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OPTIMUM OPERATION FOR DIFFERENT DESIGN CONDITIONS OF A SINGLE PRESSURE HEAT RECOVERY STEAM GENERATOR (HRSG) FOR MINIMUM ENTROPY GENERATION

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ABSTRACT

Heat Recovery Steam Generator (HRSG) is the critical link between the “topping” gas and “bottoming” steam power cycle of a combined power plant. Depending on the mass flow and property of exhaust flue gas from the gas cycle, optimum design and operation, for minimum entropy generation in a single pressure HRSG generating superheated steam is reported in this paper. It shows that an optimum pressure of steam generation exists for minimum entropy generation. The interdependence between the physical size (and hence cost) of different heat transfer components of HRSG with the thermodynamic optimum operation for minimum entropy generation is presented. Optimum pressure of steam generation increases with greater size of super heater, economizer and evaporator up to certain limit, beyond which the increase in optimum saturation pressure is marginal with increase in size of the heat transfer units. However, increasing evaporator size and degree of superheat have the adverse effect of lesser steam generation for a given economizer and super heater for minimum entropy generation.

Keywords: Heat Recovery Steam Generation (HRSG), entropy generation.

1. INTRODUCTION

As the earth’s reserves of fossil fuels and other non renewable sources of energy gradually dry up, human beings are facing a desperate need to conserve the rapidly depleting energy sources and use them in a more efficient manner. Thus, it is becoming increasingly important for engineers to shift attention towards improvement in efficiencies of power plants. An easy way to achieve higher efficiencies with power plants is by utilizing the heat from exhaust flue gases from various thermal systems, as they contain considerable amount of available energy on account of being at a higher temperature compared to the ambient. Combined cycle power plants are being developed with a vision of realising this concept. In a combined gas-steam power plant heat from exhaust flue gases of the ‘topping’ gas cycle is utilized to generate steam in the ‘bottoming’ steam in a heat recovery steam generator (HRSG) [1]. Efficiency of combined cycle power plants are seen to be considerably greater than conventional steam and gas power cycles [2]. In absence of supplementary firing, the operation of an HRSG is similar to that of a feed water heater, where heat from the exhaust flue gases of the gas turbine is used to generate superheated steam from feed water. Second law analysis of thermal systems or components is very important as it points out the sources of loss of work potential (or entropy generation) and suggests measures for improvement of thermodynamic performance [3].

Second law optimization for thermal systems can be done in different ways [4]. One method is by Entropy generation minimization in which the possibility of an optimal thermal system is explored corresponding to least irreversibility in its process of operation. Optimum design and operation of an HRSG corresponding to minimum entropy generation was studied by Nag and De [5]. Reddy et al reported the second law analysis of an HRSG using entropy generation minimization principle [6]. Exergy analysis was carried out on an HRSG by In and Lee [7]. Exergoeconomic analysis of an HRSG was carried out by Ghazi et al. using multi modal genetic algorithm for its thermoeconomic optimization [8]. Butcher and Reddy [9] studied effects of different operating parameters on thermodynamic performance and reported second law analysis of combined power plants.

In this paper variations in optimum operating conditions of an HRSG has been explored for different design conditions by studying a thermodynamic model of an HRSG based on the principle of minimum entropy generation.

2. ANALYSIS

2.1 Description Of The System

The exhaust gas from the topping gas cycle of the combined cycle power plant is fed to the super heater of the HRSG at a temperature of T_{in}.

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The flue gas exits the super heater at temperature T_1 and enters the evaporator at the same temperature. After leaving the evaporator at temperature T_2 it enters the economizer with the same temperature and finally leaves the HRSG at a temperature T_{out} into the ambient at T_0. Water is supplied to the economizer of the HRSG at the ambient temperature T_0 and at a particular pressure. It leaves the economizer at saturated liquid state and enters the evaporator at temperature T_s(saturation temperature corresponding to pressure of water initially fed). It leaves the evaporator at saturated vapour state and enters the super heater. The super heated steam then exits the HRSG at a temperature T_{sh}. The flow diagram for the assumed model and the corresponding T-s diagram are shown in Figs. 1 and 2 respectively. Various simplifying assumptions are made for this model, which are stated below.

2.2 Assumptions

1. System is in steady state.
2. Both the flue gas and the water/steam is assumed to be inviscid and incompressible.
3. Adiabatic heat exchanger is assumed.
4. Constant specific heats are assumed for water, steam and flue gas.
5. The difference in temperature between the inlet flue gas to the HRSG and the superheated steam exiting the HRSG must not be lesser than 40°C.
6. The heat exchangers (economizer, evaporator and super heater) operate under counter flow conditions.

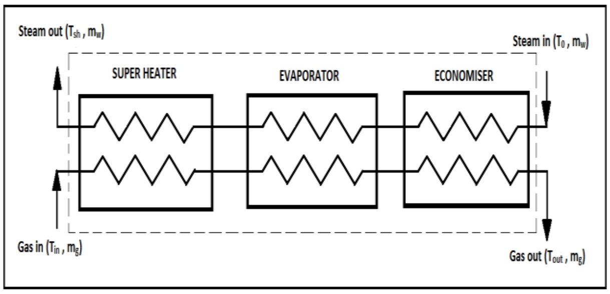

Fig.1: Schematic of the HRSG under consideration

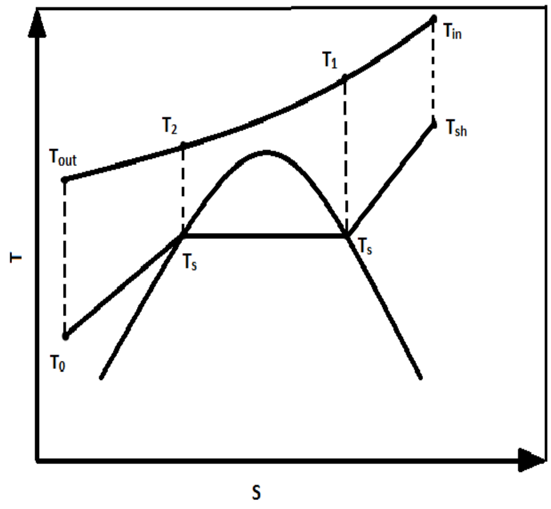

Fig. 2: T-s diagram of gas and water/steam for heat

2.3 Formulation Of The Problem

2.3.1 First law analysis

2.3.1.1 Overall HRSG

From overall energy balance of the Heat Recovery Steam Generator,

$$m_g c_g(T_{in} - T_{out}) = m_w c_w(T_s - T_0) + m_w h_{fg} + m_w c_{sh}(T_{sh} - T_s)$$

Rearranging,

$$\frac{(T_{out} - T_0)}{T_0} = (NTG - XNTS) - X_{sh}\,(NTSH - NTS) - \frac{(Xh_{fg})}{c_w T_0} \qquad (1)$$

2.3.1.2 Super Heater

Energy balance for super heater where T_g and T_w are the temperatures of gas and steam at any cross section of the super heater,

$$m_g c_g\,(T_{in} - T_g) = m_w c_{sh}\,(T_{sh} - T_w)$$

$$T_w = T_{sh} - \frac{T_{in} - T_g}{X_{sh}} \qquad (2)$$

Performing heat balance for an elemental length ‘dL_{sh}‘ of the super heater,

$$-m_g c_g dT_g = U_{sh} P_{sh}(T_g - T_w) dL_{sh}$$

Substituting 'T_w' from Eq. 2 and integrating over the entire length of the superheater,

$$ln\frac{T_1-\frac{X_{sh}}{X_{sh}-1}T_0(1+NTSH)+\frac{T_{in}}{X_{sh}-1}}{T_{in}-\frac{X_{sh}}{X_{sh}-1}T_0(1+NTSH)+\frac{T_{in}}{X_{sh}-1}} = \left(\frac{1-X_{sh}}{X_{sh}}\right)NTUSH \quad (3)$$

Applying energy balance to entire super heater and rearranging,

$$T_1 = T_0(1+NTG)-T_0X_{sh}(NTSH-NTS) \quad (4)$$

Substituting T_1 in Eq. 3,

$$1-(X_{sh}-1)\frac{NTSH-NTS}{NTG-NTSH} = e^{\frac{1-X_{sh}}{X_{sh}}NTUSH} \quad (5)$$

2.3.1.3 Evaporator

For an elemental length 'dL_e' of the evaporator heat transfer equation yields,

$$-m_gc_gdT_g = U_{ev}P_{ev}(T_g-T_w)dL_{ev}$$

Integrating for the evaporator as a whole,

$$ln\left(\frac{T_2-T_s}{T_1-T_s}\right) = -\frac{U_{ev}P_{ev}L_{ev}}{m_gc_g} = -NTUEV \quad (6)$$

For counter flow heat exchange in the evaporator,

$$m_wh_{fg} = U_{ev}P_{ev}(LMTD)_{ev} \quad (7)$$

From equation (6) & (7),

$$h_{fg} = \frac{c_w}{X}(T_1-T_s)(1-e^{-NTUEV})$$

Substituting the value of 'T_1' Eq. 4,

$$h_{fg} = \left(\frac{c_w}{X}\right)T_0\big((NTG-NTS)-X_{sh}(NTSH-NTS)\big) .(1-e^{-NTUEV}) \quad (8)$$

From energy balance of the evaporator,

$$m_gc_g(T_1-T_2) = m_wh_{fg}$$

$$T_2 = T_1-\left(\frac{X}{c_w}\right)h_{fg} \quad (9)$$

2.3.1.4 Economizer

The energy balance of the economizer is,

$$m_gc_g(T_2-T_g) = m_wc_w(T_s-T_w)$$

$$T_w = T_s-\frac{(T_2-T_g)}{X} \quad (10)$$

For an elemental length 'dL_e' of the economizer,

$$-m_gc_gdT_g = U_eP_e(T_g-T_w)dL_e$$

Substituting 'T_w' from Eq. 10 and integrating for the total heat transfer in the economizer,

$$\ln\frac{T_{out}-\left(\frac{X}{X-1}\right)\left(T_s-\frac{T_2}{X}\right)}{T_2-\frac{X}{X-1}\left(T_s-\frac{T_2}{X}\right)} = \left(\frac{1-X}{X}\right)NTUE \quad (11)$$

From Eqs. 4, 9 and 11,

$$e^{(\frac{1-X}{X})NTUE} = 1-\frac{(X-1)NTSe^{NTUEV}e^{\frac{1-X_{sh}}{X_{sh}}NTUSH}}{(NTG-NTSH)} \quad (12)$$

Equation 15 involving design and operational parameters and is obtained from overall heat balance. This equation acts as an equation of constraint that must be simultaneously satisfied during entropy generation minimization.

2.3.2 Entropy generation

The entropy generation for the assumed model can be expressed as:

$$S_{gen} = m_gc_gln\left(\frac{T_0}{T_{in}}\right)+m_gc_g\frac{(T_{out}-T_0)}{T_0} + m_wc_wln\left(\frac{T_s}{T_0}\right)+\frac{m_wh_{fg}}{T_s} + m_wc_{sh}ln\left(\frac{T_{sh}}{T_s}\right)$$

The above equation represents the entropy generation due to irreversible heat transfer occurring in the HRSG and also that due to discharge of exhaust gas at a higher temperature than the ambient. Dividing the above equation by m_gc_g, we get the entropy generation number.

Entropy generation number,

$$N_s = -\ln(1+NTG)+\frac{T_{out}-T_0}{T_0}+X\ln(1+NTS) + \frac{Xh_{fg}}{c_w}T_s+X_{sh}\ln\left(\frac{1+NTSH}{1+NTS}\right)$$

Substituting the value of $\frac{T_{out}-T_0}{T_0}$ from Eq. 1 and h_{fg} from Eq. 8 we obtain,

$$Ns = -ln(1+NTG)+Xln(1+NTS) + X_{sh}ln\left(\frac{1+NTSH}{1+NTS}\right)+NTG - XNTS-X_{sh}(NTSH-NTS) -\big((NTG-NTS)-X_{sh}(NTSH-NTS)\big)\left(\frac{NTS}{1+NTS}\right) .(1-e^{-NTUEV}) \quad (13)$$

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This equation represents the entropy generation number as a function of several non-dimensional design and operational parameters. N_s has to be at a minimum for 'optimum' set of these parameters and simultaneously satisfying the first law (i.e. heat balance, equation of constraint (i.e. Eq. 13).

3. RESULTS AND DISCUSSIONS

For our assumed model, entropy generation due to heat transfer in a heat recovery steam generator can be expressed as a function of five independent design or operational parameters. The remaining ones are inter related and their inter relation is governed by Eq. 13 that was derived from first law analysis of the HRSG, thus, serving as an equation of constraint. All the parameters along with entropy generation (they are : NTG, NTSH, NTS, NTUSH, NTUEV, NTUE, X, X_{sh}) have been expressed in terms of non-dimensional quantities, which enables us to derive generalized conclusions regardless of definite values of these parameters. Variation of entropy generation number with the variation of these eight non dimensional parameters was thoroughly checked. However, an optimum value was obtained only for the case of NTS (i.e. non dimensional saturation temperature of steam) keeping inlet flue gas temperature (NTG), maximum super heated steam temperature (NTSH), heat capacity ratios (X and X_{sh}) and number of transfer units of evaporator (NTUEV) as constants.

N_s v/s NTS was plotted for different values of NTUE (number of transfer units of economizer) and NTUEV and is shown in Figs. 3 and 4.

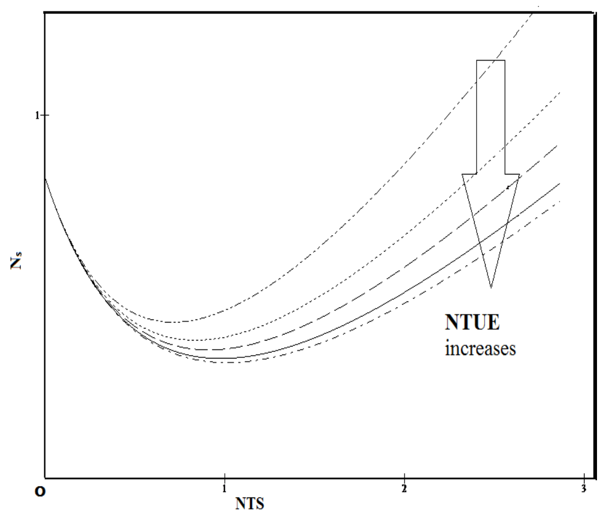

Fig. 3: N_s v/s NTS for different NTUE

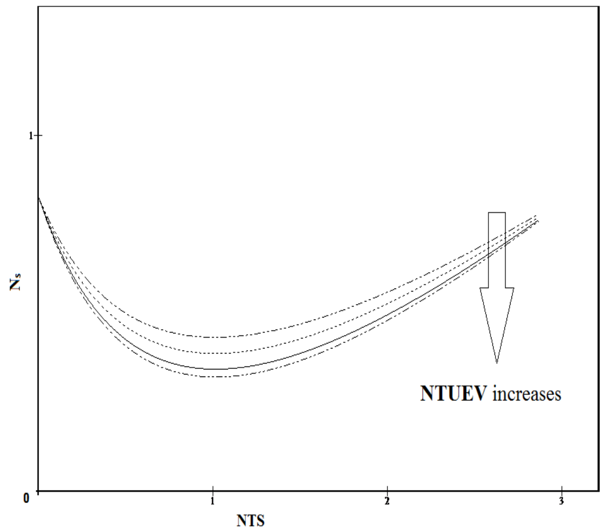

Fig. 4: N_s v/s NTS for different NTUEV

As NTUE increases, the optimum value of steam generation increases. With increasing NTUE heat transfer from the flue gas (topping gas cycle) to the steam (bottoming steam cycle) increases. As a result the temperature profiles of the gas and water tend to come closer, hence, reducing irreversibility. Therefore increasing NTUE, minimum entropy generation decreases and optimum pressure for steam generation increases.

Similarly, increasing NTUEV, also results in bringing the temperature profiles of the gas and steam closer to each other. As a result, entropy generation generally decreases with increasing NTUEV (refer to Fig 4). However, varying NTUEV keeps the optimum saturation temperature more or less the same.

On the other hand, increasing the number of transfer units of super heater results in a decrease in the value of optimum saturation temperature for steam generation even though there is an overall reduction in entropy generation in the heat exchange process.

It is however observed that beyond certain values of NTUEV, NTUSH or NTUE the reduction of entropy generation is not very significant.

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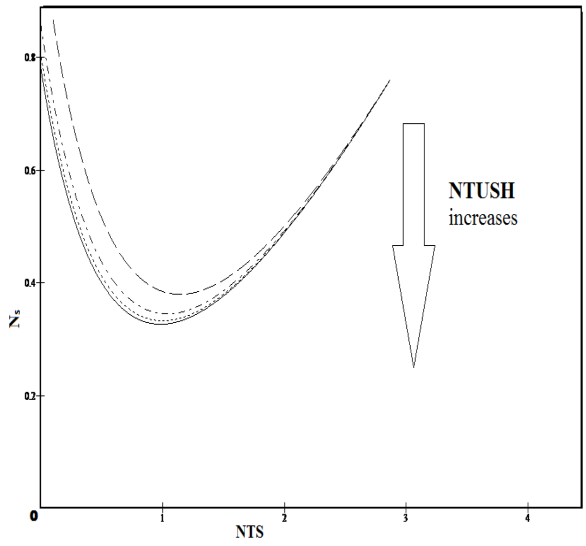

Fig. 5: N_s v/s NTS for different NTUSH

As we increase the value of maximum superheated temperature of steam (NTSH) the entropy generation decreases (shown in Fig. 5) because the exhaust flue gas temperature reduces (T_{out} decreases). However, increasing NTUE shifts the Ns v/s NTSH curve downward (shown in Fig. 6) i.e. for same value of NTSH the entropy generation is lesser for higher value of NTUE. The explanation is similar to the earlier case i.e. a greater number of transfer units helps to decrease T_{out} and hence, helps to reduce the entropy generation of the system.

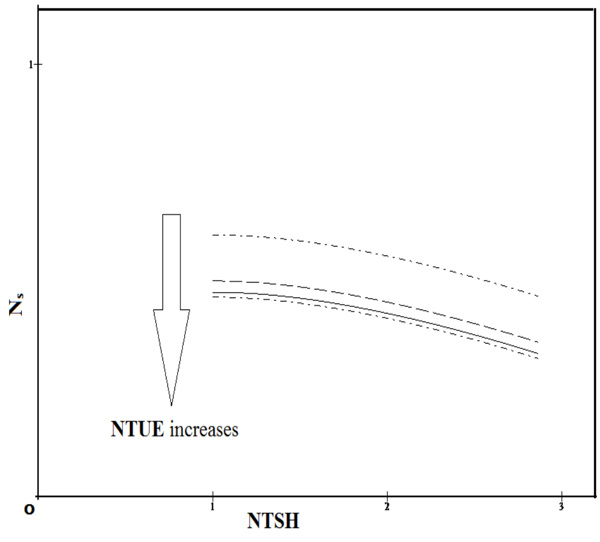

Fig. 6: N_s v/s NTSH for different NTUE

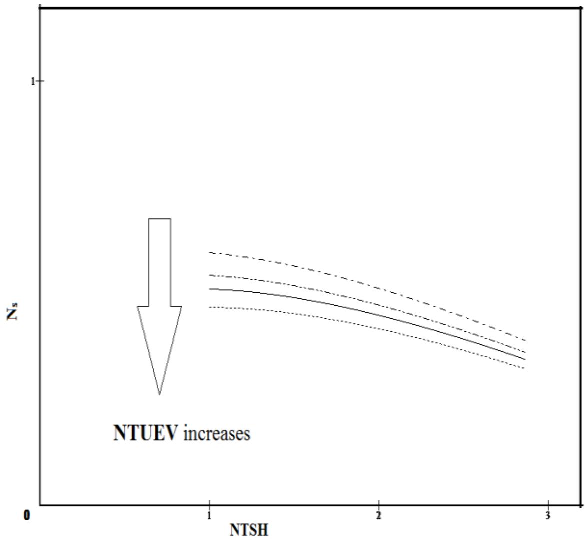

Fig. 7: N_s v/s NTSH for different NTUEV

On increasing NTUEV, a similar observation is made regarding the variation of N_s with NTSH shown in Fig. 7.

In Figs. 8 and 9, the variation of NTUE with X (heat capacity ratio of water and flue gas in the economizer) is observed for different value of NTUEV and NTUSH respectively. NTUE naturally increases with increasing values of X for constant operational parameters (NTS, NTSH) on the steam side. Also, we notice that for same value of X, increased size of the evaporator implies that a greater size of economizer is also necessary. However, for same X, as we increase the super heater size, the size of economizer required would smaller.

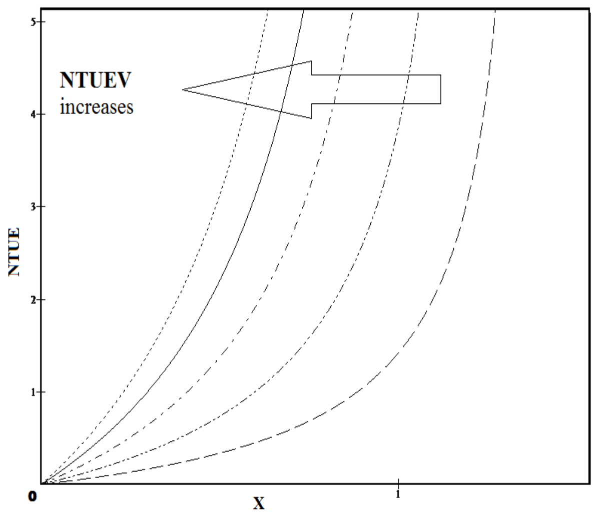

Fig. 8: NTUE v/s X for different NTUEV

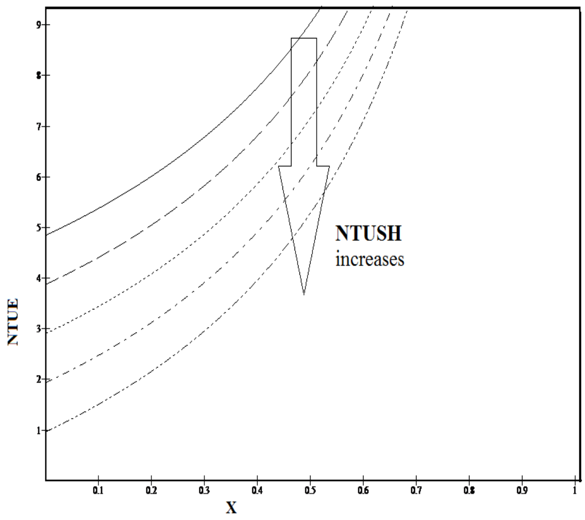

Fig. 9: NTUE v/s X for different NTUSH

4. CONCLUSIONS

A general equation for entropy generation due to heat exchange in a single pressure, unfired HRSG, generating super heated steam was developed in this paper. Using it, the variation of optimum operation of the HRSG was studied for varying design parameters. It was seen that with the increase in the number of transfer units of the individual heat exchangers (economizer, evaporator and super heater), the optimum saturation temperature increases slightly with increasing NTUE and decreases slightly with increasing NTUSH. However, with NTUEV there is no perceptible change in the optimum saturation temperature. The increase in sizes of the heat transfer components brings about overall reduction in entropy generation of the heat transfer process.

Also, we noted that as the size of the evaporator is increased, a corresponding increase in size of economizer is also necessary to maintain a constant heat capacity ratio. On the contrary, increase in size of the super heater would mean that we can operate with a smaller economizer. Thus, spending money on the super heater would be more cost effective than that for the evaporator.

NOMENCLATURE

Symbol

A = Surface area of heat transfer, m^2

c = Specific heat at constant pressure, kJ/kg K

h_{fg} = enthalpy of evaporation , kJ/kg

L = Total length of super heater, evaporator or economizer tubes, m

m = Mass flow rate, kg/s

N_s = Entropy generation number, $\frac{S_{gen}}{m_g c_g}$

NTG = Non-dimensional temperature of exhaust gas from the gas turbine, $\frac{(T_{in}-T_0)}{T_0}$, dimensionless

NTS = Non-dimensional saturation temperature of the steam generated, $\frac{(T_s-T_0)}{T_0}$, dimensionless

NTSH = Non-dimensional super heated temperature of the steam generated, $\frac{(T_{sh}-T_0)}{T_0}$, dimensionless

NTUSH = Number of transfer units of super heater, dimensionless

NTUEV = Number of transfer units of evaporator, dimensionless

NTUE = Number of transfer units of economizer, dimensionless

P = Perimeter of the heat exchanger tubes, m

S = Entropy kJ/kg K

T = Absolute thermodynamic temperature, K

U = Overall heat transfer coefficient, kW/m^2K

X = Heat capacity ratio of water or steam, $\frac{m_w c_w}{m_g c_g}$, dimensionless

X_{sh} = Heat capacity ratio of super heated steam $\frac{m_w c_{sh}}{m_g c_g}$, dimensionless

Subscript

sh = Super heater

e = Economizer

ev = Evaporator

g = Exhaust gas from the gas turbine of the topping cycle

in = Inlet of the exhaust gas to HRSG

1 = Intermediate between the super heater and the evaporator

2 = Intermediate between the evaporator and the economizer

0 = Ambient

opt = Optimum

out = Exit from the HRSG on the gas side

s = Saturation property on the steam generated

w = Water or steam

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