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Performance Analysis of the Main Cooling Water Pump at PT.X Geothermal Power Plant

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Abstract

Purpose: The purpose of this study is to analyze the performance of a spray-type condenser in a geothermal power plant, focusing on heat rate, efficiency, and energy balance.
Research Methodology: This study used daily data collection on the main cooling water pump, followed by a calculation analysis to determine motor power, pump power, and efficiency. The results were compared with the commissioning values to assess the performance of the system.
Results: The calculated turbine heat rate was 20,973.036 kJ/kWh, while the actual heat rate was 20,088.783 kJ/kWh, showing a deviation of 4.2%. The condenser efficiency from the specification data was 89.75%, whereas the actual efficiency was 82.35%. The energy balance in the design condenser was 0.02721, and the actual energy balance was 0.02505.
Conclusions: The study concluded that the actual heat rate exceeded the tolerance limit of ASME PTC 6 by 2%. The condenser's performance was compromised due to a significant pressure difference, which affected the vacuum and overall efficiency of the equipment.
Limitations: This study is limited by the specific conditions of the spray-type condenser and focuses on a single unit at PT. X Geothermal Power Plant. The analysis did not fully account for real-world operational variations, and only one turbine unit was considered.
Contributions: This research contributes to understanding spray-type condenser performance, particularly in geothermal power plants. It provides valuable insights into improving the condenser's efficiency and operational performance.

Keywords: *Condenser Efficiency, Energy Balance, Geothermal Energy Systems, Power Plant Performance, Turbine Heat Rate*

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1. Introduction

Indonesia, with its abundant geothermal resources, has a significant potential to generate geothermal electricity. This is primarily due to the country's location on the Pacific Ring of Fire, where numerous active volcanoes provide a sustainable source of heat for power generation (Erdiwansyah et al., 2020; Pambudi, 2018). Geothermal energy is considered one of the most reliable and environmentally friendly renewable energy sources, and Indonesia is strategically positioned to tap into this potential (Alhamid et al., 2016; Ayuningtyas & Ilman, 2021). Geothermal power plants are crucial in helping meet the growing energy demands of the nation, while reducing dependence on fossil fuels and minimizing carbon emissions (Hakim et al., 2020; Shezan et al., 2018; Wahyuningsih et al., 2021).

In any power generation system, the condenser plays a crucial role in the cooling process. Its primary function is to condense the turbine exhaust steam back into water. This condensation process creates pressure within the turbine, which is essential for maintaining a vacuum in the condenser (Berlian Rms & Wahyuningsih, 2021; Drożyński, 2018; Kabeyi, 2019). The vacuum, in turn, enhances the efficiency of the power plant cycle, as it allows the turbine to operate at optimal levels. The performance of the condenser directly impacts the overall efficiency of the power plant and, consequently, the overall energy production (Gribin et al., 2018; Mil'man & Anan'ev, 2020; Syahrial & Sudono, 2021).

The condenser used in the Geothermal Power Plant at PT. X is a spray-type Direct Contact Unit 3. This type of condenser operates by using water as the cooling fluid and low-pressure turbine exhaust steam as the working fluid. As the exhaust steam passes through the condenser, it is cooled and condensed back into water (Evron et al., 2020; Ndukaife & Nnanna, 2019; Saputro & Soleha, 2021; Shortall & Kharrazi, 2017). However, an increase in the temperature of the cooling water can negatively affect the condensation process. When the temperature of the cooling water rises, it reduces the effectiveness of heat transfer and leads to an increase in pressure within the condenser (Kaplanoğlu et al., 2020; Ricardianto et al., 2021). This rise in pressure ultimately affects the vacuum in the condenser, which, in turn, reduces the overall performance and efficiency of the plant's equipment (Brodov et al., 2019; Harby et al., 2016; Parmenas, 2021)

The primary issue identified in this study is that an increase in the temperature of the cold water in the system directly influences the condensation process. This leads to higher pressure values within the condenser, which disrupts the vacuum and degrades the performance of the equipment (Ibrahim et al., 2019; Susanto et al., 2021). As a result, the efficiency of the power plant decreases, affecting its ability to produce electricity at optimal levels. The purpose of this study is to investigate the impact of varying condenser temperature and pressure on condenser performance, with the aim of identifying ways to optimize these conditions and, consequently, enhance turbine performance and overall productivity in the power plant (Setyawati et al., 2021; Susanto & Parmenas, 2021). By understanding how these factors influence system efficiency, this research seeks to contribute to the design of more efficient geothermal power plants, ultimately improving their energy output and sustainability.

2. Literature Review

2.1 Review General Geothermal Power Plant

A geothermal power plant is a power plant that uses geothermal energy as its driving force. Indonesia is blessed with abundant geothermal resources owing to the many volcanoes on its large islands; only the island of Kalimantan has no geothermal potential (Bustamante et al., 2016; Setyawati & Aristiyanto, 2021; Tontu, 2020). The advantages of this technology include its cleanliness, ability to operate at lower temperatures than nuclear power plants, and safety. In fact, geothermal energy is considered one of the cleanest energy sources compared to nuclear power, oil, and coal (Avci et al., 2020; Brunetti et al., 2019; Kuncoro & Harahap, 2021).

In general, geothermal power plants are divided into three types based on the geothermal working fluid used, namely (Soltani et al., 2019):

1. Vapor dominated system,
2. Flash steam system,
3. Binary cycle system.

The process in a geothermal power plant begins with steam generated from geothermal energy, which is used to rotate the turbine. If the steam temperature exceeds 370°C, the geothermal power plant uses a vapor dominated system. In this system, steam from the geothermal reservoir is directly used to rotate the turbine. If the steam temperature ranges from 170°C to 370°C, a flash steam system is used, where the steam still contains fluid and must first be separated using a flash separator before entering the turbine. In a binary cycle system, geothermal steam is utilized to heat a secondary working fluid in a heat exchanger, and the secondary fluid is then used to rotate the turbine (Heriyanto, 2021; Tomarov & Shipkov, 2017).

2.2 Principle Work Geothermal Power Plant

Generator Electricity Power Geothermal, in principle, is the same as Generator Electricity Steam Power, only on coal-fired power plant steam made in surface use boiler, whereas on Geothermal Power Plant steam originates from the geothermal reservoir (Kablar, 2019; Keke et al., 2021). If the fluid in the head well is in the form of phase steam, the steam can be streamed directly to the turbine, and the turbine will convert geothermal energy into kinetic energy that will rotate the generator to produce electricity (Agusinta et al., 2021; Brown et al., 2015). If the fluid geothermal exits the head well US mixture fluid two-phase (phase steam and phase liquid), then the fluid separation process is first carried out. This is possible by passing the fluid through a separator so that the phase steam is separated from the phase liquid. The steam produced by this separator then flows to the turbine (Satria, 2021; Shortall et al., 2015; Solihin, 2021).

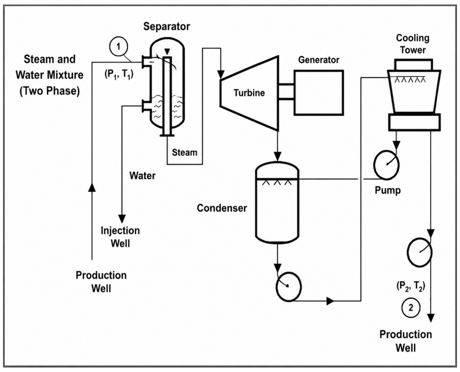

Figure 1. Flow Geothermal Power Plant Diagram

Based on Figure 1, geothermal fluids can be utilized even if the geothermal resource has a moderate temperature. For generator electricity, the generator electricity cycle binary (binary plant) was used. In the cycle generator, the secondary fluid (isobutane, isopentane, or ammonia) is heated by the geothermal fluid through a machine heat exchanger or a heat exchanger. Fluid secondary evaporates at a temperature lower than the boiling point of water at the same pressure. The secondary fluid flows to the turbine and condenses after use (Anggraini, 2021; Cai et al., 2017; Michaelides, 2016). before returning the geothermal fluids. A closed cycle in which the geothermal fluid is not mass-consumed, but only its heat (Aprillita & Perkasa, 2021; Jamero et al., 2018). is extracted by fluid second, and temporary fluid geothermal injection is returned into the The working principle of this reservoir is the system's function, which is based on components such as the separator, demister, turbine, condenser, and main cooling water pump. (MCWP), and cooling towers (Abdullah, 2021; Bošnjaković et al., 2019; Moya et al., 2018).

2.3 Spray Condenser

The condenser condenses the spent steam used to drive the turbine. The condenser used is a direct-contact condenser, where water from the cooling tower or condensation medium is sprayed directly onto the condenser. The nozzle then comes into direct contact with the spent steam, which turns the turbine. The steam condenses. and are released from the condenser along with the condensing medium. The non-condensable vapor or gas is sucked out by the ejector, which is called Non-Condensable Gas (NCG). NCG typically contains 85-90% wt CO_2, 3% wt H_2S, and the remainder is N_2 and other gases.

2.4 Parameter Performance Condenser

2.4.1 Heat Rate on Turbine

Turbine heat rate can be calculated using the following equation (ASME PTC 6):

$$TSR = \frac{m\frac{kg}{h}}{W(kW)} \quad \text{(kJ/kWh)} \quad (2.1)$$

$$THR = TSR \times h \quad \text{(kJ/kWh)} \quad (2.2)$$

Where:

- TSR: Turbine Steam Rate
- m: Main steam flow (kg/h)
- W: Output power (kWh)
- h: Output enthalpy (kJ/kg)

Condenser Efficiency

Efficiency calculation uses the output-input method:

$$\eta_{\text{cond}} = \frac{H_2}{H_1 + H_3} \times 100\% \quad (2.3)$$

2.5 Energy Balance

2.5.1 Steady Condition

$$W_s h_1 + W_c h_2 = W_{\text{cond}} h_3 \quad (2.4)$$

Where:

- W_s: Steam flow out of the turbine (kg/h)
- W_c: Cooling water flow (kg/h)

Actual Condition

$$m_2 h_2 + m_s h_s = m_3 h_3 \quad (2.5)$$

Where:

- m: Mass flow rate (kg/h)
- h: Enthalpy (kJ/kg)

3. Methodology

Research methodology is the stages of research that must be established before problem-solving can be undertaken. This allows for focused research and facilitates the analysis of problems (Sunarya et al., 2018). The flowchart of the research method is shown in Figure 2.

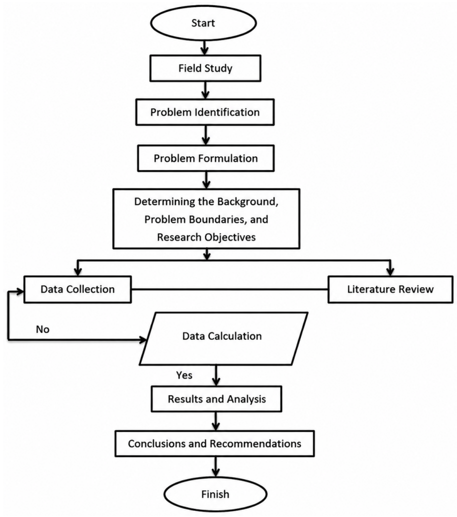

Figure 2. Research Flowchart

Based on Figure 2, the research methodology begins with the "Start" point, followed by a Field Study to identify the problem. The process continues with Problem Identification and Problem Formulation, leading to the step of Determining the Background, Problem Boundaries, and Research Objectives. The next stage involves Data Collection, which then branches into Data Processing or conducting a Literature Review. Afterward, the study proceeds to Results and Analysis, followed by drawing Conclusions and Recommendations, and concludes with the Finish point. This structured approach outlines the flow of the research process, ensuring each step is followed methodically.

3.1 Equipment Test

Data were collected using measuring instruments. The measuring instruments used are as follows: Transmitter Sensor, Temperature, pressure, and flow rate readings are used as a transmitter sensor as a tool for reading the steam turbine parameters. Gauge Sensor, In addition to using a transmitter for temperature and pressure readings, a gauge sensor is also used as a comparison of data read by the transmitter and actual field data.

3.2 Specification Design Turbine

Table 1. Specification Design Turbine Unit 3

Location	Turbine Area
Factory	Ansaldo Copelenti
Years	1992
Type	SCCF
Rating	55000 kW
Inlet steam pressure	6.5 bar
Exhaust steam pressure	0.102 bar
Temperature	162°C
Revolutions	3000 rpm
Number of stages	6+6

Based on Table 1, the turbine unit is located in the Turbine Area and was manufactured by Ansaldo Copelenti in 1992. The turbine is of the SCCF type and has a power rating of 55,000 kW. The steam entering the turbine has a pressure of 6.5 bar, while the exhaust steam pressure is 0.102 bar. The steam temperature entering the turbine is 162°C, and the turbine operates at a speed of 3000 rpm. The turbine unit features a 6+6 configuration, which indicates two stages, typically used in turbines for better efficiency at different pressures.

3.3 Spesification Design Condensor

Table 2. Specification Design Condenser Unit 3

Lokasi	Condenser Area
Internal volume	527 m^3
Size	1200x6500x6750 mm
Turbine exhaust Press	0.107 bar
Steam saturation temp	47°C
Gas weight content	1.5%
Cooling water inlet flow	3271.1
Cooling water inlet temp	27.6°C
Hot water outlet temp	44°C
Cooling water flow	2937.2 kg/s
Type of spray nozzle	Whirl jet

Based on Table 2, the condenser unit has an internal volume of 527 m^3 and dimensions of 1200x6500x6750 mm. The turbine exhaust pressure entering the condenser is 0.107 bar, with a steam saturation temperature of 47°C. The gas weight content in the condenser is 1.5%. The cooling water inlet flow is 3271.1 m^3/h, and the cooling water enters the condenser at a temperature of 27.6°C. After cooling, the hot water exits the condenser at 44°C. The condenser's cooling water flow is 2937.2 kg/s, and it uses a Whirl jet type spray nozzle, which ensures efficient cooling for the system.

3.4 Calculation Design

3.4.1 Heat Rate

By taking one sample from the research, the calculation is carried out as follows:

$$THR = TSR \times (h4)$$

Before calculating the Turbine Heat Rate, the Turbine Steam Rate is calculated to determine the specific steam consumption with the following formula:

$$TSR = \frac{m\frac{kg}{h}}{W(kW)}$$

Where:

- m = Main steam flow or steam used (kg/h)
- W = Output power or power generated (kW)

$$TSR = \frac{45000(\frac{kg}{h})}{600009(kW)}$$

$$TSR = 7.6$$

After obtaining the TSR value, the THR calculation is performed:

$$THR = TSR \times h4$$

$$THR = 7.6\,\text{kg/kWh} \times (2.759.595 \text{ kJ kg})$$

$$THR = 20.973.036\,\text{kJ/kWh}$$

3.4.2 Condenser Efficiency

The calculation uses the output – input method:

$$\eta_{cond} = \frac{H_2}{H_1 + H_3} \times 100\%$$

$$\eta_{cond} = \frac{2580.675\frac{kJ}{kg}}{2759.610\frac{kJ}{kg} + 115.711\frac{kJ}{kg}} \times 10$$

$$\eta_{cond} = \frac{2759.610}{2580 + 675} \times 100$$

$$\eta_{cond} = 89.75\%$$

3.4.3 Energy Balance (steady)

$$W_s \cdot h_1 + W_c \cdot h_2 = W_{cond} \cdot h_3$$

Where:

- $W_s : Steamflowrateoutoftheturbine(kg/h)$
- W_c : Cooling water flow rate (kg/h)
- W_{cond} : Condensate flow rate (kg/h)
- h : Enthalpy (kJ/kg)
- 450.000(kg/h)x2762,799(kJ/kg)+10.573.920(kg/h)x115,711 kJ/kg = 26.000 kg/h x 2580,675 kJ/kg
- 1,243,259,550 + 1,223,518,857 = 67,097,550
- 2,466,778,407 = 67,097,550
- 0,02721

3.5 Actual Calculation

3.5.1 Heat Rate

$$TSR = \frac{m\frac{kg}{h}}{W(kW)}$$

Where:

- m = Main steam flow (kg/h)
- W = Power output (kW)

$$TSR = \frac{43700\,(\frac{kg}{h})}{60000\,(\text{kW})} = 7.2833\,\text{kg/kWh}$$

After obtaining the TSR value, the THR calculation is continued:

$$THR = TSR \times h4$$

$$THR = 7.2833\,\text{kg/kWh} \times (2.758,198\,\text{kj/kg}) = 20.088,783\,\text{kJ/kWh}$$

Table 3. 24-Hour Heat Rate Calculation Results

Hour	**00.00**	**01.00**	**02.00**	**03.00**	**04.00**
Heat Rate	20088.87543	20182.4104	20185.60047	20182.4104	20142.7878

Hours	**05.00**	**06.00**	**07.00**	**08.00**	**09.00**
Heat Rate	20142.7878	20135.6411	20137.2325	20146.7225	20146.7225

Hour	**10.00**	**11.00**	**12.00**	**13.00**	**14.00**
Heat Rate	20100.72542	20198.22172	20194.29998	20143.5762	20089.66932

Hours	**15.00**	**16.00**	**17.00**	**18.00**	**19.00**
Heat Rate	20135.6411	20135.6411	20135.6411	20138.8239	20141.2037

Hours	**20.00**	**21.00**	**22.00**	**23.00**	**24.00**
Heat Rate	20141.9921	20187.18818	20141.2037	20141.9921	20187.97838

Based on Table 3, the heat rate calculations for each hour over a 24-hour period are shown. The values for heat rate (in kJ/kWh) are provided for each hour, demonstrating the variations throughout the day. For instance, at 00:00, the heat rate is 20,088.87543 kJ/kWh, while at 12:00, it stabilizes at 20,135.6411 kJ/kWh, and slightly drops to 20,141.9921 kJ/kWh by 24:00. The table presents how the turbine's performance fluctuates within a day, reflecting different operational conditions at different times. The fluctuations in heat rate can be attributed to changes in turbine and condenser performance, as well as varying external factors like temperature and system behavior throughout the day.

3.5.2 Condenser Efficiency

The calculation is performed using the output-input method:

$$\eta_{cond} = \frac{H_2}{H_1 + H_3} \times 100\%$$

$$\eta_{cond} = \frac{H_2}{H_1 + H_3} \times 100 = \frac{2580}{2580 + 675} \times 100 = 82.35\% \quad (3.8)$$

Table 4. 24-hour Efficiency Calculation

Hours	00.00	01.00	02.00	03.00	04.00	05.00	06.00
Efficiency Cond	82.35	82.23	82.04	82.15	82.03	82.03	82.27

07.00	08.00	09.00	10.00	11.00	12.00	13.00
82.24	82.31	82.32	82.21	82.26	82.35	82.42

14.00	15.00	16.00	17.00	18.00	19.00	20.00
82.45	82.26	82.29	82.29	82.28	82.21	82.41

21.00	22.00	23.00	00.00
82.22	82.25	82.15	82.13

Based on Table 4, the table presents the 24-hour condenser efficiency calculation, showing the efficiency of the condenser at each hour. The efficiency fluctuates slightly throughout the day, ranging between 82.15% and 82.45%. For example, at 00:00, the efficiency is 82.35%, and by 06:00, it has increased slightly to 82.27%. The table demonstrates the variations in the condenser's performance over the 24-hour period, with the efficiency tending to stabilize around 82.25% after several hours. The data shows a steady performance of the condenser with minor variations.

3.5.3 Energy Balance Calculation

$$\sum E_{in} = \sum E_{out}$$

$$m_1 \cdot h_1 + m_2 \cdot h_2 = m_3 \cdot h_3$$

437.000 kg/h x 2592,193 kJ/kg + 11.385.000 kg/h x 125,745 kJ/kg = 24.800 kg/h x 2.590,250 kJ/kg
1.132.788.341 + 1.431.606.825 = 64.238.200
2.564.395.166 = 64.238.200
= 0,02500

Table 5. 24-hour Equilibrium Calculation

Hours	00.00	01.00	02.00	03.00	04.00	05.00
Energy Equilibrium	0,0251	0,0249	0,0245	0,0249	0,0248	0,0253

06.00	07.00	08.00	09.00	10.00	11.00	12.00
0,0248	0,0249	0,0249	0,0250	0,0250	0,0247	0,0247

13.00	14.00	15.00	16.00	17.00	18.00
0,050	0,0260	0,0255	0,0255	0,0255	0,0251

19.00	20.00	21.00	22.00	23.00	00.00
0,0246	0,0249	0,0252	0,0245	0,0251	0,0244

Based on Table 5, the 24-hour energy equilibrium calculations show the variations in energy balance values at each hour. The energy equilibrium values fluctuate slightly throughout the day, ranging from 0.0246 at 19:00 to 0.0253 at 05:00. For example, at 00:00, the energy equilibrium value is 0.0251, while at 13:00, it increases to 0.0500. The fluctuations in energy equilibrium are typical of energy systems, where slight variations are observed depending on various operational factors. These values provide insight into the consistency and stability of energy flow during the day, which is crucial for optimizing the system's performance.

4. Results and Discussion

Based on the calculation results, a comparison was made between the design and actual conditions.

4.1 Comparison Heat Rate Design & Current

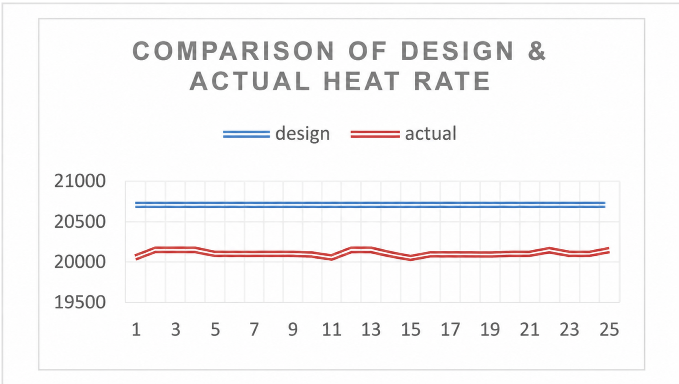

Figure 3. Chart Comparison HR Design & Actual

Based on Figure 3, it can be seen that the heat rate decreased under actual conditions. This was compared with the data design. The designed heat rate was as high as 20,973,036 kJ/kWh, whereas the actual heat rate was 20,088.783 kJ/kWh, with a deviation of 4.2%. The heat rate tolerance, based on ASME PTC 6, was 2%. Therefore, the actual heat rate exceeds the specified tolerance.

4.2 Comparison Efficiency Condenser Design & Current

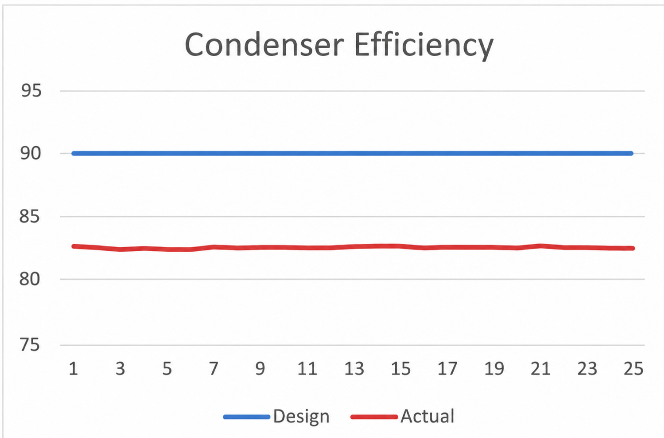

Figure 4. Chart Comparison Efficiency Design & Current

Figure 4 shows that the condenser efficiency under actual conditions was lower than that of the design. The design condenser efficiency was 89.75%, whereas the actual efficiency was 82.35%. The condenser design efficiency is better owing to the exhaust pressure, and the vacuum room conditions are better than the actual data.

4.3 Energy Equilibrium

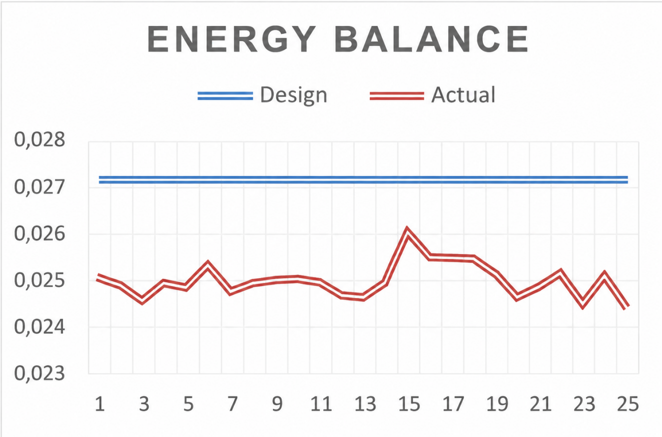

Figure 5. Chart Comparison Equilibrium Energy Design & Actual

Based on Figure 5, it can be seen that the energy balance in the condenser under actual conditions is smaller than that in the design. The design condenser energy balance value was 0.02721, whereas under actual conditions, it was 0.02505, with a difference of 0.00216. At 14:00, the system performance was very good, but at 00:00, it had the lowest value. This is caused by leaking insulation inside and outside the system, which causes the environmental temperature to affect the performance of the equipment.

5. Conclusions

From the results of the calculations and analyses that have been carried out on spray-type condenser unit 3, the following conclusions can be drawn: The heat rate of the design turbine was 20,696.965 kJ/kWh, whereas the actual heat rate was 20,046.144 kJ/kWh, with a deviation of 3.1%. The heat rate tolerance, based on ASME PTC 6, is 2%. Therefore, the actual heat rate exceeded the tolerance. The higher the heat rate, the lower the turbine efficiency, resulting in lower exhaust pressure and outlet temperature. The condenser efficiency under actual conditions was lower than that predicted by the design data. The design condenser efficiency is 89.65%, while under actual conditions it is 82.35%. The condenser design efficiency is higher owing to the exhaust pressure, and the vacuum room conditions are better than the actual data. The energy balance in the condenser under actual conditions is smaller when compared to the design. The energy balance value of the design condenser is 0.02721, while under actual conditions the actual value is 0.02505 and the difference is 0.00216. At 2:00 PM, the system performed very well, but at 12:00 AM, it had the lowest value. This is due to leaks in the insulation inside and outside the system, which can affect the equipment's performance. Further maintenance on the insulation inside and outside the system was performed to ensure that the environmental temperature did not affect the performance of the equipment, to maximize its performance and maintain the vacuum conditions in the room inside the condenser.

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Author Contributions

AP conceptualized and designed the study, performed the data collection, and analyzed the results. He also contributed to the writing of the original draft. BSH contributed to the analysis and interpretation of the data, and reviewed the manuscript. Both authors approved the final manuscript and agreed on the contents for publication.

Conflicts of Interest

The authors declare that there is no conflict of interest regarding the publication of this study. This research was conducted independently, and no financial or personal relationships influenced the results or interpretation of the findings.

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