

## Cost Effectiveness of Waste Heat Recovery and Utilization of a 450 MW Combined Cycle Power Plant

M. Sultan Khan, *khan@giki.edu.pk*

Adnan Aslam, *adnanaslam870@gmail.com*

Auneeb Iqbal, *auneb@live.com*

Ahmad J. Dar, *ahmad-gikian@hotmail.com*

Faculty of Mechanical Engineering, GIK Institute, Topi, Swabi, N.W.F.P, Pakistan

### ABSTRACT

The objective of this paper is to determine the feasibility of waste heat recovery and utilization of a 450MW combined cycle power plant. The primary fuel of the power plant was switched to natural gas and possibility of waste heat recovery to be determined. This paper discusses the procedure followed to determine the exergy of the flue gases from the heat recovery steam generator that can be utilized to preheat the natural gas to the desired temperature. A complete analysis is carried out to determine the payback period in case investment is done to recover the waste heat versus savings of fuel. The paper also discusses design of heat exchangers and selection of associated equipment in order to utilize the waste heat. The impact of utilization of this waste heat on the overall power plant efficiency is also determined. Based on the annual fuel savings a payback period is calculated to be about three years.

**Keywords:** Combined cycle power plant; Waste Heat; Exergy; Cost effectiveness

### NOMENCLATURE

$\dot{m}_f$	Mass flow rate of the fuel (kg/s)	$\dot{m}_{\text{flue gas}}$	Mass flow rate of the flue gas (kg/s)
$\dot{m}_a$	Mass flow rate of the air (kg/s)	ppm	Parts per million
$t_D$	Dew Point temperature(°C)	$P_{\text{H}_2\text{O}}$	Vapour pressure of water, equal to the mole fraction of water
$P_{\text{SO}_2}$	Vapour pressure of SO <sub>2</sub> , equal to the mole fraction of SO <sub>2</sub>	C	Conversion rate of SO <sub>2</sub> to SO <sub>3</sub> , since it was not given it is usually assumed to be 1%
$m_i$	mass fraction of the component of flue gas	$C_{p,\text{flue gases}}$	Specific heat capacity of flue gas component kJ/kg
$\Psi$	specific exergy	s	Specific entropy
$P_0$	Pressure at the dead state	$T_0$	Temperature at the dead state
P	Pressure	T	Temperature
h	Enthalpy	$R_{\text{ave}}$	Average universal gas constant value of flue gas
$K_c$	Percentage ratio of carbon in chemical composition (%)	$K_N$	Percentage ratio of nitrogen in chemical composition (%)
$K_s$	Percentage ratio of sulphur in chemical composition (%)	HRSG	Heat recovery steam generator
$\rho$	Density	$\mu$	Dynamic viscosity
$D_i$	Internal Diameter of the tube	$D_o$	Outer Diameter of the tube
t	thickness of the tube	P1(V)	Vertical Pitch of the tube
P2(	Horizontal Pitch of the tube	s	space between the fins

H)			
w	width of the fins	h	height of the fin
$D_{fin}$	Diameter of the fin (from edge to edge)	N	Total number of tubes
e	Relative Roughness	$\epsilon$	roughness of tube
R	radius of bend	a	inside radius of tube
K	Total loss coefficient	$K^*$	Loss coefficient
$v_{app}$	Velocity of Approach	$S_o$	Frontal Area
$Nu_{roach}$	Nusselt Number	$\bar{h}_{a,r}$	air side heat transfer coefficient
U	Overall heat transfer coefficient	PR	Effectiveness

## 1. INTRODUCTION

Waste heat from a power plant is defined as the energy that is being exhausted to the atmosphere, which may be recovered and used for a variety of purposes, ranging from heating of process water to heating of pre combustion gases. The quality of the exhaust gas is dependent upon the temperature, the higher the temperature the better the quality. The essential quality of heat is not the amount but rather its exergy. The strategy of how to recover this heat depends in part on the temperature of the waste heat gases and the economics involved. Large quantity of hot flue gases is generated from boilers and steam generators. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. However, not all the energy lost in waste gases can be recovered.

Combined Cycle Power Plants constitute of eight major components, Compressor, Gas Turbine (GT), Heat Recovery Steam Generator (HRSG), Pump, Steam Turbine (ST), Generators, Condenser and Cooling Towers. A combined cycle operates on two thermodynamic cycles, Brayton Cycle and Rankine Cycle. The Brayton Cycle constitutes of a compressor coupled with a GT and the Generator. The air which is compressed in the compressor enters a combustion chamber where the primary fuel is ignited and then the air is expanded in the GT; the exhaust of a GT is at a high temperature [1]. The exhaust from the gas turbine is used to supply energy to the Rankine Cycle. HRSG is used to extract energy from the exhaust of the GT and produce steam for the Rankine Cycle. The steam is then expanded in the steam turbine to produce additional electricity, which otherwise would not have been possible and thus increasing the efficiency of the plant. The expanded steam is cooled in a condenser. A condenser is a heat exchanger that exchanges the heat with cooling water and converts steam into condensed vapour. The heated water (from the condenser) is usually cooled in the cooling towers to be reused. The condensed vapour on the other hand is pumped to high pressures and sent to HRSG for conversion to steam.

Exergy is defined as the maximum theoretical useful work that can be obtained as a system interacts with an equilibrium state. The exergy is not generally conserved like energy but is destroyed in the system. The main exergy source is the flue gases which are the exhaust from HRSG which for the power plant considered in this work has a temperature of 165°C. The amount of potential available in the flue gases is about 3.327MW from one HRSG and the total of the 6.6MW from the two HRSGs of the power plant under consideration.

## 2. QUANTIFICATION OF THE FLUE GAS AND ITS POTENTIAL

The main source of exergy in our case is the HRSG exhaust, the exhaust is sent to the atmosphere at a high temperature 165°C.

### Quantification of the Available Amount of Heat:

For quantification of waste heat, the lower temperature range to which the exhaust gas temperature is determined without reaching the acid dew point. The composition of the flue gases is given in Table 1:

**Table 1. Flue gas Composition**

Flue Gas	Composition
CO <sub>2</sub>	5.0292 %
SO <sub>2</sub>	10 ppm (the highest value recorded)
N <sub>2</sub>	72.9095%
H <sub>2</sub> O	5.5547%
O <sub>2</sub>	15.2910%
Ar	1.2156
CO	8 ppm (highest value recorded)
NO <sub>x</sub>	23 ppm (highest value recorded)

$$\dot{m}_f = 10.258 \text{ kg/s}, \dot{m}_a = 465.916 \text{ kg/s}, \dot{m}_{\text{flue gas}} = 475.916 \text{ kg/s}$$

where,

$$\dot{m}_f = \text{mass flow rate of fuel}, \dot{m}_a = \text{mass flow rate of air and } \dot{m}_{\text{flue gas}} = \text{mass flow rate of the flue gas}$$

For methane, considered as fuel for the power plant running on natural gas

$$10.258 \text{ kg/s} = 10.258 \text{ (kg/s)} (1000 \text{ g/kg}) = 10258 \text{ (g/s)} (1 \text{ mole}/16 \text{g}) = 641.125 \text{ moles/s}$$

In present work air is at 265% of theoretical air, the balanced equation is as follows:



$$\frac{\% \text{ weight of flue gas component}}{100} = \frac{\text{mass flow rate of the component}}{\text{mass flow rate of flue gas}} \quad (2)$$

$$\text{no of moles of the flue gas component} \frac{\text{component}}{\text{second}} = \frac{\text{mass flow rate of the component}}{\text{atomic mass of the component}} \quad (3)$$

$$\frac{\text{ppm of flue gas component}}{10^6} = \frac{\text{mass flow rate of the component}}{\text{mass flow rate of the flue gas}} \quad (4)$$

The composition calculated in mole fraction is shown in Table 2:

**Table 2. Composition in Mole Fraction**

Composition in Mole Fraction	
CO <sub>2</sub>	3.233%
O <sub>2</sub>	13.517%
N <sub>2</sub>	73.656%
H <sub>2</sub> O	8.729%

Ar	0.861%
CO	$8.08 \times 10^{-4}$ %
NO <sub>x</sub>	$2.17 \times 10^{-3}$ %
SO <sub>2</sub>	$8.85 \times 10^{-4}$ %

In order to calculate the dew point the following equation is used: [2].

$$t_d = 203.25 + 27.6 \log(P_{H_2O}) + 1.83 \log(CP_{SO_2}) + 1.06 \{ \log(CP_{SO_2}) + 8 \}^{2.19} \quad (5)$$

where,

$t_d$  = Dew point temperature in °C

$P_{H_2O}$  = vapour pressure of water, equal to the mole fraction of water

$P_{SO_2}$  = vapour pressure of SO<sub>2</sub>, equal to the mole fraction of SO<sub>2</sub>

C = Conversion rate of SO<sub>2</sub> to SO<sub>3</sub>, since it was not given it is usually assumed to be 1%

$P_{H_2O} = 0.0872$

$P_{SO_2} = 8.851 \times 10^{-6}$

By putting these values in equation (5) we get

$$t_d = 98.58^\circ\text{C} \quad (6)$$

Using the following equation we can calculate the specific heat capacity for a mixture [1].

$$C_p = \sum_{i=1}^n (m_i c_{pi}) \quad (7)$$

$$C_p = 1.0265 \text{ kJ/kg}$$

where,

$m_i$  = mass fraction of the component of flue gas

$C_p$  = Specific heat capacity of flue gas component

The exergy of a system is the maximum useful work possible during a process that brings the system into equilibrium with a heat reservoir. Exergy is the energy that is available to be used. Unlike the energy the exergy cannot be conserved. The flow exergy of flue gas can be expressed as follows: [1]

$$\psi = (h - h_o) - T_o (s - s_o) \quad (8)$$

where,

$\psi$  = specific exergy

$s$  = specific entropy

and the subscript zero indicates the properties at the dead state of  $P_o$  and  $T_o$ .

$$s - s_o = c_p \ln \frac{T}{T_o} - R_{ave} \ln \frac{P}{P_o} \quad (9)$$

$$h - h_o = c_{p,fluegas} (T - T_o) \quad (10)$$

where,  $R_{ave}$  is the average universal gas constant value of flue gas. The average universal gas constant of combustion product is given as follows:

$$R_{ave} = \frac{K_C(0.6927) + K_N(0.2968) + K_H(4.1249) + K_S(0.2596) + K_M(0.4615) + m_{air.steo}(0.2201)}{m_{fluegas}} + \frac{(m_{air.steo} \cdot (n-1) \cdot (0.287))}{m_{fluegas}} \quad (11)$$

where;

$K_c$  =percentage ratio of carbon in chemical composition (%)

$K_N$  =percentage ratio of nitrogen in chemical composition (%)

$K_S$  =percentage ratio of sulphur in chemical composition (%)

$\bullet$   
 $m_{fluegas}$  =mass flow rate of the flue gas

$K_H$  =percentage ratio of hydrogen in chemical composition (%)

Equating equation (9) and (10) in equation (8) we get:

$$\psi = C_{p,fluegas} (T - T_o) - T_o \left( C_{p,fluegas} \ln \frac{T}{T_o} - R_{ave} \cdot \ln \frac{P}{P_o} \right) \quad (12)$$

By further simplifying it we get;

$$\psi = C_{p,fluegas} \left[ (T - T_o) - T_o \left( \ln \frac{T}{T_o} - \frac{R_{ave}}{C_{p,fluegas}} \cdot \ln \frac{P}{P_o} \right) \right] \quad (13)$$

From the data given the specific exergy flow has been calculated; the value of  $R_{ave}$  has been calculated on the basis of the flue gas composition. The values calculated are given as follows;

$$R_{ave} = 0.2633 \text{ kJ/kg.K} \quad C_p = 1.0265 \text{ kJ/kg}$$

$$T_o = 98.58^\circ\text{C} = 371\text{K} \quad (\text{as the dew point in our case is } 98.58^\circ\text{C})$$

$$T = 165^\circ\text{C} = 438 \text{ K} \quad (\text{the exhaust temperature of the flue gases})$$

$$P_o = 1 \text{ atm} \quad P = 1.015 \text{ atm}$$

By putting the above values in the equation (13) we get;

$$\psi = C_{p,fluegas} (T - T_o) - T_o \left( c_{p,fluegas} \ln \frac{T}{T_o} - R_{ave} \cdot \ln \frac{P}{P_o} \right) \quad (14)$$

$$\psi = 7.006 \text{ kJ/kg}$$

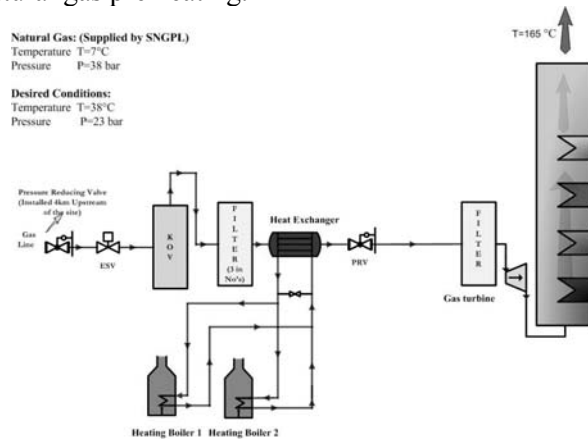
Now, multiplying the above value of specific exergy with the mass flow rate of the flue gases we get the value of specific exergy flow in MW: Specific exergy flow (MW) = 7.006 kJ/kg  $\times$  475kg/s and hence the specific exergy flow = 3.327 MW

### 3. UTILIZATION OF AVAILABLE WASTE HEAT

Although there were a number of options available but according to the needs of the plant process and due to the ease in the implementation, the option of utilization of waste heat as source of heating the natural gas was considered.

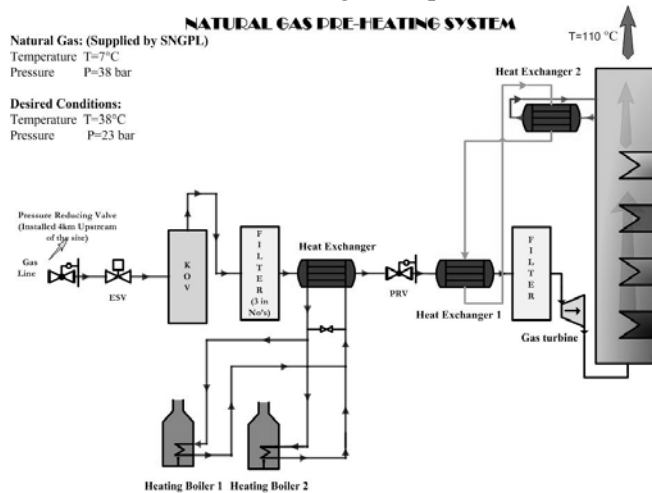
## Existing and Proposed System

The gas is provided to the plant at 7°C but the required operational temperature is 38°C. Therefore, to heat the natural gas three heat exchangers were initially installed and along with them two boilers were also installed to provide the hot water to the heat exchangers. Figure 1 shows the schematic of the existing system for natural gas pre heating.



**Figure 1 Existing Natural Gas Preheating System**

It is proposed to install new heat exchangers in the second skid prior to the filter, as the second skid has approximately half the distance from the stack as the first skid. The heating boilers currently use a large quantity of fuel to achieve the desired natural gas temperature and are very expensive to run.



**Figure 2 Proposed Natural Gas Preheating System**

The target is to fully/partially replace the boilers with the heat exchangers that will transfer the recovered heat from the flue gas to preheat the natural gas. Figure 2 shows the schematic of the proposed system.

## Quantification of the Required Heat

The amount required to heat the natural gas can be calculated as:

$$\dot{Q}_{\text{natural gas}} = \dot{m}_{\text{natural gas}} C_p (T_i - T_o) \quad (15)$$

The amount of heat required to heat the natural gas turns out to be 653 kW and the amount of heat needed at the input of heat exchanger 1 (HX<sub>1</sub>) is 850 kW

$$\dot{Q}_{\text{Heat exchanger}_1} = \frac{\dot{Q}_{\text{natural gas}}}{\eta_{\text{HX}_1}}$$

The amount of heat needed at the output of the stack heat exchanger (HX<sub>2</sub>) is 935 kW.

$$\dot{Q}_{\text{HX}_2\text{output}} = \dot{Q}_{\text{HX}_1\text{input}} + \text{losses in piping} \quad (16)$$

The amount of heat needed at the stack heat exchanger (HX<sub>2</sub>) input is 1350kW.

$$\dot{Q}_{\text{HX}_2\text{input}} = \frac{\dot{Q}_{\text{HX}_2\text{output}}}{\eta_{\text{HX}_2}}$$

Total amount needed to be extracted from the flue gas exhaust is 1350 x 2 = 2700 kW or 2.7 MW.

**Natural Gas: (Supplied by SNGPL)**

Temperature T=7°C  
Pressure P=38 bar

**Desired Conditions:**

Temperature T=38°C  
Pressure P=23 bar

**Heat Exchanger 1 (HX1):**

$\eta = 77\%$

**Natural Gas:**

Temperature inlet T<sub>i</sub>=7°C  
Temperature outlet T<sub>o</sub>=38°C  
Mass flow rate  $\dot{m}=10.258 \text{ kg/s}$

**Water:**

Temperature inlet T<sub>i</sub>=90°C  
Temperature outlet T<sub>o</sub>=70°C  
Volume flow rate  $\dot{Q}=40 \text{ m}^3/\text{s}$

**Heat Exchanger 2 (HX2):**

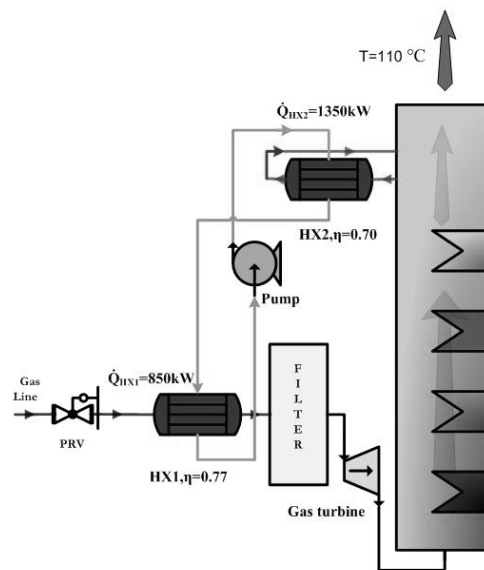
$\eta = 70\%$

**Flue Gas:**

Temperature inlet T<sub>i</sub>=7°C  
Temperature outlet T<sub>o</sub>=38°C  
Mass flow rate  $\dot{m}=191 \text{ kg/s}$

**Water:**

Temperature inlet T<sub>i</sub>=77°C  
Temperature outlet T<sub>o</sub>=93°C  
Volume flow rate  $\dot{Q}=40 \text{ m}^3/\text{s}$



**Figure 3 Specifications of the proposed system**

**Design of system [3]**

Air Cooled Heat Exchanger:

**Table 3 Water and Flue Gas Properties**

	<b>Water</b>	<b>Flue gases</b>
<b>Density(<math>\rho</math>)</b>	977.99 (kg/m <sup>3</sup> )	0.774(kg/m <sup>3</sup> )
<b>Dynamic viscosity(<math>\mu</math>)</b>	0.0004037(N.s/m <sup>2</sup> )	0.000025 (N.s/m <sup>2</sup> )
<b>Conductivity(k)</b>	0.659 (W/m.K)	0.0373(W/m.K)
<b>Prandtl number(Pr)</b>	1.963	0.686
<b>Mass flow rate(M)</b>	10.728(kg/s)	190(kg/s)

**Temperature Variance across the Year (Natural Gas)**

Since the temperature of the Natural Gas provided is not constant throughout the year, calculations have been performed to evaluate the corresponding amount of flue gas that would be needed to bleed. Calculations were also performed to evaluate the corresponding change in the water temperature

required to heat the natural gas. Table 4 shows the Natural Gas temperature provided and the corresponding temperature difference on the water side and corresponding mass flow rate of the flue gas that needs to be bled.

**Table 4 Analysis of Natural gas temperature and other parameters**

<b>Temp. of the Natural Gas supplied</b>	<b>Temp. difference required on water side</b>	<b>Mass flow rate of the flue gas required to be bled</b>
7 °C	17.55 °C	190.45 kg/s
8 °C	16.98 °C	184.30 kg/s
9 °C	16.42 °C	178.16 kg/s
10 °C	15.85 °C	172.01 kg/s
11 °C	15.28 °C	165.87 kg/s
12 °C	14.72 °C	159.73 kg/s
13 °C	14.15 °C	153.58 kg/s
14 °C	13.58 °C	147.44 kg/s
15 °C	13.02 °C	141.30 kg/s
16 °C	12.45 °C	135.15 kg/s
17 °C	11.89 °C	129.01 kg/s
18 °C	11.32 °C	122.871 kg/s
19 °C	10.75 °C	116.72 kg/s
20 °C	10.19 °C	110.58 kg/s
21 °C	9.62 °C	104.44 kg/s
22 °C	9.05 °C	98.29 kg/s
23 °C	8.49 °C	92.15 kg/s
24 °C	7.92 °C	86.00 kg/s
25 °C	7.36 °C	79.86 kg/s
26 °C	6.79 °C	73.72 kg/s
27 °C	6.22 °C	67.57 kg/s

Tube Dimensions:

The tube diameter is taken to be 1 in because increasing the diameter further results in an increase in the pressure drop for the gas side although the pressure drop on the water side reduces, but the gas side pressure drop is pretty high. Reducing the diameter results in better pressure drop on the gas side, but the pressure drop on the water side increases significantly which would increase the pumping requirements and thus our investment would increase.

**Table 5 Heat Exchanger Tube Dimensions**

D <sub>i</sub>	0.0194m
D <sub>o</sub>	0.0254m
t	0.003m
D <sub>fin</sub>	0.0514m
No of Tubes(N)	20

Fin Dimensions:

**Table 6 Heat Exchanger Fin Dimensions**

P1(V)	0.076m
P2(H)	0.088m
s	0.0055m
w	0.00127m
H	0.013m



### Water Side Heat Transfer Coefficient:

The velocity is [3]

$$v_w = \frac{\dot{M}_w}{20(\frac{\pi}{4})(D_i^2)(\rho_w)} \quad (17)$$

$$v_w = 1.855 \text{ m/s}$$

and

$$Re = \frac{v_w D_i \rho_w}{\mu_w} \quad (18)$$

$$Re_w = 8.72 \times 10^4$$

By Dittus-Boelter equation:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (19)$$

$$Nu_w = 2.7 \times 10^2$$

The heat transfer coefficient is given by:

$$\bar{h}_{w,i} = \left( \frac{Nu_w k_w}{D_i} \right) \quad (20)$$

$$\bar{h}_{w,i} = 9.18 \times 10^3 \text{ W/(m}^2 \cdot \text{K)}$$

### Water side pressure drop calculations [3]

$$\text{velocity} = \frac{\dot{m}_w}{(\frac{\pi}{4})(D_i^2)(\rho_w)N} = 1.855 \text{ m/s}$$

where,

$\dot{m}$  = mass flow rate of water = 10.728 kg/s,  $D_i$  = inside diameter = 0.0194m,  $\rho_w$  = density of water = 977.9 kg/m<sup>3</sup> and  $N$  = total number of tubes = 20

$$\text{Now } Re = \frac{\rho_w V D_i}{\mu} = 8.72 \times 10^4$$

where,

$$\mu = \text{Dynamic viscosity} = 4.037 \times 10^{-4} \text{ Pa.s}$$

$$\Delta p = 4f_p \frac{1}{D_i} \frac{V^2}{2} \quad (21)$$

$$\Delta p = 3.04 \times 10^{-2} \text{ bar or } \Delta p = 0.4468 \text{ psi}$$

To calculate the friction factor “f”, the following equation has been used: [3]

$$\frac{1}{\sqrt{4f_o}} = -2 \log \left( \frac{e}{3.7056 D_i} - \frac{5.0452}{Re} \log \left( \frac{1}{2.8257} \left( \frac{e}{D_i} \right)^{1.1098} + \frac{5.8506}{Re^{0.8981}} \right) \right) \quad (22)$$

for smooth and rough pipes in the range  $1 \times 10^4 < Re < 4 \times 10^8$

Where,

$$e = \text{relative roughness} = \frac{\varepsilon}{D_i} = 0.0073$$

$$\varepsilon = \text{roughness of tube} = 0.15 \text{ mm and therefore } f = \frac{f_0}{4} = 8.755 \times 10^{-3}$$

Pressure drop in bends:

$$Re = 8.72 \times 10^4$$

$$R = \text{radius of bend} = 0.038 \text{ m}$$

$$a = \text{inside radius of tube} = 0.0097 \text{ m}$$

$$\phi = 180^\circ$$

$$\Delta p = K \rho \frac{V^2}{2}$$

$$\Delta p = \left( \frac{4fL}{D_i} + K^* \right) \left( \rho \frac{V^2}{2} \right) \quad (23)$$

$$\Delta p = (0.3214) \left( \frac{977.99 \times 1.855^2}{2} \right) = 5.4 \times 10^{-3} \text{ bar}$$

$$\Delta p = 0.079 \text{ psi}$$

where,

$$K = \text{total loss coefficient} = \frac{4fL}{D_i} + K^*$$

$$K^* = \text{loss coefficient} = B(\phi) \left[ 0.051 + 0.38 \left( \frac{R}{a} \right)^{-1} \right]$$

$$B(\phi) = 0.7 + 0.35 \sin\left(\frac{\phi}{90}\right) = 0.712$$

$$K^* = 0.1054$$

The combined header and tube entrance losses:

$$\Delta p = \left( \frac{3\rho V^2}{2g_c} \right) (\text{No. of passes}) \quad (24)$$

$$\Delta p = 0.4 \text{ bar} = 5.94 \text{ psi}$$

Total pressure:

$$\text{Length of heat exchanger} = 29.9567 \text{ m}$$

$$\Delta p_t = (29.95 \times 0.446) + (0.079 \times 8) + 5.94$$

$$\Delta p_t = 19.956 \text{ psi} = 1.357 \text{ bar}$$

### Air-Side Heat Transfer Coefficient:

Velocity of approach  $v_{\text{approach}} = 11.20\text{m/s}$

The minimum flow area in the vertical plane of each row is given by:

$$S_{\min} = \eta_t L \left\{ p_1 - D_r - \frac{2wh}{w+s} \right\} \quad (25)$$

$$S_{\min} = 0.91\text{m}^2$$

The frontal area of the array is

$S_o = \text{length} \times \text{height}$

$$S_o = 1.558 \text{ m}^2$$

$$\sigma = \frac{S_{\min}}{S_o}$$

$$\sigma = 0.586$$

Velocity and Re of the air passing through the minimum cross sectional area:

$$v_{\max} = \frac{v_{\text{approach}}}{\sigma}$$

$$v_{\max} = 19.1\text{m/s}$$

$$\text{Re}_a = \frac{v_{\max} D_o \rho_a}{\mu_a}$$

$$\text{Re}_a = 1.5 \times 10^4$$

The air side heat transfer coefficient is obtained using ESDU correlation for high-fin staggered array heat exchangers:

$$\overline{\text{Nu}} = 0.242 \text{Re}^{0.658} \left(\frac{S}{h}\right)^{0.297} \left(\frac{p_1}{p_2}\right)^{-0.091} (\text{Pr})^{\frac{1}{3}} \quad (26)$$

$$\overline{\text{Nu}} = 93.57$$

The corresponding air side heat transfer coefficient is

$$\overline{h}_a = \frac{k_a}{D_r} \overline{\text{Nu}} \quad (27)$$

$$\overline{h}_a = 137.4 \text{ W/m.K}$$

The efficiency of high fins is given by:

$$\gamma = \frac{D_o}{2} \left( \frac{D_{\text{fin}}}{D_o} - 1 \right) \left( 1 + 0.35 \ln \frac{D_{\text{fin}}}{D_o} \right) \quad (28)$$

$$\gamma = 0.0162\text{m}$$

$$\eta_f = \frac{\tanh\left(\left(\sqrt{\frac{2\overline{h}}{wk_f}}\right)\gamma\right)}{\left(\sqrt{\frac{2\overline{h}}{wk_f}}\right)\gamma} \quad (29)$$

$$\eta_f = 0.9167$$

The high fins are selected because it gives optimum heat transfer with minimum pressure drop. The total surface area of the finned tubes is given by:

$$A = A_F + A_w$$

$$A = \frac{NL\pi}{(s+w)} \left\{ \frac{1}{2} (D_{fin}^2 - D_o^2) + D_{fin}W + D_o s \right\} \quad (30)$$

The surface area of the fins is given by:

$$A_F = \frac{NL\pi}{(s+w)} \left\{ \frac{1}{2} (D_{fin}^2 - D_o^2) + D_{fin}W \right\}, A_F = 9.87 \text{ m}^2$$

And the surface area of the tubes between the fins is given by:

$$A_w = \frac{NL\pi}{(s+w)} (D_o s), A_w = 1.29 \text{ m}^2$$

Therefore,  $A=11.168\text{m}^2$  and the total external surface area of the tube without fins is:

$$A_T = NL\pi L_o$$

$$A_T = 1.59 \text{ m}^2, \text{ therefore } \frac{A}{A_T} = 7$$

The effective air-side heat transfer coefficient based on the total surface area is given by:

$$\bar{h}_a = \left( \frac{\eta_f A_F + A_w}{A} \right) \quad (31)$$

$$\bar{h}_a = 127.3 \text{ W}/(\text{m}^2 \cdot \text{K})$$

The air side heat transfer coefficient referred to the external surface of the tube without fins is:

$$\bar{h}_{a,r} = \bar{\alpha}_a \times \frac{A}{A_t} \quad (32)$$

$$\bar{h}_{a,r} = 890.95 \text{ W}/(\text{m}^2 \cdot \text{K})$$

#### Overall Heat Transfer Coefficient:

$U_r$  is related to the external surface of the tube without fins

$$\frac{1}{U} = \frac{1}{\bar{h}_{a,r}} + \frac{D_o}{2k_f} \ln \frac{D_o}{D_i} + \frac{1}{h_{w,i}} \frac{D_o}{D_i} \quad (33)$$

$$\frac{1}{U} = 0.00128$$

$$U = 780.233 \text{ W}/(\text{m}^2 \cdot \text{K})$$

#### Air side pressure drop:

$$\sigma = \frac{p_1 - D_o - 2hw/(w+s)}{p_1} \quad (34)$$

$$\sigma = 0.602$$

$$K_a = 1 + \sigma^2$$

$$K_a = 1.362$$

$$K_f = 4.71 \text{Re}^{-0.286} \left(\frac{h}{s}\right)^{0.51} \left(\frac{p_1 - D_o}{p_2 - D_o}\right)^{0.536} \left(\frac{D_o}{p_1 - D_o}\right)^{0.36} \quad (35)$$

$$K_f = 0.325$$

$$\Delta p = (k_a + n_r K_f) \frac{1}{2} \rho v_{\max}^2$$

$$\Delta p = 5.58 \times 10^{-3} \text{ bar or } \Delta p = 0.0809 \text{ psi}$$

Heat Transfer Area:

$$Q = 1.35 \text{ MW}$$

$$Q = U A d T_m$$

$$A = \frac{Q}{U d T_m}, \text{ therefore } A = 47.8 \text{ m}^2$$

Total length of tubes is given by:

$$L = \frac{A}{\pi D_o N} \quad (36)$$

$$L = 29.96 \text{ m}$$

As,

Number of passes = 8

Therefore,

Length of heat exchanger = 3.74 m

**Table 7 Heat Exchanger Temperatures**

$T_{h,i}$	165°C
$T_{h,o}$	110°C
$T_{c,i}$	70°C
$T_{c,o}$	90°C
$dT_m$	55.68°C
$dT_m@R=2.75$	36.19°C

Effectiveness of heat exchanger:

$$PR = \frac{T_{w,in} - T_{w,out}}{T_{w,in} - T_{a,in}} \quad (37)$$

$$PR = 0.73$$

where,

$$T_{w,in} = 70^\circ\text{C}$$

$$T_{w,out} = 90^\circ\text{C}$$

$$T_{a,in} = 165^\circ\text{C}$$

#### 4. COST ANALYSIS

##### Cost of Heat Exchanger

For 1 ft<sup>3</sup> of heat exchanger surface area the cost is \$ 118.76

The total heat exchanger surface area is  $47.8 \text{ m}^2 = 514.514918 \text{ ft}^2$

The cost of the heat exchanger comes out to be  $\$ 61,103.80 \times 2 = \$ 122,208$

### **Cost of Piping**

Total length of the pipe is 280 m. the pipe is a 10 in SS pipe. [4]

The cost of the pipe = 400 euro /meter

Total cost of the piping =  $400\text{€}/\text{m} \times 1.3753\text{\$/€} \times 280\text{m} = \$ 154,034$

### **Cost of Insulation**

#### Cost of Aluminum Jacket:

Cost of Aluminum jacket =  $\$5.28/3\text{ft} = \$5.28/0.9144 \text{ m}$  [5]

Total cost of the metal jacket =  $5.28 \times 280/0.9144 = \$ 1,617$

#### Cost of Asbestos:

Cost of Asbestos unavailable, so using fiberglass of a thickness having the same thermal conductivity.

Cost of Fiberglass =  $18.98 / 3 \text{ ft} = \$18.98 / 0.9144 \text{ m}$  [6]

Total cost of fiberglass =  $18.98 \times 280/0.9144 = \$ 5,812$

#### Cost of fiber glass:

Cost of Fiber Glass =  $\$18.98 / 3 \text{ ft} = \$18.98 / 0.9144 \text{ m}$  [6]

Total cost of Fiber glass =  $18.98 \times 280/0.9144 = \$ 5,812$

#### Total Cost of Insulation:

Total cost of insulation = \$ 13,241

### **Cost of Gas**

The gas cost per 1MMBTU is Rs. 6646.43 = \$ 78.933

The total savings in fuel is  $129 + 120 \text{ ft}^3 = 249 \text{ ft}^3/\text{hr} = 0.249 \text{ MMBTU/h}$

The total savings of fuel per year is gas flow rate /hour  $\times$  no. of days of operation  $\times$  24 hours /day =  $249 \times 329 \times 24 = 2,025,864 \text{ ft}^3 = 2,025.864 \text{ MMBTU}$

## **5. PAYBACK PERIOD**

Cost of natural gas = \$ 78.933

Total savings per annum = \$ 159,907.5 per annum

Total cost for installation of the proposed system = \$ 61,103.80\*2 + cost of pump + \$ 154,034 + Cost of second heat exchangers + \$ 13,241= \$ 289,282.6 + approx. \$200,000.

Payback period for the proposed system = Total cost (Million of Rupees)/ annual savings (Million of Rupees) \$ .5 million/ \$ .160(/annum) = 3 years.

## CONCLUSION

The objectives of determining feasibility of recovering waste heat from the exhaust of a combined cycle power plant and utilizing it on site has been successfully achieved. The exergy of the flue gases has been calculated by first determining the dew point of the flue gas in order to determine the lower limit to which the temperature could be decreased. The dew point of the flue gas was found to be 98°C. The lower limit resulted in determining the amount of heat that could be recovered as well as the potential of the waste heat i.e., a total of 6.67 MW. The next step was to identify a use on site; the use identified was to preheat the natural gas being used as a fuel. The amount required for heating the natural gas was 653 kW. After incorporating all the efficiencies the amount that needed to be extracted from the exhaust of the HRSG was determined to be 1.35 MW from each HRSG exhaust. The next step was to design a system to successfully recover the waste heat and use it on site. For this, some changes were suggested in the existing system and the required heat exchanger was designed to meet the purpose. The heat transfer surface area was found to be about 48 m<sup>2</sup>. From a total available potential of 6.67 MW, 2.7 MW is proposed to be utilized in order to replace the existing gas preheating system. The amount of cost that could thus be saved is Rs. 13,464,763. The payback period worked out for the investment was evaluated to be approximately 3 years.

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