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Design and Analysis of Flue Gas Waste Heat Recovery System for Refrigeration In A Thermal Power Plant

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ABSTRACT

Huge amount of energy wasted through the flue gas in thermal power station causes a great concern in recent years. Discharging hot flue gas in the environment is not only a wastage of energy but also increases the rate of global warming. Efforts are given world-wide to harness the energy for useful purposes. The main objective of this research work focuses on tapping the unused heat energy from the flue gas in a 270 MW thermal power plant and using the same for refrigeration purpose for the large cooling load requirements in a thermal power plant. The waste heat of flue gas is utilized in vapour absorption refrigeration plant by using gas to liquid multi-pass cross flow heat exchanger that is placed in the existing space between boiler and chimney. Vapour Absorption Machine doesn't require very high temperature and also requires a little operational power as compared to Vapour Compression Refrigeration Machine. Hence, this work intends to replace existing Freon-12 refrigerant based Vapour Compression Refrigeration System during the thermal power plant operational hours and thus preventing ozone depletion. This research work also aims in minimizing global warming and also high suspended particulate matters based air pollution by reducing temperature of exhaust gas emitted to the atmosphere. The project work selects the most economic design of a heat exchanger as the final design and analyses the design by a ANSYS software on detailed design parameters of gas to liquid multi-pass cross flow heat exchanger.
Keywords: Thermal power plant, Gas to liquid heat exchanger, Vapour absorption refrigeration, Global warming, Ozone depletion, ANSYS software

I. INTRODUCTION

Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use. The sources of waste heat include streams of hot exhaust gases, exhaust steam and hot liquids, as well as through heat conduction, convection, and radiation from hot equipment surfaces. The various studies have estimated that as much as 20 to 50% of industrial energy consumption is ultimately discharged as waste heat. While some waste heat losses from industrial processes are inevitable, these losses can be reduced by improving equipment efficiency or installing waste heat recovery technologies. Waste heat recovery entails capturing and reusing the waste heat in industrial processes for various applications which include generating electricity, preheating combustion air, space heating, refrigeration and air conditioning, etc. Captured and reused waste heat is a valuable approach to improve overall energy efficiency by optimizing for costly purchased fuels or electricity and for the protection of global environment from pollution and rate of global warming.

The essential components (Figure 1) required for waste heat recovery are: 1) an accessible source of waste heat, 2) a recovery technology, and 3) a use for the recovered energy.

The main objective of this work is to utilize the waste heat available in exhaust gases coming out of the Boiler (acting as the source of waste heat) of a 270MW thermal power plant to run a Vapour Absorption Refrigeration System (VAR) during plant operational hours which can replace existing Freon-12 refrigerant based Vapour Compression Refrigeration (VCR) System present in the functional block of the plant for large cooling load requirements. The waste heat of flue gas is utilized in vapour absorption refrigeration plant by using the most economic gas to liquid multi-pass cross flow heat exchanger acting as a waste heat recovery tool that is placed in the existing space between boiler and chimney. This design is then analysed by a ANSYS software on detailed design parameters of gas to liquid multi-pass cross flow heat exchanger. This work also aims to reduce the rate of global warming and air pollution by reducing the exhaust gas temperature. This work encompasses the design and analysis of the heat exchanger only. The specification of the VAM will be adopted according to the cooling load requirement of the power station under consideration.

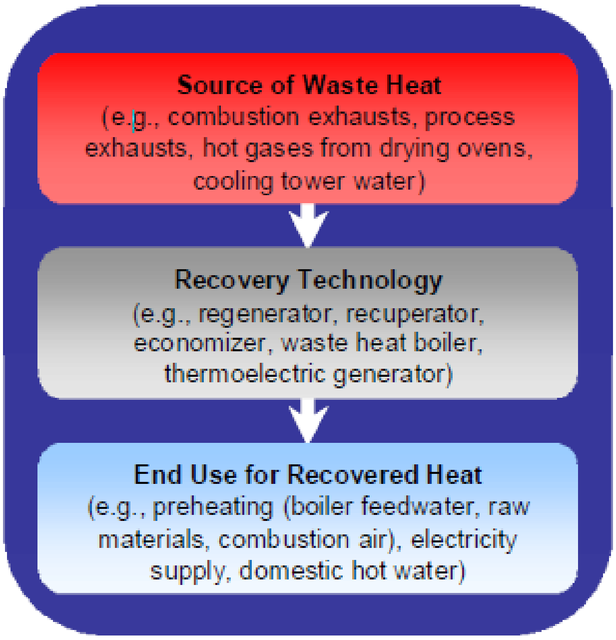


Figure 1: Three essential components required for waste heat recovery

II. METHODS AND MATERIAL

1. Description of Setup And Working Process

The system consists of a water reservoir with makeup feed arrangement, pump, heat exchanger, vapour absorption refrigeration system and cooling tower, all of which are connected by pipes of suitable dimensions. Fig. 2 shows the detailed sketch of the proposed system. The heat exchanger will be designed to provide the necessary input to the VAM so as to deliver the desired air conditioning effect. The specifications of the remaining part of the system i.e. the reservoir, the pump and the connecting pipes is to be chosen according to the flow requirement of the system.

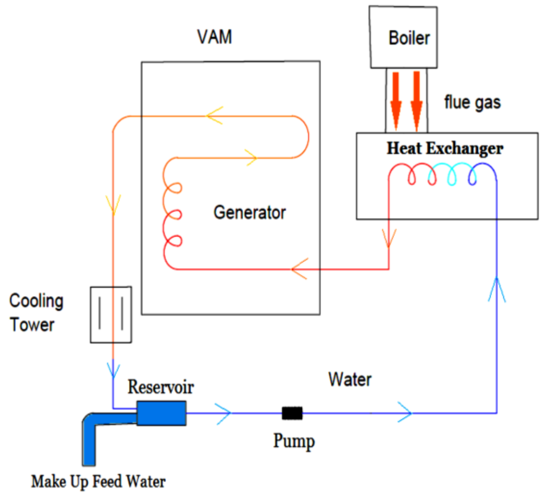


Figure 2. Detailed sketch of analytical setup and working process

The system is proposed for waste heat recovery in a cascade process so that the heat recovery process will be efficient. The working process is intended for maximum heat recovery from the flue gas which is at a temperature of 135℃. At this temperature thermal and structural effects on the heat exchanger to be designed can be neglected. The working fluid selected is water because of its relative abundance and its non-toxic nature along with its high heat capacity. The temperature of the flue gas reduces to almost 135℃ after rejecting heat from the boiler which is higher than the design value of 130℃ during power plant operation because of various losses in boiler. This flue gas carrying waste heat at the temperature higher than the design temperature is made to flow across a multi-pass cross flow heat exchanger. The flue gas after rejecting heat to the heat exchanger passes through the chimney. A pump of predetermined mass flow rate is used to transfer water from a reservoir to the heat exchanger which heats up to the desired temperature through a cascade process involving multi-pass tubes of the heat exchanger. The heated water is then pushed forward by the pressure existing in the system to the generator part of the VAM. The water coming out of VAM which is still at a considerable temperature is passed through the cooling tower which is already at the plant. The cooled water is then returned to the reservoir from where it is again pumped by the water pump. A make-up feed water arrangement is also provided with the reservoir to compensate for any losses.

2. Design Calculations for Multi-Pass Gas To Water Cross Flow Heat Exchanger

The design of flue gas to water cross flow heat exchanger and the resulting refrigerating capacity can be obtained from the thermal power plant datasheet. The total flue gas flow for 270 MW thermal power plant is 1000 tonnes/hr. There are two flue gas passes in the plant for this flow. One pass flue gas flow = 500 tonnes/hr = 138.89 kg/s.

a) Analysis of flue gas for waste heat recovery

The temperature of the exit flue gas available for waste heat recovery purpose is low and is about 135°C.So, the available energy (AE) in the exit flue gas as given by the thermodynamic equation,

$$\mathbf{AE = \dot{m}_g\, c_{pg}\left((T - T_o) - T_o \ln\left(\frac{T}{T_o}\right)\right)}$$

where, $\dot{m}_g$= exit flue gas flow rate,

c_{pg} = specific heat capacity for flue gas,

T = exit flue gas temperature,

T_o= ambient temperature.

From 270 MW thermal power plant datasheet and standard thermophysical properties tables,

$\dot{m}_g = 277.6$ kg/s ,

c_{pg} = 1.008 kJ/kg ^{o}K [3],

T = 135^{o}C = 408 K and T_o = 30^{o}C = 303 K.

Thus, **Available Energy that can be extracted AE = 4154.52 kW**

Vapour Absorption Machine (VAM) does not require very high temperature and works on the temperature in the range 80-120℃ and its COP varies from 0.6 to 0.8.

b) Heat transferred from one pass flue gas to water in heat exchanger

$\mathbf{Q = \dot{m}\, C_{pf}\, (T_{fi} - T_{fo})}$

where, Q is the heat transferred from flue gas to water,

$\dot{m}$ is the mass flow rate of flue gas = 138.89kg/s ,

C_{pf} is the specific heat capacity of flue gas = 1.008 kJ/kgK ,

T_{fi} is the inlet temperature of flue gas at heat exchanger inlet =135^{0}C and

T_{fo} is the desirable outlet temperature of flue gas at heat exchanger outlet = 130^{0}C.

Thus, **Q = 700 kW**

c) Mass flow rate of water required in heat exchanger

Now, $\mathbf{Q = m_w\, C_{pw}\, (T_o - T_i)}$

m_w is mass flow rate of water,

C_{pw} is specific heat capacity of water = 4.187 kJ/kgK,

To is desirable outlet temperature of water from heat exchanger = 95^{0}C (assumption),

T_i is inlet temperature of water at heat exchanger inlet = 25^{0}C &

Q = 700 kW.

Thus, $\mathbf{m_w = 2.38}$ **kg/s**

d) Calculation of Overall Heat transfer Coefficient using Zukauskas relation

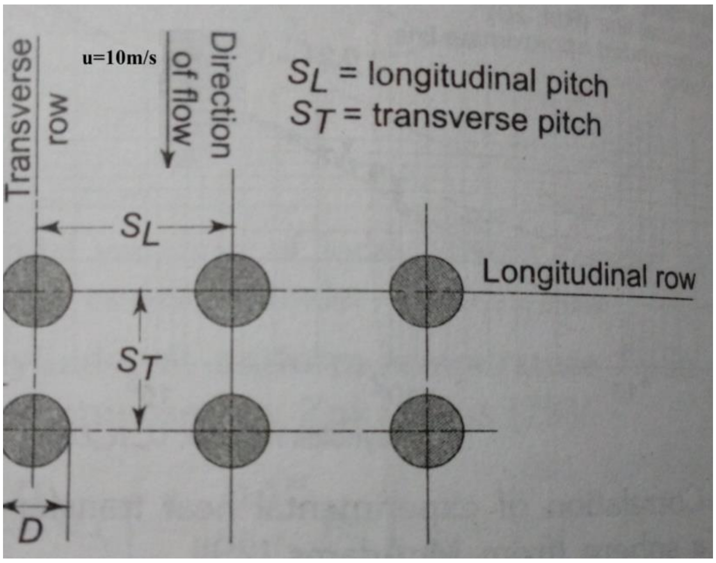


Figure 3: In-line tube arrangement in cross flow heat exchanger

The Cross Flow Heat Exchanger in this work has following specifications:

Inlet flue gas velocity u_∞= 10m/s,

Outer diameter of a water tube of Aluminium d = 0.025m = 25mm,

Thickness of water tube t = 1mm,

Inner diameter of a water tube d_i = 24mm,

Dimensions S_T & S_L as in above fig. = 0.1m = 10cm.

Now, $u_{max} = u_\infty \frac{S_T}{S_T - d}$ [6],

where, u_{max} = maximum flow velocity based on minimum flow area for flow, Substituting all values we get, u_{max} =13.3 m/s.

Reynolds number for this arrangement is given by-

$$Re_d = \frac{u_{max}\, x\, d}{\vartheta}$$

where, ν = kinematic viscosity

The various physical properties are calculated at mean film temperature of

$$\frac{135+\frac{25+95}{2}}{2} = 97.5^0 C\text{=370 K.}$$

At 370 K for flue gas from standard thermo-physical properties tables, We have,
Kinematic viscosity $\nu = 19.28 \times 10^{-6}$ m^2/s,
Thermal conductivity $k = 28 \times 10^{-3}$ W/mK,
Prandtl number Pr = 0.7.
Thus, $Re_d = \frac{u_{max}\, x\, d}{\vartheta}$ = 17245.85

Now, for transition regime $10^3 < Re_d < 2 \times 10^5$, using **Zukauskas relation** for in-line tubes [6],

$$\textbf{Nusselt Number } \mathbf{Nu_d = 0.27\, Re_d^{0.63}\, Pr^{0.36}}$$

$$\mathrm{Nu_d} = 0.27 \times 17245.85^{0.63} \times 0.7^{0.36}$$

$$\mathbf{Nu_d = 110.84}$$

Thus, heat transfer coefficient $\boldsymbol{h} = \frac{\boldsymbol{Nu_d\, k}}{\boldsymbol{d}}$,
h = 125 W/m²K.

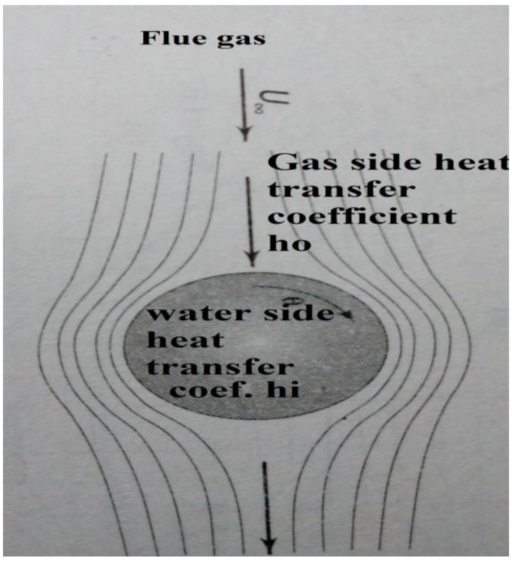


Figure 4: Flow over a circular cylinder

Now from above fig., Overall heat transfer coefficient $U = Uo = \frac{1}{h_o} + \frac{x}{k} + \frac{1}{h_i}$

Since in general, gas side heat transfer coefficient h_o<< water side coefficient h_i,

Thus, $1/h_o >> 1/h_i$,
Hence, $U \approx h_o$= 125 W/m^2K.

Overall heat transfer coefficient U = 125 W/m²K.

e) Logarithmic Mean Temperature Difference (LMTD)

The next important step is finding out the LMTD of the heat exchanger.

$$\text{LMTD}_{\text{counter flow}} = \Delta T_m = \frac{\Delta T_i - \Delta T_e}{\ln(\frac{\Delta T_i}{\Delta T_e})}$$

where, $\Delta T_i = T_{h1} - T_{c2} = 40^oC$ and $\Delta T_e = T_{h2} - T_{c1} = 105^0C$ where T_{h1} and T_{h2} are the inlet and outlet temperatures of the flue gas and T_{c1} and T_{c2} are the corresponding values of water.

Thus, $\text{LMTD}_{\text{counter flow}} = \frac{40-105}{\ln(\frac{40}{105})} = 67.35^oC$

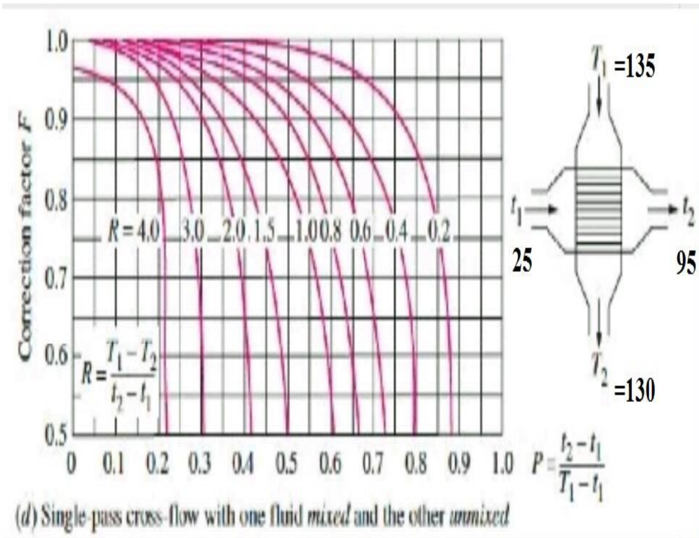


Figure 5: Correction factor calculation for cross flow heat exchanger

LMTD for cross flow = $\text{LMTD}_{\text{counter flow}}$ x correction factor = 67.35 x 0.97
LMTD $_{\textbf{cross flow}}$**= 65.32°C.**

f) Surface Area of the Heat Exchanger

Area of the heat exchanger is found out using the equation,
Heat transfer, Q = UA LMTD $_{\textbf{cross flow}}$
where, A is the surface area of the heat exchanger in m^2.
$700 \times 10^3 = 125 \times A \times 65.32$
Thus, **Surface area A = 85 m².**

g) Number of Tubes in heat exchanger

Diameter of a water tube d = 0.025 m = 25mm.

For a heat exchanger consisting of n tubes taking each tube length L =9 m, the total surface area is given by,

A = πdLn

Thus, $n = \frac{85}{\pi \times 0.025 \times 9} = 120$

Thus, **Number of tubes n = 120**

h) Velocity of water through each column of water tube

It is clear that mass flow rate,

$m_w = \rho A_c V$

where, ρ is the density of water in kg/m^3= 1000 kg/m^3,

Total mass flow water rate through 20 columns = 2.38 kg/s,

m = mass flow rate through each column =

2.38 /20 = 0.119 kg/s,

A_c is the cross sectional area of each column water tube in m^2.

$A_c = \frac{\pi \times {d_i}^2}{4} = \frac{\pi \times 0.024^2}{4} = 4.52 \times 10^{-4}\ m^2$ and

V is the velocity of water through each column in m/s.

Hence, $V = \frac{m_w}{\rho A_c} = \frac{0.119}{1000 \times 4.52 \times 10-4} = 0.263$ m/s

Thus, **Velocity of water through each column V = 0.263 m/s**

i) Calculation of water pumping power through the heat exchanger

Reynold number for water flow through a single column $Re_d = \frac{\rho \, x \, V \, x \, d_i}{\mu}$

Where, ρ is density of water = 1000 kg/m^3,

V is the velocity of water through each column = 0.263 m/s,

d_i is inner diameter of tube = 0.024 m,

μ is dynamic viscosity of water at mean film temperature of 25+95/2 = 60°C = 489×10^{-6} Ns/m^2

Thus, Re_d = 12907.97

Since, $4000 < Re_d < 10^6$,

Thus, co-efficient of friction $f = \frac{0.079}{{R_{ed}}^{1/4}}$.

Thus, $f = 7.41 \times 10^{-3}$.

Head lost due to friction $h_f = \frac{4fLV^2}{2gd_i}$

where, L is length travelled by water through a single column = 6 x 9 = 54 m.

g is acceleration due to gravity = 9.81 m/s^2

Thus, h_f = 0.235 m.

Now, water pumping power required thorough a single column $\mathbf{P = \rho Q g h_f}$

where, Q = discharge through a single column

$Q = A_c \times V = 4.52 \times 10^{-4} \times 0.263 = 1.188 \times 10^{-4}\ m^3/s$.

Thus, P = 0.274 W.

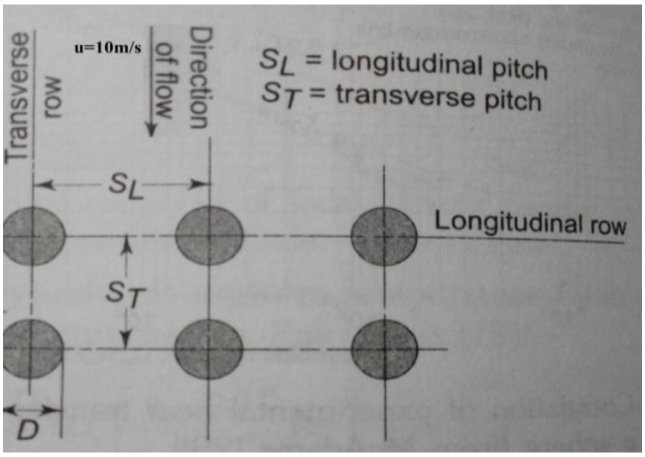
Since, there are 20 columns,

Total water pumping power required

P_{total} = 20x0.274 = 5.5 W.

3. Design Arrangement of Cross Flow Heat Exchanger In Flue Gas Path

There are 120 aluminium water carrying tubes each of 9m length in the designed cross flow heat exchanger. The above designed cross flow heat exchanger is to be directly placed in one of the flue gas duct path which is 3.750 m x 3.750 m size for 270 MW thermal power plant. For this, each water tube of 9 m length is made up from three segments each of 3m length and these are joined or welded by elbows. The centre to centre distance between successive segments is S_T = 0.1 m = 10 cm. Using this method, a column of 6 water tubes is formed. The length of the column having 6 tubes is approximately 2 m in transverse direction. In this way, all the 120 water tubes are arranged in 20 columns each having 6 water tubes placed in a single row one after the other. The centre to centre distance between successive columns is S_L= 0.1 m and thus occupying approximately 2 m in longitudinal direction. Thus, the designed cross flow heat exchanger gets comfortably fit into the flue gas duct of 3.750 m x 3.750 m. The cold water at 25°C enters into the bottom of each column from a common header placed at bottom. The hot water at about 95°C leaves from the top of each column into another common header placed at the top. Since the total water flow rate is 2.38 kg/s, the water flow rate in each of 20 columns is 0.119 kg/s. This whole design arrangement is well shown in following figures.



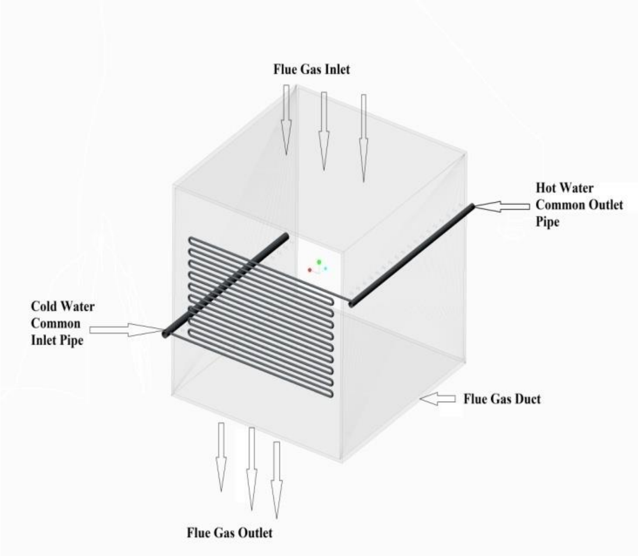


Figure 6: Sectional 3D view of a column

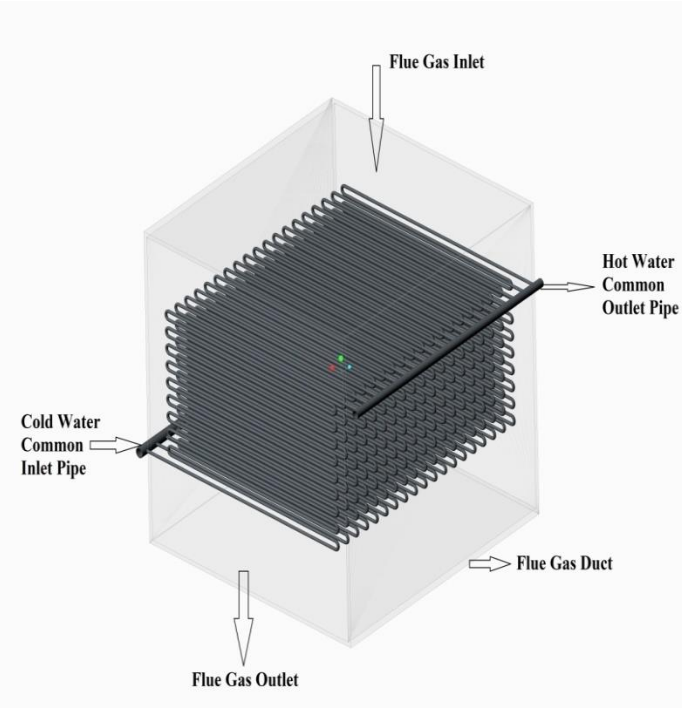


Figure 7: Designed 3D view of cross flow heat exchanger

4. Experimental Analysis

The heat exchanger analysis is being carried out using ANSYS software. Thermal and structural stress effects need not to be considered at pressures below 15 atm or temperatures below 150°C.Thus the thermal and structural analysis of the heat exchanger is not carried out in this work. The common materials used for fabrication of industrial heat exchangers are aluminum, copper, stainless steel and high carbon steel. Since this work involves low temperature heat recovery, stainless steel and high carbon steel are not considered as they are commonly used in high temperature application and are very costly. Table shows the comparison of properties of Aluminum and Copper which are the possible options for material for the heat exchanger. Since this heat recovery is in the very low temperature and also aluminum is less denser it is the most suitable choice.

Table: Comparison of Physical Properties of Materials

Material	**Density**	**Melting Point**	**Boiling Point**
Aluminium	2.70 g·cm−3	660.32 °C	2470 °C
Copper	8.96 g·cm−3	1084.62 °C	2562 °C

4.1 Velocity Analysis

Since the velocity of flue gas across the heat exchanger and the velocity of water flowing through each pipe are the same, only a single pipe including the control volume is being analysed. The solution obtained from the software is obtained as a streamline simulation.

a) Flue gas velocity analysis

Fig.8 shows the velocity streamline of flue gas over an aluminium water tube. From the figure, it can be understood that the velocity of flue gas increases when it approaches the tube and decreases when it passes the tube. This is in accordance to the design calculations involved in section 3.d of this paper, where inlet flue gas velocity is 10 m/s and increases upto 13 m/s as the flow progresses. It may be due to the reduction in area that occurs when the flue gas approaches the tube and the subsequent reduction in pressure. As the flow area increases the velocity of the flue gas decreases correspondingly.

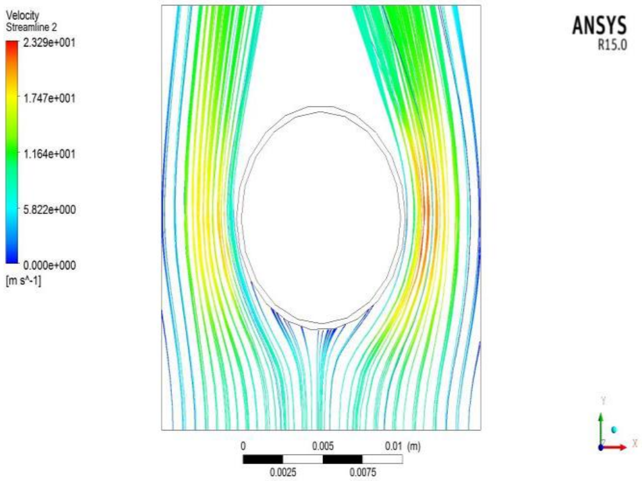


Figure 8: Velocity Streamline of Flue Gas

From the fig.8, it can also be inferred that the velocity of the flue gas directly below the tube is almost zero. This may be due to the near stagnation condition that arises at that region. This is mainly because of the fact that the particular layer of flue gas has less chance to escape due to the obstruction created by the tube and the continuous collision with fresh flue gas which nearly impedes its motion. Thus there is less chance of fresh flue gas interaction with the tube wall at this region.

b) Water tube velocity analysis

Fig.9 shows the velocity variation of water flow in the radial direction inside an aluminium water tube. It can be inferred from the figure that the velocity of water layers close to the pipe boundary is low as compared to the velocity of water layer passing through the center of the pipe. This is due to the viscous effects that become prominent as the proximity to the pipe wall increases.

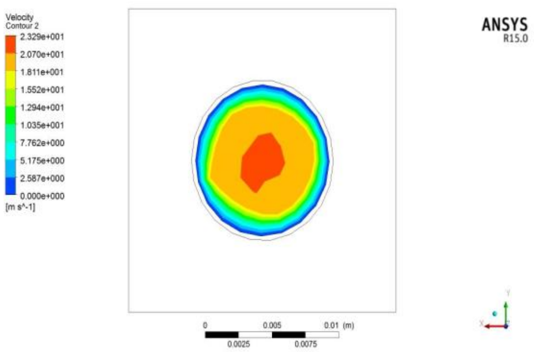


Figure 9: Water tube velocity analysis

5.2 Temperature Analysis

Temperature analysis is carried out on a single column consisting of 6 aluminium water tubes. Other columns need not to be considered as they exhibit same behavior. The following boundary conditions have been applied on a single column in the ANSYS software to obtain the temperature analysis.

a) Flue gas temperature on outer walls of water tubes at heat exchanger inlet - 135^{o}C

b) Flue gas mass flow rate - 138.89 kg/s

c) Flue gas inlet velocity – 10 m/s

d) Flue gas temperature on outer walls of water tubes at heat exchanger outlet - 130^{o}C

e) Flue gas dynamic viscosity – 185x10^{-7} Ns/m^2

f) Flue gas density – 0.92 kg/m^3

g) Water temperature on the inner wall of water tube at heat exchanger inlet - 25^{o}C

h) Water mass flow rate inside a single column – 0.119 kg/s

i) Velocity of water inside a column – 0.263 m/s

j) Overall heat transfer coefficient – 125 W/m^{2}K

Fig.10 shows the temperature contour of the inner walls of 6 aluminium water tubes of a single column.

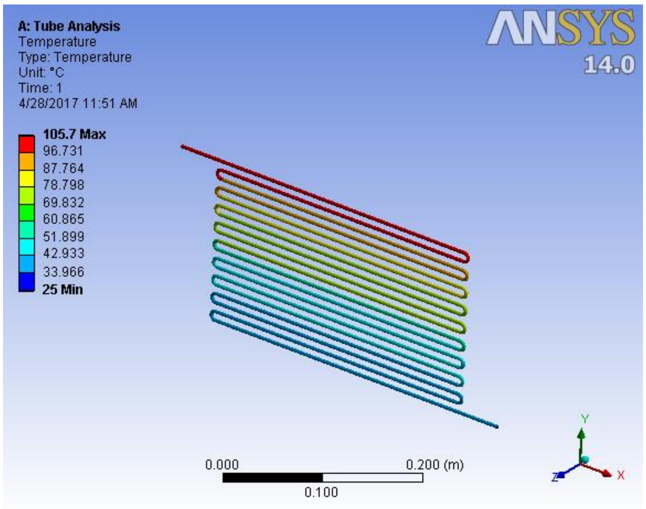


Figure 10: Temperature contour of inner walls of a column

From the figure, it can be seen that cold water at 25^{o}C as shown by blue color enters the column at bottom. The temperature of water goes on increasing inside the column as it progresses upwards through the multipass heat exchanger water tubes. This increase in temperature can be seen by different changing colors in the above figure. High temperature water in the range of 96-105°C as shown by red color is finally obtained at column outlet as a result of flue gas heat transfer to water. This outlet temperature obtained is in well accordance to the desirable outlet temperature of water assumed in section 3.c of design calculations.

Fig. 11 shows the temperature contour of outer walls of a single column as obtained from ANSYS software.

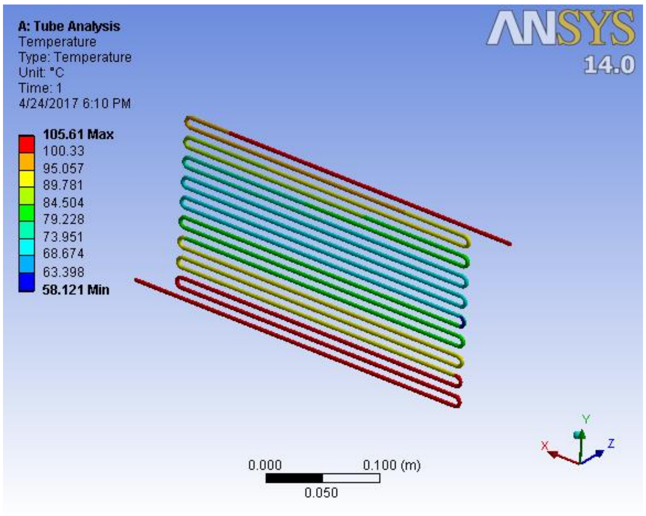


Figure 11: Temperature contour of outer walls of a column

III. RESULTS AND DISCUSSION

a) Desired hot water outlet temperature:

High temperature water in the range of 96-105°C is obtained at the outlet of each of the 20 columns from experimental analysis. This temperature is potential enough for the generator part to run a single effect Vapour Absorption Machine.

b) Quantity of heat extracted from flue gas at 135 °C:

Since the outlet temperature of water is in the range of 96-105 °C, then as much as 700 kW of heat has been transferred into the heat exchanger out of the available 4154.25 kW heat from the inlet flue gas at 135°C according to design calculations. Thus, approximately 20 percent of heat is extracted from inlet flue gas.

c) Reduced flue gas outlet temperature:

The flue gas outlet temperature is in the range of 127-130°C from design calculations and experimental analysis as a result of heat transfer of about 700kW into the heat exchanger. Thus, flue gas outlet temperature emitted to the atmosphere is reduced by 5 to 6 °C and so the desired exit temperature is achieved according to thermal power plant data sheet.

d) Achievable refrigeration load calculations:

Considering temperature of hot water in generator portion of VAM as 90°C, Maximum COP in vapour absorption system is given by-

$$\textbf{Max. COP} = \left(\frac{\mathbf{T_E}}{\mathbf{T_C - T_E}}\right)\left(\frac{\mathbf{T_G - T_C}}{\mathbf{T_G}}\right)$$

where, T_E = Evaporator temperature = 9°C=282K,
T_G = Generator temperature = 90°C=363K and
T_C = Condenser temperature = 30°C= 303K
Thus, Max. COP = 2.22
In general, Actual COP = 0.5 x Max. COP = 0.5 x 2.22 =1.11

Now, we know,

Refrigeration load = Actual COP x Heat supplied in generator

Refrigeration load = 1.11 x 700 kW = 777 kJ/s = 777/3.5 =222 TR

Refrigeration load that can be achieved from one flue gas path = 222 tonnes of refrigeration = 222 TR

e) Energy and cost savings

Power required for operating VAM = 2% of VCM
Generally for 270 MW and 200 TR,
Power consumed for operating VCM = 300 kW
Thus, energy consumed per hour = 300kWhr.
Annual energy consumption=300x24x365=2.628 x 10^6 kWhr = 2.628 million units (MU).
Thus, annual energy cost considering Rs.5 per unit = 2.628 x 10^6 x 5 = Rs.1,31,40,000
Now, Power consumed for operating VAM=2% 300 = 6 kW.

Annual energy consumption= 6x24x365=52560 kWhr
Now, annual energy cost considering Rs.5 per unit= 52560 x 5 = Rs.2,62,800.

Thus, **Annual cost savings = Rs. 1,28,77,200**

f) Reduction in suspended particulate matter at the chimney outlet

Since the outlet temperature of flue gas is reduced from 135^{o}C by 5 to 6^{o}C to 127-130^{o}C, density of the exit gas increases (as density of gas is inversely proportional to temperature) and hence its velocity reduces for the same mass flow rate of gas according to the equation $\dot{m} = \rho A_c V$, where $\dot{m}$ is flue gas mass flow rate, ρ is flue gas density and V is its velocity. Thus, electrostatic precipitators get more time for precipitation of dust particles in the flue gas and suspended particulate matters (SPM) at the chimney outlet gets reduced significantly. The value of SPM at 135 ^{o}C was observed as 50-60 mg/Nm^3.Thus, reduced value of SPM in the range of 30-40 mg/Nm^3 can be easily achieved.

IV. CONCLUSION

As the energy demand in our day to day life escalates significantly, the regeneration of energy into some beneficial work is a fantastic job. One such low grade energy is heat energy. So it is imperative that a significant and concrete effort should be taken for using heat energy through waste heat recovery. This work focuses on tapping the waste heat energy from flue gas and using the same for refrigeration requirements in a 270MW thermal power plant. The main conclusion that can be drawn from this work is that as much as 20 percent flue gas waste heat can be easily recovered of the total available waste heat energy from the exit flue gas at 135oC in a thermal power plant. This is achieved by using the most economic gas to liquid multi-pass cross flow heat exchanger acting as a waste heat recovery tool that is placed directly in the flue gas duct pass between boiler and chimney. The temperature of water exiting the heat exchanger is in the range of 96-105oC and is potential enough for the generator part to run a single effect Vapour Absorption Machine and achieve 222 tonnes of refrigeration from one flue gas path. Thus, huge annual energy savings of as much as 98 percent can be obtained since power required for operating VAM is very low (just 2%) as compared for VCM and therefore, effectively replaces the exiting Vapour Compression Machine during power plant operational hours. Thus, replaces Freon-12 refrigerant used in VCR system which causes ozone depletion. It can also be concluded that by reducing temperature of exhaust gas emitted to the atmosphere by 5 to 6 oC as a result of heat transfer in the designed heat exchanger, high suspended particulate matters (SPM) based air pollution at the chimney outlet can be reduced. The reduced value of SPM at the chimney outlet in the range of 30-40 mg/Nm3 can be easily achieved and thus the thermal power plant satisfies the pollution norm of SPM less than 50 mg/Nm3. The rate of global warming can also be minimized by reducing temperature of exhaust gas. The further reduction in exit flue gas temperature is not recommended because of cold end corrosion effect in the gas exit path.

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