Study of low pressure water binary power generation system for hot spring water

学位名	修士（工学）
学位授与機関	東京海洋大学
学位授与年度	2018
URL	http://id.nii.ac.jp/1342/00001704/

Master's Thesis

STUDY OF LOW PRESSURE WATER BINARY POWER GENERATION SYSTEM FOR HOT SPRING WATER

March 2019

Tokyo University of Marine Science and Technology

Graduate school of Marine Science and Technology

Course of Marine System Engineering

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Nomenclature.

$\vec{c}_1$: Absolute inlet velocity of steam [m/s]

$\vec{c}_{1w}$: Absolute velocity of droplet inlet[m/s]

C_p : constant specific heat of water [kJ/kgK]

D : Turbine Diameter[m]

f : Frequency[Hz]

F : Steam force[N]

h_0 : Specific enthalpy of turbine inlet [kJ/kg]

$h_0{}'$: Specific enthalpy of saturated liquid at turbine inlet pressure [kJ/kg]

$h_0{}''$: Specific enthalpy of saturated steam at turbine inlet pressure [kJ/kg]

h_1: Specific enthalpy of turbine Outlet [kJ/kg]

$h_1{}'$: Specific enthalpy of saturated water at outlet of turbine [kJ/kg]

$h_1{}''$: Specific enthalpy of saturated steam at turbine outlet [kJ/kg]

⊿H : Hot water enthalpy difference at the heat exchanger [kJ/kg]

m_c : working fluid flow rate[kg/s]

m_H : Hot water flow rate[kg/s]

N : Rotational speed [rpm]

P: Output [kW]

R : Steam gas constant [kJ/kgK]

s_0 : Specific entropy of fluid at turbine inlet [kJ/kg]

s_1 : Specific entropy of fluid at turbine outlet [kJ/kg]

$s_1{}'$: Specific entropy of saturated water at turbine outlet pressure [kJ/kg]

$s_1{}''$: Specific entropy of saturated steam at turbine outlet pressure [kJ/kg]

T : absolute temperature of the nozzle inlet steam [K]

t : turbine torque [N.m]

⊿T: Temperature difference between hot water at the inlet and outlet of the evaporator [K]

$\vec{u}$: A side turbine peripheral speed [m/s]

$\overrightarrow{w_1}$: Relative inlet velocity of steam [m/s]

$\overrightarrow{w_{1w}}$: Relative inlet velocity of droplet [m/s]

x_0 : dryness at turbine inlet [-]

x_1 : dryness at turbine outlet [-]

α_1 : absolute angle of steam at turbine inlet [deg]

β_1 : Relative angle of steam at turbine inlet [deg]

η_G : Cycle efficiency [%]

κ : Specific heat ratio of steam [-]

γ : optimization parameter [kJ/kg]

1. Preface

1.1 Introduction and background

Together with the rapid growth of the world population, the world global warming determined the trend of the world energy policy. While providing enough energy is necessary for our daily life, the consideration of the environmental impact of the produced energy should not be neglected. Limited ability to supply the non-renewable energy leads to the energy shortage in developing countries. To satisfy these requirements, the need to increase the production of renewable energy has become crucial.

Recently the geothermal is one of the most desirable renewable energy sources. As shown in fig.1 Japan has the world's third largest potential of geothermal resources (23,400MW), however has only 520MW (2.2%) to have been developed. The year production in 2010 was about 500 MW, which represents half of a large nuclear power plants power production. The Ministry of the Environment has estimated that this fraction can be raised to 23.5. There are over 3,000 hot springs in Japan if exploited can generate 8,500 MW of electricity.

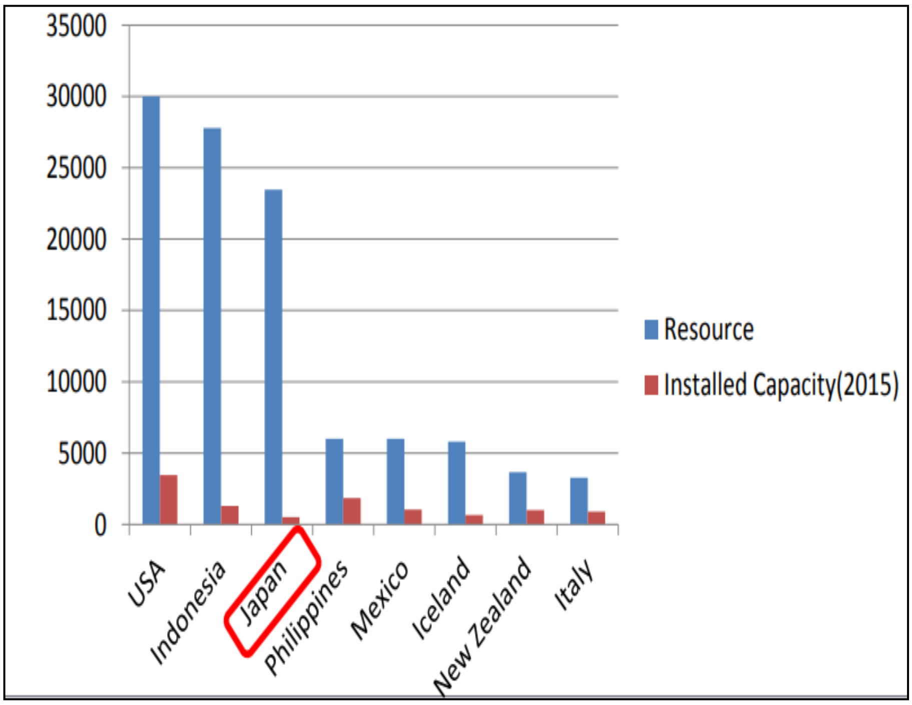

Fig.1 Geothermal energy potential.

Geothermal power generation plant can be divided into two main different types. Namely the flash steam geothermal power plant and the binary geothermal power plant as show in fig.2 below. Each type of geothermal power plant have a specific characteristic and they differ from each other by the heat cycle system formation and flow.

In the flash steam geothermal plant, the brine also known as the heat source which is a mixture of steam and liquid is taped from a production well and send to a steam-liquid separator where steam is separated from the rest of the mixture. The steam then expand through a turbine and power is generated.

There is only one heat cycle which make the particularity of such a system.

On the other hand, in the geothermal binary power generation plant there are two working fluid and or 2 heats cycle. The brine or heat source is taped from the production well and pumped to a heat exchanger where it supply heat to a secondary fluid known as the working fluid. In most cases the working fluid is an organic fluid which has a low boiling point and thus can evaporate even at a low temperature. The working fluid goes throw a turbine where it expanded and power is generated.

Our research will be based on the binary geothermal power generation system.

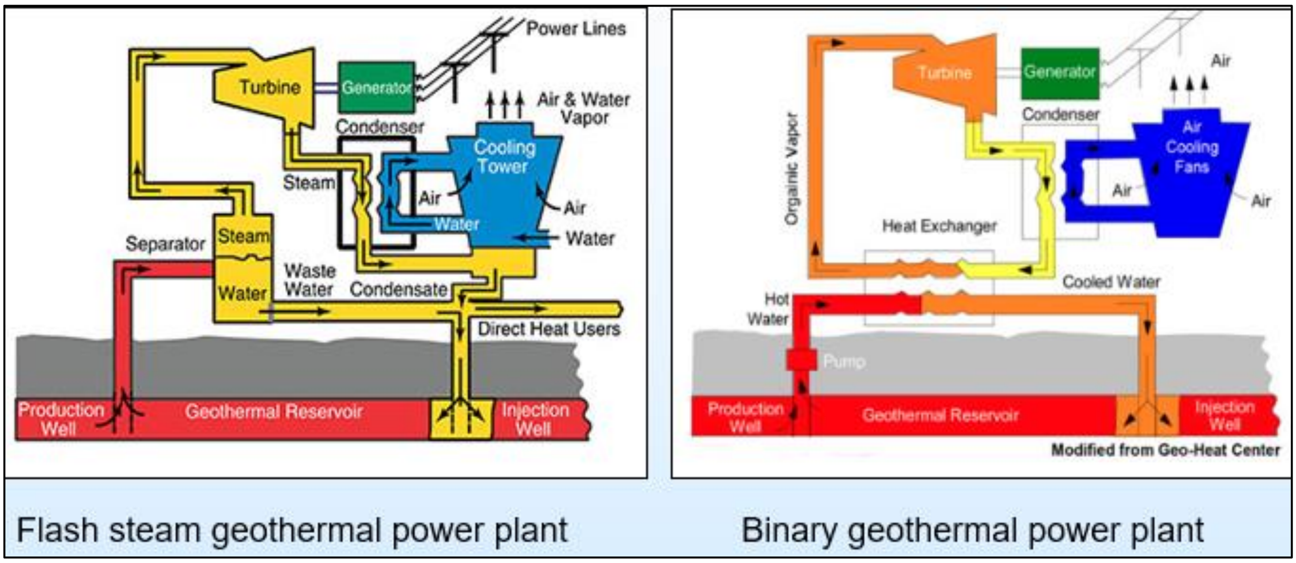

Fig2. Type of geothermal power plant

In general, based on the design conception, the binary generation system can be divided into 2 types which are the ways to recover the energy from the low temperature thermal sources, namely the two-phase turbine and steam-driven turbine. In the case of a steam driven turbine, only the steam drive the turbine while in two-phase turbine, a mixture of liquid and steam drive the turbine Fig.3 is a representation of both systems T-H diagram at the evaporator. Assuming the specific heat capacity of hot spring water is constant, the temperature changes linearly. For this reason, on a steam-driven turbine, there exists a point where the difference of temperature between the heat source and the working fluid is the lowest called the pinch. The pinch point imposes a constraint on the heat recovering system, therefore, the outlet temperature of the hot water remains higher as observed in the T-H representation. Generally, this pinch point coincides with the boiling point of the working fluid.

However, it is demonstrated that when the two-phase flow turbine is adopted, the constraint does not exist and thus it becomes possible to recover a high amount of energy or to obtain a lower outlet temperature of the heat sources. A method of recovering the heat or lowering the outlet temperature is analyzed in this paper.

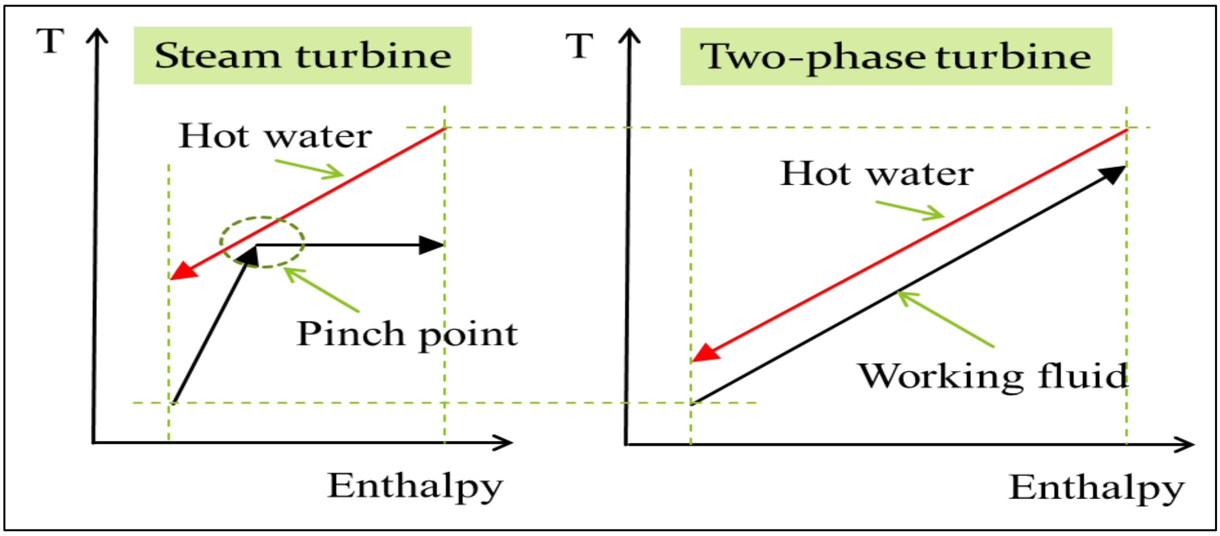

Fig.3 pinch point constraint

1.2 Research purpose

In most binary power generation system, it has been observed the low boiling point organic substance such as ammonia, HCFC123, n-Pentane and PF5050 are used as working fluid, these cycle are called ORC or organic ranking cycle. It has been proven that most of the organic working fluid used has a high global warming potential or GWP and are currently facing a ban. Therefore to help address this concern Water was adopted as working fluid. As an advantage of water binary power generation system, although the thermodynamic performance might be low, the ability to meet climate protection initiatives can be mentioned. There is no ozone depletion potential (ODP) and there is no greenhouse gas emissions known as GHG. The system can be operated safely and the cost of installation, maintenance and operation are very low compare to complex superheated steam power generation plant.

It was found that in Steam driven turbine, the pinch point imposes a constraint on the heat recovering system therefore the outlet temperature of the hot spring water which is the heat source remains higher during the heat recovering process.

A hot water binary power generation system that produce 10kW at the generating end output will be developed and analyzed by applying the heat from the hot spring water with temperature below 100 deg. Celsius. therefore the low temperature heat energy from hot spring water will be converted into mechanical work and the hot spring water Temperature will be lowered to 40℃ which is generally consider as the average bath temperature in japan.

1.3 Ranking cycle

Shown in fig.4 is the basic diagram of a ranking cycle. The Rankine cycle is a model used to predict the performance of steam turbine systems. The Rankine cycle closely describes the process by which steam-operated heat engines commonly found in thermal power generation plants generate power.

Power depends on the temperature difference between a heat source and a cold source. The higher the difference, the more mechanical power can be efficiently extracted out of heat energy, as per Carnot's theorem.The heat sources used in these power plants are usually nuclear fission or the combustion of fossil fuels such as coal, natural gas, and oil, or concentrated solar power. The higher the temperature, the better the system performance. The efficiency of the Rankine cycle is limited by the high heat of vaporization of the working fluid.

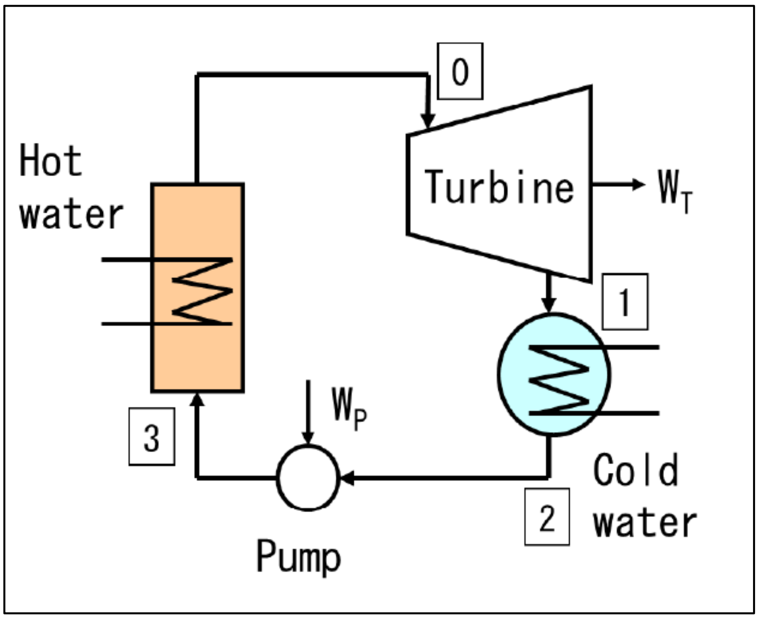

Fig.4 Ranking cycle of a binary cycle

1.3.1 Hot water binary power plant generation system T-S diagram

Shown in fig.5 is the T-S diagram of the hot spring water binary power plant at turbine inlet pressure of 30.2kPa.the x-axis represent the entropy while y-axis represent the temperature variation. The incipient point coincide with the boiling point at the evaporator of the working fluid. The system output is delimitated by the area 012340. Where 4 represent the incipient point.

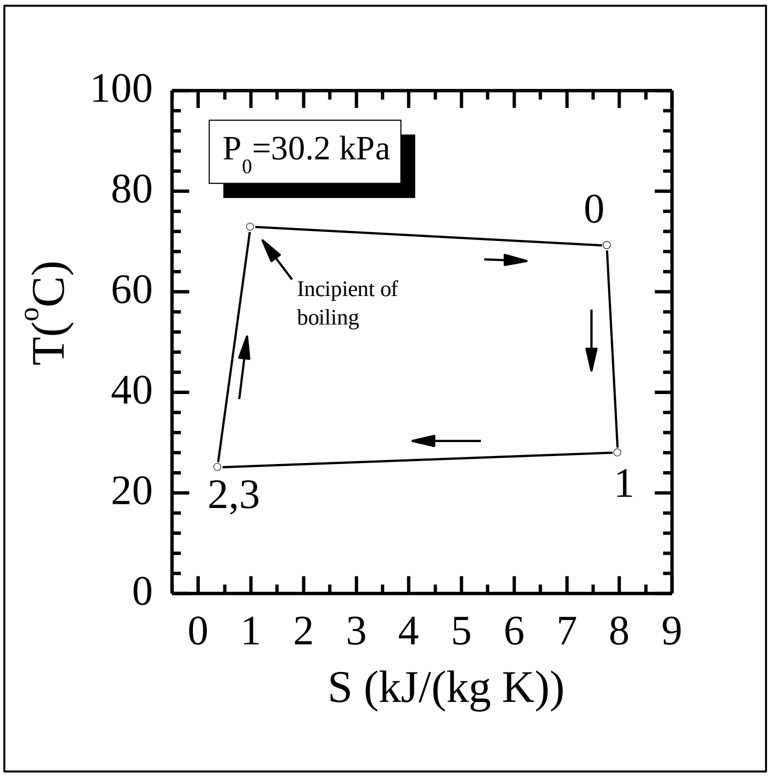

Fig.5 T-S diagram of binary cycle

1.3.2 Hot water binary power plant generation system T-H diagram

Shown in fig.6 is the T-H diagram of a water binary power generation system at the turbine inlet pressure of 30.2kPa. The red dot line represent the hot water line or heat source line while the blue dot line represent the cooling water line or the cold source line. The pinch point can be defined as the point where the hot water temperature and the working fluid temperature difference reach it minimum or smallest value.

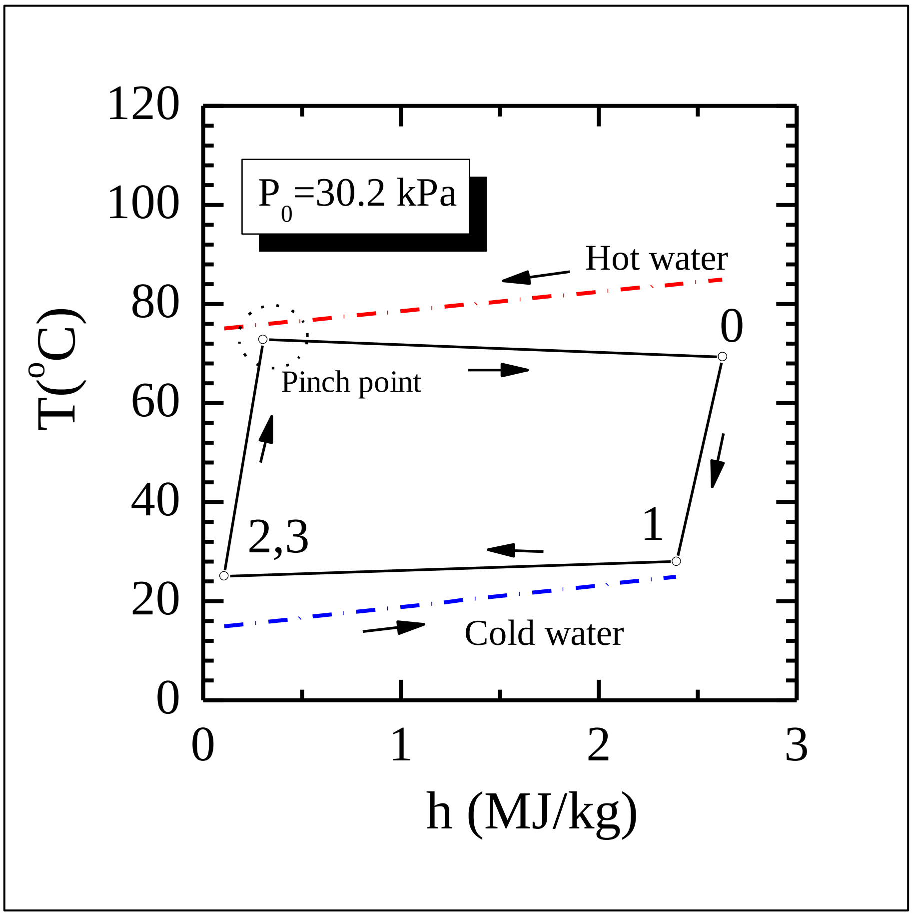

Fig.6 T-H diagram of a water binary power plant.

2. Hot water binary plant theoretical analysis

2.1 Purpose

The steam table and design specification were used to determine the performance of a water binary power plant. Shown in fig7. Is the process flow diagram of the hot water binary system which can be divided into 4 different stages as describe in the table1 below. The number 0 in the diagram represents the turbine inlet, the number 1 represent the turbine outlet. The number 2 represents the condenser outlet while the number 3 represents the evaporator inlet.

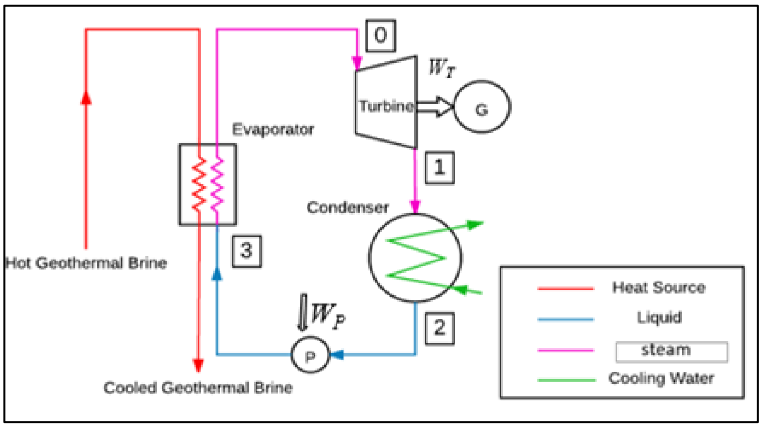

Fig7. Process flow diagram

Table1. Process flow diagram description

Stage	Location	Description
stage 1: 3 to 0	**Evaporator**	Heat is supplied to the working fluid. Liquid ➡ vapor. the temperate rises and a vacuum is maintain in the steam line
stage 2: 0 to 1	**Turbine**	The steam expands as the temperature and pressure drop and mechanical work is being produced. dry saturated steam ➡ wet steam
stage 3: 1 to 2	**Condenser**	Heat taken away and temperature drop. Wet steam ➡ Saturated liquid steam
stage 4: 2 to 3	**Pump**	Liquid low pressure and temperature to high pressure and low temperature, pump work received

Table2 represent the design specification of the system. Theoretical analysis will be carried out base on the design specification as described in the table.

Table2. Design specification.

Output [kW]	10
Hot water inlet temperature [°C]	85
Cooling water inlet temperature in condenser [°C]	15
Cooling water outlet temperature in condenser [°C]	25
Pinch point temperature difference [K]	3
Hot water pressure loss in evaporator [kPa]	5
Cooling water pressure loss in condenser [kPa]	0.6
Turbine efficiency [-]	0.8
Pump efficiency [-]	0.4
Machine efficiency [-]	0.92

2-2 Method

Based on the design specification, the steam table was used to determine the performance and efficiency of a water binary power system. Parameters such as turbine inlet and outlet temperature, pressure and turbine efficiency were used in the calculation. The turbine efficiency was kept constant during the entire calculation and analysis at 0.8 while other parameters where changed according to a specific and defined purpose in order to obtain the system must accurate characteristics.

2.3 Results analysis

2.3.1 Turbine Inlet steam pressure effect

Shown in fig.8 is the binary T-H diagram at turbine inlet pressure of 30.2kPa. The temperature variation of hot spring water and cooling water are included. The outlet temperature of hot water is limited at 75°C due to the pinch point constraint.

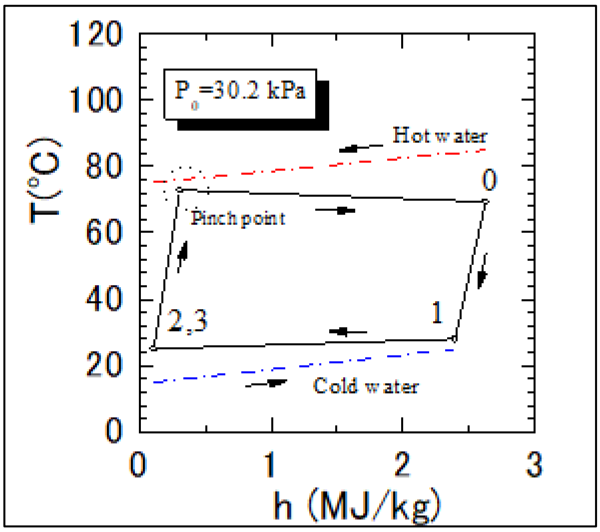

Fig.8 T-h diagram with pinch point.

Considering 1 kg of working fluid, since the system efficiency is the ratio of the generator output to the heat energy received, it can be obtained from the following equation (1)

$$\eta = \frac{W_T - W_P}{h_0 - h_3} \tag{1}$$

Where, W_T is the turbine power produced, W_P the pump received power, h_0-h_3 is the enthalpy difference at the evaporator. The working fluid flow rate m_c necessary to generate [10kw] of power is given by the following equation (2).

$$m_c = \frac{10}{W_T - W_p} \tag{2}$$

Assuming that the change in the turbine is isentropic change, the entropy of the turbine inlet and the entrance of the turbine are equal, and since the working fluid at the turbine outlet is wet steam

$$s_0 = s_1 = s_1' + x_1(s_1'' - s_1') \tag{3}$$

Here,s_1', $s_1''$are the specific entropy of saturated water and saturated steam at the turbine outlet respectively, and x_1 is the steam dryness .

From the equation (3), the dryness x_1 at the exit of the turbine in case of isentropic change can be obtained by the following equation.

$$x_1 = \frac{s_0 - s_1'}{s_1'' - s_1'} \tag{4}$$

Thus, the enthalpy of the turbine exit at isentropic change can be obtained by the following equation.

$$h_1 = h_1' + x_1(h_1'' - h_1') \text{ [kJ/kg]} \tag{5}$$

Where h_1', h_1'' are the specific enthalpy of saturated and saturated steam at the turbine outlet.

The turbine work produced W_T is enthalpy difference between turbine inlet and outlet,

$$\Delta\mathrm{h} = h_0 - h_1 \tag{6}$$

By considering the turbine efficiency η_T,the actual worked can be given as

$$\Delta h_T = \eta_T(h_0 - h_1) = \eta_T\Delta\mathrm{h} \tag{7}$$

Base on the system heat balance between the heat source and the working fluid the following equation can be obtain.

$$m_H = \frac{m_c(h_0 - h_3)}{\Delta h_H} \tag{8}$$

Analytical result shows that the hot spring water outlet temperature dropped as the turbine inlet pressure decreases. Fig.9 is a representation of the system T-H diagram, when the turbine inlet pressure is maintain at a pressure of 20kPa and 30.2kPa the pressure drops, the pinch point temperature decreases causing the hot spring water temperature slope to be steep, as a result its became possible to obtain a low outlet temperature of hot spring water.

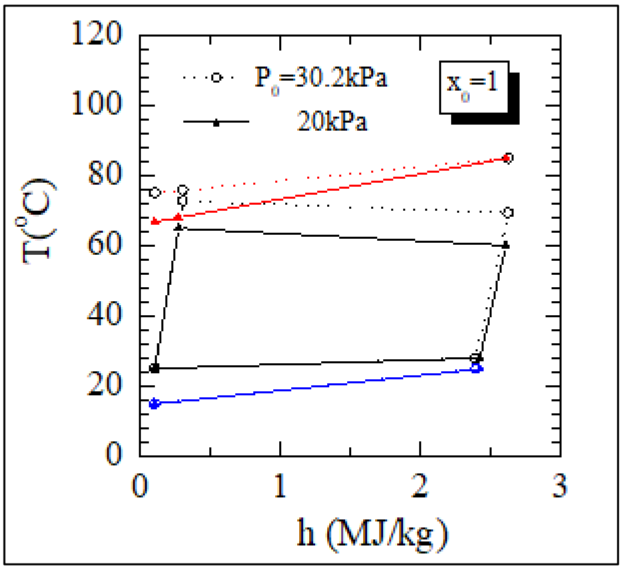

Fig.9 T-h diagram at different pressure

The pressure variation does not only affect the hot spring water temperature, calculation base on equation (2) showed the hot spring water flow rate changes as the pressure drops. Fig 10 shows the effect of the turbine inlet pressure on the system. As the turbine inlet pressure drops, the hot water outlet temperature also decreases monotonously. However, when the turbine inlet pressure drops, the hot water flow rate also decreases and reach a minimum at an inlet pressure of 15kPa. It can be predicted that for the area below the minimum pressure value, there is a need for a high amount of energy for the system to perform accurately as the system efficiency is too low.

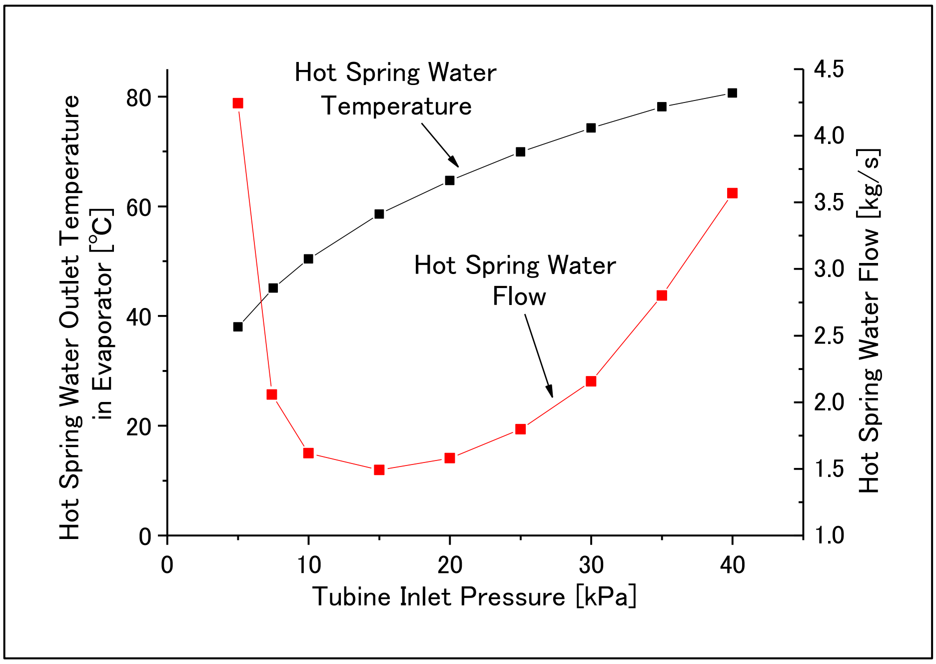

Fig10. Pressure variation effect on flow rate and temperature

Given that a high amount of energy is required when the turbine inlet pressure is low, analytical result base on the power generated per kilogram of hot water was carried out. From Fig.11 it is shown that the turbine inlet pressure dropped lowered the efficiency. As proven previously, the hot spring water flow rate m_H reach a minimum at 15 kPa. Thus, a high amount of energy is required. For this reason, the power generated per 1 kg of hot spring water *Gn* reach a peaks at a pressure of 15 kPa. It can be concluded that at this point most of the heat energy held by the hot spring water is totally converted into power.

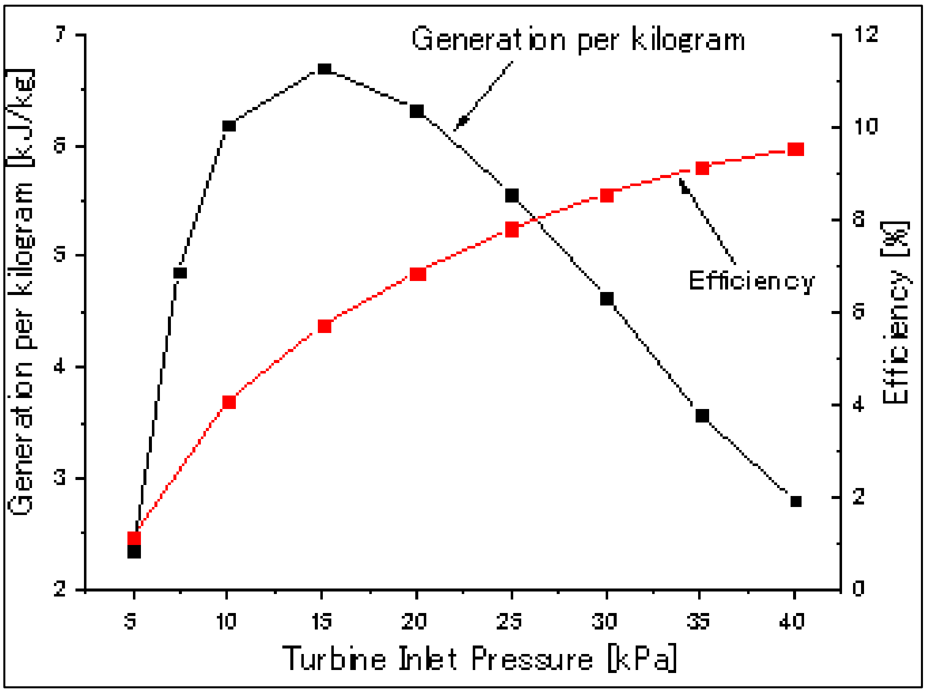

Fig.11 Pressure variation effect on efficiency and generation

Base on fig.12 the flow rate of working fluid increases with decreasing steam pressure. This can be explained by the fact that the enthalpy difference of the turbine inlet and outlet of the working fluid decreases as the steam pressure decreases and a high amount of working fluid flow rate is necessary to obtain the power generation amount of 10 kW.

In addition, the hot spring water flow rate decreases as the steam pressure decreases and reach a minimum at steam pressure P_0 = 15 kPa. It can be predicted that when the pressure falls within the range of 40 kPa to 15 kPa, the outlet temperature of the working fluid in the evaporator decreases therefore, the inlet-outlet temperature difference of the working fluid also decreases.

However, on the hot spring water side the outlet temperature rises and the inlet-outlet temperature difference Increases. Consequently a less amount of hot spring water flow rate is required. The hot spring water flow rate further increased in the low-pressure region below 15 kPa. This can be explained by the fact that the turbine inlet-outlet enthalpy difference of the working fluid becomes considerably small and the necessary flow rate of the working fluid to drive the turbine rapidly increases and the flow rate of the working fluid passing through the evaporator also increases rapidly. It became then necessary to further increase the hot spring water flow regardless of the enthalpy difference of the working fluid side and hot spring water side of the evaporator.

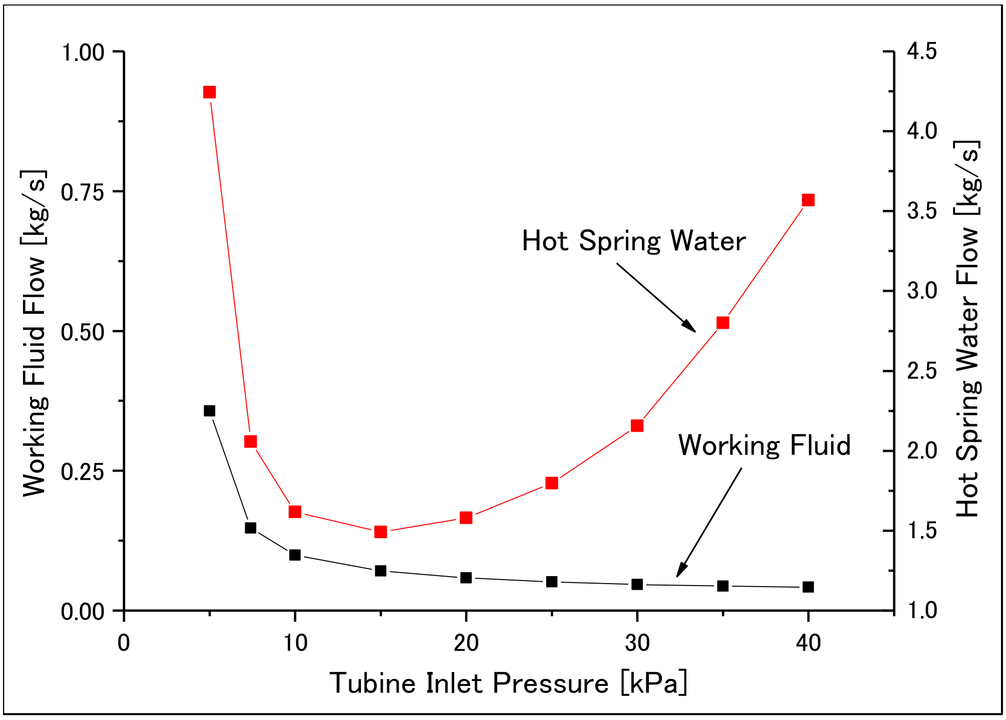

Fig.12 Turbine inlet pressure effect on flow rate.

2.3.2 Turbine Inlet steam quality effect

It has also been proven that another way of lowering the turbine outlet temperature is by decreasing the steam quality (steam dryness). In this case, the pinch point remained fixed while the enthalpy difference of the evaporator is reduced, in order word, the gradient of hot spring water per enthalpy increases and thus lowered the outlet temperature.

Since the pinch point constraint does not exist when the quality is around 0, it become possible to lower the hot spring water temperature up to a temperature equal to the sum of the pinch point temperature difference (3°C) and the cooling water inlet temperature.

It was shown that when the proportion of liquid in the steam-liquid 2 phase mixture at the inlet of the turbine increases, the turbine is not only driven by steam but also by water. In this case, it is assumed that the turbine inlet flow velocity decreases as liquid proportion increases and thus the rotation speed also decreases.

In addition, it can be predicted that the turbine efficiency will also decreases due to an increase in the proportion of liquid in steam-liquid 2 phase mixture. However, this fact had not been taken into consideration when carrying out analytical result as the turbine efficiency was kept constant.

Fig.13 shows a comparison between turbine inlet quality $x_0 = 1$ and $x_0 = 0.4$. It can be observe that by decreasing the quality, the hot spring water temperature gradient becomes steep and thus the outlet temperature can be lowered.

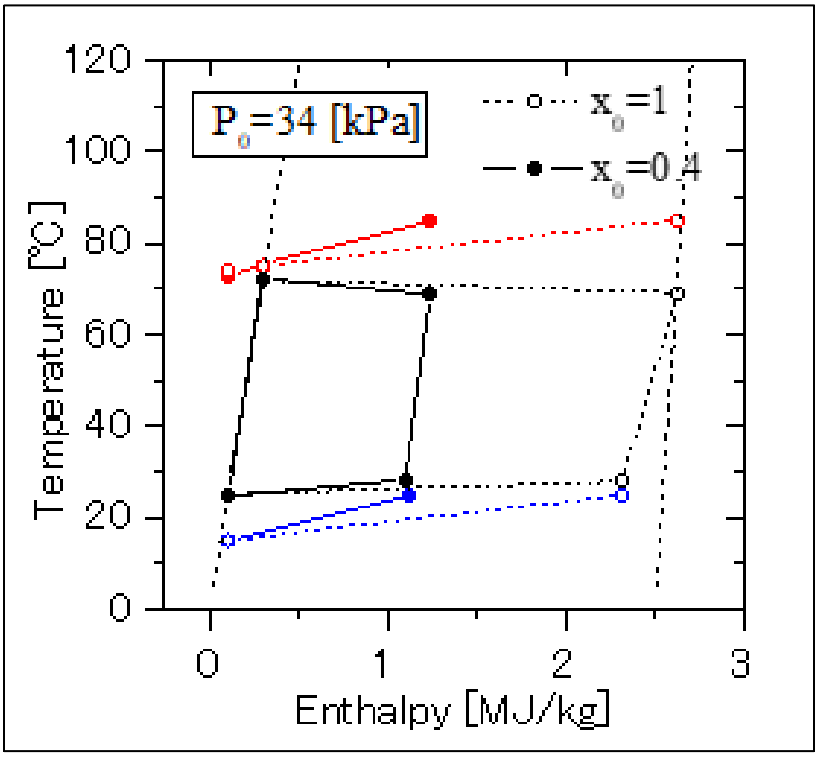

Fig.13 Effect of the steam dryness

As shown in fig.14, when the quality decreases, the hot spring water outlet temperature T_{out} decreases and the flow rate m_H slowly decreases. The hot spring water outlet temperature and flow rate rapidly decreases when the quality is around 0.2. When the quality is 0, the pinch point constraint does not exist, under this condition, the flow m_H slowly rises.

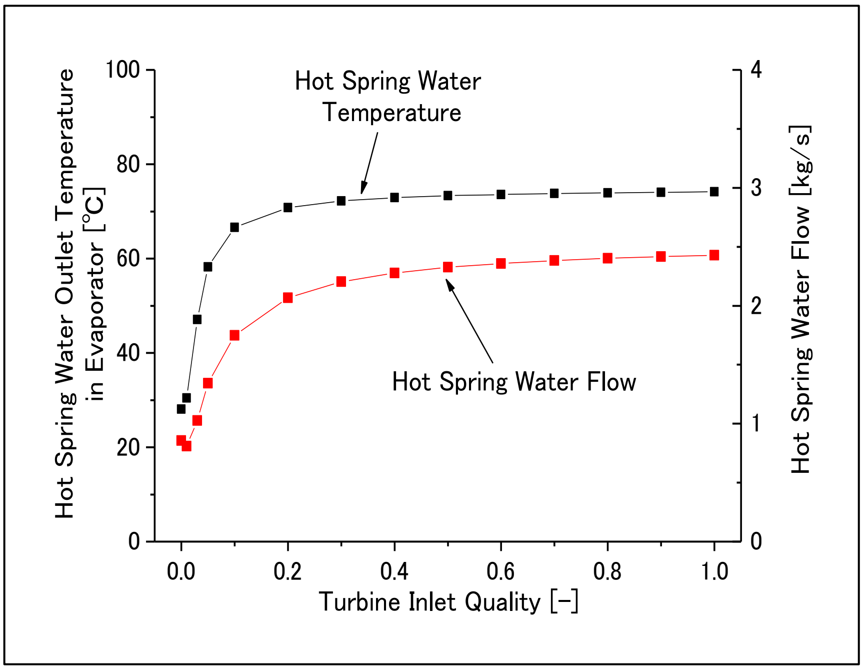

Fig.14 Turbine inlet quality effect on flow and temperature

Shown in fig.15 is the effect of turbine inlet quality on the generation *Gn* and cycle efficiency. As can be observed, when the steam dryness is less than 0.2 the efficiency falls while the power generated per kg of hot water rose. When the quality decreases just below 0.2 the efficiency rapidly decreases however since the decrease in hot spring water outlet temperature is larger, the hot spring water flow rate m_H decreases without reaching the minimum value. For this reason, the amount of electricity generation *Gn* per 1 kg of hot spring water increases and reaches a peak just before the quality becomes zero. This peak is caused by the non-existence of the pinch point constraint which then allow the hot water flow rate m_H to slightly rise. As the steam quality decreases, the efficiency and power generated per kilogram of hot water remain constant.

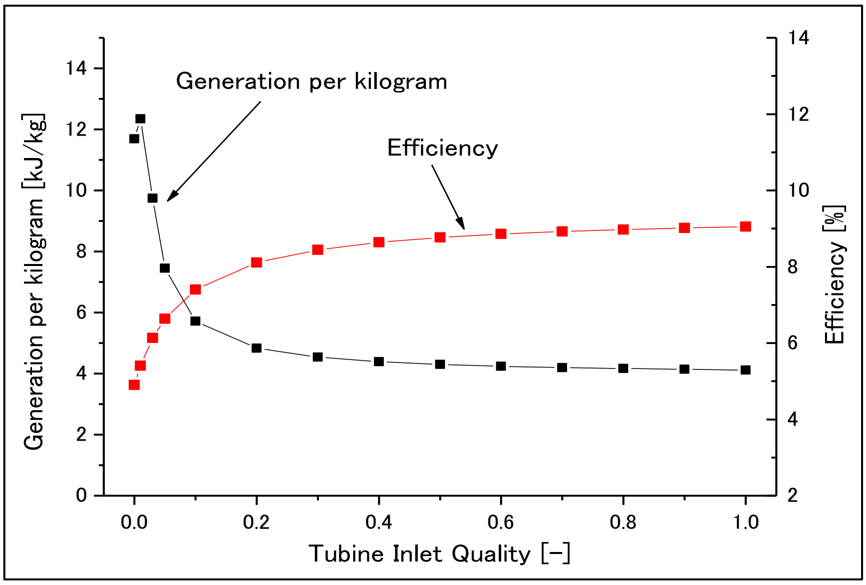

Fig.15 Turbine inlet steam quality effect on efficiency and generation.

From the above analysis decreasing the steam pressure or quality of the turbine inlet steam, the system efficiency decreases, however, the amount of power generation per kg of hot spring water increases and reach its maximum value. Therefore by considering the limited amount of hot spring water, it is necessary to pay attention to the amount of electricity generated per kg of hot spring water rather than the conventional efficiency. However, since the turbine efficiency is kept constant in this study, it has to be taken into consideration in future experiment for a more accurate pressure-efficiency and quality-efficiency relation definition.

In the conventional cycle efficiency-oriented method, since it is not possible to lower the hot spring water temperature, it was necessary to cool the hot spring water after the power generation for bathing or secondary usage. The method shown in this report is convenient for effectively using hot spring water for secondary usage.

3. Hot water binary plant experimental analysis

3.1 Low pressure hot water binary plant experiment

3.1.1 Purpose

The hot water binary power generation system will be run and operated under certain condition and load in order to determine the system performance and characteristics. Shown in fig.16 is the system general process flow diagram representation with both heat cycle and major components. Trial or test of the generation system will be carried out in order to reach the system steady state condition.

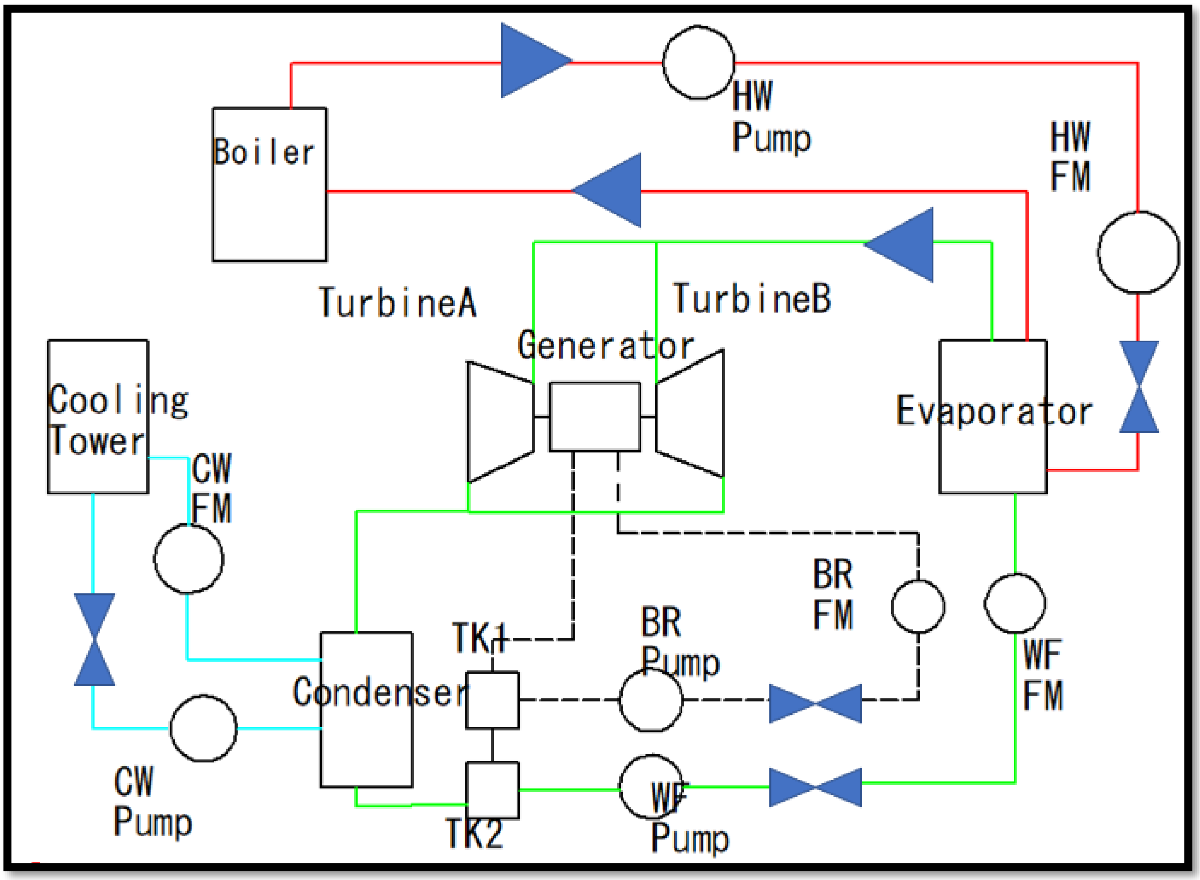

Fig.16 General Process flow diagram

3.1.2 Experiment equipment

Fig.17 represent the experiment general arrangement showing the main components and their location on the site.

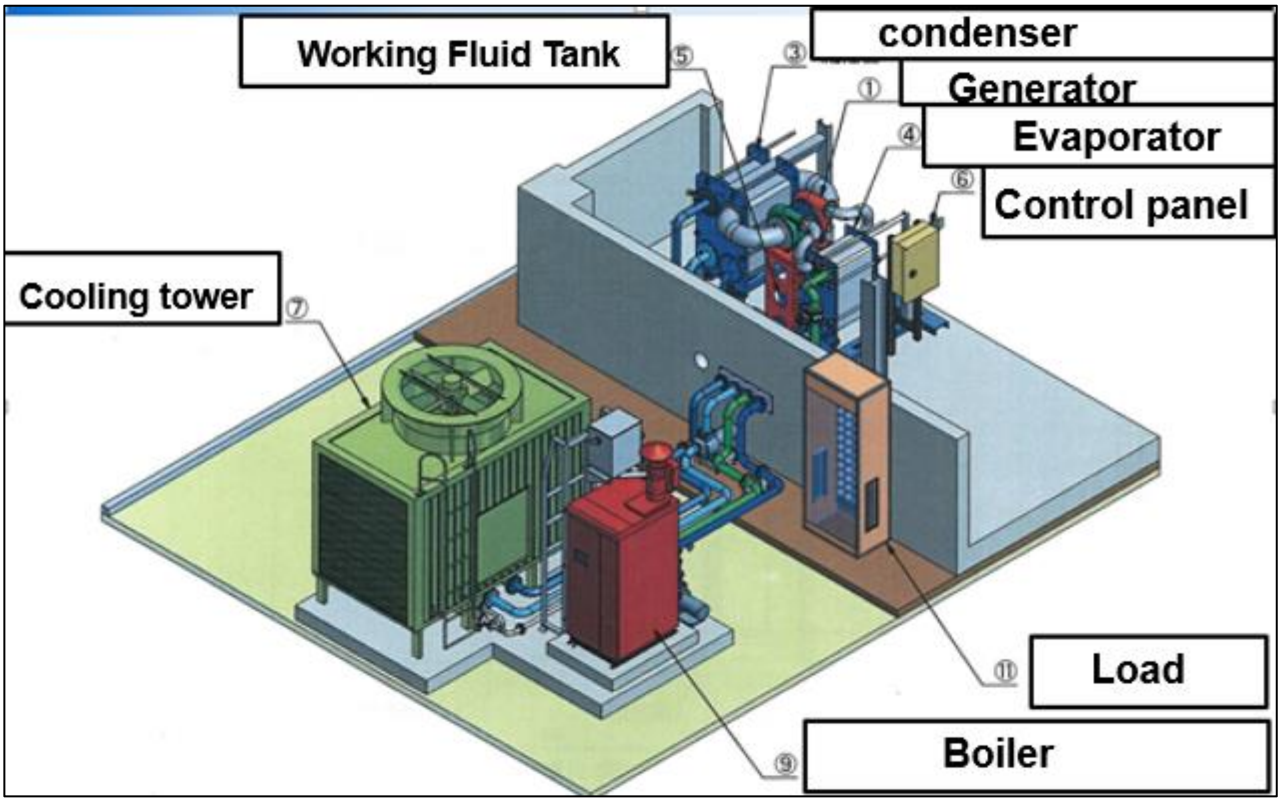

Fig.17 Experimental equipment general arrangement.

Fig.18 turbine blade

As shown in fig.18 the turbine used is the radial turbine. A turbine in which the flow of the working fluid is radial to the shaft. This flow is smoothly orientated perpendicular to the rotation axis as a result, there is less mechanical stress (and less thermal stress, in case of hot working fluids) which enables a radial turbine to be simpler, more robust, and more efficient (in a similar power range) when compared to axial turbines.

Fig.19 Turbine blade and nozzle

Fig.20 Electricity load

Shown in fig.20 is the electric load. The load was set up and program in 4 different blocks. Each block having 3 rows of light bulb.

Fig21. Cooling tower

Fig.22 Boiler

The main features of this experimental apparatus will be described

Table.2 Highlights table of boiler

Hot water boiler	
1.description	
Type	Reflux hot water heater
Maker	Miura Kogyo Co., Ltd.
Model number	UT－200H(outdoor specification)
2.specification	
Heat output	233 kW
Maximum head pressure	0.98 MPa(maximum)
Burner model	forced pushing ventilation source mixed fuel system
Used fuel oil	City gas (13 A)
Usable temperature	150°C(maximum)

Table.3 Highlights table of evaporator

Evaporator	
1 .description	
Type	Type Plate Heat Exchanger
maker	Manufacturer Alpha Laval Co., Ltd
Model	M10－BFM
2. Specification	
Maximum flow rate	2.3 m^3/min
Maximum working pressure 1.0 MPa	1.0 MPa
Operating temperature	-25℃～140℃
Adaptation standard ISO	ISO

Table.4 Highlights table of working fluid pump

Working fluid pump	
1.description	
type	small size magnet pump
maker	Iwaki Co.,Ltd
model	MD－70RZ
2.specification	
Working fluid flow	40 L/min (Max)
Total head	14.3 m

Table.5 Highlights table of lubricating oil pump

Lubricating Working fluid pump	
1.description	
model	Small magnet pump
manufacturer	Inc. Iwaki
Model number	MD－70RZ
2.Specification	
Working fluid flow	40 L/min(maximum)
maximum head	14.3 m

Table.6 Highlights table of turbine

Turbine (Twin entry turbine)	
1. Description	
Model	Twin entry radial Turbine
Manufacturer	Inc. ARCHIVE WORKS
2.Specification	
Rotor blade outside diameter	244 mm
Design rotational speed	36000 rpm
Working fluid flow rate	20 L/min (Maximum)
Maximum pressure	1 MPa
Maximum Temperature	150℃

Table.7 Highlights table of generator

Generator	
1.description	
model	concentrated winding permanent magnet synchronous generator
manufacturer	Inc. ARCHIVE WORKS
2.Specification	
Rated Output	10 kW
Rated speed	36000 rpm
Type of power supply	3 phases
Rated voltage	200 V
Rated current	29 A
Heat-resistant class H of insulator	H Type
Insulation class	H type
Running time continuous	continuous

Table.8 Highlights table of condenser

Condenser	
1.Description	
Model	Plate type heat exchanger
Manufacturer	De Laval
Model number	TS20－MFM
2.Specification	
Maximum Flow rate	10 m^3/min
Maximum pressure	2.1 MPa
Operating temperature	-25℃～140℃
Adaptation standard	ISO

Table.9 Highlights table of hot water pump

Hot water pump	
1.Description	
type	Centrifugal pump
manufacturer	Co., Ltd. Terada Pump Manufactory Co., Ltd.
Model number	CMP6－52.2R
2.specification	
Flow rate	800 L/min(maximum)
Total head height	14 m

Table.10 Highlights table of flow-control valve

Medium circulation flow control valve	
1.Description	
Type	Floating ball valve
Manufacturer	Co., Ltd. Kitz
Model type	RDH124
2.Specification	
Maximum pressure	2.1 MPa

Table.11 Highlights table of cooling water pump

Cold water pump	
1.description	
type	Centrifugal Pump
Manufacturer	Co., Ltd. Terada Pump Manufactory Co., Ltd
Type model	CMP6－52.2R
2.Specification	
Flow rate	800 L/min(Maximum)
Total head height	14 m

Table.12 Highlights table of pressure sensor

Pressure sensor	
1.description	
Type	Absolute Pressure Transmitter
Maker	Huba Control
Model Type	680 series
2.Specification	
Maximum pressure	25 bar
Maximum temperature	150℃

Table.13 Highlights table of pressure sensor

Pressure sensor	
1.Description	
Type	High precision electronic pressure sensor with digital display
Manufacturer	convum
Model type	MPS－33 series
2.Specification	
Maximum pressure	0.8 MPa

Table.14 Highlights table of temperature sensor

Temperature sensor	
1.description	
model	Thermistor
Manufacturer	Fuji Electric Co., Ltd
Type model	FTNA1HE3－A11Y
2.specification	
Measurement range	0～150℃

Table.15 Highlights table of flow meter

Hot water flow meter cold water flow meter	
1.Description	
Type	Electromagnetic non-wetted type 2 wire type electromagnetic type flow rate
	Wetted part material: alumina ceramic, SUS 316 L, FKM (standard · explosion proof type)
Manufacturer	KEYENCE
Type model	FD－UH80H
2.Specification	
Maximum pressure	1 MPa
Maximum temperature	100℃
Measurement range	3015.9 L/min (Maximum)
Flange standard	JIS10K
Response time	0.5 s　minimum

Table.16 Highlights table of measurement control device

Measurement control device	
1.Description	
Type	Cloud data retractable remote control type
	Control computer: Aldino
	Analog / digital method
Name	Name Water binary generator Operation measurement control device
Manufacturer	Inc. ARCHIVE WORKS
2.Specification	
capacity	4 Mb, main computer · 500 Mb
Temperature measurement	thermistor · SUS tube enclosed
Pressure measurement	
Power	power, voltage, current measurement
Rotational speed	rotation sensor
Power absorption	three-phase 200 V · 200 W bulbs · Number switching type
3. Control method	
generated power control l	Remote switching control of the number of light bulbs
Rotation speed control	pump rotation speed control
	Generated power (number of bulbs) control combined use

Table.17 Highlights table of data communication unit

Data trans device	

1.Description	
Type	Wireless LAN system
name	IoT_M & C Remote Monitoring Device
manufacturer	Inc. ARCHIVE WORKS
2.Specification	
configuration	LAN + main control computer
speed	communication speed ･1 control / set it to about 5 seconds

Shown in fig.23 is the system test section side view. The evaporator and the condenser are represented in blue. The twin turbine is located in between them. Two working fluid tanks are used to carry out the experiments accurately.

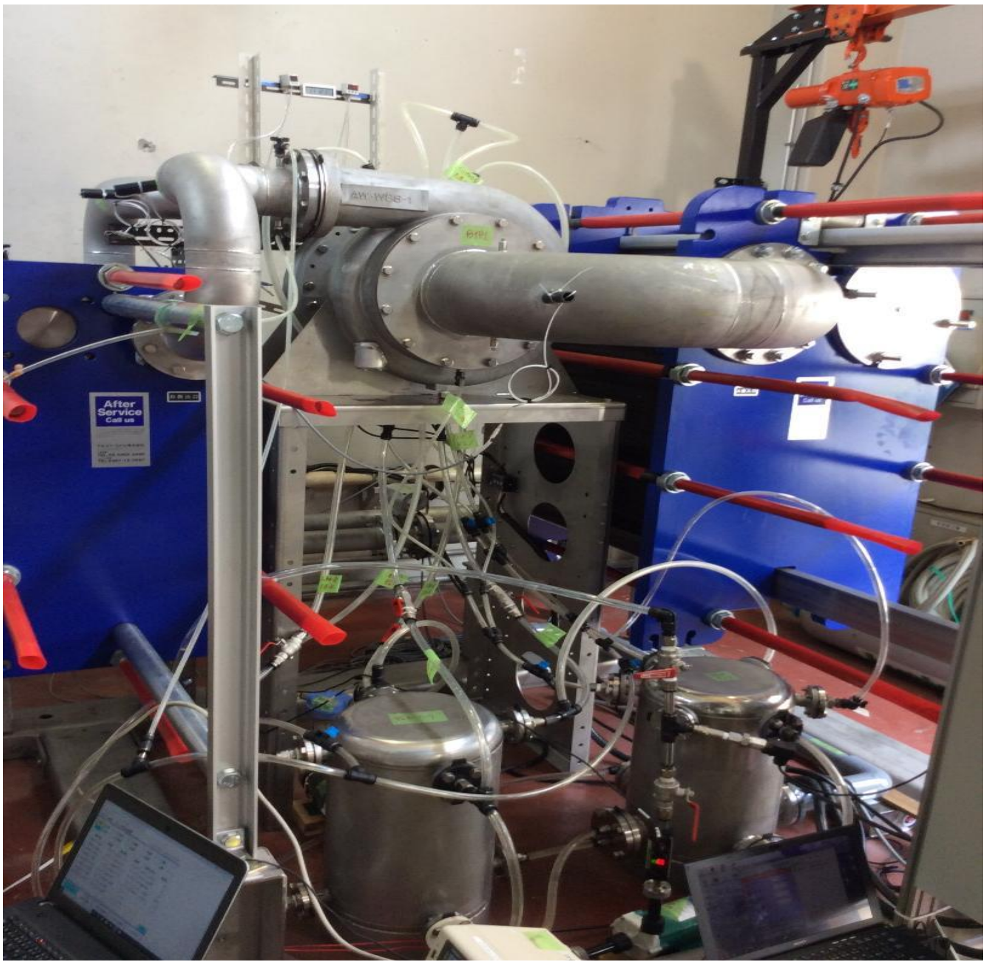

Fig.23 Test section side view

Fig.24 test section top view

3.1.3 Method

The aim of the current experiment is to acquire the data of each element that constitute the hot water binary power system.

Experiments were carried out by the following procedure.

① First of all a vacuum is maintain within the working fluid line by the use a vacuum pump.

② the boiler is started and hot water is pump to the evaporator where it supply heat to the working fluid.

③ Next, working fluid pump is run and the working fluid is pump to evaporator where it receive heat and evaporate even at a low temperature as a vacuum is maintain within the working fluid line though out the entire process. The flow rate is determined by the pump output and unfortunately could not be adjusted automatically due to the system conception. When needed the flow rate is changed manually using flow rate adjustment tools.

④Load is applied to the system to generate electricity. Load is divided into four stages.

3.2 Result analysis

3.2.1 Data frame description and cleanup process

Shown in table18. Is the summery of the dataset. Couple of experiment were carry-out in order to reach the steady state condition.as describe in the table, for each experiments carried out, some trouble were encountered. In some cases immediate solutions were applied to instantly solve the trouble and complete the experiments. Our analysis and discussion will be based on the experiments made on December 26 2018 as the data recorded were so far the most reliable and accurate compare to the other datasets.

Table18. Datasets description

Datasets					
Exp Date	Load Running Time	Max RPM	Trouble Encountered	Causes	Solutions
2018/8/31	5min	20160	shaft lubricating line failure, vibration, noise, remote control system failure, boiler temp unstable, boiler temp unstable, tank water level uneven	Wi-Fi system unstable	data install on control panel
2018/12/13	1h37min	35940	Hot water leakage, boiler temp unstable, tank water level uneven	leakage from pipe line	instant solution
2018/12/14	2h9min	30240	Hot water leakage, boiler temp unstable, turbine vibration, tank water level uneven	leakage from pipe line	welding
2018/12/25	15min	38880	boiler Trip, Turbine vibration boiler temp unstable 77to87c, tank water level uneven	Ao3 Error, interlock	Maker assistance
2018/12/26	2h30min break time 2h40min	4952	Strange noise from tank、boiler temp unstable turbine vibration, tank water level uneven	Unknown	Vacuum performed

Due to the turbine inlet pressure variation and temperature instability, the steady state condition had not been reached. Shown in the fig.25 and fig.26 below, is the temperature variation and the pressure variation respectively.

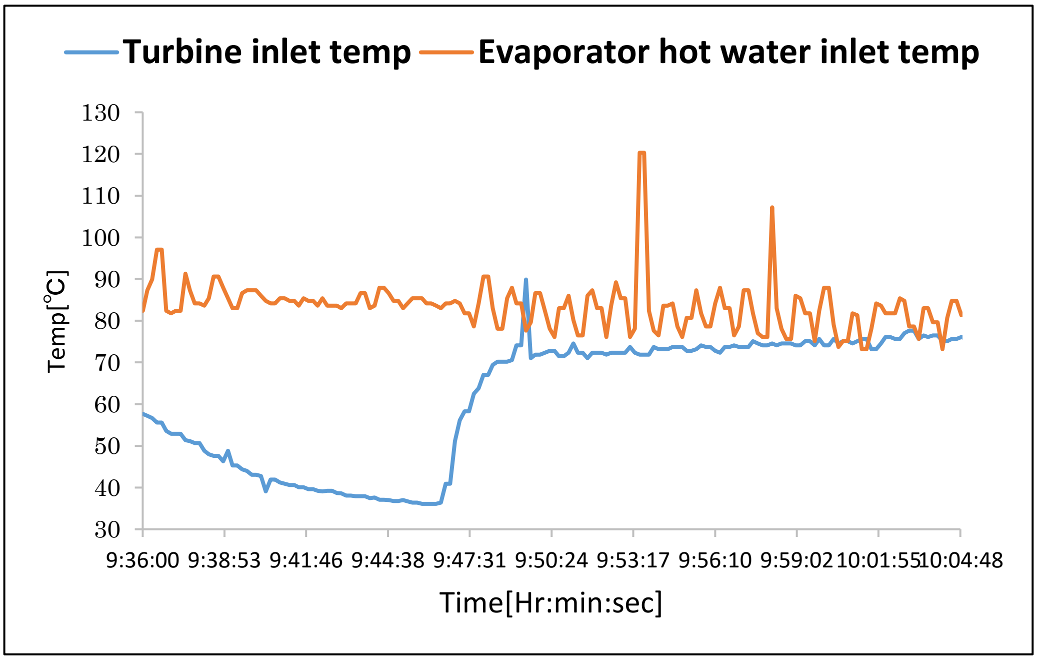

Fig.25 Inlet temperature Data instability observation.

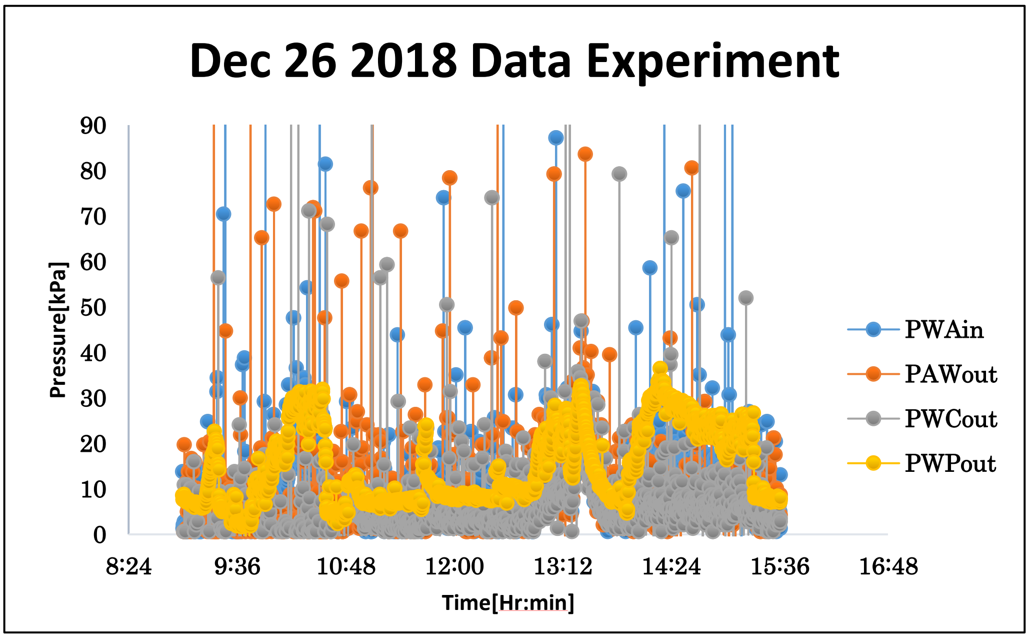

Fig.26 instability of pressure.

Base on the dataset that were recorded so far, some incorrect or unwanted data must be eliminated for accurate analysis and discussion. Shown in fig.27 and fig.28 is the cleanup data representation in the case of the turbine inlet temperature and working fluid flow rate.

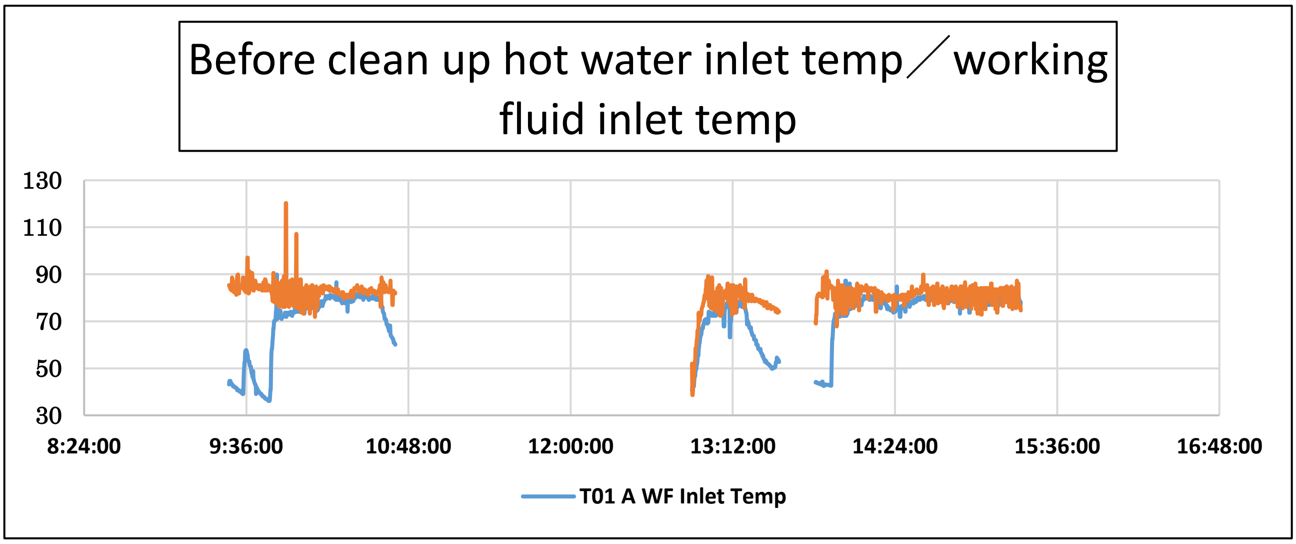

Fig.27 Before cleanup of temperature

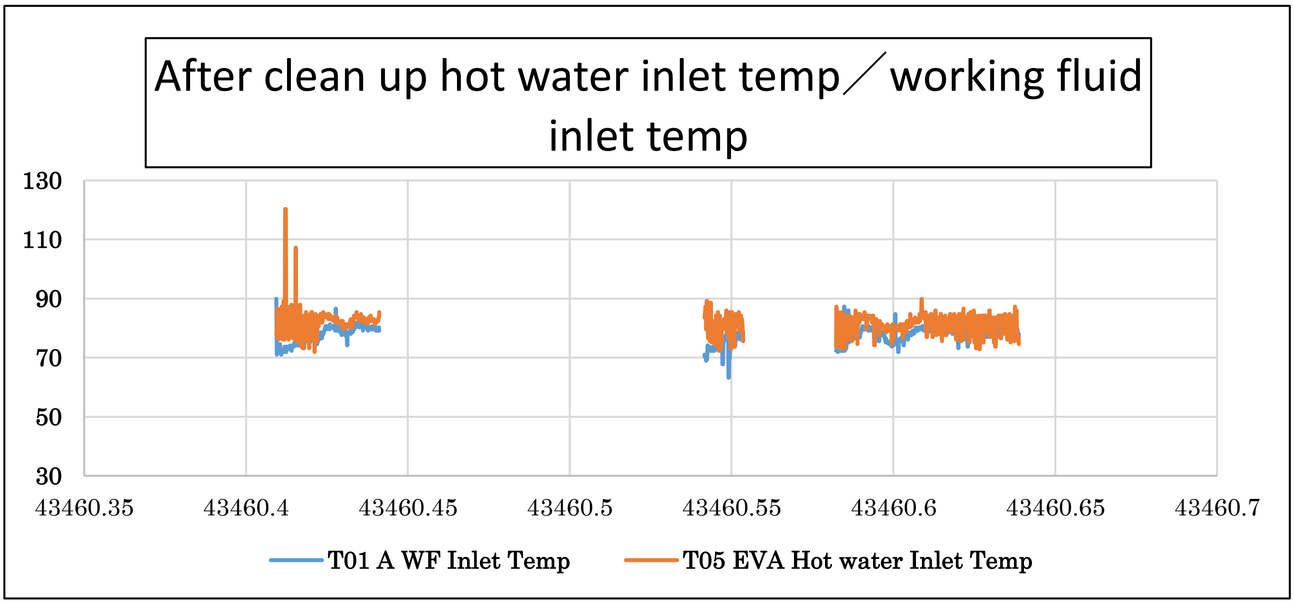

Fig.28 After cleanup of the temperature

3.2.2 Load observation.

The hot water binary power plant was run and operated under certain condition or form of load to determine the system performance. Four different form of load were applied to the system for analysis characteristic. Shown in fig.29 is the load observation.

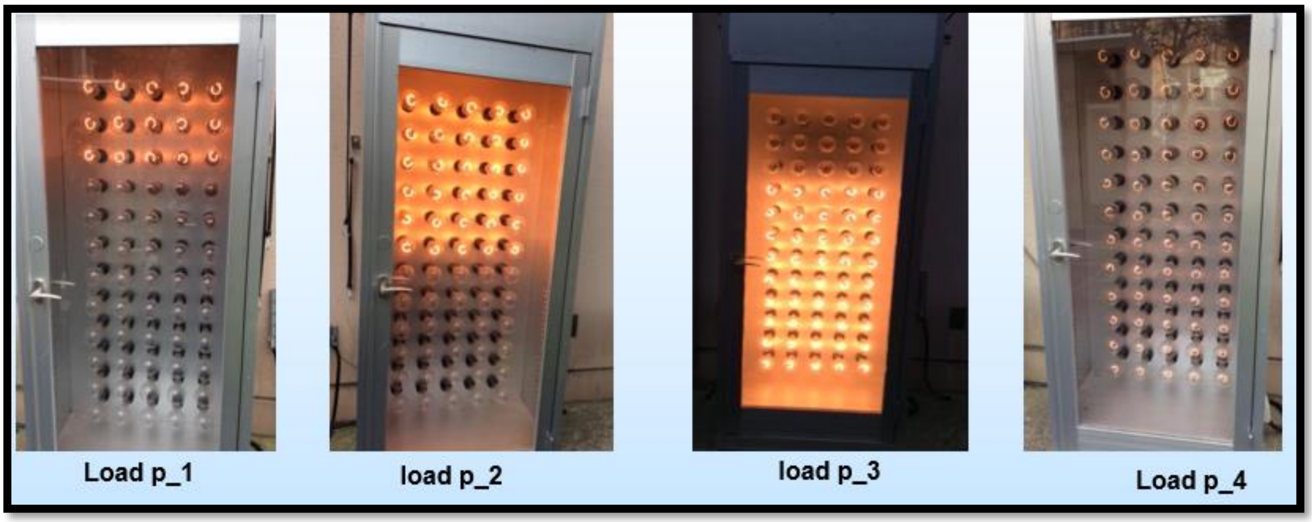

Fig.29 Load observation.

As shown in the fig.29 above, there 4 different forms of load have different characteristics.table19. Shows the load description of all 4 cases.

Table19 load characteristic

	Load
p-load0	**0**
p-load1	**3 rows**
p-load2	**6 rows**
p-load3	**9 rows**
p-load4	**12 rows**

The rows number denote the number of active light. Generally, the different form of load can be separated in 3 major group namely the light load, the medium load and the full load with the light load having the smaller number of active load and the full load have the highest number.

3.2.3 Turbine output and rotational speed relation

Base on the load observations above, the system output was analyzed. As shown in fig.30 below, for each form of load there is a specific line representation. The diagram was obtain after the cleanup of the original datasets. The unnecessary data were eliminated to have the best possible presentation. It can be predicted that for a water binary system the output is a function of the turbine revolution. However, for the same rotational speed there are different value of output, this can be explained by the difference of load that was applied on the system. Each line slope differ from each other due to flow rate variation or difference.

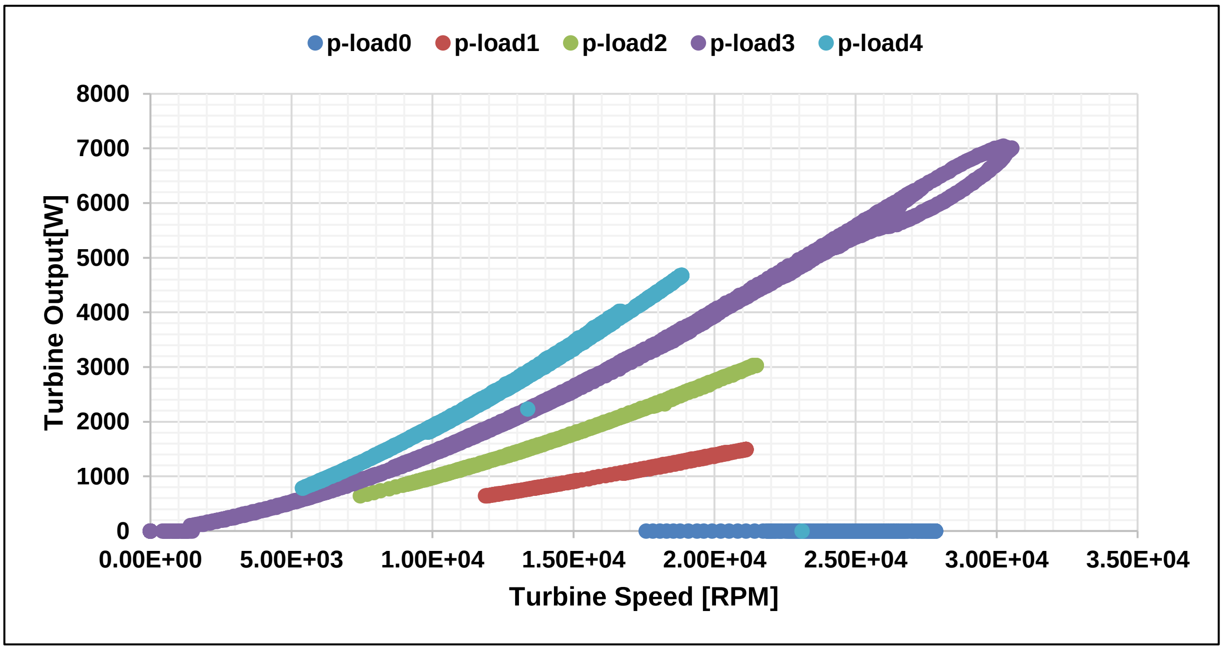

Fig.30 the turbine output increase with the rotational speed

The Rotational Speed-Output relation can be represented by the equation (9) and (10) below where F is the force applied on the turbine, R is the turbine Radius, t is turbine torque, the turbine output is P and N represents the rotational speed. An increase in turbine rotational speed lead to an increase in turbine output.Fig.31 show the action of the torque on the turbine.

$$t = F \times R \tag{9}$$

$$P = N \times F \times R \tag{10}$$

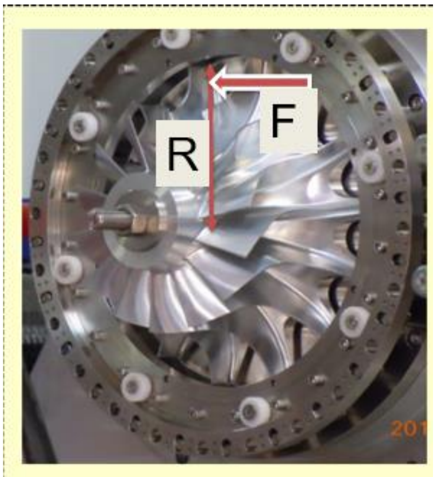

Fig.31 Torque Acting on the turbine.

3.2.4 Turbine flow rate and output

Analysis made on the flow rate showed that it a very important parameters amount other to describe the system functionality. As shown in fig.32 below, as the flow rate increases, the output also increase. Explicit observation made on load 4 representation indicated that the output start falling when the flow rate reaches certain value. This can be explained by the formation of a two-phase flow called as entrainment or carryover. It has been proven that the liquid phase of the steam-liquid mixture hit the back of the rotational turbine and thus slow down the turbine rotation.

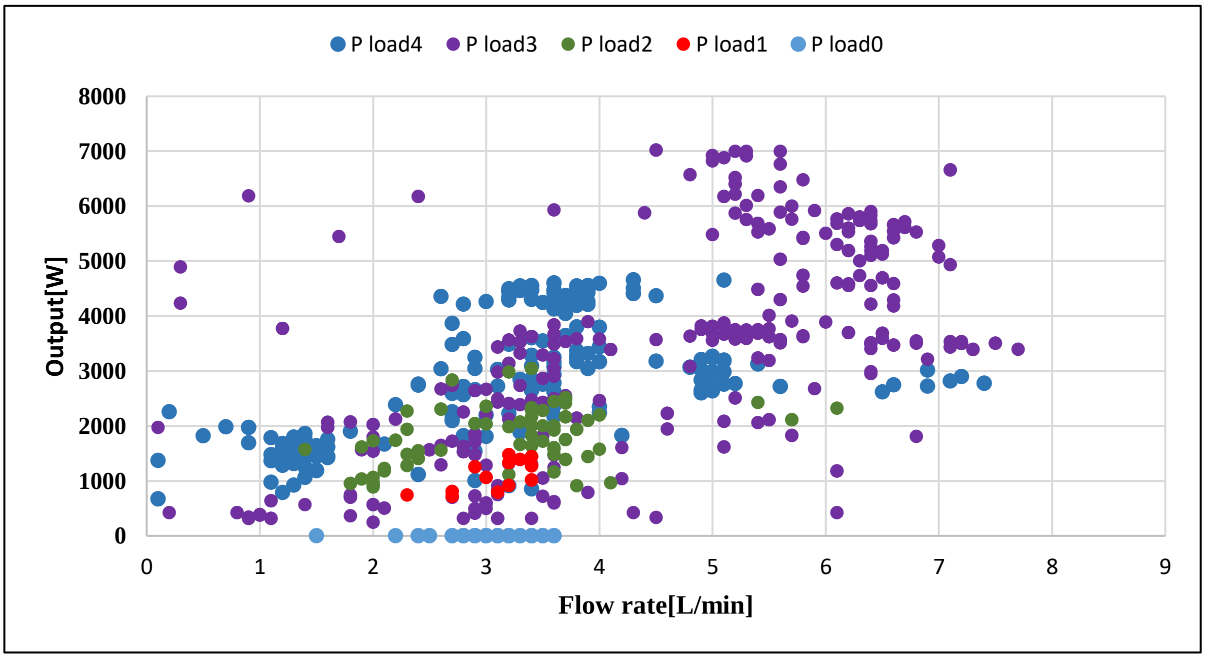

Fig.32 flow rate variation with output.

Given that the system Output increase with increasing flow rate, analysis were made on the system cycle efficiency as shown in the fig.33 below. The system cycle generally increase with the increasing flow rate. However, just above certain value of the flow rate, the system efficiency start falling. This can be explained by the formation of a two-phase flow as mentioned previously.

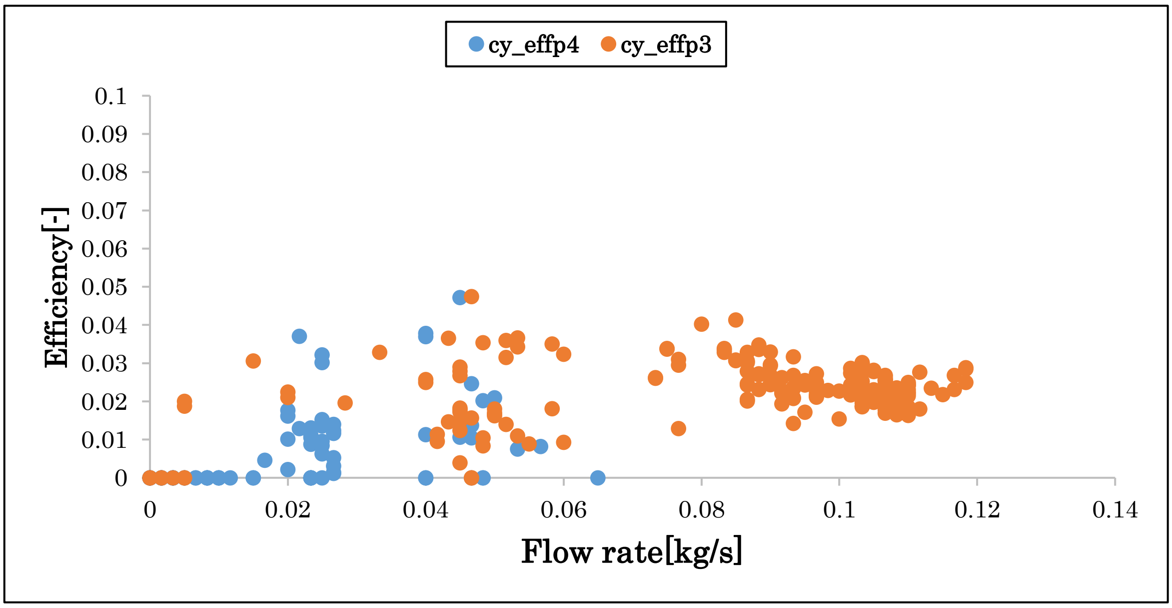

Fig.33 flow rate and efficiency

the cycle efficiency η given in equation (11) is the ratio of the output P to the enthalpy difference ΔH of the hot water at the evaporator. ΔH is consider as the total heat energy apply to the system and p the output at the generation end.

$$\eta = \frac{P}{\Delta H} \tag{11}$$

Analysis were also made concerning the effect of the system rotational speed on the cycle efficiency in order to predict the most accurate rotational speed at which the binary power generation system should be operated. As shown in fig.34 an increase in the rotational speed lead to high efficiency system.it was observed that under the load p4 the rotation stops increase when it reaches certain value, this can be explained by the weight of the load on the system. When the system is switched to a full load a high amount of power or momentum is required for the system to perform under the describe condition. Given that a high amount of momentum is required, the rotational speed does not increase due to the heavy load on the system.

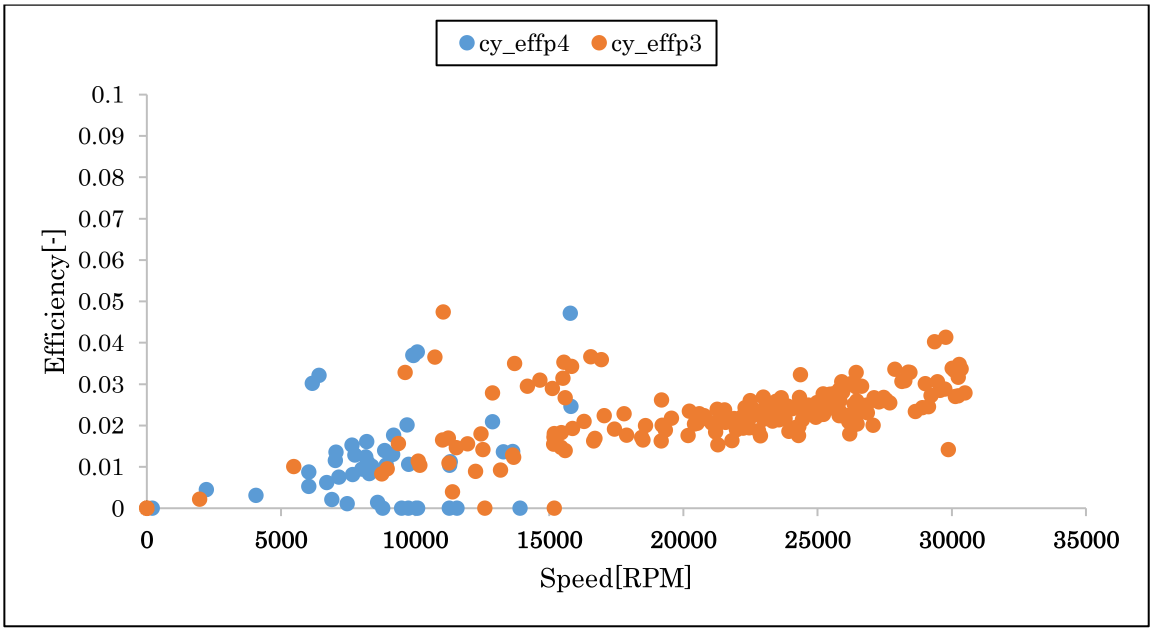

Fig.34 Rotational speed and Efficiency relation.

3.2.5 Efficiency and optimization parameter

From the analysis and discussion above, the Output-Flow rate ration called γ is very important element to determine the system general performance. The Ration Output-Flow rate can be considered as the hot water binary power plant optimization parameter. Shown in fig.35 below is the variation of the system efficiency and the Optimization parameter. The system efficiency increases with the increasing optimization parameters. The optimization parameter represents the proportion of energy in a kg of working fluid.

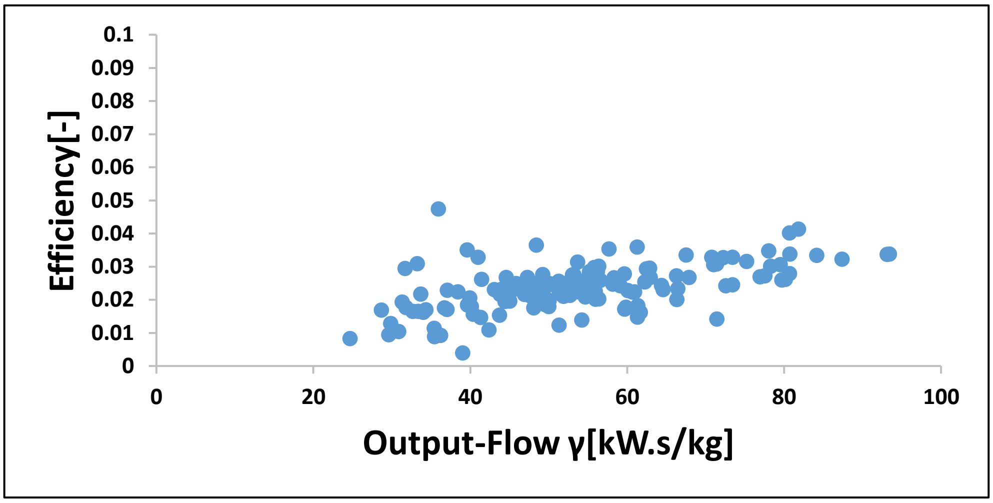

Fig.35 optimization parameter and efficiency

3.2.6 Working fluid pump effect.

Observation made on the working fluid pump pressure shows that it can be a qualitative parameter to describe the functionality of a binary power plant. As shown in fig.36

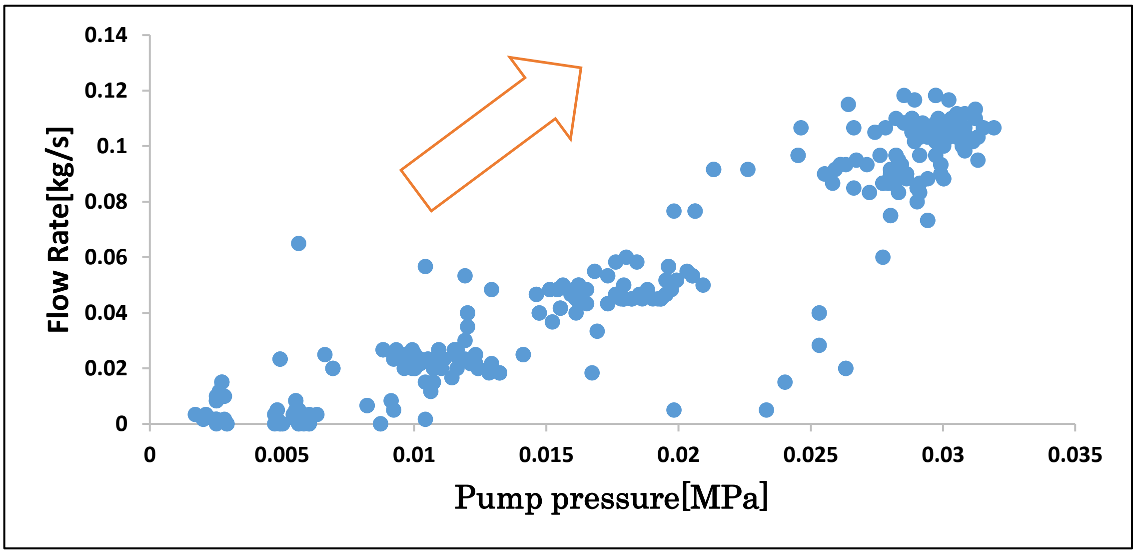

Fig.36 working fluid pump pressure and flow rate

The working fluid pump pressure not only affect the turbine inlet pressure but also the working fluid flow rate.fig.37 analysis indicated that On phase 2 the drop-in pressure can be explained by the formation of a 2-phase flow.

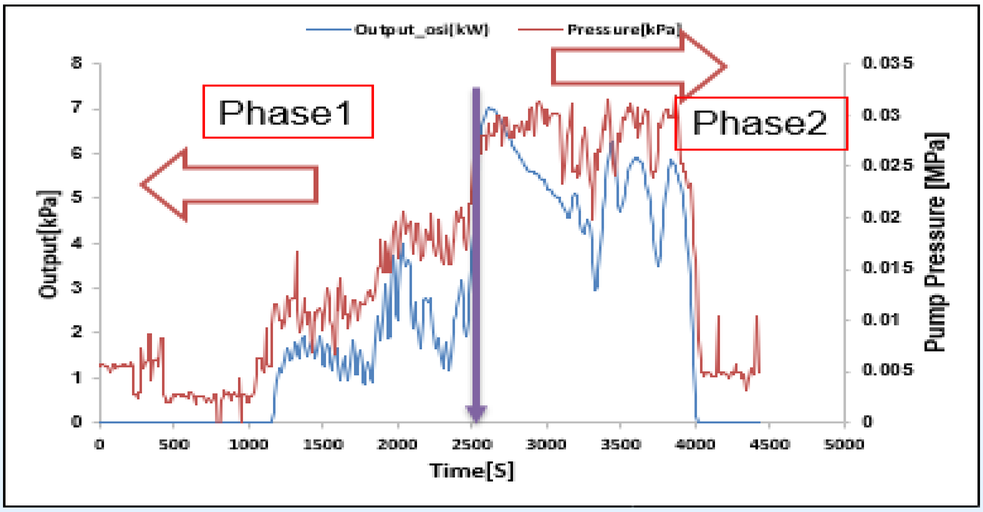

Fig.37 Output and working fluid pump relation

3.2.7 The formation of two phase flow and the effect on the turbine output

From the above analysis it was predicted that there is a formation of a two-phase flow when turbine speed reaches a certain value. The liquid formation can be explained by two main factors, the first factor of formation is during the heat exchange process. A high flow rate of the working fluid slower the formation of steam within the heat exchange therefore there is formation of carry-over or entrainment.

The second factor of formation of a 2-phase flow can be explained by a phenomena that occur within a heat exchanger as the basis for the presence of droplets. As shown in Fig.38 from Florine Giraud et al. 'S paper, the working fluid surface ripple as its passes through the heat exchanger and small bubbles are generated. From this phenomenon, it is inferred that micro droplets may be carry over to the turbine together with the generated bubbles and thus a formation of a two-phase flow.

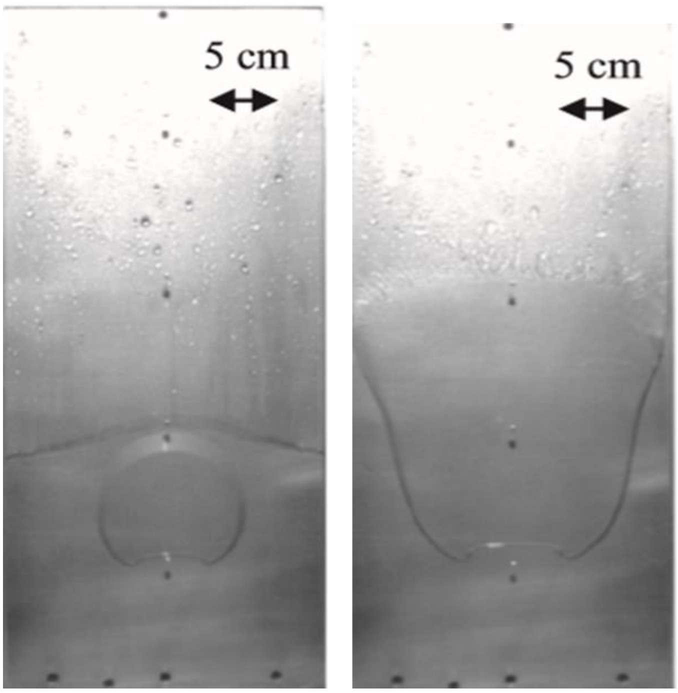

Fig.38 droplet formation.

Considering the influence of droplets contain in the steam-liquid mixture, Although it is difficult to accurately calculate the velocity of the droplet, from the steam turbine outline book, this value can be estimated to 1/10 of the steam velocity. Fig.39 shows the velocity triangle in case of a two-phase flow. The turbine peripheral speed is represented by vector u [m / s], the absolute velocity c_1 , and the relative velocity w_1. From fig. 39, the vector of the relative velocity w_{w1} of the droplet has a component in the direction opposite to the circumferential velocity, and it is observed that the droplet hit in the direction of the back of the turbine blade. The turbine inlet absolute velocity $\vec{c}_1$ of the steam at any time was obtained from the equation (12). An example of the velocity triangle at a given point is explained.

The presence of liquid in the two-phase flow or the formation of steam-liquid two phase flow also affect the turbine inlet angle. The turbine inlet steam angle was calculated as describe below. firstly, nozzle outlet absolute velocity $\vec{c}_1$ is evaluated. Specific heat ratio of wet steam κ is 1.02, the steam gas constant *R* is 0.4615 kJ/kg.K, the nozzle inlet temperature *T*

$$\vec{c}_1 = \sqrt{\kappa RT} \tag{12}$$

Turbine inlet relative velocity is obtain from equation (13)

$$\overrightarrow{w_1} = \vec{c}_1 - \vec{u} \tag{13}$$

From the formula below, absolute inlet angle and relative inlet angle is given by equation (14)

$$\overrightarrow{w_1} = \frac{\vec{c}_1 \cos\alpha_1 - \vec{u}}{\cos\beta_1} \tag{14}$$

Peripheral velocity is obtained by

$$\vec{u} = \frac{N}{60}\pi D \tag{15}$$

Turbine inlet relative velocity angle β_1 is given by

$$\beta_1 = \tan^{-1}\frac{C_1 \sin\alpha_1}{C_1 \cos\alpha_1 - u} \tag{16}$$

Where nozzle angle α_1=27.5 deg

The speed of the liquid phase of the two-phase flow is 1/10 of the speed, the absolute velocity at the turbine inlet $\vec{c}_{1w}$

$$\vec{c}_{1w} = \frac{1}{10} \times \vec{c}_1 \tag{17}$$

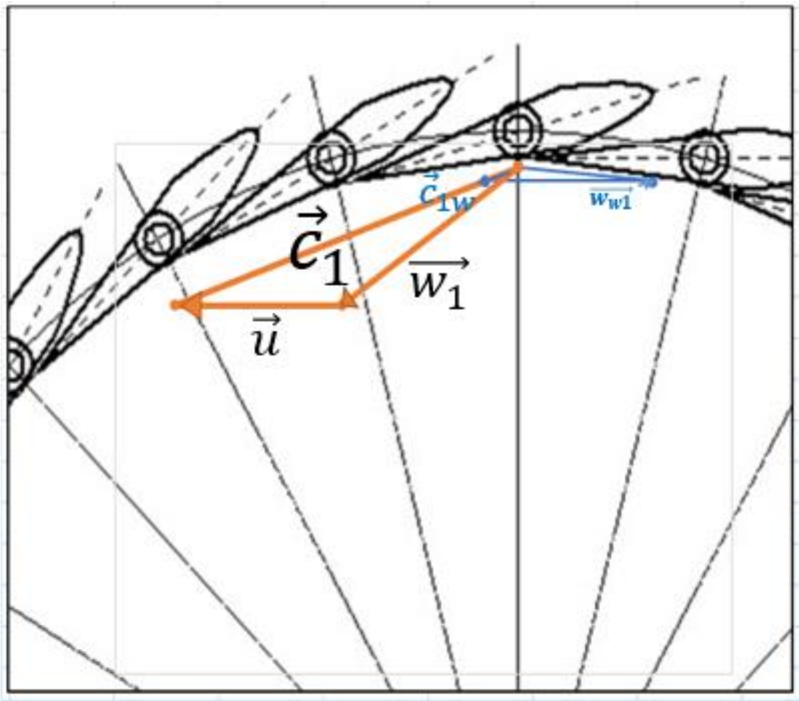

Fig.39 droplet effect on turbine rotational speed

4. Conclusion

A power generation system using the heat source from hot spring water was proposed. For the environmental safety, the working fluid is low pressure water instead of the low boiling point organic substance. The analysis indicated the amount of power generation per kg of hot spring water rose while the cycle efficiency dropped as the turbine inlet pressure or quality decreased.

Decreasing the steam pressure or quality of the turbine inlet steam, the system efficiency decreases, however, the amount of power generation per kg of hot spring water increases and reach its maximum value. Therefore, by considering the limited amount of hot spring water, it is necessary to pay attention to the amount of electricity generated per kg of hot spring water rather than the conventional efficiency. However, since the turbine efficiency is kept constant in this study, it will be taken into consideration for future experiment.

In the conventional cycle efficiency-oriented method, since it is not possible to lower the hot spring water temperature, it was necessary to cool the hot spring water after the power generation for bathing or secondary usage. The method shown in this report is convenient for effectively using hot spring water for secondary usage.

Currently, many binary power generations use polymer organic substance as working fluid instead of water. In general, since the global warming potentials (compared to carbon dioxide) of these organic substances is very high, the water binary power generation can be a suitable alternative solution.

The Experimental analysis indicated that:

Flow rate-Power relation is necessary to operated and run the water binary plant efficiently.an increase in flow rate lead to a carry-over or entrainment which slower the turbine revolution and thus the output of the system.

The system average efficiency 5% and the maximum output 7.0[kW] different from the expected 10[kW] this can be explained by loss in energy such as friction etc.

5-Acknowledgements

I thank all who in one way or another contributed in the completion of this thesis. First, I give thanks to God for protection and ability to work. I am so grateful to the Rotary Yoneyama scholarship and the Faculty of marine system engineering at Tokyo University of marine science for making it possible for me to study here. I give deep thanks to the Professors and lecturers, the librarians, and other workers of the faculty. My special and heartily thanks to my supervisor, Professor Horiki Sachiyo and Osakabe Masahiro who encouraged and directed me. Their challenges brought this work towards a completion. It is with their supervision that this work came into existence. For any faults I take full responsibility.

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