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Research Article

Mathematical modeling and performance analysis of heat recovery steam generator at Shat-Al Basra power plant

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Abstract: Energy is crucial to economic and social development. The increasing demand for electricity in the world is met by using various primary energy sources. Combined cycle gas turbines (CCGTs) are highly efficient power-generation plants due to their high temperatures and utilization of exhaust gases to generate additional power. Heat recovery steam generator (HRSG) is a very important component in CCGT, this component recovers the energy from flue gases exiting the gas turbine and generates the motive steam. In the present work, HRSG in Shatt-Al Basrah power plant has been simulated using a mathematical model to predict the temperature of the steam out of HRSG in different cases. Different parameters have been studied including the fuel of the power plant and the ambient temperature to calculate heat transfer area of each section and compare the results with actual data which shows a great agreement. The results show that maximum heat transfer occurs in the high-pressure evaporator section due to its large heat transfer area. It is also noticed that the highest LP steam temperature observed is 251.1°C at 15°C light diesel oil case and the highest HP steam temperature is 473.62°C also at 15°C light diesel oil case.

Keywords: *Combined cycle power plants, Heat exchanger, HRSG design, Waste heat recovery*

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Nomenclature					
A	m^2	Heat transfer area	K_m	w m /°C	Metal thermal conductivity
A_i	m^2/m	Inside surface area	L	m	Length of tube
A_o	m^2/m	Obstruction surface area	*LMTD*	C	Log mean temperature difference
A_t	m^2/m	Total surface area	m	kg/s	Mass flow rate
A_w	m^2/m	Area of tube wall	n	---	Number of fins per meter
b	m	Fin thickness	N_r	---	Number of tubes per row
C_P	J/kg °C	Heat capacity	N_w	---	Number of rows
d	m	Tube outer diameter	Q	w	Heat transfer rate
d_i	m	Tube inner diameter	S	m	Fin spacing
E	---	Fin efficiency	S_L	m	Longitudinal pitch
f_{fi}	m^2s °C /kJ	Inside fouling factors	S_T	m	Transferred pitch
f_{fo}	m^2s °C /kJ	Outside fouling factors	T_b	C	Outside fluid temperature
G	kg/m^2 s	Gas mass velocity	T_g	C	Exhaust gas temperature
h_o	w m^2 /°C	Outside heat transfer coefficient	T_s	C	Fin temperature
h_i	w m^2 /°C	Inside heat transfer coefficient	T_w	C	Water/steam temperature
h_f	m	Fin Hight	U	w/m^2 °C	Overall heat transfer coefficient
K	w m /°C	Fluid thermal conductivity	u	m/s	Exhaust gas velocity
Greek symbols					
η		Effectiveness	μ	Pa s	Viscosity
λ	kJ/kg	Latent heat of vaporization	ρ	Kg/m^3	Density
Subscripts					
g		Exhaust gas	*VHP*		high pressure evaporator
w		Water/steam	*SHP*		high pressure superheater
HRSG		Heat recovery steam generator	*LDO*		Light Diesel Oil
PREH		Preheater	*HFO*		Heavy Fuel Oil
VLP		Low pressure evaporator	*PP*		Pinch point
EHP		High pressure Economizer	*AP*		Approach point
SLP		Low pressure superheater			

1. INTRODUCTION

1.1. Waste Heat Recovery Systems

Waste heat recovery (WHR) is an important process for utilizing waste heat and providing additional heat and electricity. As energy demand increases, conventional energy generators based on fossil fuels and gas turbine power plants are commonly used because of their advantages; however, their thermal efficiency is low, resulting in wasted heat. WHR can solve these issues using waste energy to generate electricity or process heat. Different technologies can be applied to gas-turbine-based power plants to recover waste heat. Implementing WHR systems can increase plant efficiency, while reducing CO_2 emissions and costs [1]. In conclusion, waste heat recovery is a game-changing process having immense potential for the efficient utilization of energy resources. By effectively recovering waste heat from gas turbine power plants, we can significantly increase efficiency, reduce CO_2 emissions, and reduce costs. It is imperative to embrace and implement WHR on a large scale to pave the way for a sustainable and eco-friendly future [2,3].

1.2. Shatt Al-Basra Power Plant

Shatt Al-Basra Gas Power Plant is one of the key power stations located in Basra, in the southern Iraq. The plant consists of ten gas turbines, each of which generates 125 MW of electrical power. It operates on two types of fuel: Heavy Fuel Oil (HFO) and Light Diesel Oil (LDO). This power plant was later upgraded to a combined-cycle system by adding ten Heat Recovery Steam Generators (HRSGs) and five steam turbines. The HRSGs recover the waste heat from the gas turbines and convert it into steam, which is supplied to the steam turbines [4].

1.3. Design of Waste HRSG

With the growth of the global population and rapid expansion of industrial activities, the demand for electricity is rising significantly in both developed and developing countries. This increasing demand has driven researchers and engineers to seek more efficient and environmentally friendly power generation technologies that can reduce both electricity and pollutant emissions [5,6]. One of these technologies is waste heat recovery power plants, which combine a gas cycle (Brayton cycle) and a steam cycle (Rankine cycle) through an HRSG. In these systems, heat from the exhaust gases of the gas turbine is captured by the HRSG to generate steam at an optimal pressure and temperature. This steam is then used to drive the steam turbine, producing additional electricity with improved efficiency [7].

HRSGs are categorized into single-, dual-, and triple-pressure types depending on the number of drums in the boiler. Among these, dual-pressure HRSGs are the most used because they provide higher efficiency than single-pressure systems and are more cost-effective than triple-pressure designs [8,9]. The efficiency and power output of combined-cycle power plants depend largely on the design of the HRSG. Therefore, optimizing its design is crucial for maximizing heat recovery and improving the overall performance of the plant [10].

An HRSG is not a single heat-exchanger unit, instead, this equipment consists of a preheater, low- and high-pressure evaporators, two economizers, and a low- and high-pressure superheater in the Shatt-Al Basra power plant. The economizer heats up the condensed water using the extracts thermal energy from the exhaust gas. Then, the heated water enters the evaporator section resulting in its vaporization into steam by the flue gas. The saturated steam subsequently undergoes further heating in the superheater, where it interacts with the flue gas to become superheated. Fig. 1 illustrates a schematic diagram of the HRSG system, depicting its 7 components. The high-temperature exhaust gas exiting the gas cycle flows past the superheaters, evaporators, economizers and preheater before being exhausted to the surroundings. The temperature of the flue gas successively decreases from Tg1 to Tg2, Tg3, Tg4, Tg5, Tg6, Tg7, and Tgo. Feed water enters the preheater at temperature Tw1 and increased to Two.

Figure 1. HRSG in Shatt-Al Basra power plant.

In the current market, several standardized gas turbine systems are available. However, Heat Recovery Steam Generators (HRSG) are typically designed individually for each Combined Cycle Power Plant (CCPP). The reasons for this include the following [11]:

- *The fuels utilized are diverse. Consequently, the requirements of the relevant flue gas should be considered in the Heat Recovery Steam Generator (HRSG) design.*
- *Cycle optimization differs in the Combined Cycle Power Plant (CCPP).*
- *Ambient conditions fluctuate and influence the Gas Turbine (GT) output.*

A significant body of research has focused on enhancing the performance of combined cycle power plants through optimal HRSG design. Ahmed et al., [12] developed a mathematical model for designing an HRSG in a gas turbine power plant. The study analyzed heat flow and energy consumption using a dual-pressure system to optimize heat recovery and minimize thermal losses. The results showed that the proposed system could increase the plant's efficiency from 33% to 52% while reducing exergy losses by 35%. Nag et al., [13] designed an HRSG to generate saturated steam in a combined gas and steam cycle, aiming to minimize irreversibility in the system. Through their analysis, they clarified that operating the HRSG at full load reduces entropy generation, leading to improved thermodynamic efficiency. This suggests that maximizing the utilization of available waste heat enhances overall system performance by minimizing energy losses. In a different work, Kaviri et al., [14] investigated an optimized design of HRSG in CCPPs to minimize environmental impact. They found that increasing the HRSG inlet gas temperature up to 650°C improved both thermal and exergy efficiency.

On the one hand, Naemi et al., [15] focused on the design parameters of a dual-pressure HRSG to improve its heat recovery efficiency and ensure economic performance. They developed a mathematical model for dual pressure HRSG and the results noted that; the optimum pinch point with regard to thermodynamic aspects was 2.5 °C and 2.1 °C for HRSGs employing steam at 75 bar and 90 bar correspondingly. Feng et al., [16] worked on the role of heat exchanger design in enhancing the thermal efficiency of numerous multi-pressure HRSGs. The authors tested three configurations and emphasized the significance of optimal design of heat exchangers early in the development stage in order to improve heat recovery efficiency of the HRSG and overall efficiency of the combined cycle plant.

Durán et al., [17] proposed a method for developing the design of an HRSG, which has three sections (economizer, evaporator and one superheater) by obtaining a small heat transfer area and low-pressure losses. In a different work, Hajabdollahi et al., [18] developed a thermodynamic and thermoeconomic model for HRSG. They performed exergoeconomic analysis and multi-optimization to minimize the cost per unit of steam produced while maximizing HRSG exergy efficiency. They found that increasing the high and low-pressure drums enhances efficiency, while a higher pinch point reduced it. Additionally, increasing the HRSG inlet gas enthalpy improved exergy efficiency according to the literature. Cehil et al., [19] proposed a novel mathematical model to optimize the heat exchanger layout within HRSGs using a genetic algorithm. Their work highlighted the impact of pressure levels, reheating, and serial/parallel configurations on thermodynamic efficiency. The study emphasized the need for simultaneous optimization of layout and operating parameters to improve HRSG performance. Chantasiriwan [20] focused on determination of the optimal total heat transfer area of single-pressure HRSGs by combining thermodynamic modeling with economic evaluation. This study showed that while increasing HRSG area improved power output, the benefit diminished with higher cost, leading to an optimal economic size. These studies provided valuable insights supporting the present work, which integrates thermodynamic modeling and real plant data to evaluate HRSG performance under actual operating conditions.

The design process of an HRSG begins with estimating its steam generation capacity and determining the temperature profiles of both the gas and steam. The steam flow rate and exit temperature are typically assumed based on conventional fired steam generator specifications. These two factors play a key role in defining the required sizes of the superheater, evaporator, and economizer [21]. This study offers a validated thermodynamic model of a real HRSG system at Shatt Al-Basrah power plant using actual operational data. Unlike prior works which relied on assumed or standard data, our work confirms the model’s applicability to real-world conditions. The model was also extended to investigate the potential performance variations under alternative fuel inputs, indicating flexibility and applicability in future optimization studies. The originality lies in applying the model to a real-world, single-site facility in a developing region, reflecting actual combustion gas profiles, steam generation behavior, and system losses. Additionally, the study builds a practical framework that can later be expanded to evaluate the integration of HRSGs with downstream thermal processes, such as desalination units, offering a realistic pathway for enhancing overall energy utilization in gas-fired power stations.

2. METHODOLOGY

Mathematical models are used to simulate actual processes. The HRSG model depends on the mass flow rates, fluid dynamics, and heat transfer and energy balance [12]. The design process of HRSG uses two main parameters that are related to the temperature profiles of gas and steam side, these are pinch point temperature and approach point temperature. Pinch point is the difference between the gas temperature leaving the evaporator and saturation temperature. Approach point is the difference between the saturation temperature and the water temperature entering the evaporator Fig. 2 show a pinch point and approach point for low pressure side [22].

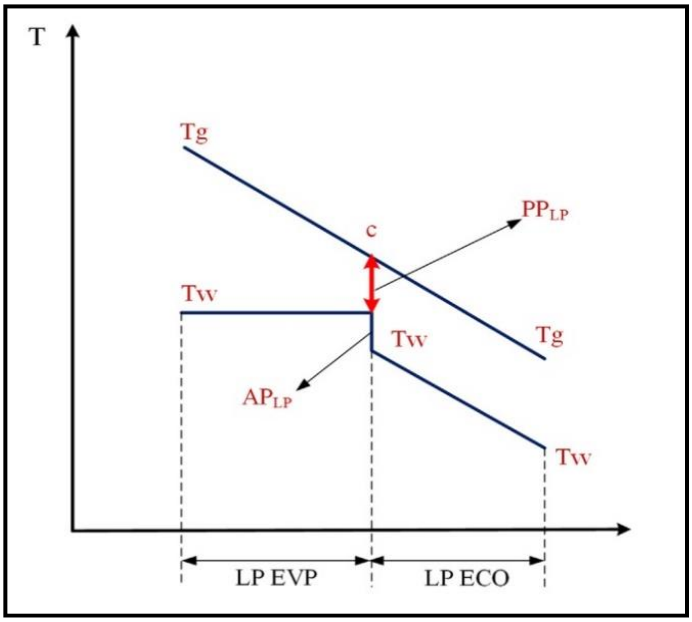


Figure 2. Pinch point and approach point.

Mathematically on ecan formulate the equations as follows for the dual pressure system,

$$PP_{LP} = T_{g\,7} - T_{w\,3} \tag{1}$$

$$AP_{LP} = T_{w\,3} - T_{w\,2} \tag{2}$$

$$PP_{HP} = T_{g\,3} - T_{w\,7} \tag{3}$$

$$AP_{HP} = T_{w\,7} - T_{w\,6} \tag{4}$$

2.1. Assumptions

In order to derive the mathematical model, some assumptions are taken, these are listed below:

1. A steady state system.
2. The flow rate, temperature and chemical composition of exhaust gas are known.
3. The tube type is staggered pitch with solid fins.
4. The natural circulation is assumed in the evaporator.
5. No heat loss in the system other than the outflow of the exhaust gas.
6. Radiation heat transfer is negligible.

2.2. Temperature Profile

In the first step of the design process, establishing the temperature profile is crucial. Through the application of energy balance principles and the utilization of an appropriate pinch point, it is possible to determine the inlet and outlet temperatures of both the working fluid and exhaust gases for each section. This calculation serves as the foundation for subsequent design steps. The equation used to develop the temperature profile is the heat balance equation according to Ref. [23]:

$$\text{Total Heat Exchange} = Q_{cold} = Q_{hot} \tag{5}$$

Energy balance in all heat transfer surfaces between the hot and cold streams (gas side and water/steam side) can be expressed as,

For high-pressure superheater,

$$m_g\, C_{P_g}(T_{g1}\text{-}T_{g2})=m_{HPw}\, C_{P_w}(T_{w\,O}\text{-}T_{w\,7}) \tag{6}$$

For high-pressure evaporator,

$$m_g\, C_{P_g}(T_{g2}\text{-}T_{g3})=m_{HPw}\, \lambda_{HP} \tag{7}$$

For high-pressure economizer 2,

$$m_g\, C_{P_g}(T_{g3}\text{-}T_{g4})=m_{HPw}\, C_{P_w}(T_{w\,6}\text{-}T_{Hw\,5}) \tag{8}$$

For high-pressure economizer 1,

$$m_g\, C_{P_g}(T_{g5}\text{-}T_{g6})=m_{HPw}\, C_{P_w}(T_{Hw\,4}\text{-}T_{w\,3}) \tag{9}$$

For low-pressure superheater,

$$m_g\, C_{P_g}(T_{g4}\text{-}T_{g5})=m_{LPw}\, C_{P_w}(T_{Lw\,5}\text{-}T_{Lw\,4}) \tag{10}$$

For low-pressure evaporator,

$$m_g\, C_{P_g}(T_{g6}\text{-}T_{g7})=m_{LPw}\, \lambda_{LP} \tag{11}$$

For the preheater,

$$m_g\, C_{P_g}(T_{g7}\text{-}T_{g\,o})=m_w\, C_{P_w}(T_{w\,2}\text{-}T_{w\,1}) \tag{12}$$

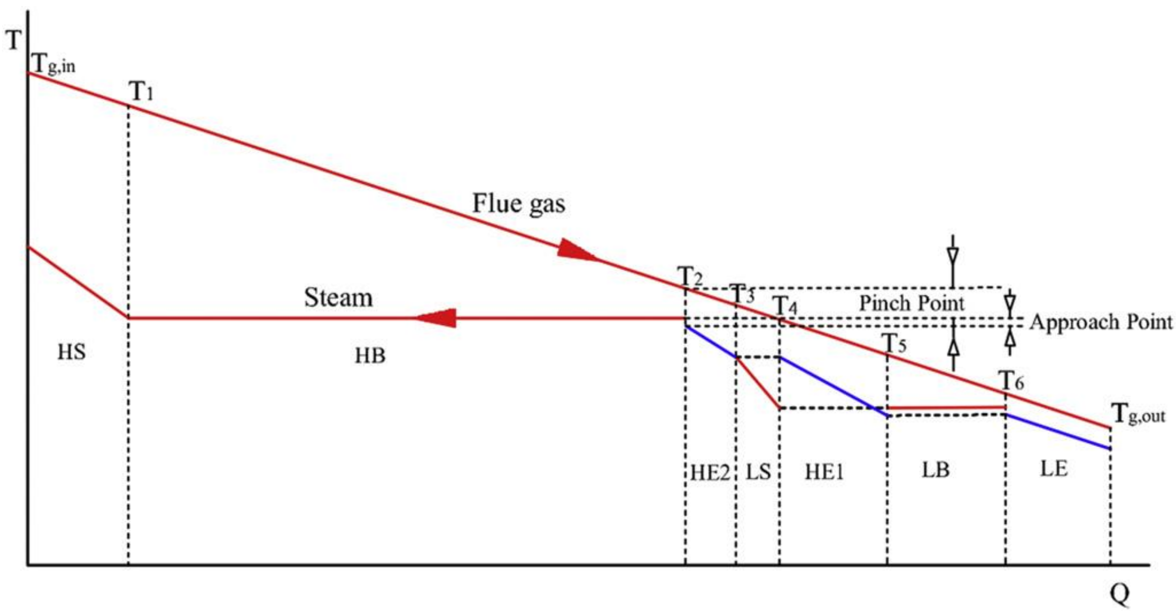


Figure 3. Gas and steam temperature profile of dual pressure HRSG after Ref. [15].

2.3. Calculating Heat Transfer Area

The design method in this study leverages the heat transfer coefficients to obtain the heat transfer area for each heat exchanger in the high- and low-pressure regions of the HRSG. The designing method is based on logarithmic means temperature difference; therefore, the heat transfer areas are obtained by:

$$Q = UA \ \ LMTD \ , \tag{13}$$

With,

$$LMTD = \frac{\left[(T_{g1}\text{-}T_{w2}\right)\text{-}(T_{g2}\text{-}T_{w1})]}{Ln\left[(T_{g1}\text{-}T_{w2}\right)\text{-}(T_{g2}\text{-}T_{w1})]} \tag{14}$$

The overall heat transfer coefficient of each section is defined by the following equation according to Ref. [22]:

$$\frac{1}{U} = \frac{A_t}{h_i A_i} + ff_i \left(\frac{A_t}{A_i}\right) + ff_o + \left(\frac{A_t}{A_w}\right)\left(\frac{d}{2K_m}\right) ln\left(\frac{d}{d_i}\right) + \frac{1}{\eta h_o} \tag{15}$$

2.4. Overall Heat Transfer Coefficient Calculation

Considering the use of finned tubes in all sections of the HRSG, the calculation of the overall heat transfer coefficient would be more complicated than that for bare tubes. The outside heat transfer coefficient of finned tubes can be calculated using various methods. Convection heat transfer coefficient is considered in the calculation of flue gas heat transfer coefficient. For instance, Ganapathy proposed a methodology for the calculation of the finned tube heat transfer coefficient [21,22]. The calculation of the inside and outside heat transfer coefficients are presented in the following sections.

2.4.1. Average inside heat transfer coefficient

The average inside heat transfer coefficient is defined by,

$$h_i = 0.024 \, Re^{0.8} P_r^{\,0.4} \frac{K}{d_i} \tag{16}$$

Reynolds number is obtained by the following equation,

$$Re_i = \frac{\rho u d_i}{\mu} \tag{17}$$

And Prandtl number is formulated by,

$$P_r = \frac{C_P \mu}{K} \tag{18}$$

2.4.2. Average outside heat transfer coefficient

Radiation heat transfer is negligible so the convection heat transfer coefficient is considered in the calculation of the average outside heat transfer coefficient. Figs. 4(a,b) illustrate the geometry and arrangement of serrated and solid finned tubes [24].

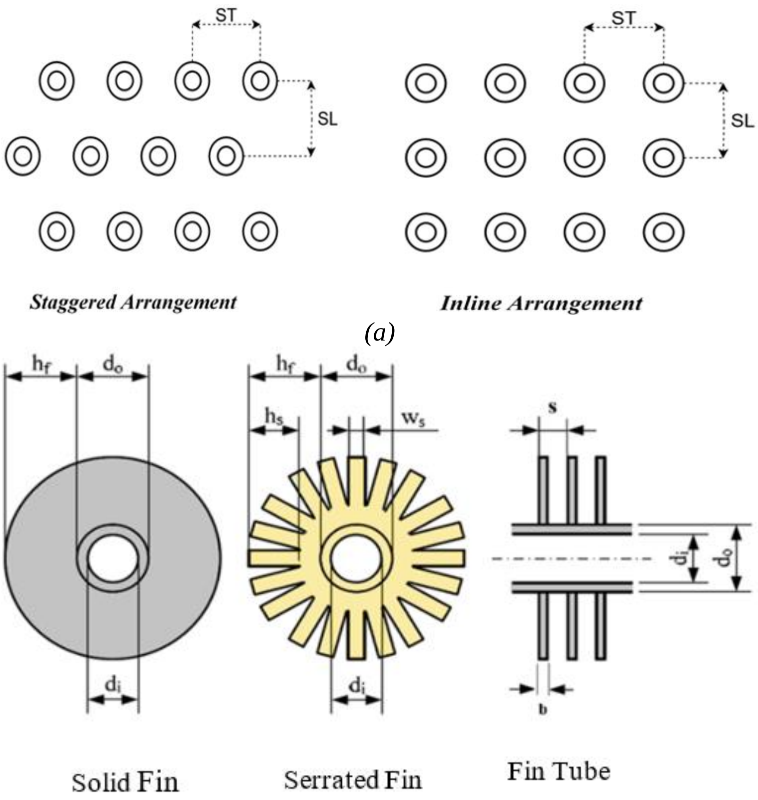


Figure 4. (a) Fin tube Arrangement and (b) fin tube geometry.

In Shatt AL-Basrah power plant, the HRSG has a solid fin and staggered tube arrangement. The average outside convection heat transfer coefficient is then calculated by the following expression,

$$ho = C1\ C3\ C5 \left(\frac{d + 2hf}{d}\right)^{0.5} \left(\frac{Tb + 273.15}{Ts + 273.15}\right)^{0.25} G\ Cp \left(\frac{K}{\mu Cp}\right)^{0.67} \tag{19}$$

Here, the coefficients C1, C2 and C3 have been calculated by,

$$C1 = 0.25\, Re_o^{-0.35} \tag{20}$$

$$C3 = 0.2 + 0.65\, e^{-\frac{0.25\, h}{s}} \tag{21}$$

$$C5 = 1.1 + \left(0.75 - 1.5\, e^{-0.7Nr^2}\right) e^{-\frac{2\, SL}{ST}} \tag{22}$$

Fin spacing (S) can be found by,

$$s = \frac{1}{n} - b \tag{23}$$

Eq. (24) calculates mass flow rate of flue gas,

$$G = \frac{M_g}{[ST - A_o] N_w L} \tag{24}$$

Where,

$$A_o = d + 2\, n\, b\, hf \tag{25}$$

In this regard, the Reynolds number is calculated with the formulation,

$$Re_o = \frac{Gd}{\mu} \tag{26}$$

tube inner surface area A_i (m^2) calculated by:

$$A_i = \pi\, d_i \tag{27}$$

average wall surface area Aw (m^2) calculated by:

$$A_w = \frac{\pi\, d + d_i}{2} \tag{28}$$

average fin temperature T_s calculated by,

$$T_s = \frac{T_{w1} + T_{w2}}{2} + 0.3\left(T_b - \frac{T_{w1} + T_{w2}}{2}\right) \tag{29}$$

Where T_b is average outside fluid temperature,

$$T_b = \frac{T_{g1} + T_{g2}}{2} \tag{30}$$

effectiveness η is,

$$\eta = 1 - (1 - E)\,\frac{A_f}{A_t} \tag{31}$$

and

$$A_f = \pi\, n(\,2d\, hf + 2hf^2 + bd + 2b\, hf) \tag{32}$$

$$A_t = A_f + \pi\, d(1 - n\, b) \tag{33}$$

$$E = \frac{1}{\left(1 + 0.002292\;\; m^2\, hf^2\, \left(\frac{d + 2\, hf}{d}\right)^{0.5}\right)} \tag{34}$$

where,

$$m = \left(\frac{2h_c}{K_f b}\right)^{0.5} \tag{35}$$

3. RESULTS AND DISCUSSION

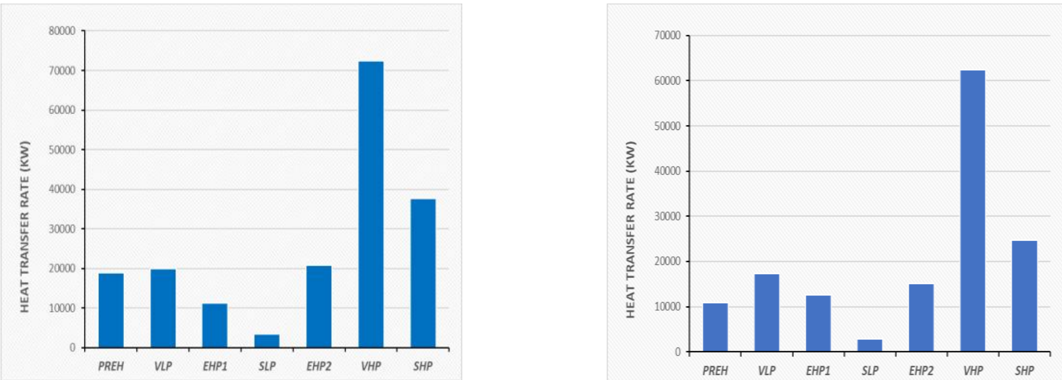
As mentioned before this power station operates on two types of fuel: (a) Heavy Fuel Oil (HFO) and (b) Light Diesel Oil (LDO). Thus, the exhaust gas temperature and composition will vary depending on the type of fuel used, and also each case will have different pinch point and approach point. Therefore, the model will incorporate the specific properties of each fuel type separately.

Table 1. Parameters of each operation cases at different fuel.

Parameter	Unit	LDO	HFO
Exhaust gas temperature	ºC	537.1	475.2
Exhaust gas flow rate	Kg/s	429.72	427.2
Low-pressure pinch point	ºC	8.28	6.05
High-pressure pinch point	ºC	7.19	5.81
Low-pressure approach point	ºC	16.4	6.65
High-pressure approach point	ºC	2.7	2.46
Component of exhaust gas:			
CO_2	V%	4.55	3.98
O_2	V%	13.44	14.56
N_2	V%	74.32	75.12
H_2O	V%	6.8	5.44
Ar	V%	0.89	0.89

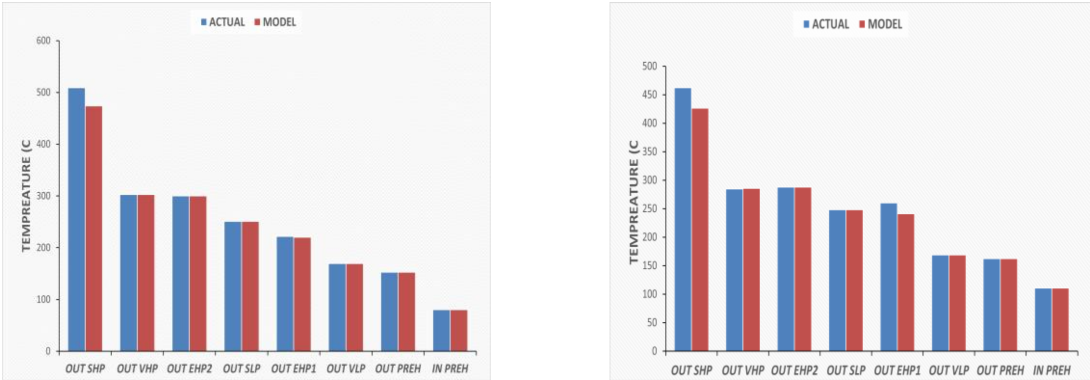
The mathematical model has been developed and solved using Excel, chosen for its user-friendly interface and high computational accuracy since it offers several advantages in solving mathematical models, including its ability to handle large datasets, perform complex calculations efficiently, and provide built-in functions for numerical analysis. Additionally, its flexibility in data visualization and ease of integration with external sources make it a practical with both datasets seamlessly integrated tool for engineering and scientific applications. The model was supplied with the properties of water, steam, and other gases from a separate Excel sheet. Fig. 5 illustrates the heat transfer from exhaust gases to steam/water within each heat exchanger of a heat recovery steam generation (HRSG) system, across various fuel types (LDO, HFO). The analysis shows that the highest heat transfer occurs in the high-pressure evaporator (VHP), while the lowest occurs in the low-pressure Superheater (SLP). Specifically, the figures show that with a LDO fuel type, the LP steam production absorbs 26,220.42 KW of energy (14.22% of the total heat recovered), while the HP steam production absorbs 158,112.72 KW (85.78%). With a HFO fuel type, the LP and HP steam production account for 22,001.91 KW (15.08%) and 123,872.16 KW (84.92%) respectively. Fig. 6 show temperatures of the water/steam within each heat exchanger in HRSG, comparing model predictions with actual plant data, demonstrating strong agreement across the operational conditions for different fuel types. The error is 2.234% for the Case #1 and 2.789 % for Case #2. Similarly in Fig. 7, temperatures of the flue gases exiting each exchanger are presented, again showing very good agreement between the model and actual data for different fuel types. The error is 3.447% for Case #1 and 4.349 % for Case #2.

Figs. 8 and 9 show the temperature profile for both the exhaust gases and the water/steam sides of each heat exchanger, along with the heat flux. The exhaust gas temperature is highest at the HRSG inlet and decreases progressively through the system, reaching its lowest value at the PREH outlet. Conversely, the water/steam temperature is lowest at the inlet and highest at the HP superheater outlet, reflecting the heat transfer from the exhaust gases. The diagrams highlight the temperature behavior in both the HP and LP sections including pinch points and approach points temperatures. Note that the temperature profiles of each case of types of fuels are seen clearly.



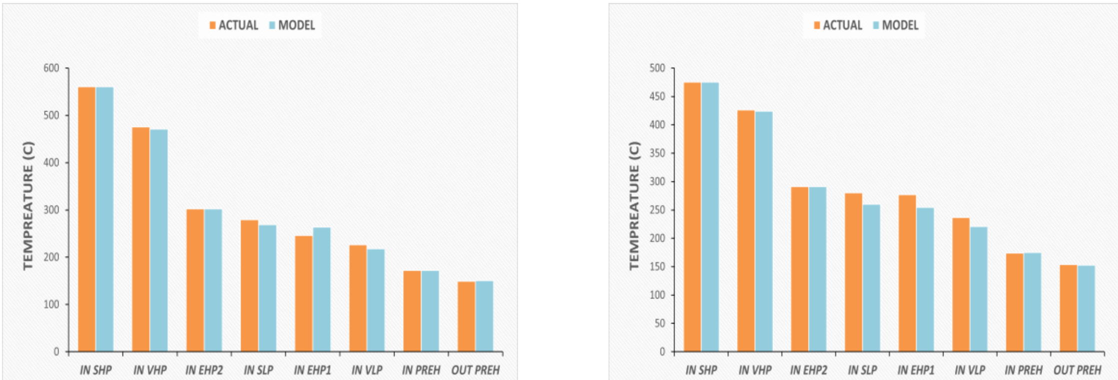
(a) *(b)*

Figure 5. The heat transfer rates at (a) Case 1 (LDO) and (b) Case 2 (HFO)



(a) *(b)*

Figure 6. Temperatures of the water/steam side at (a) Case 1 (LDO) and (b) Case 2 (HFO).



(a) *(b)*

Figure 7. Temperatures of the exhaust gas side at (a) Case 1 (LDO) and (b) Case 2 (HFO).

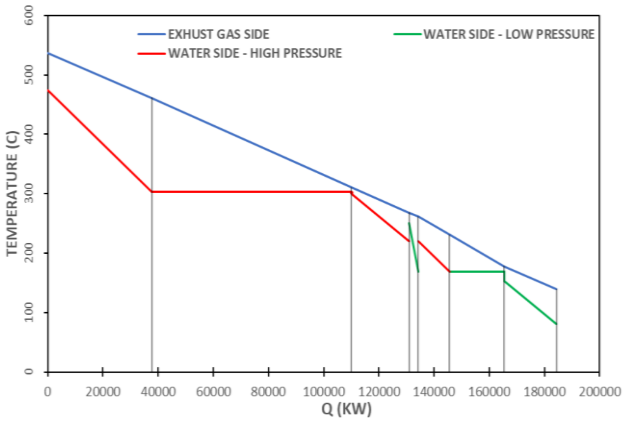


Figure 8. Temperature profile for HRSG at case 1 (LDO).

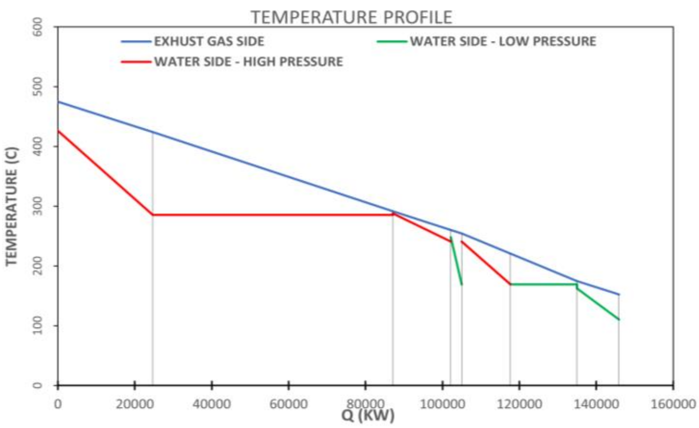


Figure 9. Temperature profile for HRSG at case 2 (HFO).

Fig. 10 shows the heat transfer surface area for each exchanger by comparing mathematical model results with actual values and showing close agreement with an error of 5%. The HP evaporator exhibits the largest surface area (i.e. 22243.56 m^2), consistent with its role as the primary heat transfer component within the HRSG.

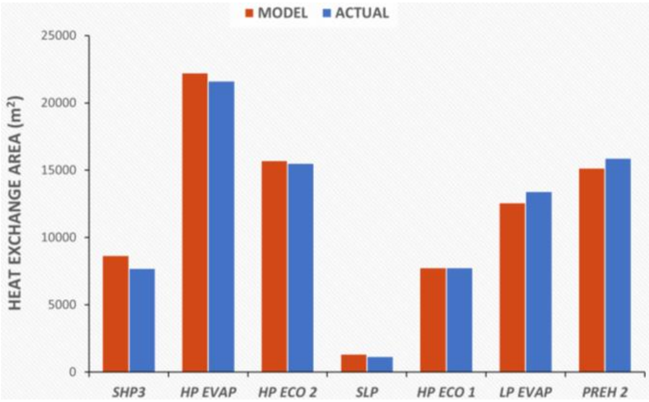


Figure 10. Heat transfer area for each heat exchanger.

4. CONCLUSION

This research established a validated mathematical model for simulating the performance of the Heat Recovery Steam Generator (HRSG) at the Shatt Al-Basra power plant, with a particular emphasis on its thermal behavior across varied operational scenarios. This model established some essential parameters like exhaust gas temperature, exhaust gas mass flow rate, and fuel type, enabling accurate calculations of steam temperatures and heat exchange areas in each level of HRSG. The good correlation between simulated and actual plant data shows that the developed model is accurate. High-pressure evaporator has the highest heat transfer rate and the greatest heat exchange area. The study also detailed how system performance is affected by fuel type, where using Light Diesel Oil (LDO) results in higher exhaust gas temperatures than Heavy Fuel Oil (HFO) will provide, increasing steam temperatures and overall efficiency.

The results of the present work will, thereby, be beneficial towards improving overall waste heat recovery, overall power plant efficiency enhancement and thus improvement in the HRSG plant design through advanced thermodynamic analysis and validation with matching plant data. These findings hold implications for engineers and researchers to enhance energy recovery, cut emissions, and advance the sustainability of power generation systems. The validated model offers a reliable foundation for optimizing heat recovery performance in gas power plants, especially in developing regions. As a forward step, this model can be extended to assess the performance of integrated HRSG-desalination systems, enabling dual-purpose energy-water optimization in arid climates.

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