Design of Heat-Recovery Steam Generator Components in Gas Turbine (70 MW) Combined Cycle Power Plants (105 MW)

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Abstract—**The gas turbine combined cycle (GTCC) power plant system combination a gas power plant and a steam power plant using Brayton and Rankine cycles. In GTCC specification, the heat-recovery steam generator (HRSG) is employed as a heat exchanger to produce superheated steam. The utilization of waste heat is a need for sustainable energy use. This paper reviews research on the recovery of waste heat by designing and simulating an HRSG construction. Therefore, this study aims to create an HRSG with high (56 bar) and low (6 bar) pressure levels at a temperature of 500 ℃ and 137.3 kg/s of gas turbines using simulation. The use of equations and design simulations can be applied to HRSG components with appropriate high and low pressure.**

Index Terms—**Economizer, evaporator, gas turbine combined cycle, heat-recovery steam generator, superheater**

I. Introduction

A Gas turbine combined cycle (GTCC) is a power-plant system that generates more efficient, significant, and improved electricity than that produced by a conventional cycle, such as a gas power plant and steam power plant. One of the benefits of GTCC is its ability to consume a limited amount of cooling water and shorter operational time than those of the two conventional plants. Industries can improve, and high efficiency in GTCC (up to 58%) can be achieved by improving the heat-recovery steam generator (HRSG) design [1]. We focus on introducing multiple pressure levels by reheating steam in the steam cycle on HRSG to recover more exhaust gas energy. Furthermore, the HRSG has broad application prospects in increasing electricity capacity and reducing its generation costs. This design aims to further improve system performance above 60%. Power plant modifications, configurations, and operating strategies can improve the unit's cycling capability [2]. Also, the combined cycle power plants are increasingly subjected to cyclic operations [3]. GTCC employed HRSG, a Heat Exchanger (HE), on the exhaust gas turbine. Researched include economizers, evaporators, and superheaters. Other research also focused on enhancement technologies by offering more design options for increasing thermal efficiency [4]. Obtaining optimal performance in HE designs requires general limitations. The boundary requirement is that the dirt factor (R_D) of the two fluids flowing in the HE should be greater or equal to the provisions of the second impurity factor, with pressure drops obtained from piping and heat exchangers [5]. The calculation of the impurity factor depends on the model value of the heat transfer coefficient, which is in progress to predict the behavior of post-boiling transitions that may occur during the anticipated operational events for boiling [6]. Furthermore, it considers the effect of the most crucial parameters, such as surface waves, evaporation, and flow configurations [7].

The value of each fluid's heat transfer is used to calculate the overall coefficient (U_C), which determines U_D to obtain R_D and provides geometrical design data from HRSG [8]. This paper summarizes the detailed HRSG design scheme and produces its height, length, and width dimensions. The feasibility analysis of the HRSG design is the thickness of the pipes used, which is different from that in previous studies on the Heat Exchanger. The research on the system of HRSG for various Na liquids produced differences in the heat transfer coefficient (h_{Na}) [9]. Research on the simple modification of the internal configuration of HRSG also makes the gas combustion flow more uniform [10].

Furthermore, recovering waste heat from a 60 MW gas turbine exhaust generates additional power of 35.14 MW [11]. Waste hot air from HRSG also indicated that the air return strategy offers significant performance improvements [12].

In this work, we propose the design and simulation of HRSG with high and low pressure by heating the exhaust heat at a temperature of 500°C. In designing this HRSG, consider the mass-energy balance equation, HRSG, and boiler identical in modeling [13]. Apart from that, they also include low and high pressures which are required to maximize heat recovery from exhaust gases. The advantage and novelty of this design are that it successfully combines steam and gas power with a capacity of 105 MW and an efficiency of 69.48%.

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II. Simulation and Design

A. Simulation

This simulation's focus is to design an HRSG with high (56 bar) and low (6.2 bar) pressure levels at a capacity of 105 MW. This parameter is the main component of the HRSG, which utilizes the gas turbine's flue to convert water into steam at a temperature of 500 ℃. Capacity and temperature are the initial values for simulating the initial design parameters. The simulation results are as shown in Tables I, II, and III.

TABLE I. Data from the Simulation Superheater Results to Start Designing

No	Parameter	Unit	*Low Pressure 6 bars*	*High Pressure 58 bars*
1	Inlet water temperature	^{0} C	151	265.5
2	Vapor temperature out	^{0} C	160	271
3	Steam mass flow rate	kg/hr	27899	38438.3
4	Exchaust gas temperature entered	^{0} C	300	464.4
5	Specific heat exhaust gases	kJ/kg ^{0} C	0.908	1.139
6	Exchaust a mass flow rate	kg/hr	494319	494319
7	Enthalpy of water enters	kJ/kg	636.56	1162.49
8	Enthalpy of steam out	kJ/kg	2756.8	2788.87

TABLE II. Data from the Simulation evaporator Results to Start Designing

No	Parameter	Unit	*Low Pressure 6 bars*	*High Pressure 58 bars*
1	Inlet vapor temperature	^{0} C	160	270.5
2	Vapor temperature out	^{0} C	213	450
3	Specific heat of steam	kJ/kg ^{0} C	2.281	2.905
4	Steam mass flow rate	kg/hr	27758.9	38438.3
5	Exhaust gas temperature entered	^{0} C	350	500.5
6	Exhaust gas temperature	^{0} C	343	464.4
7	Specific heat exhaust gases	kJ/kg ^{0} C	1.101	1.135
8	Exhaust mass flow rate	kg/hr	494319	494319

TABLE III. Data from the Simulation Superheater Results to Start Designing

No	Parameter	Unit	*Low Pressure*	*High Pressure*
			6 bars	*58 bars*
1	Inlet water temperature	^{0} C	140.25	269.06
2	Water temperature out	^{0} C	303.8	509.95
3	Specific heat of the water	kJ/kg ^{0} C	4.23	4.25
4	Water mass flow rate	kg/hr	27898.5	38438.3
5	Exhaust gas temperature entered	^{0} C	372.12	650.8
6	Exhaust gas temperature	^{0} C	334.92	572.53
7	Specific heat exhaust gases	kJ/kg ^{0} C	1.06	1.094
8	Exhaust mass flow rate	kg/hr	494319	494319

B. Design

The design uses the exhaust gas's base temperature that comes out of the turbine, which is between 450 ℃ and 630℃ [14]. Furthermore, the method uses a flue gas temperature at 500ºC and a mass flow rate of 137.3 kg/s with the initial dimensions, as shown in Table IV. The HRSG design uses a flue gas temperature at 500ºC and a mass flow rate of 137.3 kg/s and sets the initial design dimensions as in Table IV.

The design of the HRSG must go through the stages of designing HRSG, which is as follows:

C. Calculation of Heat Transfer Rate

The equation for calculating the heat transfer rate, energy balance, and LMTD [15] are:

Heat transfer rate:

$$q = U_D \times A \times \Delta t_{LMTD} \qquad (1)$$

q is the rate of heat transfer (W), A is heat transfer area (m), U_D is the design of overall heat transfer coefficient (W/(m^2) (℃)), and Δt_{LMTD} is logarithmic mean temperature difference (℃).

Energy Balance:

$$Q = M \times C_p \times (T_1 - T_2) = m \times c_p \times (t_2 - t_1) \qquad (2)$$

Q is energy balance (W), M is exhaust mass flow rate (kg/s), m is water mass flow rate (kg/s), C is the specific heat of exhaust gas (J/kg ℃), and c is the specific heat of feed water (J/kg ℃). T_1 is the temperature of hot fluid entering (℃), T_2 is the temperature of hot fluid coming out (℃), t_1 is the temperature of cold fluid entering (℃), and t_2 is the temperature cold fluid coming out (℃).

TABLE IV. DATA DIMENSION SET TO START DESIGNING

No.	Parameter	Unit	Superheater		Evaporator		Economizer	
			Low Pressure (6,2 bar)	High Pressure (56 bar)	Low Pressure (6,2 bar)	High Pressure (56 bar)	Low Pressure (6,2 bar)	High Pressure (56 bar)
1	BWG.		13	13	15	13	10	13
2	Outer Diameter (OD)	cm	1.905	1.905	2.54	2.54	2.54	2.54
3	Inner diameter (ID)	cm	1.422	1.424	2.174	2.057	1.859	2.057
4	Thick pipe	cm	0.241	0.241	0.183	0.2413	0.34	0.241
5	Distance between fin ends	cm	1.27	0.2413	1.27	1.27	1.27	1.27
6	Flow area per tube	cm	0.627	0.627	1.463	2.245	1.069	1.308
7	Surface per linear m (Outside)	m	0.059	0.059	0.079	0.099	0.079	0.079
8	Surface per linear m (inside)	m	0.045	0.045	0.067	0.085	0.058	0.065
9	Weight per linear m	kg steel	0.329	0.329	0.354	0.581	0.617	0.454
10	Arrangement of pipe		Triangular	Triangular	Triangular	Triangular	Triangular	Triangular
11	Fin height	cm	0.635	0.635	2.007	2.007	2.007	2.007
12	Fin thickness	cm	0.419	0.419	0.1016	0.102	0.102	0.102
13	Number of fins per inch	cm	10.16	10.16	20.32	20.32	20.32	20.32
14	OD '(OD + 2 x height of fins)	cm	3.81	3.175	6.5532	6.553	6.5532	6.553
15	Long pipe	cm	74.98		74.98		74.98	
16	High Duct	m	8.997		8.997		8.997	
17	Duct Width	m	3.0785		3.0785		3.0785	

The LMTD can be calculated by equation, following:

$$\Delta t_{LMTD} = \frac{\left(\Delta t_2 - \Delta t_1\right)}{\ln\left(\frac{\Delta t_2}{\Delta t_1}\right)} \tag{3}$$

LMTD is Log Mean Temperature difference (°C), Δt_1, Δt_2 is temperature difference at the cold and hot terminals, respectively (°C)

D. Calculating Equivalent Diameter

Calculation of equivalent diameter is to use Eq. (4), where the parameters needed to determine the equivalent diameter are the fin surface (A_f) [16], the bare tube area (A_o), and the projected perimeter.

$$d_e = \frac{\left(2\left(A_f + A_o\right)\right)}{\left(projected\ perimeter\right)} \tag{4}$$

With $A_f = \frac{\pi}{4}\left(OD'^2 - OD^2\right) 2\times N_f \times 12$ (5)

$$A_o = \left(1 - N_f . y\right)\pi \times OD \times 12 \tag{6}$$

A_f is the surface of thin fins (both sides) (cm^2), A_o is bare surface on the outside of finned tube (cm^2), OD' outer diameter (outer diameter of the pipe plus fin height) (cm), OD is the outside diameter of the tube (cm), and N_f is number of fins (per tube).

Then to calculate the projection perimeter (P) using the following equation:

$$P = 21\times 2?N_f \times\ 2+2\left(1-N_f \times Y\right) \tag{7}$$

where P is projected perimeter (cm/m), Y is fin thickness (cm), and L is fin height (cm)

E. Determine the Number of Pipes per Bundle

The pipes contained in the HRSG consist of pipeline arrangements in the form of rows (Bundles). The distance between the vertical center points of the channel width determines the number of pipes in a Bundle. Calculation of the number of tubes in HRSG in one bundle uses the equation:

$$N_t = \frac{Y}{ST} \tag{8}$$

N is the number of pipes per bundle (dimensionless), Y is duct width (m), and ST is the vertical distance between central points of lines (cm).

With $ST = OD + \left(2\times 1\right) + ?Dis\tan ce?between\,the\,tips\,of\ the\ fins$ (9)

F. Calculating the Heat Transfer Coefficient in Pipes (h_i)

Calculate Flow area

$$\alpha_t = \frac{N_t \times a'_t}{144} \quad (10)$$

Where α_t is flow area (m^2), N_t is the number of pipes (dimensionless), and α't = flow area (cm^2).

Calculate Water Flow Speed (V)

$$V = \frac{G_t}{3600\rho_w} \quad (11)$$

Where V is flow velocity of water (m/s), Gt is mass water velocity (kg/(m^2) (s), and ρw is water density (kg/$m^{3)}$

With Mass Velocity

$$G_t = \frac{w}{\alpha_t} \quad (12)$$

Determining Reynold Numbers in Pipes (Ret)

Determination of the Reynold number in the pipe (Ret), using the following equation [17]

$$R_{et} = \frac{D \times G_t}{\mu} \quad (13)$$

D is tube diameter, μ and is the water's viscosity (kg/ (m. s).

G. Determine the Heat Transfer Coefficient in Pipes (h_i)

In determining the heat transfer coefficient in the pipe (hi), you can use the following equation:

$$h = j_H \times \frac{k}{D} \times \left(\frac{c_p \times \mu}{k} \right)^{\frac{1}{3}} \quad (14)$$

h is heat transfer coefficient in the pipe (W/(m^2 ℃)), k_w is heat conductivity of feed water at average temperatures (W/ (m ℃)), μ is the viscosity of water at average temperature (kg/ (m. s), j_H is the Sieder-Tate and Colburn heat-transfer factor in the pipe (dimensionless), and c is specific heat (J/ (kg ℃).

H. Calculating the Exhaust Gas Pass Area

$$\alpha_s = XY - N_t \times OD.48 - N_t \left(2y.L.N_f .48 \right) \quad (15)$$

where α_s is exhaust area passage area, (m^2), X is duct height, (m), and Y is duct width, (m).

I. Calculating Mass Gas Flue Mass Speed (Gs)

$$G_s = \frac{W}{a_s} \quad (16)$$

where Gs is exhaust gas mass velocity, (kg/ (m^2.s).

J. Determine Exhaust Gas Reynolds Numbers (Res)

$$R_{es} = \frac{D_e \times G_s}{\mu_s} \quad (17)$$

Where Res is exhaust gas Reynold Number (dimensionless), De is= equivalent diameter (de/12) (m), μs is the viscosity of flue gas at average temperature (kg/m. s).

K. Determine the External Pipe Heat Transfer Coefficient (hf)

$$h_f = j_H \times \frac{k_g}{D} \times \left(\frac{C_p \times \mu}{k_g} \right)^{\frac{1}{3}} \quad (18)$$

Calculating the Heat Transfer Coefficient on the Pipe Surface (h'_{fi})

$$h'_{fi} = \left(\Omega \times A_f + A_o \right) \frac{h_f}{A_i} \quad (19)$$

where Ω is Fin effectiveness (dimensionless)

Calculating the Overall Heat Transfer Coefficient (U_D)

$$U_D = \frac{h'_{fi} \times h'_i}{h'_{fi} + h'_i} \quad (20)$$

The value of U_D is the overall heat transfer coefficient (W/(m^2) (ºC)).

Calculating Heat Transfer Area (A) [18]

$$A = \frac{Q}{U_{Di} \times \Delta t_{LMTD}} \quad (21)$$

The value of A is heat transfer area, (m^2)

Count the Number of Pipe Bundles (n)

$$n = \frac{A}{A_i \ perbundle} \quad (22)$$

where n is the number of pipe bundles (dimensionless), Ai per bundle is Ai. Nt. L (m^2)

Calculates Actual Overall Heat Transfer Coefficient (U_D. act.)

$$U_{D\,act} = \frac{\dot{Q}}{A_i \times \Delta t} \quad (23)$$

With:

$$A_i = \left(n \times N_t \right) \times a'' \times L \quad (24)$$

where n is the number of pipe bundles; a" is surface per linear m, m; L is pipe length (m)

Calculating Dirt Factor (Rd')

$$R_{d'} = R_d + Excess\ fouling\ factor \times Adding\ to\ the\ outside\ fouling\ factor \quad (25)$$

where Rd' is the Combined dirt factor calculated (m^2 ℃)/W), Rd is Combined dirt factor provisions (m^2 ℃)/W).

With excess fouling factor can be calculated using the equation:

$$Excess\ fouling\ factor = \frac{1}{U_{D\,act}} - \frac{1}{U_{Di}} \quad (26)$$

Then adding to the outside fouling factor can be calculated using the equation:

$$\text{Adding to the outside } R_D = \frac{\left(A_f + A_o\right)}{A'_i} \quad (27)$$

L. Calculating Pressure Drop

Determine the Volumetric Equivalent Diameter (D'ev)

$$D'_{ev} = \frac{(4 \times net\ \ free\ volume)}{Frictional\ surface} \quad (28)$$

With net free volume (NFV):

$$NFV = XY\frac{v_s}{12} - \frac{1}{2}\left[N_t + \left(N_t - 1\right)\right]\frac{\pi}{4}\frac{OD^2}{144} \times Y - \frac{1}{2}\left[N_t + \left(N_t - 1\right)\right]\frac{\left(OD'^2 - OD^2\right)}{144} \quad (29)$$

Where X is duct height (m), Y is duct width (m), Vs. is Volumetric section (cm^3), and y is fin thickness (in). Then the friction surface can be calculated using the equation:

$$Friction\,surface = \frac{1}{2}\left[N_t - \left(N_t - 1\right)\right] \times bare\,tube\,area \times Y \quad (30)$$

Where bare tube area (ft^2/ft).

M. Calculating the Pressure Drop on the Duct Side (ΔPs)

$$\Delta P_s = \frac{f \times G_s^{\,2} \times L_p}{5.22 \times 10^{10} \times D'_{ev} \times s \times \phi_s}\left(\frac{D'_{ev}}{S_T}\right)^{0.4}\left(\frac{S_L}{S_T}\right)^{0.6} \quad (31)$$

ΔPs is pressure drop on the duct side (bar), f is flue gas friction factor (m^2/cm^2), and s is specific gravity. ST is the vertical distance between the center points of the pipe (cm), SL is the transverse distance between pipe center points (cm), Øs is the exhaust gas viscosity ratio, and Lp is exhaust gas path length (m). The size of the path (Lp) is calculated using the following equation:

$$L_p = \frac{n \times v_s}{12} \quad (32)$$

N. Calculating Pressure Reduction in Pipes (ΔPt)

$$\Delta P_t = \frac{f_t \times G_t^{\,2} \times L \times n}{5.22 \times 10^{10} \times D \times s \times \phi_t} \quad (33)$$

where ΔPt is pressure drop on the pipe side (bar), f is water friction factor (m^2/cm^2), s is specific gravity, Øt is water viscosity ratio, and L is pipe length (m).

High temperatures convert feed water into superheated steam. Superheated steam will drive a steam turbine and produce mechanical energy. The design requirements used are a decrease in water pressure ≤ 0.689476 bar, a decrease in exhaust gas pressure, and vapor ≤ 0.137895 bar, and the calculated dirt factor must be higher than or equal to the stipulated dirt factor.

O. Calculate the Minimum Pipe Thickness **(t_{min})**

The calculation of the minimum pipe thickness can keep the pipe within safe limits if it is operated at its working pressure. The thickness of the tube is one of the requirements for designing HRSG components from the mechanical side. Calculation of minimum thickness of the pipe is performed using the following equation [19-20]:

$$t_{min} = \frac{P \times OD}{2S + P} + 0.005 \times OD \quad (34)$$

where t_{min} is a minimum thickness of the pipe, m, P is fluid working pressure, bar, S is allowable stress, bar, OD is the outside diameter of the tube, m.

P. Design Output

The HRSG design uses a high (56 bar) and low (6.2 bar) pressure base at a capacity of 105 MW. Before performing the calculation phase, the superheater design stage determines the outer diameter, BWG (Birmingham Wire Gauge), and pipe thickness. The first calculates the dirt factor, followed by selection of the pressure drop on the flue gas and vapor side. The results of this superheater design meet the design requirements with a dirt factor of 0.003057 retention stipulation (0.003) and pressure drop on the flue gas and vapor side less than or equal to 2 psi.

The economizer design calculation also meets the design requirements at a dirt factor of 0.00349 (0.003) and the pressure drop on the exhaust side ≤ 0.1378952 bar and water ≤ 0.689476 bar.

III. Results And Discussion

A. Design Results

We note the parameters of the HRSG design as follows:

Pipe thickness (Table V) HRSG HP components of a standard pipe for typical HRSG models. This condition helps in increasing pressure dynamics and variation. HRSG has many tubes (Table VI) to improve the heat exchange area with exhaust gas flow. Since heat transfer in HRSG is conducted by convection, HRSG requires as much surface area as possible.

The HP HRSG downcomer tube area is equal to 0.865 m^2 which is higher than HRSG (0.6323 m^2) [21]. This condition is possible to increase the water head in the evaporator's downcomer, and it helps enter the natural circulation process and avoid backflow problems in the evaporator.

This difference is due to the significant pressure drop in the number of tubes in the evaporator.

TABLE V. THE RESULTS OF THE HRSG DESIGN

		Superheater		Evaporator		Economizer	
Parameter	Unit	HP	LP	HP	LP	HP	L.P.
Outer diameter of the pipe	cm	1.905	1.905	2.54	2.54	2.54	2.54
Inside diameter of the pipe	cm	1.4224	1.4224	2.0574	2.1742	2.0574	1.859
Pipe length per line	m	8.9977	8.9977	8.9977	8.9977	8.9977	8.9977
Number of pipes per row		70	70	40	40	40	40
Number of rows	line	7	1	44	23	4	4
Number of fins	fin	4	4	8	8	8	8
Pressure-drop (gas)	bar	0.069	0.01131	0.01129	0.1157	0.0703	0.0989
Pressure-drop (water for ECO in and out; EVAP in/steam for EVAP out; SUP in and out)	bar	0.0291	0.0345	0.02199	0.32336	0.22295	0.12728
Dirt factor		0.00306	0.00355	0.00309	0.00305	0.00349	0.00346

B. Specification of HRSG Dual Pressure Level

The HRSG analyzed in this study supplies two levels of pressure, namely high and low pressures, to a steam turbine. Gas turbine exhaust gas flows through HRSG, thus providing the heat energy used for steam production. HRSG contains two superheaters (one HP SUP and one LP SUP), two economizers (one HP ECON and one LP ECON, and two evaporators (one HP EVAP and one LP EVAP).

TABLE VI. SPECIFICATION OF HRSG DUAL PRESSURE LEVEL

Parameter	**Unit**	**Value**
Output	MW	35
Flue Gas Flow	kg/s	137.3
Flue Gas Input Temperature	C	500.43
Flue Gas Output Temperature	C	146.8
Feed Water Flow	kg/s	18.43
Feed Water Input Temperature	C	60.14
Flue Gas From		Natural Gas
Efficiency	%	69.48

Table VI indicates that the HRSG design can increase efficiency by up to 68.48 %, which exceeds the value indicated in the design made by Wahyu et al, who designed the propulsion system using HRG, namely 48.49% [22]. The optimization of the double pressure steam cycle (SDC) for waste heat recovery by Liya et al. is and 19.75 [23].

TABLE VII. COMPARISON OF MINIMUM PIPE THICKNESSES WITH DESIGNED PIPES

Component	Minimum pipe thickness [mm]	Design pipe thickness [mm]
Superheater Low Pressure	0.147	3.40
Evaporator Low Pressure	0.196	1.83
Economizer Low Pressure	0.196	3.40
Superheater High Pressure	0.104	2.41
Evaporator High Pressure	0.734	2.41
Economizer High Pressure	0.734	2.41

C. Feasibility Analysis of the Design Dimension

It is crucial to determine the pipe's outer diameter before assessing the thickness using the BWG. It is also important to calculate the pressure drop and impurity factors of HRSG components such as economists, evaporators, and superheaters at high and low pressures. The reviewed operation's feasibility is the thickness of the feed pipe.

D. Effects of Changes in Flue Gas Temperature on the Property of Water

Fig. 1 illustrates an increase in flue gas mass speed due to a temperature rise. Changes in the mass rate of the exhaust gas increase the amount of water entering the HRSG. This condition states that the amount of energy from the flue gas is directly proportional to the evaporated feed water's mass. HRSG is a significant component of the GTCC that utilizes the temperature of the exhaust gas produced by the gas turbine to provide steam. The gas turbine's combustion temperature is not always constant during operation [8]. This condition changes the exhaust gas turbine's temperature; therefore, there is a need to design the temperature changes' HRSG components. Changes in exhaust gas temperature are very influential on the mass rate of feed water needed by HRSG.

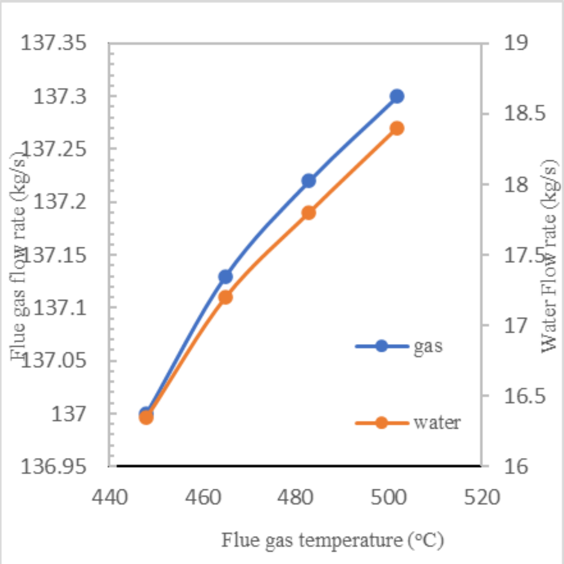

Figure 1. Effects of changes in flue gas temperature

They are used by the working fluid pressure and temperature at 56 bar and 6.2 bar, respectively. Table V indicates that the pipe thickness from design calculations is greater than the minimum allowable thickness. These data suggest that the HRSG components have met their mechanical requirements at low and high pressures.

IV. Conclusion

This paper uses equations and simulates the complete operating range of high and low pressure HRSGs. The equations for calculating the pressure drop and heat transfer in the HE use the same data. A high temperature, mass flow, and pressure for both sides, gas, and water/steam obtained for all components under investigation. The HRSG simulation shows that, although simple, the HSRG modeling is effective.

In future work, we will attempt to further improve the efficiency of HRSG by integrating simulation with the design using equations and changing the exhaust gas temperature variable. In future work, we will try to further improve the efficiency of HRSG by integrating simulation with the design using equations and changing the exhaust gas temperature variable.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Sri Wuryanti contributed up to the analysis and in charge of the whole direction and design. Rakhadian Dwi Jadmiko conducted a simulation. All authors discussed the results and contributed to the last manuscript.

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