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RESEARCH ARTICLE OPEN ACCESS

# Improving the Overall heat transfer coefficient of an Air Preheater by Design, Fabrication and CFD Analysis

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**ABSTRACT**
An air preheater is 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. This project mainly deals with design, modeling and fabrication and cfd analysis of a shell and tube air preheater. Over all heat transfer coefficient of the shell and tube heat exchanger is based on the results of effectiveness-ntu approach and lmtd approach. Drawing of various components will be presented with the help of various software's like solid works, proe, etc., even different experimental results and trails will be analyzed and tabulated. Conclusion of the project will be the complete presentation of thermal and mechanical design, fabrication model, Overall heat transfer coefficient and cfd (computational fluid dynamics) analysis for the air preheater.
***Keywords***: air preheater, cfd, fabrication .

## I. INTRODUCTION

An air pre-heater is a general term to describe any 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. They may be used alone or to replace a recuperative heat system or to replace a steam coil. In particular, this article describes the combustion air pre-heaters used in large boilers found in thermal power stations producing electric power from e.g. fossil fuels, biomasses or waste.

The purpose of the air pre-heater 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. As a consequence, the flue gases are also sent to the flue gas stack (or chimney) at a lower temperature, allowing simplified design of the ducting and the flue gas stack. It also allows control over the temperature of gases leaving the stack.There are two types of air pre-heaters for use in steam generators in thermal power stations. One is a tubular type built into the boiler flue gas ducting, and the other is a regenerative air preheater. These may be arranged so the gas flows horizontally or vertically across the axis of rotation. Another type of air preheater is the Regenerator used in iron or glass manufacture.

## II. SCOPE OF WORK

After studying the journal papers mentioned above, it is understood that there are some gaps in development of shell and tube air preheater. The project is planned with the following work scope, modeling and fabrication of shell and tube air preheater including thermal design.

## III. THEORETICAL BACKGROUND & CONSTRUCTION FEATURES

Let us see each one it their detail with their basic construction, working and related problems A bundle of vertical tubes through which the flue gas flows downward and exchanges heat with ambient air flowing horizontally across the exterior of the tubes. Ref Fig3

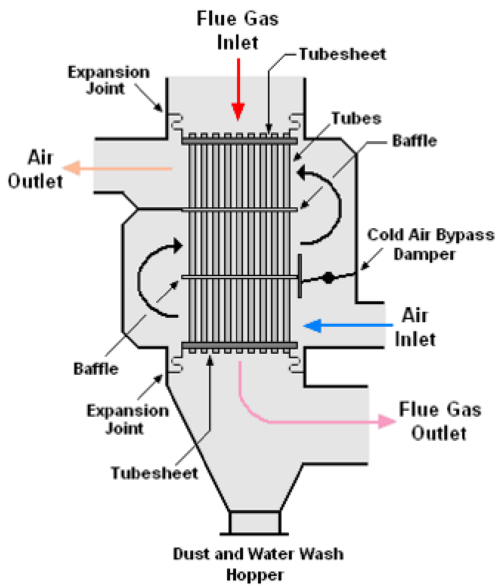



Figure : Tubular Type Air Preheater

The air enters the lower tube bundle from the right-hand side, exits on the left-hand side and then enters the middle tube bundle on the left-hand side and exits on the right-hand side. Finally, the air enters the upper tube bundle on the right-hand side and exits on the left-hand side. In essence, such a design is similar to the 3-pass design of (1) above except that the air is in the tubes rather than outside the tubes. Tubular preheaters consist of straight tube bundles which pass through the outlet ducting of the boiler and open at each end outside of the ducting. Inside the ducting, the hot furnace gases pass around the preheater tubes, transferring heat from the exhaust gas to the air inside the preheater. Ambient air is forced by a fan through ducting at one end of the preheater tubes and at other end the heated air from inside of the tubes emerges into another set of ducting, which carries it to the boiler furnace for combustion.

## IV. PROBLEMS

The tubular preheater ducts for cold and hot air require more space and structural supports than a rotating preheater design. Further, due to dust-laden abrasive flue gases, the tubes outside the ducting wear out faster on the side facing the gas current. Many advances have been made to eliminate this problem such as the use of ceramic and hardened steel.

## V. METHODOLOGY

In all of these approaches the same basic procedure is followed.

- During preprocessing
- The geometry (physical bounds) of the problem is defined.
- The volume occupied by the fluid is divided into discrete cells (the mesh). The mesh may be uniform or non uniform.
- The physical modeling is defined – for example, the equations of motions + enthalpy + radiation + species conservation
- Boundary conditions are defined. This involves specifying the fluid behaviour and properties at the boundaries of the problem. For transient problems, the initial conditions are also defined.
- The simulation is started and the equations are solved iteratively as a steady-state or transient.
- Finally a postprocessor is used for the analysis and visualization of the resulting solution.

## VI. DISCRETIZATION METHODS

The stability of the chosen discretization is generally established numerically rather than that of analytically as with simple linear problems. Special care must also be taken to ensure that the discretization handles discontinuous solutions gracefully. The Euler equations and Navier–Stokes equations both admit shocks, and contact surfaces.

Some of the discretization methods being used are:

FINITE VOLUME METHOD.The finite volume method (FVM) is a common approach used in CFD codes, as it has an advantage in memory usage and solution speed, especially for large problems, high Reynolds numberturbulent flows, and source term dominated flows (like combustion). In the finite volume method, the governing equations partial differential equations (typically the Navier-Stokes equations, the mass and energy conservation equations, and the turbulence equations) are recast in a conservative form, and then solved over discretecontrolvolumes.

This discretization guarantees the conservation of fluxes through a particular control volume. The finite volume equation yields governing equations in the form,

$$\frac{\partial}{\partial t}\iiint Q\,dV + \iint F\,d\mathbf{A} = 0,$$

where $Q$ is the vector of conserved variables, $F$ is the vector of fluxes (see Euler equations or Navier–Stokes equations), $V$ is the volume of the control volume element, and $\mathbf{A}$ is the surface area of the control volume element.

## VII. DESIGN CALCULATIONS

The optimum thermal design of a shell and tube heat exchanger involves the consideration of many interacting design. Parameters which can be summarized as follows:

**Process:-**

1. Process fluid assignments to shell side or tube side.
2. Selection of stream temperature specifications.
3. Setting shell side and tube side pressure drop design limits.
4. Setting shell side and tube side velocity limits.
5. Selection of heat transfer models and fouling coefficients for shell side and tube side.

**Mechanical:-**

1. Selection of heat exchanger TEMA layout and number of passes.
2. Specification of tube parameters - size, layout, pitch and material.
3. Setting upper and lower design limits on tube length.
4. Specification of shell side parameters – materials, baffle cut, baffle spacing and clearances.
5. Setting upper and lower design limits on shell diameter, baffle cut and baffle spacing.

## VIII. THERMAL DESIGN OF AIR PREHEATER

The aim of the following design is to calculate overall heat transfer coefficient through LMTD method. This is done by assuming the exit temperatures of hot side and cold side. Fluid Flow through Shell is air and exhaust gas from the diesel engine flows through pipes.

INPUT DATA:-

- Inlet temperature of air ($t_1$)= 33 **°C** (ambient)
- Outlet temperature of air ($t_2$) = 52.51 **°C** (calculated from heat balance)
- Inlet temperature of gas ($T_1$) = 85 **°C** (measured)
- Outlet temperature of gas ($T_2$)= 5**°C**(say)
- Tube inner diameter ($d_i$)= 19.05mm =0. 01905m
- Tube Outer diameter ($d_o$) = 22mm = 0.022m
- Shell Inner diameter ($D_i$)= 220mm = 0.22m
- No of tube (n) = 15

CALCULATION OF AIR MASS FLOW RATE ($m'_a$) AND AIR VELOCITY:

- **Volumetric flow rate** (Qa) = 2.3 $m^3$ /min = **0.0383** $m^3$/s ( From Blowerspecification)
- Air flow cross sectional area (Aa) = shell c/s – n* π/4*$d^2$**$m^2$**
- **Air velocity** (Va) = Volumetric flow rate (Qa)/ Air flow cross sectional area(Aa) = 0.0383/0.0328 = **1.167 m/s**
- **Mass flow rate of air ( $m'_a$ )** = Volumetric flow rate * density
- Density of air (from heat transfer data book) at 33°C **=** 1.11539 kg/$m^3$.
- Hence the **Mass flow rate of air ( $m'_a$)= .044 kg/s**
- **Air flow cross sectional area** (Aa)= **0.0328**

CALCULATION OF GAS MASS FLOW RATE AND GAS VELOCITY

Mass flow rate of gas side: It is calculated by experiment on diesel engine silencer the procedure is the exhaust gases in the silencer are cooled by water and reading are tabular

Table : Exhaust gas mass flow rate

| S.No. | Volumetric flow rate V cc/s | Twi**°C** | Two**°C** | Tgi**°C** | Tgo**°C** | Mw Kg/s | Mg Kg/s |
|---|---|---|---|---|---|---|---|
| 1. | V1=80 | 28.7 | 30.7 | 116.7 | 42 | 80* $10^{-3}$ | 4.26* $10^{-3}$ |
| 2. | V2=60 | 28.7 | 38 | 128 | 47 | 60* $10^{-3}$ | 13.7* $10^{-3}$ |
| 3. | V3=40 | 28.7 | 41 | 135 | 54 | 40* $10^{-3}$ | 12.09* $10^{-3}$ |

**FROM HEAT BALANCE:**

Heat lost by gas is equal to heat gained by water.

Hence,

- mg*Cp*ΔTg= mw*Cp*ΔTw
- mw1= ρ*V1=80* $10^{-3}$ Kg/s
- mw2 = ρ*V2=60* $10^{-3}$Kg/s cpw =4.18 kJ/kgk (from HT data book)
- mw3= ρ*V3=40* $10^{-3}$Kg/s cpg =2.1 kJ/kgk (from Diesel engine lab manual)
- mg1= mw*Cpw*ΔTw/Cpg*ΔTg
- mg1 =0.0426Kg/s
- mg2=0.0137Kg/s
- mg3=0.01209Kg/s

Mass flow rate of gas (mg) = 0.0137 kg/s (Higher value of above three is considered)

Density of flue gas from heat transfer data book at 85**°C =** 1.008 kg/$m^3$

Volumetric flow rate gas = mass flow rate /density =0.0137/1.008 = 0.01367 $m^3$/s

Gas flow through one tube = volumetric flow rate/no of tube =9.116* $10^{-4}$ $m^3$ /s

Velocity of gas in one tube = (gas flow through one tube/area of tube)

- Area of tube = π/4*$d^2$
- d=19.05mm
- Velocity of gas in one tube = 3.198 m/s

**CALCULATION OF AIR OUT LET TEMPERATURE:**

Heat balance heat load on gas side and air side

Q g = Q a

- mg*Cpg*(Tg1 -Tg2) = ma*Cpa*(Ta1- Ta2)
- Cpa = 1.005 KJ/kgk
- Cpg = 2.1 KJ/kgk

**Outlet temperature of air ($T_{a2}$)** =$52.51^0$C

CALCULATION OF **LMTD** (LOG MEAN TEMPERATURE DIFFERENCE):

For Counter Current flow type Shell and Tube heat Exchanger,

LMTD = (ΔT1- ΔT2)/ln(ΔT1/ ΔT2)

- ΔT1 =Tg1 –Ta2=$32.5^0$C
- ΔT2 = Tg2 –Ta1=$22^0$C

Hence,

LMTD = (32.9 - 22)/ln(32.9/22)

**LMTD = 26.90**$^0$C

CALCULATION OF GAS SIDE HEAT TRANSFER COEFFICIENT

Using **Dittus Boelter** equation:

Flow in tube Generalized equation is Nu = $0.023Re^{.8} Pr^{0.3}$Properties of air at $T_f$ = $(Tg_1+Tg_2)/2$ Ns/$m^2$ = (85+55) / 2 = 70°C

- Density ρ = 1.053 kg/$m^3$
- Absolute viscosity μ = $19.002*10^{-6}$ Ns/$m^2$
- Prandtl number P r= 0.6989
- Thermal conductivity k = 0.02872 W/mK
- Velocity of gas in tube v = 3.198m/s
- D = shell diameter = 0.220m

  Re = ρvD/ **μ**= 3376.004

  Nu = $0.023Re^{.8} Pr^{0.3}$
- Nu = $0.023(3376.004)^{0.8} (0.6989)^{0.3}$
- Nu =13.73

  hg=Nu*k/d

**Gas side heat transfer coefficient (hg**) = (13.73*0.02872/0.01905) = 20.69 W/$m^2$K

**CALCULATION OF OVERALL HEAT TRANSFER COEFFICIENT:**

We know that **1 / U = (1 / $h_a$)+ (do / $k_{metal}$*ln($d_0$/$d_i$) + (1 / $h_g$ )**

Where, U = The overall heat transfer coefficient W/$m^2$K

$k_{metal}$= Thermal conductivity of the material W/mK

h = Heat transfer coefficient W/$m^2$K

The factor (do / $k_{metal}$*ln($d_0$/$d_i$ ) is negligible, and substituting all the other values, We get,

Overall heat transfer coefficient **U = (1 / 8.22) + 0+ (1 / 20.69) = 5.716 W/$m^2$K.**

Calculation of Heat Load, Q:

Heat Load, Q = ma*cpa*(Ta1- Ta2) = 0.044*1.005*(52.51-33) = 0 .862 KW

**CALCULATION OF AREA:-**

From the fundamental equations for heat transfer we get, **Q** = **U A $\Delta T_{lmtd}$**

- Heat transfer rate Q = 0.862kw
- overall heat transfer coefficient U = 5.716 W/$m^2$K
- Corrected LMTD = 22.86
- Hence, 0.862*1000 = 5.716*A*22.86
- A= 6.59 $m^2$

  From Area =n πdL ➔ 6.59 =15* π * 0.01905 * L ➔ **L = 110.13 m**

Therefore, **Length of each tube** required is **7.342m**

*A. CONTRUCTION DETAILS OF AIR PREHEATER:*

***B.*** *The following are the constructional details of the Air-preheater model:*

*Table constructional details of the Air-preheater model:*

Figure : drawing of an air-preheater

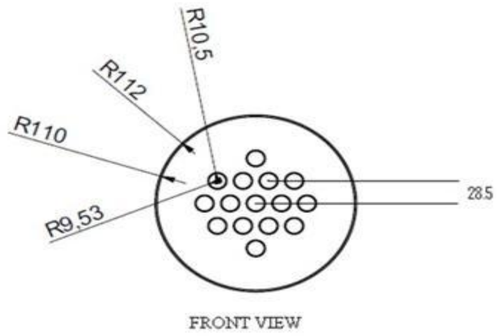



Figure : dimensional view of the shell

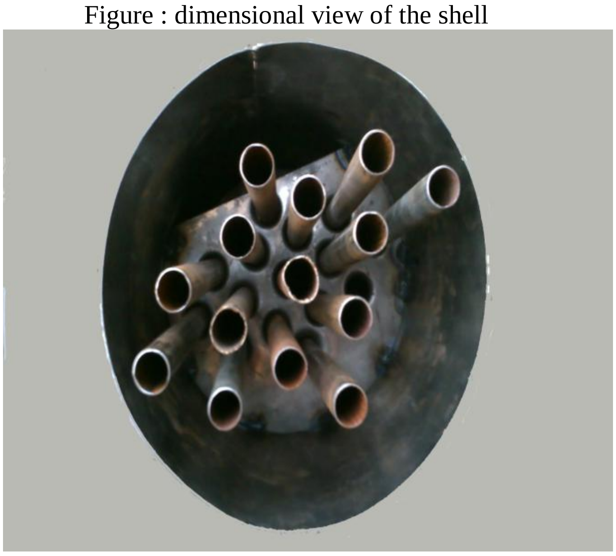
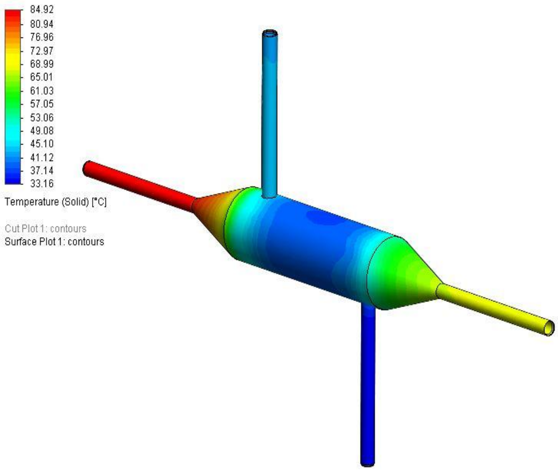
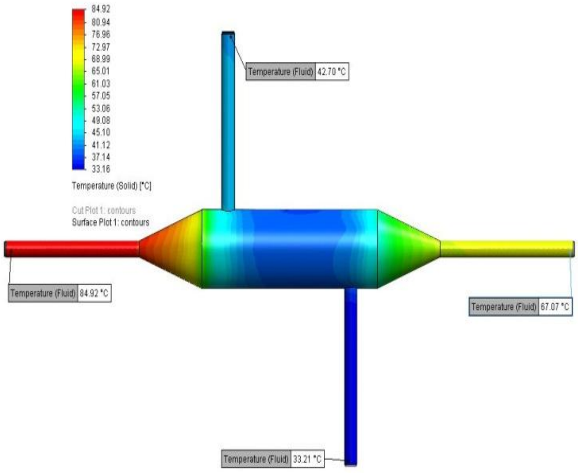


Figure :Shell with Tubes and baffle during manufacturing Internal View:

Figure:Shell with Tubes during manufacturing External View

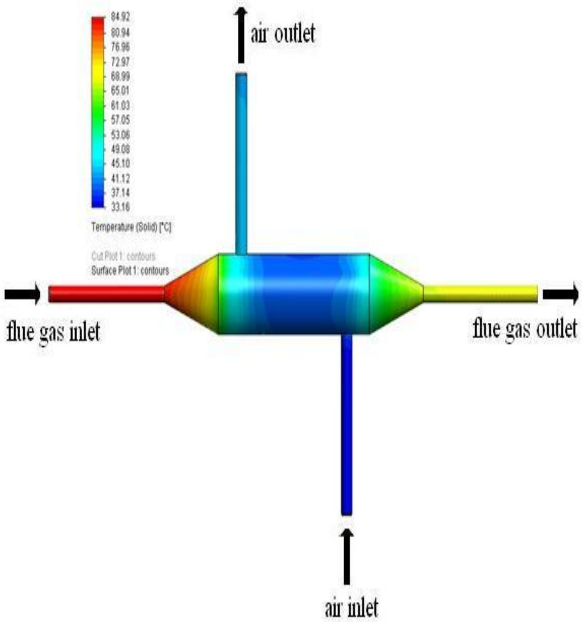
Figure : Assembly to an Engine Exhaust:



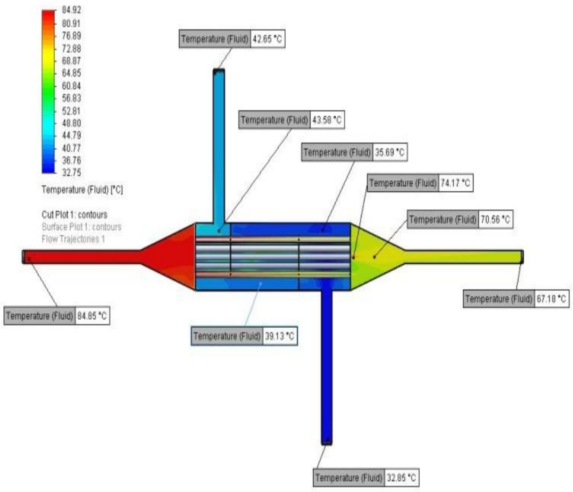


## IX. RESULTS IN CFD ANALYSIS

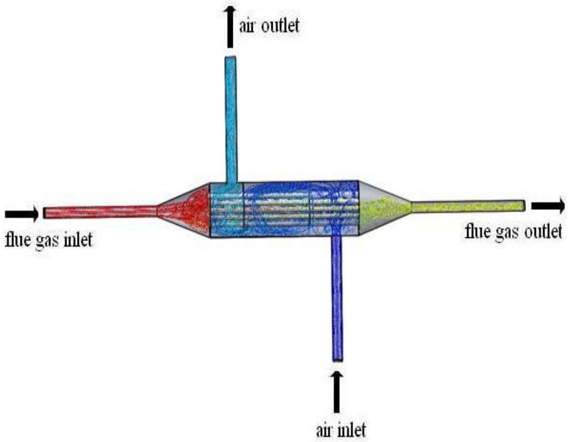
**Solid Temperature**




**Fluid Temperature**



**Animation Plot**



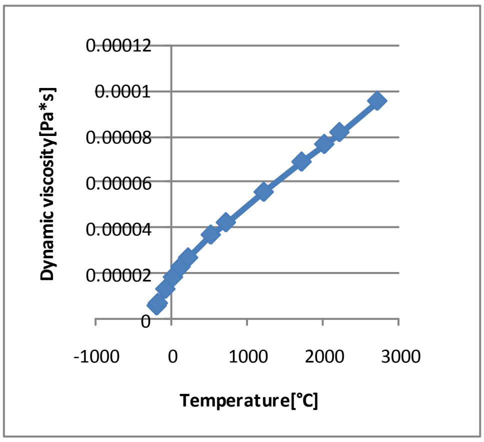

**Engineering Database**

**Gases**

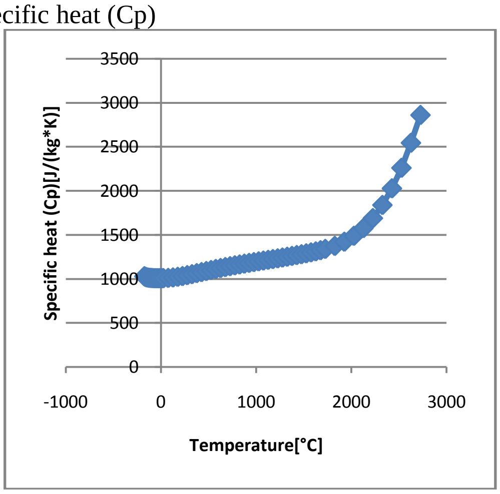
***Air***

Path: Gases Pre-Defined
Specific heat ratio (Cp/Cv): 1.399
Molecular mass: 0.0290 kg/mol

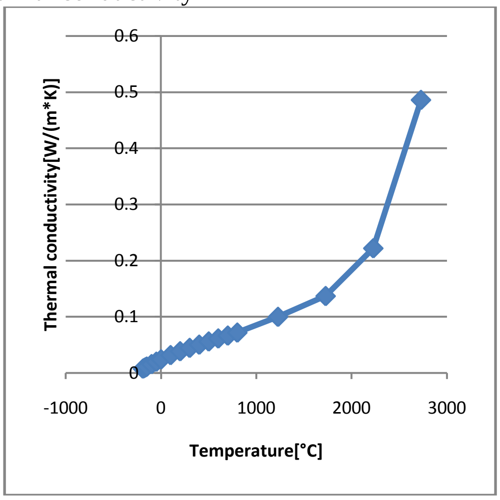
Dynamic viscosity




Specific heat (Cp)




Thermal conductivity




**EXPERIMENTAL READING AND CALCULATIONS**

The following is the experimental procedure:

- Connect the APH to the Diesel engine silencer outlet with a hosepipe, union/coupling
- Fix the blower to air inlet pipe firmly. Use tape if necessary.
- Insert thermocouple in the port provided at gas inlet pipe
- Start the diesel engine
- Switch on blower and Thermocouple
- Thermometer is also used to measure temperature
- Wait for some time until the air outlet temperature becomes steady

Based on the experimental readings, thermal design of air preheater is verified by using NTU method

AIM**:** Cross verifying the experimental air outlet temperature with that calculated by NTU method

**CALCULATION OF AIR OUTLET TEMPERATURE USING NTU METHOD:**

**Heat transfer rate, Q = mg*Cpg*ΔTg kW**

Where,**Mass flow rate** gas = 0.0137 kg/s

**Cpg** = 2.1 kJ/kgk

**Temperature inlet of gas** = 65 °C

Therefore, **Q** = 0.0137*2.1*(65-Tgo)

**Heat capacity of gas (C)** = **mg.cpg** = 0.0137*2.1= 28.77 W/K ---------------→(1)

**Heat capacity of air = ma.cpa** = 0.044*1.005= 42.22 W/K ---------------→(2)

Comparing the two values in Equation (1) & ( 2)

$\mathbf{C_{min}}$ = 28.77 w/k and $\mathbf{C_{max}}$ = 42.22 W/K

**Effectiveness, € = actual heat transfer/ideal heat transfer**

i.e.€ = Qactual/Cmin(Tg1- Ta1)

- Tg1=65, Ta1=33

- € = 0.0137*2.1*1000*(65-Tgo)/28.77*(65-33)

| S.no | Air inlet temp °c | Air out temp °c | Gas inlet temp °c | Gas outlet temp °c |
|---|---|---|---|---|
| 1. | 33.4 | 47 | 65 | 46 |
| 2. | 33.4 | 44 | 60 | 37 |

**NTU= [-1/(1+C$^2$ )][ln{(2/€)-1-C-(1+ C$^2$)}/ {(2/€)-1-C+(1+ C$^2$)}]** (From HT data book )

C = Cmin/Cmax = 0.6817

Also **NTU= UA/ Cmin** = .716*0.5924/28.77 = 0.117

Substituting all values in the above NTU equation:
0.117= [-1/(1+0.6817$^2$ )][ln{(2/0.03125(65- Tgo))-1-0.6817-(1+ 0.6817$^2$)}/ {(2/0.03125(65- Tgo))-1-C+(1+ C$^2$)}] From the above equation **gas outlet temperature** is calculated as **Tgo = 49.35°C**

From heat balance , Qg= Qa

- ➔ mg*Cp*ΔTg = ma*Cpa*ΔTa
- ➔ 0.0137*2.1*(65-49.35)= 0.044*1.005*(Tga-3)
- ➔ 0.4502=0.044*1.005*(Tga-33)

Therefore the calculated air outlet temperature of air out let Tga = 43.66°cHence the outlet temperatures are matching and design air preheater is verified.

## X. DISCUSSION OF RESULTS AND CONCLUSIONS RESULTS

1. Thermal Design of Air pre heater is done by fixing air and gas out let temperature and a heat transfer area of 6.59 m$^2$ is obtained.
2. A pro type of air pre heater (Shell and tube type) is fabricated for a heat transfer area of 0.5924 m$^2$ as for higher air outlet temperature the air preheater size increases which is not economic for a lab model.
3. Exit temperatures of Air preheater model
   - Air outlet temperature = 47°C
   - Gas outlet temperature = 46°C
4. Exit Temperature of air preheater CFD model is
   a) Solid temperature Result
      Air outlet temp = 42.70$^0$C
      Flue gas outlet Temp = 67.07$^o$C
   b) Fluid Temperature Result
      Air outlet temp = 42.65$^0$C
      Flue gas outlet Temp = 67.18$^o$C

## XI. CONCLUSIONS

1. Thermal design of Air preheater model is cross verified by NTU method
2. Air and Gas outlet temperature from experiment are 47°C and 46°C which are close to that obtained by NTU method 43.66°c and 49.35°C
3. The difference in the temperature may be due to fouling in gas and distribution of flow due to vibration of the system

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