

Analysis of the Impact of Sliding Pressure Operation on the Efficiency of a 610 MW Subcritical Steam Power Plant Boiler

¹*Hamam Suhaily, ²*Berkah Fajar Tamtomo Kiono, ³Sri Widodo Agung Suedy

¹Master of Energy, Diponegoro University, Indonesia

²Mechanical Engineering, Diponegoro University, Indonesia

³Biology, Diponegoro University, Indonesia

*Corresponding Authors E-mail: hamamsuhaily@students.undip.ac.id and fajarberkah10@gmail.com

Abstract - The 610 MW subcritical power plant was originally designed to operate at a constant base load with fixed boiler steam pressure. However, due to grid oversupply conditions, the plant frequently operates at partial loads, resulting in increased inefficiencies caused by throttle losses, particularly during minimum load operations, which significantly reduce high-pressure (HP) turbine efficiency. This study investigates the impact of sliding pressure operation, which involves varying boiler steam pressure with load, on overall plant efficiency compared to constant pressure operation. A virtual plant simulation was employed to model the heat and mass balance of the power plant and simulate operational scenarios under varying steam pressures. The optimal simulation results were validated through direct testing on the plant, followed by performance analysis. The results reveal that sliding pressure operation improves HP turbine efficiency by 5-7%, reduces the net plant heat rate (NPHR) by 57 kcal/kWh, and enhances overall thermal efficiency by 0.35% at 57% load (350 MW).

Keywords: Constant pressure, Sliding Pressure, Subcritical Boiler, Turbine efficiency, Heat rate.

I. INTRODUCTION

The oversupply of electricity in the grid has compelled power plants, previously operated as base load units running at full capacity, to adapt by operating at partial loads to meet grid demands. This shift to partial load operation significantly impacts the overall performance of power plants, with the most pronounced effect observed in the efficiency of high-pressure (HP) turbines.

A subcritical power plant with a net installed capacity of 610 MW is designed to operate in constant pressure mode, where the boiler steam pressure remains unchanged regardless of the load. However, under partial load conditions, this mode

results in significant throttle losses at the control valves, which in turn reduces turbine efficiency [1].

The Electric Power Research Institute (EPRI) has indicated that variable pressure operation can improve HP turbine efficiency [2]. Similarly, KC Cotton, in his book, highlights that variable pressure or sliding pressure operation performs better at loads below 60%, delivering improved steam temperature compared to constant pressure modes under similar conditions [3].

Energy and exergy analysis were conducted on supercritical power plants by comparing two operating modes: sliding pressure and constant pressure. The study focused on partial loads of 100%, 80%, and 60% of the normal continuous rating (NCR). The results show that exergy destruction under sliding pressure conditions for loads of 100%, 80%, and 60% are 47.66 MW, 37.88 MW, and 28.94 MW, respectively. In contrast, under constant pressure conditions, exergy destruction for the same loads are 49.16 MW, 43.22 MW, and 43.92 MW, respectively. Additionally, the power input to the Boiler Feed Pump (BFP) showed a significant reduction of 9.39%, 21.52%, and 42.5% at loads of 100%, 80%, and 60%, respectively, under sliding pressure operation compared to constant pressure operation [4].

The performance decline at partial load conditions poses a critical challenge to power plant efficiency and demands immediate attention. Accordingly, this study aims to assess the effectiveness of sliding pressure operation, compare it with constant pressure modes, and identify critical parameters necessary to maintain stable and efficient plant operation.

II. MATERIAL AND METHOD

This study compares the efficiency of a power plant operating under the constant pressure modes with its performance after transitioning to the sliding pressure modes. Constant pressure operation maintains a fixed main steam pressure regardless of the load, whereas sliding pressure

operation adjusts the main steam pressure according to the load. An illustration of the transition from constant pressure to sliding pressure operation is presented in Figure 1.

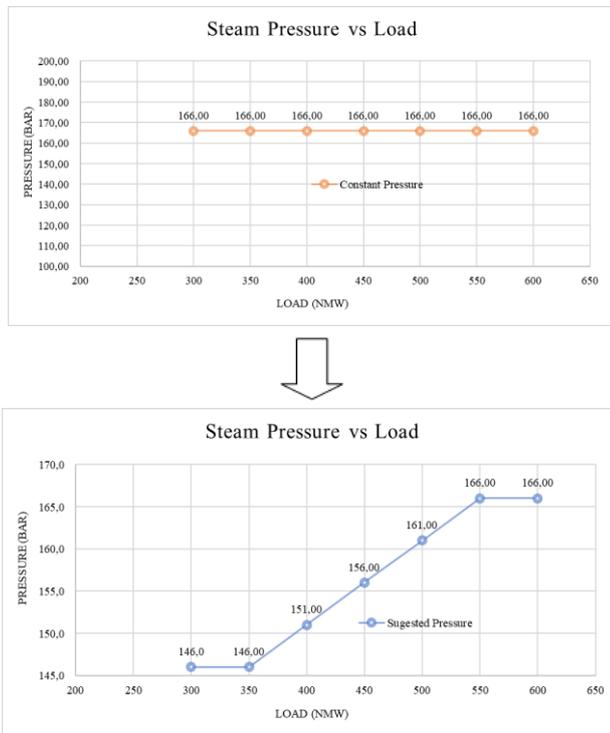


Figure 1: pressure transition from constant pressure to sliding pressure

This study is a quantitative research conducted using simulation methods on power plants. The research begins by simulating the operational processes of the power plant using the EtaPRO Virtual Plant software to analyze performance changes resulting from the implementation of the sliding pressure operation mode [5]. Subsequently, the operation modes are tested directly on the power plant to observe changes in critical operating parameters. The performance values of the power plant are expected to be the main output of this research. The sequence of the research process is shown in Figure 2.

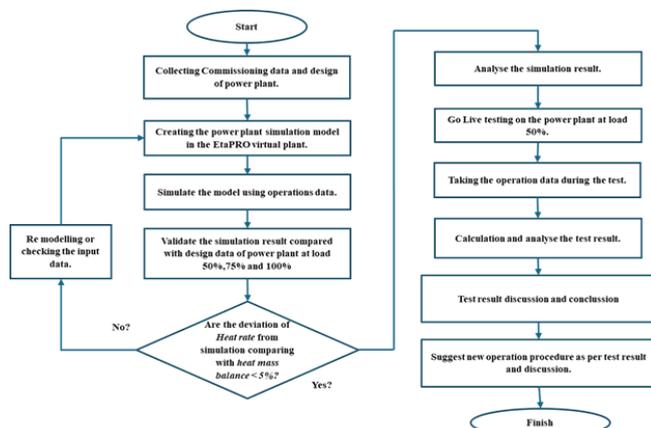


Figure 2: Flow chart of the research

The computer simulation modeling was developed using EtaPRO Virtual Plant software provided by Toshiba. The heat and mass balance documents from the power plant were used as references for building the simulation model. The main equipment components included in the virtual plant model are the boiler, high-pressure turbine (HP turbine), intermediate-pressure turbine (IP turbine), low-pressure turbine (LP turbine), condenser, condensate pump, boiler feed pump (BFP), feedwater heater, and deaerator. The results of the simulation model are presented in Figure 3. The input data for the modeling are based on the design data of the main equipment from the power plant. The main equipment included in the model is listed in Table 1 [6].

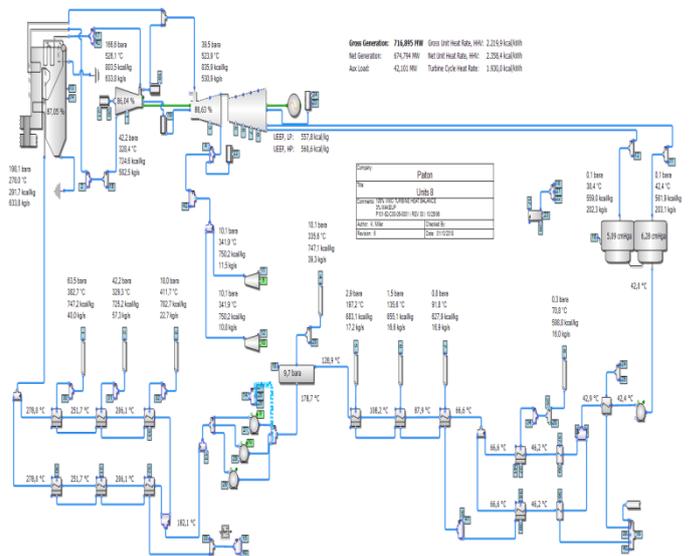


Figure 3: Power Plant Modelling Using Virtual Plant

Model validation is essential to ensure that the computer simulation model accurately represents real-world conditions. This is a critical step in simulation model development, as it ensures the reliability of the model for accurate analysis or prediction purposes. To validate the accuracy of the computer simulation model, the net plant heat rate and turbine heat rate from the simulation results were compared with the design values provided in the heat and mass balance documents of the power plant [6]. The deviation between the two values is required to not exceed 5% [7].

If the deviation exceeds 5%, the simulation model is considered inaccurate. The 5% validation threshold is based on references from similar studies that utilized computer simulations to model steam power plants. The comparison between the design heat rate and the simulation heat rate is presented in Table 2.

Table 1: The main power plant equipment used during the simulation

No	Equipment	Specification	Quantity	Manufacturer
1	Boiler	Sub-Critical Coal tangential firing boiler, Cap. 615 MW	1	ABB-CE/Alstom
2	Turbine	Tandem compound 4 flow, Consist of HP Turbine, IP Turbine, LP Turbine A & B, Rating 670 MW/3000 rpm.	1	GE Power System
3	Condenser	Dual pressure twin shell double flow divided waterboxes, Heavy duty 1,4807 x 10 ⁹	1	Hitachi Machinery & Engineering
4	Condensate Pump	Vertical barrel type multi stage souble suction, Cap. 0,271 m ³ /s	3	EBARA Corporation
5	Low Pressure Feed water heater	low pressure Shell & U-Tube one zone	5	Struthers Industries
6	Deaerator	Open Feed water heater	1	Babcock & Wilcox
7	Boiler Feed Pump	Turbine driven multistage centrifugal pump, Cap. 6450 GPM	2	EBARA Corporation
8	High pressure feed water heater	High pressure Shell & U-Tube three zone	6	Struthers Industries
9	Generator	Rating 846500 KVA	1	GE Power System

Table 2: Validation of design comparison with simulation model results

Load %	MW	Steam Press. Bar	Steam Temp. Deg. C	Net Plant Heat Rate (kcal/kwh)			Turbine Heat Rate (Kcal/kwh)		
				Design	Simulation	Dev.	Design	Simulation	Dev.
50	330	166,6	535	2583,4	2549,4	1,3%	2076,0	2030,1	2,3%
75	502	166,6	535	2409,5	2389,3	0,8%	1962,8	1934,2	1,5%
100	670	166,6	535	2365,3	2350,1	0,6%	1934,9	1917,9	0,9%

The deviation of the net plant heat rate between the simulation and the design values was 1.3%, while the turbine heat rate deviation was 2.3%. At 75% load, the deviation in net plant heat rate was 0.8%, and the turbine heat rate deviation was 1.5%. For the simulation at 100% load, the deviation of the net plant heat rate was 0.6%, and the turbine heat rate deviation was 0.9%. Based on these results, all deviations at the three load levels were below the 5% threshold. Therefore, it can be concluded that the simulation model is valid and can be used for further analysis.

III. RESULTS AND DISCUSSIONS

This study employs computer simulations using the EtaPRO Virtual Plant software to model the heat and mass balance of a power plant. The model is subsequently used to simulate the operational modes under the sliding pressure modes. The results from the computer simulation serve as a reference for direct testing at the respective power plant. The plant's performance during the testing phase is observed and analyzed to develop operational recommendations for the sliding pressure mode.

3.1 Simulation Result

The analysis of computer simulations using the EtaPRO Virtual Plant software is based on two power plant performance criteria: high-pressure turbine efficiency (HP Turbine Efficiency) and net plant heat rate (NPHR). The results of the simulation are presented as follows.

3.1.1 High Pressure (HP) Turbine Efficiency

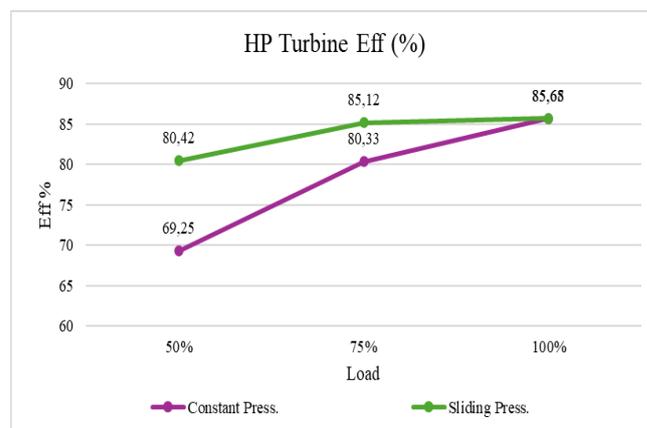


Figure 4: The trend of HP turbine efficiency based on simulation result

The results of the computer simulation shown in Figure 4 indicate a significant difference in the high-pressure turbine efficiency (HP Turbine Efficiency) graph. The sliding pressure operational mode demonstrates higher HP turbine efficiency compared to the constant pressure mode. At 50% load, the HP turbine efficiency for the sliding pressure mode is 80.42%, whereas the constant pressure modes shows an efficiency of only 69.25%, indicating an improvement of 11.17%. At 75% load, when the inlet steam pressure is increased in the sliding pressure mode, the efficiency difference between the two modes narrows to 4.79%. Meanwhile, at 100% load, there is no difference in the inlet steam pressure to the high-pressure turbine between the sliding pressure and constant pressure modes, resulting in identical HP turbine efficiencies.

3.1.2 Net Plant Heat Rate (NPHR)

The value of the Net Plant Heat Rate (NPHR) is influenced by the Turbine Cycle Heat Rate (TCHR) and boiler efficiency (%). In the computer simulations conducted, the boiler efficiency was set as an input parameter with a value of 87.05%, consistent with the boiler design. Therefore, it can be concluded that only changes in TCHR affect the simulated NPHR values. The trend of the simulated NPHR is shown in Figure 5.

The trend of the Net Plant Heat Rate (NPHR) resembles that of the Turbine Cycle Heat Rate (TCHR), indicating that the sliding pressure operational mode has a positive impact when the unit operates below 75% load. However, when the unit operates above 75% load, the NPHR values become worse compared to the constant pressure operational mode.

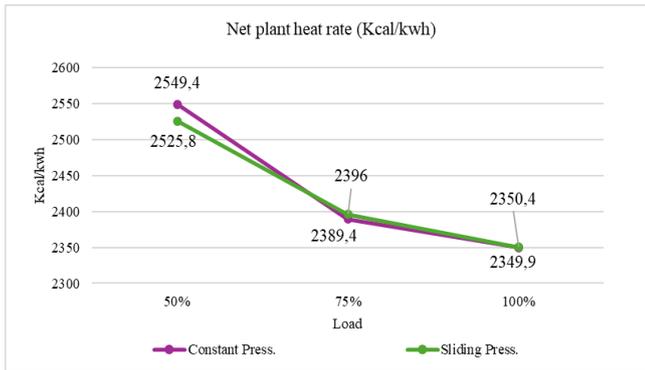


Figure 5: The trend of NPHR based on simulation result

Based on the analysis of all operational and performance parameters, the computer simulation results indicate that the sliding pressure operational mode positively impacts the high-pressure turbine efficiency (HP Turbine Efficiency) compared to the constant pressure modes when the power plant operates at partial loads.

In terms of the Net Plant Heat Rate (NPHR), the simulation results show that the sliding pressure modes enhances plant performance when applied at loads below 75%. However, when the plant operates above 75% load, the constant pressure mode delivers slightly better performance.

These computer simulation results provide an overview of the operational parameters and performance conditions when the sliding pressure mode is tested directly at the power plant.

3.2 Direct Test on the Power Plant

The second phase of this study involved conducting a direct test on the power plant to observe the effects of applying the sliding pressure operational modes at partial load conditions. The low load selected was 350 MW (57%), which represents the technical minimum load allowed for the plant to operate under normal conditions.

Previous computer simulation results indicated that the sliding pressure mode improves high-pressure turbine efficiency (HP Turbine Efficiency) and the Net Plant Heat Rate (NPHR). A more detailed discussion of the results from the direct test on the power plant is as follows:

3.2.1 HP turbine efficiency

The evaluation of high-pressure turbine (HP turbine) efficiency was conducted to analyze the impact of implementing sliding pressure operation modes on a power plant. This study utilized a real-time performance monitoring system supported by the EtaPRO platform, allowing a direct comparison of HP turbine efficiency data distributions

between constant pressure operation modes (166 Bar) and sliding pressure operation mode. As shown in Figure 6, the sliding pressure operation mode demonstrated improved HP turbine efficiency compared to the constant pressure operation mode. An efficiency improvement of 5–7% was observed at a net load of 350 MW when using the sliding pressure operation modes. These findings indicate that the results from computer simulations align with those of direct testing, confirming enhanced plant performance through improved HP turbine efficiency.

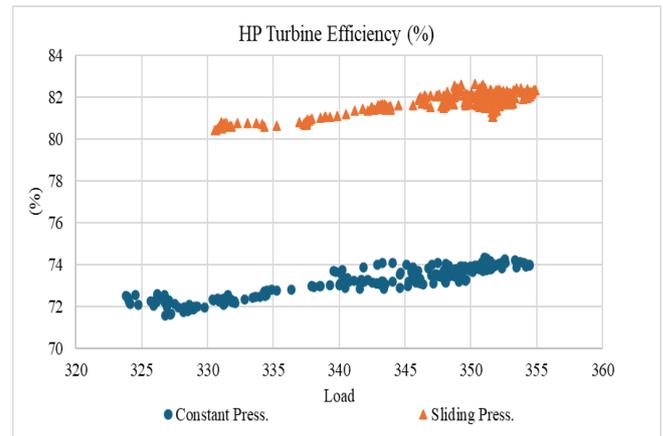


Figure 6: The trend of HP Turbine Efficiency based on direct test result

3.2.2 Net Plant Heat Rate (NPHR)

The net plant heat rate (NPHR) was evaluated to assess the overall performance improvement of the power plant [9]. The analyzed data were derived from the EtaPRO real-time performance monitoring system, which can be downloaded and further analyzed. Based on the NPHR evaluation shown in Figure 7, the implementation of the sliding pressure operation mode at a net load of 350 MW with a steam pressure of 125 bar improved the NPHR by 57 kcal/kWh, equivalent to approximately 2.32% of the power plant's design NPHR.

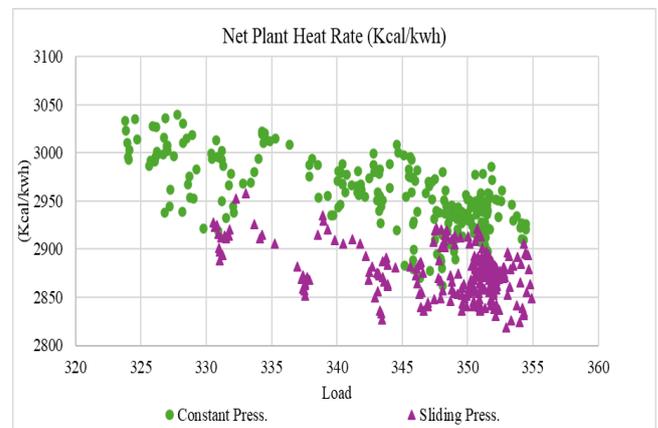


Figure 7: The trend of NPHR based on direct test result

3.2.3 Thermodynamic Analysis

Thermodynamic analysis was conducted to evaluate thermal efficiency based on mass and energy balance. To perform the thermal cycle analysis, data on the conditions at each major component of the power plant were required. The data at each point were collected during testing, where the data representing constant pressure operation were taken at a load of 350 MW with a pressure of 166 bar, while the data for sliding pressure operation were taken at the same load with a steam pressure of 125 bar. The collected data included pressure and temperature, which were then processed using a steam calculator to determine the enthalpy and entropy values at each process point. The process flow scheme of the power plant is illustrated in Figure 8.

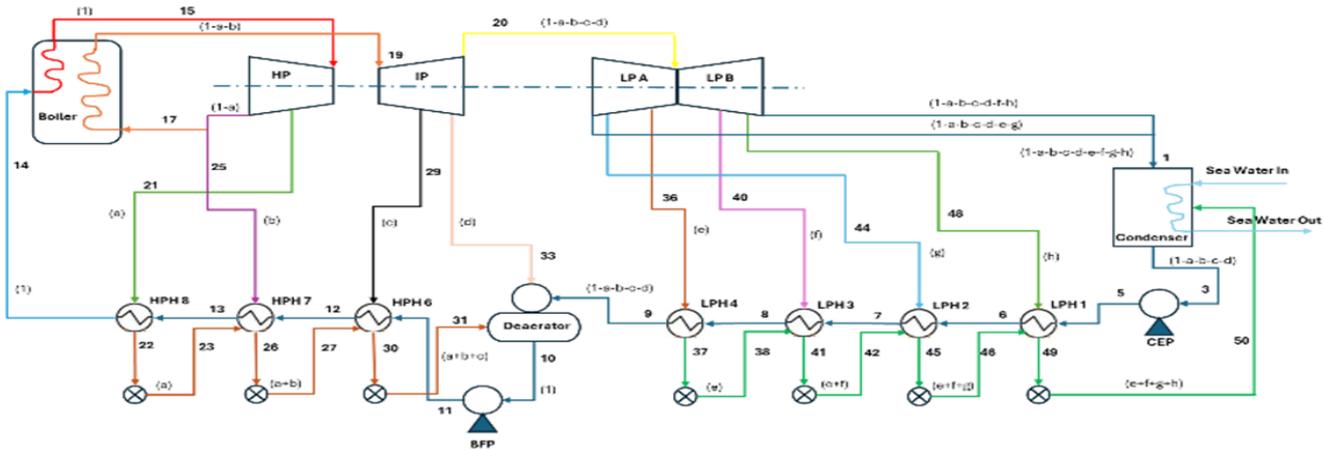


Figure 8: Process scheme of power plant

To evaluate thermal efficiency, data on the mass fraction of turbine steam extractions are required, as indicated in Figure 8 with the codes a, b, c, d, e, f, g, and h. These data are used to calculate the net power output and the heat input required by the power plant, enabling the calculation of thermal efficiency. By applying the principles of energy and mass balance [8], the mass fractions can be calculated using equations derived from the process analysis of the cycle shown in Figure 8. The resulting equations are as follows:

✓ Steam Extraction from HP Turbine (a and b).

$$a = \frac{h_{14} - h_{13}}{h_{21} - h_{23}}$$

$$b = \frac{(h_{13} - h_{12}) - a(h_{23} - h_{27})}{h_{25} - h_{27}}$$

✓ Steam Extraction from IP Turbine (c dan d)

$$c = \frac{(h_{12} - h_{11}) - (a + b)(h_{27} - h_{31})}{h_{29} - h_{31}}$$

$$d = \frac{(h_{10} - h_9) + (a + b + c)(h_{31} - h_9)}{h_{33} - h_9}$$

✓ Steam extraction from LP Turbine (e,f,g and h)

$$e = \frac{(1 - a - b - c - d)(h_9 - h_8)}{h_{36} - h_{38}}$$

$$f = \frac{(1 - a - b - c - d)(h_8 - h_7) + e(h_{38} - h_{42})}{h_{40} - h_{42}}$$

$$g = \frac{(1 - a - b - c - d)(h_7 - h_6) + (e + f)(h_{42} - h_{46})}{h_{44} - h_{46}}$$

$$h = \frac{(1 - a - b - c - d)(h_6 - h_5) + (e + f + g)(h_{46} - h_{50})}{h_{48} - h_{50}}$$

✓ The energy produced by *HP turbine* (W_{hp}) is expressed in kj/kg.

$$W_{hpt} = (h_{15} - h_{21}) + (1 - a)(h_{21} - h_{17})$$

✓ The energy produced by *IP Turbine* (W_{ip}) is expressed in kj/kg.

$$W_{ipt} = (1 - a - b)(h_{19} - h_{29}) + (1 - a - b - c)(h_{29} - h_{33}) + (1 - a - b - c - d)(h_{33} - h_{20})$$

✓ The energy produced by *LPA Turbine* (W_{LpA}) is expressed in kj/kg.

$$W_{LpA} = (1 - a - b - c - d)(h_{20} - h_{36}) + (1 - a - b - c - d - e)(h_{36} - h_{44}) + (1 - a - b - c - d - e - g)(h_{44} - h_1)$$

✓ The energy produced by *LPB Turbine* (W_{LpB}) is expressed in kj/kg.

$$W_{LpB} = (1 - a - b - c - d)(h_{20} - h_{40}) + (1 - a - b - c - d - f)(h_{40} - h_{48}) + (1 - a - b - c - d - f - h)(h_{48} - h_1)$$

✓ The Energy needed by *CEP* (W_{CEP}) is expressed in kj/kg.

$$W_{CEP} = (1 - a - b - c - d)(h_5 - h_3)$$

✓ The Energy needed by *BFP* (W_{BFP}) is expressed in kj/kg.

$$W_{BFP} = (h_{11} - h_{10})$$

✓ The heat energy Q_{in} absorbed by the system is expressed in kJ/kg

$$Q_{in} = (h_{15} - h_{14}) + (1 - a - b)(h_{19} - h_{17})$$

✓ The net work W_{netto} produced by system is expressed in kj/kg.

$$W_{netto} = W_{hpt} + W_{ipt} + W_{LpA} + W_{LpB} - W_{CEP} - W_{BFP}$$

✓ Thermal efficiency (%)

$$\eta_{thermal} = \frac{W_{netto}}{Q_{in}}$$

To facilitate understanding of the thermodynamic conditions, a T-S (temperature-entropy) diagram of the power plant cycle is provided as Figure 10.

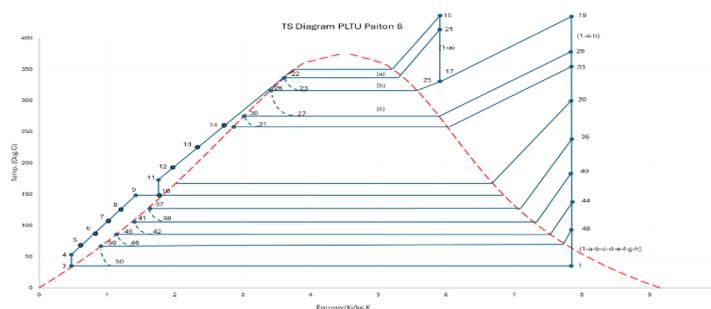


Figure 10: T-S diagram of power plant

The enthalpy data for each state, as illustrated in the T-S diagram of the power plant cycle, are as follows (table 3).

Table 3: Specific enthalpy under operating conditions (a) constant pressure (b) sliding pressure

(a)				(b)			
Point	H (Kj/Kg)						
h1	2485,71	h25	2992,80	h1	2489,16	h25	3025,47
h3	168,21	h26	939,85	h3	168,27	h26	938,26
h4	180,11	h27	939,85	h4	180,94	h27	938,26
h5	183,24	h29	3286,04	h5	184,12	h29	3290,78
h6	238,74	h30	745,43	h6	239,47	h30	744,69
h7	293,33	h31	745,43	h7	294,98	h31	744,69
h8	369,76	h33	3103,99	h8	372,29	h33	3111,32
h9	456,75	h36	2855,30	h9	457,67	h36	2864,49
h10	625,17	h37	470,35	h10	626,93	h37	472,75
h11	644,02	h38	470,35	h11	639,81	h38	472,75
h12	766,64	h40	2755,30	h12	764,11	h40	2760,13
h13	941,96	h41	390,32	h13	937,44	h41	391,78
h14	1042,22	h42	390,32	h14	1038,05	h42	391,78
h15	3384,09	h44	2637,74	h15	3428,98	h44	2641,96
h17	2999,84	h45	300,93	h17	3033,40	h45	302,33
h19	3520,72	h46	300,93	h19	3523,01	h46	302,33
h20	3103,99	h48	2607,83	h20	3111,32	h48	2608,64
h21	3088,06	h49	242,13	h21	3121,74	h49	242,15
h22	1041,49	h50	242,13	h22	1039,10	h50	242,15
h23	1041,49			h23	1039,10		

Subsequently, by inputting the data from Table 3 into the aforementioned equations, the mass fraction of turbine steam extractions and the thermal efficiency values were calculated for the sliding pressure operation test compared to the constant pressure operation. The mass fraction values of turbine steam extractions are presented in Table 4, while the thermal efficiency calculations are shown in Table 5.

Table 4: The calculation results of turbine extraction fraction

Fraksi Ekstraktion	Equipment	Nilai	
		Constant Pres	Sliding Pres
a	HPH 8	0,0489896819	0,0483105521
b	HPH 7	0,0829764736	0,0807093470
c	HPH 6	0,0381645913	0,0390106548
d	DEAERATOR	0,0821744356	0,0819589519
e	LPH 4	0,0272708329	0,0267720916
f	LPH 3	0,0250885025	0,0253982360
g	LPH 2	0,0197622698	0,0197373127
h	LPH 1	0,0193348253	0,0193699628

Table 5: The calculation results of energy in the main equipment and thermal efficiency

Energy Equipment	Nilai (kj/kg)	
	Constant Press.	Sliding press.
W HP Turb	212,12	223,16
W IP Turb	354,79	351,57
WLP A Turb	449,21	453,57
WLP B Turb	453,16	457,43
Wcep	11,23	11,89
Wbfp	18,85	12,88
Qin	2794,00	2817,37
Daya Netto	1439,20	1460,96
Thermal Eff	51,51%	51,86%

Based on the calculation results, it was observed that at a net load of 350 MW, the sliding pressure operation mode produced better thermal efficiency compared to the constant pressure operation mode. The thermal efficiency increased from 51.51% to 51.86%, representing an improvement of 0.35%.

IV. CONCLUSION

After conducting research on the effects of sliding pressure operation mode on the performance of steam power plants, the study began with simulations of sliding pressure and constant

pressure operation mode using the EtaPRO Virtual Plant application. This was followed by direct testing on a steam power plant. Based on the research, the following conclusions were drawn:

1. The results of computer simulations and direct testing on the power plant indicate that under low-load conditions (350 MW), the sliding pressure operation mode improves the power plant's performance compared to the constant pressure operation mode.
2. Direct testing on the power plant demonstrates that the sliding pressure operation mode increases HP turbine efficiency by 5–7%, thereby improving the net plant heat rate by 57 kcal/kWh when operating at a load of 350 MW.
3. Thermodynamic analysis at a load of 350 MW shows that the sliding pressure operation mode increases thermal efficiency by 0.35% compared to the constant pressure operation mode.

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AUTHORS BIOGRAPHY



Hamam Suhaily,
Master of Energy, Diponegoro
University, Indonesia.



Berkah Fajar Tamtomo Kiono,
Mechanical Engineering,
Diponegoro University,
Indonesia.



Sri Widodo Agung Suedy,
Biology, Diponegoro University,
Indonesia.

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