Heat Transfer Analysis in High Pressure Heater (HPH) of Circulating Fluidized Bed (CFB) Boiler on the Steam Turbine PLTU Teluk Sirih Unit 2

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Abstract: ***A pressure heater (HPH) is a heat exchange device that is widely used in industry, including power plants such as PLTU Teluk Sirih Unit 2, which uses steam as the working fluid. HPH works by changing the phase and temperature of the fluid with other fluids, such as the results of steam extraction from a turbine. HPH functions as a preheater to feed water before it enters the boiler. By operating the HPH, the boiler workload will be lighter because the water entering the boiler has a high temperature. This research was conducted to determine the initial performance of HPH - 1 and HPH - 2 shell and tube heat exchanger types and determine work efficiency performance using thermodynamic and heat transfer calculation methods. From the results of the research carried out, it was found that the increase in terminal temperature difference (TTD) in the HPH 1 preheater was 2, 453 °C, and HPH 2 was 13, 097 °C. The increase in drain cooler approach (DCA) for the initial heater (HPH 1) is 18.2 °C; for the initial heater 2 (HPH 2), it is 32.77 °C. The increase in temperature rise (TR) at HPH 1 was 3.24 °C; for HPH 2 it was 9.42°C. In the log mean temperature difference, there was an increase in HPH 1 of 26.84 °C and in HPH 2 of 39.68 °C. Heat exchanger effectiveness decreased at HPH 1 by 0.079% and at HPH 2 by 0.169%.***

Keywords: Heat exchanger, high - pressure heater, log mean temperature difference

1. Introduction

The Electric Steam Power Plant (ESPP) Teluk Sirih Unit - 2 has a capacity of 2 x 112 MW and has several Feedwater Heater components, one of which is a Preheater. This equipment is a shell and tube type heat exchanger which functions to provide high - pressure preheating of the fill water. The extraction steam flows in the shell, while the filling water flows in the tubes. Of the two existing units, each unit has two levels of Preheater, namely High - Pressure Heater (HPH) - 1 and HPH - 2. The boiler feed water will first be heated in HPH 2 and then go to HPH 1. Steam extraction at HPH - 1 and HPH 2 comes from the High - Pressure turbine [6], [7].

The performance of the preheater can affect the efficiency of water heating in the boiler if the heat transfer capability of the feedwater heater is not good. This can be caused by several factors, such as damage to the pipe or fouling on the pipe surface, and others [11].

A decrease in the performance of the preheater or high - pressure heater will affect the increase in coal consumption for the steam formation process in the boiler and affect the overall efficiency of the generating cycle. To determine the performance of this tool, it is necessary to carry out a mathematical analysis of the various parameters monitored during operation, based on commissioning (trial) data for HPH 1 FW Outlet Temperature 232.99 ºC and HPH 2 FW Outlet Temperature 203.64 ºC [7]. However, currently it is visually visible that it has decreased. Based on these observations, appropriate action must be taken immediately so that the preheater can operate normally if a decrease in performance occurs.

2. Literature Survey

High Pressure Heater (HPH) is a high - pressure water heater that uses hot steam from turbine extraction as a heating medium before entering the economizer in the boiler. This component consists of a cylindrical shell on the outside and a number of tubes (tube bundles) on the inside where the fluid temperature inside the shell is different from the temperature inside the tube, resulting in heat transfer between fluid flows. As a component in The Electric Steam Power Plant (ESPP), HPH has a very important role in maintaining the temperature of the fill water entering the boiler. The better the efficiency value of the HPH, the greater the efficiency of the boiler so that it can save daily operational costs for the the electric steam power plant [8].

In industrial applications, HPH is an example of a heat exchanger known as a heat exchanger (HE). HE is a heat exchange device that functions to change the phase temperature of a type of fluid. This process occurs by utilizing the heat/heat transfer process from a high - temperature fluid to a low - temperature fluid [6].

In a circulating fluidized bed (CFB) type furnace boiler, the gas speed is faster than in a fluidized bed boiler with a bubbling system [8]. So that the density in the furnace, namely the bed material, can be lifted and flow, a minimum gas velocity value is needed so that the particles can be lifted and leave the furnace [10].

Burning of solid fuel in the furnace occurs due to turbulence, colliding with the burning medium, namely sand. The remaining solid fuel that has not been burned will circulate through the cyclone/compact separator [12].

Several studies that have been carried out regarding HPH Heat Exchanger (HE) analysis are as follows:

Min Li and Alvin [1] conducted a thermodynamic performance analysis on a u - Tube borehole ground heat exchanger using the entropy generation minimization method. This research develops two explicit expressions for length dimensions and Reynolds number by minimizing entropy generation. The research results conclude that the optimal thermodynamic parameters of the borehole heat exchanger can be determined using the entropy generation minimization method.

Bizzi [3] carried out design calculations for the dimensions of a shell and tube - type heat exchanger using the Heat Transfer Research Inc. (HTRI) computerized analysis method and manual calculation analysis methods. The results of the dimensional calculation analysis show that the designed heat exchanger meets the minimum requirements for the specified fouling factor. The quality of the heat exchanger will increase in proportion to the decreasing fouling factor value, decreasing pressure drop value, and the size of the heat exchanger dimensions.

Veriyawan [2] carried out optimization on the design of a shell and tube type heat exchanger using the particle swarm optimization (PSO) algorithm. The aim is to optimize the overall heat transfer coefficient value by getting the best value. From the optimization results of 3 HE units, the A and U values for HE E - 1111 were 472 W/m^{2}C and 289 m^2, respectively; for HE E - 1107, 174 W/m^{2}C and 265 m^2; and for HE E - 1102, 618 W/m^{2}C and 574 m^2. The overall heat transfer value that has been optimized is in accordance with the objective function, and it can be said that HE Shell and Tube reaches the optimal point.

Sudrajat [4] carried out HE analysis to determine the effect of fouling on the actual heat transfer rate and HE effectiveness. Analysis is carried out by calculating the required parameters. The results of calculations and analysis show that there is a decrease in the heat transfer rate of 0.411kW, or 19.45%, equivalent to the energy produced from using 0.036 l of diesel fuel for 1 hour. Fouling that occurred increased by 0.561 m^2 K/kW. Meanwhile, effectiveness decreased by 3.7%.

Devia and Didik [5] conducted research that focused on the effective heat transfer rate, overall heat transfer coefficient, effectiveness value, and the amount of pressure drop on HPH. The research results show that the effective heat transfer rate in the HPH occurs at a value of 37.013kW due to an imbalance between the shell and tube sides. The overall heat transfer coefficient value at HPH is 502.48 because it is influenced by the convection coefficient that occurs on the shell side and tube side with the assumption that radiation heat transfer is ignored. The effectiveness value is 0.47; there is a decrease in performance caused by the age of the equipment and lack of equipment maintenance. Pressure drop of 23, 498.06 Pa.

Based on the research above, there are many problems that occur in heat exchangers that are caused by several factors, including pressure drop, effective heat transfer, impurity factors, etc. So heat exchanger performance evaluation needs to be carried out to improve the performance of the heat exchanger so that it meets the expected operating conditions.

3. Methods/Approach

In this research, the method used is a thermodynamic and heat transfer analysis method using data obtained from operational data and HPH performance configuration data and other supporting data so that its performance can be determined.

a) Preheater Installation on CFB Boilers

The HPH preheater installation used to increase efficiency at the Teluk Sirih PLTU can be shown with a pipe instrument diagram (PI and D) as shown in figure 3.1.

Figure 3.1: PLTU Teluk Sirih Preheater Installation Line (HPH)

b) Preheater Specification Data (HPH)

To evaluate the performance of the preheater (HPH), it is necessary to know the specification data that shows the characteristics of the equipment; in Figure 3.2 the HPH is shown, and in Table 3.1 the HPH - 1 and HPH - 2 specifications.

Figure 3.2: High Pressure heater (HPH) PLTU Teluk Sirih Unit 2

Table 3.1: Preheater Specification (HPH)

Items	Unit	HPH 1	HPH 2
Extraction Steam Pressure	Mpa	3, 213	1, 823
Extraction Steam Temperature	°C	395, 95	322, 59
Extraction Steam Flow Rate	T/h	23, 59	31, 06
Water Temperature of heater inlet	°C	203, 64	160, 08
Water Temperature of heater outlet	°C	232, 99	203, 64

c) Analysis Steps

The analysis steps carried out are as follows:

1) Calculation of Terminal Temperature Difference (TTD), using equation 1:

$$TTD = Tsat - Tfeedwater\ out \quad (1)$$

Where:
Tsat = Extraction steam saturation temperature at initial pressure (°C)
Tfeedwater out = HPH outlet feedwater temperature (°C)

2) Calculation of Drain Cooler Approach Temperature, can be calculated using Equation 2.

$$DCA = Tdrain - Tfeedwater\ in \quad (2)$$

Where:
T_{drain} = Extraction steam drain outlet temperature (°C)
$T_{feedwater\ in}$ = Feedwater inlet temperature (°C)

3) Calculation of Temperature Rise, can be calculated using equation 3.

$$TR = Tfeedwater\ out - Tfeedwater\ in \quad (3)$$

Where:
$T_{feedwater\ out}$ = Feedwater outlet temperature (°C)
$T_{feedwater\ in}$ = Feedwater inlet temperature (°C)

4) LMTD calculation, can be calculated using equation 4.

$$LMTD = \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (4)$$

$$= \frac{(T_{steam\ in} - T_{feedwater\ out}) - (T_{steam\ out} - T_{feedwater\ in})}{ln\left(\frac{(T_{steam\ in} - T_{feedwater\ out})}{(T_{steam\ out} - T_{feedwater\ in})}\right)}$$

Where:
$T_{steam,\ in}$ = Incoming extraction steam temperature (^{0}C)
$T_{feedwater,\ out}$ = Outgoing feed water temperature (^{0}C)
$T_{steam,\ out}$ = Outgoing extraction steam drain temperature (^{0}C)
$T_{feedwater,\ in}$ = Inlet feed water temperature (^{0}C)

5) Calculation of Q Requirements, can be calculated using the equation:

Because the high - pressure heater consists of three zones, namely subcooling, condensing, and desuper heating. Then we will also calculate the heat required for each zone so that the equation becomes:

- Heat Transfer Rate in the Desuperheating Zone

$$Q_{desuperheating} = \dot{m}hi\ (hhi - hg) \quad (5)$$

- Heat Transfer Rate in the Condensing Zone

$$Q_{condensing} = \dot{m}hi\ (hg - hf) \quad (6)$$

- Heat Transfer Rate in the Subcooling Zone

$$Q_{subcooling} = \dot{m}hi\ (hf - hho) \quad (7)$$

Where:
$Q_{desuperheating}$ = Desuperheating zone heat transfer rate (kJ/s)
$Q_{condensing}$ = Condensing zone heat transfer rate (kJ/s)
$Q_{subcooling}$ = Subcooling zone heat transfer rate (kJ/s)
mhi = Mass flow rate of extraction steam (kg/s)
hhi = Enthalpy of incoming extraction steam (kJ/kg)
hg = Vapor enthalpy of saturation vapor (kJ/kg)
hf = Liquid saturation vapor enthalpy (kJ/kg)
hho = Enthalpy of drain water (kJ/kg)

6) Calculation of Heat Capacity Ratio, can be calculated using the equation:

It is a comparison between the smallest and largest heat capacities for the two fluid flows where the C* value <1.

$$C^* = \frac{C_{min}}{C_{max}} \quad (8)$$

Where:
C_{steam} = ṁsteam. Cpsteam
$C_{feedwater}$ = ṁfeedwater. Cpfeedwater

7) Calculation of effectiveness (ε), can be calculated using the formula:

$$\epsilon = \frac{Q}{Q_{max}} \quad (9)$$

$$Q = (m\ cp)_{steam}\ (T_{steam\ in} - T_{steam\ out})$$

$$Q = (m\ cp)_{feed\ water}\ (T_{feedwater\ out} - T_{feedwater\ in})$$

$C_{steam} > C_{feedwater},\ (T_{steam\ in} - T_{steam\ out}) < (T_{feedwater\ out} - T_{feedwater\ in})$

$C_{steam} > C_{feedwater},\ (T_{steam\ in} - T_{steam\ out}) > (T_{feedwater\ out} - T_{feedwater\ in})$

Because $C_{steam} < C_{feedwater}$, then

$$Q_{max} = (m\ cp)_{steam}\ (T_{steam\ in} - T_{feedwater\ in})$$

$$\epsilon \frac{Q}{Q_{max}} \quad (10)$$

$$NTU = \frac{1}{(1+C^{*2})^{1/2}} \ln\left[\frac{2-\varepsilon\left[1+C^* - (1+C^{*2})^{1/2}\right]}{2-\varepsilon\left[1+C^* + (1+C^{*2})^{1/2}\right]}\right]$$

For $C_h = C_{min}$

$$\epsilon \frac{C_{steam}\ (T_{steam\ in} - T_{steam\ out})}{C_{min}\ (T_{steam\ in} - T_{feedwater\ in})} \quad (11)$$

Number Transfer Unit (NTU)

$$\text{Or, } NTU = \frac{UA}{C_{min}} \quad (12)$$

4. Results/Discussion

The results obtained from the comparative data (commissioning) of PLTU Teluk Sirih unit 2 based on the performance test document/initial testing before the unit's first commercial operation:

1) The data obtained consists of several tests, namely loads of 50%, 75%, and 100% of the installed capacity.
2) Next, the data is interpolated with a polynomial graph to adjust to the load being tested at this time as shown in Table 4.1.

Table 4.1: Comparison Data Agustus 2020

Extraction Steam to Deaerator				
Pressure	Pex3	Measured	bara	6.00
Temperature	Tex3	Measured	oC	382.40
Enthalpy	Hex3	Steam Table	kJ/kg	3, 233.85
Drain Water to HP 1 Heater				
Pressure	Pd1	Calculated	Bara	26.07
Temperature	Td1	Measured	oC	209.17
Enthalpy	Hd1	Steam Table	kJ/kg	894.10
Drain Water to HP 2 Heater				
Pressure	Pd2	Calculated	Bara	15.06
Temperature	**Td2**	**Measured**	oC	168.24
Enthalpy	Hd2	Steam Table	kJ/kg	711.80
Deaerator Shell				
Pressure	Pds	Measured	bara	6.50
Temperature	Tds	Measured	oC	159.80
Enthalpy	Nd	Steam Table	kJ/kg	674.63
Make up Water				
Pressure	Pmu	Measured	bara	
Temperature	Tmu	Measured	oC	36.2
Enthalpy	Hmu	Steam Table	kJ/kg	151.737
Exhaust Turbine				
Pressure	Pexh	Measured	bara	0.103
Temperature	Texh	Measured	oC	41.54
Enthalpy	Hexh	Steam Table	kJ/kg	173.97
Mean Steam Flow				
Final Feed Water Flow	Mf	[(1+K4) ”Mcw - Mis]/ (1 - A+K5*A)	kg/h	301, 019.39
Make up Water Flow to Condenser	Mm	Measured	kg/h	0.00
Superheat Spray Flow	Mis	Measured	kg/h	45, 712.39
Main Steam Flow (at main stop valve inlet)	M1	Mf+ Mis - (Mm+ Mds+Mht)	kg/h	344, 923.72
HP# 1 Extraction Steam Flow	Mext1	K1’Mf	kg/h	23, 817.80
HP# 2 Extraction Steam Flow	Mext2	(K2 - K1*K3) ”MF	kg/h	25, 413.00

3) Carry out performance tests according to standard operating procedures (SOP) as shown in the results in Table 4.2.

Table 4.2: Test Data September 2024

Extraction Steam to HP 1 Heater				
Pressure	Pex1	Measured	bara	29.13
Temperature	Tex1	Measured	oC	416.29
Enthalpy	Hex1	Steam Table	KJ/kg	3, 270.08
Extraction Steam to HP 2 Heater				
Pressure	Pex2	Measured	bara	18.41
Temperature	Tex2	Measured	oC	338.37
Enthalpy	Hex2	Steam Table	KJ/kg	3, 115.17
Extraction Steam to Deaerator				
Pressure	Pex3	Measured	bara	9.05
Temperature	Tex3	Measured	oC	268.37
Enthalpy	Hex3	Steam Table	KJ/kg	2, 986.46
Drain Water to HP 1 Heater				
Pressure	Pd1	Calculated	bara	28.25
Temperature	Td1	Measured	oC	222.59
Enthalpy	Hd1	Steam Table	KJ/kg	955.75
Drain Water to HP 2 Heater				
Pressure	Pd2	Calculated	bara	47.86
Temperature	**Td2**	**Measured**	oC	188.18
Enthalpy	Hd2	Steam Table	KJ/kg	799.71
Deaerator Shell				
Pressure	Pds	Measured	bara	-
Temperature	Tds	Measured	oC	155.40
Enthalpy	Nd	Steam Table	Hi/kg	655.62
Make up Water				
Pressure	Pmu	Measured	bara	
Temperature	Tmu	Measured	oC	36.2
Enthalpy	Hmu	Steam Table	KJ/kg	151.737
Exhaust Turbine				
Pressure	Pexh	Measured	bara	0.127882

Temperature	Texh	Measured	oC	51.6896
Enthalpy	Hexh	Steam Table	KJ/kg	5, 594.45
Mean Steam Flow				
Final Feed Water Flow	Mf	[(1+K4) ”Mcw - Mis]/ (1 - A+K5*A)	kg/h	355, 545.32
Make up Water Flow to Condenser	Mm	Measured	kg/h	2, 752.00
Superheat Spray Flow	Mis	Measured	kg/h	28, 025.20
Main Steam Flow (at main stop valve inlet)	M1	Mf+ Mis - (Mm+ Mds+Mht)	kg/h	415, 753.30
HP# 1 Extraction Steam Flow	Mext1	K1’Mf	kg/h	27, 045.80
HP# 2 Extraction Steam Flow	Mext2	(K2 - K1*K3) ”MF	kg/h	26, 649.14

After calculating the comparison and testing parameter data, the calculation results were obtained as shown in Table 4.3 as follows:

Table 4.3: Calculation Results

S. No	Parameter	August 2020		September 2024	
		Data Pembanding		Data Pangujian	
		HPH - 1	HPH - 2	HPH - 1	HPH - 2
1	Terminal Temperature Difference (TTD)	0.063^{0}C	0.94^{0}C	2.516^{0}C	14.037^{0}C
2	Drain Cooler Approach Temperature (DCA)	10.2^{0}C	6.16^{0}C	28.4^{0}C	38.93^{0}C
3	Temperature Rise (TR)	32.28^{0}C	36.89^{0}C	35.52^{0}C	44.94^{0}C
4	Log Mean Temperature Difference (LMTD)	50.08^{0}C	40.72^{0}C	31.24^{0}C	80.4^{0}C
5	Capacity Heat Ratio	0.04	0.0431	0.0388	0.0522
6	Heat Exchange Thermal Effectiveness	0.951	0.963	0.872	0.794

4.1 Terminal Temperature Difference (TTD)

Terminal Temperature Difference (TTD) is defined as the extraction steam saturation temperature minus the feed water outlet temperature. An increase in TTD indicates a reduction in heat transfer in heat exchanger (HE) equipment, while a decrease indicates improved performance in HE equipment. Typical ranges for TTD on high - pressure preheaters with and without desuperheat zones are - 3°C to - 5°C and below 1°C.

Analysis of Terminal Temperature Difference (TTD) Calculation Results

Based on the calculation results obtained, the difference in terminal temperature difference (TTD) on HPH - 1 and HPH - 2 is shown in figure 4.1 and figure 4.2:

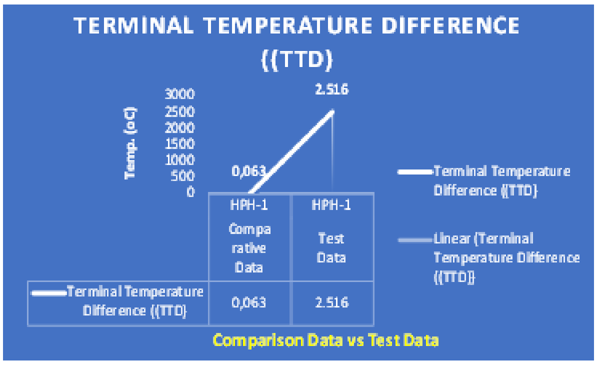

Figure 4.1: Comparison diagram of HPH - 1 TTD Value

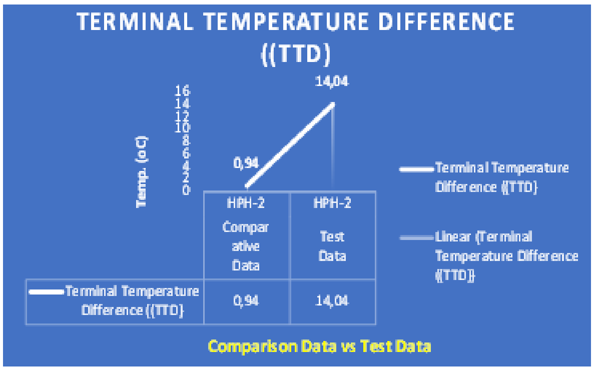

Figure 4.2: Comparison diagram of HPH - 2 TTD Value

Figure 4.1 shows that there was an increase in the HPH - 1 TTD value from 0.063 °C to 2, 516 °C with a difference of 2, 453 °C. Likewise, in Figure 4.2, there is an increase in the HPH - 2 TTD value from the comparative data of 0.94 °C to the test of 14, 037 °C with a difference of 13, 097 °C. This increase in TTD value causes the initial heater performance to decrease.

An increase in the TTD value can be caused by several factors, such as fouling in the pipe, high extraction steam drain levels, decreased or increased extraction steam pressure, and temperature.

4.2 Drain Cooler Approach Temperature (DCA)

Calculation of the drain cooler approach (DCA) value is one of the methods used to determine the initial value.

Analysis of Drain Cooler Approach (DCA) Calculation Results

According to the calculation results obtained in Table 4.3, the comparison diagram of DCA values for HPH - 1 and HPH - 2 is shown in Figure 4.3 and Figure 4.4:

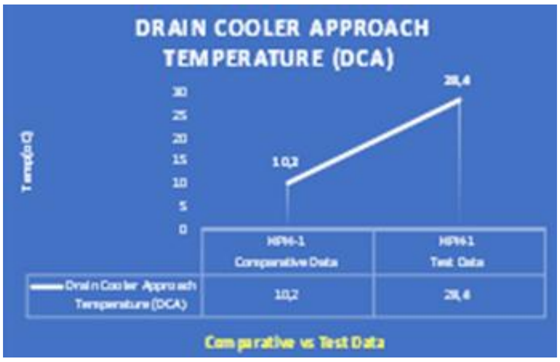

Figure 4.3: Comparison diagram of HPH - 1 DCA Value

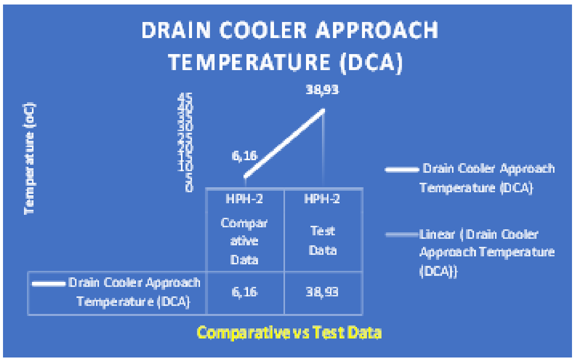

Figure 4.4: Comparison diagram of HPH - 2 DCA Value

Based on the results of the Drain Cooler Approach (DCA) calculation shown in Figure 4.3, comparative data of 10.2 °C and test data of 28.4 °C, there was an increase in DCA on the initial heater (HPH - 1) of 18.2 °C. In Figure 4.4 it is shown that the comparative data is 6.16°C and the test data is 38.93°C; there is an increase in DCA for preheater 2 (HPH 2) heater performance of 32.77°C. These data show that the performance of the preheaters (HPH - 1 and HPH - 2) decreases. The initial decrease in HPH can be caused by several factors, such as fouling or leaks in the pipe, high extraction steam drain levels, extraction steam pressure, and temperature not suitable for commissioning.

4.3 Temperature Rise (TR)

Temperature rise is the difference between the inlet feed water temperature and the outlet feed water temperature of the preheater (HPH). Good HPH performance must meet the factory design specifications with the drainage water level setting in the preheater properly maintained in accordance with the procedures. If there is an increase in the rise temperature, then the heat absorption performance of the HPH equipment is good.

Analysis of Temperature Rise (TR) Calculation Results

According to the calculation results in Table 4.3, the TR Diagram of Comparative Data and test results are shown in Figures 4.5 and 4.6:

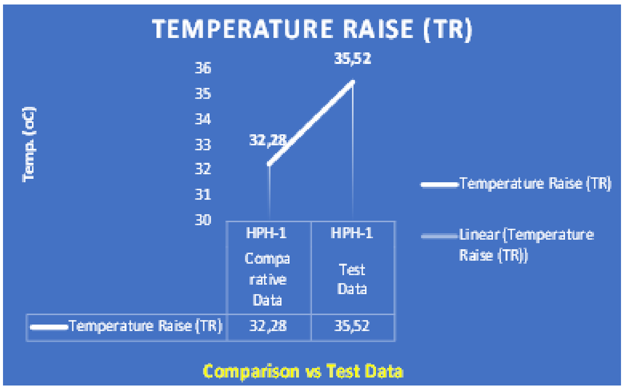

Figure 4.5: Comparison diagram of HPH - 1 TR Value

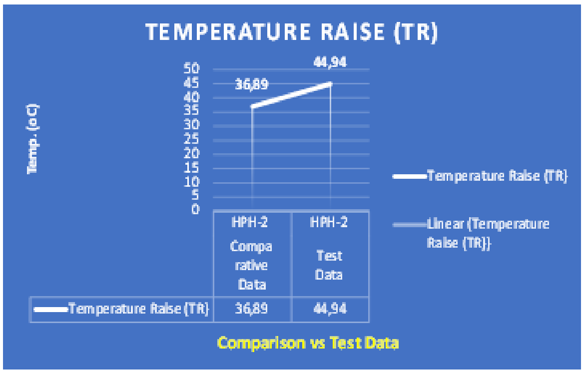

Figure 4.6: Comparison diagram of HPH - 2 TR Value

Comparison of the Temperature Rise (TR) value shown in Figure 4.5, namely comparative data of 32.28 °C and test data of 35.52 °C, there was an increase in the TR value on HPH - 1 by 3.24 °C. Figure 4.6 shows comparative data of 36.89 °C and test data of 44.94 °C. There was an increase in the TR value on HPH - 2 of 9.42 °C. Based on the results of these calculations, it can be stated that the performance of HPH - 1 and HPH - 2 due to heat absorption in the heater is very good.

4.4 LMTD (Log Mean Temperature Difference)

According to the calculation results in Table 4.3, a comparison graph of the Log Mean Temperature Difference (LMTD) values is shown in Figure 4.7 and Figure 4.8.

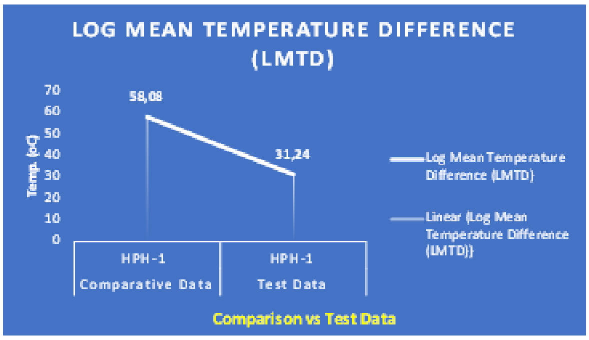

Figure 4.7: Comparison diagram of HPH - 1 LMTD Value

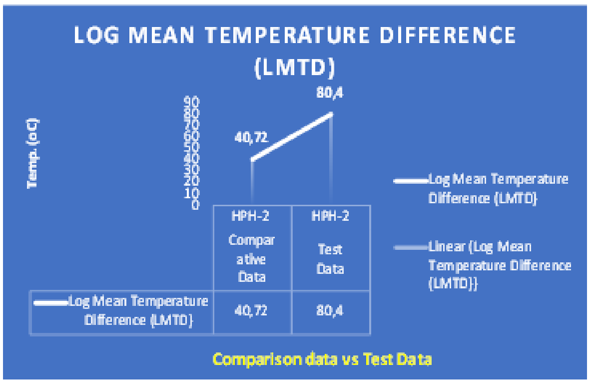

Figure 4.8: Comparison diagram of HPH - 2 LMTD Value

In Figure 4.7 it is shown that the LMTD value for the comparative data is 58.08 °C, and the test data is 31.24 °C. There is a decrease in the LMTD value on HPH - 1 of 26.84 °C, while in Figure 4.8 it is shown that the comparative data is 40.72 °C and the test data is 80.4 °C. C, there was an increase in the TR value for HPH - 2 by 39.68 °C.

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Based on these results, it can be stated that the performance of the HPH - 1 preheater shows poor heat absorption, while the performance of HPH - 2 shows very good heat absorption.

4.5 Heat Exchanger Thermal Effectiveness

Heat Exchanger Thermal effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate for given flow and temperature conditions. The maximum possible heat transfer speed can be achieved if the fluid with the minimum heat capacity ratio (HCR) value experiences a maximum temperature increase in the preheater (HPH). The closer the effectiveness value is to 1, the better the performance of the preheater (HPH), and conversely, if the effectiveness value decreases, the performance of the preheater (HPH) decreases.

Analysis of Heat Exchanger Effectiveness Calculation Results

Based on the calculation results in Table 4.3, the graph of the Heat Exchanger Effectiveness calculation results is shown in Figure 4.9 and Figure 4.10:

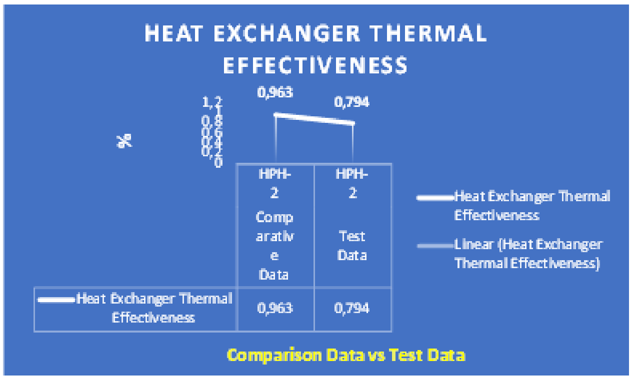

Figure 4.9: Comparison diagram of HPH - 1 HE Thermal Effectiveness Value

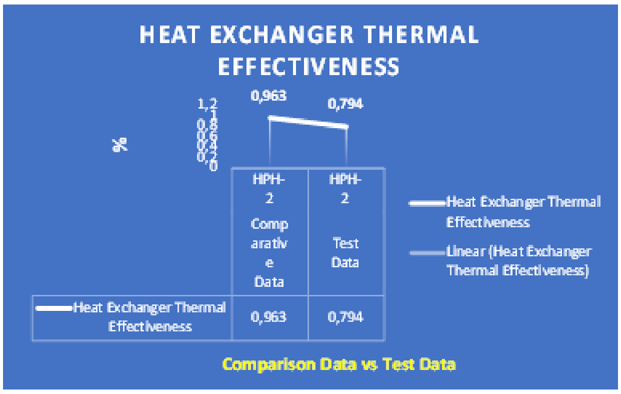

Figure 4.10: Comparison diagram of HPH - 2 HE Thermal Effectiveness Value

The results of the heat exchanger effectiveness calculation for the comparative data were 0.951%, and the test data was 0.872%. As shown in Figure 4.9, there was a decrease in HPH - 1 of 0.079%, while in Figure 4.10 the comparative data was 0.963% and the test data was 0.794%. There was a decrease in heat exchanger effectiveness on HPH - 2 ofs 0.169%. Comparison of the calculation results shows that the performance on HPH - 1 and HPH - 2 has decreased. A decrease in the effectiveness value is usually caused by impurities in the heat exchanger pipe, causing the equipment to not absorb maximum heat.

5. Conclusion

Based on the results of thermodynamic analysis and heat transfer at HPH - 1 and HPH - 2 PLTU Teluk Sirih Unit 2, several conclusions can be drawn as follows:

1) From the results of calculations and analysis, it was found that there was a decrease in performance in the initial heaters (HPH - 1 and HPH - 2). This is shown in an increase in terminal temperature difference (TTD) and drain cooler approach temperature (DCA) and a decrease in the thermal effectiveness value of pre - heaters 1 and 2 (HPH - 1 and HPH - 2). However, in contrast to the temperature rise and log mean temperature difference (LMTD) values, which have increased, this shows that the heat absorption in the preheater (HPH - 1 and HPH - 2) is very good.
2) Factors that influence the reduction in heat transfer are as follows:
   a) An increase in the temperature rise and log mean temperature difference (LMTD) values can be caused by several factors, such as inaccurate readings of the feed water temperature measuring instrument entering the preheater (HPH - 1 and HPH - 2).
   b) Increases in terminal temperature difference (TTD) and drain cooler approach temperature (DCA) can be caused by high drainage water levels, leaks in the heat exchanger pipe, and plugging in the preheater (HPH - 1 and HPH - 2) so that heat transfer is not optimal.
   c) The decrease in effectiveness of the preheater (HPH - 1 and HPH - 2) can be caused by impurities or fouling in the heat exchanger pipe so that heat absorption is not optimal.

3) Actions to restore heat transfer performance in the preheater (HPH - 1 and HPH - 2) can be carried out during routine maintenance activities, such as cleaning the tube, checking tube condition, checking and calibrating tank level, checking or calibrating measuring instruments, checking the flange or connection of each pipe junction.

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