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Development of Micro Grid Kalina Cycle® System – The First Demonstration Plant in Hot Spring Area in Japan

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ABSTRACT

Hot springs are a part of Japanese culture. They have various temperatures, and some of them have over 50deg.C. It is expected that utilization of the hotospring energy will be used for electrical power generation. A binary power generation technology has been used low temperature heat source. The Kalina Cycle® (Ammonia-Water Cycle) - a breakthrough technology provides strong energy conversion performance on low temperature and low brine flow rate conditions. So it is expected that the technology will be adequate for hot spring heat sources. GERD has conducted “Development and Verification Test of Hot Spring Power Generation System” which is a commissioned project by Ministry of Environment. The program included technical development associated with power grid coordination and technical development for determining the effects to the hot springs. Designed heat source condition for 50kW power generation as an annual average is a brine inlet temperature of 98deg.C and brine flow rate of 388L/min. In order to evaluate the power generation performance of a turbine generator, a loading test was conducted by using nitrogen gas at factory. It was confirmed that the generated power was 72.5 kW of electricity with a turbine rotation speed of 50,000 rpm. The demonstration plant was constructed at Matsunoyama hot spring area, Tokamachi city, Niigata pref. in Japan, and verification testing has been conducted since 2012. Since we could not use enough brine heat source to achieve the designed performance due to protect of hot spring resource and high concentration of non-condensable gas, we improved the system to use unutilized steam and brine heat from the well and to adopt higher efficiency heat exchanger. As a result of trial test, we confirmed 44.5kW power generation. In this case, inlet temperature of heat source was lower than designed condition (98deg.C). We are trying to increase inlet heat to designed condition and to operate continuously.

1. INTRODUCTION

"The hot spring" is one of the Japanese representative cultures, and there are approximately 27,500 places of hot springs in Japan (Ministry of Environment, 2013). The high temperature hot springs are distributed over the whole country and especially in Hokkaido, Tohoku and Kyusyu regions (Figure 1). If we can use these hot spring resources for power generation, there is a possibility that micro-grid binary system will be getting popular as dispersion type power source. In large number of high temperature hot springs in Japan, temperature difference energy is remained under-utilized or unused most of times. There are large potential for micro-grid power generation by the existing temperature difference energy of hot springs. It is estimated that there is big heat capacity which can generate electricity of 723,000kW only in the existing hot springs in the country. (Muraoka, 2007).

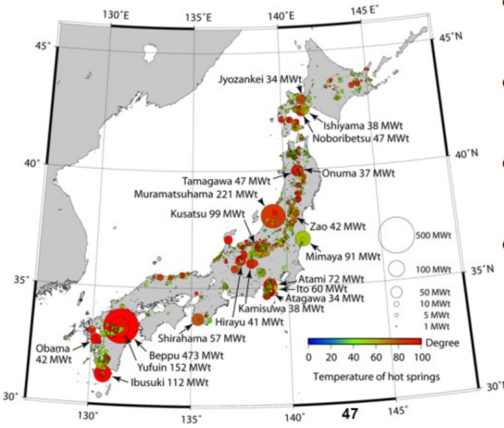

Figure 1 Resources of Hot Spring

The first binary cycle power plant in Japan was Hatchoubaru geothermal power plant which was built in Oita Pref. in 2004. The heat source is low pressure steam that cannot connect to the steam line of the geothermal power plant. The capacity of power generation is 2,000kW. No geothermal binary cycle power plant was built since then. However, geothermal energy was re-evaluated as low carbon and domestic base-load power after "The Great East Japan Earthquake in March 2011" as a motivation, and Feed-in Tariff was started in 2012. Accordingly, compact size binary cycle power plant using low boiling temperature working fluids are becoming more popular. First generation micro grid binary power plants are summarized in Table 1.

Table 1 Micro grid binary power plant in Japan

Year	Location	Power	Temperature	Working Fluid
Dec. 2011	Matsunoyama in Niigata Pref.	87kw	98℃	NH3-H2O
Feb. 2013	Beppu Onsen in Oita Pref.	60kW		HFC-245fa
Feb. 2013	Obama Onsen in Nagasaki Pref.	60kW	105℃	HFC-245fa
Feb. 2013	Yamagawa Geothermal Power Plant (Kyusyu EPC)	250kW		HFE

2. MICRO GRID BINARY POWER PLANT BY HOT SPRING

2.1 Binary cycle

A binary power cycle is a popular method of electricity generation utilizing energy from geothermal hot springs. Electricity is generated by steam energy of low boiling point medium which is evaporated by hot spring heat source. It is called binary cycle because there are two cycles the geothermal heat source system and the closed loop power cycle system.

Binary cycle system include bothe Organic Rankine Cycles (ORC) and Kalina Cycle® for electrical power generation. The basic theory of ORC is conventional Rankine Cycle technology, which is used in steam power station, but it is used low boiling point organic medium for working fluid in binary geothermal applications. In Kalina cycle systems, the working fluid is ammonia-water mixture, which includes the added characteristic of a variable boiling temperature mixture for heat recovery. Advantages of ammonia media are low boiling point and recuced available energy losses during heat transfer processes in comparison with Rankin Cycle systems. But it is necessary to handle carefully due to flammable and odor characteristics.

2.2 Kalina cycle® power plant

2.2.1 Basic concept of Kalina Cycle®

Kalina cycle® is high efficiency power generation cycle designed by Dr. Kalina (Kalina, 1989). There are many type cycles according to heat source temperature and applications. The common characteristics are as follows.

(1) The working fluid is an ammonia-water mixture. The boiling point of the mixture is lower than water, so it will be able to use low temperature resources for power generation.

(2) When ammonia-water mixture is heated at constant pressure, evaporating temperature is variable from the bubble point temperature to dew point temperature. Similarly, condensate temperature is variable from the dew point temperature to bubble point temperature. Because of this characteristic, temperature difference between ammonia water mixture and heat source will be recuced in comparison with ORC systems. Accordingly, it will minimize irreversible energy loss by heat exchange and Kalina cycle systems are appropriate for heat recovery from sensible heat sources.

2.2.2 Kalina cycle system for Hot spring heat source

Kalina cycle system for hot spring heat source includes a simple system which is consisted of Ammonia-feed pump, Evaporator, Recuperator, Turbine-generator and Condenser (Figure 2). Ammonia water mixture is pumped to evaporator. Ammonia is evaporated in evaporator by hot spring heat energy. Ammonia steam and lean ammonia liquid is separated by separator. Ammonia steam is use for power generation by turbine generator. After generation, ammonia steam is condensed to ammonia water mixture at condenser and pumped to evaporator again.

The characteristics of this cycle are as follows:

(1) The evaporator is fed with high concentrated ammonia-water mixture, and it will generate ammonia steam using heat from the geothermal heat source.

(2) Ammonia water mixture is heated to near temperature of heat source by non-isothermal evaporation.

(3) In Rankine cycle, working fluid is evaporated to dry saturated steam. But in Kalina cycle, evaporate is finished at a condition of wet saturated steam.

(4) Wet saturated steam is separated by separator. Remaining heat energy of separated liquid phase (lean ammonia water) is recovered by Recuperator and used for evaporation of ammonia water mixture.

(5) Turbine exhausted ammonia steam is mixed with lean ammonia water and condensed to ammonia liquid.

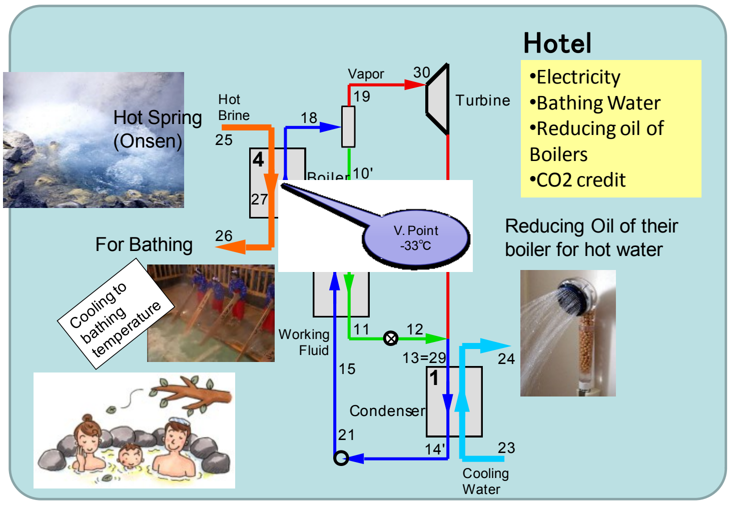

Figure 2 Kalina cycle system for Hot spring heat source

Figure 3 includes an example of temperature profile of ammonia water mixture (77wt%) and HFC245fa for ORC in evaporator in Figure 3. In the figure, the assumed temperature of hot spring is 95 deg.C.

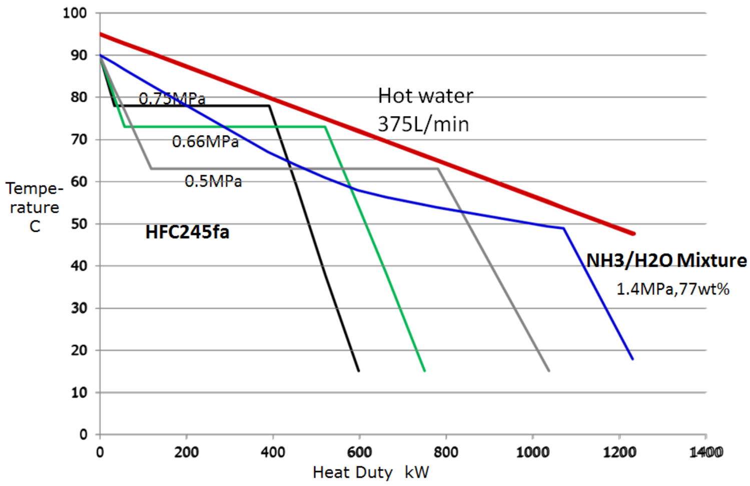

Figure 3 Temperature profile in Evaporator

In case of HFC245fa, the boiling region of energy transfer is based on constant temperature for a single pressure. In case of ammonia water mixture, non-isothermal evaporation results in reduced available energy losses, which improves power generation. Conversely, for a constant electrical production, the Kalina Cycle® requires less brine flow rate

3. DEMONSTRATION PLANT OF MICRO GRID KALINA CYCLE POWER PLANT

3.1 Background

The development of a hot spring power generation system and the installation and trial operation of the system in Matsunoyama, Tokamachi City, Niigata Prefecture were conducted by commissioned project under the auspices of the Ministry of Environment. Technical development associated with power grid coordination and technical development for determining the effects on the hot springs were also conducted in the project.

The objective of this project is to develop a prototype system for hot spring power generation system, and is expected to be highly effective in terms of the reduction of CO2 emissions. We have a cooperation of Tokamachi City that is an owner of hot spring well. It shows over view of Kalina Binary Power Plant in Matsunoyama in Figure 4.

3.2 Outline of demonstration plant

Electrical power plant capacity of this system is 87kw at a condition of heat source 97deg.C, 388L/min and cooling water 7.2 deg.C, 1,250 L/min. The designed net output power is 50kW. At first, we have a plan to use only hot spring water and exchanged the heat with ammonia water in the evaporator. Unfortunately, the production rate of hot spring well was lower than designed rate and it proved difficult to test power plant with the actual available brine flow rate. Well head temperature of the hot spring well is about

130 deg.C and we have a chance to use not only brine but also steam. General power generation systems with hot springs use hot water as the heat source, but a combination with steam heat can decrease production rate from hot spring well. In FY 2013, we introduced a system that utilized both hot water and steam heat produced from the hot spring by installing a steam heat exchanger to heat pure water as an intermediate medium between the geothermal well and the power plant. The concept of the revised heat source system is shown in Figure 5. In this sytem, we introduced pure water as intermediate media. It means that it has no chance to contact hot spring water and ammonia water, so it can reduce risk for mixing of ammonia to hot spring water.

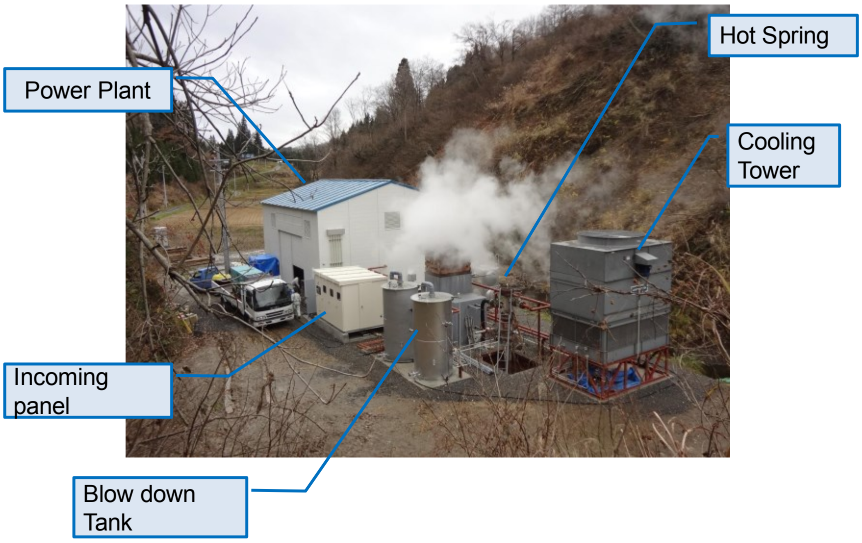

Figure 4 Over view of Kalina Binaly Power Plant in Matsunoyama

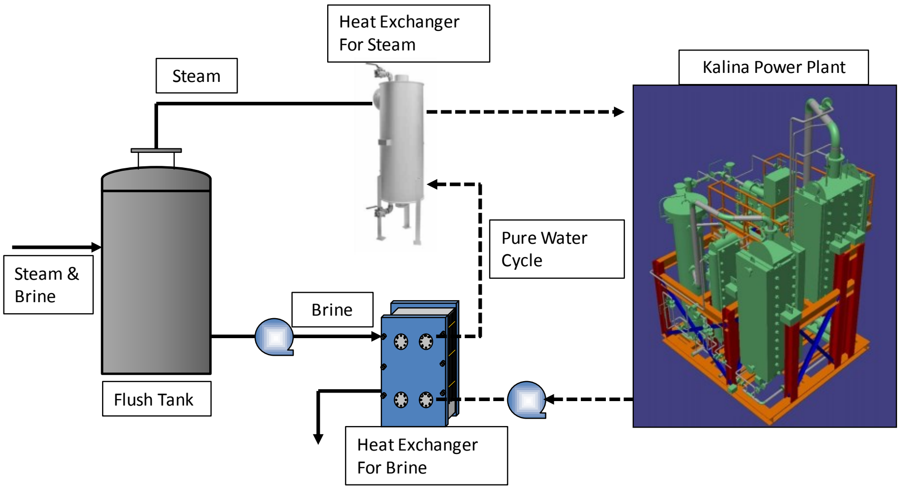

Figure 5 Concept of Heat Source System

3.1 Turbine generator

One of the characteristics of ammonia water media is that the latent heat of vaporization is greater than a working fluid of Organic Rankin Cycle. It leads to decrease circulation volume of working fluid for the same heat source. And it will make high pressure because of the low boiling point of ammonia. Consequently, it is necessary for Kalina-cycle® to adopt high pressure and low flow rate. We selected outer flow type radial turbine called Nanosteam Turbine generator made by Energent Corp. (Phil, 2011). Turbine has connected to generator with single shaft. The generator is a permanent magnet synchronous electric motor producing 87kW at 50,000rpm and converts rotation frequency to grid frequency. This turbine is compact and high efficiency (Figure 6).

4. TRIAL OPERATION OF DEMONSTRATION PLANT

4.1 Trial operation and Result

After finishing construction of power plant, we have conducted trial operation. Hot water was introduced to Evaporator and the flow rate was increased in proportion to circulation rate of ammonia water. When the flow rate was about 310L/min, we could not increase the flow rate further due to gas lock of hot water pump by non-condensable gas. The flow rate of hot water was about 80% compared with rated flow rate, but ammonia temperature at the separator exhaust was 67.4 deg.C which is 27.1 deg.C lower than rated temperature. The ammonia cycle data was shown in Table 2.

Figure 6 Nanosteam Turbine Generator

Table 2 Input heat data for ammonia cycle

	Items	Unit	Design	Test data
NH3 Flow	Flow rate	kg/s	1.301	1.050
	Concentration	%	75.9	76.5
Hot Water	Flow rate	L/min	388	310
	Temperature	deg.C	98.0	98.0
Separator	Pressure	MPaG	1.482	1.432
	Temperature	deg.C	94.5	67.4
	Steam flow rate	kg/s	0.812	0.565
	Liquid flow rate	kg/s	0.489	0.485

We have tried to start turbine under this condition, and the output power was about 10kW. Figure 7 shows the measured output power and the design calculated power as a function of cooling water temperature. The output power is about 20kW lower than designed power (Figure 7).

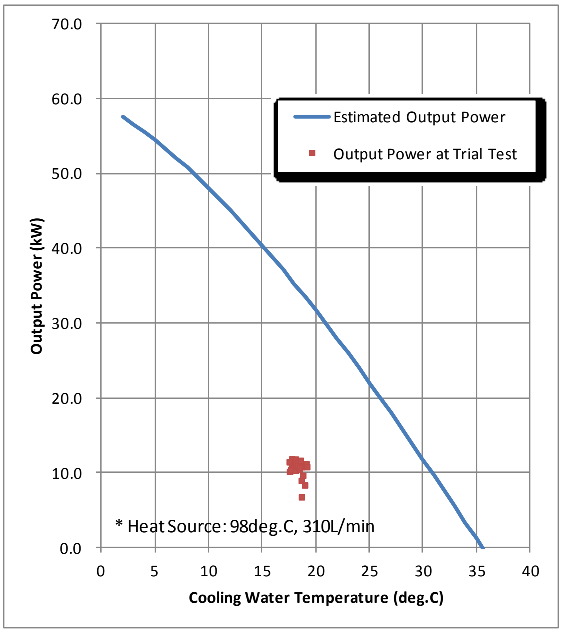

Figure 7 Comparison of output power between calculated value and test data

We also simulated output power based on turbine input and output measured data and the result was about 12.3kW that the difference was about 2.2kW. The running condition was on off-design point, and then the turbine efficiency was decreased compared with rated condition. It means that the performance of turbine was good condition. So the main reason of low output power was lack of input heat to ammonia cycle.

4.2 Examination of decreasing output power

We have investigated performance of the evaporator, the examination result of evaporator are described on Table 3. Case A is exchanged heat data based on 310L/min of brine, and Case B is based on 388L/min of brine. On case A, overall heat transfer coefficient is about 0.195 kW/m^{2}K, which is only 21% of designed value. It leads to decrease ammonia vapor volume. I this case, input volume of brine is lower that rated brine flow rate. Then we tried to input rated volume of brine to evaporator in order to evaluate an effect of physical volume of heat source. Case B shows the test result under rated flow rate. However, the temperature is about 90 deg.C. The overall heat transfer coefficient is about 0.200, which is same as Case A. The ammonia temperature of separator is about 67 deg.C, which is also same as CaseA.

Table 3 Examination for performance of Evaporator

Items	Unit	Design	Test Results	
			Case A	Case B
Surface Aria "A"	m2	119.2	119.2	119.2
Inlet Temp. (Hot Water) Th1	deg.C	98.0	98.2	90.0
Outlet Temp. (Hot Water) Th2	deg.C	52.3	63.3	65.0
Flow rate (Hot Water) qw	kg/s	5.166	5.166	6.420
Inlet Temp. (NH3 aq) Tc1	deg.C	33.2	29.2	31.0
Outlet Temp. (NH3 aq) Tc2	deg.C	94.5	67.4	67.0
Flow rate (NH3 aq) qnh3	kg/s	1.050	1.050	1.100
⊿T1(=Th1-Tc2)	deg.C	3.50	30.80	23.00
⊿T2(=Th2-Tc1)	deg.C	19.11	34.10	34.00
LMTD ⊿tm ⊿tm=(⊿T1-⊿T2)/ln(⊿T1/⊿T2)	deg.C	9.2	32.4	28.1
Heat exchange rate (Th2-Th1)×qw×4.186	kW	989	755	672
Overall heat transfer coefficient, U	kW/m2･K	0.902	0.195	0.200

Overall heat transfer coefficient is described as equation 1. The decrease of overall heat transfer coefficient is correlated with heat transfer rate (hhw) between heat source and heat transfer plate of evaporator and heat transfer rate (hwf) between ammonia water and heat transfer plate of evaporator. The heat transfer rate is related to mass flux of flow media. Then it is necessary for increasing overall heat transfer coefficient to increase mass flux of heat source and ammonia water.

$$U = \frac{1}{\frac{1}{h_{hw}} + \frac{\delta}{\lambda} + \frac{1}{h_{wf}}} \quad \cdots\cdots\cdots \quad (1)$$

U: Overall heat transfer coefficient (W/m2･K)

hhw: Heat transfer coefficient between hot water and plate of Heat Exchanger (W/m2･K)

hwf: Heat transfer coefficient between NH3aq and plate of Heat Exchanger (W/m2･K)

δ : Thickness of plate (m)

λ : Thermal conductivity (W/m･K)

And another reason of decreasing of exchanged heat transfer is that evidence points to a problem with proper fluid distribution in the boiling ammonia water side of the Evaporator. This “mal-distribution” is caused by the combination of mechanical design factors consisting of 1) poor inlet distribution, 2) vertical up-flow, 3) different fluid densities and 4) low pressure drop. This mal-distribution results in vertical up-flowing portions of the heat exchangers having one or a combination of the following problems – very low flow, stagnant flow, and/or recirculation flow. These portions, therefore, contribute little to zero in heat transfer duty of the exchangers. In other words, the Evaporator’s effective heat transfer surface that actually contributes to boiling duty is less than the surface provided.

There is an appreciable difference in the density of the ammonia-water mixture as the liquid is heated, and as the lighter ammonia component boils off from heavier lean liquid. Consequently, since the existing heat exchangers are “vertical” up-flowing units, there is a "gravity" effect upon the different fluid densities that simply cannot be overcome. This gravity effect will hold down the heavier (leaner) and cooler ammonia water fluids, while buoying up the lighter (richer), hotter and vapor factions. The existing evaporator has 720 individual vertical up-flowing paths. It is inconceivable that each of these hundreds of individual vertical paths would have equal fluid densities throughout its full vertical height, and thus equal flow. Just a "very slight difference" in fluid densities is sufficient in the 2600 mm high Evaporator to cause mal-distribution. The ammonia water pressure drop in the Evaporator is in the order of just 0.9 kPa. This very low pressure drop is not sufficient to overcome even slight density differences among the hundreds of individual vertical flow paths.

The new heat exchanger is significantly different in mechanical design than the existing exchangers. The new unit also resolves the design deficiencies of the existing exchangers that contribute to flow mal-distribution. The new unit has a considerably higher mass flux and fluid velocity. These factors increases the heat transfer coefficient and pressure drop values appreciably. The higher heat transfer coefficient reduces the required surface area while the higher pressure drop assures proper fluid distribution to all the surfaces. Flow direction between the existing and new heat exchanger is described in Figure 8.

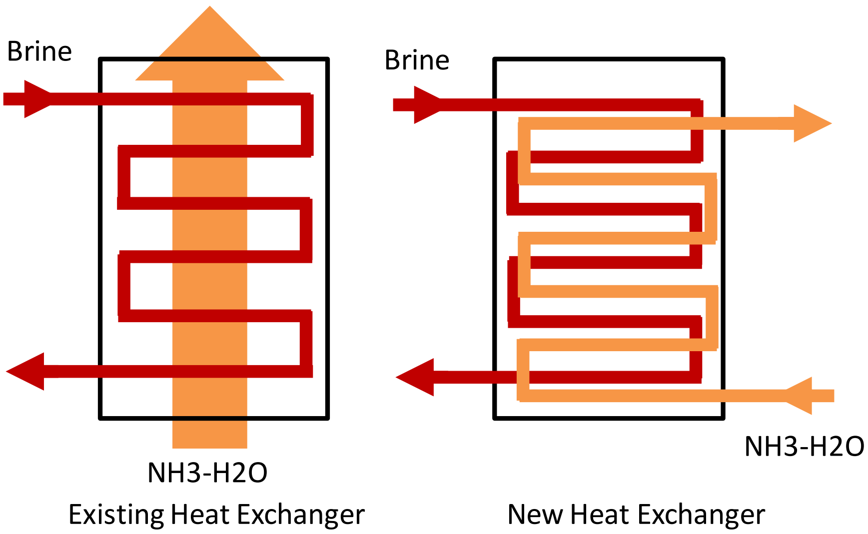

Figure 8 Flow direction between the existing and new heat exchanger

We have conducted heat run testing after the existing evaporator was replaced to new one. The test is measured heat exchanged rate and overall heat transfer coefficient at the condition of 150 – 320L/min of heat source. Test result between hot water flow rate (heat source) and exchanged heat is described in Figure 9 and Table 4. The heat exchanged rate is about 1,115kW and overall heat transfer coefficient is about 1.053W/m^2K. And the result shows that the new heat exchanger has an enough performance based on the design.

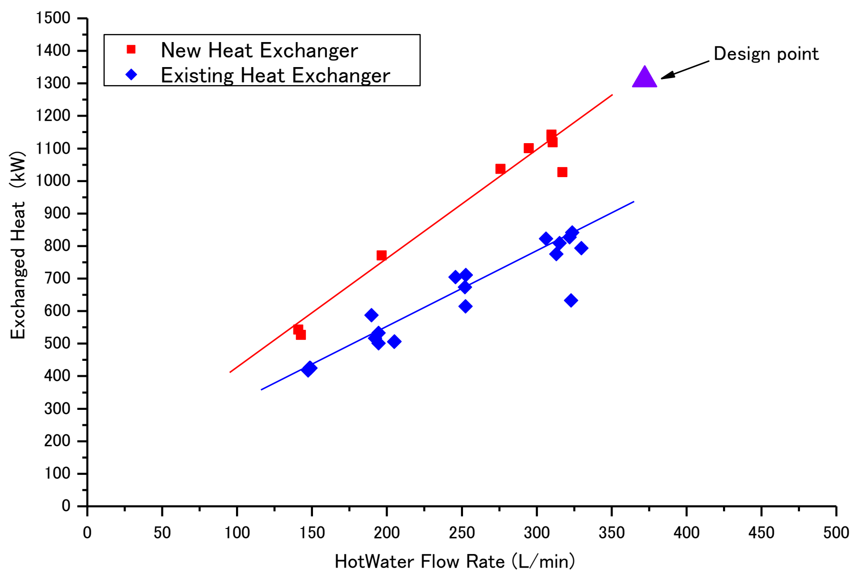

Figure 9 Exchanged Heat respected to Hot water flow rate

Table 4 Examination for performance of New Heat Exchanger

Items	Unit	Design	Existing H.E.	New H.E.
Surface Aria "A"	m2	119.2	119.2	62.7
Inlet Temp. (Hot Water) Th1	deg.C	98.0	98.2	92.6
Outlet Temp. (Hot Water) Th2	deg.C	52.3	63.3	40.6
Flow rate (Hot Water) qw	kg/s	5.166	5.166	5.123
Inlet Temp. (NH3 aq) Tc1	deg.C	33.2	29.2	17.8
Outlet Temp. (NH3 aq) Tc2	deg.C	95.0	67.4	80.5
Flow rate (NH3 aq) qnh3	kg/s	1.050	1.050	0.986
⊿T1(=Th1-Tc2)	deg.C	3.00	30.80	12.10
⊿T2(=Th2-Tc1)	deg.C	19.11	34.10	22.79
LMTD ⊿tm ⊿tm=(⊿T1-⊿T2)/ln(⊿T1/⊿T2)	deg.C	8.7	32.4	16.9
Heat exchange rate (Th2-Th1)×qw×4.186	kW	989	755	1,115
Overall heat transfer coefficient, U	kW/m2･K	0.953	0.195	1.053

4.3 Generation test result

We have conducted generation test with new Evaporator. The results are shown in Figure 10 and Table 5. Table 5 is included previous test result using the existing heat exchanger. At that time, flow rate of heat source is about 80% of rated rate, then ammonia water flow rate is set 1.05 kg/s which is same ratio of rated rate. The temperature of heated ammonia water is about 80.5deg.C and temperature of cooling water is 7.2 deg.C. The generation test was conducted under the condition. Average of output power was about 44.5kW. Output power was increased 4.6 times in comparison with the output power measured with the original Evaporator. In comparison with data using the original Evaporator, the difference of ammonia steam enthalpy at outlet of turbine is about 6.8 kJ/kg, and the difference at inlet of turbine was increased from 1386.4kJ/kg to 1539.4kJ/kg. In case of using new Evaporator, the output power was increased enthalpy drop in the turbine. In future, we have a plan to conduct generation test based on rated heat source and also conducted continuous generation test.

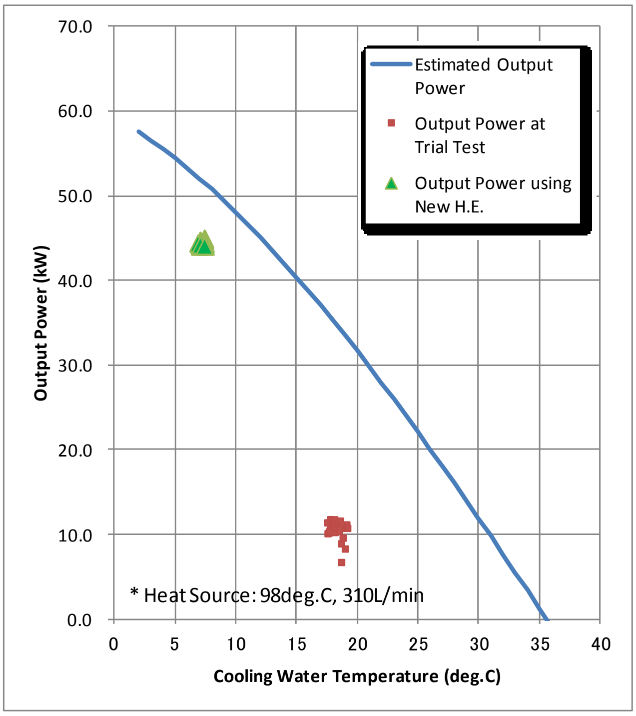

Figure 10 Result of Output power with new heat exchanger

5. CONCLUSION

We have developed micro grid Kalina-cycle® system which utilizes hot spring water for heat source, and conducted the first verification tests at Matsunoyama in Tokamachi City, Niigata Prefecture, Japan. Since we could not use enough brine heat source to achieve the designed performance due to limitations of the hot spring resource, we improved the system to utilize not only brine but also steam from the well. This helps to reduce production rate from hot spring well. We have examined a type of evaporative heat exchanger which is suitable for low temperature hot spring resources. The overall heat transfer coefficient of the original Evaporator (first type) was much lower than designed value. The reason is that a low mass flux of heat source and ammonia water, and mal-distribution in the heat exchanger. We designed block type heat exchanger which is able to increase mass flux and fluid velocity. The heat exchanged rate is about 1,115kW and overall heat transfer coefficient is about 1.053W/m^2K, and the result shows that the new heat exchanger has an enough performance for hot spring resources.

As a result of verification test of Kalina-cycle® system, we confirmed 44.5kW power generation. In this case, inlet temperature of heat source was lower than designed condition (98deg.C). We are trying to increase inlet heat to designed condition and to operate continuously.

Table 5 Result of generation test with new heat exchanger

Items		Unit	New H.E.	Existing H.E.
NH3 Flow	Flow rate	kg/s	1.05	1.00
Separator	Pressure	MPaG	1.35	1.31
	Temperature	deg.C	80.5	66.7
Turbine inlet	NH3 steam flow	kg/s	0.722	0.500
	Temperature	deg.C	80.0	66.0
	Pressure	MPaG	1.35	1.41
Turbine outlet	Temperature	deg.C	54.0	48.0
	Pressure	MPaG	0.41	0.83
Turbine	NH3 steam enthalpy	kJ/kg	1359.2	1352.4
	Heat drop	kJ/kg	80.2	30
	Average Power	kW	44.5	10.9

6. ACKNOLEGEMENT

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REFERENCES

Kalina, A., Leibowitz H: Applying Kalina cycle technology to high enthalpy geothermal resources. GRC Council Transactions,Vol.13, pp.605-611. (1989).

Ministry of the Environment: State of hot spring uses in Japan. (2008). (in Japanese)

Muraoka, H., Sasaki, M., Yanagisawa, N. and Osato, K.: Development of a Small and Low-Temperature Geothermal Power Generation System and Its Marketability in Asia, Proceedings of the 8th Asian Geothermal Symposium, December 9-10, 2008

Phil Welch, Patrick Boyle, Marc Sells, Michelle Giron, Construction and Startup of Low Temperature Geothermal Power Plants, GRC Conference-SanDiego,CA, October 26th , P13 & P19, (2011).