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# Exergetic Optimization of Designing Parameters for Heat Recovery Steam Generators Through Direct Search Method

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**Abstract:** In this paper, first a sensitivity analysis of parameters affecting the performance of the steam generator is performed. This process includes the pinch temperature difference and the gas velocity (width of the HRSG) as the operating parameter and geometrical parameters (fin's thickness, fin's height, number of fins in each inch, tube transverse pitch) for each pressure parts is carried out, and then these parameters have been optimized. Geometry details of the fins were considered separately for each pressure part of the steam generator (economizer, evaporator and superheater). The considered objective function is the rate of irreversibility in the steam generator which constitutes exergy destruction rate and exergy loss rate. After the optimization of total irreversibility rate, thermal and exergetic efficiencies and components of irreversibility, including exergy destruction for each part and exergy loss are calculated and compared with the situation prior to optimization.

**Keywords:** Boiler, Difference, Exergy, HRSG, Optimization, Pinch Point

## 1. Introduction

Daily increase of the energy demand in the world has encouraged researchers and engineers to look for different methods for more heat recovery. Among these methods, heat recovery steam generators (HRSGs) are regarded as very effective ones. This heat recovery is carried out in a more effective way when design parameters are chosen in their optimum state. HRSGs have numerous applications such as cogeneration of hot steam utility, generation of the steam needed for injecting into the combustion chamber, and also a more common application, in combined cycle power plants. In combined cycle power plants, exhaust gas of the gas turbine enters a chamber named heat recovery steam generator and transfer its thermal energy to the water which is flowing in that chamber. In this way, the water flowing in the chamber is converted to steam and later is used in a steam turbine (Fig. 1). High efficiency, low energy loss and long life of these power plants are some of the factors distinguishing them from other facilities of electricity generation and in the recent years a large number of these power plants have been employed in different locations of the world [1]

HRSGs are one of the most important components of combined cycles, the design of which has a remarkable influence on the total efficiency of the power plant [1,2]. One of the important parameters in designing of heat recovery steam generators is the pinch temperature difference Fig. 2. Reducing the pinch temperature difference causes the gas flow pressure drop within the boiler to increase and consequently it increases the rate of irreversibility.

On the other hand, this reduction results in a reduction of temperature difference of gas and steam flow and irreversibility due to the temperatrue difference of gas and steam when heat transfer is reduced, which in turn causes reduction of exergy destruction. So it is clear that there is always an optimum pinch temperature difference for which the rate of total irreversibility is minimum. This fact is examined through the sensitivity analysis that follows.

Other important parameters in the design of HRSGs are geometrical parameters of HRSG as will be seen in the sensitivity analysis section. Increasing thickness, height of fins or number of fins per inch of fins, increases the pressure drop resulting from the increase of the velocity of the exhaust gases. On the other hand, as absence of fins decreases heating area per unit of tube length, it prolongs the path of the gas and thus increases frictional pressure drop. Considering the high

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influence of pressure drop in the rate of irreversibility, there needs to be an optimum amount for fin properties. Different objective functions including irreversibility, thermo economical function or weight may be used to optimize thermal systems. The thermoecomoic approach is the most complete method in which both kinds of irreversibility (exergy loss and exergy destruction) in terms of money and capital cost are considered. The method needs detailed information about payments for constructing a new system as a function of designing variables. This information is sometimes hard to be obtained and depends on many parameters such as where the system is to be constructed and used. Total irreversibility as an objective function is used by many authors for optimizing thermal systems [3, 4] Using this objective function means maximizing efficiency of the system.

In different references, different values have been proposed for the pinch temperature difference. Alessandro Franco and Alessandro Russo [5] and C. Casarosa et al. [6] performed thorough exergy and thermo economic analyses of the heat recovery steam generator and calculated the optimum pinch temperature difference. The pinch temperature difference obtained in these references through exergy analysis is zero degrees celsius. This value is in no way logical and the reason for such a result is neglecting the effects of pressure drop in steam generator's boiler. In reference [7] the effect of the pinch point on the first and second law efficiency of HRSG is studied.

Alessandro Franco and Nicola Giannini [8,9] optimized design parameters of an HRSG in two steps. The first step can be aimed to the minimization of the pressure drop for a given heat flow. The second step leads to a reduction of the overall dimensions, maintaining the imposed performance of the HRSG in terms of heat flow and pressure drop. In their method pinch temperature is not optimized. In reference [4] exergy was used as the objective function. The paper mostly concerns on the arrangement of a combined cycle i.e; the number of pressure levels and reheats In this paper, a methodology based on using irreversibility as the objective function is suggested for optimizing heat recovery steam generators which, as stated before, has already been used in reference [5,6]. The optimization of this work provides the following advantages in comparison to these references:

1- In references [5,6] arrangement of heat transfer surfaces are not specified. In their work, a typical heat transfer coefficient is considered for economizer, evaporator and superheater. In the present work a more accurate analysis is used considering radiation and convective heat transfer mechanisms as a function of designing variables.

2- Geometrical parameters such as the arrangement of tubes, tubes' diameter and arrangement of the fins are not optimized in reference [5,6]. In the present study these parameters along with other parameters are optimized

3- The pressure drop inside the boiler is neglected in references [5,6]. This assumption makes the system's optimum pinch temperature difference zero. However, in this work, pressure drop and its effect on the performance of the heat recovery steam generator are considered.

4- Increasing number of designing variables, an optimization method (direct search method) is used to minimize the objective function.

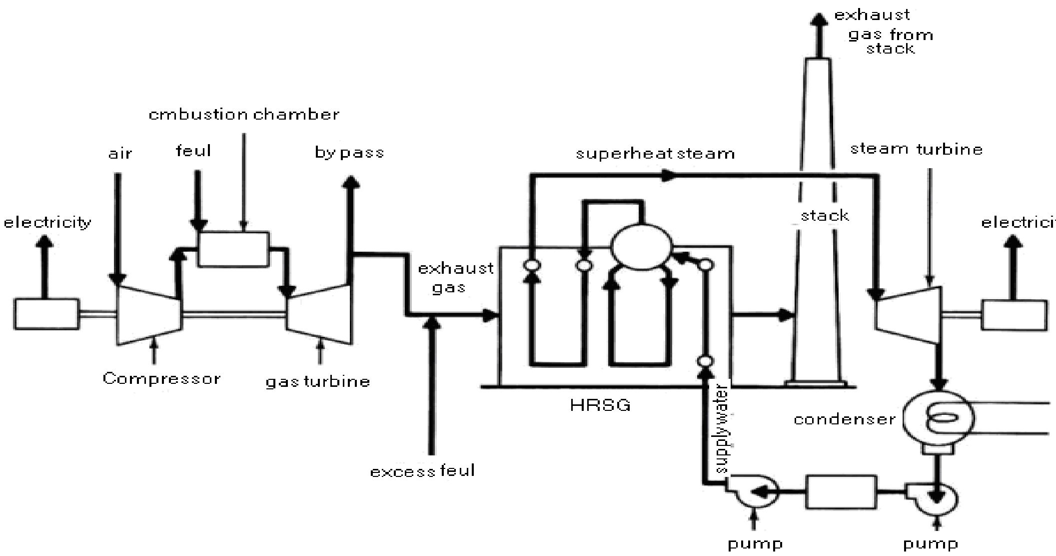


**Fig. 1. Schematic of a combined cycle power plant**

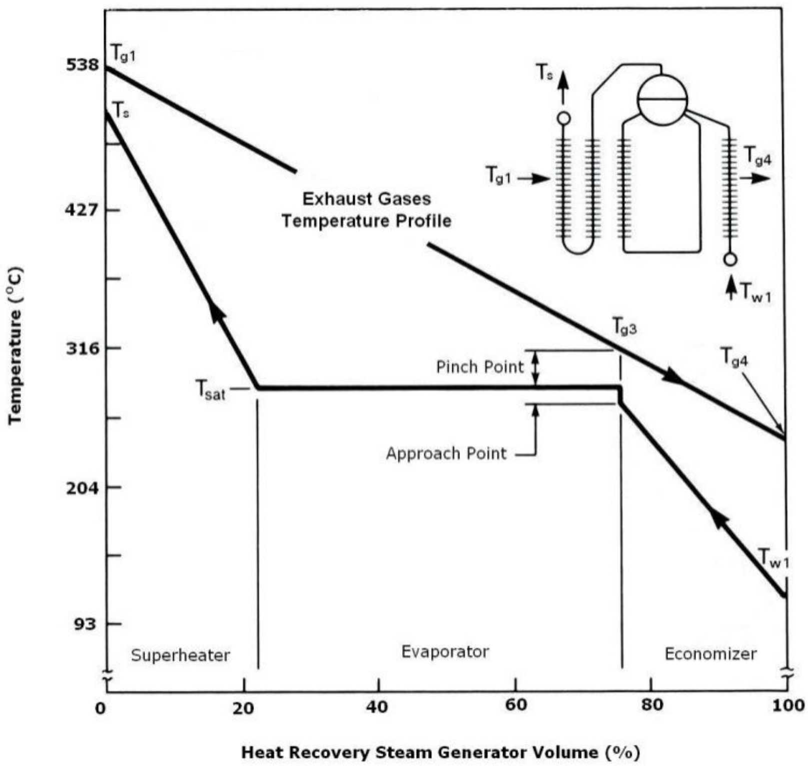


**Fig. 2. Temperature profile in a heat recovery steam generator**

## 2. Thermodynamic analysis

In this section, first, the mass flow rate of the generated steam and the exhaust gas temperature of the HRSG are calculated using the fist-law of thermodynamic and then, different components of exergy in the HRSG are calculated using the second law of thermodynamic.

### 2.1. Analysis using the first law of thermodynamic

Using the first law of thermodynamics for the superheater and evaporator and knowing the steam temperature of the superheater, the temperature of the feed water entering the drum and the pinch temperature difference and using Eq. (1), mass flow rate of the generated steam in the boiler is calculated. Also, once the first law of thermodynamics is used for the economizer, as in Eq. (2), and knowing the temperature of the feed water entering the boiler, temperature of exhaust gas at the stack is obtained.

$$\dot{m}_g\left(h_{g_1}-h_{g_3}\right)=\dot{m}_s\left(h_s-h_{w_2}\right) \quad (1)$$

$$\dot{m}_g\left(h_{g_3}-h_{g_4}\right)=\dot{m}_s\left(h_{w_2}-h_{w_1}\right) \quad (2)$$

### 2.2. Exergy analysis

In order to carry out exergy analysis in a heat recovery steam generator, first, one has to specify different components of exergy in it and then, employ the exergy balance equations. In this work, exergy loss and exergy destruction are considered as the components of the total irreversibility of the heat recovery steam generator which are extensively discussed in the following paragraphs:

#### 2.2.1. Exergy loss

Exhaust of hot gases at the stack denotes heat energy loss and thus wasting of a part of the fuel exergy, Eq. (3). Clearly, the higher temperature of the exhaust gases, the higher will be the rate of exergy loss as a result of the stack heat loss. Therefore, it is always attempted to reduce this temperature. One way to reduce the stack temperature is to reduce the pinch temperature difference.

$$E_{Loss}=m_g\ [(h_{g3}-h_0)-T_0(s_{g4}-s_0)] \quad (3)$$

#### 2.2.2. Exergy Destruction

Friction and temperature differences are two exergy destruction causes, the presence of which in any process reduces the capacity to produce work (exergy). In heat recovery steam generators, temperature difference between the gas and steam and pressure drop of the gas (as a result of friction) cause exergy destruction and therefore a portion of the fuel energy to be consumed in order to overcome these sources of exergy destruction. Thus, because of the mentioned factors, a part of the fuel exergy ($\dot{E}_{Fuel}$) is always destroyed. This exergy is called destructed exergy ($\dot{E}_{Destruction}$) and is calculated using the exergy balance equation for each component, as shown in Eq. (4):

$$\dot{E}_{Destruction} = \dot{E}_{Fuel} - \dot{E}_{Product} \tag{4}$$

where, $\dot{E}_{Product}$ and $\dot{E}_{Fuel}$ are the amount of exergy which are produced and consumed respectively. Equations (5) and (6) are written for each component:

$$\dot{E}_{Fuel} = \dot{m}_g \left[ \left( h_{gi} - h_{go} \right) - T_0 \left( s_{gi} - s_{go} \right) \right] \tag{5}$$

$$\dot{E}_{Product} = \dot{m}_s \left[ \left( h_{wo} - h_{wi} \right) - T_0 \left( s_{wo} - s_{wi} \right) \right] \tag{6}$$

## 3. Heat Transfer

### 3.1. Heat transfer equations

In order to design a heat recovery steam generator, it is essential to first calculate the overall heat transfer coefficient (U). Then, using the available correlations, boiler's heat area and also the number of tubes along the depth of the boiler ($N_d$) are obtained and using that, gas flow pressure drop can also be calculated. For calculation of U, Eq. (7) is used:

$$\frac{1}{U} = \left( \frac{A_t}{A_i} \times \frac{1}{h_i} \right) + \left( \frac{A_t}{A_i} \times ff_i \right) + \left( \frac{A_t}{A_w} \times \frac{d_o}{200k_t} \times \ln \frac{d_o}{d_i} \right) + \left( ff_o \right) + \left( \frac{1}{\eta h_o} \right) \tag{7}$$

Methods for calculating $A_t$, $A_i$ and $A_w$ are presented in [7].

For calculating the internal and external heat transfer coefficients of tubes, ($h_i$ and $h_o$) and also the efficiency of finned heat transfer areas ($\eta$), correlations presented are used in [11]:

$$h_i = \frac{Nu.k}{d_i} \tag{8}$$

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \tag{9}$$

$$h_o = h_r + h_c \tag{10}$$

$$h_r = \sigma \varepsilon_g \frac{T_g^4 - T_t^4}{T_g - T_t} \tag{11}$$

$$h_c = C_1 C_3 C_5 \times \left( \frac{d_o + 2h_f}{d_o} \right)^{0.5} \times \left( \frac{T_g}{T_f} \right)^{0.25} \times GC_{P_g} \times \left( \frac{k}{\mu C_{P_g}} \right)^{0.67} \tag{12}$$

In above equations, $\varepsilon_g$ can be obtained using charts in references [8,9]. Coefficients $C_1$, $C_3$, and $C_5$ are calculated by using the correlations in [7].

### 3.2. Fins and tubes' arrangement

As can be seen in Fig. 3, there are two arrangements for tubes in heat recovery steam generators: In-line and Staggered. One of the advantages of the Staggered arrangement is higher overall heat transfer coefficient and less required heat transfer area. Consequently, using this arrangement might lead to a decrease of capital cost. Pressure drop in this arrangement can be more or less than the In-line arrangement [1,2]. In this work In-line arrangement is taken into account.

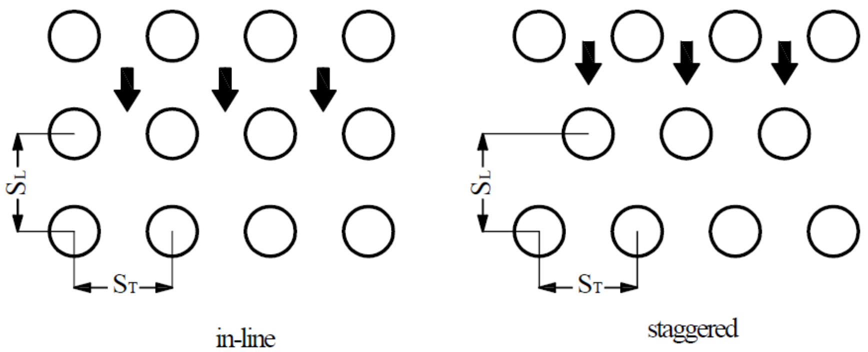


**Fig. 3. Arrangement of tubes in heat recovery steam generators**

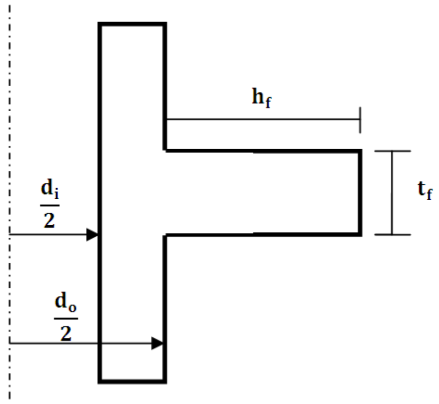


**Fig. 4. Schematic of fin geometry**

As $S_L$ has something to do with evaporator circulation and it is not a design parameter, this parameter isn't analyzed or optimized; But $S_T$ as a design parameter which influences the HRSG's performance is analyzed in the sensitivity analysis section. Likewise, as diameters of tubes (inside and outside) are function of different parameters such as material resistance which is out of the scope of this work, these parameters are neither analyzed nor optimized.

In the HRSG which is under consideration of this work, in all parts, annular fins with constant thickness, as shown in Fig. 4, were used. Important geometrical parameters of these fins are height and thickness and the number of fins per inch. More over Transverse pitch, $S_T$, is a parameter which affects the HRSG's performance. So, four parameters are considered for each pressure part as geometry details. Besides, two operating parameters including pinch temperature difference and gas velocity (width of HRSG) are regarded in the sensitivity analysis and optimization. So 14 parameters are available, including 12 geometry parameters and 2 operating parameters, to be analyzed and optimized.

### 3.3. Gas pressure drop

The gas which enters heat recovery boiler is the output of a gas turbine. In the perfectly ideal state, this gas should leave turbine at the atmospheric pressure and enter the boiler but because of the gas flow pressure drop, gas turbine's exhaust gas is always expelled from the turbine in a pressure higher than the atmosphere pressure to compensate for this pressure drop and finally get out of the HRSG at the atmospheric pressure. Therefore, it is clear that the higher the pressure drop inside the boiler, the higher will be the turbine's output pressure and the lower the turbine's output work. Thus, it has always been attempted to decrease the gas flow pressure drop inside the boiler and thus increase the work output of the turbine. This pressure drop is calculated by using the following equation:

$$\Delta P_g = \left(f_g + a\right)\frac{G^2 N_d}{500\,\rho} \tag{13}$$

Where

$$\rho = \frac{12.2M}{T_g} \tag{14}$$

Methods for calculation of parameters $f_g$ and $a$ are also provided in [11].

## 4. Optimization

The first step to optimize a thermodynamic system is to determine a proper objective function. The next step is to minimize this objective function which can be done using a number of different numerical, analytical and graphical methods. Among these methods, direct search has been selected as a numeric method. Using the equations presented in the sections of thermodynamic and heat transfer analyses, the objective function is optimized as a function of pinch temperature difference and some geometrical parameters of HRSG. Here, the rate of total irreversibility is chosen as the objective function which represents thermodynamic performance of the system and is the sum of rate of exergy destruction for each component and rate of exergy loss Eq. (15). Also, using Eq. (16) and Eq. (17), important thermodynamic quantities which clarify the thermal state of the system like thermal efficiency and exergetic efficiency are determined to enrich the comparison between the optimum state and the initial state more thoroughly.

$$I = \sum \dot{E}_{Destruction} + \dot{E}_{Loss} \tag{15}$$

$$\Psi = \frac{E_{s1} - E_{s4}}{E_{g1}} \tag{16}$$

$$\eta = \frac{\left(h_{wo} - h_{wi}\right)}{\left(h_{gi} - h_o\right)} \tag{17}$$

### 4.1. The optimization method employed

The utilized method is Conjugate Directions Method, or as it is also named after its innovator, Powell's Method. The main idea in this optimization method is one dimensional search to find the optimum value which is a function of variables X1, X2,… . When there is a starting point, it seems reasonable to look for the optimum value of X1, while X2, X3 … are held constant. Next, after having found the optimum value for X1, this process is repeated for the other dimensions. Uni-variate search method doesn't necessarily require numerical differentiations and thus is advantageous for the functions which are not easily differentiable However, searching with changing one variable at a time is not necessarily effective, as shown in the left part of Fig. 5. Search along a series of directions which are perpendicular to the objective function, as shown in the right part of Fig. 5, seems more effective.

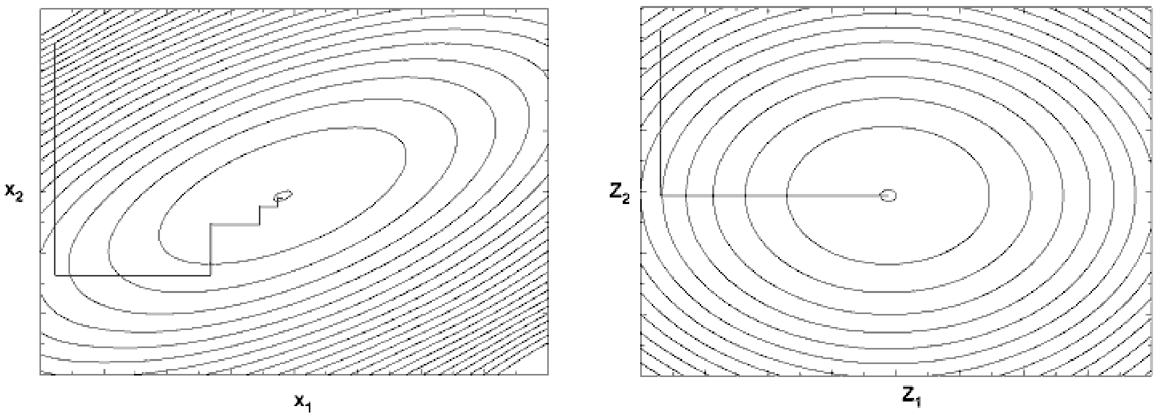


**Fig. 5. Two dimensional representation of direct search and uni- variate search**

**Table 1. Data of a water-tubed heat recovery steam generator**

| Property | Value |
|---|---|
| boiler input feed water temperature | 110°C |
| temperature of the generated steam | 480°C |
| Pressure of the generated steam | 10MPa |
| emperature of the turbine's exhaust gases | 575°C |
| Flow rate of the turbine exhause gases | 30Kg / s |
| Pinch temperature difference $\Delta T_{pp}$ | 5°C |
| Gas analysis | |
| $CO_2$ | 3٪ |
| $H_2O$ | 7٪ |
| $N_2$ | 75٪ |
| $O_2$ | 15٪ |
| geometry property | |
| Tubes' external diameter | 0.05m |
| Tubes' internal diameter | 0.0425m |
| Arrangement type | In-Line |
| Transverse pitch between tubes | 0.1m |
| Dimensions of the economizer and the evaporator | 2 m ×4 m |
| Dimensions of the superheater | 2 m ×3 m |
| Fins' properties in economizer | 4×0.0015×0.015 (m) |
| Fins' properties in evaporator | 5×0.002×0.01 (m) |
| Fins' properties in superheater | 2×0.001×0.01 (m) |

**Table 2. Results obtained using initial values of decision variables in**

| Thermodynamic parameters | Values |
|---|---|
| Rate of irreversibility ($\dot{I}$ (Kw )) | 1959 |
| Thermal efficiency (η) | 72.53٪ |
| Exergetic efficiency (ψ) | 75.81٪ |

## 5.Results

In this analysis, a designed HRSG is considered as the test case which its data are brought in Table 1. Exergy analysis is performed to calculate exergy destruction and exergy loss at different parts of the boiler. At the second stage, changing the design variables, the sensitivity of the objective function with respect to the parameters is studied. In the third stage, the design variables are optimized and the results of optimization are compared with non-optimized state.

### 5.1. Exergy analysis of the initial design

Initial values for fins' geometrical parameters and operating parameters are shown in table 1. Based on these values, thermodynamic decisive variables as shown in Table 2. are calculated.

### 5.2. Sensitivity analysis of decision variables

This sensitivity analysis includes analysis of changes of decision variables and observing the effect of each of these variables on the objective function, while other parameters are held constant according to the data given in table 1. Since the geometry details are similar for the three pressure sections, curves of each geometry parameters are brought in one figure for all pressure parts.

#### 5.2.1. Sensitivity analysis on the pinch temperature difference

When the pinch temperature difference approaches zero, because the temperature difference between the exhaust gas flow of the generator and water flow is decreased, exergy destruction caused by that is also decreased and also the temperature of the exhaust gases and the involved exergy loss is decreased. However, at the same time, heat transfer area and the number of tubes standing against the gas flow is increased and this in turn results in increased pressure drop and its resulting exergy destruction. As can be seen in Fig. 6, this increase in exergy destruction related to pressure drop at amounts of pinch temperature difference smaller than 1.2$^o$C, dominates the reductions that happen in irreversibility. The opposite holds about pinch temperature differences larger than 1.2$^o$C. For the sensitivity analysis of the pinch temperature difference against factors of irreversibility, pinch temperature difference has been varied between 0.1$^o$C and 5$^o$C.

As the width of HRSG determines the gas velocity (which is an operating parameter), this parameter is regarded among operating parameters. As can be seen in Fig. 7, by increasing the width of the steam generator, pressure drop results in decrease of irreversibility. The sensitivity analysis is carried out in the range of 1.5-4 m. The optimum value of this parameter should therefore be obtained via thermoeconomic optimization.

#### 5.2.2. Sensitivity analysis of geometric parameters of HRSG

Choosing the fin configuration, especially in gas flows with lower particles' concentration, is a function of many parameters including heat transfer coefficient of each part of the tube, overall size, cost and gas flow pressure drop which all affect the operational cost [1,2]. The optimal values calculated in the sensitivity analysis are evaluated while all other values were held constant base a on the data given in table 1. The geometric parameters which are analyzed include; height and thickness of fins, number of fins per inch and transverse pitch.

Decreasing the height and thickness of the fins results in decreasing of the heat transfer area per unit length of the tube and as a result, frictional pressure drop will be increased. On the other hand, excessive increase of the height and/or thickness of the tubes will result in a reduction of heat transfer coefficient and thus, increase of heat transfer area which in turn causes pressure drop. That excessive increase also increases gas velocity in tubes and results in more pressure drop.

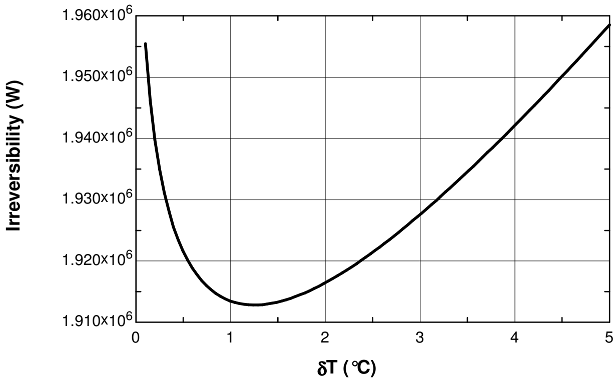


**Fig. 6. Sensitivity analysis for pinch temperature difference**

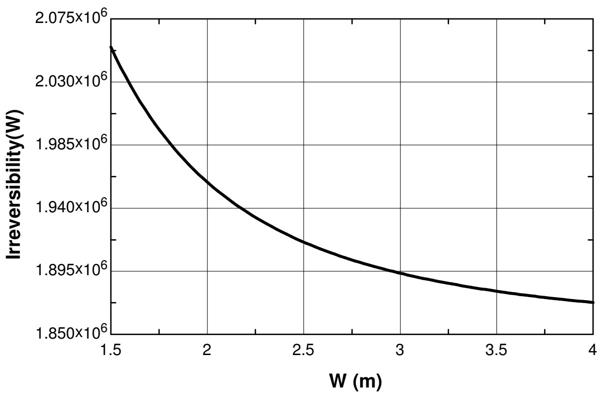


**Fig. 7. Sensitivity analysis for width of the steam generator**

Consequently, geometric parameters of these fins in economizer and evaporator have optimum values. The fin thickness is varied between 0.0005 m and 0.0015 for the three pressure parts in Fig. 8.

Sensitivity analysis for height of the fins of all three pressure parts is carried out in the range 0.001m-0.015, as shown in Fig. 9. When the ratio of the heat transfer coefficient inside and outside the fins is reduced, one can more effectively take advantage of the ration of external extended areas to internal extended areas [2]. Since the heat transfer ratio in the tube part of the superheater is less than that of the economizer and the evaporator and is close to heat transfer coefficient of the gas, using fins in this part may lack thermodynamic justification. The Fig. 9 confirms this truth.

The number of fins per inch has a considerable effect on pressure drop and consequently, value of irreversibility in the heat recovery steam generator. Increasing the number of fins results in the reduction of the area available for gas flow and the increase of gas speed and consequently, increase of exhaust gas speed in gas passage tubes. Reduction of the number of fins per inch in turn results in the reduction of area per unit length, which means increase for the fins' height and increase of frictional pressure drop. Therefore, there should be an optimum value. As it can be seen in Fig. 10, the curve confirms this.

As it can be seen in Fig. 11, transverse pitch of fins has a magnificent effect on the rate of irreversibility. Increasing transverse pitch of fins, results in lowering the exhaust velocity and consequently, reduction of pressure drop and irreversibility. Therefore, transverse pitch, from the thermodynamic point of view, doesn't show an optimum value. The optimum value of this parameter should be looked for through thermo economic optimization and analysis.

### 5.3. Results related to optimal values of decision variables

A suitable algorithm for optimization is needed. In this work, the direct search method has been selected. In this method, a range for each of the decision variables should be chosen. These ranges are shown in table 3. The data given in this table including lower and upper limits are based on common values used in industrial boilers. The given values for geometry details are employed for the three pressure parts.

Optimal values of the geometrical parameters and also pinch temperature difference are shown in table 4. Although in sensitivity analysis optimum values were found for most parameters, but some of these parameters don't exhibit optimum values in optimization. The reason is that the other parameters experience considerable changes and the behavior of this parameter changes accordingly. So the optimum value may occur out of the bounds defined in table 3. In addition, although these values optimize the irreversibility function, they do not seem to be practical. The reason is that, the objective function is merely thermodynamic and doesn't consider the economic aspects of the problem. To achieve this goal, one needs to define a suitable thermo economic objective function which considers both thermodynamic and economic issues and is a way too complicated and far beyond the scope of this work.

**Table 3. Upper and lower limits of decision variables for the three pressure parts**

| Parameters | Lower limit | Upper limit |
|---|---|---|
| $\Delta T_{PP}$ | 0.1 | 5 |
| W | 1.5 | 4 |
| $h_f$ | 0.001 | 0.015 |
| $n_f$ | 2 | 8 |
| $S_{tf}$ | 0.06 | 0.15 |
| $t_f$ | 0.0005 | 0.0015 |

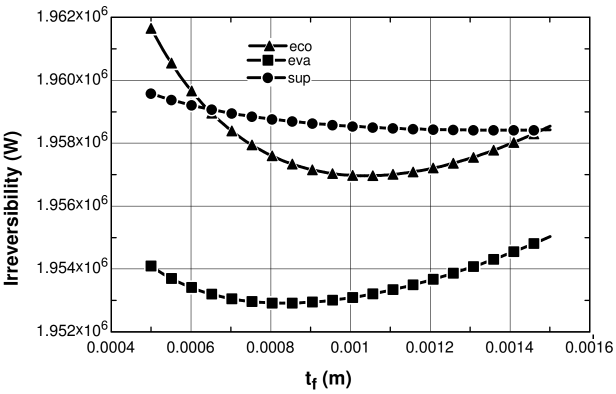


**Fig. 8. Sensitivity analysis for thickness of fins of the economizer**

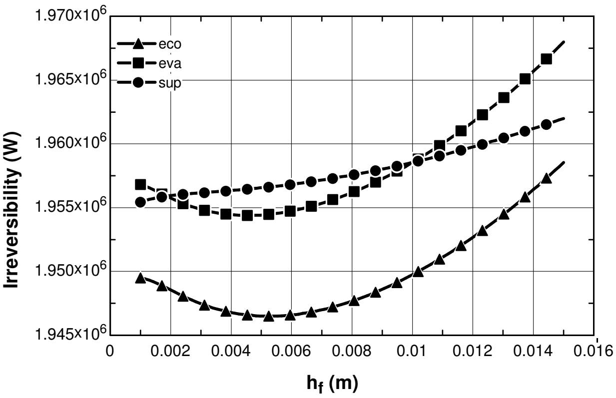


**Fig. 9. Sensitivity analysis for height of fins of the economizer**

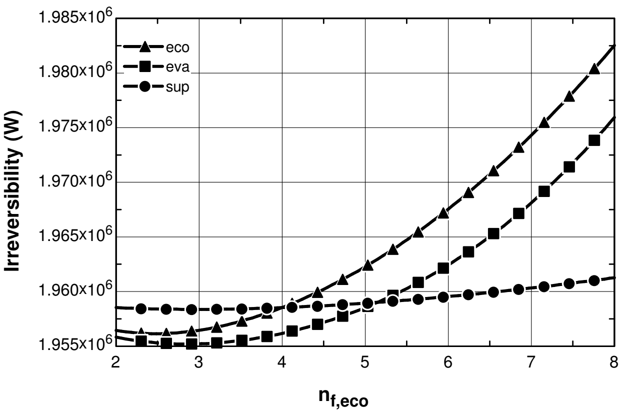


**Fig. 10. Sensitivity analysis for the number of fins per inch in the economiz**

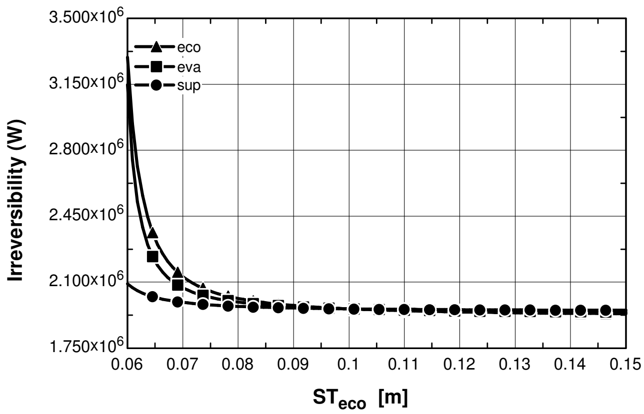


**Fig. 11. Sensitivity analysis for transverse pitch of the fins in the economizer**

**Table 4. Operating parameters geometric detail in its optimum state for each part of the heat recovery steam generator**

| Thermodynamic parameters | Value | | | |
|---|---|---|---|---|
| Pinch temperature difference ($^oC$) $\Delta T_{pp}$ | 0.1045 | | | |
| W | 4 | | | |
| Geometry parameters | Fin's thickness $t_f$ (m) | Fin's height $h_f$ (m) | Number of fins per inch $n_f$ | Fin's transverse Pitch $S_T$ (m) |
| Fin properties in economizer | 0.0005 | 0.007206 | 8 | 0.015 |
| Fin properties in evaporator | 0.0005553 | 0.00769 | 8 | 0.015 |
| Fin properties in superheater | 0.0005 | 0.005123 | 8 | 0.015 |

**Table 5. Results related to initial values of decision variables**

| Thermodynamic parameters | Values |
|---|---|
| Rate of irreversibility ( $\dot{I}$ ) | 1756 kW |
| Thermal efficiency (η) | 73.86 % |
| Exergetic efficiency (ψ) | 78.06 % |

As it can be seen in the table 4, the height of fins in superheater is less than that of economizer and evaporator. This confirms the results presented in the sensitivity analysis section.

Pinch temperature difference is much lower than the recommended amounts. In fact, the pinch temperature difference is obtained through a balance of irreversibility caused by pressure drop and sum of the irreversibility values caused by temperature difference and exergy loss which is not practical. The balance which results in the values recommended in references is actually a balance between the cost of irreversibility and the initial cost.

As it can be seen in the first column of Fig. 12, the amount of exergy loss is considerable compared to the other forms of irreversibility. Moreover, as the result of optimization doesn't look practical, it is favorable to optimize the system from thermo economic point of view. Obviously, the optimization algorithm tends to decrease exergy loss to decrease irreversibility value to a higher degree. The second column of the above chart confirms this.

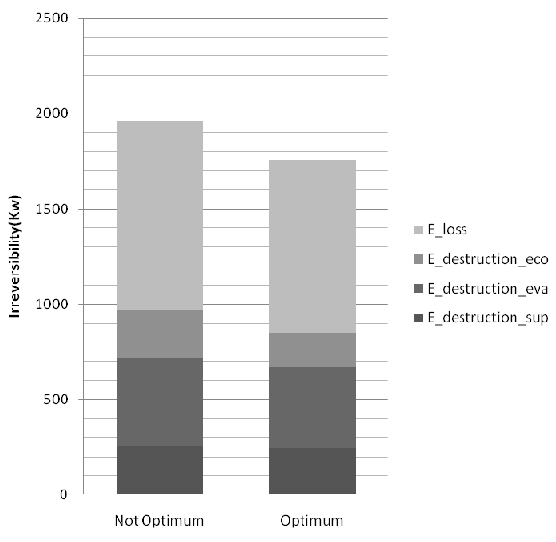


**Fig. 12. Irreversibility in each of the parts of the generator, before (left) and after (right) the optimization**

## 6. Conclusions

In sensitivity analysis, it is found that some parameters possess optimal points relative to the objective function, whereas some others don't show such a behavior. In addition, the above results show that geometry details as well as operating parameters considerably influence the thermodynamic state of the system. As a result, the rate of irreversibility reduces by 203 kw and thermal and exergetic efficiencies raises by 1.33 and 2.25 percent respectively. The optimum values differ greatly from the common values in industry and this is merely because of having ignored the economic issues in choosing the objective function. It was also shown that the bigger part of the irreversibility rate is due to exergy loss which can be lowered through thermodynamic optimization.

In this work thermodynamic analysis is carried out and consequently the obtained results are not close to reality because economic issues are not considered. For improvement and development of this work, a thermo economic objective function is needed which in turn requires extensive data about the economic parameters of the problem.

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## Nomenclature

| | |
|---|---|
| A | area per cm of tube length $cm^2/cm$ |
| $C_p$ | specific heat (J/kg.k) |
| D | tube diameter (m) |
| $\dot{E}_{Destruction}$ | time rate of exergy destruction (kw) |
| $\dot{E}_{Fuel}$ | time rate of fuel exergy (kw) |
| $\dot{E}_{Loss}$ | time rate of exergy loss (kw) |
| $\dot{E}_{product}$ | time rate of product exergy(kw) |
| $\dot{I}$ | time rate of total irreversibility(kw) |
| ff | fouling coefficient inside the tube ($m^2$.k/w) |
| $f_g$ | friction factor |
| G | gas mass velocity(kg/$m^2$.s) |
| h | enthalpy entropy (kJ/kg), heat transfer coefficient (w/$m^2$.k) |
| $h_c$ | convective heat transfer coefficient (w/$m^2$.k) |
| $h_f$ | fin height (m) |
| $h_r$ | radiative heat transfer coefficient (w/$m^2$.k) |
| k | thermal conduction factor (w/m.k) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $N_d$ | number of tubes in front of gas (deep direction) |
| Nu | Nusselt number |
| Pr | Prandtl number |
| Re | Reynolds number |
| s | entropy (kJ/kg.k) |
| T | temperature (k) |
| $t_f$ | fin thickness (m) |
| U | total heat transfer coefficient (w/$m^2$.k) |
| W | width of the HRSG |
| $\varepsilon$ | emissivity |
| $\eta$ | efficiency of heat transfer area, Thermal efficiency |
| $\mu$ | viscosity (Pa.s) |
| $\rho$ | density (kg/$m^3$) |
| $\sigma$ | Stefan-Boltzmann constant (w/$m^2$.$k^4$) |
| $\Psi$ | exergy efficiency |
| $\Delta P$ | pressure drop in HRSG (kPa) |

**Subscripts**

| | |
|---|---|
| f | fin |
| g | gas |
| i | inlet, inside |
| L | longitudinal pitch |
| o | outlet, outside |
| pp | pinch point |
| 0 | restricted dead state |
| s | steam |
| sat | saturation state |
| t | total, tube |
| T | transverse pitch |
| w | water, wall |

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