Design of Shell and Spiral Tube Heat Exchanger for Waste Heat Recovery

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***Abstract*— One of the most important parts of saving energy is getting rid of waste heat. Many factories, businesses that provide services, industrial processes, and machines waste a lot of energy that goes up the stacks and into the air or is lost in some other way. This waste heat will be liquid, gaseous, or a mix of the two, and its temperature will range from a little above room temperature to a few hundred degrees Celsius. Waste heat recovery (WHR) is the process of capturing and moving waste heat with the help of a gas or liquid and sending it back to the system as an extra source of energy. Recovery of waste heat is a very important part of saving energy and living in a sustainable way. Depending on the temperature of the waste heat, different methods are used to get the heat back. In this paper, a heat exchanger called a "pancake" is proposed. This type of heat exchanger can be used to recover heat from different manufacturing and industrial processes, such as those in the food and chemical industries. A shell and pancake type tube heat exchanger is a proposed heat exchanger that will be made to improve the rate of waste heat recovery and the effectiveness of the heat exchanger. When compared to a traditional shell-and-tube heat exchanger, the size of the unit will be smaller if pancake tubes are used instead of shell-and-tube tubes.**

***Keywords* - Waste heat recovery, Shell and pancake type tube heat exchanger, overall heat transfer coefficient**

I. INTRODUCTION

With growing concern about global warming, the engineering industry is being tasked with reducing greenhouse gas emissions and improving the efficiency of their sites. In this regard, the use of waste heat recovery systems in industrial processes is one of the major areas of research to reduce fuel consumption, harmful emissions and improve efficiency. Heat exchangers are widely used in the process industries and chemical industries for various applications including refrigeration and air conditioning systems, food industries, chemical processing etc. Hence the use of high-performance heat exchanger is very important for energy conservation.
Different methods are used to increase the thermal performance of heat exchanger; one of them is augmentation heat transfer. Heat transfer by augmentation is broadly classified into three categories: Active, Passive, and Compound technique (combination of active and passive). In active techniques external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer is observed. This can be achieved by mechanical aids, surface vibration, and electrostatic fields, etc. while Passive Techniques do not require any direct input of external power; these use power from the system itself which ultimately leads to sufficient exchange of heat at the cost of noticeable fluid pressure drop [1]. This technique promotes higher heat transfer coefficients by disturbing or altering the existing flow behavior except for extended surfaces. When two or more such techniques are employed simultaneously to enhance heat transfer coefficient then that technique is called compound technique.
According to the different industrial processes, industries produce different quality and quantity of waste heat. In order to utilize the waste heat produced by the industry it is necessary to analyze the industrial processes used in major energy consuming industries to investigate and apply suitable waste heat recovery methods. During research it was found that cement, ceramics, iron and steel, refineries, glassmaking, chemicals, paper and pulp, and food and drink industries are the major source of waste heat [2]. According to the availability of waste heat temperature, different techniques are used to recover heat loss. Heat loss is classified into high, medium and low temperature. According to the temperature range waste heat recovery (WHR) system is introduced to get maximum efficiency [3]. High temperature, WHR system recovers waste heat above 400ºC, medium temperature range is between 100 to 400ºC and low temperature range is below 100ºC [4].

II. LITERATURE REVIEW

In waste heat, heat rejected at any temperature can be utilized; conventionally, at high temperature, higher quality of heat obtained which can be optimized easily in waste heat recovery process. Hence it is important to discover and recover the maximum amount of heat from high potential process to obtain maximum efficiency

The quantity or the amount of available waste heat can be calculated using the equation shown below.

$$Q = v \times \rho \times CP \times \Delta T \qquad (1)$$

where, Q (J) is the heat content, v is the flow rate of the substance (m^3/s), ρ is density of the flue gas (kg/m^3), Cp is the specific heat of the substance (J/kg.K) and ΔT is the difference in substance temperature (K) between the final highest temperature in the outlet (T_{out}) and the initial temperature in the inlet (T_{in}) of system [3]

A. Heat Recovery Systems

Heat recovery systems are usually used to exchange heat to and from gases, liquids, vapours and solid particle streams. They can be grouped as follows:

- Gas/gas;
- Gas/liquid;
- Liquid/liquid;
- Liquid/vapour(evaporation);

NOMENCLATURE

A	Area for heat transfer, m^2
A_{fl}	Area available for flow, m^2
C_i	Constant in Richard et.al. correlation
T	Temperature, ˚C
a	Constant in equation of spiral
Cp	Specific heat, kJ / kg k
ΔT_{lm}	Log mean temperature difference, ˚C
D	Diameter of shell, m
U_o	Overall Heat Transfer coefficient, W/m^2 K
De	Dean Number
V	Velocity, m/s
v	Volume flow rate m^3/s
d	Diameter of tube, m
f	Friction factor
h	Heat transfer coefficient, W/m^2 k
L	Length of tube for Pancake, m
P	Pressure, bar
p	Pitch
Nu	Nusselt Number
Re	Reynolds Number
q	Total Heat Duty, kW
R	Radius of spiral, m
m	Mass Flow rate, kg/s
Pr	Prandtl number
d_{ho}	header at outlet
C_c	heat capacity of cold fluid
C_h	heat capacity of hot fluid

Greek Symbol

K	Thermal conductivity, W/m˚K
ρ	Density, kg/m^3
μ	Dynamic Viscosity, kg/ms
$\in$	Effectiveness
θ	Angle in radians in case of spiral

Subscript

O	outlet
w	water
a	air
i	inlet
h	hot
c	cold

- Vapour/liquid (condensation); and
- Heat pumps

The most appropriate device selection will depend on many factors including:

- Maximum operating pressure:
- Minimum and maximum temperatures.
- Fluid limitations; and
- Available equipment sizes.

Depending on the type and source of waste heat and in order to justify which waste heat recovery system can be used. Also, it is important to investigate the amount and temperature of heat that can be recovered from the process.

During the study of different types of heat exchanger, it was found that shell with different shape tubes is often used in industries. Many researchers have proved that, helically coiled tubes are superior to straight tubes for heat transfer application. [10], [11]. Helically coiled tubes with compact heat exchangers are also used in various industrial applications such as petroleum, refrigeration, HVAC (Heating, Ventilation and Air Conditioning), food, nuclear, solar systems, etc. [12]. Spiral tube heat exchanger has many advantages including high overall heat transfer coefficient, high effectiveness, compact size etc. Also, helical geometry permits handling of high temperatures and extreme temperature differentials without high induced stresses or costly expansion joints. Spiral tubes can handle high pressure fluid effectively and it is also found in the studies that fouling in spiral tubes is low at high turbulence. Usually, clean streams of gases from production processes were used in iron and steel industry for waste heat recovery. For example, [13] in an experiment demonstrated that flat heat pipe (FHP) in steel production can recover heat up to 15.6 (kW) for a source temperature of 450°C. In this experiment the dimension of 1m height x 1m width innovative FHP model was constructed and tested. The model of flat heat pipe which was inclined at an angle to horizontal was charged with water flow rate 0.38 kg/s and hot gases dispersing from the process imparted on that flat heat pipe model.

While studying the thermal performance and frictional characteristics of shell and helically coiled heat exchanger it is found that many researchers have proposed various correlations for Nusselt number and friction factor for both shell and tube side. During the

investigation of thermal performance and pressure drop of helical coil heat exchanger with and without helical crimped fins [14] found that, mass flow rate of hot and cold-water inlet and hot water inlet temperature have significant effect on the heat exchanger effectiveness. In the experiment, heat exchanger was formed from shell and helically coiled tube with two different coil diameters. During the experiment the inlet temperature of cold and hot water ranged from 15 to 25ºC and between 35 to 45ºC respectively. The mass flow rate of cold and hot water ranges from 0.10 to 0.22 kg/s and 0.02 to 0.12 kg/s, respectively.

Similarly, a Shell and Pancake type heat exchanger for waste heat recovery from producer gas was designed, fabricated and tested [15], during experimentation eight stainless steel tube (SS-316 seamless) pancakes were judiciously assembled in a mild steel shell of 0.54 m ID to form a shell and pancake heat exchanger. Pancakes type tubes are used for cold fluid (water) and shell is used for hot fluid (exhaust gas). Length of the tube for one pancake was approximately 6 m and ID & OD was 0.20 & 0.24m respectively. In result it was found that heat transfer coefficient for water and gas was 1145.7 W/m^2K and 97.18 W/m^2K respectively also the overall heat transfer coefficient was 77.5 W/m^2K which is in the range of 12% of theoretical calculation with 8% accuracy. Also, the pressure drop was within the acceptable range. To scale up this laboratory scale model for industrial use, theoretically it was found that seven pancakes are required just to sensibly heat the water and twenty-three pancakes are required for latent heating of water at 2.5 bar pressure (gauge).

In an experimental study of thermal performance of shell-and-coil heat exchangers, [16] shell and three different spherical coils of copper were fabricated and tested at different hot water temperatures and different mean mass flow rates of shell and coil. In investigation it was found that effectiveness of heat exchanger decreases when the ratio of mass flow rate of tube-side to shell-side (R_m) increases. When R_m is equal to one, shell and coil side temperature profile from top to bottom of H.X was linear. When R_m is less than one shell side temperature distribution was linear (i.e., h is constant) and coil side temperature profile was concave i.e., Coil side surface temperature is higher at the top and it drops at a faster pace while moving towards the bottom of heat exchanger. And when R_m is greater than one, shell side temperature profile was not linear or logarithmic and coil side temperature profile changes from concave up to concave down i.e., coil surface is not much hot at the top part and its temperature gradually decreases when it reaches at the bottom of heat exchanger.

An experiment to find optimal and critical values of geometrical parameters of shell and helically coiled tube heat exchanger was performed [17]. During the experiment dimensionless geometrical parameters p/dt, dc/dt, Hc/f, Hsh/dsh, dv/f was considered to find coefficient of design, (COD). And it was found that optimal values for these numbers are 1.54, 10, 0.909, 2.4 and 0.107 similarly critical values obtained was 3.077 < p/dt < 4, 7, 0.479, 1.091, 0.083. These optimal and critical values of dimensionless parameters are obtained to maximize and minimize the heat COD.

In investigation on thermal analysis of conical coil heat exchanger [18] fifteen conical coils of different cone angles (0° (helical), 45°, 90°, 135°, 180° (spiral)), with three different tube sizes (ID x OD, 8 x 10, 10 x 12 and 12 x 15) are fabricated and analyzed. Hot water of 70ºC and cold water of 25.5ºC with a flow rate of 10 to 100 lph and 30 to 90 lph respectively was used to perform experiments. And it was found that the Nusselt number (Nu) increases when Renault number (Re) increases inside the tube for constant shell side water flow. It was found that Nusselt number is high at 0º and reduces from 0º to 180º. Similarly, effectiveness of heat exchanger decreases when Renault number increases, and it is maximum at 0º and minimum at 180o. Also, it was found that Nusselt number decreases when inside diameter increases for the same constant hot and cold-water flow rate. And friction factor decreases when Renault number increases i.e., friction factor is high in spiral coil and reduces as the angle reduces from 180º to 0º.

Similarly, during the study of intensification of the heat transfer in helically coiled tube heat exchangers via coiled wire inserts [12]. Helically coiled tube heat exchanger with coiled wire inserts of various cross sections and dimensions was investigated. In investigation it was found that wire diameter of 0.008 m, enhanced Nusselt number & friction factor by 131.9 – 340.9% & 338.9 – 536.1% respectively as compared to empty smooth coiled tube. The inserts with the cross sections of "two rectangular", "square", "four circular cross section wire with diameter equal to 0.0002 m" and "concentric circular wire with diameter equal to 0.002 m" increases the heat transfer rate by 12.3%, 11.4%, 2.5% and 1.1%, respectively for the inlet flow rate equal to 0.05 kg/s. Furthermore, these cross sections intensify the heat transfer for the inlet mass flow rate of 0.075 and 0.1 kg/s.

III. PROPOSED HEAT EXCHANGER

Medium temperature waste heat recovery is considered to design a suitable heat exchanger which can recover the sensible heat from different industrial processes. Water and hot air are considered as a working fluid. Also, it is considered that hot air will flow from shell and water will flow from pancake. Total heat available is.

$$q_a = m_a \times Cp_a \times (T_{h.i} - T_{h.o}) \quad (2)$$

If mass flow rate of hot air and water is considered constant, then heat absorbed by the water is

$$q_w = m_w \times Cp_w \times (T_{c.o} - T_{c.i}) \quad (3)$$

Flow arrangement is selected as counter flow arrangement and by applying the effectiveness-NTU method outlet temperature of hot and cold fluid can be calculated. Hot air is flowing through the shell and water is flowing through the tubes. Hot air side heat transfer coefficient can be calculated by using the following equations

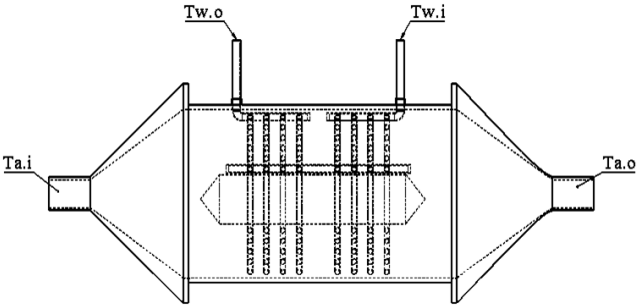

Fig: 1 Proposed Model of Shell and Pancake Tube Heat Exchanger

$$m_a\, Cp_a\ (T_{h.i} - T_{h.o}) = m_w\ Cp_w\ (T_{c.o} - T_{c.i}) \quad (4)$$

$$\in = \frac{Q_{actual}}{Q_{max}} \quad (5)$$

$$= \frac{C_h \times (T_{h.i} - T_{h.o})}{C_{min} \times (T_{h.i} - T_{c.i})} \quad (6)$$

Where, C_h and C_c are heat capacity of hot and cold fluid. Now assuming geometrical dimensions viz. length of pancake tube, core diameter of pancake, outer diameter of pancake tube, dho and Ri , and using the equation of spiral (r = aθ, where a = p/2 and p = 2do), the value of Ro is found out using equation (7) and shell diameter is calculated using equation (8).

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

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

This shell diameter is modified as per the nearest value available in TEMA standards. Further the available area for hot air flow is

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

$$V_a = \frac{\dot{m}}{\rho_a \times A_{f.a}} \quad (10)$$

$$R_e = \frac{\rho_a \times V_a \times d_o}{\mu_a} \quad (11)$$

Considering aligned tube arrangement and using Richard et.al Correlation for Flow across banks of the tubes is

$$Nu_a = 0.192 \times Re^{0.81} \times Pr^{1/3} \quad (12)$$

Valid for 2000 < Re < 4000

$$h_a = \frac{Nu_a \times k_a}{d_o} \quad (13)$$

Similarly, water side heat transfer coefficient can be calculated by using the following equations.

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

Where n= no of pancakes mounted on header

$$V_w = \frac{\dot{m}}{\rho_w \times A_{f.w}} \quad (15)$$

$$Re = \frac{\rho_w \times V_w \times d_o}{\mu_w} \quad (16)$$

Nusselt number calculation for water side flow is

$$Nu_w = 0.012 \times (Re_w^{0.87} - 280) \times Pr_w^{0.4} \times [1 + (\frac{d_i}{L_s})^{\frac{2}{3}}] \quad (17)$$

Valid for 2000 < Re < 106

$$h_w = \frac{Nu_w \times k_w}{d_i} \quad (18)$$

In water side heat transfer, tubes are pancake type tubes, so it is necessary to consider curvature ratio for such tubes. Also, during the fluid flow a secondary flow is induced because of centripetal force. Dean number (De) shows the relation between the centripetal force and tube curvature ratio. It can be calculated by using the following equation.

$$De = \frac{\sqrt{\frac{1}{2}(\text{Inertia Fore})\times(\text{Centripetal Force})}}{\text{Viscus Force}}$$

$$= Re \times \sqrt{\frac{d_i}{2R_c}} \quad (19)$$

Tube side pressure drop, and friction factor can be calculated as

$$f = 2\,\frac{\Delta P}{L} \times \frac{d_i}{\rho v^2} \quad (20)$$

$$f = \frac{64}{Re} \quad (21)$$

Nusselt number correlation can also be obtained by using Dean Number. Purandare[18] has studied and developed the correlation between Nusselt number and Dean number for spiral tubes.

$$Nu = 0.124\,De^{0.80} \times Pr^{-0.259} \quad (22)$$

The value of Nusselt number obtained from equation no (17) and (22) will give the same value. The overall heat transfer coefficient based on outside surface area is thus estimated as

$$U_o = \frac{1}{\frac{1}{h_i} + \frac{d_o}{2k}\ln\frac{d_o}{d_i} + \frac{1}{h_o}\frac{d_o}{d_i}} \quad (23)$$

From the charts of NTU vs. effectiveness, for single shell pass with multiple tube passes and 0.25 capacity ratio, NTU obtained is 1.1. After getting the outlet temperature of cold and hot fluid with Counter flow arrangement and with LMTD method the number of pancakes turns required will be calculated as

$$\Delta T_{lm} = \frac{(T_{h.i} - T_{c.o}) - (T_{h.o} - T_{c.i})}{\ln\frac{T_{h.i} - T_{c.o}}{T_{h.o} - T_{c.i}}} \quad (24)$$

$$N = \frac{q_w}{U_o(\pi d_o L)\Delta T_{lm}} \quad (25)$$

IV. Conclusion

Eventually, the proposed design of a shell and pancake type heat exchanger will help to safeguard the environment from greenhouse gases and global warming by efficiently recovering waste heat from medium and large temperature heat sources. You might think of this heat exchanger as compact and effective. With 8 pancake tubes, the estimated output temperature of water is 82°C, which is roughly more than 50% of heat recovery if the hot stream input (air) is considered at 160-oC and the cold stream inlet (water) is considered at 30°C. This heat exchanger should have an efficiency more than 0.55, according to projections. This exchanger will be manufactured and tested in a relatively short amount of time. If the results are verified by experimental testing, the design can be modified to work with a wider range of temperatures and a wider variety of fluids. Cleaning methods for heat exchangers and tubes can be tailored to either soot development or other types of fouling. After the model has been validated, modifications can be made to increase or decrease its size as needed for the intended industrial use, taking into account the constraints imposed by the construction of the spiral tube. Recovering heat from hot streams that would otherwise be lost by being released into the atmosphere is a good first step towards energy conservation. The proposed heat exchanger geometry will increase the rate at which such waste heat is recovered and put to use in the process industry. Indirectly, this far-out technology could be a huge boon to the movement towards renewable energy and ecological preservation.

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