OPTIMIZATION AND DESIGN OF TATARA BINARY GEOTHERMAL POWER PLANT IN *BEPPU, JAPAN*

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ABSTRACT

The Tatara binary geothermal power plant is a small scale plant located in Beppu, Oita prefecture, Kyushu Island, Japan, with installed capacity of 70 kW. This plant started operation in August 2014 using geothermal fluid at about 106ºC from geothermal well of 550m, and R245fa as a working fluid. It has the capacity to produce 60 kW as gross turbine work output. In this plant the geothermal fluid produced from a well is first heat exchanged with ground water which is further heat exchanged with the working fluid. To avoid any chemical complications in the heat exchanger, ground water was used as a heat medium with the working fluid, R245fa. In this plant, heat depleted geothermal fluid after heat exchange and cooling water are used for hot water supply by mixing with groundwater. Mathematical models for energy flow was developed and implemented in Engineering Equation Solver program. This study shows the working conditions of the Tatara binary plant and it was found that the plant is already operating closely to the optimal working conditions, its overall thermal efficiency was found to be 8.3% according to the developed model while the system efficiency was found to be 5.8%.

1. INTRODUCTION

Since the last years, new concepts for electricity generation using low-enthalpy / low temperature sources are under investigation to reduce the electricity generating costs and to satisfy the rising energy demand within a decentralized electricity production approach (Herfurth, et al., 2015).

Binary cycle power plants are the most in operation comprising of about 50% of the total number (613) of geothermal plants worldwide (Bertani, 2015). The difference between binary cycle and other geothermal power generation systems is that binary plants are able to operate on low temperature geofluids. Binary cycle plants or Organic Rankine Cycle (ORC) as they are also known, are used to generate electricity from medium to low temperature geothermal resources (Valdimarsson, 2010) and they help to increase efficiency of geothermal fluid through recovery of heat from waste fluid of steam flash power plants. Binary power plants use a secondary working fluid, which is organic, to rotate turbine and produce electricity. The secondary working fluid has a low boiling point temperature and a high vapour pressure at low temperatures when compared to water (Maghiar & Antal, 2001).

Tatara binary plant uses an already existing geothermal well which produce steam-water mixture and used as heat source. This geothermal fluid is first heat exchanged with the ground water which is further heat exchanged with the working fluid and finally depleted geothermal fluid, as well as cooling water are both mixed with ground water to make hot water for heating purposes. The main objective of this study is to evaluate the thermal efficiency and the utilization of the Tatara binary plant in order to archive higher energy efficiency.

2. GEOTHERMAL POWER PLANTS IN JAPAN

Japan is one of the most tectonically active countries in the world, with nearly 200 volcanoes and the evidence of tremendous geothermal energy resources (IGA, 2015). Its geothermal development started in 1925, with an experimental unit, and the first commercial plant on Matsukawa started in 1966. However, due to environmental concerns, geothermal power plants in Japan has not been increasing. The feed-in Tariff Scheme for Renewable Energy which was launched in Japan in 2012, sets the purchase price from geothermal power generation of less than 15 MW at 42 yen/kWh (Mitsubishi Heavy Industries, 2014) which is higher compared with the price of electricity produced with fossil fuel such as 8.9 yen/kWh for nuclear power, 12.3 yen/kWh for coal fired power and 13.5 yen/kWh for LNG fired power generation (Matsuo, 2015).This attracted lots of investment in small-scale binary power generation facilities including the Tatara binary power plant.

3. TATARA BINARY PLANT

3.1. Location of Tatara binary plant and Geology of the area

Figure 1: Location map of Tatara binary power plant

The Tatara binary plant is located in Beppu, Oita prefecture. Beppu is one of the large hot spring resorts in Japan (Fig.1). There are numerous fumaroles and hot springs scattered on a fan-shaped area, extending 5 km from East to West and 8 Km from north to south. Some of the thermal manifestations are called "Jigoku (Hell)" and are of interest to visitors. (Geothermal Research Society of Japan, 1988). It is located at the eastern end of the Beppu-Shimabara Graben. The basement rocks are probably the same as those of Hatchobaru geothermal field: Paleozoic crystalline schists and cretaceous granite intrusions. This is based on the fact that such rocks are found as xenoliths in rocks such as the Yufugawa pumice flow, the Hohi volcanic rocks and the Yufu volcanic rocks (Kikkawa and Moriyama, 1974). Miocene to early Pleistocene volcanic rocks and formations (the Kankaiji and the Setouchi volcanic rocks, Oita formations, and Hohi volcanic rocks in ascending order) are mainly situated in the southern part of the field. (Geothermal Research Society of Japan, 1988).

3.2 Characteristics of Tatara Binary Plant

The Tatara binary plant is composed of 3 loops; geothermal hot water, ground water and working fluid loop. The first loop contains geothermal hot water from the well, the second loop is for ground water which heat exchanges with geothermal fluid, the third loop is for the working fluid which exchanges heat with the ground water and finally being cooled by circulating cooling water. Both cooling water and heat depleted geothermal water are mixed with ground water to produce hot water for heating purposes. The average temperature, mass flow rate and pressure for geothermal hot water measured during the testing period was 106ºC, 1.7kg/s and 0.5 bar (g) respectively.

4. METHODOLOGY

This paper concerns the performance analysis and optimization of Tatara binary geothermal power plant as shown in Fig.2. Mathematical model formulations and their solutions for energy analysis were conducted in detail for all major components of the plant including turbine, heat exchangers and pumps which were thermodynamically analysed. Then the parasitic load was deducted from the total work output i.e., power consumed by pumps and other auxiliary loads in the power station. The analysis was performed using the Engineering Equation Solver (EES), which is a general equation-solving program that can numerically solve several coupled non-linear algebraic equations (Jalilinasrabady et al., 2011).

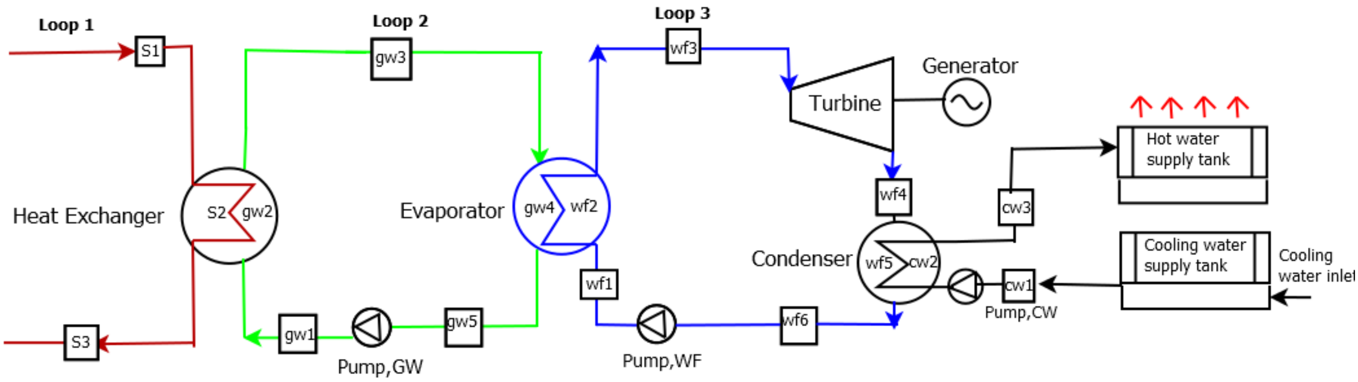

Figure 2: Process flow diagram of Tatara binary plant

The Tatara plant flow diagram is depicted in Fig.2. The heat source cycle is designated with a subscript s on all parameters concerned. The ground water cycle, working fluid and the cooling cycle are designated with subscripts gw, wf, and cw respectively. The main parameters used for simulation were determined by the heat source characteristics, and to make calculations possible, some assumptions were also made. The input data to the developed model were obtained from the power plant measurements and the unavailable data were calculated by equations (1-14) in the developed model using EES. Table 1 shows the working conditions of the Tatara binary plant, including the measured and calculated parameters for cycles in Fig.2.

Parameters (Unit)	Value
Geothermal fluid temperature (HEX inlet) (°C)	106
Geothermal fluid temperature (HEX outlet)(°C)	81.32
Geothermal fluid pressure (Bar (g))	0.50
Geothermal fluid mass flow rate (kg/s)	1.70
Ground water temp. (Evaporator inlet)(°C)	82.79
Ground water temp. (Evaporator outlet)(°C)	73.26
Ground water pressure (bar)	0.50
Ground water mass flow rate (kg/s)	17.0
Turbine inlet temperature(°C)	74.95
Working fluid mass flow rate (kg/s)	4.50
Turbine inlet pressure (bar(g))	6.95
Condenser pressure (bar(g))	1.96
Ambient temperature (°C)	20
Turbine efficiency (%)	85
Pumps efficiency (%)	75

Table 1: Boundary conditions for the developed model

4.1 Thermodynamic analysis

Analysis of the heat exchangers is straightforward and well described in many engineering books such as Shah and Sekulic, 2003).

Standard methodology assumes:
a) Steady state operating conditions
b) No heat losses from heat exchangers
c) Pure countercurrent flow in heat exchangers
d) Constant overall heat transfer coefficient
e) Constant specific heat of heat source fluid

As heat losses in heat exchangers are neglected, the amount of the heat added to the ground water is taken to be equal to the heat extracted from the geothermal fluid.

4.1.1 Heat exchange between geothermal hot water and the ground water (Loop 1 in Fig.2)

$$Q_S = Q_{gw} \tag{1}$$

$$Q_S = m_s \ (h_{s1} - h_{s3}) \tag{2}$$

The heat balance across the heat exchanger is given by

$$m_S(h_{S1} - h_{S3}) = m_{gw}\big(h_{gw3} - h_{gw1}\big) \tag{3}$$

where m_s is the mass flow rate (kg/s) of geothermal fluid, Q_s is the heat lost by the geothermal fluid (kJ/s), h_x is the enthalpy at point x (kJ/kg), m_{gw} is the ground water mass flow rate (kg/s) and $\mathrm{Q_{gw}}$ is the rate at which heat is gained by the ground water (kJ/s).

4.1.2 Heat exchange process between ground water and the working fluid (Loop 2 in Fig.2)

Energy and mass balances between the two kinds of fluids becomes;

$$Q_{gw} = Q_{wf} \tag{4}$$

The mass balance across the evaporator can be written as

$$m_{gw}\left(h_{gw3} - h_{gw5}\right) = m_{wf}\left(h_{wf3} - h_{wf1}\right) \tag{5}$$

Temperatures at point gw_4 and wf_2 as indicated in Fig.2 are sensitive to pinch point temperature in the evaporator.

The pinch point temperature denotes the smallest difference in temperature that can be reached between the primary fluid and the secondary fluid (Valdimarsson, 2010) and is usually provided by the manufacturer of the heat exchanger. The relationship of temperatures at these two points is therefore given with respect to the pinch point as follows:

$$T_{gw4} = T_{wf2} + T_{pinch\_Evap} \tag{6}$$

where T_{pinch_Evap} is the pinch point temperature difference in evaporator (°C)

4.1.3 Turbine Analysis

It is ideally perceived that the process of expansion of vapour in the turbine is isentropic i.e. entropy at the output of the turbine is the same as entropy at the turbine inlet ($S_3 = S_4$). However, in actual application the process is not isentropic since the expansion is irreversible and the process increases the fluid entropy. Both the specific enthalpy hs_{wf4} and the actual enthalpy h_{wf4} are assessed at the point wf4.

Relationship between enthalpy and isentropic turbine efficiency is given as:

$$\eta_{tur} = \frac{h_{wf3} - h_{wf4}}{h_{wf3} - hs_{wf4}} \quad (7)$$

where η_{tur} is the turbine isentropic efficiency, and hs_{wf4} is the isentropic enthalpy at point wf4 (kJ/kg). Efficiency of turbine is generally provided by the turbine manufacturer and it is common practice to use 85% (Valdimarsson, 2010).

Work done by the vapour which is the mechanical power output from the turbine is given by turbine efficiency, mass flow rate of the fluid passing through the turbine and the enthalpy drop across the turbine as:

$$W_{tur} = \eta_{tur}\, m_{wf}\left(h_{wf3} - hs_{wf4}\right) \quad (8)$$

where W_{tur} is the mechanical power output of the turbine (kW).

From the turbine, the vapour is led to a condenser inlet at point wf4 where pressure is kept as low as possible with the aim of extracting more energy from the turbine process. The condenser is coupled to the water cooling system which performs three tasks: de-superheating, condensing and sub-cooling the working fluid.

4.1.4 Condenser Analysis

In the condenser, heat Q_{wf} is rejected from the working fluid between stations wf4 and wf6, to the cooling medium in the condenser. Station wf5 is the dew state where the working fluid is fully de-superheated and is saturated vapour at condensation temperature. The cooling medium accepts heat from the working fluid as Q_{cw} across the stations cw1 and cw3 as shown in Fig 2. Hence

$$Q_{wf} = Q_{cw} \quad (9)$$

where Q_{wf} is the heat from working fluid in condenser (kJ/s), Q_{cw} is the heat to cooling medium in condenser (kJ/s).

It is assumed that all the rejected heat from the working fluid is accepted by the cooling medium.

The rejected heat from the working fluid is found by using the mass flow rate of the fluid and the change in enthalpy across the condenser at stations wf4 and wf6. Point wf5 is the condensation point inside the condenser. The relationship is expressed by:

$$Q_{wf} = m_{wf}\left(h_{wf4} - h_{wf6}\right) \quad (10)$$

Q_{cw} is found by multiplying the cooling fluid mass flow rate and the change in enthalpy in the cooling water across the condenser to give:

$$Q_{cw} = m_{cw}\,(h_{cw3} - h_{cw1}) \quad (11)$$

where h_{cx} is the cooling fluid enthalpy at point x and m_{cw} is the cooling fluid mass flow rate.

After being condensed, the working fluid is directed to a fluid circulation pump at point wf6 where the fluid pressure is increased. The fluid then flows to the evaporator at wf_1 and the cycle then repeats.

Work done by the working fluid circulation pump can be calculated as;

$$W_P = m_{wf}\,\eta_P\,\left(hs_{wf1} - h_{wf6}\right) \quad (12)$$

where W_P is the work done by the working fluid circulation pump (kW).

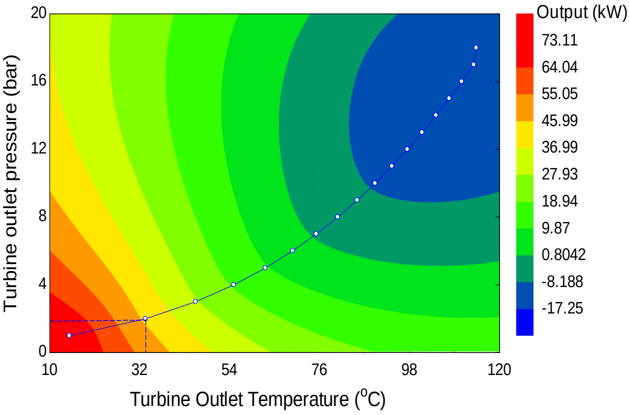

Figure 4: Variation of turbine work output and turbine outlet temperature with turbine outlet pressure

In the cycle, heat from the heated ground water is added to the working fluid as it passes through the evaporator, and heat is removed from the fluid as it passes through the condenser, after the fluid has driven a turbine. The cycle includes a parasitic load required to drive a circulation pump and related equipment depending on the fluid's pressure requirements. The thermal efficiency is therefore determined using Eq. (13) as presented by Marcuccilli & Thiolet (2010),

$$\eta_{th} = 1 - \frac{\Delta h_{wfc}}{\Delta h_{wf_{Evap}}} * 100 \quad (13)$$

where η_{th} is the cycle's thermal efficiency, Δh_{wfc} is the enthalpy difference in the condenser and $\Delta h_{wf_{Evap}}$ is the enthalpy difference across the evaporator.

This study also considered the overall efficiency of the plant.

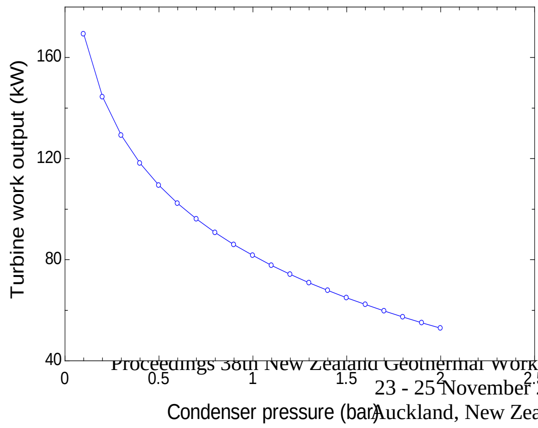

Figure 3: Relationship between Condenser pressure and turbine work output

The efficiency of a power station is evaluated as the net electricity produced/energy input (Ibrahim et al., 2005). In geothermal power plants, the energy input can be defined as total mass flowrate of fluid (kg/s) multiplied by the average enthalpy (kJ/kg), so that the overall efficiency is given by:

$$\eta_{system} = \frac{W_{net}}{m_s h_s} * 100 \quad (14)$$

where W_{net} *is* the running capacity (kW), m_s is the total mass flow rate (kg/s), and h_s is the reservoir enthalpy (kJ/kg).

5. RESULTS AND DISCUSSION

An energy analysis of the Tatara binary geothermal power plant has been conducted using the EES program which solves the simultaneous equations in the developed model. Parameter values in Table 1 are used as initial data for the analysis, then sensitivity studies on turbine output were

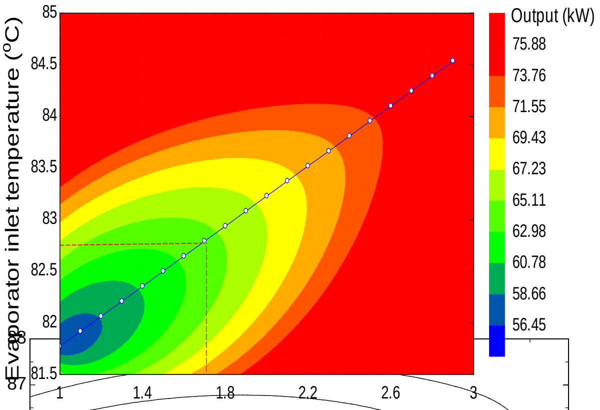

Figure 5: Variation of turbine work output and mass flow rate of geothermal fluid with evaporator inlet temperature

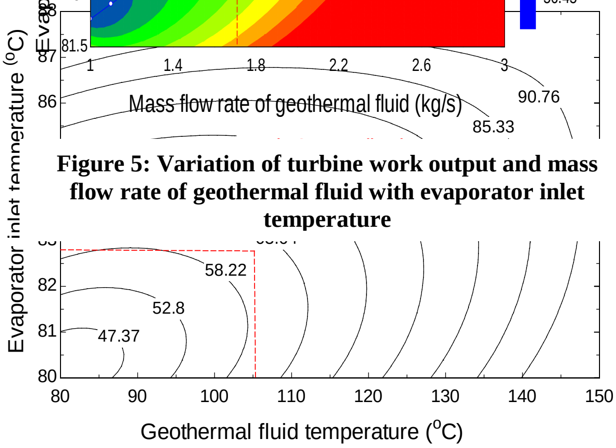

Figure 6: Variation of turbine work output and geothermal fluid temperature with evaporator inlet temperature

conducted by changing parameter values such as condenser pressure, mass flow rate of geothermal water and so on. It is very important to clearly define pressures in the cycles. Geothermal fluid, ground water and cooling water pressures change only slightly, while working fluid pressure changes significantly.

Figure 3 presents the turbine work output versus condenser pressure. Work output quickly decreases with an increase in condenser pressure in its low range, but the rate of decrease of work output becomes moderate at higher condenser pressures.

This figure clearly indicates that condenser pressure significantly affects the turbine work output. This is due to the fact that it is very important to maintain a near-vacuum in the condenser so that steam can easily flow and more work can be extracted from steam in the turbine.

Figure 4 shows the effects of turbine outlet pressure and turbine outlet temperature on work output. The blue line indicates turbine work output. This figure shows that an increase in turbine outlet pressure leads to an increase in turbine outlet temperature and consequently reduces the rate of steam flow in the turbine. Basically, increasing turbine outlet temperature reduces the enthalpy difference across the turbine and reduces the turbine work output. The lower the turbine outlet temperature, the higher the enthalpy difference across the turbine and the higher the turbine work output. Turbine outlet temperature of about 33 °C and turbine outlet pressure of 1.96 bar indicate the operating point for this cycle.

Figure 5 shows the effects of mass flow rate of geothermal fluid and evaporator inlet temperature on turbine work output. The blue line presents the maximum desirable temperature at a given mass flow rate. From this figure, we can say that an increase in mass flow rate of geothermal fluid leads to a slight increase in the evaporator temperature. This increment causes the turbine work output also to increase. In this study, about 2 kg/s of geothermal fluid and evaporator inlet temperature of 82.79 °C are required to produce a turbine output of 62.26 kW.

Figure 6 shows the effects of geothermal fluid temperature and evaporator inlet temperature on turbine work output. The contours denote iso-work output. An increase in geothermal fluid temperature leads to an increase in the evaporation temperature and pressure of the working fluid in the evaporator. This increases the operating pressure of the evaporator and consequently leads to an increase in the temperature at which heat is transferred to the steam and thus raises the turbine work output. The geothermal fluid at 106 °C and the evaporator inlet temperature of 82.79 °C give the required turbine work output of 62.26 kW.

Figure7 presents the turbine work output versus required working fluid mass flow rate. The turbine work output increases linearly with an increase in working fluid flow rate. The developed model requires 4.5 kg/s of working fluid to produce turbine work output of 62.26 kW.

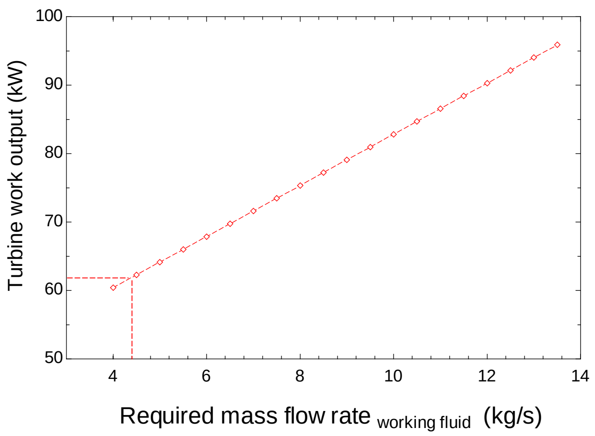

Figure 7: Relationship between working fluid mass flow rate and turbine work output.

Figure 8 presents required mass flow rate of ground water and turbine work output. The figure shows that the turbine work output linearly increases with an increase in ground water mass flow rate. Depending on the developed model characteristics, the required mass flow rate of ground water is 17 kg/s to produce turbine work output of 62.26 kW

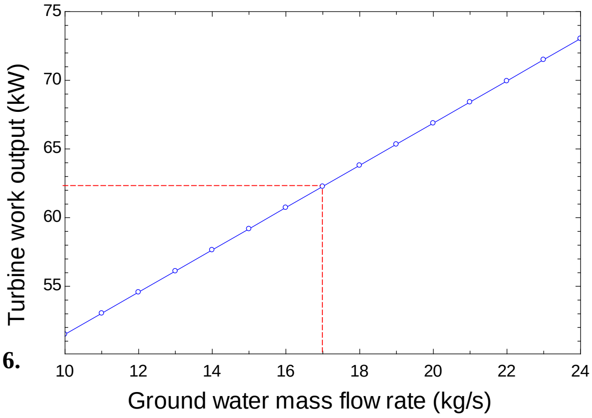

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Figure 8: Relationship between ground water mass flow rate and turbine work output

CONCLUSION

A performance evaluation of the Tatara binary geothermal power plant and a sensitivity analysis have been conducted using a newly developed model. The results are summarized as follows;

1. The plant is operating close to its optimal conditions, with 106°C as the heat source temperature and 82.79 °C as the temperature of the ground water at the evaporator inlet.
2. The required mass flow rates for geothermal fluid, ground water and working fluid are 1.7, 17 and 4.5 kg/s respectively, which gives a turbine work output of 62.26 kW. According to our model, the calculated thermal efficiency of the cycle was 8.3% while the overall efficiency of the system was 5.8%. System efficiency seems to be low compared with that of other ordinary binary cycles due to ground water loop between geothermal fluid and working fluid.
3. Evaporator temperature and pressure have significant impacts on the turbine work output and thermal efficiency in general.
4. Condenser pressure should be kept as low as possible as long as it is higher than atmospheric pressure, to avoid any infiltration into the system; otherwise the lower the pressure, the higher the turbine efficiency and better work output.

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