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International Journal of Heat and Mass Transfer 45 (2002) 1807–1814

International Journal of HEAT and MASS TRANSFER

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# Second law analysis of a waste heat recovery steam generator

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Received 21 November 2000; received in revised form 11 September 2001

## Abstract

In recent years a great deal of attention is focussed on the efficient utilization of energy resources with minimum heat loss. There is a growing interest on second law analysis to minimize the entropy generation in various thermal units and thereby to improve and optimize the design and performance. In the present work, a waste heat recovery steam generator is considered, which consists of an economizer, an evaporator and a super heater. The unit produces superheated steam by absorbing heat from the hot flue gases. A general equation for the entropy generation has been proposed, which incorporates all the irreversibilities associated with the process. By using suitable non-dimensional operating parameters, an equation for entropy generation number is derived. The effect of various non-dimensional operating parameters, on the entropy generation number are investigated. The role of gas specific heat, non-dimensional inlet gas temperature difference ratio ($\tau$), heat exchanger unit sizes (NTUB, NTUS, NTUE) on entropy generation number are also reported. The results will help to understand the influence of different non-dimensional operating parameters on entropy generation number, which in turn will be useful to optimize the performance of the unit. 

## 1. Introduction

The persistent need to conserve the fast depleting energy resources, and to use them in a more efficient manner has renewed the interest in devices that can utilize heat from waste flue gases from various industries. The flue gases on the virtue of being at a higher temperature relative to the surroundings and having a higher mass flow rate, posses considerable amount of available energy, which if not utilized properly will lead to huge undesirable energy loss. During the last two decades there has been considerable attention, on to utilize heat from flue gases for various applications and to optimize the units which are used to absorb heat from waste flue gases.

Bejan [1] gave a comprehensive review of second law analysis of heat and mass transfer processes and devices, the overall objective being to determine the optimum conditions under a variety of situations in which the system destroys the least amount of energy, while performing the fundamental engineering function. London and Shaw [2] discussed the costs of irreversibilities associated with both fluid flow and heat transfer in heat exchanger design. A second law analysis of the optimum design and operation of thermal energy storage system was given by Krane [3].

It has been shown by Bejan [4] that irreversibility can be reduced by bringing the inlet temperature of the stream closer to the temperature of the liquid bath and by keeping the exhaust gas temperature as low as possible. This can be done by a series of storage units. Second law optimization of a sensible heat thermal energy storage system has been studied in greater details by Taylor et al. [5].

Nag and Mukherjee [6] made a thermodynamic optimization of convective heat transfer in an isothermal duct. Nag and Naresh Kumar [7] studied the same for a duct with constant heat flux. Entropy generation in thermal radiation was discussed by Arpaci [8]. San et al. [9] evaluated entropy generation, so as to minimize

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0017-9310/02/$ - see front matter 
PII: S0017-9310(01)00293-9

**Nomenclature**

| | |
|---|---|
| $A_O$ | surface area for heat transfer ($m^2$) |
| $A_{OB}$ | surface area for heat transfer of the evaporator ($m^2$) |
| $c_1$ | specific heat of water (kJ $kg^{-1}$ $K^{-1}$) |
| $c_2$ | specific heat of steam (kJ $kg^{-1}$ $K^{-1}$) |
| $c_{pg}$ | specific heat of hot flue gas (kJ $kg^{-1}$ $K^{-1}$) |
| $f$ | friction factor |
| $h_o$ | external convection heat transfer coefficient (W $m^{-2}$ $K^{-1}$) |
| $h_{Sat}$ | enthalpy of saturated steam (kJ $kg^{-1}$) |
| $h_{Sup}$ | enthalpy of superheated steam (kJ $kg^{-1}$) |
| $h_{WO}$ | enthalpy of water at the entry of economizer (kJ $kg^{-1}$) |
| $h_{fg}$, $L$ | latent heat of vaporization (kJ $kg^{-1}$) |
| $l_S$ | length of superheater heat exchanger (m) |
| $l_B$ | length of evaporator (boiler) heat exchanger (m) |
| $l_E$ | length of economizer heat exchanger (m) |
| $LMTD_B$ | logarithmic mean temperature difference for the evaporator |
| $M_W$ | mass flow rate of water (kg $s^{-1}$) |
| $m_g$ | mass flow rate of flue gas (kg $s^{-1}$) |
| NTUE | number of transfer units of economizer |
| NTUB | number of transfer units of boiler (evaporator) |
| NTUS | number of transfer units of superheater |
| $N_S$ | entropy generation number |
| $P_o$ | ambient pressure (kPa) |
| $P_B$ | perimeter of evaporator or boiler heat exchanger (m) |
| $P_E$ | perimeter of economizer heat exchanger (m) |
| $P_S$ | perimeter of superheater heat exchanger (m) |
| $r$ | radius of gas tube (m) |
| $St$ | Stanton number |
| $S_{gen}$ | rate of entropy generation (kW $K^{-1}$) |
| $T_g$ | temperature of flue gas at the considered location (K) |
| $T_O$ | ambient temperature (K) |
| $T_O$ | temperature of water at the inlet of economizer (K) |
| $T_\alpha$ | flue gas temperature at the inlet of superheater (K) |
| $T_{Out}$ | flue gas temperature at the outlet of economizer (K) |
| $T_X$ | flue gas temperature at the exit of superheater (K) |
| $T_Y$ | flue gas temperature at the exit of evaporator (K) |
| $T_{Sat}$ | saturation temperature of water/steam (K) |
| $T_{Sup}$ | temperature of superheated steam (K) |
| $T_W$ | temperature of water at the considered location (K) |
| $U_{OB}$ | overall heat transfer coefficient for the evaporator (W $m^{-2}$ $K^{-1}$) |
| $U_O$ | overall heat transfer coefficient of combined the economizer, evaporator and superheater (W $m^{-2}$ $K^{-1}$) |
| $V$ | flue gas velocity (m $s^{-1}$) |
| $V^*$ | non-dimensional gas velocity |
| $\Delta P_g$ | pressure drop (kPa) |
| $\tau$ | non-dimensional hot flue gas inlet temperature difference ratio |
| $\tau_s$ | non-dimensional water saturation temperature difference ratio |
| $\tau_h$ | non-dimensional superheated steam temperature difference ratio |

irreversibility as a combined heat and mass transfer process.

Nag and Mazumder [10] reported the thermodynamic optimization of a waste heat recovery boiler with an economizer and evaporator and producing saturated steam. They have reported the effect of different operating parameters on entropy generation for the waste heat recovery boiler. Rosen [11] discussed second law analysis, the approaches and implications. He has presented a comparison of the main approaches to second law analysis reported in the literature. Garimella and Garimella [12] investigated the utilization of waste heat from commercial process steam boiler for air conditioning using absorption cooling systems. An ammonia-water generator absorber heat exchanger system was developed and modeled to use waste heat from the boiler flue gases and to deliver chilled water.

Zubair and Al-Naglah [13] presented an analytical model for the second law based thermo-economic analysis and optimization of a sensible heat storage system, in which the storage element is both heated and cooled by flowing streams of gases. They have also studied the influence of important unit cost parameters on $NTU_{s,opt}$ and $\theta_{S,opt}$. Saboya and da Costa [14] applied from second law of thermodynamics, the concepts of irreversibility, entropy generation, and availability to counter flow, parallel flow and cross flow heat exchangers. In the case of cross flow configuration they have considered four types of heat exchangers: (1) both fluids unmixed, (2) both fluids mixed, (3) fluid of maximum heat capacity rate mixed and the other unmixed, (4) fluid of minimum heat capacity rate mixed and the other unmixed. The counter flow heat exchanger is compared with the other five types of heat exchangers, to know the one with minimum irreversibility. Juan and Bejan [15] reported on the entropy minimization in parallel-plates counterflow heat exchangers.

Juan and Bejan [15] have presented the conditions for minimum irreversibilty and for minimum entropy generation. Entropy generated and the exergy destroyed in lithium bromide thermal compressors driven by the exhaust gases of an engine was discussed by Izquierdo et al. [16].

In many situations hot flue gases are used to produce superheated steam. The steam thus produced can be used for electric power generation or for process heat purpose. In the literature not much is reported on the second law analysis of a such unit. Reddy et al. [17] reported some preliminary results on the effect of various operating parameters on entropy generation for a waste heat recovery boiler producing superheated steam. In the present work, a waste heat recovery boiler with an economizer, an evaporator and a superheater is considered. The unit absorbs heat from hot flue gases and produces super heated steam. For this unit general equations for entropy generation and entropy generation number are proposed. The effect of different non-dimensional operating parameters on entropy generation number are investigated.

## 2. Formulation of the problem

The unit with an economizer, an evaporator and a superheater as shown in Fig. 1 is considered for the analysis and the same is presented on $T-S$ diagram (Fig. 2). The direction of hot flue gas and water/steam in the waste heat recovery steam generator is shown in Fig. 1. The counter flow arrangement is considered for the analysis. The temperature profiles and pressure conditions at the entry and exit of the economizer, evaporator and superheater are shown in Fig. 2.

The following assumptions are made in the formulation:

The system is in steady state.

The pressure drop in the water steam line is neglected.

There is no extraneous heat loss

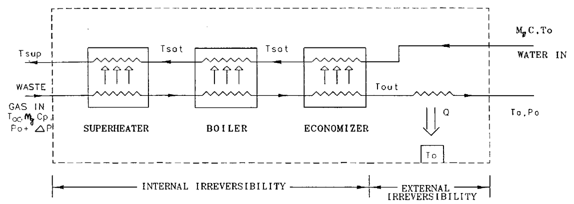


Fig. 1. Thermodynamic model of a waste heat recovery steam generator.

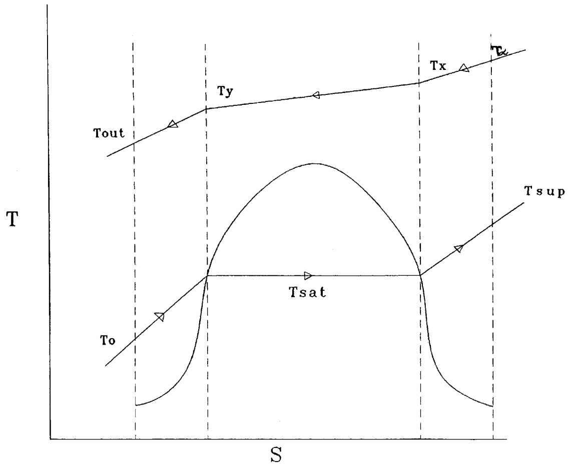


Fig. 2. $T-S$ diagram of waste heat recovery boiler.

The actual specific heat of the flue gas and the flue gas composition are taken into account in the analysis.

The pressure drop in the flue gas line is considered.

### 2.1. Analysis

The energy balance for the waste heat recovery unit gives

$$m_g c_{pg}(T_\alpha - T_{Out}) = M_W[(h_{Sat} h_{WO}) + h_{fg} + (h_{Sup} - h_{Sat})]. \quad (1)$$

The latent heat of vaporization $h_{fg}$ is denoted as $L$, in the analysis.

In terms of specific heat, Eq. (1) can be written as

$$m_g c_{pg}(T_\alpha - T_{Out}) = M_W[c_1(T_{Sat} - T_O) + L + c_2(T_{Sup} - T_{Sat})], \quad (2)$$

where $c_1$ is specific heat of water and $c_2$ is the specific heat of steam (approximated).

There are six kinds of irreversibilities associated with heat transfer in the unit. These are quantified by the entropy generation terms as given below:

$$S_{gen} = m_g c_{pg} \ln\frac{T_O}{T_\alpha} + m_g R \ln\frac{(P_O + \Delta P_g)}{P_O} + m_g c_{pg}\frac{(T_{Out} - T_O)}{T_O} + M_W c_1 \ln\frac{T_{Sat}}{T_O} + \frac{M_W L}{T_{Sat}} + M_W c_2 \ln\frac{T_{Sup}}{T_{Sat}}, \quad (3)$$

where the first two terms represent the rate of entropy change experienced by the hot flue gas stream from waste heat boiler inlet to exit. The third term accounts for the entropy generation due to heat transfer from the gas leaving the economizer to the environment at temperature $T_O$. The other terms account for the entropy change for water and steam from inlet to outlet due to sensible heating of water in the economizer, boiling of water in the evaporator and superheating of steam in the superheater.

Dividing Eq. (3) with $m_g c_{pg}$

$$\begin{aligned}\frac{S_{gen}}{m_g c_{pg}} &= N_S \\ &= \ln\frac{T_O}{T_\alpha} + \frac{R}{c_{pg}}\ln\frac{(P_O+\Delta P_g)}{P_O} + \frac{(T_{Out}-T_O)}{T_O} \\ &\quad + \frac{M_W c_1}{m_g c_{pg}}\ln\frac{T_{Sat}}{T_O} + \frac{M_W L}{m_g c_{pg}}\frac{}{T_{Sat}} \\ &\quad + \frac{M_W c_2}{m_g c_{pg}}\ln\frac{T_{Sup}}{T_{Sat}}.\end{aligned} \tag{4}$$

The following non-dimensional parameters are defined.

The non-dimensional hot flue gas inlet temperature difference ratio ($\tau$), between the inlet and ambient temperatures of the hot flue gas is defined as

$$\tau = \frac{T_\alpha - T_O}{T_O}. \tag{5}$$

The non-dimensional water saturation temperature difference ratio ($\tau_s$), between water saturation temperature and water inlet temperature to the economizer as

$$\tau_S = \frac{T_{Sat} - T_O}{T_O}. \tag{6}$$

The non-dimensional superheated steam temperature difference ratio ($\tau_h$), between superheated temperature of steam and the water entry temperature to the economizer as

$$\tau_h = \frac{T_{Sup} - T_O}{T_O}. \tag{7}$$

Let the ratio of heat capacities of water and gas stream be defined as

$$X_1 = \frac{M_W c_1}{m_g c_{pg}}. \tag{8}$$

Let the ratio of heat capacities of steam and gas stream be defined as

$$X_2 = \frac{M_W c_2}{m_g c_{pg}}. \tag{9}$$

The non-dimensional entropy generation number is defined as

$$N_S = \frac{S_{gen}}{m_g c_{pg}}. \tag{10}$$

From Eq. (1), the temperature of the flue gas leaving the economizer $T_{Out}$ can be written as

$$T_{Out} = T_\alpha - \frac{M_W}{m_g c_{pg}}\left[c_1(T_{Sat} - T_O) + L + c_2(T_{Sup} - T_{Sat})\right]. \tag{11}$$

### 2.2. Superheater

The energy balance across the superheater can be written as

$$M_W(h_{Sup} - h_{Sat}) = m_g c_{pg}(T_\alpha - T_X) \tag{12}$$

or

$$M_W c_2(T_{Sup} - T_{Sat}) = m_g c_{pg}(T_\alpha - T_X). \tag{13}$$

$T_X$ can be simplified as

$$T_X = T_O[1 + \tau - X_2(\tau_h - \tau_s)]. \tag{14}$$

By considering a small elemental length '$dx$' for the superheater and by making energy balance for the element and integrating between the limits $T_X$ and $T_\alpha$ and by further simplification, the temperature of the flue gas at the exit of superheater, $T_X$ can be expressed, in terms of other operating parameters as

$$\begin{aligned}T_X = {} & (T_O X_2(\tau - \tau_h)\exp[\mathrm{NTUS}(1 - X_2)] \\ & + T_O X_2(1 + \tau_h) - T_O(1 + \tau))/X_2 - 1.\end{aligned} \tag{15}$$

### 2.3. Evaporator

The energy balance across the evaporator can be written as

$$M_W L = m_g c_{pg}(T_X - T_Y). \tag{16}$$

$T_Y$ can be expressed as

$$T_Y = T_X - \frac{M_W L}{m_g c_{pg}}. \tag{17}$$

From Eq. (14)

$$T_X = T_O[1 + \tau - X_2(\tau_h - \tau_s)]. \tag{18}$$

$T_Y$ can be expressed as

$$T_Y = T_O[1 + \tau - X_2(\tau_h - \tau_s)] - \frac{L X_1}{c_1}. \tag{19}$$

By making an energy balance for an elemental length $dx$ of the evaporator and by integrating between the limits $T_Y$ and $T_X$ and by further simplification, $T_Y$ can be written in terms of other operating parameters as

$$\begin{aligned}T_Y = {} & \frac{T_O X_2(\tau - \tau_h)}{X_2 - 1}\exp[\mathrm{NTUS}(1 - X_2)\exp(-\mathrm{NTUB})] \\ & + \frac{\exp(-\mathrm{NTUB})}{X_2 - 1}[T_O X_2(1 + \tau_h) - T_O(1 + \tau)] \\ & + T_{Sat}[1 - \exp(-\mathrm{NTUB})].\end{aligned} \tag{20}$$

The latent heat of vaporization $L$, can be expressed as

$$L = \frac{U_{OB}A_{OB}(\mathrm{LMTD})_B}{M_W}. \tag{21}$$

$L$ is simplified as

$$L = \frac{c_1 T_O[1-\exp(-\mathrm{NTUB})]}{X_1(X_2-1)}[X_2(\tau-\tau_h) \times \exp\mathrm{NTUS}(1-X_2) + X_2(1+\tau_h) - (1+\tau) - (1+\tau_s)(X_2-1)]. \tag{22}$$

### 2.4. Economizer

Energy balance for the entire waste heat recovery unit can be written as

$$m_g c_{pg}(T_\alpha - T_{Out}) = M_W[c_1(T_{Sat}-T_O) + L + c_2(T_{Sup}-T_{Sat})]. \tag{23}$$

The temperature of the flue gas at the exit of economizer $T_{Out}$ can be written as

$$T_{Out} = T_O(1+\tau) - X_1 T_O \tau_s - \frac{X_1 L}{c_1} - X_2 T_O(\tau_h - \tau_s). \tag{24}$$

By making the energy balance for an elemental length $dx$ for the economizer and by further simplification $T_{Out}$ is obtained in terms of $\tau$, $\tau_h, \tau_s, X_1, X_2$, NTUE, NTUB, NTUS.

By utilizing the above derived relations, the equation for entropy generation number ($N_S$) in terms of $\tau, \tau_h, \tau_s, X_1, X_2$, NTUE, NTUB, NTUS can be expressed as

$$\begin{aligned} N_S = &-\ln(1+\tau) + \frac{R}{c_{pg}}\frac{\ln(P_O+\Delta P_g)}{P_O} \\ &+ \frac{[1-\exp(-\mathrm{NTUE}(1+X_1))](X_1+1)}{X_1} \\ &+ X_2(\tau-\tau_h)\exp[\mathrm{NTUS}(1-X_2)]\exp(-\mathrm{NTUB}) \\ &\times\left[\frac{\frac{X_1}{X_1+1}\exp[-\mathrm{NTUE}(1+X_1)] + \frac{1}{X_1+1}}{X_2-1}\right] \\ &+ [X_2(1+\tau_h)-(1+\tau)]\exp(-\mathrm{NTUB}) \\ &\times\left[\frac{\frac{X_1}{X_1+1}\exp[-\mathrm{NTUE}(1+X_1)]X_1 + \frac{1}{X_1+1}}{X_2-1}\right] \\ &+ (1+\tau_s)[1-\exp(-\mathrm{NTUB})] \\ &\times\left[\frac{X_1}{X_1+1}[\exp(-\mathrm{NTUE}(1+X_1))]X_1 + \frac{1}{X_1+1}\right] \\ &- 1 + X_1\ln(1+\tau_s) + \frac{[1-\exp(-\mathrm{NTUB})]}{(X_2-1)(1+\tau_s)} \\ &\times [X_2(\tau-\tau_h)\exp[\mathrm{NTUS}(1-X_2)] + X_2(1+\tau_h) \\ &- (1+\tau) - (1+\tau_s)(X_2-1)] + X_2\ln\frac{(1+\tau_h)}{(1+\tau_s)}. \end{aligned} \tag{25}$$

The pressure drop of the gas, may be expressed by the relation

$$\frac{\Delta P_g}{P_O} = \frac{f(l_E+l_B+l_S)\rho V^2}{P_O r} = \frac{f(\mathrm{NTUE}+\mathrm{NTUB}+\mathrm{NTUS})\rho V^2 h_O}{2St P_O U_O}. \tag{26}$$

Let the non-dimensional gas velocity be defined as

$$V^{*2} = \frac{V^2}{[(2St\,U_O P_O c_{pg})/(f h_O \rho R)]}, \tag{27}$$

where $U_O$ is the average overall heat transfer coefficient for the economizer, evaporator and superheater together. For low gas velocity $V$, $\Delta P_g \ll P_O$ and therefore

$$\begin{aligned} \frac{R}{c_{pg}}\ln\left(1+\frac{\Delta P_g}{P_O}\right) &= \frac{R\Delta P_g}{c_{pg}P_O} \\ &= \frac{R}{c_{pg}}\frac{f(\mathrm{NTUE}+\mathrm{NTUB}+\mathrm{NTUS})\rho V^2 h_O}{2St P_O U_O} \\ &= (\mathrm{NTUE}+\mathrm{NTUB}+\mathrm{NTUS})V^{*2}. \end{aligned} \tag{28}$$

Finally, the entropy generation number for the waste heat recovery steam generator, $N_S$ in terms of non-dimensional gas velocity and other non-dimensional operating parameters can be expressed as

$$\begin{aligned} N_S = &-\ln(1+\tau) + (\mathrm{NTUE}+\mathrm{NTUB}+\mathrm{NTUS})V^{*2} \\ &+ \frac{[1-\exp(-\mathrm{NTUE}(1+X_1))](X_1+1)}{X_1} \\ &+ X_2(\tau-\tau_h)\exp[\mathrm{NTUS}(1-X_2)]\exp(-\mathrm{NTUB}) \\ &\times\left[\frac{\frac{X_1}{X_1+1}\exp[-\mathrm{NTUE}(1+X_1)] + \frac{1}{X_1+1}}{X_2-1}\right] \\ &+ [X_2(1+\tau_h)-(1+\tau)]\exp(-\mathrm{NTUB}) \\ &\times\left[\frac{\frac{X_1}{X_1+1}\exp[-\mathrm{NTUE}(1+X_1)]X_1 + \frac{1}{X_1+1}}{X_2-1}\right] \\ &+ (1+\tau_s)[1-\exp(-\mathrm{NTUB})] \\ &\times\left[\frac{X_1}{X_1+1}[\exp(-\mathrm{NTUE}(1+X_1))]X_1 + \frac{1}{X_1+1}\right] \\ &- 1 + X_1\ln(1+\tau_s) + \frac{[1-\exp(-\mathrm{NTUB})]}{(X_2-1)(1+\tau_s)} \\ &\times [X_2(\tau-\tau_h)\exp[\mathrm{NTUS}(1-X_2)] + X_2(1+\tau_h) \\ &- (1+\tau) - (1+\tau_s)(X_2-1)] + X_2\ln\frac{(1+\tau_h)}{(1+\tau_s)}, \end{aligned} \tag{29}$$

$$N_S = f(\tau, \tau_s, \tau_h, \mathrm{NTUE}, \mathrm{NTUB}, \mathrm{NTUS}). \tag{30}$$

Eqs. (25) and (29) represent the dependence of entropy generation number, $N_S$ on various other non-dimensional operating parameters. The equations will help to estimate $N_S$ for given hot flue gas and water/steam flow conditions. The equations will also be useful to find the optimum conditions and heat exchanger sizes at which the entropy generation is minimum for the given gas flow rate and water/steam conditions.

## 3. Results and discussion

The $N_S$ equation will give an idea on the role of various operating parameters on the entropy generation rate. In the present work an attempt is made to present the effect of certain non-dimensional operating parameters on the entropy generation number. The results may give an idea on the role of non-dimensional operating parameters on $N_S$ for the waste heat recovery steam generator. Optimum values of the design parameters like NTUE, NTUB, NTUS for a given $\tau, \tau_s, \tau_h$ can be estimated by utilizing Eqs. (25) and (29) for the given gas flow rate and water/steam conditions. These could also be obtained by plotting $N_S$ against the relevant operating parameters. The effect of certain non-dimensional operating parameters on the $N_S$ are demonstrated in Figs. 3–8, respectively. The present results are basic results, which will give an idea on the role of various parameters on entropy generation number. The different set of parameter values are chosen to investigate their effect on $N_S$. The main aim of the present work is to investigate the $N_S$ variation trend with different operating parameters. This will give an idea on the role of operating parameters on $N_S$ and their optimization trend.

The $N_S$ variation with NTUB for two different non-dimensional hot flue gas inlet temperature difference ratios ($\tau$) is presented in Fig. 3. For a particular $\tau$ value, as the NTUB increases, the $N_S$ decreases for other fixed set of non-dimensional parameters. This is due to more heat absorption from the flue gas, which may result in reduced irreversibility and heat loss to the ambient. The trend is same for two different non-dimensional hot flue gas inlet temperature difference ratios. However, for lower value of $\tau$, the decrease in $N_S$ with NTUB is faster than that for the higher value. This could be probably due to the reason that, for a designed NTUB value, the temperature difference between flue gas and water/steam ($\Delta T$) may be low. This results in reduced irreversibility and entropy generation. Also the heat loss to the ambient may be less, resulting in reduced external

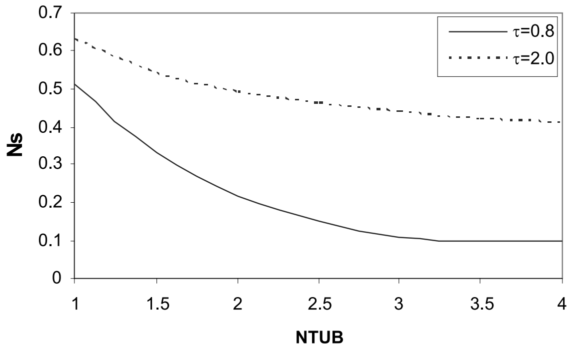


Fig. 3. $N_S$ variation with NTUB for different values of $\tau$, $\tau_s = 0.25$, $\tau_h = 0.35$, $X_1 = 0.35$, $X_2 = 0.15$, NTUE$\,=0.7$, NTUS$\,=0.4$, $\Delta P/P_O = 0.1$.

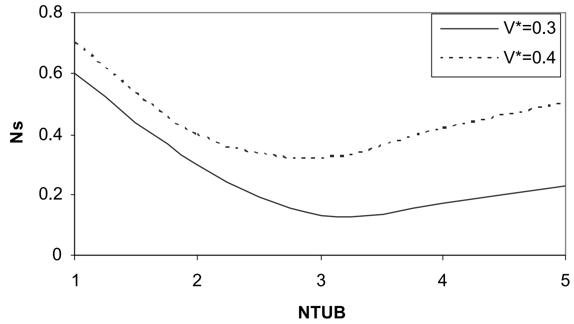


Fig. 4. Variation of $N_S$ with NTUB, $\tau = 1.55$, $\tau_s = 0.6$, $\tau_h = 1.056$, $X_1 = 0.556$, $X_2 = 0.669$, NTUE$\,=0.6$, NTUS$\,=0.4$.

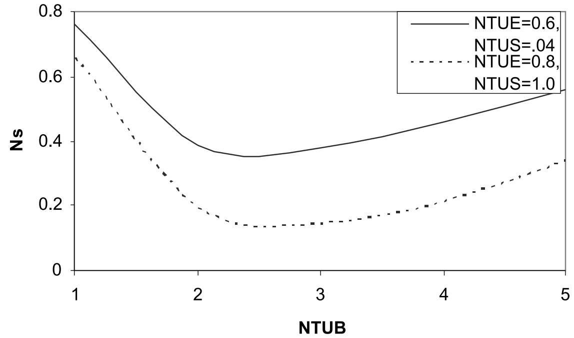


Fig. 5. Variation of $N_S$ with NTUB, $\tau = 1.55$, $\tau_s = 0.6$, $\tau_h = 1.056$, $X_1 = 0.556$, $X_2 = 0.669$, $V^* = 0.4$.

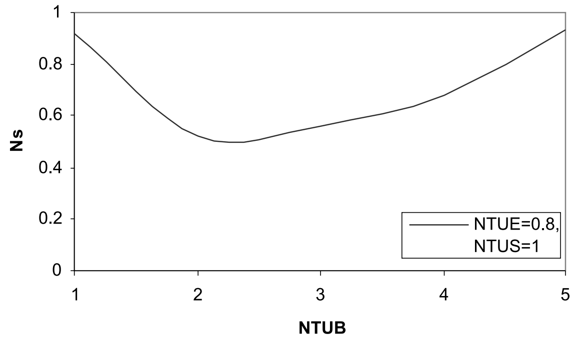


Fig. 6. Variation of $N_S$ with NTUB, $\tau = 1.55$, $\tau_s = 0.6$, $\tau_h = 1.056$, $X_1 = 0.556$, $X_2 = 0.669$, $V^* = 0.5$.

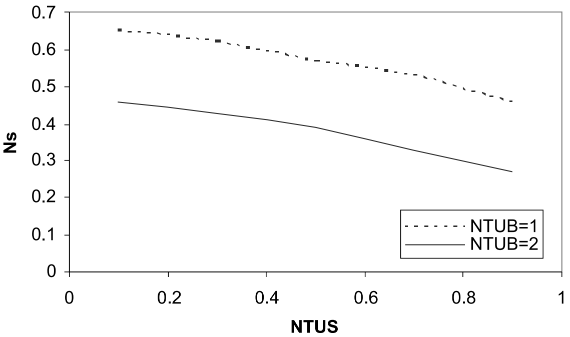


Fig. 7. Effect of NTUS on $N_S$ for different NTUB values, $\tau = 1.6$, $\tau_s = 0.25$, $\tau_h = 0.4$, $X_1 = 0.35$, $X_2 = 0.15$, NTUE = 0.7, $c_{pg} = 1.005$.

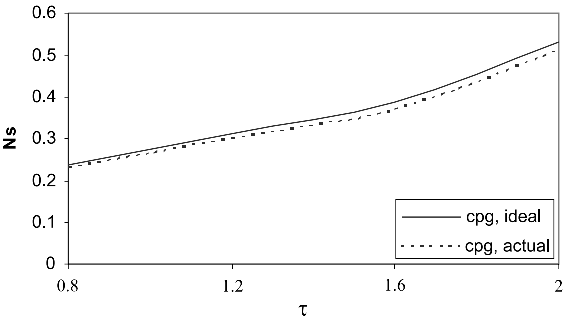


Fig. 8. $N_S$ variation with $\tau$ for ideal and actual gas specific heat ($c_{pg}$) values, $\tau_s = 0.25$, $\tau_h = 0.4$, $X_1 = 0.35$, $X_2 = 0.17$, NTUE = 0.7, NTUS = 0.4, NTUB = 2.0.

irreversibility. This clearly shows that temperature difference between stream-to-stream ($\Delta T$) has a dominating effect on entropy generation rate than frictional pressure drop ($\Delta P$). However, if the flue gas inlet temperature difference ratio is high, then the heat loss and the irreversibility will be high. Also, for higher hot flue gas inlet temperature difference ratio, $\tau$, the temperature of the gas at the exit of evaporator is high, which ultimately results in the gas leaving at higher temperature at the economizer exit. This results in increased irreversibility and entropy generation. So, for a set NTUB value, if the flue gas inlet temperature difference ratio,$\tau$, increases, it results in higher $N_S$ values. For minimum $N_S$ values $\tau$ should be on the lower side. However, this is decided based on the hot flue gas conditions and steam conditions that are required. However for given hot gas conditions and water/steam conditions, the optimum unit size can be selected where the $N_S$ is minimum.

The variation of $N_S$ with NTUB for two different non-dimensional gas velocities, for a given set of other non-dimensional operating parameters is shown in Figs. 4–6, respectively. In all the cases, it is clearly shown that, for a set of parameters, the $N_S$ tends to approach clearly a minimum value at a particular NTUB value and again shows increasing trend. This possibly suggests that for a particular gas flow and water/steam conditions, at a particular NTUB the temperature profiles ($\Delta T$), pressure drops ($\Delta P$) in the unit are such that, the total irreversibility is low. This results in low entropy generation. However, it can be thought that the above mentioned reasons are the possible ones, which might result in this type of variation of $N_S$ with NTUB. The entropy generation number increases for higher non-dimensional gas velocities ($V^*$). The observations in Figs. 3–6 suggest that for the specific chosen parameters, the optimum value of NTUB for minimum $N_S$ is different for all the cases. The reason for selecting different set of operating parameters in Figs. 3–6 is to investigate whether optimum NTUB changes or not in each case. The optimum NTUB at minimum $N_S$ value is different in each case as observed in Figs. 3–6.

Fig. 7 demonstrates the variation of$N_S$ with NTUS for a particular non-dimensional hot flue gas inlet temperature difference ratio ($\tau$). Here, the interest is to study, how the $N_S$ varies, if the NTUS is changed. The results clearly presents that, for a given NTUB, if NTUS is increased $N_S$ decreases. This may be due to the reason that, higher the NTUS, more heat is absorbed from the flue gas, which results in low temperature of the flue gas at the exit of economizer. This results in two things. Low temperature of the gas means low irreversibility and also the external irreversibility gets reduced. The overall effect of them results in lower $N_S$ values. However, the temperature of the flue gas at the exit of economizer cannot be lower than, $T_O$. This is because for heat transfer to take place between flue gas and water, the temperature at the exit of economizer has to be greater than $T_{O,}$ otherwise, heat transfer from hot flue gas to water is not possible.

The influence of gas specific heat ($c_{pg}$) on entropy generation number, $N_S$ is presented in Fig. 8. The main aim of this part is to study how much is the deviation in$N_S$ value, if one takes the actual specific heat value of flue gas taking the gas composition and temperature in to account and an ideal gas specific heat value. The two cases are compared here. In one case, actual value of the flue gas specific heat, taking into gas composition is used. In another case, an ideal specific heat value for the gas is used. The actual gas specific heat taking in to gas composition into account is estimated from the relations available in the thermodynamic literature. The ideal gas specific heat is taken for that of air which is available in standard thermodynamics books. The calculated entropy generation numbers are indicated in Fig. 8. The results clearly presents that, the role of gas specific heat ($c_{pg}$) on entropy generation number is not that significant at lower $\tau$ values. However, for higher $\tau$ values, there is a slight decrease in $N_S$ predicted value with actual gas specific heat, than that with an ideal gas specific heat value.

## 4. Conclusion

A general equation for entropy generation number ($N_S$) for a waste heat recovery steam generator producing superheated steam is proposed.

An attempt has been made to present some basic results on the effect of certain combination of non-dimensional operating parameters on entropy generation number. It has been observed that for a particular non-dimensional inlet gas temperature difference ratio ($\tau$) and for other fixed parameters, $N_S$ is minimum at a particular NTUB suggesting a possible optimum value for the design, where the total irreversibility is low and the performance will be better for the waste heat recovery steam generator.

The entropy generation number increases with increase in non-dimensional hot flue gas inlet temperature difference ratio ($\tau$) due to higher temperature difference $\Delta T$) between stream-to-stream (flue gas and water/steam) which increases total irreversibility.

The specific sources that are responsible for entropy generation are the temperature difference between stream-to-stream ($\Delta T$), heat loss to the surroundings, ambient temperature and frictional pressure drop in the unit ($\Delta P$). The temperature difference between stream-to-stream ($\Delta T$) (hot gas to water/steam) has dominating effect on the entropy generation rate.

The deviation in predicted $N_S$ value, with actual specific heat of the gas taking into gas composition into account ($c_{pg}$) and with ideal specific heat ($c_{pg}$) of the gas is not significant at low hot flue gas inlet temperature difference ratios ($\tau$). However, for higher $\tau$ values, there is a slight reduction in $Ns$ value with actual gas specific heat values.

## Acknowledgements

The authors acknowledge the encouragement and help from Dr. V.N. Vedamurthy, ex-Principal of Vellore Engineering College, Vellore, India.

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