EMS623U

Heat Exchange and Waste Minimisation

Individual design project: Heat Exchanger Design

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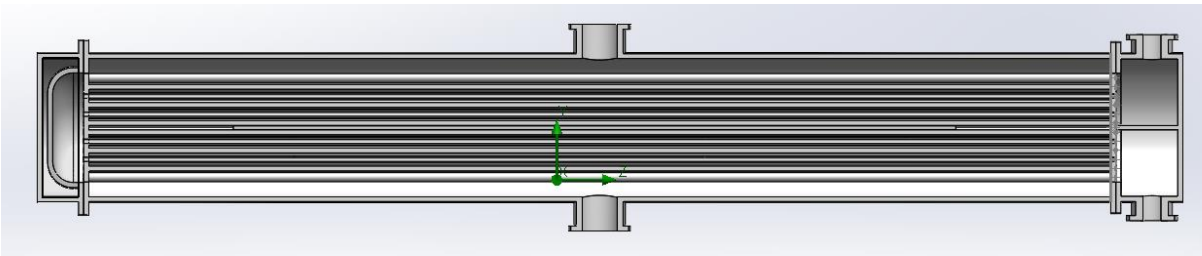

Introduction

It has been shown in studies [1] that temperature changes in an office environment can cause a negative effect on the productivity up to 9%. With that being stated in this study a high efficiency and appropriately sized shell and tube heat exchanger will be designed.

There are two shell and tube exchangers in the loop first one being, the one that take in the outside air at a temperature of 0 °C and heats it to 20 °C is using water on the tube side. The second one is the one that will be designed. This one recover heat from the air in the office at 20 ºC before it is ejected outside at the same intake rate on 90 m^3/min this energy is transferred using the Shell and tube heat exchanger to the tube side water pre heating it. This process called a run around system which provides 10kW of energy in this case, which is the heat transfer rate for this second heat exchanger. This run around system also reduces heating costs.

In the design for this shell and tube heat exchanger there are a few variables which will need to be assumed to a reasonable degree, namely flowrate, intake and exit temperatures of the tube side water. A suitable geometry complying with the TEMA standards will be achieved in this study. Lastly heat transfer coefficients for the shell side and tube side must be estimated.

Methodology

To solve the proposed problem this study will use a multi-step process inspired by one found in [2, pg. 865].

Step 1 Specification and Tube side assumptions:

Identify the specification of the Heat exchanger in this case, on the shell side air is moving at a volumetric flowrate of 1.5 m^3/s, intake temperature of 20ºC, Specific heat capacity of 1005 J/Kg ^{0}C and a density of 1.225 Kg/m^3, Lastly the energy output of the heat exchanger is 10 kW

Using the formula for Duty below, the outgoing temperature for air is found:

$Q=\dot{m}_{hot}C_{p,hot}(T_{in,hot}-T_{out,hot})=\dot{m}_{cold}C_{p,cold}(T_{out,cold}-T_{in,cold})$

Using the same formula knowing that the specific heat capacity of the water tube side is 4200 J/Kg ^{0}C and setting the intake temperature to 5 ^{0}C and the out-going temperature to 15 ^{0}C. A mass flowrate of 0.238 Kg/s is found.

Step 2 Assumption for Overall heat transfer Coefficient:

A reasonable assumption for the Overall heat transfer coefficient of 200 W/m^2 ^{0}C as its on the higher end of the 20 - 300 range taken from [2, pg. 826]

Step 3a initial configuration:

A configurating for the Heat exchanger is chosen following the TEMA guidelines and application. For the first iteration is a 1 shell 2 pass Tema G type heat exchanger

3b Calculating ΔT_{lm}, P, R and F:

Calculating ΔT_{lm} , P and R using their respective formula below which in this case yields 7.05 yielding 0.36 and 1.85 respectively for this scenario.

$$\Delta T_{\mathrm{lm}} = \frac{(T_1 - t_1) - (T_2 - t_2)}{Ln(\frac{(T_1 - t_1)}{(T_2 - t_2)})} \qquad P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1}$$

With the parameters above A correction factors can be determined this can be done manually using the tables [2] or using an online tool [3] which was used in this study. For this case the correction factor is 0.9548.

Step 4 Calculating area Required:

Use formula to the right using values from previous steps. In this case Area required is 7.4 m^2.

$$A_{req} = \frac{Q}{(F\ \Delta T_{lm}\ U)}$$

Step 5 Tube side layout and tube size:

Decision for length, inner and outer tube diameter , material ,pitch and head type must be decided. In this initial design a length of 3.5 m, inner diameter of 14.8mm and outer of 19mm with a pitch of 1.25 equalling 23.75mm, aluminium and a floating head type was chosen.

Step 6 Number of tubes and tube side velocity:

The cross-sectional area of and individual tubes is calculated using SA = π r l and then divided by the A_{req} found in step 4 in this case a minimum of 35.5 was required so 36 tubes will be used.

In this step we also calculate the tube side velocity by finding the tubes cross sectional area and then multiplying it by the number of tubes in one pass this is then divided by the volumetric flow rate of the fluid in this case 0.8m/s.

Step 7 Bundle diameter:

This is calculated by using the formula to the right from [2, pg. 837].

In this case 181mm given a floating head is being used 56mm of clearance is added giving 237mm.

$$D_b = d_0 (\frac{N_t}{K_1})^{\frac{1}{n_l}}$$

Step 8 Tube side coefficient:

Since water is being used tube side , This formula on the right [2 , pg. 850] can be used to calculate the heat transfer coefficient tube side. In this example h_i= 456 W/ $m^2\,{}^0C$

$$h_i = \frac{4200(1.35 + 0.02t){u_t}^{0.8}}{{d_i}^{0.2}}$$

Step 9 Shell side coefficient:

Firstly, Baffle spacing is chosen which is between 0.1-0.5 as well as the baffle cut between 15 and 45. In this instance 0.2 and 25 respectively. Cross flow is then Calculated using the A_s formula. Next the shell-side equivalent diameter (hydraulic diameter) de is found using its respective formula. Lastly Reynolds number is also calculated. These values are then entered in the h_s formula giving us the Shell side heat transfer coefficient h_s= 891 W/$m^2\,{}^0C$ [2,pg. 857]

$$A_s = \frac{(p_t - d_o)D_S l_B}{p_t} \qquad d_e = \frac{1.1}{d_o}({p_t}^2 - 0.917{d_o}^2) \qquad Re = \frac{u_s d_e \rho}{\mu} \qquad h_s = \frac{k_f}{d_e} Re\ Pr^{0.33}$$

Step 10 Overall heat transfer coefficient

The formula below [2, pg. 824] is used to calculate the Overall heat transfer coefficient. OHTC = 227 W/$m^2\,{}^0C$

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o \ln(\frac{d_o}{d_i})}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_i}$$

Step 11 Pressure drop

The formulas below [2, pg. 851, pg. 857] are used to calculate the pressure drops on the tube and shell side respectively.

$$\Delta P_t = 8J_f(\frac{L'}{d_i})\frac{\rho {u_t}^2}{2} \qquad \Delta P_s = 8J_f(\frac{D_s}{d_e})(\frac{L}{l_B})\frac{\rho {u_s}^2}{2}$$

Results:

Using the formulas provided in the methodology a python code was written in such a way that values can be input and multiple configurations simulated in a short amount of time which can be found in the appendix of this study (Fig. 4-6). The assumption made in the methodology section of this study will carry over to the configuration tested and only the independent variable highlighted have been changed.

Results				
Test No.	1	2	3	4
Configuration	G	E	G	Tube in tube
Number of passes	2	2	2	1
Tube length (m)	3.5	3.5	2	479.638023
Tube outer diameter (mm)	19	19	16	19
Tube inner diameter (mm)	14.8	14.8	10.4	14.8
Area required	7.43	9.19	7.43	7.09864274
Number of tubes	36	48	96	1
tube side velocity (m/s)	0.076	0.057	0.072	1.39
Shell diameter (mm)	237	262	272	28.5
Tube side coeffient (W/ m^2 ^oC)	456	362	464	200
Shell side heat transfer coeffient (W/ m^2 ^oC)	891	770	872	200
OHTC (W/ m^2 ^oC)	227	188	201	98
Pressure drop Tube (Pa)	115	36	23	177000
Pressure drop Shell (KPa)	70914	34083	32001	117080

Fig. 1 Results table

The table (Fig. 1) above shows the different configuration that were simulated using the code. The Independent variable is the row in the light green and dependent variable are in light orange except for the tube and tube which maintain the same standard except for number of tubes and tube length.

The above scenarios have been ran using aluminium as the tube material as in an office environment the incoming water can be monitored to ensure its not in contact with other material to eliminate the risk of aluminium galvanic corrosion. This was done as a cost reducing measure as copper piping would be magnitude of time more expensive while only having a modest more conductivity. When running test outside of the above presented no major difference in OHTC was found when keeping other parameters identical.

Carbon steel shell at 9.3 mm shell thickness was also chosen as it amongst the cheapest material that can be used which ensuring that it can withstand the pressure inside the shell excluding the tube in tube heat exchanger.

The first system is based exactly on the sample calculation in the methodology and will be the base configuration. Given that the air is being discharged at a high-rate pressure drop on the shell side is sure to be high. The G type configuration was chosen as the base configuration as it is used when minimising shell side pressure drop is a priority [5]. It also yielded the highest correction factor for the set temperature which corresponds to a lower A_{req} amongst all the 2 pass configurations. This configuration yielded a higher than predicted Overall heat transfer coefficient. However, it came at the cost of having the highest pressure drop shell side at 70 MPa. This is due to requiring the least number of tubes.

Following this system an alternative configuration was run which is the most commonly used [5] that being E type. All other parameters stayed the same including tube size and length. This resulted in lower tube side velocity, more tube needed, lower tube side and shell side coefficient and therefore lower Overall heat transfer coefficient.

The third system was a result of a few attempts through trial and error with the code in the appendix (Fig 2-4). This system uses a G type Shell with 2 passes however, the tube length is 2 meter the outer diameter of the tube is 16mm and the tube thickness is 2.8mm. This configuration requires more tubes than the first system which increases the shell diameter which consecutively yield the lowest shell pressure drop at 32MPa. This is true while also having a higher tube side heat transfer coefficient and matching and slightly exceeding the estimated overall heat transfer coefficient at 201 W/ m^2 °C. This Shell and tube exchanger is the most balanced out of the others presented.

Looking at the tube in tube design, we can observe that given all the heat need to be dissipated by one singular tube the system justly becomes extremely long. Consequently, shell for this heat exchanger is quite small at 28mm, the major problems with this design continue as the overall heat transfer coefficient is 98 W/ m^2 °C which is less than 50% the originally assumed Heat transfer coefficient. Finally, the pressure drop is also astronomical both tube side and shell side at 177000 Pa tube side and 117080 kPa shell side.

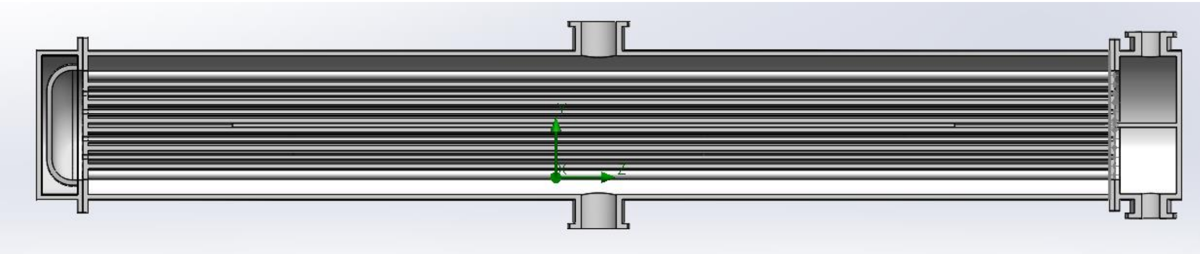

Fig. 2 System 3 cross section

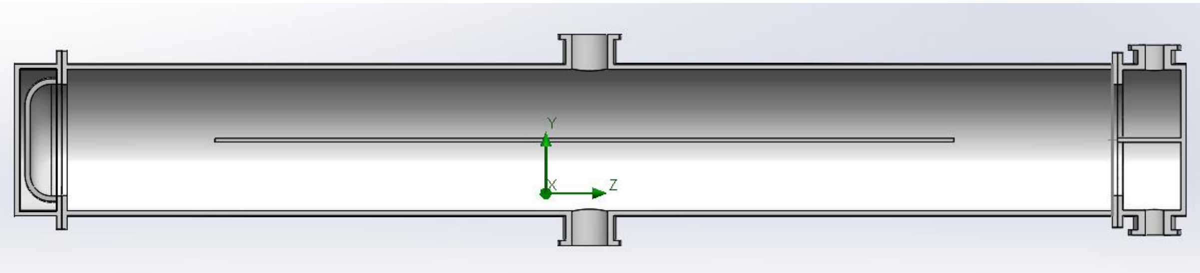

Fig. 3 System 3 cross section without tubes

Conclusions:

This study had the aim to design a shell and tube heat exchanger to recover waste heat from the office air as part of a run around system for an office environment which is used to reduce the energy consumption when heating the office in the colder months. This aim was successfully archived by analysing the given scenario and a cost efficiency and up to standard system has been proposed which recovers 10kW of energy that would have otherwise been wasted to the environment. This proposed design adheres closely with the TEMA standards. This was possible by making reasonable assumptions based on current design that are being used in the real world and prioritising the overall heat transfer coefficient.

The findings from the several iterations simulated systems highlights the relationship between heat transfer coefficients and pressure drop. As system 1 a G-type shell with 2 passes produced the highest Overall heat transfer at 227 W / m^2 °C however it also had the highest pressure drop on the shell side at 70 MPa. This suggests that optimalization is required to balance the two variables.

The E-type shell was the second system tested as it is the most used in the industry which resulted in the lowest Overall heat transfer at 188 W / m^2 °C which is approximately 17% less than system 1 while requiring 33% more tubes.

The tube in tube was practically unusable for the given scenario as it had overall heat transfer at 98 W / m^2 °C while requiring a length of 480m which is unfeasible as nearly all offices do not have a HVAC room that is half a kilo meter in length.

The most balanced and therefore the proposed design comes in the form of system 3 which is a G type shell and tube with 2 passes a smaller tube length of 2 meters and tube inner and outer diameters of 10.4mm and 16mm respectively. This system has the lowest shell side pressure drop at 32 MPa and slightly higher overall heat transfer at 201 W / m^2 °C than required.

To reduce the cost of the system which ensuring adequate heat transfer material selection played a key role and therefore aluminium tubes and carbon steel will be used for shell, with the only drawback being that the quality of the water must be controlled to avoid galvanic corrosion inside the tubes.

To improve this proposed design simulation should be ran using more tube passes in the shell. This would decrease pressure drop while increasing surface area and could lead to a higher overall heat transfer for the system. Testing with larger diameter thicker walled tubing could also lead to an increase in overall heat transfer due to the enhanced thermal conductivity. Lastly changing materials for the tubing to perhaps copper would be more costly upfront but reduce the maintenance from galvanic corrosion and given it has a higher thermal conductivity could lead to a higher overall heat transfer coefficient

References:

[1] Seppanen, O., Fisk, W., & Lei, Q. (2006). Effect of temperature on task performance in office environment. Lawrence Berkeley National Laboratory. Retrieved from https://escholarship.org/uc/item/45g4n3rv

[2] Towler, Gavin Sinnott, Ray. (2022). <i>Chemical Engineering Design - Principles, Practice and Economics of Plant and Process Design (3rd Edition) - 19.1 Introduction.</i> (pp. 823). Elsevier. Retrieved from
https://app.knovel.com/hotlink/pdf/id:kt012WNNH3/chemical-engineering/heat-trans-introduction

[3] LMTD Correction Factor Charts (no date) CheCalc ‐ Logarithmic Mean Temperature Difference LMTD Correction Factor Charts. Available at: https://checalc.com/solved/LMTD_Chart.html (Accessed: 10 November 2024).

[4] G. van Rossum, Python tutorial, Technical Report CS-R9526, Centrum voor Wiskunde en Informatica (CWI), Amsterdam, May 1995.

[5] Enerquip (2023) 7 shell configurations to consider when designing a shell and tube heat exchanger, Enerquip Thermal Solutions. Available at: https://www.enerquip.com/7-shell-configurations-what-you-need-to-know-when-designing-a-shell-and-tube-heat-exchanger/ (Accessed: 25 October 2024).

Appendix – Sample calculation

623U.py - H:\Y3\623U Heat\623U.py (3.13.0)

File Edit Format Run Options Window Help

```python
import numpy as np
from math import e
pi = np.pi
ln = np.log  # assign the numpy log function to a new function called ln
ln(e)

print ('Step 1: Specification and Tube side assumptions')
# Q = m_hot * C_hot (T_hot_in - T_hot_out) = m_cold * C_cold (T_cold_in - T_cold_out)
Q = 10000 # W
m_hot = 1.5 * 1.225 # kg/s
C_hot = 1005 # J/kg C
T_hot_in = 20
T_hot_out = 0
m_cold = 0 # kg/s
C_cold = 4200 # J/kg C
T_cold_in = 5
T_cold_out = 15
print ('Q: ', Q,' \nm_hot: ', m_hot, ' \nC_hot: ', C_hot)
print ('T_hot_in: ', T_hot_in,' \nT_hot_out: ', T_hot_out, ' \nm_cold: ', m_cold)
print ('C_cold: ', C_cold,' \nT_cold_in: ', T_cold_in, ' \nT_cold_out: ', T_cold_out ,'\n')


T_hot_out = T_hot_in - (Q / (m_hot * C_hot))
print ('T_hot_out: ' , T_hot_out)

m_cold = Q / (C_cold * (T_cold_out - T_cold_in))
print ('m_cold (kg/s): ' , m_cold)
m_cold_vol = (m_cold / 1000) #m^3/s
print ('m_cold (m^3/s): ' , m_cold_vol)

T1 = T_hot_in
T2 = T_hot_out
t1 = T_cold_in
t2 = T_cold_out

diff_temp = (T1-t2)/ (T2-t1)
Delta_T_lm = ((T1-t2) - (T2-t1)) / ln(diff_temp)
print ('Delta_T_lm: ' , Delta_T_lm)

P = (T2 - T1) / (t1 - T1)
R = (t1 - t2) / (T2 - T1)

print ('P: ' , P,' R: ' , R)

print ('G type 2 pass F =     0.9546') #15 0.9546 10 0.9881
F = 0.9546#float( input ('input F: '))
Delta_T_lm_Corrected = Delta_T_lm * F
print ('Delta_T_lm_Corrected: ' , Delta_T_lm_Corrected)


print ('\nStep 2: Assumption for Overall heat transfer Coefficient')
print ('U_assumed (20-300) =     200')
U_assumed = 200#float( input ('input U_assumed: '))
A_req = Q / (Delta_T_lm_Corrected * U_assumed)
print ('A_req: ' , A_req,' m^2')


print ('\nStep 4: layout and tubes')

print ('d_o (mm)=     19')
d_o = 19#float( input ('input d_o (mm): '))
d_o_m = d_o / 1000
print ('tube thickness (mm)=     2.1')
d_t = 2.1#float( input ('input tube thickness (mm): '))
d_i = d_o - (d_t * 2)
print ('tube inner diameter (mm): ',d_i)
d_i_m = d_i / 1000

print ('tube lenght (m)=     3.5')
t_l = 3.5#float( input ('tube lenght (m): '))
t_l_mm = t_l * 1000

pitch = d_o * 1.25
pitch_m = pitch / 1000

n_pass = 2

t_sa = pi * d_o_m * t_l
print ('single tube surface area(m^2): ' , t_sa)

n_tubes = A_req / t_sa
print ('Number of tube required: ' , n_tubes)
n_tubes = float( input ('Number of tube required (muliple of 6 recomended): '))

t_crossarea = (pi/4)* (d_i_m ** 2)
print ('tube cross area (m^2): ' , t_crossarea)

A_per_pass = (n_tubes / n_pass) * t_crossarea
print ('Area per pass (m^2): ' , A_per_pass)

u_t = m_cold_vol / A_per_pass
print ('fluid velocity tube side (m/s): ' , u_t)
```

Fig. 4 IDLE script in python [4] part a

```python
print ('\nStep 7: bundle and shell diam')
K_1 = 0.249
n_1 = 2.207

D_d = d_o * ((n_tubes / K_1) ** (1/n_1))
print ('Shell diameter without clearence (mm): ' , D_d)
D_dt = D_d + 56
print ('Shell diameter with clearence (Split ring floating head)(mm): ' , D_dt)
D_dt_m = D_dt / 1000


print ('\nStep 8: tube side coeffient')
h_i = (4200 * (1.35 + (0.02 * t1)) * (u_t ** 0.8))/ (d_i ** 0.2)
print ('tube side heat transfer coeffient (W/ m^2 C): ' , h_i)


print ('\nStep 9: shell side heat transfer coeffient')
print ('baffle spacing as percentage of shell diam (0.1-0.8)=    0.2')
b_s_p = 0.2#float( input ('input baffle spacing as percentage of shell diam (0.1-0.8)= '))
b_s = D_dt * b_s_p
A_s = ((pitch - d_o) / pitch) * D_dt * b_s
A_s_m = A_s / 1000000
print ('Area of shell(mm^2): ' , A_s,' Area of shell(m^2): ' , A_s_m)

d_e = (1.1 / d_o) * ((pitch ** 2) - (0.917 * (d_o ** 2)))
d_e_m = d_e / 1000
print ('diameter e (mm): ' , d_e,' diameter e (m): ' , d_e_m)

m_hot_vol = 1.5
u_s = m_hot_vol / A_s_m
print ('Air velocity shell side (m/s): ' , u_s)

Re_s = (1.225 * u_s * d_e_m ) / ((1.79 * (10 ** -5)))
print ('Re shell side (Dimentionless): ' , Re_s)

Pr_s = 0.707
print ('Prandtl number (Dimentionless): ' , Pr_s)

print ('j_h for 25% cut : 1.9 * (10 ** -3)')
j_h = 1.9 * (10 ** -3)#eval( input ('j_h (from table): '))

k = 0.0257 #Thermal Conductivity W/(m·K
h_s = (k / d_e_m) * (j_h) * Re_s * (Pr_s ** 0.33)
print ('shell side heat transfer coeffient (W/ m^2 C): ' , h_s)


print ('\nStep 10: OHTC')
f_c_w = 0.0002 #W/ m^2 C
f_c_a = 0.0002 #W/ m^2 C
print ('thermal conductivity of aluminum (W/m^2 C) : 203')
k_w = 203#eval( input ('thermal conductivity of the tube wall material: '))

u_o = ((((1/h_i) + f_c_w) * (d_o/d_i)) + ((d_i_m * (ln(d_o/d_i))/ (2 * k_w)) + ((1 / h_s) + f_c_a))) ** -1
print ('OHTC (W/m^2 C) : ', u_o)

print ('\nStep 11: Pressure drop  ')

Re_t = (1000 * u_t * d_i_m ) / ((1.52 * (10 ** -3)))
print ('Re tube side (Dimentionless): ' , Re_t)
print ('j_f_t for Re_t of 1122 : 9 * (10 ** -3)')
j_f_t = 9 * (10 ** -3)#eval( input ('j_f_t (from table): '))
p_d_t = (n_pass * (((8 * (j_f_t * (t_l_mm / d_i))) + 2.5))) * ((1000 * (u_t ** 2))/ 2)
print ('Pressure drop tube side N/m2 (Pa): ' , p_d_t)


print ('j_f_s for Re_t of 811435 25% : 2.5 * (10 ** -2)')
j_f_s = 2.5 * (10 ** -2)#eval( input ('j_f_s (from table): '))
p_d_s = (((8 * (j_f_s * (D_dt_m / d_e_m))) * (t_l_mm/b_s))) * (((1.225 * (u_s ** 2))/ 2))
p_d_s_kpa= p_d_s/1000
print ('Pressure drop shell side N/m2 (Pa): ' , p_d_s ,' (KPa)' ,p_d_s_kpa)
```

Fig. 5 IDLE script in python [4] part b

```
===================== RESTART: D:\Y3\623U Heat\623U test.py ====================
Step 1: Specification and Tube side assumptions
Q:  10000
m_hot:  1.8375000000000001
C_hot:  1005
T_hot_in:  20
T_hot_out:  0
m_cold:  0
C_cold:  4200
T_cold_in:  5
T_cold_out:  15

T_hot_out:  14.584898636071344
m_cold (kg/s):  0.23809523809523808
m_cold (m^3/s):  0.0002380952380952381
Delta_T_lm:  7.045551093968567
P:  0.36100675759524375  R:  1.8466875
G type 2 pass F =    0.9546
input F: 0.9546
Delta_T_lm_Corrected:  6.725683074302394

Step 2: Assumption for Overall heat transfer Coefficient
U_assumed (20-300) =    200
input U_assumed: 200
A_req:  7.434189129583116  m^2

Step 4: layout and tubes
d_o (mm)=    19
input d_o (mm): 16
tube thickness (mm)=    2.1
input tube thickness (mm): 2.8
tube inner diameter (mm):  10.4
tube lenght (m)=    3.5
tube lenght (m): 2
single tube surface area(m^2):  0.10053096491487339
Number of tube required:  73.94924674082424
Number of tube required (muliple of 6 recomended): 78
tube cross area (m^2):  8.4948665353068e-05
Area per pass (m^2):  0.003312997948769652
fluid velocity tube side (m/s):  0.07186700438002369

Step 7: bundle and shell diam
Shell diameter without clearence (mm):  216.281567982594
Shell diameter with clearence (Split ring floating head)(mm):  272.28156798259397

Step 8: tube side coeffient
tube side heat transfer coeffient (W/ m^2 C):  463.91056040844825

Step 9: shell side heat transfer coeffient
baffle spacing as percentage of shell diam (0.1-0.8)=    0.2
Area of shell(mm^2):  2965.490090522398  Area of shell(m^2):  0.002965490090522398
diameter e (mm):  11.360800000000001  diameter e (m):  0.0113608
Air velocity shell side (m/s):  505.8185845213063
Re shell side (Dimentionless):  393266.31979951565
Prandtl number (Dimentionless):  0.707
j_n for 25% cut : 8.5 * (10 ** -4)
j_h (from table):  1.1 * (10 ** -3)
shell side heat transfer coeffient (W/ m^2 C):  872.7946082118075

Step 10: OHTC
thermal conductivity of aluminum (W/m^2 C) : 203
OHTC (W/m^2 C) :  200.7725252190413

Step 11: Pressure drop
Re tube side (Dimentionless):  491.7216089159515
L/Di:  192.3076923076923
j_f_t for Re_t of 1122 : 1.9 * (10 ** -2)
j_f_t (from table): 2.5 * (10 ** -2)
Pressure drop tube side N/m2 (Pa):  211.56087035633217
j_f_s for Re_s of 811435 25% : 2.5 * (10 ** -2)
Re_s:  393266.31979951565
j_f_s (from table): 2.9 * (10 ** -2)
Pressure drop shell side N/m2 (Pa):  32001823.628211297  (KPa) 32001.823628211296
```

Fig. 6 Script ran in python [4]