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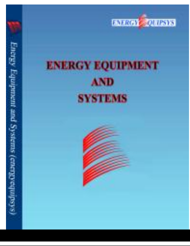

Multi-objective optimization of heat recovery steam generators

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ABSTRACT

In this paper, a multi-objective method is used to optimize a heat recovery steam generator (HRSG). Two objective functions have been used in the optimization, which are irreversibility and HRSG equivalent volume. The former expresses the exergetic efficiency and the latter demonstrates the cost of the HRSG. Decision variables are geometric and operational parameters of the HRSG. The results of the multi-objective optimization are shown in a famous curve called the Pareto curve. The resulting Pareto curve can be used as a decision making tool by designers. Different optimal parameters are presented for different weight coefficients in the function. Volume and exergy optimization are special cases of the proposed algorithm. It is also shown that thermoeconomic and multi-objective optimizations can also be specific cases of the proposed algorithm if the proper weight coefficient is used. This weight coefficient depends on local prices of energy and construction costs of the HRSG.

Keywords: Combined Cycle, HRSG, Multi-objective Optimization, Thermoeconomic.

1. Introduction

Employing combined cycle plants is one of the most prevalent methods of power generation. High efficiency, low heat loss and relatively long lives of combined cycle power plants are among the advantages distinguishing them from other types of power plants, and that is why in recent years a large number of these power plants have been employed in different parts of the world [9].

Figure 1 shows a schematic view of a combined cycle. As is shown in the figure, the exhaust gas of the gas turbine superheats the water required for the steam turbine in the heat exchanger. The temperature profile in an HRSG is as shown in Fig. 2. The difference between the saturation temperature and the temperature of the gas at the evaporator outlet is defined as the pinch temperature difference. Reducing the pinch temperature difference signifies reducing stack losses, increasing pressure drop and the cost of the HRSG.

For a system, particularly a combined cycle, there are two methods to optimize energy consumption. One of these methods is to optimize the whole system as an integrated unit. The second method is the optimization of each component of the whole system individually. In this method the optimization is focused on the component in which more avoidable irreversibility occurs from this point of view.

HRSGs are the most important parts of these power plants. Some papers have been published about the optimization of HRSGs. P.K Nag S. De (1997) presented a design method of an HRSG with minimum possible

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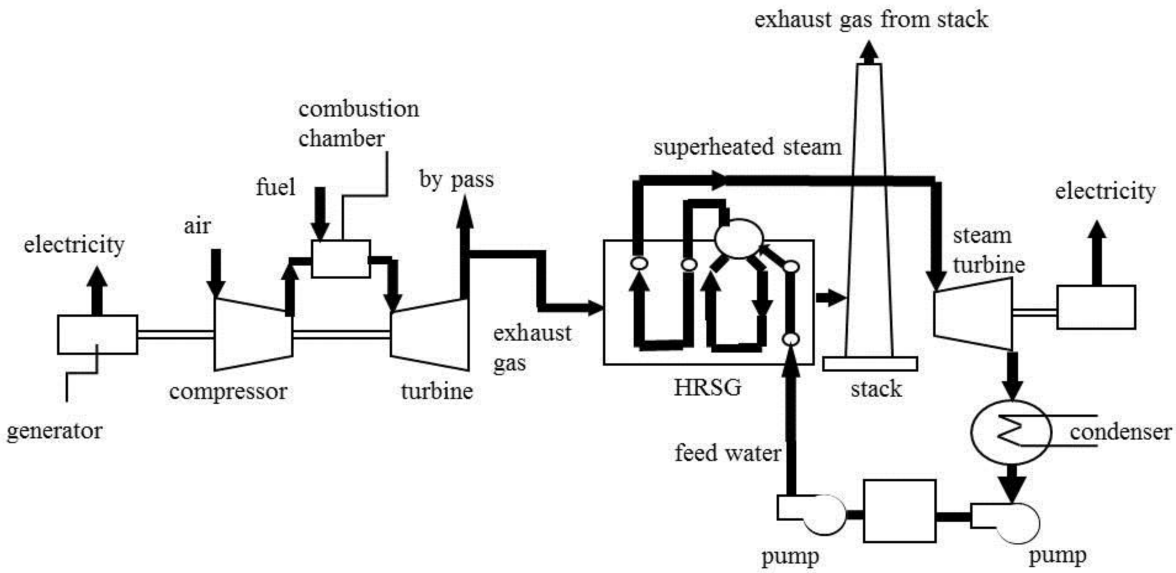

Fig. 1. Schematic view of a combined cycle

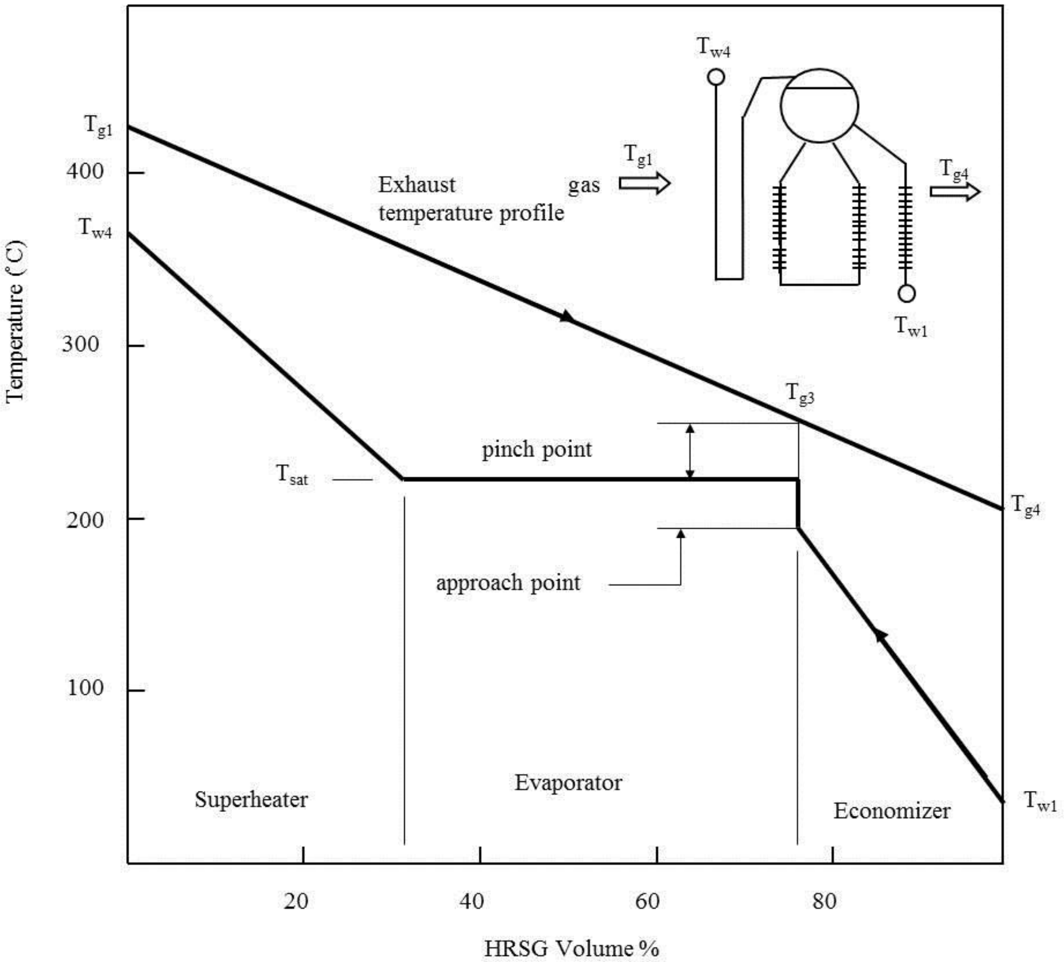

Fig. 2. A schematic view of the temperature profile in an HRSG

irreversibility and analysed the sensitivity of operating parameters on HRSG irreversibility. Butcher and Reddy (2007) investigated the effects of different operating conditions on second law efficiencies of HRSGs without any optimization. Casarosa and Donatini (2004) and Russo (2002) carried out thermoeconomic optimization in which the pinch temperature difference is optimized. As noted by the authors, the pinch temperature difference value calculated in this paper is unreasonably small. Franco and Giannini (2006, 2007) have proposed a two-step algorithm, the first step of which is minimizing the pressure drop in a constant heat transfer load. The second step is geometric minimization of HRSGs' compactness factor, while thermal parameters are held constant based on the former optimization. Mohagheghi and Shayegan (2009) proposed a merely thermodynamic optimization method, which includes optimization of pressure values of multi-pressure generators. Behbahani et al. (2010) optimized a fire tube HRSG from thermodynamic and thermoeconomic point of views through genetic algorithm. Tajik Mansouri et al. investigated the effect by using dual pressure and two arrangement of triple pressure HRSGs. They showed that increasing the number of pressure parts to lead to a lower level of exergy losses [15]. Kaviri et al. (2011) considered the effect of thermodynamic optimization of an HRSG in a combined cycle. They concluded that using a gas inlet temperature up to 650 ℃ creates the best conditions for the whole combined cycle efficiency (Kaviri, 2013). Rovira et al. (2011) developed a thermoeconomic model in order to obtain an optimum design for the HRSG. Carapellucci and Giordano (2013) optimized an HRSG through a thermoeconomic and exergetic method in a combined cycle. The parameters included levels of pressure and HRSG layouts.

Some scientists have also investigated some energy systems from a multi-objective point of view. Sanaye and Hajabdollahi (2012) thermally modelled and optimized a compact heat exchanger through multi-objective optimization. Kaviri et al. (2011) modelled a dual pressure combined cycle power plant and presented a multi-objective method to optimize the equipment. Hajabdollahia et al. (2011) investigated a dual pressure HRSG enhanced with firing in a combined cycle and optimized the exergetic efficiency and total capital cost in a multi-objective optimization. The investigated capital cost in their paper is based on the amount of heat transfer surfaces, like almost all mentioned studies- since detailed simulations have not been under consideration, no better criteria were available- which may not be as logical as volume, which is considered in this paper. In their paper the unit cost of steam is selected for the objective function, while in this paper, by considering the same unit cost of electricity on exergy loss and destruction, the total annual capital cost is formed. All decision variables in their paper are operational ones, which is due to simple rough estimation of the heat transfer coefficients. Najafi et al. (2009) considered a multi-objective optimization of a fire tube HRSG. In their paper, some geometrical parameters are considered as decision variables, and pinch temperature difference is considered as the operating one. The capital investment is considered in terms of heating surfaces instead of mass.

In this work, a new method is presented for optimizing heat recovery steam generators based on a multi-objective optimization method. The objective functions are calculated through a detailed design procedure, where every characteristic of the system, including pressure drop, plays a role in the objective functions. The simulation is carried out for a gas mixture as real gas instead of ideal, and the properties of the flue gas are calculated based on consideration of the pressure drop through trial and error. As design characteristics are taken into account, every parameter can be chosen as a design parameter. In this work, both operational and geometrical parameters are considered for optimization. The two objective functions chosen in this work are irreversibility and an equivalent volume of the HRSG. Therefore, the presented Pareto front is independent of the region. The generated Pareto curve may be used for decision making for the entire world. The equivalent volume is defined as the sum of volumes of the evaporator, the economizer and the superheater, while for the volume of the superheater a multiplier is used due to its better material and higher price. The construction costs of an HRSG depend on its volume, and minimizing its volume results in the minimization of construction costs. Minimizing irreversibility, furthermore, results in the maximization of efficiency. The objective function that is optimized in this work is the sum of these two functions multiplied by their weight coefficients. It is also shown in this work that the thermoeconomic approach is a specific case of this algorithm if suitable values are chosen for those coefficients. Which instead of depend on local costs of exergy and construction costs.

Nomenclature

A	area per cm of tube length cm^2/cm
C_p	specific heat (J/kg.K)
D	tube diameter (m)
$E_{Destruction}$	time rate of exergy destruction (kW)
E_{Fuel}	time rate of fuel exergy (kW)
E_{Loss}	time rate of exergy loss (kW)
$E_{product}$	time rate of product exergy(kW)
E_{irr}	time rate of total irreversibility(kW)
ff	fouling coefficient inside the tube (m^2.K/W)
f_g	friction factor
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)
m	mass of the material used (kg)
$\dot{m}$	mass flow rate (kg/s)
N_d	number of tubes in front of gas (deep direction)
Nu	Nusselt 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

Greek symbols

ΔP	pressure drop in HRSG (kPa)
ε	emissivity
η	efficiency of heat transfer area
ρ	density (kg/m^3)
σ	Stefan-Boltzmann constant ($W/m^2.K^4$)

Subscript

e	electricity
eq	equivalent
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

2. Thermodynamic analysis

In this section, initially the first law of thermodynamics is used to calculate the mass flow rate of the steam produced, intake gas temperature in each of pressure parts and the temperature of exhaust gases at the stack; then, using the second law of thermodynamics, different components of exergy in an HRSG are calculated.

2.1 First law analysis

Once the first law of thermodynamics is written for the superheater and the evaporator, the mass flow rate of the steam produced can be obtained as in Eq. (1). Moreover, if the first law of thermodynamics is written for the economizer, as in Eq. (2), the temperature of exhaust gases at the stack can be calculated.

$$\dot{m}_g C_{P_g} H_L (T_{g_1} - T_{g_3}) = \dot{m}_s (h_s - h_{w_2}) \quad (1)$$

$$\dot{m}_g C_{P_g} H_L (T_{g_3} - T_{g_4}) = \dot{m}_s (h_{w_2} - h_{w_1}) \quad (2)$$

2.2 Exergy analysis

In order to carry out exergy analysis in an HRSG, first different components of exergy have to be evaluated in it and then the equation of exergy balance should be used. In this work, exergy loss and exergy destruction are discussed in detail as components of HRSG's irreversibility.

2.2.1 Exergy loss

When the hot gases leave the stack, thermal energy, and therefore a part of the exergy of the fuel, is lost. The value of this exergy can be obtained by Eq. (3) [14]. It is obvious that the higher the temperature of the exhaust gases, the more exergy will be lost. Thus, reduction of this temperature is always attempted, and one of the ways to reduce this temperature is to decrease the pinch temperature difference.

$$\dot{E}_{Loss} = \dot{m}_g[(h_{g_4} - h_0) - T_0(s_{g_4} - s_0)] \quad (3)$$

2.2.2 Exergy destruction

Friction and temperature difference are among the factors that reduce capability to give work. In HRSGs, exergy is dissipated through both heat transfer through finite temperature difference and gas pressure drop, and a part of exergy of the fuel is used to overcome these sources of irreversibility. Therefore, these factors always result in the partial loss of fuel exergy. This loss is called exergy destruction, and is calculated by the exergy balance equation, Eq. (4).

$$\dot{E}_{Destruction} = \dot{E}_{Fuel} - \dot{E}_{Product} - E_{Loss} \quad (4)$$

where $\dot{E}_{Product}$ and $\dot{E}_{Fuel}$ are produced and consumed exergy, respectively. These parameters can be calculated using Eq. (5) and Eq. (6).

$$\dot{E}_{Fuel} = \dot{m}_g[(h_{gi} - h_{g0}) - T_0(s_{gi} - s_{g0})] \quad (5)$$

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

3. Heat transfer

3.1 Heat transfer equations

In order to design an HRSG, the overall heat transfer coefficient (U) has to be calculated. Then, the surface area required for heat transfer and the number of tubes in the gas side (N_d) will be determined. U is evaluated using Eq. (7).

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

A_t, A_i and A_w are determined as shown in [10]. Moreover, the external heat transfer coefficient of the pipes equals the sum of convection and radiation heat transfer coefficients.

$$h_o = h_r + h_c \quad (8)$$

The equations in [8] are used to calculate the efficiency of finned thermal surfaces (η). Heat transfer coefficients inside and outside the tubes (h_i and h_o) are calculated using the Eqs. (9, 10):

$$h_i = \frac{Nu.k}{d_i} \quad (9)$$

$$h_r = \sigma\varepsilon_g \frac{T_g^4 - T_t^4}{T_g - T_t} \quad (10)$$

where ε_g can be obtained using the charts given in [25]. Moreover, convection heat transfer of the outside part of the tubes, depending on whether the tubes are finned or bare, is calculated using different equations given in Ganapathy (2003).

Then, pressure drop is calculated:

$$\Delta P_g = (f_g + a)\frac{G^2 N_d}{500\rho} \quad (11)$$

f_g and a are evaluated as explained in [10].

3.2 Fins and tubes

Tube arrangement significantly affects the performance and cost of an HRSG [1]. As can be seen in Fig. 3, there are two choices for tube arrangement: in-line and staggered. An advantage of the staggered arrangement is its higher heat transfer coefficient and thus lowers heat transfer area. Therefore, using the staggered arrangement results in a reduction of capital costs. However, the pressure drop of the staggered arrangement can be both more and less than the in-line arrangement [10]. In this work, the in-line arrangement is considered.

In HRSGs, it is customary to use finned tubes. Using finned tubes may result in a reduction of capital costs, as well as pressure drop, which influences operational costs. Using fins is, however, effective only when the ratio of heat transfer coefficient outside the tube to its value inside the tube is small[10]. This fact holds for economizer and evaporator tubes, but not superheaters. In this work, the tubes of the evaporator and the economizer are assumed to be enhanced with fins but those of the superheater are assumed bare. In Fig. 4 the geometric parameters of fins are given.

4. Economical analysis

The total annual cost of an HRSG is defined in this work as the sum of the yearly cost of construction and irreversibility costs, as follows:

$$C_{tot} = C_{cc} + C_{irr} \tag{12}$$

The construction costs of thermal surfaces in an HRSG depend mainly on its weight. The total cost of construction of an HRSG includes other costs such as shell, the casing and the pipework. These costs are considered proportional to the cost of thermal surfaces material. Thus, the annualized capital cost of construction of an HRSG is as Eq. (13):

$$C_{cc} = CRF \times BMF \times PEC \tag{13}$$

where CRF is the capital recovery factor and BMF is a factor accounting for the lateral costs to calculate the total capital cost of the HRSG. PEC is cost of the heat transfer surface area and, as mentioned earlier, it is assumed to be a function of its equivalent volume and is calculated by Eq. (14).

$$PEC = c \times \rho \times V_{eq} \tag{14}$$

c is cost of unit mass of material used in the evaporator and economizer in which construction costs are taken into account. ρ is the density of material used in manufacturing the pipes and fins and Veq is the equivalent volume of the heat transfer area, which is slightly different from its volume. The difference is due to the fact that the cost of material varies for different pressure parts. For instance, in the super heater the average wall temperature is higher and tubes should have a higher mechanical resistance, which results in the necessity to use more expensive alloys, which increases the cost of this section.

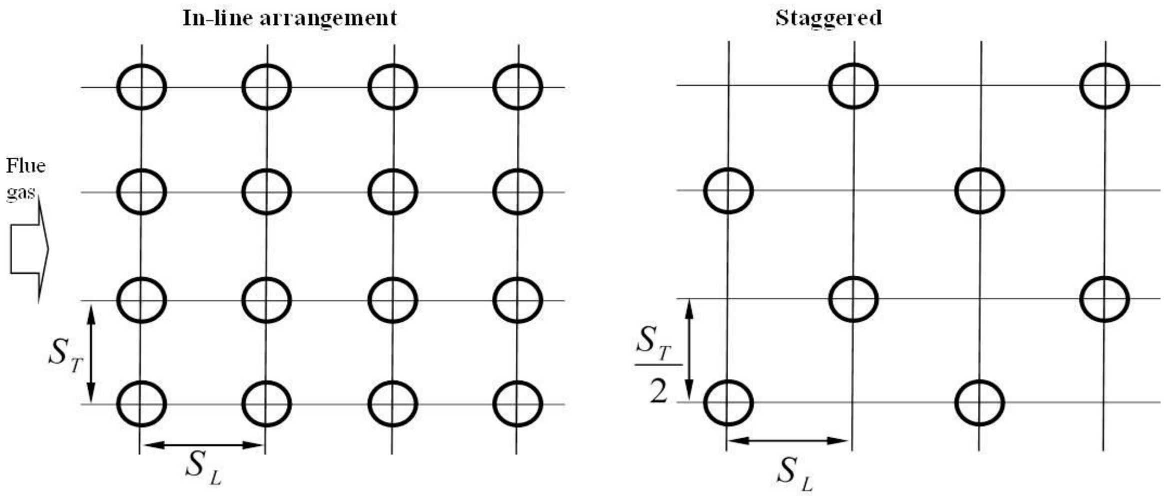

Fig. 3. Tube arrangements in HRSGs

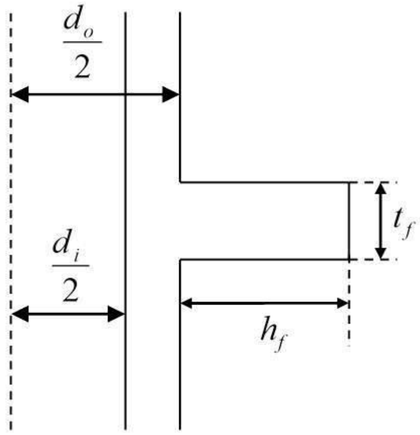

Fig. 4. A schematic sectional view of a fin with consistent thickness

$$V_{eq} = V_{eco} + V_{eva} + 1.86 \times V_{\sup} \quad (15)$$

where 1.86 is a coefficient representing the fact that, in the superheater, the tubes are bare and made from a different alloy.

Since increasing the irreversibility means the lower electricity production of the cycle, the costs of the irreversibility, as shown in Eq. (16), can be calculated by multiplying an exergetic efficiency of the combined cycle by the unit cost of electricity:

$$C_{irr} = C_e \times (\zeta \times E_{irr}) \times H \quad (16)$$

where ζ is the exergetic efficiency of a combined cycle and H is the number of working hours in a year.

5. Optimization

5.1 The objective function

As mentioned earlier, a double objective function is considered for this optimization. One of the functions is irreversibility and the other is the equivalent volume of the HRSG material. In order to generalize these results, these values are made dimensionless as in Eq. (17).

$$F = \alpha(\frac{E_{irr}}{E_{\max}}) + (1-\alpha)(\frac{V_{eq}}{V_{eq_{\max}}}) \quad (17)$$

where α is the weight coefficient and E_{irr} is the total irreversibility of the HRSG, which, as shown in Eq. (18), includes the exergy destruction of all pressure parts and also the exergy loss resulting from the discharge of combustion products to the atmosphere.

$$E_{irr} = \sum E_{Destruction} + E_{Loss} \quad (18)$$

$V_{eq\,\max}$ is the equivalent volume of the HRSG when the pinch temperature difference is 10^{-4} ^{o}C, which is estimated to be 20.54m^3.

Regarding the value of irreversibility as the objective function means that, by decreasing the irreversibilities in the HRSG, the energy production in the combined cycle power plant increases. That reduction is proportional to the additional electricity production.

E_{max} is assumed to be the exergy input of the HRSG by combustion products. Thus, the efficiency of this generator can be calculated in terms of dimensionless irreversibility, using Eq. (19):

$$\zeta = 1 - \frac{E_{irr}}{E_{in}} \quad (19)$$

5.2 Design variables

In this study, in addition to the important operational parameter, pinch temperature difference, geometrical parameters such as width of the HRSG, longitudinal and transverse tube pitches for all pressure parts, and also fin parameters including height, thickness and number of fins per inch are taken into account. Therefore, 16 decision variables are introduced to the optimization process.

The length of tubes in the evaporator is not considered in the optimization because their length influences the circulation of the fluid in the evaporator, and so is not regarded as a decision variable. Moreover, the internal and external diameters of tubes are not considered in the optimization since these values are determined by mechanical factors like their internal pressure, their mechanical resistance of the material used, and the procedure of assembly.

By changing the value of the weight coefficient, the value of each of these functions changes against the other in the multi-objective objective function. When α=1, the term related to the volume of material is cancelled out and the optimization will be exergetic. On the other hand, by taking α=0, the optimization will be merely volumetric, taking only the capital cost into account.

5.3 Different states of optimization (multi-objective)

5.3.1 Exergetic optimization (α=1)

By taking α=1, only the irreversibility term will remain. This means capital cost is negligible, which is usually not the case. Irreversibility in a heat exchanger results from either heat transfer through a finite temperature difference or frictional pressure drop. Although many researchers have used this method, the results of such an optimization may lead to unreasonable capital cost.

5.3.2 Optimization of the equivalent volume

By taking α=0, only the equivalent volume will be optimized. This means the cost of irreversibilities is negligible, which can in some, but not all cases, be true. While one tries to minimize the volume, pinch temperature difference tends to increase to a maximum that result in the minimization of the width of the heat exchanger, so an upper bound should be considered for this parameter. This happens to many parameters with different weight coefficients. Therefore, upper and lower bounds are chosen for all geometrical parameters. Such an optimization leads to minimization of the capital cost with a specified duty that is fixed by the upper bound of the pinch temperature difference.

5.3.2 Reference point approach

The two above optimizations lead to two extreme conditions that produce the reference point from which the multi-objective solution is the nearest point of the Pareto front [24]. Calculations imply that this point is the result of equal consideration of thermal and economical objective functions, α=0.5.

5.3.3 Thermoeconomic optimization

The total cost defined in Eq. (12) may be used as the thermoeconomic objective function. Therefore, by considering construction costs of an HRSG as proportional to its volume, one can determine coefficient α in Eq. (16) in a way that the resulting function equals the thermoeconomic objective function.

Using Eq. (12) and normalizing irreversibility and equivalent volume, the objective function is written as follows.

$$F=\frac{C_{tot}}{\beta}=\alpha(\frac{E_{irr}}{E_{\max}})+(1-\alpha)(\frac{V_{eq}}{V_{eq_{\max}}}) \qquad (20)$$

where α and β can be obtained using the Eqs .(21,22).

$$\alpha=\frac{C_e\times(\zeta\times E_{\max})\times H}{C_e\times(\zeta\times E_{\max})\times H+CRF\times BMF\times c\times\rho\times V_{eq_{\max}}} \qquad (21)$$

$$\beta=C_e\times(\zeta\times E_{\max})\times H+CRF\times BMF\times c\times\rho\times V_{eq_{\max}} \qquad (22)$$

Costs of irreversibility and equipment can be evaluated using the Eqs. (23, 24):

$$C_{irr}=\alpha\beta(\frac{E_{irr}}{E_{\max}}) \qquad (23)$$

$$C_{cc}=(1-\alpha)\beta(\frac{V_{eq}}{V_{eq_{\max}}}) \qquad (24)$$

6. Results

In order to analyse an HRSG, one should first determine the details of variables and parameters used in the simulation. The parameters of an HRSG are given in Table 1.

Table 1. Properties of the HRSG used in the simulation

Property	Value
Boiler input feed water temperature	110°C
Temperature of the generated steam	480°C
Pressure of the generated steam	10MPa
Temperature of the turbine exhaust gases	575°C
Flow rate of the turbine exhaust gases	30Kg / s
Pinch temperature difference ΔT_{pp}	5°C
Gas analysis	
CO_2	٪3
H_2O	٪7
N_2	٪75
O_2	٪15
Geometry property	
Tube external diameter	0.05 m
Tube internal diameter	0.0425 m
Arrangement type	In-line
Transverse pitch between tubes	0.1 m
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)
Arrangement	In-line

The results of optimization for different values of the weight coefficient create the Pareto curve. The curve presents irreversibility as a function of normalized volume. In this work, in order to make the results more sensible, exergetic efficiency as a function of normalized volume is presented (Fig. 5).

As stated before, the curve may be used as a decision making tool for designers. Eq. (13) may be used to estimate the capital cost of the boiler by knowing the volume of the heat transfer area. The coefficient of this equation depends on local expenses. Using Fig. 5, designers may estimate the capital cost of an optimized boiler with exergetic efficiency. As can be seen, increasing the capital cost does not considerably increase efficiency above 75%. Two ends of the curve are optimized designs, corresponding for choosing a volume and exergy loss as the objective function, respectively. As can be seen in the figure using these points, the unreachable ideal point is derived. The optimum multi-objective point is the nearest point on the curve to the ideal point. In this HRSG, the point is shown in Fig. 5. This figure shows that, by increasing the equivalent volume of the HRSG, it is possible to increase the exergetic efficiency up to a limit, which is 76.34%.

For calculating the total equipment cost, knowing the cost of heat transfer area is not enough. A coefficient should be used that represents lateral costs. The details of this coefficient are given in Table 2.

As mentioned earlier, for carrying out the optimization, upper and lower bounds should be placed on design parameters. These bounds have been chosen based on the suggested values in the literature and industrial documents which are given in Table 3.

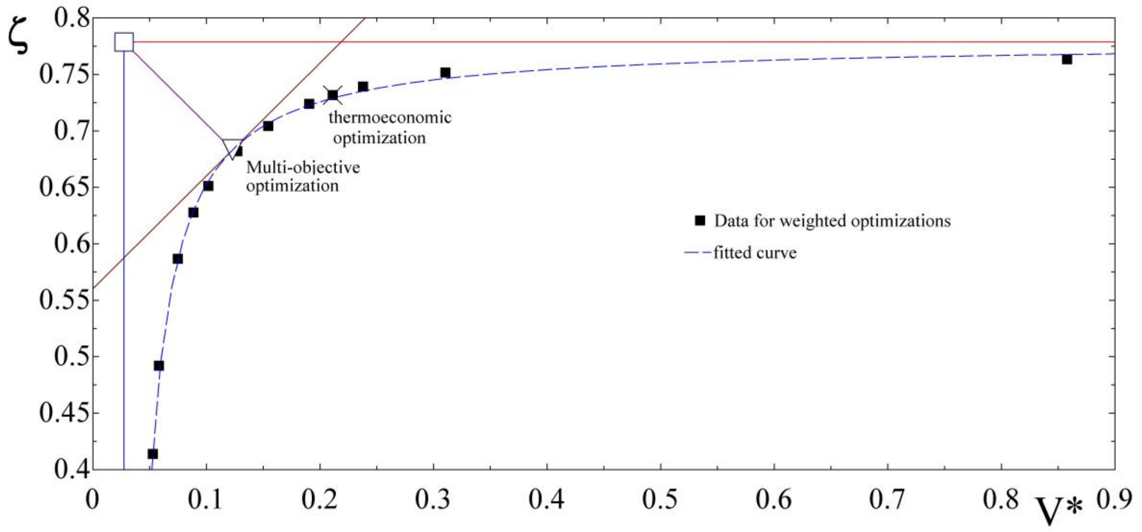

Fig. 5. Variations of normalized irreversibility and exergetic efficiency versus normalized equivalent volume

Table 2. Value of the BMF and its components

Item	Associated value for BMF
Tube volume	1
Casing and structure	0.205
Processing equipment	0.216
Piping and insulation	0.078
Control and instrumentation	0.098
Electrical panels and wiring	0.093
Engineering and supervision	0.031
Tax	0.075
Insurance	0.115
Profit of the project	0.095
Other costs	0.125
Total	2.31

Table 3. Upper and lower bounds of design parameters for multi-objective optimization of the HRSG

Parameters		Lower limit	Upper limit
General	W	0.4	4
	Pinch	-	30
Economizer	S_T	0.06	0.2
	S_L	0.06	0.2
	h_f	0.005	0.03
	t_f	0.0005	0.003
	n_f	2	8
	L	2	6
Evaporator	S_T	0.06	0.2
	S_L	0.06	0.2
	h_f	0.005	0.03
	t_f	0.0005	0.003
	n_f	2	8
Superheater	S_T	0.06	0.2
	S_L	0.06	0.2
	L	2	6

The value of the exergetic efficiency can be calculated using Eq. (19) and normalized irreversibility. Figure 5 shows variations of exergetic efficiency versus equivalent volume. A hyperbolic equation, Eq. 25, is suggested for which root square criterion is 0.9981. This signifies an accepted match between the data and the curve.

$$\zeta = \frac{-0.009131}{V^{*} - 0.02732} + 0.7786 \qquad (25)$$

This equation is applicable when the dimensionless volume is greater than zero, and implies that even if the dimensionless volume tends to infinity, the exergetic efficiency will not exceed 0.7786.

Using this diagram, a designer can choose exergetic efficiency and obtain its corresponding volume, and thus the cost of the HRSG and vice versa. Thermoeconomic calculation is carried out based on the economic values given in the following table. These values are obtained in Iran, in a specific case, and are given in Table 4.

Using the values given in Table 4, α and β can be obtained. The optimal values of design parameters with an annual cost of exergy loss/destruction and capital cost for exergetic, volumetric, thermoeconomic (α=0.749) optimizations, and a reference point approach resulting in equal consideration of both objective functions (α=0.5), are given in Table 5. Moreover, the value of β is obtained at 3.697 E 6. The zero value for capital cost means this optimization is precious only when the cost of manufacturing and assembling is negligible; on the other hand, zero irreversibility costs indicates the energy cost to be of no importance. The data resulted from these two extreme conditions, although may not be applicable, can be utilized to determine the reference point.

Table 4. Values considered in thermoeconomic calculations

Constant parameters	Values
Ce	9×10^{-5}
ζ	0.475
H	8000 hr
BMF	2.31
n	10 $year$
i	0.1
ρ	8054 kg/m^3

Table 5. Optimal values of different design parameters with different weighing coefficients

Parameters		$\alpha = 1$	$\alpha = 0.5$	$\alpha = 0.749$	$\alpha = 0$
General	W	4	1.114	1.787	0.4
	Pinch	0.007798	18.38	5.922	30
Economizer	S_T	0.2	0.06226	0.06197	0.06
	S_L	0.1443	0.09181	0.9258	0.2
	h_f	0.009731	0.005	0.005	0.005
	t_f	0.000585	0.0005	0.0005	0.0005
	n_f	8	8	8	8
	L	6	6	6	2
Evaporator	S_T	0.2	0.06219	0.06221	0.06
	S_L	0.1318	0.0902	0.09258	0.2
	h_f	0.1059	0.005	0.005307	0.005
	t_f	0.000721	0.0005	0.0005	0.0005
	n_f	8	8	8	8
Superheater	S_T	0.1695	0.07762	0.0766	0.06
	S_L	0.2	0.2	0.2	0.2
	L	6	3.526	3.663	2
C_{irr}		874710.2	588007.85	743412.2802	0
C_{cc}		0	234944.35	195733.5244	194979.8

7. Conclusion

In this paper a multi-objective optimization was performed in which two objective functions, exergetic efficiency and total capital cost, were chosen. Different geometric and operating parameters were taken into account. It has been demonstrated that the increase in exergetic efficiency in an interval of the equivalent volume is small enough to be neglected. For instance, increasing the normalized equivalent volume from 0.31 to 0.85 increases the efficiency only by 1.15%. A curve for the Pareto front is also fitted that simplifies the procedure of estimating the efficiency of the proposed HRSG with its specified equivalent volume. As noted before, some parameters, especially when optimization is the single objective, tend to stick to the upper or lower bounds of their intervals. Different values of α and β can be found by using local economical values, thus helping to make economic decisions.

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