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Optimization of the Performance of the Steam Turbine SCSF-31.2" with 6 Atages of Axial Exhaust in "X" Geothermal Power Plant (PLTP "X")

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Abstract

The SCSF-31.2" 6-stage axial exhaust turbine in PLTP "X" is the main equipment for the power plant unit that functions to convert heat energy from steam into kinetic energy (rotation). Kinetic energy is transmitted by the shaft to the generator and then converted into electrical energy. The SCCF-31.2" 6-stage axial exhaust turbine currently operates at 7.95 bar (a) inlet pressure, 169.7 °C inlet temperature, and 112.504 kg/s steam mass rate. After evaluating the turbine performance with current actual operating data, the efficiency value is 78.8%, and it produces a gross generator power of 54.867 MW. The current efficiency value has decreased by 9.2% when compared to the initial condition. The decrease in efficiency is expected to occur due to an increase in temperature and pressure in the condenser. The temperature and pressure inside the condenser were initially at 40 °C and 0.074 bar (a) and are currently at 45.2 °C and 0.098 bar (a). Therefore, optimization is done by cleaning the channel sprayer in the condenser with the hope of increasing the water-cooling flow rate and creating a wider contact area. As a result, the condenser temperature and pressure decreased so that the turbine efficiency value increased to 81.2% and the gross generator power increased to 57,224 MW

Keywords:
steam turbine, evaluation, maintenance, performance, optimization

1 Introduction

In geothermal power plants (PLTP), heat energy in the form of steam is used as a working fluid to drive a steam turbine. A steam turbine is a rotary device that converts heat energy from steam into kinetic energy in the form of rotation, which is then forwarded by a shaft that rotates a power generator to produce electricity [1]. In geothermal power plants (PLTP), heat energy in the form of steam is used as a working fluid to drive a steam turbine. A steam turbine is a rotary device that converts heat energy from steam into kinetic energy in the form of rotation, which is then forwarded by a shaft that rotates a power generator to produce electricity [2].

Electricity supply companies strive to maintain and increase production to meet consumer needs. Maintenance is also aimed at returning a system to its original condition so that it can function properly, extending the useful life of the machine, and minimizing failure as much as possible [3]. One way is to continue to perform maintenance on all the equipment owned. One of the maintenance efforts is to evaluate the performance of the equipment, Evaluate can be done both as a whole system (powerplant system) and on each piece of equipment as well as the evaluation of a steam turbine [4].

Evaluation of PLTU or turbine performance has been implemented at PT. PJB UBJOM PLTU Pulang Pisau, Central Kalimantan [5], the PLTU with a capacity of 1500 kW at a company in Kalimantan [6], a power plant with a capacity of 660 MW [7], the New Paiton Steam Power Plant (Unit 9) [8], PT PLN Pembangkitan Asam-asam [9], PLTP Lahendong [10], PT. PLN (Persero) PLTP 1 Ulubelu [11], and PLTU Tarahan [12].

Optimization of the turbine or powerplant has also been carried out, such as the use of LP (Low Pressure) Auxiliary steam at the ejector desalination plant. [13], double reheat regenerative [14], regenerative cycle with 3 pieces of feed water heater [15], and turbine washing [16].

In this study, the performance of the turbine in PLTP "X" located in Bandung will be analyzed based on the turbine's working conditions, and efforts will be made to optimize performance. In other words, the objective of this study is to recover the thermal efficiency of the turbine and boost the gross generator output.

2 Research Methods

The study was preceded by the collection of data and the development of eqs to calculate turbine performance. Observations on the ground and existing references were used to collect data. Then, depending on the collected data, performance calculations are performed using existing eqs and applicable standards. This study's optimization involved cleaning the sprayer on the condenser.

With the use of a thermodynamic state diagram, the geofluid's process may be observed in detail. Fig. 1 depicts a simplified single-flash power plant, where as Fig 2 depicts the temperature-entropy diagram [17].

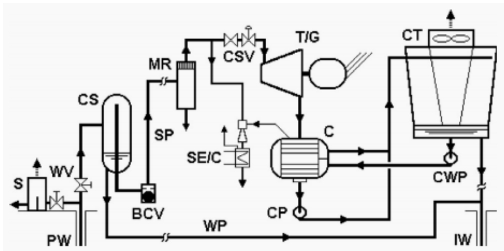

Fig 1. Simplified single-flash power plant schematic

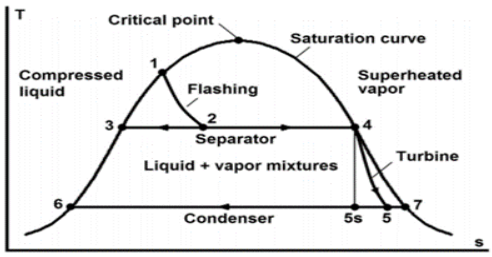

Fig 2. Diagram of Temperature-Entropy state for a single-flash plant

2.1 Flashing Process

The flashing process is depicted by the process from point 1 to point 2 in fig 1. The flashing process begins with a pressured geofluid in a saturated condition (point 1), which undergoes a pressure reduction $\eta_{tw}$to the separator's pressure (point 2). The flashing process is analogous to a constant enthalpy since it occurs spontaneously and in a stable condition without any external work. During the flashing process, either the kinetic or potential energy of the fluid is disregarded eq.(1). The assertion in the eq above is valid [17], where h_1 is Enthalpy at point 1 (kJ/kg) and h_2 is Enthalpy at point 2 (kJ/kg).

$$h_1 = h_2 \tag{1}$$

2.2 Separation Process

The separation process is described in a constant pressure state. The quality of the fluid at point 2 after the flashing process can be determined with the eq [2], where X_2 is steam quality at point 2, h_3 is enthalpy at point 3 (kJ/kg) and h_4 is enthalpy at point 4 (kJ/kg). It can be determined the mass of steam entering the turbine inlet (eq. (2)).

$$x_2 = \frac{h_2 - h_3}{h_4 - h_3} \tag{2}$$

2.3 Turbine Expansion Process

The work done by the turbine per unit mass of steam can be determined by the eq. (3), where W_t is Turbine actual work (kJ/kg) and h_5 is enthalpy at point 5 (kJ/kg).

$$W_t = h_4 - h_5 \tag{3}$$

Assuming there is no heat output and disregarding the kinetic and potential energy entering and leaving the turbine. If the turbine functions in an adiabatic and reversible (constant entropy) way, the maximum value can be attained. From this statement, the following eq. (4) can be used to compute the turbine's isentropic efficiency [4], where n_t is steam turbine efficiency (%) and h_{5s} is enthalpy at point 5s (kJ/kg).

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}} x100\% \tag{4}$$

The power generated by the turbine can be determined by the eq. (4), where $\dot{W}_t$ is Steam turbine generated power (MW), $\dot{m}s$ is total mass rate of steam entering turbine (kg/s) and X_4 is steam quality at point 4.

$$\dot{W}_t = \dot{m}_s w_t = x_4 \dot{m}_{total} w_t \tag{5}$$

The gross mechanical power generated by the turbine is obtained, while the gross electrical power value can be determined from the eq. (5), where W_e is generator-generated power (MW) and n_g is generator efficiency (%).

$$\dot{W}_e = \eta_g \dot{W}_t \tag{6}$$

The net electrical power can be determined by subtracting the gross electrical power from all the auxiliaries used to operate the plant [6].

The ideal turbine's efficiency can be affected by humidity; the higher the humidity, the lower the ideal turbine's efficiency. Using the Baumann rule, one can calculate the effect. According to the norm, 1% humidity generates a 1% reduction in efficiency value.

Geothermal turbines are typically situated in wet areas, and the performance loss must be estimated. The turbine efficiency is determined using the Baumann rule as follows **[17]**, where η_{td} is Dry turbine efficiency (%), η_{tw} is Wet turbine efficiency (%), x_4 is steam quality point 4 and x_5 is Steam quality point 5, where the value of dry turbine efficiency is assumed to be constant at 85%.

$$\eta_{tw} = \eta_{td}\, x \left[\frac{x_4 + x_5}{2}\right] \tag{7}$$

The quality of the steam at the turbine bin outlet (point 5) depends on the efficiency of the turbine (Fig. 2). Point 5 can be determined using the following eq. (8), where h_6 is enthalpy at point 6 (kJ/kg), h_7 is enthalpy at point 7 (kJ/kg), S_5 is entropy at 4 (kJ/kg. K), S_6 Entropy at 6 (kJ/kg. K) and S_7 Entropy at 7 (kJ/kg. K),

$$h_{5s} = h_6 + [h_7 - h_6]x \left[\frac{s_4 - s_6}{s_7 - s_6}\right] \tag{8}$$

where the entropy value describes the dryness of steam under ideal turbine conditions. By entering the Bauman rule we can determine the value of actual enthalpy at the turbine outlet with the eq. (9), where: h_4 is enthalpy at point 4 (kJ/kg), h_6 is Enthalpy at point 6 (kJ/kg), h_7 is enthalpy at point 7 (kJ/kg) and h_{5s} is enthalpy at point 5s (kJ/kg).

$$h_5 = \frac{h_4 - A\left[1 - \frac{h_6}{h_7 - h_6}\right]}{1 + \left[\frac{A}{h_7 - h_6}\right]} \tag{9}$$

$$A = 0{,}435x(h_4 - h_{5s}) \tag{10}$$

2.4 Condensing Process

Thermodynamics Law relates to the flow or mass rate required for cooling water in the condenser as eq. (11)[4], where m_{cw} is mass rate required (kg/s), m_{total} is mass rate leaving the turbine (kg/s), h_5 is enthalpy at point 5 (kJ/kg), Enthalpy at point 6 (kJ/kg), c is Specific heat Constanta of cooling water (4.2 kJ/kg.K),T_6 is The temperature inside the condenser (°K) and T_{cw} is water sprayer temperature (°K).

$$\dot{m}_{cw} = \dot{m}_{total}\left[\frac{h_5 - h_6}{\acute{c}(T_6 - T_{cw})}\right] \tag{11}$$

To produce electricity, this geothermal power plant which operates in an area in Bandung utilizes energy from several production wells to drive its turbine. Fig.3 is a process flow diagram of the PLTP “X”.

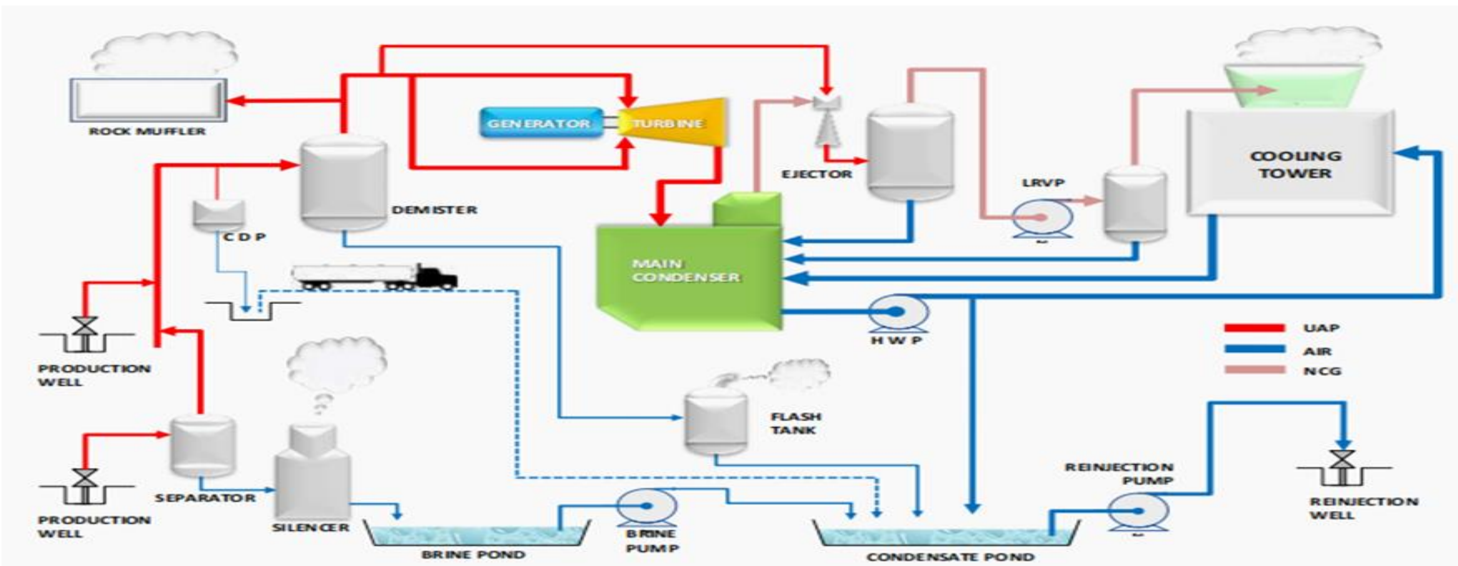

Fig 3. Process flow diagram of PLTP “X”

The turbine that will be evaluated in this research is the SCSF-31.2” 6-stage axial exhaust turbine, namely a single casing, single flow, axial exhaust, condensing turbine, impulse design, and frequency is 50 Hz turbine.

The fluid data used to drive the steam turbine in this PLTP “X” are as Inlet Pressure is 7.95 Bar (a), main steam temperature is 169.7°C, exhaust pressure is 0.098 Bar (a), exhaust temperature is 45.2 °C and output rate is 59,880 kW. For power plant operation data, it can be seen in table 1.

Table 1. PLTP “X” operation data

Parameter	Initial	Operation	Unit
Inlet Steam Mass **R**ate	112,97	112,504	Kg/s
Inlet Steam Temp**.**	170,03	169,7	°C
Inlet Steam Pressure	8,61	7,95	Bar abs
Inlet Steam Enthalpy	2767,92	2767,6	kJ/kg
Outlet Steam Pressure	0,099	0,098	Bar abs
Outlet Steam Temp**.**	43,37	45,2	°C
Cooling **W**ater Temp**.** **i**n Condenser	20,4	20,4	°C
Mass **R**ate **C**ooling **W**ater in Condenser	2234,46	2117,07	Kg/s
Turbine Rotation	3000	3000	rpm
Efficiency	88,06	-	%
Power	60,60	-	MW

3 Results and Discussion.

The evaluation of steam turbines begins with the completion of enthalpy data to determine the ideal and actual work of the turbine. The enthalpy value for each fluid position in the power plant system can be calculated using the steam-water properties table based on temperature and pressure using the given operational data. procedures where the fluid is employed in evaluations of steam turbines.

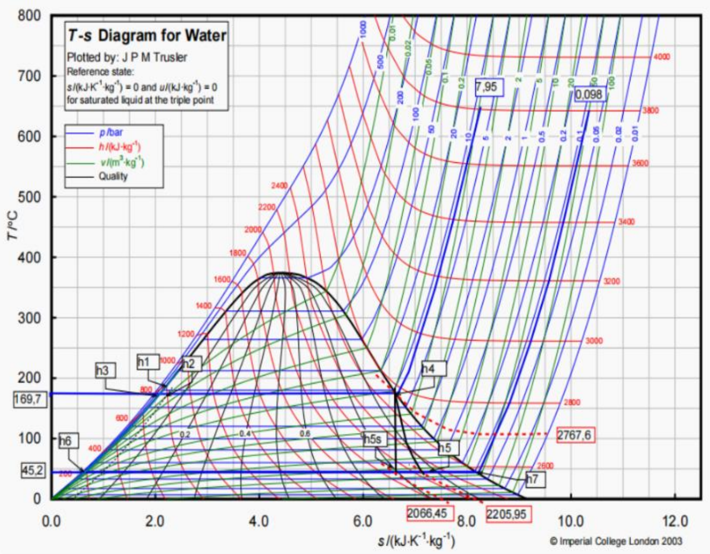

Fig 4. State point on the T-s diagram.

Table 2. Steam properties data for each state

State Point	P (bar)	T (°C)	X	h (kJ/kg)	s (kJ/kg.°K)
1	8	172	0	800	2,2
2			0,04	800	2,2
3	7,95	169,7	0	720	2
4			1	2767,6	6,6
6	0,098	45,2	0	189,28	0,64
7			1	2582,75	8,16

P: Pressure, T: Temperature, X: Steam Quality, H: Eentalphy, S: Entrophy

The thermal efficiency of the turbine is calculated by comparing the actual conditions and ideal conditions and the results are obtained as shown in table 3.

Table 3**.** Results of turbine performance evaluation

Parameter	Value	Unit
Ideal Working	681,37	kJ/kg
Actual Working	536,6	kJ/kg
Thermal Efficiency	78,8	%
Power	54,867	MW

Based on data from table 3, it can be seen, the turbine operates with a work of 536.6 kJ/ kg, a thermal efficiency of 78.8 % and the power generated is 54.867 MW. This means that the turbine efficiency has decreased from 88.06% to 78.8%. Gross generator power, which was initially valued at 60.6 MW, fell to 54,867 MW

Based on current conditions, it can be assumed that one of the causes of the decrease in efficiency and gross generator power is due to insufficient cooling and vacuum in the condenser. This may occur due to the presence of dirt that clogs the sprayer channel in the condenser so that the heat exchange in the condenser becomes less than the maximum.

Optimization in this study was carried out by cleaning the sprayer on the condenser to facilitate the flow and expand the contact area of the cooling water with the steam. After calculating the cooling water requirement, the comparison results before and after optimization are obtained, shown in table 4. With the new temperature and pressure, the turbine efficiency is recalculated with the enthalpy value in table 5

Table 4**.** Optimized condenser operation data

Parameter	Before	After
Cooling water feed (kg/s)	2234,456	2678,737
Steam temperature (°C)	45,2	40
Condenser pressure (bar)	0,098	0,074

Table 5**.** Properties of optimized steam

State Point	P (bar)	T (°C)	X	h (kJ/kg)	s (kJ/kg°K)
1	8	172	0	800	2,2
2			0,04	800	2,2
3	7,95	169,7	0	720	2
4			1	2767,6	6,6
6	0,074	40	0	167,53	0,57
7			1	2573,5	8,26

P: Pressure, T: Temperature, X: Steam Quality, H: Eentalphy, S: Entrophy

Likewise, with turbine performance, evaluation is carried out based on data from optimization results and obtained results as shown in table 6.

Table 6. Optimized turbine performance

Parameter	Value	Unit
Ideal Working	681,37	kJ/kg
Actual Working	559,65	kJ/kg
Thermal Efficiency	82,1	%
Power	57,224	MW

The comparison of the results of the turbine performance evaluation before and after optimization can be seen in table 7 where the turbine efficiency value increases by 3.3% and the gross generator power increase by 2.357 MW.

Table 7. Comparison of conditions before and after optimization

Parameter	Before	After
Steam Flow Rate (kg/s)	112,504	112,504
Condenser Cooling Feed Water (kg/s)	2234,456	2678,737
Condenser Temperature (°C)	45,2	40
Condenser Pressure (bar)	0,098	0,074
Turbine Aktual Working (kJ.kg)	536,6	559,65
Turbine Efficiency (%)	78,8	82,1
Gross Generator Power (MW)	54,867	57,224

4 Conclusions

After completing the evaluation of SCSF-31.2 turbine in PLTP "X" It was determined that the turbine's efficiency has reduced from 88.06% to 78.8%, and that the gross generating power has decreased from 60.6 MW to 54.867 MW.

It was discovered that the flow rate of feed water cooling rose from 2234.456 kg/s to 2678.737 kg/s as a result of cleaning the sprayer channel to provide more feed water cooling and a larger contact area for heat transfer. In addition, optimization reduces the condenser temperature from 45,2°C to 40°C. And the pressure decreased from 0.09 to 0.07 bar.

This improvement boosted turbine efficiency to 82.1% and gross generating power to 57.224 MW.

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