the performance of Kalina cycle when it is just proposed. The research showed that Kalina cycle is obviously excellent. Zhang (2007) studied the thermodynamic property of ammonia-water mixture and the thermal performance of Kalina cycle. The different heat sources response different types of Kalina cycle, so that the thermal efficiency can be improved. Wang (2008)studied the merits and demerits of Kalina cycle and Rankine cycle. The performance of Kalina cycle is better than that of Rankine cycle without the type of heat sources. Fu (2013) proposed a cascade utilization system included Kalina cycle during the oil production process, and the economic efficiency was improved.

A series of ammonia-water power cycles based on Kalina cycle were proposed by some other researchers. Xu (2000) proposed the absorption power and refrigeration combined cycle and the thermal performance was studied. Wu (2003) and Zheng (2002) analyzed some combined cycles based on Kalina cycle with ammonia-water mixture as working medium. Liu (2006) proposed a novel power and refrigeration combined cycle and the cycle was analyzed and optimized by thermal efficiency and exergetic efficiency as indicators. Zheng (2012) proposed a new refrigeration and heating system which combined geothermal and solar energy through ammonia absorption cycle, and the influence of heat source temperature to the refrigeration and heating efficiency was studied. Liu (2012)optimized the ammonia absorption refrigeration system using the pinch analysis method and pointed out the performance coefficient had an increase by 11.58%. Chen (2012) proposed triple pressure ammonia-water power cycle for waste heat recovery and the thermal performance was studied.

Currently the researches on Kalina cycle and ammonia-water power cycle based on Kalina cycle are almost staying in theory state. Only several Kalina power plants are in operation throughout the world. Among them the most famous Kalina geothermal power plant is in Husavik(Ogriseck, 2009), in which the temperature of geothermal water from well is about 122oC and the temperature of tail water is about 80oC, the actual power output is about 1950 kW (USvika, 2009) . Adopted it as prototype, the KC-AHT is proposed in this paper. Besides, the simulation and parametric study on KC-AHT are carried out and a comparison between the Kalina cycle power plant and KC-AHT is made.

2. DESCRIPTION AND MODELING

2.1 Description

According to the actual conditions in Husavik, the temperature of tail water is about 80oC and amount of energy is still contained. It is a waste of thermal energy and easy to cause thermal pollution if discharged directly. However, it is not possible to decrease the geothermal tail water by Kalina cycle itself, and some measure is needed to achieve the target. So a district heating system is developed and good economic benefit is achieved in Husavik, Iceland. But for general regions, the ambient temperature is higher than Iceland and 60oC is enough as the heat source of district heating. In the premise of satisfying district heating, some measures can be adopted to decrease the temperature of geothermal tail water for the target to improve the power output. It is possible to increase the net power output by recycling and re-injecting the energy contained by tail water into the power cycle with some measures. The basic Kalina cycle system (KCS) is shown in Fig.1.

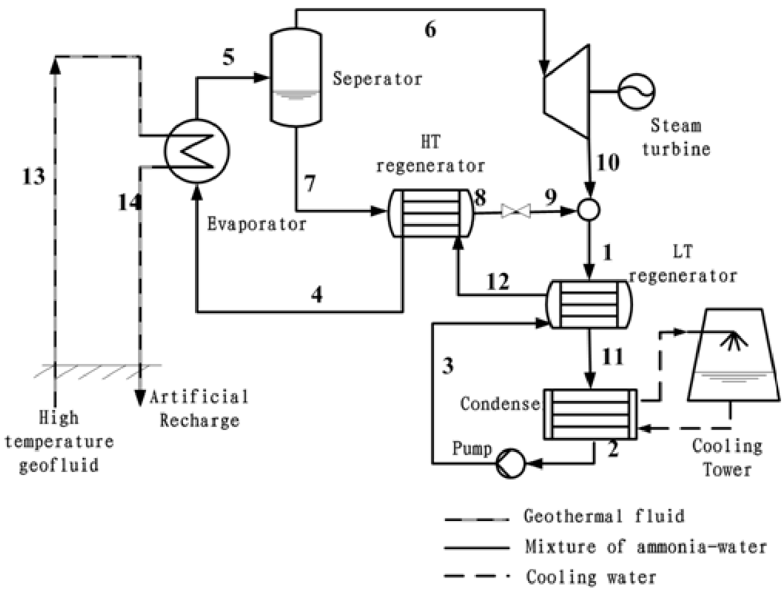

Fig.1. Basic Kalina cycle system (KCS)

Absorption heat transformer is a kind of system that can transfer energy from low-grade heat source to high-gradheat source (Fang, et al., 2008 Huang, et al., 2008 and Gao., et al). It is possible to recycle the tail water of 80oC because the temperature of driven heat source used by ammonia absorption heat transformer is only about 50oC~90oC generally. Consequently, the KC-AHT is possible in theory. Based on the theory, the schematic diagram of KC-AHT is developed as shown in Fig.2, in which the improved parts are shown in a red dashed box.

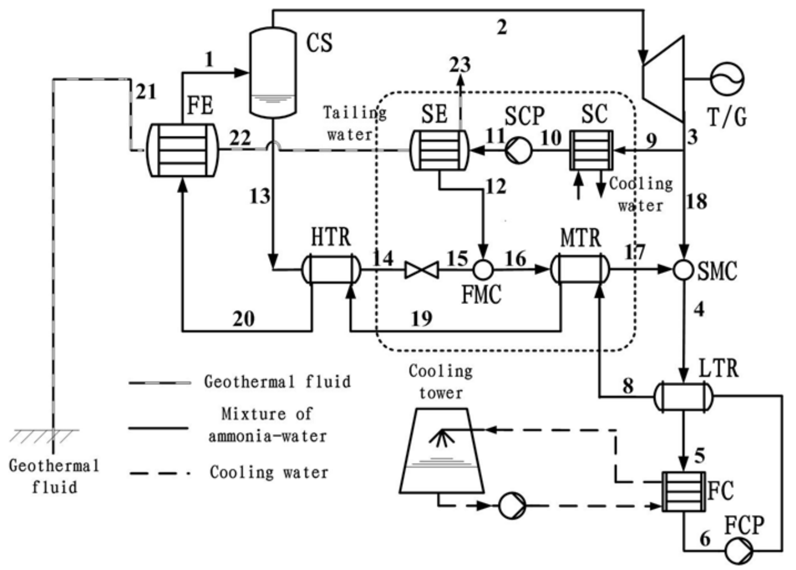

Fig.2 Schematic diagram of KC-AHT

The basic solution (ammonia-water mixture) becomes vapor-liquid two-phase mixture (2) after heat exchange with geothermal water in the generator, then leaves the generator as a saturated mixture and enters the separator afterwards. The ammonia vapor (3) separated from the top of separator is expanded in the turbine to generate power as well as the ammonia-poor solution (10) separated from the bottom of separator flows through the high-temperature (HT) recuperator (11) and throttle (12) to be cooled and depressurized to 15bar, respectively. At the same time, the turbine exhaust (4) is split into two streams and the split ratio is 1:1. One of the two streams (5) is cooled to be saturated liquid (7) in condenser 2st and pressurized (8) to be middle pressure (15bar) in pump 2st, then enters into the evaporator to recycle the remaining energy content of geothermal water which has heat exchange with basic solution in the generator. Then stream (9) is mixed with stream (12) in the mixture 2st which is adiabatic. The mixing process of ammonia-water is an exothermic process, but in order to simplify the calculation program, it is only a mixing process without exothermic process in the mixture 2st but exothermic in the absorber. The stream (13) enters into the absorber to release dissolution heat which is absorbed by low temperature basic solution (19). At the same time, the other stream (6) of turbine exhaust is diluted with stream (14) in the mixture 1st which is adiabatic and be condensed (15, 16, 17) in the low-temperature (LT) recuperator and condenser 1st by the low temperature basic solution and cooling water, respectively. The saturated basic solution (17) leaving condenser 1st is pressurized (18) to high pressure (32.3bar) in the pump 1st. Then high pressure basic solution is sent to LT recuperator (18), absorber (19) and HT recuperator (20) in turn to recycle thermal energy from high temperature solution. At last, the basic solution (1) enters into the generator and the whole process starts again.

The absorption heat transformer subsystem is the key component for the KC-AHT. The advantages of KC-AHT are mainly shown in two aspects. Firstly, the benefit comes from the absorption heat transformer subsystem, because the remaining energy content of geothermal water which has heat exchange with basic solution in the generator can be recycled by the condensed turbine exhaust and the temperature of final discharged tail water is decreased. Secondly, the dissolution heat released in the absorber can be absorbed by the low temperature basic solution to increase the generator inlet temperature of basic solution, which leads to a higher mass flow of ammonia vapor and the net power output is increased finally.

2.2 Modeling

According to the actual parameters of Kalina cycle power plant, the input parameters and theoretical calculation equations in each unit of the KC-AHT can be confirmed as shown in Tab.1 (a) and (b). The basic models for all units involve mass, energy and component conservation equations. In order to make the calculation simplify, the following assumptions are made in this paper:

a) The cycle is operated under a steady state all the time.

b) The turbine and pumps have isentropic efficiencies.

c) There is no pressure drop along the pipeline.

d) Neglect the power consumption of cooling water pump.

Table1 (a) Input parameters of KC-AHT

Temperature of geothermal water/^{o}C	122	High pressure/bar	32.3
Temperature of middle tail water/^{o}C	80	Turbine isentropic efficiency	0.87
Temperature of final tail water/^{o}C	60	Generation efficiency	0.96
Cooling water inlet temperature/^{o}C	5	Pump isentropic efficiency	0.98
Mass flow of geothermal water/kg/s	89	Split ratio of turbine exhaust	1:1
Ammonia content of basic solution	0.82	Middle pressure/ bar	15
Minimum temperature difference of generator/^{o}C	6	Minimum temperature difference of evaporator/^{o}C	6
Minimum temperature difference of recuperator/^{o}C	5	Minimum temperature difference of condenser/^{o}C	3
Pressure drop of heat exchanger/bar	1	Pressure drop of pipeline/bar	0

In the equations as shown above, Q represents energy, kJ; m represents mass flow of solution, kg/s; x represents ammonia concentration; W represents power, kW; η represents thermal efficiency; h represents enthalpy, kJ/kg; s represents entropy, kJ/kg; p represents pressure, bar; v represents specific volume, m3/kg.

The calculation model of KC-AHT is a series of nonlinear equations. EES is a kind of software that can solve hundreds of coupled non-linear algebraic equations and initial value differential equations. A major characteristic of EES software is the high accuracy of thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability. Because the thermodynamic property of ammonia-water mixture can be got easily through EES, EES is adopted and the thermodynamic calculation program is written based on the assumptions and equations shown above. With the input parameters, EES software calls other functions and packages immediately to calculate the results. At last running the simulation program and getting the calculation results of each state point. The solving process of KC-AHT by EES is shown as Fig.2.

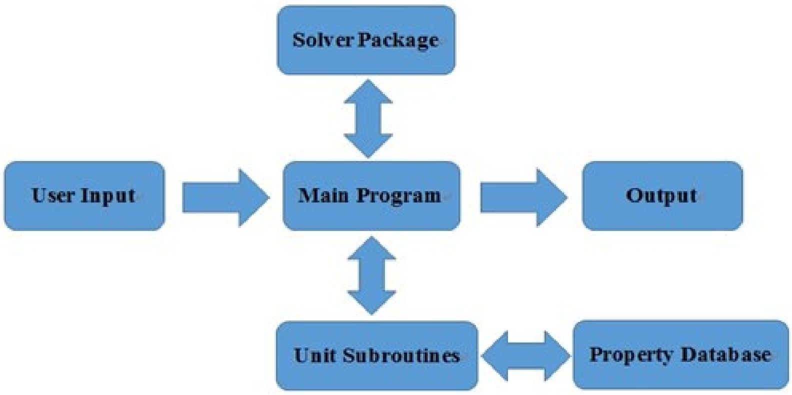

Fig.3 EES solving process of KC-AHT

Table1 (b) Control equations

	Equations
Energy conservation equation	$\Delta_{out}^{in}\sum_i Q_i = 0$
Mass conservation equation	$\Delta_{out}^{in}\sum_i m_i = 0$
Generator	$m_{gen}\cdot(h_{in}-h_{mid}) = m_{basic}\cdot(h_2-h_1)$
Evaporator	$m_{gen}\cdot(h_{mid}-h_{out}) = m_5\cdot(h_8-h_9)$
Separator	$m_{basic} = m_5 + m_{10}$ $m_2\cdot x_{basic} = m_3\cdot x_3 + m_{10}\cdot x_{10}$
Turbine	$W_{tur} = m_3\cdot(h_3-h_4)\cdot\eta_e$ $\eta_s = (h_3-h_4)/(h_3-h_{4e})$
Condenser 1st	$m_{cool\_1}\cdot(h_{24}-h_{23}) = m_{basic}\cdot(h_{16}-h_{17})$
Condenser 2 nd	$m_{cool\_2}\cdot(h_{26}-h_{25}) = m_5\cdot(h_7-h_5)$
Pump 1st	$s_{17} = s_{18}$ $W_{pump\_1} = m_{basic}\cdot v_{17}\cdot(p_{18}-p_{17})\cdot 100/\eta_{pump}$
Pump 2nd	$s_7 = s_8$ $W_{pump\_2} = m_5\cdot v_7\cdot(p_8-p_7)\cdot 100/\eta_{pump}$

3. RESULTS AND DISCUSSION

3.1 Validation

The same input parameters and operating conditions of Kalina cycle studied in reference [19] are adopted in this paper as shown in Tab.1 (a). Under the same boundary conditions, the comparison of main parameters among Kalina power plant in Husavik, Kalina cycle in reference [19] and KC-AHT in this paper is shown in Tab.2

Table2 Main parameters of Kalina cycle system

No.	*T* (℃)		*P* (bar)		*x*		Vapor fraction	
	Simulation	Ref.[3]	Simulation	Ref.[3]	Simulation	Ref.[3]	Simulation	Ref.[3]
1	45.9	46	6.77	6.6	0.82	0.82	0.63	0.64
2	7.9	8	4.77	4.6	0.82	0.82	0	0
3	8.1	8	35.3	35.3	0.82	0.82	0	0
4	62.5	63	33.3	33.3	0.82	0.82	0	0
5	116	116	32.3	32.3	0.82	0.82	0.67	0.68
6	116	116	32.3	32.3	0.972	0.97	1	1
7	116	116	32.3	32.3	0.511	0.50	0	0
8	44.7	46	31.3	31.3	0.511	0.50	0	0
9	45.1	—	6.77	—	0.511	—	0	—
10	42.1	43	6.77	6.6	0.972	0.97	0.944	0.94
11	29.7	30	5.77	5.6	0.82	0.82	0.543	0.56
12	39.7	41	34.3	34.3	0.82	0.82	0	0
13	122	122	—	—	—	—	—	—
14	80	80	—	—	—	—	—	—

A simple way to evaluate the alternative power cycles during preliminary power cycle design is to compare the performance of any new proposed cycle which produces the power output under the same conditions, so the thermodynamic comparison between Kalina and DA cycle is made. The parameters are reported in Table 3 and Table 4 under the same conditions, respectively. The simulation results of the Kalina cycle have been validated with the results in the Ref. [3, 4]. The DA cycle compares to the Kalina cycle in a T-s plane in Figs. 3 and 4 [15]. The total power output of Kalina cycle is lower than that of the DA cycle under the same condition. The work output increases by 221kW, about 10.5%. Because the tail water is adopted more temperature drop, the DA cycle is designed to use the heat of geothermal fluid from 122oC to 60oC, while only 122oC to 80oC for Kalina cycle. The thermal efficiency of DA cycle is lower, because the total heat amount of DA cycle is much more. The geothermal fluid below 80oC is difficult to use except for heating, but the heat of geothermal fluid can be used to generate electricity in the DA cycle. It is inferred that the DA cycle, which is superior in recovering low-grade waste heat, is suitable for waste heat recovery in cement plant. Compared to the Kalina cycle, the DA cycle can generate more electricity.

Table 3 Results of KC-AHT

No.	***T*** **(oC)**	***P*** **(bar)**	***x***	**Vapor fraction**
1	116	32.3	0.82	0.6708
2	116	32.3	0.9718	1
3	42.33	6.769	0.9718	0.944
4	40.62	6.769	0.82	0.5923
5	28.28	5.769	0.82	0.5273
6	8	4.769	0.82	0
7	8.428	36.3	0.82	0
8	35.38	35.3	0.82	0
9	42.33	6.769	0.9718	0.944
10	8.934	5.769	0.9718	0
11	9.009	8.769	0.9718	0
12	52.79	7.769	0.9718	0.9538

13	116	32.3	0.5107	0
14	68.45	31.3	0.5107	0
15	——	——	——	——
16	55.6	7.769	0.7434	0.5
17	40.38	6.769	0.7434	0.4136
18	42.33	6.769	0.9718	0.944
19	59.02	34.3	0.82	0
20	74	33.3	0.82	0
21	122	——	——	——
22	80	——	——	——
23	60	——	——	——

Comparison results

	KCS	**KC-AHT**
Work output	2108 kW	2329 kW
First pump power	74.38 kW	101 kW
Second pump power	——	3.57 kW
Efficiency, Z (%)	13.43	9.42

2.2 Discussion

2.2.1 Limitation

It is difficult to condense the turbine exhaust completely due to the ammonia concentration of turbine exhaust is too high (0.97), which is almost pure ammonia vapor. Consequently, the temperature of bubble point is low and the turbine exhaust only can be condensed to liquid completely by the cooling water whose temperature is lower than the bubble point. The high requirement on the temperature of cooling water caused a great limit on the application of KC-AHT in general regions.

The pressure of turbine exhaust that flow out the condenser #2 is 5.64 bar and the vapor fraction is 0 in the simulation, whose temperature of bubble point is only 8.39oC. Because the Countercurrent mode is adopted during the condensation process, so in order to make the high ammonia concentration turbine exhaust condensed to liquid completely, the temperature of cooling water must be lower than 8.39oC. The temperature of cooling water used in the simulation is only 5oC, which comes from Iceland and can complete the condensation process perfectly.

However, the temperature of cooling water like that in Iceland is not easy to get in general regions all the year, which always change with seasons. Consequently, the limit caused by the cooling water cannot be ignored.

When the ammonia concentration is 0.82, the pressures of condensed turbine exhaust that flow out the condenser 2st are changed from 5.64 bar to 11.21 bar as well as the temperatures of cooling water are changed from 5oC to 30oC. At the same time, the temperatures of bubble point are changed from 8.39oC to 29.73oC as shown in Fig.4. When the temperature of cooling water is lower than 8oC, the minimum temperature difference of condenser 2st is higher than 3oC which is meet the assumption used during the simulation. Once the temperature of cooling water is higher than 8oC, it will become more difficult to condense the turbine exhaust completely and has a higher requirement on the condenser, which caused a great limit on the application of KC-AHT especially in summer.

In order to avoid the limit, some measures are necessary to increase the temperature of bubble point such as change the ammonia concentration. During the simulation the pressure of turbine exhaust would increase with the increase of ammonia concentration, which leads to an increase of bubble point temperature. Consequently, it is possible to adopt the higher temperature of cooling water to complete the condensation process. As shown in Fig.3, when the ammonia concentration is 0.9, the pressures of condensed turbine exhaust that flow out the condenser 2st are changed from 6.13 bar to 12.52 bar as well as the temperatures of cooling water are changed from 5oC to 30oC. At the same time, the temperatures of bubble point which flow out the condenser 2st are changed from 10.81oC to 33.49oC. It is obviously the minimum temperature difference of condenser 2st is higher than 3oC in the whole range of cooling water temperature, which meets the assumption and break the limit caused by the cooling water temperature.

But what cannot be ignored is that 0.9 concentration of ammonia-water mixture is not the best concentration during the heat exchange process under the heat source which is 122oC, and the net power output is little lower than that of 0.8 concentration. However, the great advantage is that it breaks the application limit caused by the cooling water temperature. So for general regions, 0.9 concentration of ammonia-water mixture is the best concentration under the heat source and pressure used in the simulation, not 0.82 concentration of ammonia-water mixture which is used in Iceland.

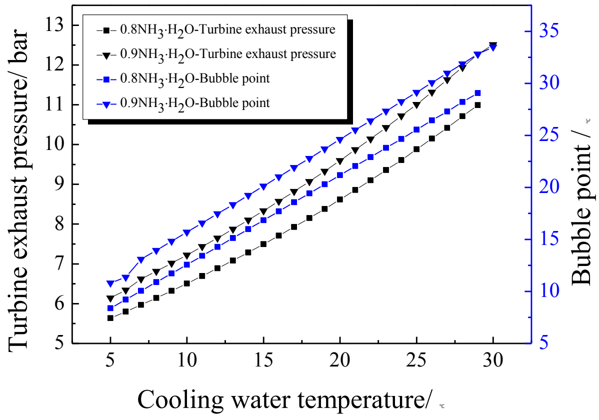

Fig.4 Limitation of cooling water temperature

2.2.2 Effect of final tail water temperature

The temperature of final tail water is about 80oC in Husavik. It can be taken full advantage by KC-AHT that proposed in this paper. The temperature of final tail water can be dropped to 60oC through heat exchange with the condensed turbine exhaust in the evaporator. At last the net power output is increased by 14.9 % and 6.8 %, respectively, which shows a great superiority. In the course of research, it is found that the final tail water temperature has a great effect on whole system. Consequently, a series of data are got through changing the final tail water temperature by EES under the same condition and the results are shown in Fig.5 (a) and (b).

As is shown in Fig.5 (a), the net power output is increased from 2234.5 kW to 3656.2 kW as well as the thermal efficiency is increased from 9.6 % to 13.6 % when the temperature of final tail water is decreased from 60oC to 50oC. Obviously, the lower final tail water temperature is, the higher net power output and thermal efficiency are. It is mainly because more energy can be recycled and re-injected into the cycle by KC-AHT with the decrease of final tail water temperature, which leads to a higher generator inlet temperature of basic solution. At the same time, the relationship between the temperature of final tail water and the mass flow of basic solution is studied. As is shown in Fig.5 (b), with the decrease of final tail water temperature, the mass flow of basic solution is increased. It is because that the generator inlet temperature of basic solution is increased while the generator outlet temperature of basic solution and the thermal load of generator are constant, it must be a higher mass flow of basic solution. At last the mass flow of ammonia vapor which enters into turbine is increased and a higher power output can be generated.

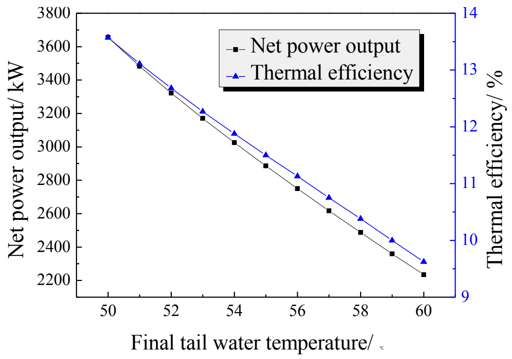

Fig.5 (a) Effect of final tail water temperature

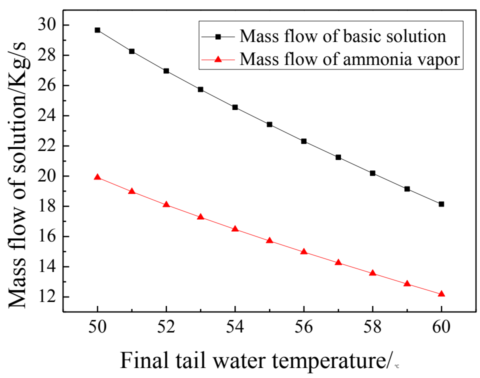

Fig.5 (b) Effect of final tail water temperature

Obviously, great benefits can be obtained with the decrease of final tail water temperature. The lower final tail water temperature is, the higher net power output and thermal efficiency are. Besides, the net power output and thermal efficiency can be increased by optimizing evaporator such as strengthening heat transfer coefficient, increasing heat exchange areas.

2.2.3 Effect of cooling water temperature

The effect of cooling water temperature cannot be ignored. Generally, in order to guarantee the condensation process could be carried out smoothly under ambient condition, the higher cooling water temperature is, the higher turbine exhaust pressure is, which leads to a higher turbine exhaust temperature. The cooling water temperature of 5oC adopted during the simulation process is derived from reference [19]. But for general regions, such a low cooling water temperature is not easy to get all the year. Especially in China where the annual average temperature is about 20oC and the cooling water temperature changes with seasons. So the research on cooling water temperature is necessary. In the calculation program, the turbine exhaust pressure has a relationship with cooling water temperature and it is determined by the cooling water temperature only.

According to the actual ambient temperature in China, cooling water temperature of 5oC~30oC is selected as research object in this paper. At the same time, according to the discussion shown above, only one kind of concentration (0.9) is selected to analyze the effect of cooling water temperature. The simulation result is shown in Fig.6.

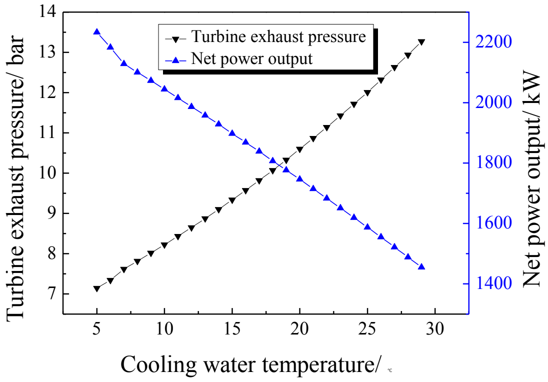

Fig.6 Effect of cooling water temperature

As is shown in Fig.6, turbine exhaust pressure is increased and the net power output is decreased almost straightly with the increase of cooling water temperature. It reveals that the performance of KC-AHT is consistent with the second law of thermodynamics. For example, when the ammonia concentration of basic solution is 0.9, with the condition of cooling water temperature is decreased from 30oC to 5oC, the turbine exhaust pressure is decreased from 13.3 bar to 7.1 bar as well as the net power output is increased from 1454.9 kW to 2233.1 kW. It also reveals that different cooling water temperatures correspond to different optimum turbine exhaust pressures. In China, when the power plant is operated with the cooling water temperature of 20oC all the year, the optimum turbine exhaust pressure is about 10.6 bar, the net power output and thermal efficiency are about 1746.1 kW and 7.52 %, respectively. The effect of cooling water temperature is great and in order to guarantee the best power performance, different turbine exhaust pressures should be adopted according to actual cooling water temperature.

2.2.4 Effect of turbine inlet pressure

The pressure ratio of turbine is an important parameter which can affect the power output obviously. Under the condition of constant cooling water temperature, the turbine exhaust pressure is constant. Consequently, the pressure ratio is decided by the turbine inlet pressure only and the power output could be changed by adjusting the turbine inlet pressure. At the same time, the mass flow of basic solution and the power consumption of feed pumps have a relationship with the turbine inlet pressure. Besides, the optimum pressure is different with the change of ammonia concentration. Consequently, it is necessary to study the effect of turbine inlet pressure.

Four kinds of basic solution with different concentrations (0.6, 0.7, 0.8 and 0.9) are selected and the results are shown in Fig.7 (a). As shown in Fig.7 (a), the net power output increases first and then decreases with the increase of turbine inlet pressure. When the turbine inlet pressure is lower than the optimum pressure, the net power output is increased with the increase of turbine inlet pressure obviously. Once the turbine inlet pressure is higher than the optimum pressure, the net power output will be decreased. On the one hand, with the increase of turbine inlet pressure, the mass flow of basic solution is increased sharply as shown in Fig.7 (b), which can lead to a higher mass flow of ammonia vapor. Also the power output is increased with the increase of pressure ratio within a certain limit. On the other hand, the mass flow of ammonia vapor which generated in the generator will be limited if the turbine inlet pressure is beyond the optimum pressure, and also the power consumption of feed pumps has a corresponding increase with the increase of turbine inlet pressure as shown in Fig.7 (c), which leads to a decrease of net power output. Both the two reasons result in such a trend as shown in Fig.7 (a).

Besides, the corresponding optimum pressures are 29 bar, 36 bar, 41 bar and 45 bar, respectively, when the ammonia concentrations of basic solution are 0.6, 0.7, 0.8 and 0.9, respectively. It obviously shows that the optimum pressure is increased with the increase of ammonia concentration. The optimum turbine inlet pressure is existent to guarantee the highest net power output under a certain condition.

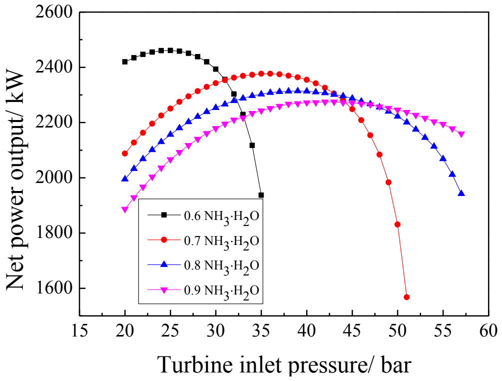

Fig.7 (a) Turbine inlet pressure and net power output

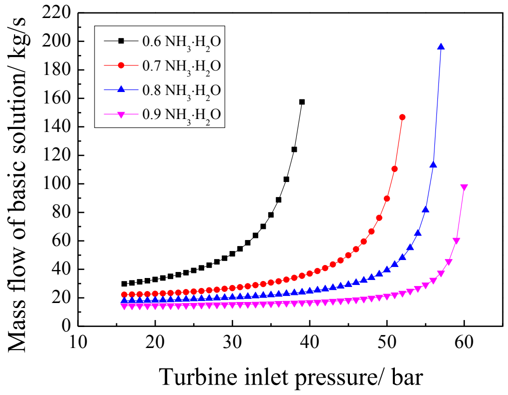

Fig.7 (b) Turbine inlet pressure and mass flow of basic solution

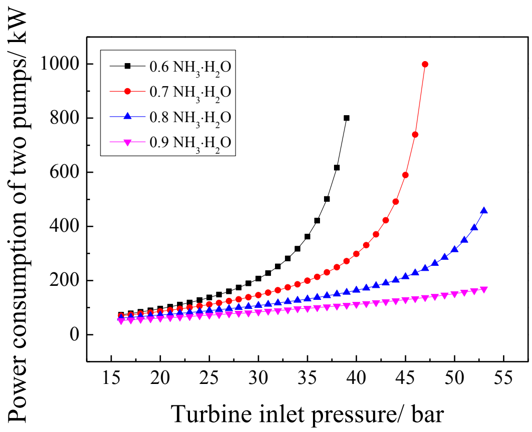

Fig.7 (c) Turbine inlet pressure and power consumption of two pumps

2.2.5 Effect of ammonia concentration of basic solution

The ammonia concentration of basic solution is a key parameter which affects the mass flow of ammonia vapor directly. It is important to study the effect of concentration of basic solution and seek the optimum ammonia concentration under a certain condition.

Four kinds of turbine inlet pressure (30 bar, 35 bar, 40 bar and 45 bar) are selected and the results are shown in Fig.8 (a). The trend of net power output is that it increases sharply first and then decreases gradually with the increase of ammonia concentration. It is mainly caused by two reasons as shown in Fig.8 (b) and (c). Firstly, with the increase of ammonia concentration it has a better match between the basic solution and heat source during the heat transfer process in the generator, which leads to a lower irreversible loss and more ammonia vapor can be generated to expand in the turbine. Consequently, the net power output is increased sharply. Secondly, the ammonia concentration cannot be too high. The advantage of variable evaporation temperature will be weakened if the concentration is close to 100 %, because the ammonia-water mixture is almost pure quality which leads to a high irreversible loss as shown in Fig.8 (b). Also the enthalpy difference of ammonia vapor that flow through the turbine is decreased which leads to a decrease of power output as shown in Fig.8 (c). The synthetic action results in the change trend as shown in Fig.8 (a).

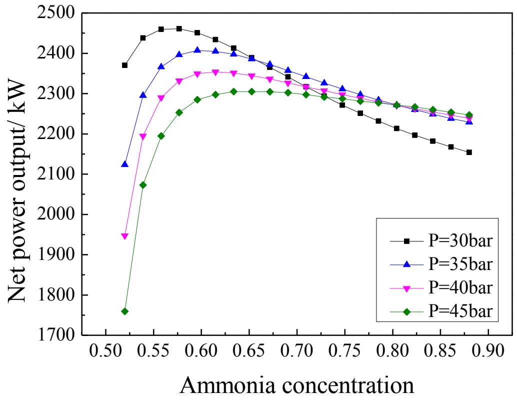

Fig.8 (a) Ammonia concentration and net power output

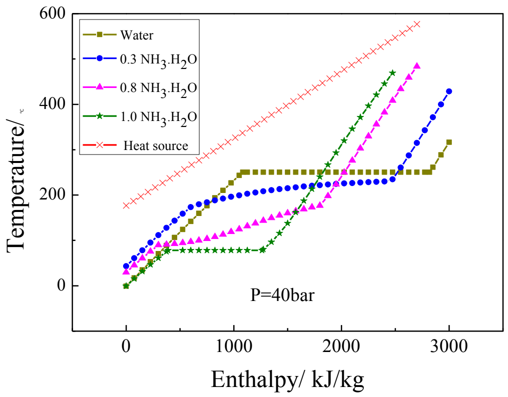

Fig.8 (b) Heat transfer process between different concentrations of basic solution and heat source

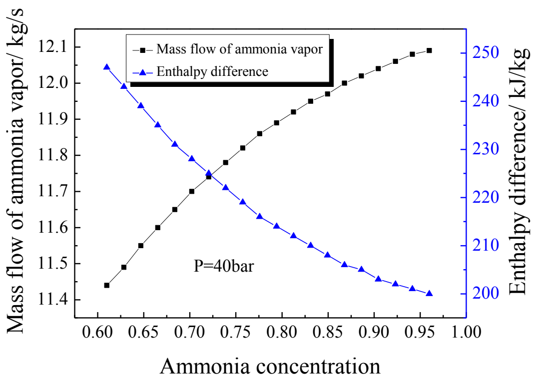

Fig.8 (c) Ammonia concentration and ammonia vapor

At the same time, it can be found the optimum ammonia concentration is existent under a certain turbine inlet pressure. When the turbine inlet pressures are 30 bar, 35 bar, 40 bar and 45 bar, respectively, the optimum ammonia concentrations are 0.59, 0.64, 0.70 and 0.79, respectively. It obviously shows that the optimum ammonia concentration is increased with the increase of turbine inlet pressure.

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