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HEAT TRANSFER ANALYSIS ON SHELL AND TUBE HEAT EXCHANGERS

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Abstract

In the present study inlet temperature of shell and tube side are taken as input parameters with a given bundle arrangement of square pitch. The thermal analysis is done firstly taking water inside the tube and steam on shell side. The design of shell and tube exchanger using Kern method for water and steam combination is validated by well-known Dittus-Boelter equation of turbulent flow inside tube. The analysis is extended using the above Kern method with different fluid combinations such as sulphur-dioxide on the tube side steam on shell side and carbon-dioxide side on tube side and steam on shell side, Parameters such as heat transfer coefficient, friction coefficient, length, area and pressure drop are determined. "C" Program is written to evaluate the above parameters. Graphs are drawn to depict the behavior for different fluid combinations. The results are tabulated.

Keywords: Shell and tube Heat Exchanger; Dittus-Boelter equation; Kern Method.

1. INTRODUCTION

1.1 Heat Exchanger:

Heat exchanger is equipment which transfers energy from a hot fluid to cold Fluid. While the fluid is passing through the heat exchanger, the temperature of Fluid changes along the length of heat exchanger. The present study focuses mainly on design of heat exchanger known to be shell and tube heat exchanger. A shell and tube heat exchanger is a class of heat exchanger designs. It is most common type of heat exchanger in oil refineries and other large chemical processes and is suited for higher pressure applications. This type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid run-through the tubes and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids.

Heat exchangers are devices to recover heat between two process streams. The use of heat exchangers is extensive in power plants, refrigeration and air-conditioning systems, Space

Chemical, Nuclear, Petrochemical, and Cryogenic industries.

Heat exchangers appear in variety of shapes and sizes It can be as huge as a power plant condenser transferring hundreds of Megawatts of heat or as tiny as an electronic chip cooler which transfers only a few Watts of thermal energy.

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The heat exchangers are classified upon the following factors:

- Construction
- Flow arrangement
- Number of shells
- Contact between the processing streams
- Compactness
- Heat transfer mechanism

Shell and tube heat exchangers consist of a series of tubes. One set of these tubes contains the fluid that must be either heated or cooled. The second fluid runs over the tubes that are being heated or cooled so that it can either provide the heat or absorb the heat required. A set of tubes is called the tube bundle and can be made up of several types of tubes: plain, longitudinally finned, etc. Shell and tube heat exchangers are typically used for high pressure applications (with pressures greater than 30 bar and temperatures greater than 260^{0}C)

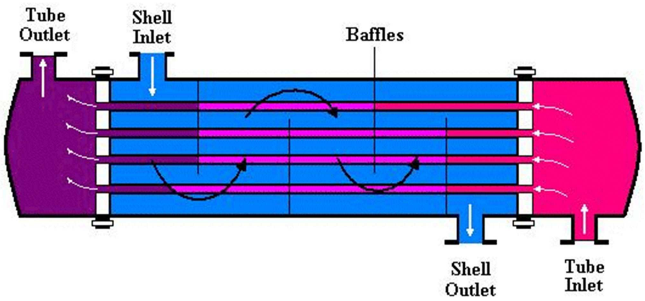

Fig.1.1. Shell and Tube Heat Exchanger

1.2 Literature Review:

In designing of shell and tube heat exchangers, authors [1-20] paid attention in analyzing this problem both experimentally and theoretically.

Kern [1] provides correction factor charts for different number of shells and even number of tube passes. He presented the correction factor F, as a function of two variables R and **S**, which depends on the inlet and exit temperatures of the heat exchanger of both the fluids. Roetzel and Nicole [2] has recognized the potential usefulness of explicit representations of LMTD Correction factors in developing computerized packages for heat exchanger design. Tinker [3] has suggested a schematic flow pattern, which divided the shell –side flow into a number of individual streams. Tinkers model has been the basis of “Stream analysis method”, which utilizes a rigorous reiterative approach and it is particularly suitable for computer calculations rather than hand calculations.

Saunders [4] in his book proposed practical method and simple design factors are provided and the method is used rapidly for fixed set of geometrical parameters. In his work the correction factors are for heat transfer and pressure drop correlations. Wills and Johnston [5] in his article developed a stream analysis method that is visible for hand calculations. They developed a new and accurate hand calculation method for shell and tube pressure drop and flow distributions

Reppich and Zagermann [6] in his paper offers a computer- based design model to determine the optimum dimensions of segment ally baffled shell-and-tube heat exchangers by calculating optimum shell-side, tube – side pressure drops from the equation provided in his work. The six optimized dimensional parameters are number of tubes, tube length, shell diameter, number of baffles, baffle cut, and baffle spacing. The proposed model also carries out also cost analysis. Lam and Lo [7] in his article presented the flow patterns and corresponding analysis of four circular cylinders subjected to a cross flow. Experiments were carried out at sub critical Reynolds number of 2100. Square arrangement of the cylinders with varying spacing ratios and angles of incidents were examined.

Motos et al [8] explained in their study a two dimensional heat transfer analysis in circular and elliptical tube heat exchangers. A finite element method is used to fluid flow and heat transfer equations and Two dimensional Isoperimetric, Four nodded, linear element was implemented for the FEA progress**.** An and

Choi [9] developed procedure for the detailed phenomena in shell and tube heat exchangers to predict the heat and mass transfer characteristics of shell and tube heat exchangers.

Moghadassi and Hosseini[10] had given the importance of heat exchangers in chemical and petrochemical industries, Heat exchangers analysis and heat translate calculations are provided. The conventional and prevalent methods (such as KERN method) are presented. Liljana Markovska and Vera Mesko [11] gave optimization of shell-and-tube-heat exchanger is accomplished by use of the optimizer software package. The objective function is defined together with the implicit constraint. The simultaneous equation solving method is used to solve the equations that describe the process. Lebele-Alawa and Victor Egwanwo[12] presented a numerical method of solution, capable of accounting for temperature dependent variation of fluid properties and heat transfer. Field data were collected for three different industrial heat exchangers and basic governing equations were applied. The parameters analyzed include: the outlet temperatures, the heat transfer coefficients and the heat exchanger effectiveness.

Naik and Matawala [13] studied design and assessment of counter flow shell and tube heat exchanger by entropy generation minimization method. McAdams [14] was one of the earliest workers to quantitatively demonstrate this. His analysis was simple and based on tubular heat exchanger. By taking into account the cost of power and fixed cost of the exchanger, per unit heat transferred, simple expressions for estimating the optimum mass velocities for both inside tubes and outside tube fluids are developed. Peter and Timmerhaus [15] recognized the importance of optimizing tube side pressure drop; shell side pressure drop and heat transfer area simultaneously. Consequently, they produced the most detailed and useful work to date on a single shell-and-tube heat exchanger optimization. The problem with their method however, is that it is restricted to shell-and-tube heat exchangers fitted with plain tube. Extension to other exchanger types requires new equations. No guidance is given on how to generate these equations.

2. DESIGN PROCEDURE OF SHELL AND TUBE HEAT EXCHANGER

2.1 Step wise Procedure for Calculation:

A heat exchanger can be designed by the LMTD when inlet and out let conditions are specified. When the problem is to determine the inlet and outlet temperatures for a particular heat exchanger, the analysis is performed more easily by using a method based on effectiveness of the heat exchanger and number of transfer units (NTU)

The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\epsilon = \frac{actual\ heat\ trensfer}{maximum\ possible\ heat\ trensfer} = \frac{Q}{Q_{max}} \qquad (2.1)$$

The actual heat transfer rate Q can be determined by energy balance equation,

$$Q = \dot{m}_h c_{ph}(t_{h1} - t_{h2}) = \dot{m}_c c_{pc}(t_{c2} - t_{c1}) \qquad (2.2)$$

The fluid capacity rate C:

$\dot{m}_h c_{ph} = c_h$ = hot fluid capacity rate

$\dot{m}_c c_{pc} = c_c$ = cold fluid capacity rate

c_{min} = the minimum fluid capacity rate (c_h or c_c)

c_{max} = the maximum fluid capacity rate ($c_h$or c_c)

The number of transfer units (NTU) $= \frac{UA}{C_{min}}$ (2.3)

Where,

U = overall heat transfer coefficient in $W/m^2.K$

A = surface area in m^2

The effectiveness $\epsilon = \frac{c_h\,(t_{hi} - t_{ho})}{c_{min}(t_{hi} - t_{ci})} = \frac{c_c\,(t_{co} - t_{ci})}{c_{min}(t_{hi} - t_{ci})}$ (2.4)

The governing equations for design problem are usually given as follows:
Heat rate
$Q = C_h(T_{hi} - T_{ho}) = C_c(T_{co} - T_{ci})$ (2.5)
Where,
Q= heat duty of heat exchanger, W
C_h= specific heat of the hot fluid, J/kgK
C_c = specific heat of the cold fluid, J/kgK
T_{hi}= temperature of the hot fluid inside, K
T_{ho}= temperature of the hot fluid outside, K
T_{ci}= temperature of the cold fluid inside, K
T_{co}= temperature of the cold fluid outside, K
Where heat capacity rate for hot or cold fluid
$C = \dot{m}C_p$
Where,
$\dot{m}$= mass flow rate, kg/sec
C= heat capacity
Log mean temperature difference for pure counter flow
$\Delta T_{lm,cf} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{ln[(T_{hi} - T_{co})/(T_{ho} - T_{ci})]}$ (2.6)
The effective mean temperature difference for cross flow
$\Delta Tm = F\Delta T_{lm,cf}$
Where F=correction factor
Shell-side area is calculated by: $A_s = \frac{D_s\,CB}{P_t}$ (2.7)
Where,
D_s=shell diameter, m
C= clearance, m
B= baffle spacing, m
$G_s = \frac{\dot{m}_s}{A_s}$ (2.8)
Where, $\dot{m}_s$= mass flow rate of shell side, kg/s
A_s= area of the shell, m^2
$D_e = \frac{4(P_t^2 - \frac{\pi D_o^2}{4})}{\pi\, D_o}$ (2.9)
Where P_t= tube pitch, m
D_o= outer Diameter of the tube, m
$Re_s = \frac{D_e\, G_s}{\mu}$ (2.10)
Where D_e= equivalent diameter, m
μ= dynamic viscosity, Ns/m^2
Shell side Nusselt number is given by Kern
$Nu = 0.36\left[\frac{D_e\, G_s}{\mu_b}\right]^{0.55}\left[\frac{C_p\, \mu_b}{k}\right]^{0.33}\left[\frac{\mu_b}{\mu_w}\right]^{0.14}$ (2.11)
Where
k= thermal conductivity, W/mK
μ_w= dynamic viscosity of water fluid, Ns/m^2
μ_b= shell fluid dynamic viscosity at average temperature, Ns/m^2
G_s= mass velocity of shell side, $kg/m^2.s$

D_e= equivalent diameter of shell side,m

The shell-side heat transfer coefficient, h_o, is then calculated as:

$h_o = \frac{Nu.k}{D_e}$ (2.12)

where

h_o= heat transfer coefficient, W/m^2.k

k= thermal conductivity, W/mK

Tube-side heat transfer coefficient by:

$A_t = \frac{\pi d_i^2}{4}$ (2.13)

Where $_{Di}$= tube inner diameter, m

$A_{tp} = \frac{N_t A_t}{no.of\ passes}$

Where N_t= number of tubes

$G_t = \frac{\dot{m}_t}{A_{tp}}$ (2.14)

Where G_t= mass velocity of tube, kg/m^2.s

A_{tp}= heat transfer area based on tube surface, m^2

$u_t = \frac{G_t}{\rho}$ (2.15)

Where

ρ= density of fluid at average temperature, kg/m^3

$Re_t = \frac{u_t \rho d_i}{\mu}$ (2.16)

Where

d_i= inner diameter of tube, m

Using the petukhov and kirillov correlation:

$Nu = \frac{\left(\frac{f}{2}\right) RePr}{1.07+12.7\left(\frac{f}{2}\right)^{1/2}(Pr^{2/3}-1)}$ (2.17)

Where

f= friction factor of flow

Re= Reynolds number

Pr= prandtl number

Where $f = (1.58 \ln Re - 3.28)^{-2}$ (2.18)

The tube-side heat transfer coefficient, h_i, is then found as:

$h_i = \frac{Nu.k}{d_i}$ (2.19)

The shell-side pressure drop can be calculated from equations

$\Delta P = \frac{f G_s^2 (N_b+1) D_s}{2\rho D_e \phi_s}$ (2.20)

Where

ΔP= pressure drop for shell side, Pa

N_b= number of baffles

ϕ_s= viscosity correction factor for shell side fluid

$N_b = L/B$

$f = \exp(0.576 - 0.19 \ln R_s)$ (2.21)

The tube-side pressure drop can be calculated from Equation:

$\Delta P_t = (4f \frac{LN_p}{d_i} + 4N_b) \frac{\rho u_m^2}{2}$ (2.22)

where N_p= number of passes

f= friction factor of tube side

2.2 Step Wise Procedure for calculation:

The following steps are adopted for the calculation of parameters of shell and tube heat exchanger

1. The outlet temperatures of shell and tube heat exchanger are computed by equations (2.4) and (2.5)
2. The log mean temperature difference to the shell and tube are computed using equations (2.6)
3. Reynolds numbers on shell side using the following equations (2.10) is caluclated
4. Nusselt number on shell side using the equation (2.11) by using Macadam's correlation is computed
5. Heat transfer coefficient on shell side using the equation (2.12) is calculated.
6. Pressure drop on shell side using the following equation (2.20) is calculated.
7. Tube side pressure drop by using equations (2.16), (2.18), (2.22) is calculated.

3. NUMERICAL SOLUTION PROCEDURE

3.1. Assumptions:

1. Area of heat exchanger is taken as $14m^2$
2. Number of tubes is taken as 81
3. Number of passes is 1 (single pass)
4. Shell side diameter is (Ds) taken as 0.38735 m
5. Tube side outer diameter and inner diameter are taken as 0.0254 m and 0.02291m.
6. Mass flow rates of cold fluid and hot fluid that is m_c and m_h are 56000 kg/h and 80000 kg/h.
7. Bundle arrangement of the tubes is square pitch.

3.2. Input Data for water-steam combination:

Inlet temperature of shell side (T_{hi}) = 393 K

Inlet temperature of tube side (T_{ci}) = 303 K

Mass flow rate of shell side ($\dot{m}_s$) = 15.5 kg/s

Mass flow rate of tube side ($\dot{m}_t$) = 22.2 kg/s

Outer diameter of the tube side (D_o)= 0.0254 m

Inner diameter of the tube side (D_i) = 0.0229 m

Shell diameter (D_s) = 0.38735 m

Pitch size (P_t) = 0.03175 m

Clearance (C) = 0.00635 m

Baffle spacing (B) = 0.3048 m

Thermal conductivity of material for steel (k) = 54 W/m.K

Specific heat of the shell side fluid $(Cp)_s$ = 2135 J/Kg.K

Absolute viscosity of the shell side fluid (μ_{bs}) =0.00001196 Kg/m.s

Thermal conductivity of the shell side fluid (K_s) = 0.02373 W/m.K

Density of the shell side fluid (ρ_s) = 0.598 Kg/m^3

Prandtl number of the shell side fluid (Pr_s) =1.08

Specific heat of the tube side fluid (C_{pt}) = 4178J/Kg.K

Absolute viscosity of the tube side fluid (μ_{bt}) = 0.0006537 Kg/m.s

Thermal conductivity of the tube side fluid (K_t) = 0.6280 W/m.K

Density of the tube side fluid (ρ_t) = 995 Kg/m^3

Prandtl number of the tube side fluid (Pr_t)= 4.340

The exit temperatures on shell and tube side :

Assuming overall heat transfer coefficient for water steam, U = 3000 W/m^2.K

Hot fluid capacity rate,c_h =15.5(2206) =34193 W/K

Cold fluid capacity rate,c_c = 22.2(4178) =92751 W/K

Then $\frac{C_{min}}{C_{max}} = \frac{34193}{92751} = 0.3686$

Number of transfer units,NTU = $\frac{UA}{C_{min}} = \frac{3000(14)}{34193}$

$= 1.2283$

For the calculated values of $\frac{c_{min}}{c_{max}}$ = 0.3686 and NTU = 1.2283 from the fig , we get

Effectiveness ϵ =0.64

$$\epsilon = \frac{c_h\,(t_{hi}-t_{tho})}{c_{min}(t_{hi}-t_{ci})} = \frac{c_c\,(t_{co}-t_{ci})}{c_{min}(t_{hi}-t_{ci})}$$

$$= \frac{2206(120-tho)}{34193(120-30)} = \frac{4178(tco-30)}{34193(120-30)}$$

Solving the equation,

Then t_{co}= 51 C^o= 324 K

t_{ho}=62 C^o = 335 K

Shell-side Area:

$$A_s = \frac{D_s\,CB}{P_t}$$

$$A_s= \frac{(0.38735\ X\ 0.00635\ X\ 0.3048)}{0.031375}$$

A_s= 0.02361 m^2

Mass velocity, $G_s = \frac{\dot{m}_s}{A_s} = \frac{15.5}{0.02361}$

G_s= 656.507 kg/m^2.s

$$D_e = \frac{4(P_t^2-\frac{\pi D_O^2}{4})}{\pi\,D_O}$$

Equivalent Diameter,$D_e = \frac{4(0.03175^2-\pi\,.\,0.0254^2/4}{\pi\,.\,0.0254}$

D_e = 0.02513 m

$Re_s = \frac{D_e\,G_s}{\mu} = \frac{0.02513\ X\ 656.507}{0.0000119} = 1379433$

Therefore, the flow of the fluid on shell side is turbulent. By using McAdam'scorrelation

$$Nu = 0.36\left[\frac{D_e\,G_s}{\mu_b}\right]^{0.55}\left[\frac{C_p\,\mu_b}{k}\right]^{0.33}\left[\frac{\mu_b}{\mu_w}\right]^{0.14}$$

$$Nu = 0.36\left[\frac{0.02513\ X\ 656.507}{0.0000119}\right]^{0.55}\left[\frac{2135\ X\ 0.000119}{0.02373}\right]^{0.33}\left[\frac{0.0000119}{0.0006523}\right]^{0.14}$$

Nu_s = 502

The shell-side heat transfer coefficient, $h_{o,}$ is then calculated as:

$h_o = \frac{Nu.k}{D_e} = \frac{502 \text{ x } 0.02373}{0.02513} = 474 \text{ W/m}^2\text{.K}$

Tube-side heat transfer coefficient by:

$A_t = \frac{\pi D_i^2}{4} = \frac{\pi \; X \; 0.02291^2}{4} = 0.0004122 \text{ m}^2$

$A_{tp} = \frac{N_t \, A_t}{no.of \; passes} = \frac{81 \; X \; 0.0004122}{1} = 0.0339 \text{ m}^2$

$G_t = \frac{\dot{m}_t}{A_{tp}} = \frac{22.2}{0.03339} = 654 \text{ kg/m}^2\text{.s}$

$u_t = \frac{G_t}{\rho} = \frac{654}{995} = 0.658 \text{ m/s}$

$Re_t = \frac{u_t \rho d_i}{\mu} = \frac{0.658 \; X \; 0.02291}{0.00000657} = 22951.3$

Therefore, the flow of the fluid on tube side is turbulent. By using the Petukhov and Kirillov correlation:

$$Nu_t = \frac{\left(\frac{f}{2}\right) Re Pr}{1.07 + 12.7\left(\frac{f}{2}\right)^{1/2} \left(Pr^{2/3} - 1\right)}$$

Where f = $(1.58 \ln Re - 3.28)^{-2}$

$f = (1.58 \; X \; ln(22951.22) - 3.28)^{-2} = 0.006313$

$$Nu = \frac{\left(\frac{0.006313}{2}\right) X \; 22951.3 \; X \; 4.34}{1.07 + 12.7\left(\frac{0.006313}{2}\right)^{1/2} x \; (4.34^{2/3} - 1)} = 141$$

The tube-side heat transfer coefficient, $h_{i,}$ is then found as:

$h_i = \frac{Nu.k}{d_i} = \frac{141 \text{ X } 0.6280}{0.02291} = 3853 \text{ W/m}^2\text{.K}$

The shell-side pressure drop can be calculated from equations

$\Delta P_s = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \, \emptyset_s} = \frac{161873.17}{0.0171705} = 9427674 \text{pa}$

N_b = L/B = 7 baffles

f = exp(0.576 – 0.19 ln Re_s)

f = exp (0.576 – 0.19 ln 1379433.21) = 0.1212

The tube-side pressure drop can be calculated from equation:

$\Delta P_t = \left(4f \frac{L N_p}{d_i} + 4 N_b\right) \frac{\rho u_m^2}{2}$

$\Delta P_t = \left(\frac{4 \text{ X } 0.0063 \text{X } 2.15 \text{ X } 1}{0.02291} + 4 \text{ X } 7\right) \text{X } 995 \text{ X } \frac{0.0658^2}{2}$

ΔP_t = 6.5 Kpa

Results:

Reynolds number of the shell side (Re_s)	=1379433.2
Nusselt number of the shell side (Nu_s)	=502.12
Heat transfer coefficient of shell side (h_s)	= 474.148 W/m^2.K
Reynolds number of the tube side (Re_t)	=22951
Nusselt number of the tube side (Nu_t)	= 140.56

Heat transfer coefficient of tube side (h_t) = 385 W/m^2.K

Friction factor of tube side (f_t) =0.006313

Friction factor of shell side (f_s) =0.1212

Pressure drop of shell side (ΔP_s) =9427.6 KPa

Pressure drop of tube side (ΔP_t) =6.5 KPa

Effectiveness ($\in$) =0.64

3.2. Flow Diagram of the Design Program:

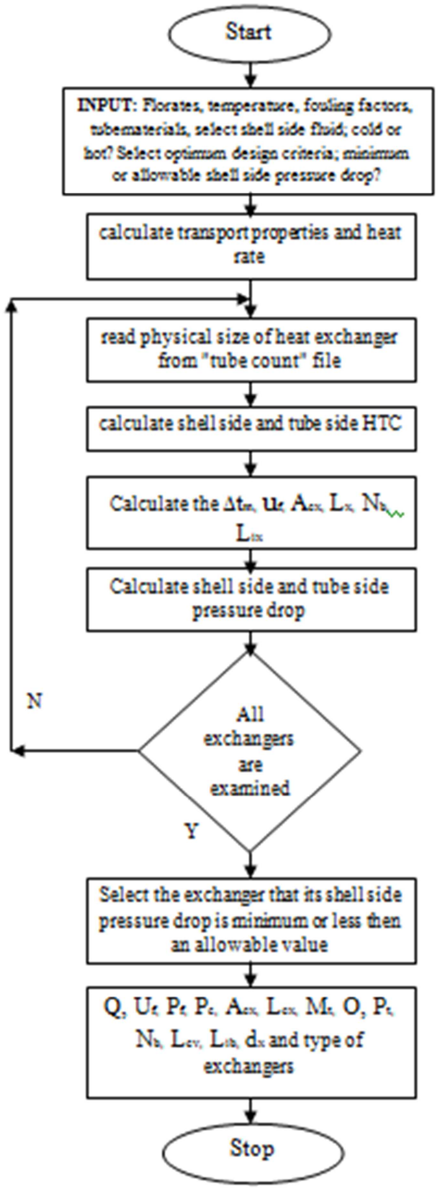

Fig.3.1. Flow Diagram of the Design Program

3.3. C- Program:

The following “C” program is written to evaluate output parameters for three different fluid combinations in general

```
//program to evaluate the Res no & ΔPs in the case of shell and tube heat exchanger
//initial temperatures & fluid properties are to be defined initially
#include<stdio.h>
#include<conio.h>
#include<math.h>
main()
{
int i,nop,nt,ch;
float di,d0,ac,tw,mt,tti,ft,tto,tco,tci,thi,tho;
float cpt,ubt,kt,pt,prt,dt1,dt2,dt3,dt4,dt5,dt6,dp;
float cps,pst,ubs,ks,ps,prs,de,x1,x2,x3,t1,t2,t3,tf,a1,a2,a3,a4,a5,a6,a7,res;
float as,gs,nus,ho,at,atp,gt,ut,ret,nut,e1,e2,e3,e4,f1,hi,m1,m2,m3,m4,u1,kw,uu,dps,dpt;
float uw,c,b,ds,ms,fs,tsi,tso,pmax,k,lmtd,area,len,q,r,p,f,k1,fp,nb;
clrscr();
printf("Enter the Cold Fluid Inlet Temp :tci \n");
scanf("%f", &tci);
//tci=20;tco=25;thi=35;
printf("Enter Cold Fluid Outlet Temp : tco \n");
scanf("%f", &tco);
printf("Enter Hot Fluid Inlet Temp: thi \n");
scanf("%f", &thi);
//printf("Enter Tubeside Specifications\n");
//printf("Enter Outer Dia of Tube: do \n");
//scanf("%f",&d0);
d0=0.0254;
//printf("Enter Inner Dia of Tube:di\n");
//scanf("%f",&di);
di=0.0229108 ;
//printf("Enter Flow Area of Tube-side: ac \n");
//scanf("%f",&ac);
ac=0.00041226;
//printf("Enter Wall Thickness of Tube-side: tw \n");
//scanf("%f",&tw);
tw=0.0012446;
//printf("Enter Mass Tube Side fluid mass flow rate: ms \n");
//scanf("%f", &mt);
  mt=38.89;
printf("Enter the Tube Side Fluid Properties\n");
printf("Enter the specific heat of tubeside fluid:cp\n");
scanf("%f", &cpt);
//cpt=4.179*1000;
printf("Enter Viscosity :ubt \n");
scanf("%f", &ubt);
//ubt=0.00095;
printf("Enter Thermal Conductivity : kt \n");
scanf("%f",&kt);
//kt=0.6065;
printf("Enter Density : pt \n");
scanf("%f",&pt);
//pt=997;
printf("Enter Prandtl Number : prt \n");
scanf("%f",&prt);
```

```
//prt=6.55;
//printf("Enter Shell side specifications");
//printf("Enter Pitch Size :pst \n");
//scanf("%f", &pst);
pst=0.03175;
//printf("Enter Clearance: c \n");
//scanf("%f", &c);
c=0.00635;
//printf("Enter Baffle spacing: b \n");
//scanf("%f", &b);
b=0.3048;
//printf("Enter Shell side Diameter:ds ");
//scanf("%f", &ds);
ds=0.38735;
//printf("Enter Mass Shell Side fluid mass flow rate: ms \n");
//scanf("%f", &ms);
ms=22.22;
printf("Enter the Shell Side Fluid Properties\n");
printf("Enter Spceific Heat : cps \n");
scanf("%f", &cps);
//cps=4.1785*1000;
printf("Enter Viscosity : ubs\n");
scanf("%f", &ubs);
//ubs=0.000797;
printf("Enter Thermal Conductivity : ks \n");
scanf("%f", &ks);
//ks=0.614;
printf("Enter Density : ps \n");
scanf("%f", &ps);
//ps=995.7;
printf("Enter Prandtl Number: prs \n");
scanf("%f", &prs);
//prs=5.43;
printf("Calculations of LMTD and Mass Flow Rates");
q =mt*cpt*(tco-tci);
tho=thi - (q/(ms*cps));
printf("Heat Transfered = %f\n",q);
printf("Hot Fluid inlet Temp is %f\n",thi);
printf("Hot Fluid Outlet Temp is %f\n",tho);
printf("Cold Fluid inlet Temp is %f\n",tci);
printf("Cold Fluid Outlet Temp is %f\n",tco);
dt1=dt2=dt3=dt4=dt5=dt6=0.000;
dt1=thi-tho;
printf("dt1 is %f\n",dt1);
dt2=tho-tci;
printf("dt2  is %f\n",dt2);
dt3=thi-tco;
printf("dt3 is %f\n",dt3);
dt4=tco-tci;
printf("dt4 %f\n",dt4);
dt5=thi-tci;
printf("Enter the type of Flow: 1. Parallel Flow 2. Counter Flow");
```

```
scanf("%d",&ch);
switch(ch)
{
case 2:
lmtd =((thi-tco)-(tho-tci))/(log((thi-tco)/(tho-tci)));
printf("LMTD is %f\n",lmtd);
break;
case 1:
lmtd =((thi-tci)-(tho-tco))/(log((thi-tci)/(tho-tco)));
printf("LMTD is %f\n",lmtd);
break;
}
r=dt1/dt4;
printf("r value %f\n",r);
p=dt4/dt5;
printf("p value %f\n",p);
a1=sqrt((r*r)+1);
printf("a1 value %f\n",a1);
a2=(1-p);
printf("a2 value %f\n",a2);
a3=(1-(p*r));
printf("a3 value %f\n",a3);
a4=p*((r+1)-a1);
printf("a4 value %f\n",a4);
a5=p*((r+1)+a1);
printf("a5 value %f\n",a5);
a6=log((2-a4)/(2-a5));
printf("a6 value %f\n",a6);
a7=log(a2/a3);
printf("a7 value %f\n",a7);
f=(a1*a7)/((r-1)*a6);
printf("Correction Factor %f\n", f);
printf("Calculation of Shell Side Heat Transfer Co-efficient\n");
as=(ds*c*b)/pst;
printf("as%f\n",as);
gs=ms/as;
printf("gs %f\n",gs);
de=((4*pst*pst)-(3.14*d0*d0))/(3.14*d0);
printf("de %f\n",de);
x1=(de*gs)/ubs;
printf("x1 %f\n",x1);
x2=(cps*ubs)/ks;
printf("x2 %f\n",x2);
//scanf("%f", &uw);
uw=0.00086;
x3=(ubs/uw);
printf("x3 %f\n",x3);
t1=pow(x1,0.55);
printf("t1 %f\n",t1);
t2=pow(x2,0.33);
printf("t2 %f\n",t2);
t3=pow(x3,0.14);
```

```
printf("t3 %f\n",t3);
nus=0.36*t1*t2*t3;
printf("nus  %f\n",nus);
ho=(nus*ks)/de;
printf("ho %f\n",ho);
printf("Calculation of Tube side Heat Transfer Co-efficient");
at=(3.14/4)*di*di;
//scanf("%d", &nop);
//scanf("%d", &nt);
nop=1;nt=81;
atp=(nt*at)/nop;
printf("atp %f\n",atp);
gt=mt/atp;
ut=gt/pt;
ret=(gt*di)/ubt;
printf("ret %f\n",ret);
f1=((1.58*log(ret))-3.28);
printf("f1 %f\n",f1);
tf=pow(f1,-2);
printf("f %f\n",tf);
e4=tf/2;
e1=sqrt(e4);
e2=pow(prt,0.67);
e3=12.7*e1*(e2-1);
nut=(e4*ret*prt)/(1.07+e3);
printf("nut %f\n",nut);
hi=(nut*kt)/di;
printf("hi %f\n",hi);
printf("Calculation of Overall Heat Transfer Coefficient");
m1=d0/(di*hi);
m2=log(d0/di);
m3=1/ho;
//scanf("%f",&kw);
kw=54;
m4=(d0*m2)/(2*kw) ;
u1=m1+m4+m3;
printf("1 %f\n",u1);
uu=1/u1;
printf("uu %f\n",uu);
area= (q)/(uu*f*lmtd);
len=area/(nt*3.14*d0);
nb=len/b;
k1=log(x1);
fp=exp(0.576 - 0.19*k1);
printf("fpis %f\n",fp);
printf("Heat Transfered = %f\n",q);
len=area/(nt*3.14*d0);
        printf("len %f\n",len);
nb=len/b;
printf("nb %f\n",nb);
printf("res = %f\n",x1);
k1=log(x1);
```

```
fp=exp(0.576 - 0.19*k1);
printf("fpis %f\n",fp);
dp=(fp*gs*gs*(nb+1)*ds)/(2*ps*de*x3);

printf("pr drop = %f\n",dp);
printf("------------------------------------------------------------");
printf("\n");
printf("Heat Transfered = %f\n",q);
printf("Hot Fluid inlet Temp is %f\n",thi);
printf("Hot Fluid Outlet Temp is %f\n",tho);
printf("Cold Fluid inlet Temp is %f\n",tci);
printf("Cold Fluid Outlet Temp is %f\n",tco);
printf("LMTD is %f\n",lmtd);
printf("Correction Factor-Shell Side %f\n", f);
printf("Heat Transfer Coefficient -Shell Side %f\n",ho);
printf("Renold number-Shell Side %f\n",x1) ;
printf("Nusselt number-Shell Side %f\n",nus) ;
printf("Correction Factor-Tube Side %f\n",tf);
printf("Heat Transfer Coefficient -Tube Side %f\n",hi);
printf("Renold number-Tube Side %f\n",ret) ;
printf("Nusselt number-Tube Side %f\n",nut) ;
printf("Overall Heat Transfer Co-efficient = %f\n",uu);
printf("sheell side friction factor = %f\n",fp);
printf("tube side friction factor = %f\n",tf);
printf("Area = %f\n",area);
printf("Length = %f\n",len);
printf("Number of baffles = %f\n",nb);
printf("pressure drop = %f\n",dp);
printf("------------------------------------------------------------");
printf("\n");
}
```

4. RESULT AND DISCUSSION:

The analysis is compared with the equations mentioned and drawn in Figs (4.1-4.3)

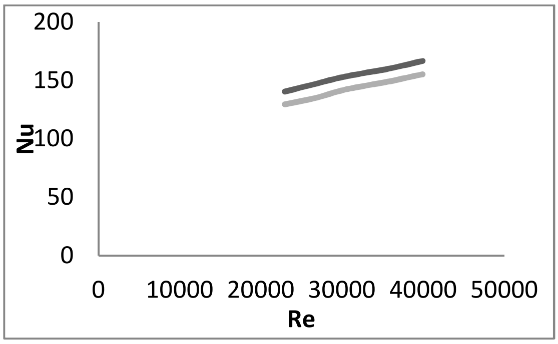

Fig. 4.1 Variation of Nu vs. Re

Fig.4.1 shows the comparison between Nusselt number on tube side using Petukhov-kirrilov correlation in Kern method by well known Dittus-Boelter equation. As per the above plot it was found that there is close agreement between two equations with a deviation of around 10 percent. Hence this equation in kern-method is used further to calculate the Nusselt number on tube side.

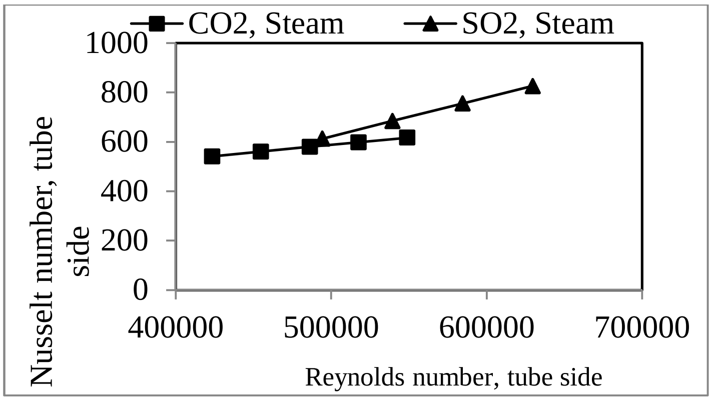

Fig. 4.2 Variation of Nu_t Vs Re_t

The above graph (Fig.4.2.) shows the variation between the Reynolds number on tube side and Nusselt number on tube side. It was found that as Reynolds number increases the Nusselt number increases. It is found that there is a significant increase of Nusselt number for SO_2 and Steam combination when compared to CO_2 and Steam combination for the given inlet temperatures.

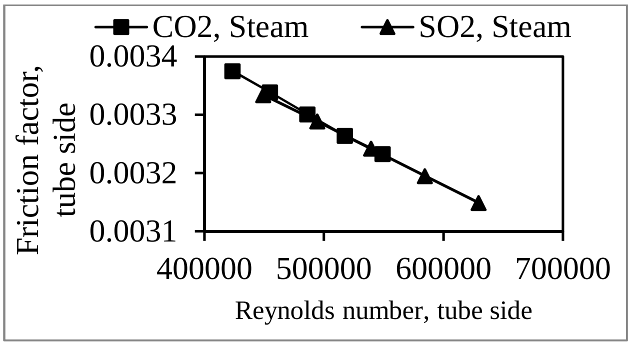

Fig.4.3. Variations of f and Re

The graph (Fig. 4.3.) shows the variation between the Friction factor on tube side and Reynolds number on tube side. The graph is drawn with combinations of fluids which CO_2 - steam and SO_2 - steam. From the above it was found that the Reynolds number of fluid on the tube side increases the friction factor decreases. However it is found that more or less for the above two combinations of fluids the friction factor remains unaffected.

5. CONCLUSIONS:

The following conclusions are arrived from the transfer analysis of shell and tube heat exchanger for three different fluid combinations (water-steam, CO_2-steam and SO_2-steam) using kern's method.

1. Taking the input parameters the values for Nusselt number, Reynolds number, heat transfer coefficient, and pressure drop and friction factor are determined.

2. Validation for Nusselt number on tube side for water using (petukhov equation) is compared with well known Dittus-Boelter equation with a deviation of 10 percent.

3. From the data arrived and drawn it is found that as Reynolds number increases Nusselt number increases and friction factor decreases both tube and shell side fluids.

Vol.2 Issue.1, January 2014. Pgs: 11-26

4. Taking steam on shell side and CO_2 and SO_2 on tube side. It was found that nusselt numbers shows steep increases for SO_2 steam combination than CO_2 steam combination. However taking above two combinations the friction factor almost remains same.

5. Taking Area of heat exchanger fixed, the length of the heat exchanger is found to be 2.16 m and the maximum effectiveness of shell and tube heat exchanger, from the calculations is found to be 0.65.

6. FUTURE SCOPE OF WORK

1. This work can be extended for different bundle tube configurations such as triangular pitch, and for different tube layouts for heat transfer analysis on shell and tube heat exchanger for particular applications.
2. Apart from Kern method considered in this work, it can be carried out by different methods such as Bell-Delaware method, Taborek method in designing/analyzing the shell and tube heat exchanger.

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