EVALUATION OF HEAT TRANSFER ON AIR RECUPERATOR

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ABSTRACT-Industrial waste heat refers to energy that is generated in industrial process without being put to practical use. It is estimated that somewhere between 20 to 50% industrial energy input is lost as waste heat in the form of hot exhaust gases, cooling water, heat lost from hot equipment surface and heated products. While some waste heat losses from industrial processes are inevitable. Facilities can reduce these losses by improving equipment efficiency or installing waste heat recovery technologies.

In the steel making process, Rolling mills play a vital role in transforming the steel to finished Products. Rolling mills at Visakhapatnam Steel Plant consists of Light and medium merchant mill (LMMM), Wire Rod Mills (WRM) and Medium Merchant Structural Mills (MMSM). Air Recuperator is present in Medium Merchant Structural Mills (MMSM) section of the plant for the purpose of waste heat recovery.

In this Medium Merchant Structural Mills (MMSM) section, air is pre-heated using the flue gas which enters the recuperator at high temperature from furnace. The pre-heated air is used in the furnace for heating the blooms. Volume flow rates and temperature of both flue gas and air are taken as input data. Heat transfer coefficient, friction factor and pressure drop on both air side and flue gas are estimated effectiveness of air recuperator is calculated.

Graphs are drawn to show the performance of air recuperator the data arrived from the calculations with varying flow rates of air and flue gas are consolidated as table and put in appendix.

Key words: recuperator, volume flow rate, heat transfer coefficient , friction factor, pressure drop.

I INTRODUCTION

Air Recuperator:

The flue gases coming out of the furnace are sent through the air recuperator so that the excess heat energy from the flue gases can be absorbed by the incoming air. The recuperator is shell and tube type cross-flow heat exchanger. Waste gases which are to be circulated in the furnace in a direction opposite to that of bloom movement will be exhausted through the openings provided at the charging end, after passing through the recuperative zone. Products of combustion, thereafter, will pass through the flue channel leading to chimney with an appreciable quantity of heat. The higher the temperature of gases, the lower will be the coefficient of heat utilization in the furnace. Therefore this heat has to be recovered, for which an air recuperator is installed in the flue channel to preheat the combustion air. **Sarma et al. (1)** has analyzed laminar convective heat transfer enhancement using twisted tapes in tubes of thermal devices in particular. **Sarma et al. (2)** has discussed about momentum and thermal eddy diffusivities in tubes that creates turbulent flow. In this paper the above parameters are evaluated.

Shah and Sekulic [1] in their book provided an introduction to direct transfer type exchangers. They gave detailed information of recuperative heat exchangers, basic thermal design and theory for recuperator. **Incropera [2]** in his book discussed the internal flow in circular tubes. He discussed fully developed conditions for turbulent flow. **Cengel [3]** in his book provided flow across the bank of tubes. He had discussed about different tube arrangements. **Thirumaleswar [4]** in his book provided flow across the bank of tubes. He discussed various flow regimes and relation of friction factor. **Rao[5]** in his paper discussed air preheater as a heat exchanger device designed to heat air before another process (for example, combustion in a boiler) with the primary objective of increasing the thermal efficiency of the process. The purpose of the air preheater is to recover the heat from the boiler flue gas which increases the thermal efficiency of the boiler by reducing the useful heat lost in the flue gas. In his paper he mainly dealt with design, modeling and fabrication and CFD analysis of a shell and tube air preheater. In this paper overall heat transfer coefficient of the shell and tube heat exchanger is based on the results of effectiveness-NTU approach and LMTD approach. The results showed the complete presentation of thermal and mechanical design. **Suresh et al. [6]** in their paper conducted theoretical performance of the air heater to increase the thermal efficiency of the process. For every 200^o C drop in the flue gas exit temperature, the boiler efficiency increased by 1%. The air heater performance procedure provided a systematic approach for conducting routine air heater performance tests on tubular and rotary regenerative air heater. Various performances indices like air heater leakage, gas-side efficiency, x-ratio etc are determined using this procedure. **Karamarkovi et al. [7]** in their paper presented a mathematical model that defines the geometry of the heat exchanger so as the heat transfer from the kiln to the combustion air is to be equal to the heat dissipated from the bare kiln. The exchanger decreases fuel consumption of the kiln for 12.00%, and increases its energy and exergy efficiency by 7.35% and 3.81%, respectively. They also stated that to obtain a better performance, the air flow and geometry of the exchanger should be arranged to achieve the smallest possible temperature difference between the kiln surface and the preheating air, whose amount should always be kept at the optimal value for the used fuel. **Oyelami and Adejuyigbe [8]** in their article studied the design of a radiation-recuperative heat exchanger for a 200 kg capacity rotary furnace. A new design of recuperator, which is a heat exchanger, incorporated into the rotary furnace is done in such a way that it returns substantial part of the waste heat back into the furnace. A radiation type recuperator was designed to achieve this objective. It was established that the design results in 34.78% increase in the thermal efficiency of the furnace system and also leads to 44.4% reduction in the fuel consumption. **Gui et al. [9]** conducted experimentally a three dimensional periodic numerical model for fully developed flow in a cross wavy primary surface recuperator for a microturbine system. The performance of flow and heat transfer is analyzed. The fields of flow and temperature in a gas and air channel are obtained. Different working conditions are numerically simulated. Numerical results are compared with experimental data. Results showed that the flow in the gas and air channel is axis-symmetry along the center of channel. The flow of fluid is fluctuant. The flow velocity of gas is much higher than that of air. The thermal ratio of cross wavy primary surface recuperator can reach 95.2%. The thermal ratio decreases with the improvement of gas inlet temperature. When gas inlet temperature increases by 100K, the thermal ratio is found to decrease by about 1%. The thermal ratio increases with the reduction of flow rate in the channel. When flow rate reduces by 40%, the thermal ratio increases by about 4%. The research results can be used to guide and check the performance of a recuperator. **Akbari and Oman [10],** in their work investigated the impact of a heat recovery ventilator (HRV) on the energy use and indoor radon in residential buildings. In their study, a multi zone model of a detached house is developed in IDA Indoor Climate and Energy (IDA ICE 4.0). The model is validated using measurements regarding use of energy for heating, ventilation and whole energy. The result shows that the measurements and dynamic simulation increases by 74% energy savings of the ventilation loss, amounted about $30kWh.m^{-2}$ per year.

II) DESCRIPTION AND WORKING OF RECUPERATOR

A. Recuperator

Recuperator is a special purpose cross flow energy recovery heat exchanger positioned within the supply and exhaust air streams and flue gas system in order to recover the waste heat.

***B.* Description of Recuperator**

Recuperator Fig 1 consists of straight tubes and bent tubes. Inner and outer diameter of the tubes are 46mm and 50.8mm respectively. Inner diameter of the tube is made of carbon steel seamless, whereas outer diameter of the tubes is made of ferric stainless steel which are aligned in the direction of fluid flow.

It is a cross flow heat exchanger in which the two fluids (hot and cold) cross one another in space at right angles. Cold fluid (air) passes through separate tubes and there is no mixing of fluids. Hot fluid (flue gas) is perfectly mixed as it flows through the shell.

Recuperator is a shell and tube, two tube pass heat exchanger. There are 440 tubes in each pass. Cold fluid (air) passes through the tube and hot fluid (flue gas) passes through the shell. The inlet tube (one pass) is bent in order to increase the length of air flow.

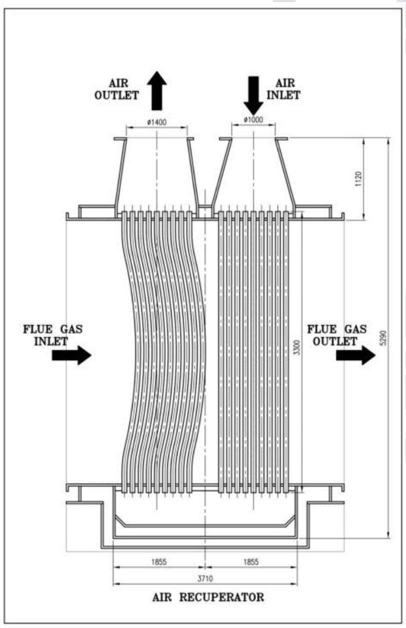


Fig 1Air Recuperator

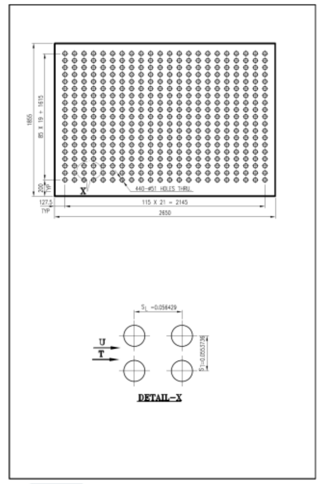


Fig 2 inline alignment of tubes

***C.* Working of Recuperator**

The Fig 2 gives process flow of flue gas and air in air recuperator. The flue gas which enter air recuperator at high temperature losses it heat to air i.e. the tubes of air recuperator by the convection principle. The air and flue gas flow cross section to each other which allows flue gas to have high contact area and high residence time for convection to take place effectively. Tubes arranged in air recuperator are in inline configuration. Spacing between two centers of tubes is called pitch, there are two types of pitch one is longitudinal pitch and other is transverse pitch. The overall longitudinal length for single pass is 1855 mm and the overall transverse length is 2650 mm.

The Air Recuperator behaves as two pass. The effectiveness of a recuperator varies with temperature of both air and flue gas.

III ANALYSIS OF THE RECUPERATOR

***A.* ASSUMPTIONS:**

(1) Recuperator is treated to be a heat exchanger under steady state.

(2) The wall resistance and fouling factors are negligible

(3) Fully developed conditions at tube outlet.

(4) Bent tubes are considered as straight tubes.

CALCULATIONS OF FLUE GAS PROPERTIES**:**

Heat capacity rate of flue gas is calculated as (C_h)

$C_h = [\frac{C_{c(T_{co}-T_{ci})}}{T_{hi}-T_{ho}}]$ (1)

where,

$C_h = m_h C_{ph}$ = heat capacity rate for hot fluid, $(\frac{m^3}{hr} \times \frac{kJ}{kgK})$

m_h=mass flow rate of hot fluid (flue gas), (kg/s)

C_{ph} = Specific heat of hot fluid (flue gas), (kJ/kgK)

$C_c = m_c C_{pc}$ =capacity rates for cold fluid, $(\frac{m^3}{hr} \times \frac{kJ}{kgK})$

m_c = mass flow rate of cold fluid (air), (kg/s)

C_{pc}= specific heat of cold fluid (air side), (kJ/kgK)

Effectiveness is calculated as $(\in)$

$\in = \frac{C_c(T_{co}-T_{ci})}{C_{min}(T_{hi}-T_{ci})}$ (2)

C_{min}= minimum heat capacity rate of fluid, $(\frac{m^3}{hr} \times \frac{kJ}{kgK})$

Specific heat of hot fluid is calculated as (C_{ph})

$C_{ph} = \frac{m_c C_{pc}(T_{co}-T_{ci})}{m_h(T_{hi}-T_{ho})}$ (3)

Density of flue gas (ρ_f) is got by using the heat balance equation.

$$m_h C_{ph}(T_{hi} - T_{ho}) = m_c C_{pc}(T_{co} - T_{ci})$$

$$\rho_f = \frac{V_c C_{Pc}(T_{co} - T_{ci})\rho_c}{V_h C_{Ph}(T_{hi} - T_{ho})} \quad (4)$$

ρ_f= density of flue gas, (kg/m^3)

V_c =volume flow rate of cold fluid i.e air, (m^3/hr)

ρ_c= density of cold fluid i.e air, (kg/m^3)

V_h =volume flow rate of hot fluid i.e flue gas, (m^3/hr)

***B.*CALCULATIONS OF HEAT TRANSFER COEFFICIENT ON AIR SIDE**

Cross sectional area of tubes (A) = number of passes×number of tubes in each pass × cross sectional area of tube

$A_c = N_1 \times N_2 \times \frac{\pi}{4} D_i^2$

N_1=number of passes

N_2=number of tubes in each pass,

D_i= inner diameter of the tube, m

Mass velocity, (G) = $\frac{W}{A}$

Where,

W=air flow rate, kg/s

A_c=cross sectional area of tubes, m^2

Reynolds number of air (Re_a) is expressed as

$$Re_a = \frac{D_i G}{\mu_a} \quad (5)$$

Where,

μ_a =viscosity of air, kg/m-s

From Dittus Boelter equation, Nusselt number of air (Nu_a) is taken as

$Nu_a = 0.023 Re_a^{0.8} Pr_a^{0.4}$ (6)

Where,

Pr_a=Prandtl number of air

Heat transfer coefficient on air side is calculated as

Heat transfer coefficient on air side (h_a)

$h_a = \frac{Nu_a \times k_a}{D_i}$ (7)

Where,

k_a=thermal conductivity of air, W/m k

Friction coefficient is expressed as

Friction coefficient of air (f) =0.046 $Re^{-0.2}$ (8)

Friction factor = 4*friction coefficient

Pressure drop of air inside the tube is expressed as

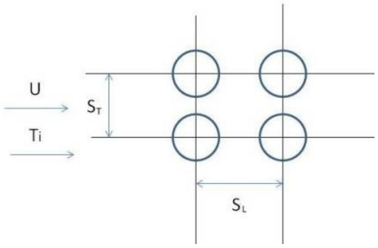
Pressure drop$(\Delta P) = \frac{4fLG^2}{2\rho_a D_i}$ (9)

Where,

L=length of the tube, m

ρ_a = density of air, kg/ m^3

C. CALCULATIONS OF HEAT TRANSFER COEFFICIENT ON FLUE GAS SIDE:



The above figure shows square type tube layout as inline arrangement

Cross sectional area (A) is calculated as follows:

$$A= N_T S_T L \qquad (10)$$

Where,

A= free flow area

N_T =Number of tubes in the transverse direction

S_T =Transverse pitch

L =length of the tubes, m

Free stream velocity (U) is calculated as

$$U =\frac{m_f}{\rho_f A} \qquad (11)$$

where,

m_f = mass flow rate of flue gas, kg/s

ρ_f =density of flue gas, kg/ m^3

Maximum velocity(U_{max}) is calculated as

$$U_{\max} = \frac{S_T}{S_T - D_O} U \qquad (12)$$

D_o =outer diameter of tube, m

Reynolds number is calculated as

$$Re_f = \frac{\rho_{f} \times U_{max} \times D_o}{\mu_f} \qquad (13)$$

where,

μ_f = viscosity of flue gas, kg/m-s

ρ_f =density of flue gas, kg/ m^3

Nusselt number (Nu_f) is calculated as

For transition regime (i.e1000< $\mathrm{Re}_f < 2\times10^5$)

$$Nu_f=0.27(\mathrm{Re}_f)^{0.63} (Pr_f)^{0.36} (\frac{Pr}{Pr_w})^{0.25}$$

Neglecting $(\frac{Pr}{Pr_w})^{0.25}$term, Nusselt Number is given as

$$Nu_f=0.27(\mathrm{Re}_f)^{0.63} (\mathrm{Pr})^{0.36} \qquad (14)$$

Nu_f =Nusselt number of flue gas

Heat transfer coefficient of flue gas is calculated as (h_f)

$$h_f = \frac{Nu_f k_f}{D_o} \qquad (15)$$

k_f =thermal conductivity of flue gas

Friction factor of flue gas is calculated as (f)

$$f= \left[0.044 + \frac{0.08 \times (\frac{S_L}{D})}{\left(\frac{S_T - D}{D}\right)^{0.43+1.13\frac{D}{S_L}}} \right] \times \mathrm{Re}_f^{-0.15} \qquad (16)$$

IV RESULTS AND DISCUSSIOND

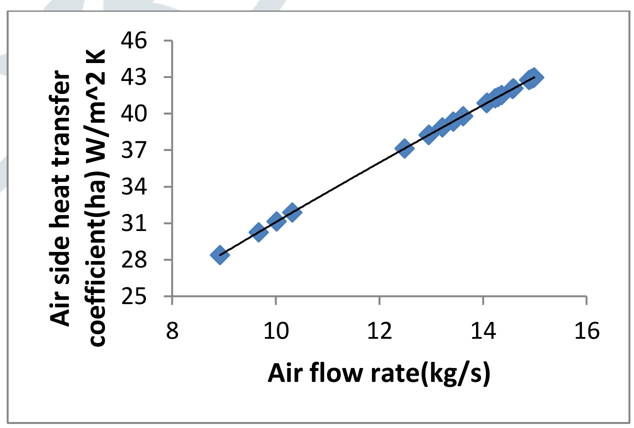


Fig 3 Air side heat transfer co-efficient with air flow rate

The above fig.3 shows the variation of air side heat transfer coefficient with respect to airflow rate through the recuperator as air flow rate increases heat transfer coefficient increases. It is seen that heat transfer coefficient is calculated using amount of heat transfer and temperature difference between the inlet and outlet of the air. The amount of heat transferred also depends on the surface area and the residence time of air.

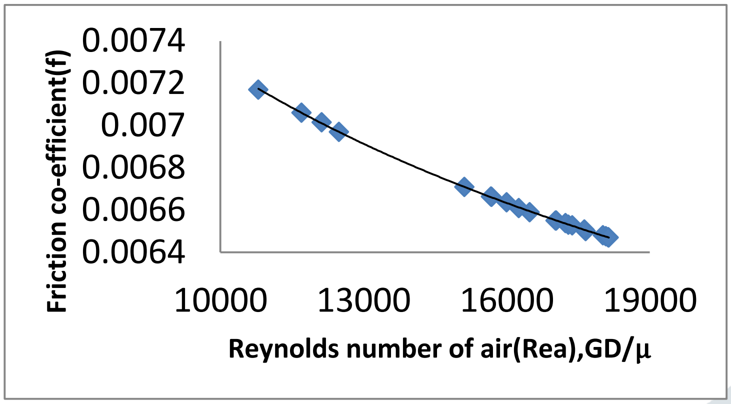


Fig.4 Friction factor with Reynolds number of air

The above Fig .4 shows the variation of friction coefficient with Reynolds number of air. The results show that the friction co-efficient decreases with increase in Reynolds number of air. Friction factor of air is directly proportional to the residence time through the tubes of recuperator with increase in flow rates of air residence time decreases.

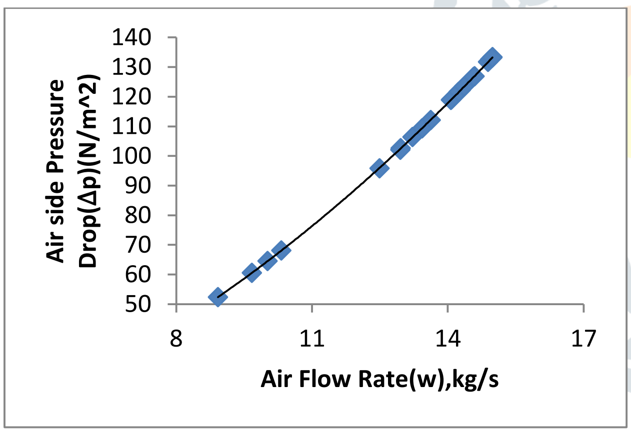


Fig 5 Air side pressure drop with air flow rate

The above Fig.5 is plotted for air side pressure drop with respect to air flow rate. With the increase in air flow rate the pressure drop across the tubes of the recuperator increases. Both pressure drop and air flow is linearly proportional to flue gas flow rate. As there is temperat erature difference across the tubes the pressure drop increases with the increase in temperatures across the tubes of air recuperator.

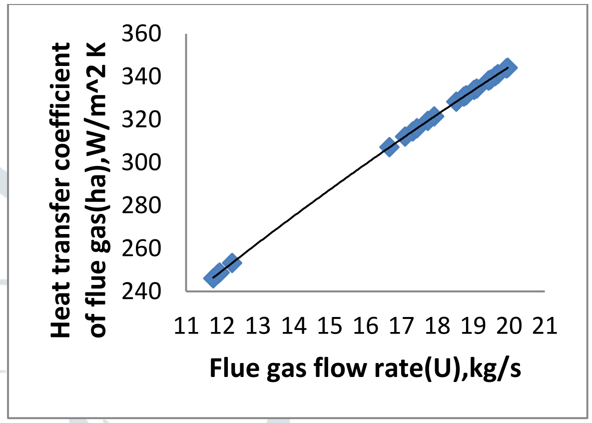


Fig 6 Heat transfer co-efficient o with flue gas flow rate

The above fig. 6 shows a variation of Heat Transfer Co-efficient of Flue gas with respect to Flue gas flow rate through recuperator. Heat Transfer Co-efficient increase with the increase in flow rate of flue gas. Heat Transfer Co-efficient of flue gas is also effected by the cross sectional area with increase in cross sectional area the heat transfer coefficient of flue gas also increases

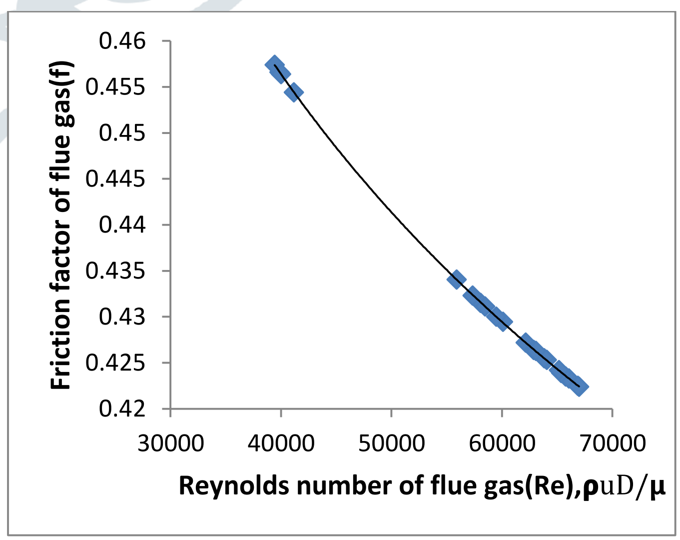


Fig 7 Friction factor of flue gas with Reynolds number of flue gas

The above graph 7 plotted shows the trend between friction factor of flue gas with Reynolds number of flue gas .The frictional factor of flue gas is calculated using values of longitudinal Pitch, transverse pitch, outer diameter of tubes and Reynolds number of flue gas .The friction factor of flue gas decreases with the increase in Reynolds Number of flue gas.

V CONCLUSIONS:

In this work analysis is carried out on heat recovery taking air and flue gas. The heat transfer co-efficient, friction factor and pressure drop of air is calculated under turbulent flow condition and heat transfer co-efficient, friction factor of flue gas is studied under transition flow condition.

Based on the above, the following conclusions are arrived

1. For increase in mass flow rate of air from (8.92-14.99 kg/s) heat transfer co-efficient of air side increases by 33.96%
2. Friction factor is effected by mass flow rate of air. Hence with an increase in Reynolds number of air from (21575.22-36283.18), friction co-efficient decreases by 9.768%
3. For increase in air flow rate from (8.923-14.99 kg/s) pressure drop across the tube of air increases by 60.73%
4. For increase in mass flow rate of flue gas from (11.76-19.94 kg/s) heat transfer co-efficient of flue gas increased by 28.37%.
5. Friction factor is effected by mass flow rate of flue gas. Hence with an increase in Reynolds number of flue gas 38595.75-65427.59 friction factor decreased by 7.65%
6. Effectiveness of the recuperator is maintained in between 0.411 and 0.572

NOMENCLATURE

A_c	Cross sectional area, (m^2)
A_f	free flow area, (m^2)
C	heat capacity rate, ($\frac{m^3}{hr} \times \frac{KJ}{kgK}$)
c_p	Specific heat at constant pressure, (J/kg K)
D	Diameter, (m)
Δp	pressure drop, (N/ m^2)
f	friction factor
G	mass velocity, (kg/s m^2)
h	heat transfer coefficient, (W/ m^2 K)
k	thermal conductivity, (W/mK)
L	length of the tube, (m)
m	mass flow rate, (kg/s)
N	number of tubes
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
S	pitch
T	fluid temperature, (^oC)
U	free stream velocity, (m/s)
W	air flow rate, (kg/s)

GREEK SYMBOLS

$\in$	effectiveness
μ	dynamic viscosity, (Ns/m^2)

υ	kinematic viscosity, (m^2/s)
ρ	density, (kg/m^3)

SUBSCRIPTS

a	air
b	bulk mean
c	cold fluid
F	free flow area
f	flue gas
h	hot fluid
l	longitudinal
max	maximum
t	transverse
w	wall temperature
1	number of passes
2	number of tubes in each pass

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