Modeling and optimization of finless and finned tube heat recovery steam generators for cogeneration plants

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Abstract
This article aims to model and optimize the fire tube Heat Recovery Steam Generator (HRSG) used in a gas microturbine cogeneration of heat and power system. Here, six cases including a finless and finned tube (solid and serrated) HRSGs with the inline and staggered arrangements are optimized and compared using techno-economic assessment. Optimization of HRSG was carried out by applying two objective functions of minimizing the total annual cost as well as maximizing the exergy efficiency. Nondominated Sorting Genetic Algorithm II is used to find out the optimal values of design variables. The TOPSIS decision-making process is employed to choose the optimum point of the Pareto front. A 600 kW gas microturbine cogeneration plant is considered as a case study. The results revealed that the finless tube HRSG with an inline arrangement has the lower pinch and approach temperatures (3.5°C and 3.1°C, respectively), the higher steam mass flow rate (12.7%), the lower total annual cost (64096$/year), and the higher exergy efficiency (60.6%). It is also found that using the staggered arrangement tube banks along with the solid fins in HRSG leads the total annual cost, the steam mass flow rate, and exergy efficiency increase up to 41.6%, 100%, and 12.6%, respectively. This means that the thermodynamic performance improvement will compensate the total annual cost increment. Furthermore, it is demonstrated that HRSG with solid fins and staggered arrangement with total annual costs of (90810$/year), exergy efficiency of (73.2%), and steam mass flow rate of (4680 kg/h) has the best performance in comparison with other finned tube cases.

KEYWORDS
cogeneration of heat and power systems, gas microturbine, genetic algorithm, heat recovery steam generator, inline and staggered arrangement, multiobjective optimization

1 | INTRODUCTION

In combined heat and power (CHP) systems, the prime mover waste heat is recovered to produce hot water/steam for heating or power generation. Heat recovery along with power generation, increases the efficiency of CHP systems up to...
80%. The most important equipment which is used for the waste heat from recovery is the heat recovery steam generator (HRSG). Hot exhaust gases move through the HRSG to make warm water or steam. Steam can expand through a turbine of a Rankine cycle to generate electricity. HRSGs are well known as two types of heat exchangers: water tube and fire tube. In the water tube boiler, the water goes through the pipes and the gas goes through the shell while it is opposite in the fire tube boiler.

There are numerous research activities in which authors have modeled and/or optimized water tube or fire tube HRSGs. These HRSGs can be used in Combined Cycle Power Plants (CCPP), CHP systems, or in other cycles (tri-generation, hybrid, and so on) which are surveyed in the following.

1.1 Application of HRSG in CCPP

Since the prime mover of CCPP (power output higher than 25 MW) is gas turbine, the temperature and mass flow rate of exhaust gas are relatively high and the waste heat should be recovered in two or three levels. Therefore, double or triple pressure level water tube HRSG is frequently used in this application.

Srinivas studied a double pressure water tube HRSG in order to maximize the heat recovery and improve the CCPP performance. Each of the heating section in HRSG is solved from the local flue gas condition with an aim of getting minimum possible temperature difference. It was shown that the optimum compressor pressure ratio decreases with an increase in HRSG high pressure level. Feng and Zhong studied the impact of HRSG’s components layout on the amount of recovered heat. In this regard, a dual pressure HRSG under three different layouts was considered and water/steam temperature and pressure, steam mass flow rate, and heat efficiency of different heat exchangers layout of HRSG were analyzed. The results show that the optimization of heat exchangers layout of HRSGs has a great significance for waste heat recovery and energy conservation. Bahrampoury and Behbahani employed the multiobjective optimization method in order to optimize a water tube HRSG. Two objective functions including irreversibility and the equivalent volume of HRSG in which the former one represents the exergy efficiency and the later one represents the HRSG costs, were considered. The results revealed that by increasing the equivalent volume of HRSG, the exergy efficiency is increased up to (76.3%) and also increasing the normalized equivalent volume from 0.31 to 0.85 increases the efficiency only by 1.15%. Sharma and Singh studied a finned water tube HRSG for a 663 MW CCPP. They found that increase in fin height by 3 units offers the reduction in heat transfer by 20% and fin segmentation improves gas penetration resulting in higher heat transfer.

Saidur et al. modeled industrial water tube steam generators from the energy and exergy-economic points of view and determined the exergy and energy efficiencies. They found that heat exchanger and combustor are the main parts that contributed loss of exergy. It was also shown that the lower pinch point leads to a higher steam mass flow rate.

Parametric analysis of water tube HRSG used in CCPP is performed by Sharma and Singh. They investigated effects of fin density, fin height, fin type, and tube diameter on pressure drop and overall heat transfer coefficient in HRSG. The results showed that the higher fin densities for any surface does not make any difference in heat transfer analysis as it acts as a solid fin design and increasing fin height by 18% offers a reduction in heat transfer by 15%.

Ghazi et al. optimized a water tube HRSG in order to find the optimum design parameters using single objective optimization. Total cost per unit of produced steam exergy was considered as the objective function. The objective function includes investment cost, operational and maintenance cost, and the corresponding cost of the exergy destruction. Design variables were high and low drum pressures, steam mass flow rates, pinch point temperature differences, and the duct burner fuel consumption flow rate. The results showed that with increasing the exergy unit cost, the optimum values of design parameters are selected such that to decrease the objective function. In addition, it was shown that the higher inlet gas enthalpy increases the required heat transfer area and consequentially the start-up cost. Also it was demonstrated that the lower pinch point results in a bigger heat transfer area, higher exergy efficiency, and lower operational cost.

Casarosa and Donatini optimized single, double, and triple pressure water tube HRSGs considering the number of pressure levels, the pressures, the mass flow rate ratio, and the inlet temperatures to HRSG as design parameters. Thermodynamic optimization was carried out based on the minimization of exergy losses. It was shown that just by optimizing the heat recovery and the steam cycle operating parameters, it seems possible to reach overall combined cycle efficiencies close to 60% on existing plants.

Nadir and Ghenai optimized a water tube HRSG using particle swarm optimization (PSO) algorithm. A thermodynamic comparison between the optimums of three configurations of HRSG operating at exhaust gas temperature from 350°C to 650°C was accomplished. They found that adding another pressure level allows achieving a higher pressure at
the inlet of high pressure turbine, producing more steam quantities, destroying less exergy and finally producing more specific work. Franco and Russo\textsuperscript{10} optimized a water tube HRSG using thermodynamic and thermo-economic objective functions. Thermodynamic optimization has the purpose to diminish energy losses while the aim of the thermo-economic optimization is the minimization of a cost function, sum of the cost of exergy inefficiencies, and the cost of the HRSG. The results show that utilizing thermo-economic objective function leads to the plant’s thermal efficiency increment up to 60%.

Exergy analysis of the HRSG for calculating exergy losses, heat transfer, and pressure losses for different physical components is conducted by Sharma and Singh.\textsuperscript{11} Their results showed that fin density, fin thickness, fin height, tube diameter, and fin spacing have a noticeable effect on exergy loss minimization. Mehrgoo and Amidpour\textsuperscript{12,13} and Norouzi et al\textsuperscript{14} presented a method for design and optimization of water tube HRSG which is known by constructal theory. They optimized water tube HRSGs using genetic algorithm considering total entropy generation and capital cost as objective functions. Their results showed that the use of several pressure levels in HRSGs causes a considerable increase in the power production, declines irreversibility in HRSGs, and allows producing higher steam flow rate.

Behzadi et al\textsuperscript{15} performed multiobjective optimization and exergo-economic analysis of waste heat recovery from Tehran’s waste-to-energy plant considering exergy efficiency and total product unit cost as objective functions. The optimum exergy efficiency and total product unit cost were obtained 19.61% and 24.65$/GJ, respectively. Nikbakht Naserabad et al\textsuperscript{16} carried out multiobjective optimization of single and double pressure level water tube HRSGs for full repowering of Bandar Abbas steam power plant. Optimization is performed considering exergy efficiency and cost of electricity generation as objective functions and using genetic algorithm method. The results of the multiobjective optimization of repowering with two single-pressure and double pressure level HRSGs indicate an increase in the exergy efficiency of the plant up to a level above 46%. Mokhtari et al\textsuperscript{17} investigated the effect of gas turbine and HRSG on power generation in order to reduce exergy destruction and power loss in the gas turbine. Power loss due to pressure drop in the gas turbine and the electricity cost are considered as two objective functions for optimization. It was concluded that with an increment in compressor pressure ratio, the steam generation increases while electricity cost decrease.

Rezaie et al\textsuperscript{18} optimized a water tube HRSG including two super-heaters, one evaporator and one economizer using genetic algorithm considering flue gas pressure drops, heat transfer surface area, and heat transfer rate as objective functions. The optimum values of parameters and optimal HRSG arrangement were obtained in order to minimize the cost of HRSG. Results show that the capital cost and the total heat transfer surface area of the HRSG were decreased by 24.3% and 22.3%, respectively. 4E analysis and optimization of double pressure water tube HRSG using genetic algorithm were accomplished by Mokhtari et al.\textsuperscript{19} Optimization results showed that the exergy and thermal efficiencies increased from 42% and 47.6% to 47.28% and 48.94% respectively and the optimum pinch and approach temperatures were 35°C and 20°C respectively. Najjar et al\textsuperscript{20} analyzed a water tube HRSG in a CCPP at full and partial load operations. The results showed that the degradation rate increases with time and with the operational load. They also found that the maximum degradation of the HRSG components was in the low-pressure stage of the HRSG at the full load and reached −6.73%, while the maximum overall degradation of the HRSG reached −6.02% at the full load.

1.2 Application of HRSG in CHP systems

Since the prime movers of CHP systems are often microgas turbines and gas or diesel engines (with power output of up to 25 MW), the temperature and mass flow rate of exhaust gas is not as much as those in CCPP and the waste heat can be recovered in one level. Therefore, a one-pressure level fire tube HRSG is frequently used in this application. Despite the wide application of fire tube HRSGs in CHP systems, few researches are available in literature in which fire tube HRSGs are optimized or analyzed.

Behbahani-nia et al\textsuperscript{21} optimized a small cogeneration system including a gas microturbine and a fire tube HRSG using thermodynamic, thermo-economic, and multiobjective optimization. The HRSG system was optimized considering the sum of the exergy losses and the exergy destruction and the sum of annualized values of the capital cost as objective functions. The results showed that the thermodynamic optimization does not lead to major improvement of the total cost of the HRSG although the thermo-economic and multiobjective methods improve the total cost of the system due to decrease in the pinch point.

Gutiérrez\textsuperscript{22} developed a dynamic model for analyzing the performance of the fire tube HRSG Using MATLAB. The mathematical model developed is based on the first principles of mass, energy, and momentum conservations. In the
model, the two parts of the boiler (fire/gas and water/steam sides), the economizer, the superheater, and the heat recovery are considered. The model developed can capture the dynamics of the boiler level and boiler pressure with confidence.

1.3 Application of HRSG in novel cycles

Recently, HRSG is utilized in novel cycles or in a novel configuration which from those important ones the following can be referred. Mahdavi and Khalilarya\textsuperscript{23} investigated a cogeneration system including gas turbine-HRSG-ORC in solar applications thermodynamically. Energy and exergy analysis of an innovative combined gas and steam cycle including a heat exchanger for preheating air in air bottoming cycle and a HRSG for generating steam are conducted by Khan and Tliti.\textsuperscript{24} Ghaebi and Ahmadi\textsuperscript{25} evaluated a new hybrid tri-generation system including Solid Oxide Fuel Cell (SOFC), HRSG and HDH desalination unit from energy and exergy point of view. Exergy and economic optimization of a novel U-typed once-through HRSG are carried out by Li et al.\textsuperscript{26} The results showed that the highest exergy efficiency reached to 54.61\% and the annual income and payback period are 121.242 Euro and 1.1 years respectively.

It is worthy to notify that in most of researches steady state operation is modeled. X-steam function or REFPROP library is used to calculate the pressure/temperature-dependent thermophysical properties of water and steam during the modeling and optimization procedures.

It should be noted that because of various geometrical and thermodynamic design parameters and their wide range of variations, numerous cases will be generated for optimization process which is not possible to be conducted by numerical simulation and since there is no commercial software for optimization of HRSGs, it is just possible to do the optimization process using thermodynamic modeling and genetic algorithm similar to what is accomplished in the above references.

According to researches addressed above, the optimization and comparison of finless and finned tube HRSGs in gas microturbine cogeneration plants were not considered.

According to the available literatures, HRSG modeling of a cogeneration plant are performed with different approaches including energy modeling,\textsuperscript{27-29} energy and exergy modeling,\textsuperscript{30,31} exergo-economic modeling,\textsuperscript{8,10,18,21} energy-economic modeling,\textsuperscript{32,33} and the major prime movers used in cogeneration or CHP systems such as gas turbine\textsuperscript{8,10,34,35} or diesel engine.\textsuperscript{32,36-42} Accordingly, the comprehensive modeling and optimization of fire tube HRSG in energy, exergy, and economic aspects for a gas microturbine cogeneration plant were not found in open literature. Although the design procedure for water and fire tube HRSGs are almost similar and the only difference returns to different heat transfer coefficients, but the main focus of research activities is on water tube HRSGs for their wide usage in CCPP. Furthermore, the comparison of techno-economic optimization results for finless and finned tube HRSGs considering geometrical parameters and types of fin as design parameters has not been considered, yet. Moreover, techno-economic assessment is required to investigate whether the heat transfer rate increment (due to adding fins) will compensate the investment and maintenance cost.

In this study, a fire tube HRSG comprises economizer, evaporator and super-heater is analyzed considering the tube arrangement and fin type, simultaneously. The investigation consists of energy, exergy, and economic (3E analysis) analysis of the system. Furthermore, two objectives optimization of the fire tube HRSG including the minimizing of the total annual costs (investment, maintenance, and exergy destruction rate costs) and the maximizing of the exergy efficiency. The optimization is done to find the geometric and thermodynamic design variables of HRSG. Last but not least, this analysis is performed on a cogeneration system including a 600 kW microgas turbine to show its effectiveness.

2 PHYSICAL DESCRIPTION AND CHARACTERISTICS OF HRSG

Schematic of a fire tube HRSG is shown in Figure 1. Water enters the economizer at a certain temperature. Then moves into the evaporator (in where the water enters the shell and the gas passes through the pipes) at a higher temperature and saturated steam is produced by heat transfer between gas and water. Finally, steam goes through the super heater and superheat steam with higher energy content is produced.

Tube banks in HRSG equipment have two common arrangements including inline and staggered as are shown in Figure 2. Each arrangement has different characteristics from heat transfer and fluid flow point of view. Furthermore, to increase the heat transfer rate, the steam production and the energy recovery, the external surfaces or fins are used which are in two common types of solid and serrated as is demonstrated in Figure 3.\textsuperscript{43}
Two important parameters of HRSG which have significant effect on the performance are the pinch point and the approach point. The pinch and approach points are shown in Figure 4. Pinch point is the difference between the gas temperature leaving the evaporator and saturation temperature and the approach point is the difference between saturation temperature and water temperature entering the evaporator. Although Decrement of these two parameters diminishes energy destruction due to heat transfer process, but pinch and approach points cannot be arbitrarily selected. A low pinch or approach point naturally increases the surface area required for the evaporator and economizer and thus the HRSG cost while a large value for these variables decreases the amount of energy recovered. Therefore, the most appropriate value for pinch and approach points should be obtained by optimization.
In this study, fire tube HRSG is optimized considering tube arrangement, fin type, and pinch and approach points as design parameters.

3 | ENERGY ANALYSIS

3.1 | HRSG modeling

In this section, thermal modeling of HRSG components (economizer, evaporator, and super-heater) is presented in order to obtain the inlet/outlet temperature of gas and water/steam, steam generation mass flow rate, and the heat transfer surface areas.

The following assumptions are considered in modeling section:

1. Steady state operation is modeled.
2. The radiation heat transfer is neglected due to the heat source temperature lower than 600°C.
3. The superheater outlet gas temperature is higher than its dew point temperature. (To prevent condensation of water content in exhaust gas.)
4. The exhaust gas is assumed to be an ideal gas.
5. The momentum equation for heat exchangers is ignored since the heat transfer process is considered isobaric.
6. Heat exchangers are assumed to be isolated and therefore there is no heat loss from heat exchangers to the environment.

For thermal modeling of HRSG components (economizer, superheater, evaporator), first law of thermodynamics is applied considering a control volume around each component and using $\varepsilon$-NTU/logarithmic mean temperature difference (LMTD) methods. Relations for modeling of HRSG components are presented in Table 1.

Similar to relations used for superheater, Equations (2) and (3) in Table 1 are also used for economizer. Unlike superheater and economizer in which $\varepsilon$-NTU method is used, in evaporator, the energy equation is solved just for the gas flow (since the saturation temperature of steam is known by having steam pressure). So for reducing the computation time and simplicity of the model, the LMTD method is used for evaporator design. $U$ in Table 1 is the overall heat transfer coefficient for finless tubes in which the inner and outer convective heat transfer coefficients are available in References 44–47. The overall heat transfer coefficient for finned tubes is presented in Appendix A.

The overall heat transfer rate in HRSG is obtained using the following equation:

$$Q_{Total} = Q_{sh} + Q_{Eva} + Q_{Eco}$$ (11)

| Equipment type | Energy balance | Description |
|----------------|----------------|-------------|
| **Superheater** | \[Q_{sh} = \dot{m}_{g} C_{p}\left(T_{\text{in},sh} - T_{\text{out},sh}\right) = \dot{m}_{s}(h_{\text{in},sh} - h_{\text{out},sh})\] | First Law of Thermodynamics |
| | \[\varepsilon = C_{\text{Min}}(T_{\text{in},sh} - T_{\text{sat}})\] | (1) Heat transfer units |
| | \[C_{\text{Min}} = \begin{cases} C_1 = \dot{m}_{g} C_{p}\left(\text{if } C_1 < C_2 \right) \\ C_2 = \dot{m}_{s} C_{p}\left(\text{if } C_2 \leq C_1 \right) \end{cases}\] | (4) Minimum heat capacity |
| | \[\frac{1}{U} = \frac{d}{h_{\text{di}}} + \frac{1}{h_{\text{io}}} + \frac{d}{\Delta h \ln \left[\frac{T_{\text{in},sh} - T_{\text{sat}}}{T_{\text{in},sh} - T_{\text{out},sh}}\right]} \times \ln \frac{d}{a} + \frac{f_{\text{di}}}{a} + \frac{f_{\text{io}}}{a}\] | (5) Total heat transfer coefficient |
| **Evaporator** | \[Q_{Eva} = \dot{m}_{g} C_{p}\left(T_{\text{out},eva} - T_{\text{in},eva}\right) = FAU\Delta T_{\text{lm}}\] | Using LMTD method |
| | \[\Delta T_{\text{lm}} = \frac{T_{\text{out},eva} - T_{\text{in},eva}}{\ln \left(\frac{T_{\text{out},eva} - T_{\text{sat}}}{T_{\text{in},eva} - T_{\text{sat}}}\right)}\] | (7) logabratic mean temperature difference |
| | \[T_{\text{in},eva} = T_{\text{sat}} + \left[\frac{T_{\text{out},eva} - T_{\text{sat}}}{}\times \exp \left( - \frac{U\times d}{\dot{m}_{g} C_{p}} \right) \right]\] | (8) inserting Equation (7) in (6) leads to Equation (8) |
| **Economizer** | \[Q_{Eco} = \dot{m}_{g} C_{p}\left(T_{\text{out},eco} - T_{\text{in},eco}\right) = \dot{m}_{s}(h_{\text{in},eco} - h_{\text{out},eco}) = \varepsilon C_{\text{Min}}(T_{\text{in},eco} - T_{\text{out},eco})\] | Heat transfer in economizer using $\varepsilon$-NTU method |
| | \[C_{1} = \dot{m}_{g} C_{p}\] |
| | \[C_{2} = \dot{m}_{s} C_{p}\] |
| | \[C_{\text{Min}} = \begin{cases} C_1(\text{if } C_1 < C_2) \\ C_2(\text{if } C_2 \leq C_1) \end{cases}\] | (10) Minimum heat capacity |
4 | EXERGY ANALYSIS

Exergy is the maximum attainable useful work of any kind of energy. The irreversibility (exergy destruction rate) in a system can be evaluated using exergy analysis. Total exergy destruction rate of HRSG is sum of the equipment exergy destruction rate due to heat transfer and friction losses and the exergy losses due to discharge of hot exhaust gasses to the environment.

Exergy losses rate is calculated by Equation (12):

\[ E_L = \dot{m}_g [(h_{g_{out,env}} - h_{g_{in,env}} - T_{atm}(s_{g_{out,env}} - s_{g_{in,env}})] \]

Assuming exhaust gas as an ideal gas, Equation (12) can be rewritten:

\[ \dot{E}_L = \dot{m}_g \left[ C_p (T_{g_{out,env}} - T_{atm}) - T_{atm} \left( C_p \ln \left( \frac{T_{g_{out,env}}}{T_{atm}} \right) \right) \right] \]

The exergy destruction rate is defined as the difference between inlet and outlet exergy rate for each component:

\[ E_D = \dot{E}_{in} - \dot{E}_{out} \]

where \( \dot{E}_{in} \) and \( \dot{E}_{out} \) are inlet and outlet exergy rate which are calculated through Equations (15)-(20) for superheater, evaporator, and economizer respectively:

\[ \dot{E}_{in,sh} = \dot{m}_g [(h_{g_{in,sh}} - h_{g_{out,sh}}) - T_{atm}(s_{g_{in,sh}} - s_{g_{out,sh}})] \]
\[ \dot{E}_{out,sh} = \dot{m}_{wa} [(h_{s_{out,sh}} - h_{s_{in,sh}}) - T_{atm}(s_{s_{out,sh}} - s_{s_{in,sh}})] \]
\[ \dot{E}_{in,eva} = \dot{m}_g [(h_{g_{in,eva}} - h_{g_{out,eva}}) - T_{atm}(s_{g_{in,eva}} - s_{g_{out,eva}})] \]
\[ \dot{E}_{out,eva} = \dot{m}_{wa} [(h_{s_{in,eva}} - h_{s_{out,eva}}) - T_{atm}(s_{s_{in,eva}} - s_{s_{out,eva}})] \]
\[ \dot{E}_{in,eco} = \dot{m}_g [(h_{g_{in,eco}} - h_{g_{out,eco}}) - T_{atm}(s_{g_{in,eco}} - s_{g_{out,eco}})] \]
\[ \dot{E}_{out,eco} = \dot{m}_{wa} [(h_{s_{out,eco}} - h_{s_{in,eco}}) - T_{atm}(s_{s_{out,eco}} - s_{s_{in,eco}})] \]

Finally, the total exergy destruction rate is calculated using Equation (21):

\[ \dot{E}_{Des} = (\dot{E}_{in,sh} + \dot{E}_{in,eva} + \dot{E}_{in,eco} + \dot{E}_L) - (\dot{E}_{out,sh} + \dot{E}_{out,eva} + \dot{E}_{out,eco}) \]

5 | ECONOMIC ANALYSIS

The total annual cost of HRSG is sum of the annual equipment investment and maintenance costs and the exergy destruction rate cost:

\[ C_{\text{Total}} = C_{\text{Capital-Cost}} + C_{\text{Exergy-Destruction-Rate}} \]

Annual investment cost of HRSG equipment directly depends on the heat transfer area which is calculated through Equations (23)-(28) for superheater, evaporator, and economizer respectively:

\[ C_{\text{Capital-sh}} = (10^{k_{1sh} + k_{2sh} \log_1(A_{sh}) + k_{3sh} (\log_2 A_{sh})^2)} \]

\[ \begin{align*}
  k_{1sh} &= 4.1884 \\
  k_{2sh} &= -0.2508 \\
  k_{3sh} &= 0.1974
\end{align*} \]
\[
C_{\text{Capital-eva}} = (10^{k_1_{eva}} + k_2_{eva} \log_{10}(A_{eva}) + k_3_{eva}(\log_{10}A_{eva})^2)
\]
(25)

\[
\begin{align*}
k_{eva} &= 5.0238 \\
k_{2_{eva}} &= 0.3475 \\
k_{3_{eva}} &= 0.0703
\end{align*}
\]
(26)

\[
C_{\text{Capital-eco}} = (10^{k_1_{eco}} + k_2_{eco} \log_{10}(A_{eco}) + k_3_{eco}(\log_{10}A_{eco})^2)
\]
(27)

\[
\begin{align*}
k_{1_{eco}} &= 2.76 \\
k_{2_{eco}} &= 0.7282 \\
k_{3_{eco}} &= 0.0783
\end{align*}
\]
(28)

Finally, the total annual investment cost is obtained using Equation (29):

\[
C_{\text{Capital-Cost}} = CRF \times (C_{\text{Capital-sh}} + C_{\text{Capital-eva}} + C_{\text{Capital-eco}})
\]
(29)

where CRF is the capital recovery factor and is calculated by Equation (30):

\[
CRF = \left( \frac{i(1+i)^n}{(1+i)^n - 1} \right)
\]
(30)

where \(i\) is the interest rate and \(n\) is the equipment useful lifetime.

The capital cost increment through the time is calculated by Equation (31):

\[
C_2 = C_1 \left( \frac{I_2}{I_1} \right)
\]
(31)

where \(C\) is capital cost and \(I\) is cost index, 1 refers to 2006 and 2 refers to 2017. The cost index ratio \(\left( \frac{I_2}{I_1} \right)\) for computing the equipment cost in 2017 is 1.32.

On the other hand, the exergy destruction rate is part of the energy content which is not used in heat recovery process. The equivalent cost of this energy loss is calculated by the following equation:

\[
C_{\text{Exergy-Destruction-Rate}} = C_E \times \dot{E}_{\text{Des}} \times H
\]
(32)

where \(H\) is the annual working hours of HRSG and \(C_E\) is the exergy unit price in $/kWh.

6 | OPTIMIZATION METHODOLOGY

In this article, optimization is performed using multiobjective genetic algorithm considering two objective functions including the total annual cost of HRSG (Equation (22), which is minimized) and the exergy efficiency of HRSG (Equation (33), which is maximized).

\[
\eta_{ex} = \frac{\dot{E}_{\text{out,Sh}} - \dot{E}_{\text{out,Eco}}}{\dot{E}_{\text{in,Sh}} - \dot{E}_{\text{in,Eco}}}
\]
(33)

Design parameters (decision variables) for finless tube HRSG cases are the pinch point temperature \((\Delta T_{\text{Pinch}})\), the approach point temperature \((\Delta T_{\text{Approach}})\), the working pressure \((P_{\text{working}})\), and the steam mass flow rate \((m_{\text{st}})\) and for finned tube HRSG cases are the four above parameters as well as the fin height \((h_f)\), the fin thickness \((t_f)\), and the number of fins per meter \((n_f)\). Dependent parameters such as the gas/steam temperatures and the heat transfer areas of equipment will be calculated using the design parameters. List of the design parameters, their range of variation, and the system’s constrains are presented in Table 2. The modeling and optimization procedure flowchart is shown in Figure 5.
**TABLE 2** The list of design parameters, their range of variation and the system’s constrains.\(^{21,44,50}\)

| Parameter                                | Variation range |
|------------------------------------------|-----------------|
| \(\Delta T_{\text{Pinch}}\)             | 3-30°C          |
| \(\Delta T_{\text{Approach}}\)          | 10-40°C         |
| Working pressure \((p_{\text{working}})\) | 2–8 bar         |
| \(m_{\text{g}}\)                        | 0.5-1.36 kg/s   |
| \(h_f\)                                 | 0.005-0.03 m    |
| \(t_f\)                                 | 0.0005-0.003 m  |
| \(n_f\)                                 | 78-312 perm     |

**Constrain** \(T_{\text{Eout}} > T_{\text{dew point}}\)  
Reason: Preventing from condensation of water content in exhaust gas

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**Energy analysis**
Thermodynamic properties of water/steam in HRSG (\(X_{\text{steam}}\) function), steam mass flowrate, exhaust gas and water/steam temperatures in economizer, evaporator and superheater, heat transfer surface and heat transfer rate in economizer, evaporator and superheater and heat transfer coefficient in finless and finned tube heat exchangers (Eq. 1 to 10 in table 1 and Equations in appendix 1).

**Exergy analysis**
Exergy losses in HRSG, exergy destruction rate in economizer, evaporator and superheater, total exergy destruction rate and exergy efficiency of HRSG (Equations 2 to 1 and Eq. 23).

**Economic analysis**
Total annual investment cost of HRSG, annual maintenance cost of HRSG, cost index for considering the capital cost increment through the time and exergy destruction rate cost (Equations 12 to 22).

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**FIGURE 5** The procedure of modeling and optimizing fire tube HRSG by two-objective genetic algorithm optimization
6.1 The genetic algorithm method for multiobjective optimization

The evolutionary cycle in genetic algorithm includes the following steps:

1. Generating the initial population.
2. Selecting the parents and applying cross over for generating the offspring population.
3. Selecting the population members for mutation and generating the mutant population.
4. Merging the initial, offspring and mutant population for generating the new population.
5. Repeating steps 2-4 until the termination condition is achieved.

The above steps are shown in Figure 6.

The optimization problems are divided into two groups of single objective and multiobjective. In single objective optimization problems, optimizing the objective function with satisfaction of equal and unequal constrains which is defined from a domain (design parameters) to the real number set which is regular is conducted. In single objective optimization problems, there is only one optimum answer. The mathematical expression of the above concept is:

$$\text{Min or Max} f(x) \Rightarrow f : X \rightarrow IR$$ (34)

$$g_j(x) \geq 0 \quad j = 1, 2, \ldots, m$$ (35)

$$n_k(x) = 0 \quad k = 1, 2, \ldots, n$$ (36)

where $f(x)$ is the objective function, $X$ is the design parameters vector, $g_j(x)$ is the unequal constrain and $n_k(x)$ is the equal constrain.

In multiobjective optimization problems, objective functions are optimized with satisfaction of equal and unequal constrains which is defined from a domain (design parameters) to the real number set which is irregular, that is, in mathematical expression:

$$\text{Min or Max} f_i(x) \Rightarrow f : X \rightarrow IR^m$$ (37)

$$g_j(x) \geq 0 \quad j = 1, 2, \ldots, m$$ (38)

$$n_k(x) = 0 \quad k = 1, 2, \ldots, n$$ (39)

To clarify the concept of irregular set of answers, suppose that $x_1$, $x_2$, and $x_3$ are the three possible answers from the real number set in two-dimensional objective space (Figure 7). It is possible to compare $x_1$ and $x_2$, but it is impossible to compare $x_1$ and $x_3$ or $x_2$ and $x_3$. In other word, $x_3$ is lower and higher than $x_1$ and $x_2$ with respect to objective functions ($f_1$) and ($f_2$) respectively. So, in multiobjective optimization problems, there is a set of irregular answers which should
be sorted. In this article, the Nondominated Sorting Genetic Algorithm II (NSGA II) is used for sorting the population in order to find the optimum answer.

To explain the NSGA II method, it is necessary to clear the concept of dominance. According to Figure 8, $x$ is dominated by members of $C$ and members of $A$ are dominated by $x$, but there is no definite statement about $B$ and $D$. Mathematically:

$$\forall a \in A \rightarrow a \succ x$$  \hspace{1cm} (40)

$$\forall c \in C \rightarrow c \prec x$$  \hspace{1cm} (41)

The NSGA II method sorts the answers in a way that the objective functions are not dominated by other answers.\textsuperscript{51} These answers are called Pareto front. Pareto front is the locus of optimal non-dominated answers which is not dominated by any other answer. All points on the Pareto front are acceptable as solution for multiobjective optimization problems. But eventually for practical reasons, one point should be chosen by the designer as the optimum solution.

Since the dimension of objective functions in this article are different (the exergy efficiency is nondimensional and the total annual cost is in $/\text{year}$), the objective functions should be normalized first and then the optimum point should be selected using TOPSIS method. In TOPSIS method, an ideal point (where the functions have their best values) and a nonideal point (where the functions have their worst values) are considered. The optimum point should have the least and the most distance from the ideal and nonideal points respectively.\textsuperscript{52}

7 | CASE STUDY

In the present study, a 600 kW gas microturbine is considered as prime mover of cogeneration plant. Technical specification of the gas microturbine is presented in Table 3. Benefits of using gas microturbine as prime mover are high efficiency
TABLE 3  Technical specification of 600 kW gas microturbine exhaust gas\textsuperscript{21}

| Load\% | $P$ (kW) | $T_{g1}$ ($^\circ$C) | $m_g$ (kg/h) |
|--------|-----------|----------------------|-------------|
| 100    | 600       | 570                  | 18 000      |

FIGURE 9  Schematic for a cogeneration plant including a gas microturbine, a HRSG and a stack

TABLE 4  List of input parameters (constant parameters) for optimization\textsuperscript{21,49,50}

| Parameter                     | Numerical value |
|-------------------------------|-----------------|
| $T_{g1}$ ($^\circ$C)         | 570             |
| $T_{wa,ent}$ ($^\circ$C)      | 20              |
| $T_{dew\ point}$ ($^\circ$C) | 105             |
| $d$ (cm)                     | 5.08            |
| $m_g$ ($\frac{kg}{hr}$)      | 18 000          |
| $H$ ($\frac{hr}{year}$)      | 7000            |
| CRF                           | 0.2385          |
| $C_E$ ($$/kWh$$)             | 0.004           |

at full and partial load operation, low investment cost, short start up and shut down, and low water usage.\textsuperscript{50} The schematic of a cogeneration plant including a gas microturbine, a HRSG and a stack is shown in Figure 9. The microturbine exhaust gas is recovered in HRSG and the steam is generated. The purpose is to find the optimum values of design parameters and objective functions for optimum design of HRSG. Therefore, modeling and optimization of other components such as gas microturbine and stack are not included in this study. Input parameters which are constant during optimization process are presented in Table 4.

8  | RESULTS AND DISCUSSION

8.1  | Model verification

Main parameters of HRSG equipment (economizer, evaporator, superheater) including the steam mass flow rate ($m_{st}$), the heat transfer area ($A$), and the total heat transfer rate ($Q$) are obtained from present modeling and compared with FIRECAD\textsuperscript{53} commercial software results. Modeling and FIRECAD results and their difference are presented in Table 5. This table shows a good agreement between modeling and FIRECAD results with maximum relative error of about 3%.

8.2  | Optimization results

HRSG optimization is accomplished by developing a code in MATLAB environment which takes about 1 hour to complete the running process for each case separately. REFPROP is used to read the thermodynamic properties of water/steam during computation process.
TABLE 5 Comparison of present modeling and FIRECAD results for HRSG equipment

| Items          | FIRECAD results | Present modeling results | Maximum relative error |
|----------------|-----------------|--------------------------|------------------------|
|                | $Q_{\text{total}}$ (kJ/h) $\times 10^6$ | $m_{\text{wa}}$ (kg/h) | $A$ (m$^2$) | $Q_{\text{total}}$ (kJ/h) $\times 10^6$ | $m_{\text{wa}}$ (kg/h) | $A$ (m$^2$) |                  |
| Economizer     | 6.56            | 130 926.4               | 98.26                 | 6.66         | 134 235.4           | 99.12                 | 2.52%            |
| Evaporator     | 22.53           | 8518.51                 | 869.3                 | 22.20        | 8473.3             | 845.9                 | 2.68%            |
| Superheater    | 12.39           | 29 483.5                | 1989                  | 12.20        | 29 558.8           | 1966                  | 1.54%            |

NSGA II is used in order to minimize the total annual cost and maximize the exergy efficiency. Six cases including finless and finned tube HRSG (solid and serrated) with inline and staggered arrangement are optimized and compared. Four design parameters (including pinch point temperature, approach point temperature, working pressure, and steam mass flow rate) are considered for finless tube HRSG and seven design parameters (the four above parameters as well as the fin height ($h_f$), the fin thickness ($t_f$), and the number of fins per meter ($n_f$)) are considered for finned tube HRSG. Genetic algorithm tuning parameters are set at generation population of 40 for finless tube HRSG and 70 for finned tube HRSG, crossover probability of 0.7, mutation probability of 0.4, and mutation rate of 0.02.

8.3 Finless tube HRSG cases

The nondimensional Pareto front curves of finless tube HRSG with an inline and a staggered arrangement are shown in Figures 10 and 11 respectively. Ideal and nonideal points are determined in the graphs. In ideal point, the total annual cost

![Non dimensional Pareto front for total annual cost-exergy efficiency optimization of finless tube HRSG (inline arrangement)](image10)

![Non dimensional Pareto front for total annual cost-exergy efficiency optimization of finless tubes HRSG (staggered arrangement)](image11)
has its lowest value and the exergy efficiency has its highest value while in the nonideal point the total annual cost has its highest value and the exergy efficiency has its lowest value. The optimum point is selected using TOPSIS method based on the least and most distance from the ideal and nonideal points respectively. The optimum values of design parameters, dependent parameters, objective functions, and exergy destruction rates are presented in Tables 6-8 respectively.

As it can be observed in Table 6, finless tube HRSG with an inline arrangement has lower pinch and approach point (3.5°C and 3.1°C) in comparison with a staggered arrangement due to the improved tube bundle arrangement which leads to better use of exhaust gas energy content and consequently more heat transfer rate. In addition, finless tube HRSG with an inline arrangement has higher working pressure (14%) due to lower pinch point and higher saturation temperature

| TABLE 6 | The optimum values of design parameters for multiobjective optimization of finless tube HRSG (inline and staggered arrangement) |
|------------------|------------------|
| **Arrangement** | **Design parameters** |
|                 | Inline | Staggered |
| ΔT_Pinch (°C)   | 17.5   | 21.01     |
| ΔT_Approach (°C)| 21.2   | 24.3      |
| P_Working (bar) | 3.1    | 2.7       |
| m_d (kg/h)      | 2232   | 1980      |

| TABLE 7 | The optimum values of independent parameters (temperatures and heat transfer areas) and objective functions for multiobjective optimization of finless tube HRSG (inline and staggered arrangement) |
|------------------|------------------|
| **Arrangement** | **Dependent Para.** |
|                 | Inline | Staggered |
| A_S, Sh (m²)    | 15.81  | 16.25     |
| Tube number Sb (number) | 100 | 102 |
| A_Eva (m²)      | 35.8   | 38.3      |
| Tube number Eva (number) | 225 | 241 |
| A_Eco (m²)      | 22.36  | 23.42     |
| Tube number Eco (number) | 141 | 147 |
| T_out, Sh (°C)  | 110.08 | 112.33    |
| T_out, Eva (°C) | 152.14 | 150.97    |
| T_out, Eco (°C) | 435.22 | 433.66    |
| T_sat (°C)      | 113.44 | 108.66    |
| C_Total (€/year)| 64.096 | 65.286    |
| η_EX            | 60.6%  | 58%       |

| TABLE 8 | The optimum values of exergy destruction rate in equipment for multiobjective optimization of finless tube HRSG (inline and staggered arrangement) |
|------------------|------------------|
| **Arrangement** | **Exergy rate** |
|                 | Inline | Staggered |
| E_L (kW)        | 78.81  | 80.53     |
| E_{F, Sb} (kW)  | 150.03 | 152.10    |
| E_{F, Eva} (kW) | 558.12 | 570.35    |
| E_{F, Eco} (kW) | 76.01  | 81.20     |
| E_{F, Sb} (kW)  | 118.20 | 113.14    |
| E_{F, Eva} (kW) | 220.87 | 213.43    |
| E_{F, Eco} (kW) | 73.31  | 72.01     |
| E_{Destruction} (kW) | 450.59 | 485.6    |
and steam mass flow rate (12.7%) due to higher working pressure and heat transfer rate. Therefore, the HRSG with an inline arrangement has better performance than that of the HRSG with a staggered arrangement because of the lower pinch and approach point, the higher steam mass flow rate, and consequentially higher power output.

As can be observed in Table 7, the heat transfer surface area of HRSG with an inline arrangement is lower (5.4%) than that of HRSG with a staggered arrangement due to lower pinch point and higher heat transfer rate. The equipment gas and water/steam output temperatures are approximately the same for both inline and staggered arrangements.

Total annual cost of HRSG with an inline arrangement is (1.98%) lower than that of HRSG with a staggered arrangement because of the lower heat transfer surface area.

Exergy efficiency of HRSG with an inline and a staggered arrangement are 60.6% and 58% respectively. The higher exergy efficiency is due to the higher heat transfer rate and lower exergy destruction rate in HRSG equipment because of lower heat loss (of exhaust gas energy content) to the environment. Therefore, HRSG with a staggered arrangement has better performance from the second law of thermodynamics point of view (2.6% higher exergy efficiency).

As it is observed in Table 8, the most exergy destruction rate occurs in the evaporator (337.25 kW) because of the fluid change of phase, steam enthalpy increment, and a significant drop in the evaporator gas temperature due to the more energy recovery.

Total exergy destruction rate of HRSG with an inline arrangement (450.59 kW) is 7% lower than of HRSG with a staggered arrangement (485.6 kW) because of the higher working pressure and the lower heat transfer surface area.

Therefore, finless tube HRSG with an inline arrangement compared to the finless tube HRSG with a staggered arrangement has lower total annual cost and heat transfer surface area by 1.98% and 5.4%, higher steam mass flow rate and exergy efficiency by 12.7%, and 2.6% and lower pinch and approach points which means for better thermodynamic performance.

8.4 | Finned tube HRSG cases

In this section, four cases are optimized which are as follows:

1. Finned tube (solid type) HRSG with an inline arrangement (SOI).
2. Finned tube (solid type) HRSG with a staggered arrangement (SOS).
3. Finned tube (serrated type) HRSG with an inline arrangement (SEI).
4. Finned tube (serrated type) HRSG with a staggered arrangement (SES).

The optimum values of design parameters, dependent parameters, objective functions, and equipment exergy destruction rates for the above four cases are presented in Tables 9-11. Looking at Tables 9-11, it is perceived that the finned tube HRSG (solid type) with a staggered arrangement has the best performance from techno-economic point of view regarding the lower heat transfer surface area and fin number (3.7%, 7.1%, and 4.1% lower than SOI, SEI, and SES respectively) because of higher heat transfer coefficient and consequently higher heat transfer rate. Lower total annual cost (2.9%, 8.8%, and 5.8% lower than SOI, SEI, and SES respectively) is due to the lower heat transfer surface area and exergy destruction rate cost. Higher steam mass flow rate (8.3%, 28.7%, and 17.6% higher than SOI, SEI, and SES respectively) is because of higher working pressure, lower pinch and approach points, and higher heat transfer rate. Higher exergy efficiency (2.8%, 13.6%, and 7.9% higher than SOI, SEI, and SES respectively) is due to the lower exergy destruction rate because of higher heat transfer rate and lower heat loss to the environment. It is also perceived that the exergy efficiency obtained by our optimization (73.12%) is higher than those of Sharma and Singh (55% at maximum)\(^1\) and Behbahani-nia et al\(^2\) (60% at maximum) in which fin parameters were considered in the design process. This is due to the finned tube HRSG (solid type) with an inline arrangement, the finned tube HRSG (serrated type) with a staggered arrangement, and the finned tube HRSG (serrated type) with an inline arrangement are in second, third, and fourth places respectively.

8.5 | Comparison of performance for finned tube HRSG and finless tube HRSG

In this section, performance of the finned tube (solid type) HRSG with a staggered arrangement (best finned tube case) and the finless tube HRSG with an inline arrangement (best finless tube case) at their optimum point are compared. As it is observed in Table 9, the pinch and approach points are slightly increased from 17.5°C to 20.02°C and from 21.2°C to
TABLE 9  The optimum values of design parameters for multiobjective optimization of finned tube HRSG (inline and staggered arrangement)

| Fin type | Solid | Serrated |
|----------|-------|----------|
| Arrangement | Design parameters | Inline | Staggered | Inline | Staggered |
| ΔT.Pin (°C) | 21.5 | 20.02 | 23.6 | 22.55 |
| ΔT.Approach (°C) | 29.3 | 28.62 | 29.5 | 29.4 |
| P.Working (bar) | 6.1 | 6.5 | 5 | 5.55 |
| m₀ (kg/h) | 4320 | 4680 | 3636 | 3978 |
| h_fg (m) | 0.0069 | 0.007 | 0.0064 | 0.0067 |
| n_f (per m) | 112 | 96 | 130 | 120 |
| t_f (m) | 0.0022 | 0.002 | 0.0019 | 0.002 |

TABLE 10  The optimum values of independent parameters (temperatures and heat transfer areas) and objective functions for multiobjective optimization of finned tube HRSG (inline and staggered arrangement)

| Fin type | Solid | Serrated |
|----------|-------|----------|
| Arrangement | Dependent para. | Inline | Staggered | Inline | Staggered |
| A_sh (m²) | 30.44 | 27.34 | 35.44 | 32.94 |
| A_iva (m²) | 46.34 | 48.36 | 43.02 | 44.68 |
| A_ivo (m²) | 37.61 | 34.51 | 39.61 | 37.1 |
| T_Eva,eco (°C) | 136.88 | 141.92 | 130.99 | 133.93 |
| T_Eva,eva (°C) | 177.97 | 182.00 | 175.42 | 176.7 |
| T_Eva,sh (°C) | 470.41 | 478.12 | 465.78 | 468.1 |
| T_sat (°C) | 159.47 | 161.98 | 151.82 | 155.80 |
| C_Total ($/year) | 93 389 | 90 810 | 98 819 | 96 104 |
| η_EX | 71.2% | 73.2% | 64.4% | 67.8% |

28.62°C for finless and finned tube HRSG respectively. This is due to the large increment of working pressure as well as higher outlet gas temperature due to the heat transfer surface area augmentation (because of adding fins). The working pressure and steam mass flow rate has been increased by about 100% (6.5 bar and 4680 kg/h) due to the heat transfer surface area and heat transfer rate increment. Also, the optimum values of the fin height, fin thickness, and the fin number per meter are (0.007 m, 0.002 m, and 96) respectively.

Looking at Table 10, it is realized that the total heat transfer surface area of the finned tube HRSG is increased by 48% because of adding fins and increasing of working pressure and pinch and approach points. This results in total annual cost increment of 41.6%.

Gas and water/steam side temperatures of HRSG are increased (up to 35°C) due to the increase of working pressure (saturation temperature) and pinch and approach points.

It is also observed that the exergy efficiency is increased from 60.6% to 73.2% because of more energy recovery and higher steam mass flow rate due to the heat transfer surface area increment.

Exergy destruction rate in equipment are presented in Table 11. It is found that the total exergy destruction rate is decreased from 450.59 to 423.17 kW due to the higher outlet exhaust gas temperature and lower heat loss to the environment because of the heat transfer rate increment.

Therefore, it can be concluded that using solid fins in HRSG with a staggered arrangement causes the total annual cost increment up to 41.6%, however it also increases the steam mass flow rate and exergy efficiency up to 100% and 12.6% respectively. This means that the thermodynamic performance improvement will compensate the total annual cost increment and this type of fins can be utilized in practical applications.
TABLE 11 The optimum values of exergy destruction rate in HRSG equipment for multiobjective optimization of finned tube HRSG (inline and staggered arrangement)

| Fin type Arrangement | Solid Exergy des. | Serrated Exergy des. |
|----------------------|------------------|----------------------|
|                      | Inline | Staggered | Inline | Staggered |
| $E_L$ (kW)           | 105.8  | 113.4     | 95.66  | 100.73    |
| $E_F,Sh$ (kW)        | 180.36 | 194.05    | 175.36 | 177.86    |
| $E_F,Evo$ (kW)       | 595.13 | 580.32    | 609.40 | 602.26    |
| $E_F,Eco$ (kW)       | 80.41  | 93.23     | 71.28  | 75.84     |
| $E_P,Sh$ (kW)        | 150.08 | 155.44    | 144.34 | 147.21    |
| $E_P,Evo$ (kW)       | 294.77 | 320.76    | 290.41 | 292.59    |
| $E_P,Eco$ (kW)       | 78.85  | 81.63     | 77.25  | 78.05     |
| $E_{DeSTRUCTION}$ (kW) | 438   | 423.17    | 441.7  | 439.8     |

9 | CONCLUSION

A comprehensive modeling and optimization of fire tube HRSG (comprises economizer, evaporator, and super-heater) in energy, exergy, and economic aspects for a gas microturbine cogeneration plant were implemented in this article. Comparison of techno-economic optimization results for finless and finned tube HRSGs considering geometrical parameters and types of fin as design parameters was also conducted. In this regard, six cases including finless and finned tube (solid and serrated) HRSGs with inline and staggered arrangement were studied. Optimization of HRSG was carried out by choosing a group of two objective functions including total annual cost (which was minimized) and exergy efficiency (which was maximized). The results for a 600 kW gas microturbine cogeneration plant (as a case study) showed that using tube banks with a staggered arrangement along with solid fins in HRSG causes total annual cost increment up to 41.6%, however this also increases the steam mass flow rate and exergy efficiency up to 100% and 12.6% respectively. This means that the thermodynamic performance improvement will compensate the total annual cost increment.

The presented method can be used for optimization of HRSG’s utilized in CHP systems or CCPP with other prime movers such as gas turbines, gas engines, diesel engines, and fuel cells, so further studies can be conducted in order to compare the performance of the optimized HRSG with different prime movers. The performance of the optimized HRSG can also be investigated in partial load operation of prime movers. In addition, the performance comparison of optimized water tube and fire tube HRSGs with the same input condition can be performed.

PEER REVIEW INFORMATION

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CONFLICT OF INTEREST

The authors declare no potential conflict of interest.

NOMENCLATURE

- $A$: heat transfer area (m$^2$)
- $C_P$: specific heat capacity (kJ/kg K)
- $CRF$: capital recovery factor
- $C_{Total}$: total annual costs of HRSG ($/year)
- $d$: diameter of the tubes (cm)
- $E$: rate of exergy (kW)
- $F$: logarithmic mean temperature difference correction factor
- $ff_i$: internal fouling factor (m$^2$ K/W)
- $ff_o$: external fouling factor (m$^2$ K/W)
- $G$: mass velocity (kg/m$^2$ s)
- $H$: annual working hours of HRSG (h/year)
- $h$: enthalpy (kJ/kg)
$$h_i \quad \text{heat transfer coefficient inside tubes (W/m}^2\ \text{K})$$

$$h_o \quad \text{heat transfer coefficient outside tubes (W/m}^2\ \text{K})$$

$$k \quad \text{thermal conductivity (W/m K)}$$

$$L \quad \text{length of the tubes (m)}$$

$$\dot{m}_g \quad \text{gas mass flow rate (kg/s)}$$

$$\dot{m}_s \quad \text{steam flow rate (kg/s)}$$

$$Q \quad \text{heat transfer rate (kW)}$$

$$Re \quad \text{Reynolds number}$$

$$T \quad \text{temperature (°C)}$$

$$U \quad \text{overall heat transfer coefficient (W/m}^2\ \text{K)}$$

**GREEK SYMBOLS**

$$\eta \quad \text{efficiency}$$

**SUBSCRIPT**

$$\text{atm} \quad \text{atmosphere}$$

$$\text{CHP} \quad \text{combined heat and power}$$

$$\text{Eco} \quad \text{economizer}$$

$$\text{Eva} \quad \text{evaporator}$$

$$\text{in} \quad \text{inlet}$$

$$\text{g} \quad \text{gas}$$

$$\text{HRSG} \quad \text{heat recovery steam generator}$$

$$\text{LMTD} \quad \text{logarithmic mean temperature difference}$$

$$\text{NSGA} \quad \text{nondominated sorting genetic algorithm}$$

$$\text{NTU} \quad \text{number of transfer unit}$$

$$\text{O} \quad \text{outlet}$$

$$\text{sat} \quad \text{saturation phase}$$

$$\text{Sh} \quad \text{super heater}$$

$$\text{st} \quad \text{steam}$$

$$\text{wa} \quad \text{water}$$

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APPENDIX A: OVERALL HEAT TRANSFER COEFFICIENT FOR FINNED TUBES

The overall heat transfer coefficient for finned tubes is calculated using following Equation 44:

\[ U = 0.85 \times \eta \times h_c \]  
\[ (A1) \]

\( \eta \) in Equation (A1) is calculated by the following equation:

\[ \eta = 1 - (1 - E) \frac{A_f}{A_t} \]  
\[ (A2) \]

For solid fins:

\[ E = \frac{1}{\left\{ 1 + 0.002292 m^2 h^2 \left( \frac{d + 2h_f}{d} \right) \right\}} \]  
\[ (A3) \]

\[ m = \left( \frac{24h_f}{k_b} \right)^{0.5} \]  
\[ (A4) \]

\[ A_f = \pi n_f \times \frac{4dh_f + 4h_f^2 + 2t_f d + 4t_f h_f}{24} \]  
\[ (A5) \]

\[ A_t = A_f + \pi \frac{d(1 - n_f t_f)}{12} \]  
\[ (A6) \]

And for serrated fins:

\[ E = \frac{\tanh(mh)}{mh} \]  
\[ (A7) \]

\[ m = \left( \frac{24h_s(t_f + w_f) / k_{bw_s}}{k_{bw_s}} \right)^{0.5} \]  
\[ (A8) \]
where \( w_f \) is the fin width and is obtained from Equation (A9):

\[
s = \frac{1}{n_f} - t_f \tag{A9}
\]

\[
A_f = \pi d n_f \times \frac{2h(w_f s + t_f) + t_f w_f s}{12 w_f s} \tag{A10}
\]

\[
A_t = A_f + \pi d \left( 1 - \frac{n_f t_f s}{12} \right) \tag{A11}
\]

\( h_c \) is the convection heat transfer coefficient which is calculated by the following equation:

\[
h_c = C_1 C_2 C_3 \left( \frac{d + 2h_f}{d} \right)^{0.5} \times \left( \frac{t_g + 460}{t_f + 460} \right)^{0.25} \times G C_p \left( \frac{k}{\mu C_p} \right) \tag{A12}
\]

where \( t_g \) and \( t_f \) is the average gas temperature and the average fin temperature respectively, \( h_f \) is the fin height, \( k \) is the thermal conductivity of gas, \( \mu \) is the dynamic viscosity of gas, and \( G \) is the gas mass velocity and is computed through Equation (A13):

\[
G = \frac{\dot{m}_g}{[(S_f/12) - A_o] N_W L} \tag{A13}
\]

\[
A_o = \frac{d}{12} + \frac{n_f t_f h_f}{6} \tag{A14}
\]

where \( t_f \) and \( n_f \) are the fin thickness and the number of fins per meter respectively.

Constants in Equation (A2) are presented in Table A1:

| Fin Type     | Tube bundle arrangement | Constants                                                                 |
|--------------|------------------------|---------------------------------------------------------------------------|
| Solid        | Inline                 | \( C_1 = 0.053(1.45 - 2.9 S_f/\nu)^{-2.3} Re^{-0.21} \)                     |
|              |                        | \( C_2 = 0.2 + 0.65e^{(-0.25 h_f/\nu)} \)                                  |
|              |                        | \( C_3 = 1.1 - (0.75 - 1.5e^{-0.75 S_f/\nu})e^{-2 S_f/S_t} \)               |
|              | Staggered              | \( C_1 = 0.091 Re^{-0.25} \)                                              |
|              |                        | \( C_2 = 0.35 + 0.65e^{(-0.25 h_f/\nu)} \)                                |
|              |                        | \( C_3 = 0.7 + (0.7 - 0.8e^{-0.15 S_f/\nu})e^{-1 S_f/S_t} \)               |
|              | Serrated               | \( C_1 = 0.053(1.45 - 2.9 S_f/\nu)^{-2.3} Re^{-0.21} \)                     |
|              | Inline                 | \( C_2 = 0.25 + 0.6e^{(-0.2 S_f/\nu)} \)                                   |
|              |                        | \( C_3 = 1.1 - (0.75 - 1.5e^{-0.75 S_f/\nu})e^{-2 S_f/S_t} \)               |
|              | Staggered              | \( C_1 = 0.053(1.45 - 2.9 S_f/\nu)^{-2.3} Re^{-0.21} \)                     |
|              |                        | \( C_2 = 0.35 + 0.65e^{(-0.17 h_f/\nu)} \)                                |
|              |                        | \( C_3 = 0.7 + (0.7 - 0.8e^{-0.15 S_f/\nu})e^{-1 S_f/S_t} \)               |
|              | Staggered              | \( C_1 = 0.053(1.45 - 2.9 S_f/\nu)^{-2.3} Re^{-0.21} \)                     |
|              |                        | \( C_2 = 0.35 + 0.65e^{(-0.17 h_f/\nu)} \)                                |
|              |                        | \( C_3 = 0.7 + (0.7 - 0.8e^{-0.15 S_f/\nu})e^{-1 S_f/S_t} \)               |