Extended Abstracts of The Second Pacific Rim Thermal Engineering Conference
December 13-17, 2019, Maui, Hawaii, USA

PRTEC- 24201

THEORETICAL AND EXPERIMENTAL INVESTIGATION OF WATER BINARY POWER GENERATION

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ABSTRACT

Binary power generation system using low-pressure water as a working fluid has been proposed. This system is aimed to recover the low-grade waste heat such as the hot spring water; therefore, the assumed temperature of heat source is about 85 °C. The effects of turbine efficiency on the cycle performance were discussed with the theoretical calculations based on the steam table. The calculations showed that the cycle efficiency and the electric power generation per 1kg hot water became almost double if the turbine efficiency increases by 0.2. Furthermore, the 10kW pilot plant with radial twin turbines and plate-type heat exchangers for the evaporator and condenser was constructed. The power increase due to the increase of turbine inlet pressure was expected but the experimental result did not show the clear increase. The higher turbine inlet pressure resulted not only as the increase of critical flow but also the decrease of turbine efficiency. It was considered that the decrease of efficiency was due to the decrease of steam superheat. The more water droplet could be generated in the nozzles and turbines at the lower superheat and resulted as the decrease of turbine efficiency.

KEYWORDS: Binary power generation, Turbine efficiency, Superheat, Low grade heat recovery, Rankine cycle

1. INTRODUCTION

There is abundant geothermal resource in specific areas of Japan and 500MW electricity was generated in 2010, whereas spa is also geothermal resource and widely spreads across the country. In some spa facilities, hot water is cooled for the bathing since the temperature from production wells is too high. In this process, the low-grade heat is wasted; however, it can be recovered and generate 720MW electricity according to some trial calculation [1].

Binary power generating system seems to be a promising technology to recover this kind of low-grade waste heat. However, organic compounds, which are often used as a working fluid in the conventional binary cycle, are not suitable in spa resorts due to their flammability and toxicity. Although alternative Freon is also used as a working fluid, it has high global warming potential property that could exert a bad influence on the earth and the impression on spa resorts. Therefore, the binary power generation system using the low-pressure water as a working fluid has been studied. Osakabe has studied characteristics of water binary cycle based on the steam table [2]. However, the turbine efficiency of water binary cycle is not well known.

In this research, the effects of turbine efficiency on the cycle performance were studied with the theoretical calculations based on the steam table. Furthermore, the experiment with 10kW pilot plant was conducted and the results were discussed.

2. THEORETICAL CALCULATIONS

Theoretical water binary cycle performance was calculated based on the steam table. Table 1 shows the calculated conditions. Each subscript of 0, 1, 2 and 3 shows the locations of the turbine inlet, the turbine

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outlet, the condenser outlet and the evaporator inlet, respectively. In the experiment, the superheated steam was generated in the evaporator; therefore, the theoretical calculations were conducted assuming the superheated steam would be generated. The cooling water outlet temperature in the condenser was set at 25℃ and the pinch point constraint was fixed at 3K; thus, the turbine outlet temperature was on 28℃ and the turbine outlet pressure was at the saturated pressure of 28℃. The turbine inlet entropy can be expressed as

$$s_1 = x_1 s_g + (1 - x_1) s_l \tag{1}$$

If the adiabatic change in turbine is assumed, the entropy at turbine inlet and outlet would be the same. Then, the quality at turbine outlet can be expressed as

$$x_1 = (s_0 - s_l)/(s_g - s_l) \tag{2}$$

The enthalpy at turbine outlet can be given as

$$h_1 = x_1 h_g + (1 - x_1) h_l \tag{3}$$

The actual enthalpy drop in turbine Δh_T is expressed with

$$\Delta h_T = \eta_T \Delta h \tag{4}$$

where Δh is the enthalpy drop in adiabatic change and η_T is the turbine efficiency. The quality at turbine outlet can be expressed as

$$x_1 = (h_1 - h_l)/(h_g - h_l) \tag{5}$$

The pump inlet pressure P_2 was calculated from the turbine outlet pressure P_1 minus the pressure loss in condenser. The pump outlet pressure P_3 was calculated from the turbine inlet pressure P_0 minus the pressure loss in evaporator. When η_p is the pump efficiency, the enthalpy increase Δh_p at pump can be given as

$$\Delta h_p = V_2 (P_3 - P_2)/\eta_P \tag{6}$$

The pump outlet temperature was calculated to fit this enthalpy increase. When the machine efficiency was η_M, the turbine output W_T, the electric demand W_P and the cycle efficiency η_c can be respectively given as

$$W_T = \eta_M \Delta h_T \tag{7}$$

$$W_P = \Delta h_P/\eta_M \tag{8}$$

$$\eta_c = (W_T - W_P)/(h_0 - h_3) \tag{9}$$

When W_{set} is the designated output, the working flow rate m_w can be given as

$$m_w = W_{set}/(W_T - W_P) \tag{10}$$

When Δh_H is the enthalpy drop between the hot water inlet and outlet, the required hot water flow rate m_H can be given by

$$m_H = m_w (h_0 - h_3)/\Delta h_H \tag{11}$$

The generation per 1kg hot water G_n can be given as

$$G_n = W_{set}/m_H \tag{12}$$

The turbine pressure ratio η_p is presented by

$$\eta_p = p_1/p_0 \tag{13}$$

Critical pressure ratio η_c is calculated with

$$\eta_c = \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}} \tag{14}$$

If the turbine pressure ratio is lower than the critical pressure ratio, the critical pressure ratio is adopted in the calculations of mass flow rate. If not, the turbine pressure ratio is adopted. When G is nozzle outlet mass flux, the dimensionless nozzle outlet mass flux G^* is expressed by

$$G^* = \frac{G}{\sqrt{p_0/v_0}} = \left[2\eta^{\frac{2}{k}} \frac{1}{1-1/k}\left(1-\eta^{1-\frac{1}{k}}\right)\right]^{\frac{1}{2}} \tag{15}$$

The nozzle outlet mass flux G is calculated as

$$G = G^*\sqrt{p_0/v_0} \tag{16}$$

When A is the total area at nozzle outlet, the mass flow of the pilot plant m can be given as

$$m = GA \tag{17}$$

The turbine efficiency η_t in the experiment is defined as

$$\eta_t = \frac{W}{m(h_0-h_1)} \tag{18}$$

3. EXPERIMENTAL APPARATUS

Figure 1 shows the overview of experimental apparatus. The low-pressure water, which is the working fluid, was stored in the tanks after coming out from the condenser. Then, the water was pumped into the evaporator and the superheated steam was generated. This steam hit the radial twin turbines and flew out into the condenser. Hot water was generated by the boiler and pumped into the evaporator. Cooling water was chilled in the air-cooling system and pumped into the condenser. Each turbine, which has degree of reaction, was

Table 1. Theoretical calculation condition

Output	[kW]	10
Hot Water Inlet Temperature	[°C]	85
Turbine Inlet Temperature	[°C]	78
Cooling Water Inlet Temperature	[°C]	15
Cooling Water Outlet Temperature	[°C]	25
Pinch Point Constraint	[K]	3
Pressure Loss in Evaporator	[kPa]	5
Pressure Loss in Condenser	[kPa]	0.6
Pump Efficiency	[-]	0.4
Machine Efficiency	[-]	0.92

Table 2. Experimental condition

Hot Water Inlet Temperature	[°C]	75-90
Hot Water Outlet Temperature	[°C]	20-30
Hot Water Flow Rate	[kg/s]	3-7
Cooling Water Inlet Temperature	[°C]	10-18
Cooling Water Outlet Temperature	[°C]	12-24
Cooling Water Flow Rate	[kg/s]	7-12
Working Flow Rate	[kg/s]	0.03-0.12
Nozzle Flow Area	[m^2]	2.7×10^{-3}
Specific Heat Ratio of Superheated Steam	[-]	1.3

installed at both ends of the turbine shaft and the generator located at the middle. Plate-type heat exchangers were adopted for the evaporator and the condenser.

Table 2 shows the experimental condition. The analyzed data were picked up from the power-stable data ranges and averaged in 30 seconds.

4. RESULTS

Figure 2 shows the relation between cycle efficiency and turbine inlet pressure obtained with the theoretical calculations. It is obvious that if the turbine inlet pressure increases by only 0.2, the cycle efficiency becomes almost double.

Figure 3 shows the relation between generation per 1kg hot water and turbine inlet pressure with the calculations. As well as the cycle efficiency, it is evident that if the turbine inlet pressure increases by only 0.2, the generation per 1kg hot water becomes almost double.

One of the operation methods to set the turbine inlet pressure at around 15kPa is preferable to obtain the higher generation per 1 kg hot water which contributes to the more efficient use of geothermal resource. Even if the turbine inlet pressure is increased, the cycle efficiency is slightly increased to about 8% with the turbine efficiency 0.8.

Figure 4 shows the relation between the power output and the revolutions per minute in the experiment. The green dots show all measuring data and blue dots are the power-stable data used for data analysis. It is observed that the output increases with the revolutions per minutes and the maximal output is about 8kW.

Figure 5 shows the relation between the output and turbine inlet pressure. It was expected that the higher turbine inlet pressure contributed to the larger output since the equation (16) showed the larger steam flow rate into the nozzles at the higher turbine inlet pressure. However, the output does not show the clear increase with the pressure.

Figure 6 shows the relation between turbine efficiency and turbine inlet pressure. The turbine efficiency decreases with the pressure increase. It is suggested that the increase of turbine inlet pressure do not contribute to the power increase due to the decrease of efficiency.

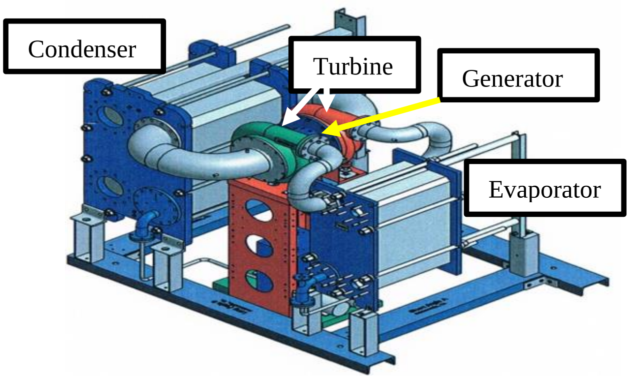

Fig. 1 Overview of experimental apparatus

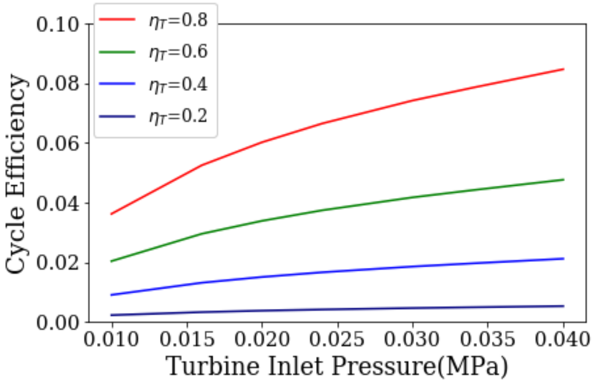

Fig. 2 Effects of turbine efficiency

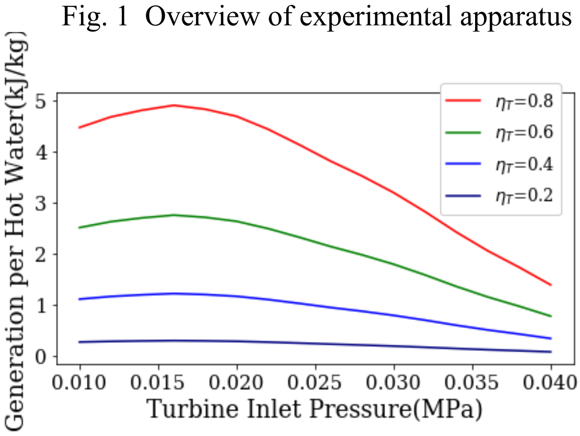

Fig. 3 Effects of turbine efficiency

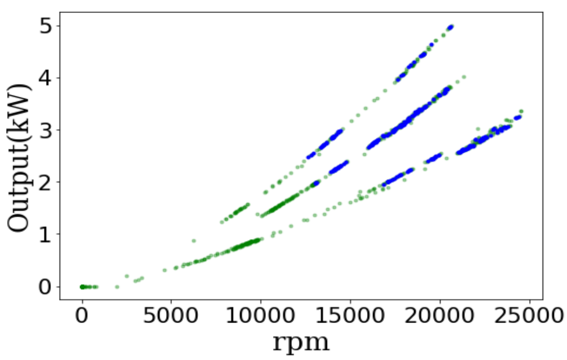

Fig. 4 Output in the experiment

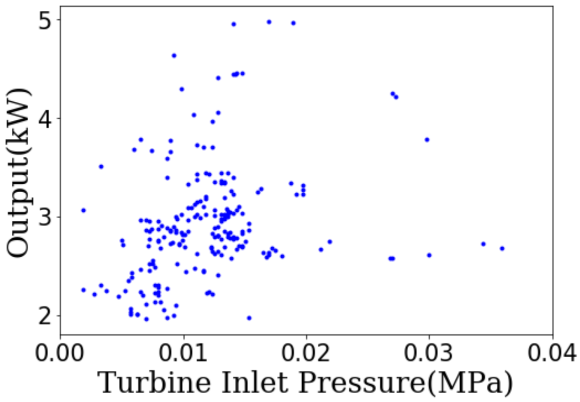

Fig. 5 Effects of turbine inlet pressure

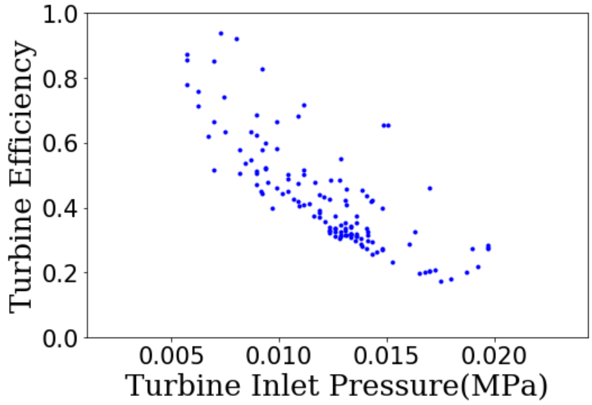

Fig. 6 Turbine efficiency

In this experiment, the hot water temperature supplied to the evaporator is approximately the same. So when the turbine inlet pressure is increased, the superheat of the steam decreases. Figure 7 shows the relation between turbine efficiency and superheat. The turbine efficiency tends to decrease with the decrease of the steam superheat. It can be considered that the turbine efficiency decreases due to the decrease of superheat, since the lower superheat allowed the generation of the more water droplets in the nozzles and the turbines.

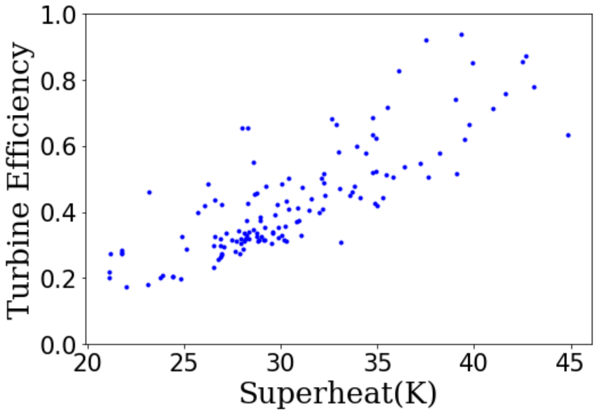

Fig. 7 Effects of superheat

5. CONCLUSION

(1) In the theoretical calculations, 0.2 increase of turbine efficiency doubles the cycle efficiency and the generation per 1kg hot water. The importance of high turbine efficiency is quantitatively clarified. Moreover, it is pointed out that the generation per 1kg hot water is considered to be important for the more application of geothermal resource.

(2) It was expected that the higher turbine inlet pressure would generate more power due to the increase of steam flow rate. However, the clear relation between the pressure and the power was not confirmed in the present experiment.

(3) The turbine efficiency decreased as the turbine inlet pressure was increased in the experiment. This decrease was considered to prevent the power increase due to the increase of turbine inlet pressure

(4) The superheat decreased as the turbine inlet pressure was increased in the experiment where the temperature of the hot water supplied to the evaporator was given. It was considered that the decrease of superheat resulted in the decrease of turbine efficiency, because the more amounts of water droplets would be generated in the nozzles and the turbines with decreasing superheat.

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