ID: 2016-ISFT-362

Exergy Based Parametric Analysis of Dual Pressure HRSG in Gas/Steam Combined Cycle Power Plant

Meeta Sharma[1], Onkar Singh[2]

[1,2]*Amity University, Noida (U.P.), India; M. M. M. University of Technology, Gorakhpur (U.P.) India*
[1]*contactmeeta@yahoo.com*

Abstract: *This paper presents the exergy analysis of a dual pressure HRSG which generates steam at 60 bar and 6 bar as HP and LP steam generation in the gas/steam combined power plant. The study is based on the plant data taken from a gas/steam combined cycle power plant at Auraiya (U.P.), India. The major physical parameters of the dual pressure HRSG which are considered for analysis are tube diameter, fin thickness, fin height, tube pitch distance and fin density. The exergy analysis shows that in the dual pressure HRSG the exergy losses get minimized by up to 20 % by choosing correct fin height which also improvises the exergy efficiency by around 5-15%. The results obtained present the parametric analysis of HRSG from the exergy perspective.*

Keywords: *Heat recovery steam generator, combined cycle power plant, energy recovery, exergy loss, exergy efficiency*

1. INTRODUCTION

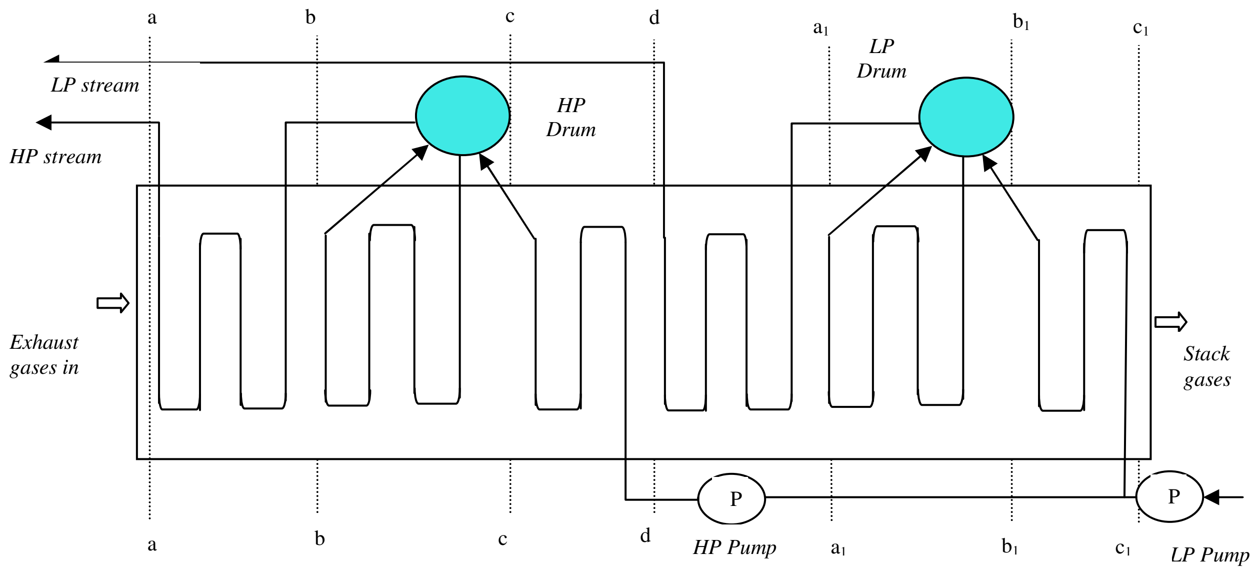
Present state of civilization has enormous dependence on energy leading to the growing demand of electricity and thus the world wide challenge of energy shortage. The energy scenario has further become grim due to the decreasing availability of fossil fuels and other conventional sources of energy.This energy crisis can be resolved by harnessing new sources of energy, effective utilization of present sources of energy and enhancing the efficiency of existing processes and equipments used for power generation. Presently, combined cycle power plants (CCPPs) are amongst the most preferred options mainly because of their capability to operate at flexible load conditions, high thermal efficiencies, less emissions in respect to power produced and lesser time for construction. Thus, the CCPPs have the potential of getting their efficiency improved through efficient utilization of fuels through use of waste heat of the exhaust gases in heat recovery steam generators (HRSG)for augmenting power output by steam turbine. In view of the scope of improving HRSG performance through identification of inefficiencies, there is need to have exergy based analysis which has the capability of analyzing the thermal systems through quantification of quality of energy and locating the areas of inefficiencies in them. Present paper deals with exergy based parametric analysis of dual pressure HRSG in gas/steam combined cycle power plant.

Some of the pioneering work done in this area is given ahead. Bejan [1, 2] described the basic techniques of exergy analysis, entropy generation minimization and also discussed the evaluation methods along with the ideas of irreversibility, entropy generation or exergy destruction. Reddy and Butcher [3] conducted the study of the performance of waste heat recovery power generation system based on second law analysis for various operating conditions and concluded that the first and second law efficiencies of HRSG for such system reduces at higher pinch point. Carapellucci and Giordano [4] described a method of comparison between exergetic and economic criteria for obtaining the heat recovery steam generator of gas-steam plants.Cornelissen and Hirs [5] illustrated the optimization of energy system design through combination of exergy analysis and life cycle analysis where there is a trade-off between energy saving during process and exergy utilized throughout the construction of energy system. Deng and Chia-Chin [6] discussed the engineering design and exergy analysis of a gas turbine combined cycle power plant based upon power generation system. Dincer and Rosan [7] described the exergy, energy and sustainability development for various systems. Escoa manual [8] gives empirical relation for finned tube boiler design. London and Shah [9] discussed the cost of irreversibilities in a heat exchanger. Moran and Sciubba [10] discussed the basic principle and practices based upon on exergy analysis. Nag and De [11] described the design and optimization of heat recovery steam generator with minimum irreversibility. Norouzi and Amidpour [12] discussed the optimal thermodynamic and economic volume of a heat recovery steam generator by constructal design. Reddy and Ramkiran [13] discussed the second law analysis of a waste heat recovery boiler. Rosan [14] also discussed the second law analysis approaches and implications for different systems.

This paper investigates the performance of the HRSG based upon its physical parameters such as fin height, fin thickness, fin density, and tube diameter, tube pitch distance. In the present study the HRSG chosen for the study has segmented fins arranged in staggered manner on its tube surfaces and undertakes steam generation at two different pressures and fixed superheated temperature conditions.

2. MATHEMATICAL MODELING FOR THE DUAL PRESSURE HRSG BASED EXERGY ANALYSIS

HRSG has three basic sections namely the super heater, the evaporator and the economizer and are considered as heat exchangers in a cross flow arrangement. Schematic diagram of the dual pressure HRSG for the HP and LP streams with its sections is shown in Fig. 1. This study considers the existing plant data of the NTPC's combined cycle plant at Auraiya (U.P.), India as baseline conditions. The thermodynamic parameters considered in the dual pressure HRSG have been detailed in Table 1.



Section (a-a to b-b) HP super heater
Section (b-b to c-c) HP evaporator
Section (c-c to d-d) HP economizer
Section (d-d to a_1-a_1) LP super heater
Section (a_1-a_1 to b_1-b_1) LP evaporator
Section (b_1-b_1 to c_1-c_1) LP economizer

Fig. 1. Schematic Diagram of Dual Pressure HRSG

The exergy loss and the exergy efficiency are functions of the enthalpy, entropy and exergy flow, along the different HP and LP streams, and are quantified using the governing equations detailed ahead. HRSG in consideration is of unfired type therefore effects of the chemical changes have been neglected and also the kinetic and potential energies have been ignored.

The enthalpy function h is given by equation (1). The enthalpies for HP and LP streams are evaluated as h_{HP} and h_{LP} at corresponding pressure and temperature conditions.

$$h = \int_{To}^{T} Cp(T)dT \tag{1}$$

The entropy function Φ is given by equation (2), entropy function for HP and LP streams also evaluated as Φ_{HP} and Φ_{LP} at corresponding pressure and temperature conditions.

$$\Phi = \int_{To}^{T} Cp(T)\frac{dT}{T} \tag{2}$$

The entropy flow, s is shown by equation (3), entropies for HP and LP streams also evaluated as s_{HP} and s_{LP} at corresponding pressure and temperature conditions.

$$s = \Phi - R.\ln(\frac{p}{po}) \tag{3}$$

The gas side pressure drop along the along length of HRSG is shown by equation (4) the pressure drop of HP and LP stream condition are evaluated as ΔP_{HP} and ΔP_{LP} for respective conditions. The effects of the tube side pressure losses have been neglected as they are much lesser in comparison to tube side pressure drop.

$$\Delta P = \frac{(f_o + A)G_o{}^2 N_r}{\rho_b \times 1.083 \times 10^9} \tag{4}$$

Exergy flow, Ω is given by equation (5), exergy flow for HP and LP streams as Ω_{HP} and Ω_{LP} evaluated under given pressure and temperature conditions.

$$\Omega = h - T_o s = h - T_o\left[\Phi - R\ln\left(\frac{p}{p_o}\right)\right] \quad (5)$$

Exergy in = (Exergy out in products + Exergy emitted with waste) + Exergy destruction (6)

By using above expressions for the exergy flow between two points in a section of the HRSG, the exergy loss can be quantified. The exergy loss comprises of the energy loss to the surroundings and exergy destruction associated with the internal irreversibilities. The exergy efficiency is expressed as a ratio of exergy output in product to exergy input. The exergy loss and exergy efficiency at the three sections of the HRSG for dual pressure HRSG are given ahead. As a-b, b-c and c-d are the HP sections; d-a1, a1-b1 and b1-c1 are the LP sections of the HRSG for the dual pressure steam conditions.

The exergy loss and the exergy efficiency for super heater, evaporator and economizer for HP and LP streams of HRSG are evaluated as:

Exergy loss ($\Delta\Omega_{sHP}$) and exergy efficiency (ψ_{sHP}) for the HP super heater,

$$\Delta\Omega_{sHP} = (\Omega_a - \Omega_b)_g - (\Omega_b - \Omega_a)_w \text{ and } \psi_s = (\Omega_b - \Omega_a)_w/(\Omega_a - \Omega_b)_g \quad (7)$$

Exergy loss ($\Delta\Omega_{evHP}$) and exergy efficiency (ψ_{evHP}) for the HP evaporator,

$$\Delta\Omega_{evHP} = (\Omega_b - \Omega_c)_g - (\Omega_c - \Omega_b)_w \text{ and } \psi_{evHP} = (\Omega_c - \Omega_b)_w/(\Omega_b - \Omega_c)_g \quad (8)$$

Exergy loss ($\Delta\Omega_{ecoHP}$) and exergy efficiency (ψ_{ecoHP}) for the HP economizer,

$$\Delta\Omega_{ecoHP} = (\Omega_c - \Omega_d)_g - (\Omega_d - \Omega_c)_w \text{ and } \psi_{ecoHP} = (\Omega_d - \Omega_c)_w/(\Omega_c - \Omega_d)_g \quad (9)$$

Exergy loss ($\Delta\Omega_{sLP}$) and exergy efficiency (ψ_{sLP}) for the LP super heater,

$$\Delta\Omega_{sLP} = (\Omega_d - \Omega_{a1})_g - (\Omega_{a1} - \Omega_d)_w \text{ and } \psi_{sLP} = (\Omega_{a1} - \Omega_d)_w/(\Omega_d - \Omega_{a1})_g \quad (10)$$

Exergy loss ($\Delta\Omega_{evLP}$) and exergy efficiency (ψ_{evLP}) for the LP evaporator,

$$\Delta\Omega_{evLP} = (\Omega_{a1} - \Omega_{b1})_g - (\Omega_{b1} - \Omega_{a1})_w \text{ and } \psi_{evLP} = (\Omega_{b1} - \Omega_{a1})_w/(\Omega_{a1} - \Omega_{b1})_g \quad (11)$$

Exergy loss ($\Delta\Omega_{ecoLP}$) and exergy efficiency (ψ_{ecoLP}) for the LP economizer,

$$\Delta\Omega_{ecoLP} = (\Omega_{b1} - \Omega_{c1})_g - (\Omega_{c1} - \Omega_{b1})_w \text{ and } \psi_{ecoLP} = (\Omega_{c1} - \Omega_{b1})_w/(\Omega_{b1} - \Omega_{c1})_g \quad (12)$$

The HRSG has been examined for the chosen input conditions by estimating the exergy loss and the exergy efficiency for varying physical parameters. The mathematical model consisting of governing equations detailed herein have been used for obtaining the results described ahead.

TABLE 1: Input Condition and Different Values Taken for Dual Pressure Analysis for the Combined Cycle Power Plant (NTPC, CCPP, Auraiya, U.P., India)

Input conditions	Values taken
Atmospheric Condition	298 K (25^{o}C) and 1 bar
Cycle Pressure Ratio	14:1
Turbine Inlet Temperature	1473 K or 1200^{o}C
Steam Superheat Temperature (High Pressure)	768 K or 495^{o}C
Gas Turbine Exhaust Pressure	1.2 bar
Condenser Pressure	0.05 bar
Polytropic Efficiency of Compressor and Gas Turbine	90%
Isentropic efficiency of steam turbine	90%
Mass Flow Rate of Exhaust Gases	400 kg/s
Temperature at Inlet of HRSG	800 K
Pinch Point Temperature Difference	10
Steam Pressure (High Pressure)	60 bar
Steam Pressure (Low Pressure), bar	6 bar
Steam Superheating Temperature (Low Pressure)	437 K or 164^{o}C
Feed water inlet temperature	315 K or 42^{o}C

3. RESULTS AND DISCUSSION

The results of the dual pressure exergy analysis are based on the input conditions given in Table 1 and Table 2.The above mentioned parameters and their considered range for simulation and their effects are analyzed and discussed as under.

TABLE 2: Physical Parameters Taken for Analysis

Description	Variants of HRSG
Diameter of tube (mm)	50.8
Thickness of the tube(mm)	3.05
Tube type	Segmented Spiraled fins
Fin density(no. of fin/ m)	236
Thickness of fin (mm)	1.25
Height of the fin (mm)	19
Tube bundles	Staggered
Longitudinal pitch (mm)	101.6
Transverse pitch (mm)	101.6

Fig. 2 shows the effect of the fin height on the exergy loss in the dual pressure HRSG. For the HP steam condition of the HRSG, the evaporator and the super heater sections have higher exergy loss than economizer whereas for the LP steam conditions, exergy losses are higher for super heater than evaporator and economizer for the fin height variations.

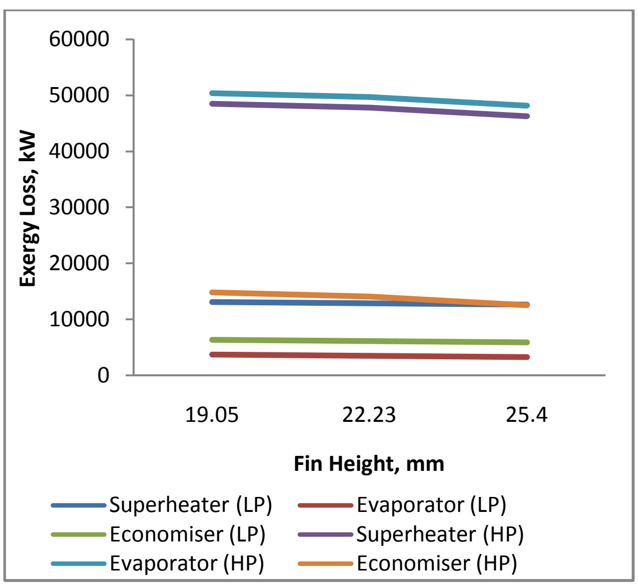


Fig. 2. Variationof Exergy loss for different Fin Heights in Dual Pressure HRSG

As the fin height increases from 19.05 mm to 25.4 mm the exergy loss decreases for both HP and LP steam conditions. It is observed that these losses decrease by 4.6% for super heater, 4.4% for evaporator and 15.15% for economizer section under HP steam of the HRSG and by 5.15% in super heater, 23.17% in evaporator and 15.76% in economizers for LP stream of the HRSG, for the above range of fin heights.

Fig. 3 shows the variations of exergy losses with different fin densities in the sections of dual pressure HRSG. This study shows that as the fin density in the HRSG sections increases, the exergy loss decreases. This tendency of decrease in exergy loss with fin density is similar in both HP and LP streams of the HRSG. The above graphical representation of fin density variation suggests that for HP steam condition, the exergy loss reduces by 1.3% in super heater, 1.2% in evaporator and 4.3% in economizer for varying fin densities from 79 to 236 fins/m. Whereas for similar range of fin density, under LP steams conditions of the HRSG, these losses are reduced by 1.9% in super heater, 6.37% in evaporator and 3.96% in economizer. This shows that with increase in fin density the exergy losses are further minimized in the dual pressure HRSG.

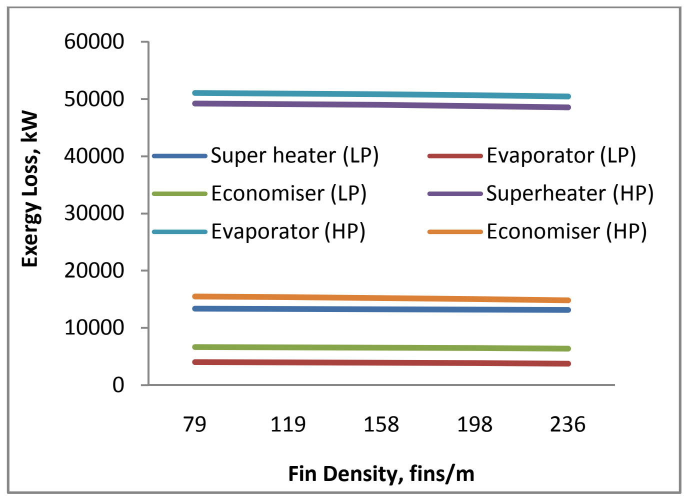


Fig. 3. Effect of Fin Densities on Exergy Loss in Dual Pressure HRSG

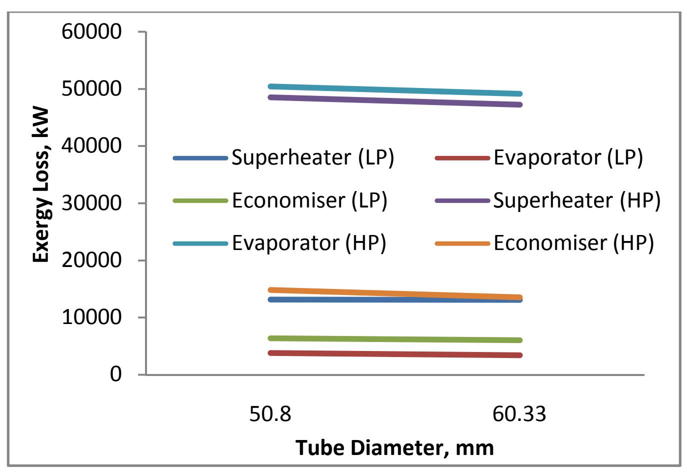


Fig. 4. Variation in Exergy Loss with Tube Diameters in Dual Pressure HRSG

Fig. 4 shows the variation of exergy loss with tube diameters for dual pressure HRSG in its sections. Upon considering various tube diameters for HP and LP streams. It is observed that the exergy loss decreases in both the streams when the tube diameters are increased from 50.8 mm to 60.33 mm. As observed there is decrease in exergy losses by 2.7 % in super heater section, 2.5 % in evaporator section and 8.6 % in economizer section when the tube diameter increases from 50.8 mm to 60.33 mm for HP stream of HRSG. Similarly, there is decrease in exergy losses by 0.22 % in super heater section, 9.65 % in evaporator section and 5.58 % in economizer section for the same variation of tube diameters for LP stream of HRSG. This decrease in exergy loss is small for larger surface areas because exhaust gas pressure drop is higher i.e. for outer tube diameter of 60.33 mm or above. It is thus observed that for dual pressures HRSG, by changing the tube outer diameters a considerable amount of exergy loss can be minimized in its sub-sections.

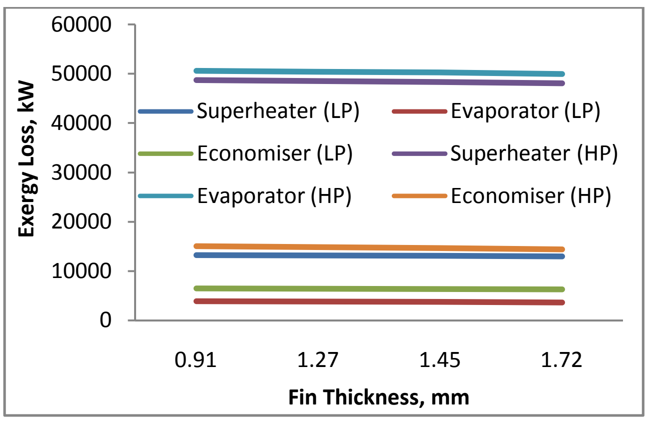


Fig. 5. Effect of Fin Thickness on Exergy Loss in Dual Pressure HRSG

Fig. 5 illustrates the variation of exergy losses with fin thickness for the dual pressure HRSG in its sub sections. At HP conditions of the HRSG, exergy loss is higher for the super heater and the evaporator than the economizer. However, the exergy losses are comparatively of lesser magnitude under LP conditions. The results obtained for LP conditions also show that the super heater has the maximum losses as compared to the evaporator and the economizer.

Upon considering increasing fin thicknesses from 0.91 mm to 1.72 mm for HP and LP steam conditions, it shows that the exergy loss gets decreased in both conditions. While there is decrease in exergy losses by 1.3 % in the super heater, the evaporator section and 4 % in the economizer section if the fin thickness increases from 0.91 mm to 1.72 mm for HP stream of the HRSG. The decrease in exergy losses is by 1.64 % in the super heater section, 5.73 % in the evaporator section and 3.37 % in the economizer section for LP stream of the HRSG. It is also observed that at dual pressures a sizeable amount of exergy loss can be reduced in the HRSG sub sections by changing fin thickness.

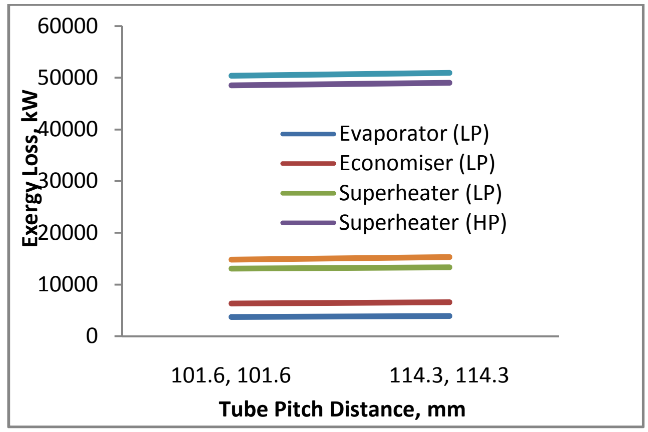


Fig. 6. Variation of Exergy Loss with Pitch Distance (Square) in Dual Pressure HRSG

Fig. 6 describes the variation of exergy loss with tube pitch distance in sub-sections of the dual pressure HRSG. Amongst different possible tube layouts usedin HRSG, the square finned tube arrangement has been chosen for both the streams of the HRSG. It is observed that for both HP and LP conditions there is an increase in exergy loss when the pitch distance is changed from 101.6 mm to 114.3 mm (S_l=101.6, S_t=101.6 to S_l=114.3, S_t=114.3). This exergy loss increases by 1.06 % for super heater, by 1.02 % for evaporator and by 3.4% for economizer for HP streams and by 5.94 % in super heater, by 23.35% in evaporator and by 12.80 % in economizer for LP streams. In LP stream of HRSG this increase in exergy loss is substantial due to higher pressure drop and a poor configuration.

As observed from Fig. 2 to Fig. 6, for the HP conditions of the HRSG, exergy losses are higher for super heater and evaporator than economizer. However, the exergy losses are comparatively of lesser magnitude under LP conditions. This is due to the fact that at high temperatures and pressures, the irreversibilities increase due higher entropy generation. The results obtained for LP conditions also show that the super heater has the maximum losses as compared to evaporator and economizer. At high pressure the major part of energy absorbed is in super heater as the latent heat absorbed by evaporator decreases. Due to high pressure conditions and fluid friction the generated frictional losses cause higher exergy losses for evaporator and super heater. Under LP conditions the fluid friction losses are comparatively very less, exergy losses for economizer and evaporator are less than that in super heater due to temperature irreversibility between exhaust gases and superheated condition of steam.

It is also evident from exergy analysis that for higher exergy losses, the exergy efficiency becomes lower for any thermodynamic system. By calculating the exergy efficiency for HRSG sub-sections at the various pressure levels an optimum design can be obtained which rejects lesser amount of waste heat to the surroundings with the minimum internal irreversibilites.

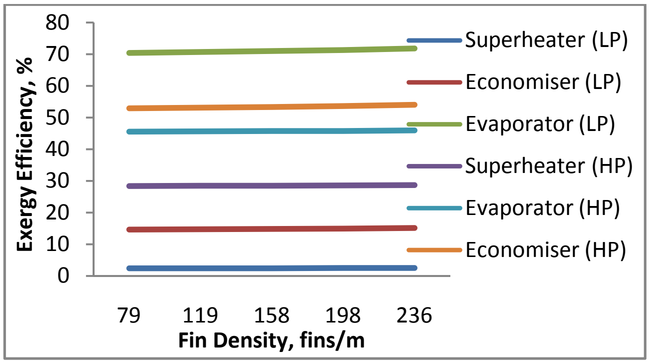


Fig. 7. Effect of Fin Densities on Exergy Efficiency in Dual Pressure HRSG

The Fig. 7 shows the variation of exergy efficiency for different fin densities in a dual pressure HRSG. An increase in fin density for HP and LP streams of HRSG modifies the exergy efficiency in its sub-sections. It is observed that for both HP and LP conditions, when the fin density increases from 79 to 236 fins/m, the exergy efficiency increases. This increase is by 1 % for super heater, by 0.7 % for evaporator and by 2.08 % for economizer at HP conditions and by 4.5% for super heater, by 1.9 % for evaporator and by 3.5 % for economizer for LP conditions.

In Fig. 8 variation of exergy efficiency for different fin thicknesses of dual pressure HRSG has been shown. It indicates that when the fin thickness increases from 0.91 mm to 1.72 mm, the exergy efficiency increases in both HP and LP streams of the HRSG. The HP stream has increase in exergy efficiency by 0.90 % in super heater, by 0.76 % in evaporator and by 2.14 % in economizer. In the similar trend LP stream has increased exergy efficiency by 4.8 % in super heater, by 1.65 % in evaporator and by 3.0 % in economizer. Although higher fin thickness shows rigid design but loss of gas pressure and material wastage limits this to optimal fin thickness.

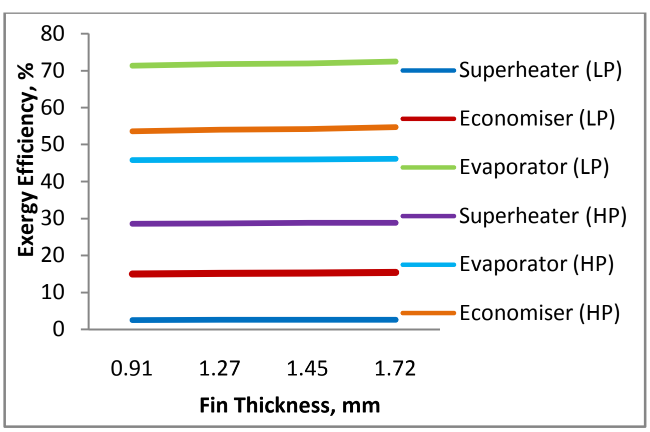


Fig. 8. Variation of Exergy Efficiency with Fin Thicknesses in Dual Pressure HRSG

Fig. 9 shows the variations of exergy efficiency for different fin heights in a dual pressure HRSG. The above graphical representation indicates that when the fin height increases from 19.05 mm to 31.75 mm, the exergy efficiency increases at both pressures conditions in its sub-sections. While HP stream shows increase in exergy efficiency by 3.69 % in super heater, by 2.41 % in evaporator and by 7.5 % in economizer, the LP stream shows increase in exergy efficiency by 5.1 % in super heater, by 7.0 % in evaporator and by 15.4 % in economizer. The HP and LP streams improve the exergy efficiency of HRSG evaporator by around 15% which enhances the overall energy recovery by HRSG. In a dual pressure HRSG, the fin height of more than 31.5 mm increases the complexities with high pressure drop.

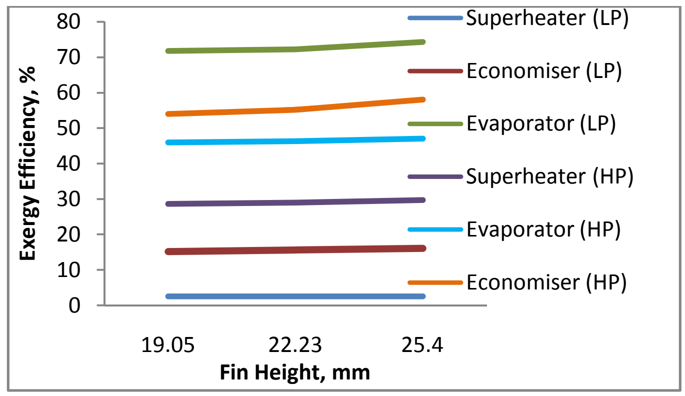


Fig. 9. Variation of Exergy Efficiency with Fin Heights in Dual Pressure HRSG

Fig. 10 shows the variation of exergy efficiency for different tube diameters in a dual pressure HRSG. The larger tube diameter shows enhanced exergy efficiency in HP and LP conditions of the HRSG. When tube diameter increases from 50.8 mm to 60.33 mm, HP steam shows that the exergy efficiency is enhanced by 2.38 % for super heater, by 1.4 % for evaporator and by 4.16 % by economizer. This increase in LP stream is by 0.4 % for super heater, by 2.8 % for evaporator and by 5 % in the economizer. Moreover, the tube diameters of larger than 60.33 mm increases the internal irreversibities which causes enormous gas pressure drop for above pressure conditions.

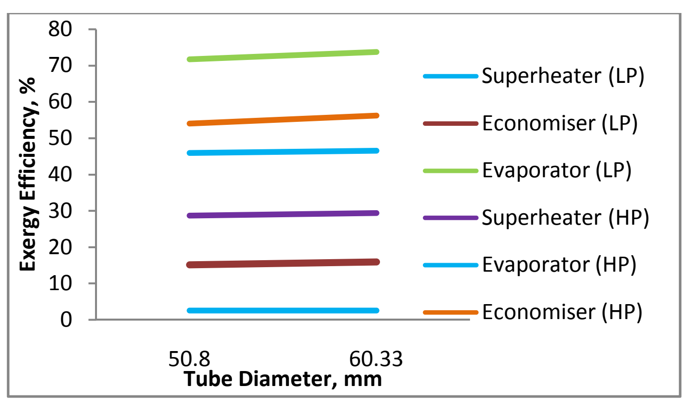


Fig. 10. Variation of Exergy Efficiency with Tube Diameters in Dual Pressure HRSG

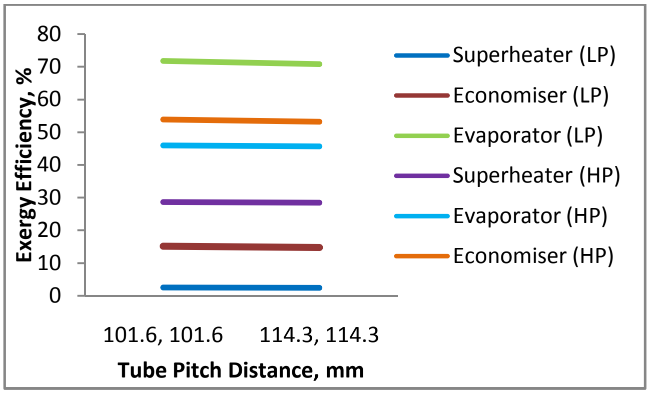


Fig. 11. Effect of Tube Pitch Distance (Square) on Exergy Efficiency in Dual Pressure HRSG

Fig. 11 shows the effects of variation in tube pitch distances (square) on exergy efficiency in a dual pressure HRSG. It is observed that with the increased tube pitch distances from 101.6 mm to 114.3 mm (Sl=101.6, St=101.6 to Sl=114.3, St=114.3), the exergy efficiency decreases for both HP and LP conditions of the HRSG. The HP steam shows that exergy efficiency is reduced by 0.73 % for super heater, by 0.54 % for evaporator and by 1.39 % for economizer. This reduction of exergy efficiency in LP stream is by 8.0 % for super heater, by 5.54 % for evaporator and by 10 % in the economizer.

4. CONCLUSIONS

Present study shows that the exergy analysis of dual pressure HRSG provides following important conclusions.

- Fin height can minimize the exergy losses in LP evaporator to the extent of 20 % and in HP economizer to the extent of 15 % for the chosen HP and LP streams. This reduction in exergy losses improves the exergy efficiency up-to 3-8 % for HP and 5-15 % for LP stream, which eventually increases the generation of steam in the HRSG.

- Outside tube diameter also reduces exergy losses nearly up-to 5-8 % in HP and LP economizer and increases exergy efficiency of HP and LP economizer by around 4-5 %.

- In dual pressure HRSG the variation of fin density can reduce exergy losses from 5-6% in LP evaporator and up-to 4 % in HP economizer which can improve exergy efficiency of dual pressure HRSG up-to 2-5 %.

- Fin thickness also minimizes exergy losses marginally up-to 4% in HP and LP evaporator with little improvement in exergy efficiency of the HRSG. The tube pitch distance increases the exergy loss marginally in its sections under HP conditions but this loss increases 5–20 % under LP condition which is substantial and reduces the exergy efficiency by 6-10%.

Thus appropriate selection of above physical parameters can help in estimating the optimum parametric conditions of HRSG for maximizing the recovery of heat, improvement in steam generation and overall combined cycle performance.

REFERENCES

[1] Bejan, A. Entropy generation through heat and fluid flow. Wiley, New York, 1982.

[2] Bejan, A. Thermodynamic design of heat and mass transfer processes and devices. Heat and Fluid Flow, Butterworth, 1987, 8 (4), 258-278.

[3] Butcher, C. J.; Reddy, B. V. Second law analysis of heat recovery based on power generation system. International Journal of Heat and Mass Transfer, 2007, 50, 2355-2363.

[4] Carapellucci, R.; Giordano, L. A comparison between exergetic and economic criteria for obtaining the heat recovery steam generator of gas-steam plants. International Journal of Energy, 2013, 58, 458-472.

[5] Cornelissen, R. L.; Hirs, G. G.; Thermodynamic optimization of heat exchanger. International Journal of Heat and Mass Transfer, 1999, 42, 951-959.

[6] Deng, C. S.; Chia-Chin, C. Engineering design and exergy analysis of a gas turbine combined cycle power plant based upon power generation system. International Journal of Energy, 2004, 29, 1182-205.

[7] Dincer, I.; Rosan, M. A. Exergy: energy environment sustainable development, Elsevier.

[8] Escoa finned tube manual, Escoa Corporation, Tulsa.2007,

[9] London, A. L.; Shah, R. K. Costs of the irreversibility's in heat exchanger design, International Journal of Heat Transfer Engineering, 1983, 4(2), 59-73.

[10] Moran, M. J.; Sciubba, E. Exergy analysis-Principles and Practice. ASME Journal of Gas Turbines and Power, 1994, 116, 285-290.

[11] Nag, P. K.; De, S. Design and operation of heat recovery steam generator with minimum irreversibility. International Journal of Applied Thermal Engineering, 1996, 17(4), 385-391.

[12] Norouzi, E.; Amidpour. M. Optimal thermodynamic and economic volume of a heat recovery steam generator by constructional design. International Communication in Heat and Mass Transfer, 2012, 39, 1286-1292.

[13] Reddy, B. V.; Ramkiran, G. Second law analysis of a waste heat recovery boiler. International Journal of Heat and Mass Transfer, 2002, 45, 1807-1814.

[14] Rosan, M. A. Second law analysis approaches and implications. International Journal of Energy Resources, 1998, 23, 415-429.