DESIGN OF HEAT EXCHANGER FOR WASTE HEAT RECOVERY FROM PRODUCER GAS

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Abstract: - An innovative Pancake type heat exchanger is designed for a particular process industry. A new arrangement for flow of hot and cold fluids is employed for design—hot fluid flows in axial path while the cold fluid flows in a spiral path. To assess the performance of the prototype, its model is suitably designed and fabricated so as to perform experimental tests. Also, the design is conveniently programmed to conduct simulation tests on computer. Overall Heat Transfer Coefficient is taken as the measure of performance for comparing experimental results with theoretical values. Percentage accuracy is also estimated.

Keywords: - Waste heat recovery, pancake heat exchanger, overall heat transfer coefficient

1 Introduction

Heat exchanger is an indispensable component in an industrial system especially in process industry. Lot many commercial designs and types of heat exchangers are available in market for exchange of heat as well as for recovery of waste heat for the process plants. Since it has been a desperate need of the industry to spend less energy and extract maximum work possible out of it, it is this prejudice thought which has accelerated the use of waste heat recovery systems in the industries. Waste heat recovery deals with the extraction of heat from flue gases or other sources, which are otherwise termed as waste, and utilize it for worth production or process.

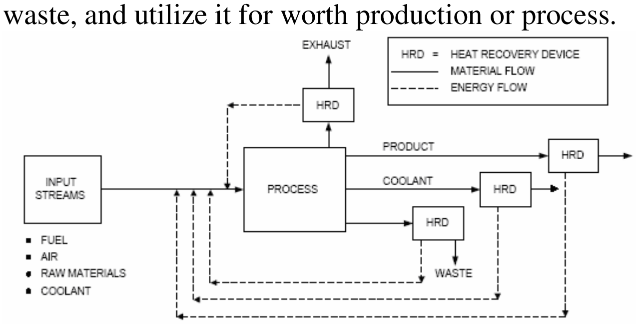

Fig.1 Schematic of possible single process heat recovery energy flow [1]

Waste heat recovery benefits can be broadly categorized into ‘Direct Benefits’ and ‘Indirect Benefits’. Direct Benefits include—increase in process efficiency arising from reduction in utility consumption & costs and process costs; and Indirect Benefits include—reduction in pollution, equipment size, auxiliary energy consumption etc.[2]

Today, various heat exchange systems are available for recovery of waste heat. Selection of the system for a particular use depends on various factors such as heat potential available for recovery, possible use of recovered heat, space available for installing the unit, and of course, cost!

The relative increase in heat transfer in spiral tube over a straight tube was noted 40 % higher at Re 2000. This shows that spiral tubes offer better performance as heat exchangers than straight tubes [3]. Also, there are numerous benefits of using spiral tube heat exchanger for process industries, which are listed by Minton [4]. These include—specially suited for flow of small heat loads, particularly effective for laminar flow because laminar flow heat transfer is much higher with spiral tubes, flows can be counter current, no problems associated with differential thermal expansion, compact and easy installation. The smaller length of entire unit is an added advantage of this system as compared to tube-in-tube or coil type heat exchangers. There are few limitations, which are easily overridden by the benefits.

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NOMENCLATURE

A	Area for heat transfer, m^2
A_{fl}	Area available for flow, m^2
a	Constant in equation of spiral
C_1	Constant in Grimmison's correlation
C_i	Constant in Richard et.al. correlation
Cp	Specific heat, kJ / kg k
D	Diameter of shell, m
De	Equivalent diameter of annulus, m
d	Diameter of tube, m
F	The forced convective multiplier
f	Friction factor
h	Heat transfer coefficient, W/m^2 k
h_{FC}	The forced convective coefficient
h_{NB}	The nucleate boiling coefficient (supressed)
h_{FZ}	The Forster- Zuber coefficient
L	Length of tube for Pancake, m
m	Mass Flow rate, kg/s
$\dot{m}$	mass flux
m	Constant in Grimmison's correlation
N	Number of spirals (Pancakes)
Nu	Nusselt Number
Nu_c	Nusselt number for straight tube
P	Pressure, bar
Pr	Prandtl number
Pr_w	Prandtl number at wall temperature
P_T	Turbulent spiral factor
P_L	Laminar Spiral Factor
q	Total Heat Duty, kW
R	Radius of spiral, m
Re	Reynold Number
S	The suppression factor
SPN_L	Laminar spiral number
S_L	Longitudinal pitch
S_T	Transverse pitch
T	Temperature, °C
ΔT_{lm}	Log mean temperature difference, °C
U_o	Overall Heat Transfer coefficient, W/m^2 °K
V	Velocity, m/s
X_{tt}	Lockhart-Martinelli parameter

Greek Symbols

κ	Thermal conductivity, W/m°K
ρ	Density, kg/m^3
μ	Viscosity, kg/ms
θ	Angle in radians in case of spiral
ν	Volume flow rate
$\Phi_v^{0.14}$	Viscosity ratio
σ	Surface tension, N/m

Subscript

g	Gas
w	Water
O	Outer
i	Inner
h	Header tube
c	Core pipe
m	Mean
sat	Saturation
s	Steam, Spiral
sp	Spiral
atm	Atmosphere
vg	Average
L	Liquid phase

It is found that very little information on spiral tube heat exchangers is available in the existing literature, that too for 'both fluids flowing in spiral paths'. Pancake is a spirally wound tube following the geometry of *Archimedean spiral* [5]. The present design is different than the existing one in the sense that the direction of hot fluid is in axial path while the cold fluid flows in a spiral path.

2 Problem Formulation

The problem is to design a suitable heat exchange system for waste heat recovery from a producer gas and use this heat to generate steam for a particular process industry. The data regarding producer gas composition, temperature, pressure and volume flow rate of gas, is collected from the actual site. Using this data, heat potential available in the producer gas is calculated using energy balance equation.

$$q_g = m_g \, Cp_g \, \Delta T \quad (1)$$

Quantity of steam generated is estimated from the following equation:

$$q_s = \text{Sensible heat} + \text{Latent heat} = [Cp_w \times (T_{sat} - T_{wi})] + hfg \quad (2)$$

$$m_s = m_w = \frac{q_g}{q_s} \quad (3)$$

This unit is designed for two stages i.e. sensible heating and latent heating. The set of equations and procedure adopted for sensible heating stage are as under:

$$q_{sensible} = m_w \times Cp_w \times \Delta T_w \quad (4)$$

The temperature of producer gas at the end of sensible heating of water is estimated using energy balance equation.

$$m_w\ Cp_w\ \Delta T_w = m_g\ Cp_g\ \Delta T_g \quad (5)$$

Flow arrangement selected is counter flow type and accordingly LMTD for this stage is calculated.

$$\Delta T_{lm} = \frac{(T_{gi} - T_{wo}) - (T_{go} - T_{wi})}{\ln\left[\frac{(T_{gi} - T_{wo})}{(T_{go} - T_{wi})}\right]} \quad (6)$$

Making assumptions for geometrical dimensions viz. L, d_c, d_o, d_{ho} and R_i, and using the equation of spiral ($r = a\theta$, where $a = p/2\pi$ and $p = 2\ d_o$), the value of R_o is found out using equation (7) and shell diameter is calculated using equation (8) [6-16].

$$L = \frac{1}{2a} \times [R_o^2 - R_i^2] \quad (7)$$

$$D_i = (R_0 + d_{ho}) \times 2 \quad (8)$$

This shell diameter is modified as per the nearest value available in TEMA standards.

Further, gas side film heat transfer coefficient is calculated using equations (9) to (13).

$$A_{fl.g} = \frac{\pi}{4} D_i^2 - \left[(L \times d_o) + \left(2 \times \frac{\pi}{4} \times d_{ho}^2\right) + \left(\frac{\pi}{4} d_c^2\right)\right] \quad \dots (9)$$

$$Vg = \frac{\dot{m}_g}{\rho_g \times A_{fl.g}} \quad (10)$$

$$\text{Re} = \frac{\rho_g V_g D_e}{\mu_g} \text{ where } De = do \quad (11)$$

Using Grimson's Correlation for Nusselt number for flow across banks of the tubes:

$$Nu = 1.13 \times C_1 (R_{e\max})^m \Pr^{1/3} \quad (12)$$

$$h_g = \frac{Nu \times k}{d_o} \quad (13)$$

The water side heat transfer coefficient is calculated using equations (14) to (19) assuming n pancakes mounted on one header.

$$A_{fl.w} = \frac{\pi}{4} d_i^2 \times 4 \quad (14)$$

$$Vw = \frac{\dot{m}_w}{\rho \times A_{fl.w}} \quad (15)$$

$$\text{Re} = \frac{\rho_w V_w d_i}{\mu_w} \quad (16)$$

Using Gnielinski's Correlation for Nusselt Number

$$Nu_s = \frac{\left(f/2\right) \times (R_e - 1000) \times P_r}{1 + 12.7 \times \left(f/2\right)^{1/2} \times \left(P_r^{2/3} - 1\right)}$$

valid for $0.5 \le P_r \le 2000$ and

$$2300 \le R_e \le 5 \times 10^6 \quad (17)$$

where, $f = \frac{1}{4} \times (1.82 \times \log R_e - 1.64)^{-2}$

valid for Re from transition to 10^6.

Nusselt number for spiral coil is estimated using Mikheev correlation:

$$\frac{Nu_c}{Nu_s} = \left(1 + 3.54 \frac{a}{R}\right) \times \left(\frac{\Pr_m}{\Pr_w}\right)^{0.25}$$

valid for $\frac{R}{a} > 6$ here $\frac{R}{a} = \frac{R_{avg}}{a}$ (18)

where, a = tube inside radius

$$h_w = \frac{Nu_c \times k}{d_i} \quad (19)$$

The water side heat transfer coefficient may also be calculated using Rangarao correlation for spiral coil as under[3]:

$$Nu_{sp} = 0.296 \times R_e^{0.63} \times P_r^{0.3} \times P_T^{-0.25} \times \Phi_v^{0.14}$$

for Re> 2300 (20)

where $\quad P_T = \frac{R_o^{1.92} - R_i^{1.92}}{P \times d_i^{0.92}}$

$$h_w = \frac{Nu_{sp} \times k}{d_i} \quad (21)$$

The overall heat transfer coefficient based on outside surface area is thus estimated.

$$U_o = \frac{1}{\frac{1}{h_i} + \frac{d_o}{2k} \times \ln \frac{d_o}{d_i} + \frac{1}{h_o} \times \frac{d_o}{d_i}} \tag{22}$$

The number of pancakes required for sensible heating is:

$$N = \frac{q_{sensible}}{U_o \left(\pi do \times L\right) \Delta T_{lm}} \tag{23}$$

The set of equations and procedure adopted for latent heating stage are as under:

$$q_{latent} = m_s \times h_{fg} \tag{24}$$

The calculations for gas side film heat transfer coefficient are similar to that of sensible heating stage. The calculations for water side or steam side film heat transfer coefficient (nucleate boiling coefficient) [10] are done by using equations (25) to (34). For two phase heat transfer, the heat transfer coefficient at a point in a tube is estimated for an assumed dryness fraction of 0.5.

Reynold's number for the liquid phase flowing alone in the tube

$$R_{eL} = \frac{\dot{m} \times (1-x) \times d_i}{\mu_L} \tag{25}$$

HTC for the liquid phase flowing alone

$$h_L = 0.023 \times \frac{k_L}{d_i} \times R_{eL}^{0.8} \times P_r^{1/3} \tag{26}$$

The Lockhart-Martinelli parameter

$$X_{tt} = \left[\frac{(1-x)}{x}\right]^{0.87} \times \left(\frac{\rho_g}{\rho_L}\right)^{0.5} \times \left(\frac{\mu_L}{\mu_g}\right)^{0.125} \tag{27}$$

The forced convective multiplier F

$$F = 2.35 \times \left(\frac{1}{X_{tt}} + 0.213\right)^{0.736} \tag{28}$$

The forced convective coefficient

$$h_{FC} = F \times h_L \tag{29}$$

The two phase Reynold's Number as defined by Chen

$$R_e = R_{eL} \times F^{1.25} \tag{30}$$

The supression factor

$$S = \frac{1}{1 + 2.53 \times 10^{-6} \times R_e^{1.17}} \tag{31}$$

The nucleate boiling coefficient (supressed) is given by

$$h_{NB} = S \times h_{FZ} \tag{32}$$

where h_{FZ} is calculated using Forster- Zuber correlation as follows

$$h_{FZ} = \frac{0.00122 \times \Delta T_{sat}^{0.24} \times \Delta P_{sat}^{0.75} \times C_{pL}^{0.45} \times \rho_L^{0.45} \times k_L^{0.75}}{\sigma^{0.5} \times h_{fg}^{0.24} \times \mu_L^{0.29} \times \rho_g^{0.24}}$$

where $\Delta T_{sat} = \left(T_w - T_{sat}\right)$

ΔP_{sat} is the pressure at ΔT_{sat} (33)

Total Heat Transfer Coefficient for two phase flow

$$h_i = h_{FC} + h_{NB} \tag{34}$$

The overall heat transfer coefficient based on outside surface area and then number of pancakes required for latent heating is estimated adopting the same procedure as that of sensible heating stage.

Finally, pressure drop is calculated using equation:

$$\Delta p = \frac{4 f l V^2 \rho}{2 D_e} \quad \text{where } f = \frac{16}{R_e} \tag{35}$$

3 Problem Solution

As per the data collected from actual site and the formulae stated in above section, if the producer gas is cooled from 300°C to 140°C, the heat potential available is 86.8 kW. From this heat, around 119 kg/hr steam can be generated at 2.5 bar gauge pressure. Sensible heat duty required to heat the water from 25 °C to 139 °C is 15.8 kW. LMTD for counter flow arrangement is 123°C and during sensible heating gas is being cooled down to 271°C. Length of the tube for one pancake is assumed as 6 meters with inside diameter of the spiral 0.2032 m. Outer diameter of spiral is calculated to be 0.2425 m, accordingly shell diameter required to fit this spiral is 21 inches which modified by nearest shell diameter available in TEMA standards is 21 ¼ inches (0.5397 m).Using the set of equations stated in earlier section, calculated value of gas side film heat transfer coefficient is 97.18 W/m^{2}K. The water side heat transfer coefficient is calculated by using two separate correlations viz. Mikheev's correlation [11] and Rangarao's correlation [3], both predicting approximately same values i.e. 1145.7 W/m^{2}K and 1144.9 W/m^{2}K respectively. Value obtained using Mikheev's correlation is used for calculation of overall heat transfer coefficient based on outer surface area, which is 77.5 W/m^2K. Subsequently the number of pancakes required for sensible heating is estimated as 7. If 4 pancakes are mounted on one header then 2 headers are required for sensible heating of water from 25°C to 139°C. Similarly calculations are made for latent heating. The latent heat duty required to generate dry saturated steam at 2.5 bar gauge pressure is 71 kW. Calculated value of gas side film heat transfer coefficient is 103.7 W/m^2K and the same for water side is 14376.2 W/m^2K. The overall heat transfer coefficient for latent heating stage is therefore 88 W/m^2K and number of pancakes required for this work is 23. If 4 pancakes are mounted on one header then 6 headers are required for latent heating. Pressure drop at gas side is around 0.0004 bar for this heat exchanger.

4 Conclusion

An analytical model is developed for carrying out design simulations of the Pancake type heat exchanger. For the validation of the analytical model, pancake type heat exchanger is fabricated in the laboratory with two sets of four pancakes. A total of eight pancakes are used in the heat exchanger. Stainless steel (SS-316 seamless) is used for tubes and mild steel for shell. This pancake bundle of eight spirals is placed in a shell. This heat exchanger is tested for the heat transfer between engine exhaust through shell and water through tubes. The experimental results obtained are presented in Fig.2 to Fig.5.

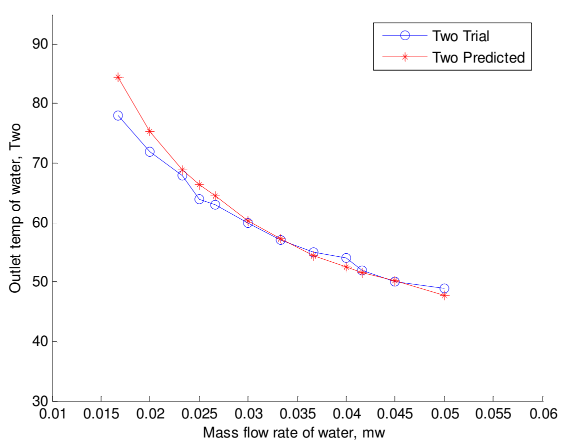

Fig. 2 T_{wo} Vs Mass Flow Rate of Water

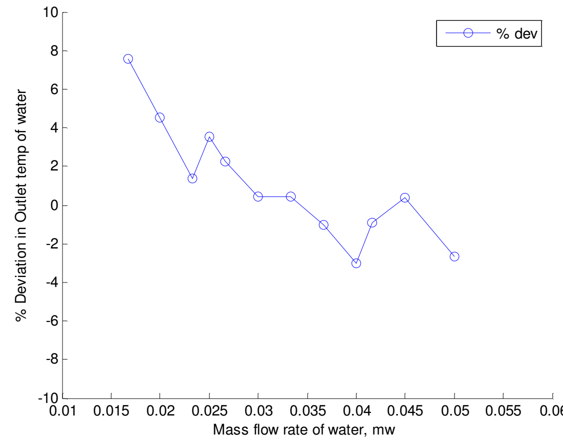

Fig. 3 % Deviation in outlet temp. of water

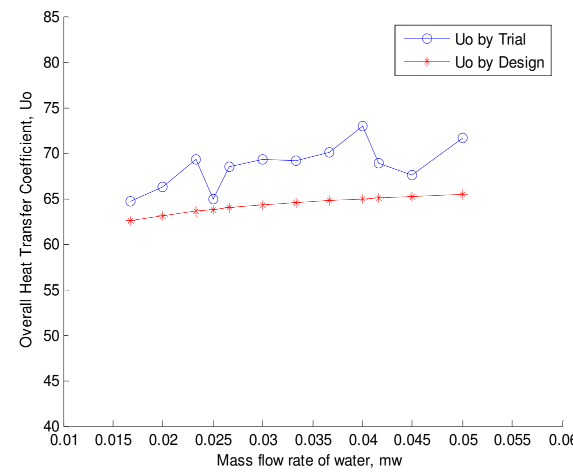

Fig. 4 U_o Vs Mass Flow Rate of Water

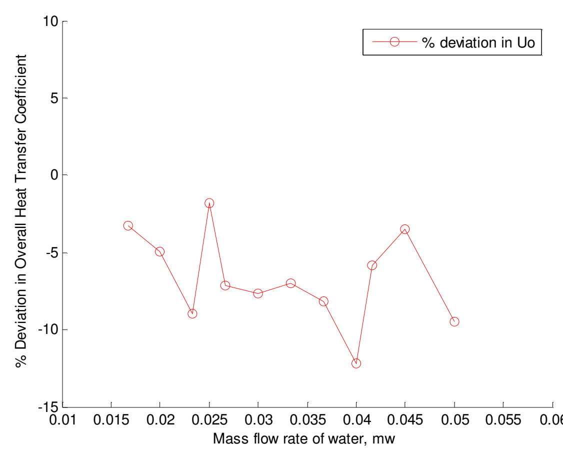

Fig. 5 % deviation in U_o Vs Mass Flow Rate of Water

The experimental results show that the deviation between calculated values of overall heat transfer coefficient (from the experimental results) and theoretical values obtained (from the analytical model) are within 12%. Also, the accuracy is found to be within ±8 % in approximation. The pressure drop estimated is also compared with actual values observed during experimentation, which is found in acceptable range. It is stated in the existing literature that each correlation is reasonable over a certain range of conditions, but for most engineering calculations one should not expect accuracy to much better than 20% [13].

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