Assignment 3 Heat transfer

Design of an air-cooled exchanger

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1 Introduction

The organic Rankine cycle is an altered version of the Rankine cycle developed in the 1950s. It uses an organic fluid with a liquid-vapour phase change occurring at a temperature lower than the liquid-vapour phase change of water.

Using a fluid with a lower boiling point enables the recovery of heat from flue gasses (exhaust gasses leaving into the atmosphere through a pipe) at low temperatures. The organic Rankine cycle is often used to increase the efficiency of power plants. Nowadays this cycle is also applied to harvest or recover the heat from geothermal sources, biomass combustion, and industrial waste heat.

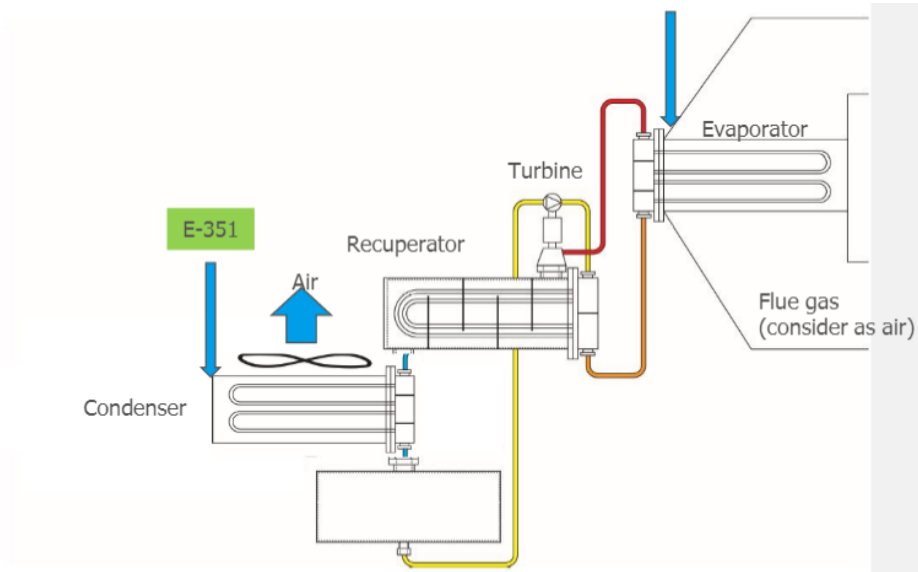

Figure 1: Schematic depiction of an organic Rankine cycle

In this assignment we will design an air-cooled heat exchanger (E-351) for the condenser in the organic Rankine cycle. In this type of heat exchanger a fluid flowing through a bundle of finned tubes is cooled by an air flow flowing over this bundle of tubes. This air flow can be induced by a fan or be forced by a fan. In the case of an induced flow the fan will be installed above the tubes and in the case of a forced flow the fan will be placed below the tubes.

Air coolers are common heat exchangers. Whether to cool with water or air depends on the process temperatures and the climatic conditions. In a moderate climate air cooling will be more economical for process temperatures above 65 degrees °C. In tropical climate zones, meaning high temperatures and high humidity, air cooling will most of the times be more economical than water cooling.

An advantage of air cooling is that it is more environmentally friendly than cooling with water. Air coolers also don't require so much maintenance and the parts are widely available.

There are however also drawbacks to air coolers. First of all the overall heat transfer coefficient in air coolers is low compared with other heat exchangers. This leads to comparatively large equipment. This equipment is also very loud compared to other types of heat exchangers.

The different configurations of air cooled heat exchangers also have certain advantages and disadvantages with respect to each other. Air cooled heat exchangers which have a fan that forces an air flow consume less electrical power than their counterparts with an induced flow. Besides they are easily accessible for maintenance. A downside of a fan forcing an air flow is an uneven flow distribution. This leads to recirculation of air through the bundle of tubes. Air cooled heat exchangers with an induced air flow have a very even flow distribution. Compared to heat exchangers with forced flows however they produce a lot of noise and the height of the installation is greater.

Finally there is the option to use natural convection as a cooling mechanism. Although this saves in electricity considerably the downsides are the installation height of about 20-30 m and the reduced flexibility of the cooling process. In

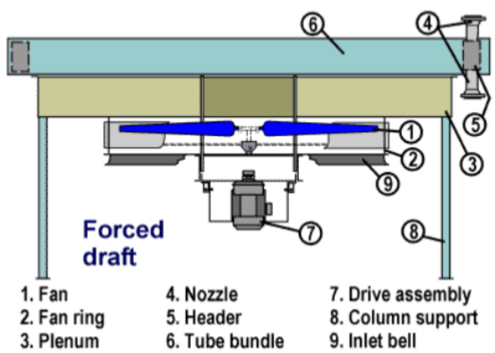

Figure 2: Schematic of air-cooled heat exchanger with forced flow

figure 2 a air-cooled heat exchanger has been schematically depicted concisely. In this case the air flow is forced through the bundle of tubes. In the case an air flow is induced the fan will be on top of the installation.

2 Duty of the air-cooled heat exchanger

In the air-cooled heat exchanger we are designing, an actetone flow in the tube bundle will be cooled by an air flow. In the table below the given properties of both flows have been denoted. These properties have been retrieved with Refprop.

Acetone	Inlet	Outlet
m(kg/s)	4.5	
T(K)	313.15	
P(bar)	0.56582	
Quality	1	0
Cp(kJ/kg-K)	1.4733	2.1822
h(kJ/kg)	483.25	-35.480
$\rho(kg/m^3)$	1.3102	767.66
$\mu(10^{-6}Pa-s)$	8.4186	267.05
Pr	3.9093	0.80653

Table.1: Acetone Mass flux, temperature and enthalpy at in- and outlet of tube

The first step in designing an air-cooled heat exchanger is to determine how much heat should be transferred per second. This characteristic of the heat exchanger referred to as duty can be found with the energy balance below:

$$\dot{Q} = \dot{m}_{ac}(h_{ac,in} - h_{ac,out}) = \dot{m}_{air}(h_{air,out} - h_{air,in}) \tag{1}$$

As all the characteristics of the acetone flow are given the duty can be calculated. We find a duty of **2.33 MW**. Now with the same formula the outlet enthalpy of the air flow can be found. Knowing the duty the outlet temperature of the air flow can also be calculated.

$$T_{air,out} = T_{air,in} + \frac{\dot{Q}}{\dot{m}_{air}c_{p,air}} \tag{2}$$

Air	Inlet	Outlet
m(kg/s)	115	
T(K)	293.15	308.11
P(bar)	1	
Cp(kJ/kg-K)	1.0061	1.0067
h(kJ/kg)	419.41	434.47
$\rho(kg/m^3)$	1.1888	1.1309
$\mu(10^{-6}Pa-s)$	18.205	18.926
Pr	0.70794	0.70606

Table.2: Air Mass flux, temperature and enthalpy at in- and outlet

3 Tube side heat transfer coefficient

The acetone is flowing through the tubes in a cross flow arrangement. The tubes are high finned tubes installed in a staggered arrangement. The tube model we have chosen is produced by the German manufacturer Schmöle and has the code number 9 45 25. In the table below the properties of this tube model have been denoted.

	Base Tube		Finned Tube					
Code nr.	$d_o[mm]$	$t[mm]$	$d_f[mm]$	$h_{fin}[mm]$	$f_t[mm]$	$f_p[mm]$	$A_{ratio}[m^2/m]$	$L[m]$
9 45 25	25.0	1.5	45.0	10.0	0.4	2.82	0.87	12

Table 1: Properties of Schmöle nr. 9 45 25

In order to clarify the staggered arrangement figure 3 has been incorporated.

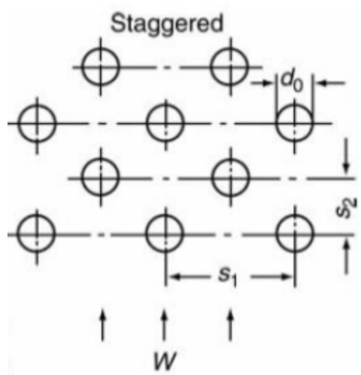

Figure 3: Staggered arrangement

The length of the tubes in our heat exchanger is 12 m. The number of fins n_f on one tube can be calculated by dividing the length L of the tube by the fin pitch.

$$n_f = \frac{L}{f_p} \tag{3}$$

Now the total fin area of one tube A_f can be calculated, here we neglect the thickness of the fins.

$$A_f = 2 * \frac{1}{2}\pi(d_f^2 - d_o^2) * n_f \tag{4}$$

The total surface area of one pipe A can be calculated by adding the area of the spaces between the fins to the fin area.

$$A = A_f + \pi(f_p - f_t)(n_f - 1) \tag{5}$$

The internal area A_i of on tube can be calculated with the next formula:

$$A_i = \pi L(d_o - 2t) \tag{6}$$

The next step is to compute the equivalent outside heat transfer coefficient. This can also be referred to as the virtual heat transfer coefficient. For the **air** heat transfer coefficient, it is **assumed a value of 70 W/m^{2}K**. **Stainless steel** is chosen as the heat exchanger material with thermal conductivity of **16 W/m-K**

$$\alpha_{virtual} = \alpha_{air}[1 - (1-\eta)\frac{A_f}{A}] \tag{7}$$

Finally the overall heat transfer coefficient can be computed. The formula below gives the overall heat transfer coefficient.

$$U_o = \left(\frac{1}{\frac{A}{\alpha_{ac}A_i} + \frac{1}{\alpha_{virtual}} + \frac{1}{F_{air}} + \frac{1}{F_{ac}} + \frac{At}{2A_i\lambda_{wall}}}\right) \tag{8}$$

We estimate an overall heat transfer coefficient of **38.41 W/m^{2}K**.

$$\Delta T_{lm} = \frac{(T_1 - T_{2in}) - (T_1 - T_{2out})}{ln\frac{T_1 - T_{2in}}{T_1 - T_{2out}}} \tag{9}$$

To find the total required heat transfer surface A$_t$ for the duty we can use the following relation with F=0.9:

$$A_t = \frac{\dot{Q}}{FU\Delta T_{lm}} \tag{10}$$

Now, for stagggered arrangement the proper air side heat transfer co-efficient is calculated using the following,

$$Nu = 0.38Re^{0.6}(\frac{A}{A_{to}})^{-0.15}Pr^{1/3} \tag{11}$$

The Re here is calculated for staggered arrangement as following,

$$Re = \frac{\rho w_s d_o}{\mu}, \text{where} \quad w_s = w_{in}\frac{A_{in}}{A_{smallest}} \tag{12}$$

$$\frac{A_{in}}{A_{smallest}} = \frac{s_1 \times f_p}{2(f_p(\sqrt{s_2^2 + (0.5s_1)^2} - d_o) + (\sqrt{s_2^2 + (0.5s_1)^2} - d_f)f_t} \tag{13}$$

For **circular fins,**

$$\frac{A}{A_{to}} = 1 + \frac{2h_{fin}(h_{fin} + d_o + f_t)}{sd_o} \tag{14}$$

Fin efficiency is calculated using the relation,

$$\eta_{fin} = \frac{tanhX}{X} \tag{15}$$

where

$$X = \phi\frac{d_o}{2}\sqrt{\frac{2\alpha}{\lambda f_t}} \tag{16}$$

$$\phi = (\frac{d_f}{d_o} - 1)[1 + 0.35ln\frac{d_f}{d_o}] \tag{17}$$

Now we know the total required heat transfer surface and the total area of one tube we can find the number of tubes in the heat exchanger. Dividing the total surface by the surface of one tube we find the number of tubes. We will have four rows of tubes. Dividing the total number of tubes by the number of rows we find the number of tubes in a row. We find number of tubes in a row. The number of tubes in a row we will call n_{tr}. The overall heat transfer co-efficient after the iterations is found to be **15.34 W/m-K**

We can now calculate the area of the tube bundle.

$$A_{bundle} = s_1 L n_{tr} \tag{18}$$

4 Pressure drop

In this section of the report we will compute the pressure drop over the tubes of the air-cooled heat exchanger and the power of the fan.

First we will calculate the tube-side pressure drop. The formula below gives the pressure drop inside the tubes:

$$\Delta p_{tubes} = \frac{1}{2} n_p (8j \frac{L}{d_i} (\frac{\mu_{ac,in}}{\mu_w})^{-0.14} + 2.5) \rho_{ac,in} v_{in}^2 \tag{19}$$

In this formula n_p is the number of passes in the heat exchanger. In our design the heat exchanger has only one pass. The factor j_f is equal to 3.3 10^{-3}. The pressure drop over the **tubes** is found to be **0.0366 bar**. This is within the limit of 0.3 bar. The following formula holds for the pressure drop of the air flow over the tubes:

$$\Delta p_{air} = \xi n_{mr} \frac{\rho w_e^2}{2} \tag{20}$$

$$\xi = \xi_{lam} + \xi_{turb} F_v \tag{21}$$

$$a = \frac{s_1}{d_o}, b = \frac{s_2}{d_o}, c = \frac{s_d}{d_o} \sqrt{(\frac{a}{2})^2 + (b)^2} \tag{22}$$

$$F_v = 1 - exp(-\frac{Re + 200}{1000}) \tag{23}$$

$$\xi_{lam} = \frac{f_{a,l,v}}{Re} \tag{24}$$

As b is greater than $0.5\sqrt{2a+1}$ the following relation holds for the friction factor $f_{a,l,v}$.

$$f_{a,l,v} = \frac{280\pi[(b^{0.5} - 0.6)^2 + 0.75]}{(4ab - \pi)a^{1.6}} \tag{25}$$

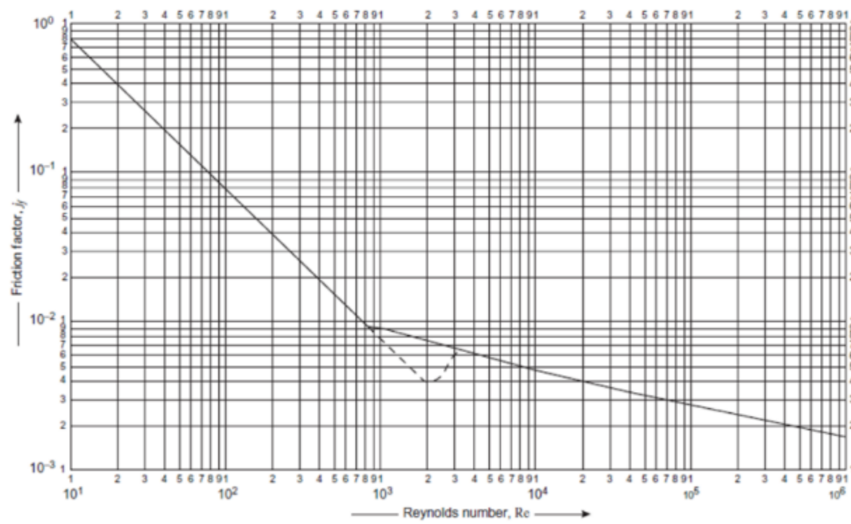

Figure 4: j_f factor

The friction factor $f_{a,t,v}$ is defined as:

$$f_{a,t,v} = 2.5 + \frac{1.2}{(a-0.85)^{1.08}} + 0.4(\frac{b}{a}-1)^3 - 0.01(\frac{a}{b}-1)^3 \tag{26}$$

$$\xi_{turb} = \frac{f_{a,t,v}}{Re^{0.25}} \tag{27}$$

Now, the air side pressure drop has been estimated to be **9.4837 Pa** which is lesss than 150 Pa. Therefore, the design is accepted

5 Fan power

Now we know the pressure drop of the air flow we can find the power of the fan. We assume the efficiency of the fan to be 50% and the efficiency of the motor to be 95%.

$$\dot{W}_{fan} = \frac{w_{approach}(s_1 n_t L)\Delta p}{\eta_{fan}\eta_{motor}} \tag{28}$$

In this formula n_t is the number of tubes in a row. The power of the fan should be equal to **2.67 kW**

6 Cost estimation

When designing a system of heat exchangers the following formula can be used to estimate the cost a single heat exchanger(RobinSmith 2005):

$$Cost = Cb * (A/Ab)^M * 1.29 * 0.82 \tag{29}$$

A is the heat exchange area of a heat exchanger. As the chosen type of heat exchanger is the Air cooled, $Ab = 200$, $c_b = 1.56 * 10^5$ and $M = 0.89$. As the original formula gives a value in dollars in 2000, it has been inflation adjusted (times 1.29) and converted to euros (times 0.82). The total cost is found to be: **381,962.3 Euros**

7 Cell method

In cell method, one column containing 4 rows is taken for analysis, since it is a cross flow heat exchanger and the values resemble the same for subsequent columns. Now, each tube is discretised into **100 cells**. The energy balance is done for each cell,

$$Q = m_{air} C_p \Delta T = m_{acetone} x \Delta h = U A \Delta T_{lm} \tag{30}$$

in which x is the vapor fraction
U is initially assumed and air outlet temperature from the cell is calculated by solving air side heat transfer and Overall heat transfer. ΔT_{lm} is approximated as $T_{acetone} - T_{air\_in}$ due to small temperature change otherwise and the heat transfer is per unit cell

$$T_{air\_out} = \frac{U * A * \Delta T_{lm}}{(m_{air} * cp)} + T_{air\_in} \tag{31}$$

The vapor fraction is calculated by equating the Q with $m_{acetone} x \Delta h$.
Now, the above shown method is used to calculate U value and the T_{air_out} is updated. This iteration process continues till the error is in specified range.Similarly, the outlet x value is used for successive cells of the same tube, whereas the T_{air_out} is used as T_{air_in} for next tube. After calculating for few tubes, the average heat transfer co-efficient is found to be **24.21 W/mK** . The entire calculations are attached as python code in the appendix

8 Appendix

```
#!/usr/bin/env python
# coding: utf-8

# In[36]:


import numpy as np
import math as m
import sympy as sy
#Condensing
#Acetone 0.3bar limit
m1=4.5
T1=40+273.15
h1_out=-35.480
h1_in=483.25
rho1o=767.66
rho1i=1.3102
rho1=0.5*(rho1o+rho1i)
k1o=149.07*10**-3
k1i=15.378*10**-3
k1=0.5*(k1o+k1i)
mu1o=267.05*10**-6
mu1i=8.4186*10**-6
delmu=mu1o-mu1i
delrho=rho1o-rho1i
mu1=0.5*(mu1o+mu1i)
p1=0.56582*10**5
Q=m1*(h1_in-h1_out)
h1=3000 #Fig.19.1 condensation organic vapors
F1=5000 #Table 19.2 organic vapor
pr1i=0.80653
pr1o=3.9093
pr1=0.5*(pr1i+pr1o)
print('Duty: '+str(Q/1000)+' MW')


#Air
m2=155
T2_in=20+273.15
h2_in=419.41
cp2=1.0061
T2_out=T2_in+(Q/(m2*cp2))
print('T2_out: '+str(T2_out-273.15)+' C')
T2=0.5*(T2_in+T2_out)
```

```
rho2=1.155
k2i=25.873*10**-3
k2o=26.984*10**-3
k2=0.5*(k2i+k2o)
mu2i=18.205*10**-6
mu2o=18.926*10**-6
mu2=0.5*(mu2i+mu2o)

p2=1*10**5
pri=0.70794
pro=0.70606
pr=0.5*(pri+pro)
d=20*10**-3
h2=70 #Fig.19.1 air
F2=10000 #Fouling table 19.2

#Tube choice in slide
#Model 9 45 25
dbo=25*10**-3 #base tube dia
t=1.5*10**-3 #thickness
db=dbo-2*t
df=45*10**-3 #finned tube dia
fh=10*10**-3 #Fin height
ftm=0.4*10**-3 #Fin mean thickness
fp=2.82*10**-3 #Fin pitch

#staggered
s1=s2=50*10**-3 #slide

L=12 #Tube length maximum Laser fin catalogue
#https://www.schmoele.de/wp-content/uploads/2020/03/Schmoele_LASERFIN_TUBES_Web.pdf
Kst=16 #stainless steel thermal conductivity
eff=0.65 #Slides
Nf=L/fp #Number of fins
#circular fin
AAt=1+(2*fh*(fh+dbo+ftm)/(fp*dbo))
Ao=3.14*dbo*L
Att=AAt*Ao
Af=0.5*3.14*(df**2-dbo**2)*Nf #Fin area
At=Af+3.14*dbo*(fp-ftm)*(Nf-1) #Total area
h2v=h2*(1-(1-eff)*(Af/At)) #Virtual coefficient
Ai=3.14*db*L #internal area
U=(((At/(h1*Ai))+(1/h2v)+(1/F1)+(1/F2)+(At*t/(Ai*2*Kst)))**-1) #Overall
print(U)
Ua=U
del_Tm = ((T1-T2_in)-(T1-T2_out))/(np.log((T1-T2_in)/(T1-T2_out))) #LMTD
```

```
print(del_Tm)
F=0.9
#Loop begins

A=Q*1000/(F*U*del_Tm) #Area
Nt=np.ceil(A/At)
Nr=4 #Number of rows less than 10
Ntr=np.ceil(Nt/Nr) #No.of tubes in a row

#altering Nt
Nt=Ntr*Nr
print(Nt)
#bundle Area
Ab=s1*Ntr*L
#Air side flow rate
v2=m2/(rho2*Ab) #in range
den=2*(fp*((s2**2+(s1*0.5)**2)**0.5-dbo)+((s2**2+(s1*0.5)**2)**0.5-df)*ftm)
AA=s1*fp/den
vs=AA*v2
Re=vs*rho2*dbo/mu2
Nu=0.38*(Re**0.6)*(AAt**-0.15)*(pr**(1/3))
ha=Nu*k2/dbo
print(ha)
phi=((df/dbo)-1)*(1+0.35*np.log(df/dbo))
print(phi)
X=phi*(0.5*dbo)*(2*ha/(Kst*ftm))**0.5
eff=np.tanh(X)/X
print(eff)
h2v=ha*(1-(1-eff)*(Af/At)) #Virtual coefficient
#Condensation inside horizontal tubes
np=1#2passes
m1t=m1/(Nt/np)
tulo=m1t/(L) #tube_loading
hst=0.76*k1*(rho1o*(rho1o-rho1i)*9.81/(mu1o*tulo))**(1/3) #stratified
vt=m1t/(rho1*0.25*3.14*db**2)
vt1=m1t/(np*rho1i*0.25*3.14*db**2)
Rea=rho1*vt*db/mu1
han=0.021*(k1/db)*(Rea**0.8)*(pr1**0.43)*(1+(rho1o/rho1i)**0.5)/2 #annular
h1=max(han,hst)
U=(((At/(h1*Ai))+(1/h2v)+(1/F1)+(1/F2)+(At*t/(Ai*2*Kst)))**-1) #Overall
error=abs((Ua-U)*100/U)
print(str(error)+" %")
print("U = "+str(U))

#Pressure drop
a=s1/dbo
```

```
b=s2/dbo
sd=s1/(2*m.sin(m.atan(s1/(s2*2))))
c=(sd/dbo)*((a/2)**2+b**2)**0.5
print(b,0.5*(2*a+1)**0.5) #b>
we=v2*a/(a-1)
Ff=1-m.exp(-(Re+200)/1000)
falv=280*3.14*((b**0.5-0.6)**2+0.75)/((4*a*b-3.14)*a**1.6)
elam=falv/Re
fatv=2.5+(1.2/(a-0.85)**1.08)+0.4*(b/a-1)**3-0.01*(a/b-1)**3
etur=fatv/(Re**0.25)
e=elam+etur*Ff
delp_air=0.5*e*Nr*rho2*we**2
print("Air side delP: "+str(delp_air)+' Pa') #<150 pa
eff_fan=0.5
eff_motor=0.95
W_fan=v2*(s1*Ntr*L)*delp_air/(eff_fan*eff_motor)
print("Fan Power(kW): "+str(W_fan/1000))
#tubeside delP
Tw=T1-U*(T1-T2)/h1
muw=278.35*10**-6
j=3.3*10**-3
delp_tube=np*0.5*(8*j*(L/db)*((mu1i/muw)**-0.14)+2.5)*rho1i*vt1**2
print("Tube side delP: "+str(delp_tube*10**-5)+' bar') #<0.3 bar

Cb=1.56*10**5
Qb=200
M=0.89
C=Cb*(A/Qb)**M
print(C*1.29*0.89,A)
#Control volume method
nc=100 #number of cells in one tube
delh=h1_in-h1_out
mt=m1/Nt
ma=m2/(Ntr*nc)

Tr0=[T2_in]*(nc+1)
Tr1=[0]*(nc+1) #row 1
x1=[0]*(nc+1) #x 1
u1=[0]*(nc+1) #U 1
Tr2=[0]*(nc+1) #row 2
x2=[0]*(nc+1) #x 2
u2=[0]*(nc+1) #U 2
Tr3=[0]*(nc+1) #row 3
x3=[0]*(nc+1) #x 3
u3=[0]*(nc+1) #U 3
Tr4=[0]*(nc+1) #row 4
```

```
x4=[0]*(nc+1) #x 4
u4=[0]*(nc+1) #U 4

#Assume U=20

Aa=At/nc

i=0
while(i<=nc):
    Ua=20
    T2_in=T2_out
    error=40
    x=0
    a=x
    while(error>30):
        LMTD=T1-T2_in #Approx
        T2_out=((Ua*Aa*LMTD)/(ma*cp2*1000))+T2_in
        Q=Ua*Aa*LMTD
        x=a+Q/(mt*delh*1000)
        #Air side flow rate
        Nu=0.38*(Re**0.6)*((AAt/nc)**-0.15)*(pr**(1/3))
        ha=Nu*k2/dbo
        phi=((df/dbo)-1)*(1+0.35*m.log(df/dbo))
        X=phi*(0.5*dbo)*(2*ha/(Kst*ftm))**0.5
        eff=m.tanh(X)/X
        print(eff)
        h2v=h2*(1-(1-eff)*(Af/At)) #Virtual coefficient
        #Condensation inside horizontal tubes
        np=1#2passes
        m1t=m1/(Nt/np)
        tulo=m1t/(L) #tube_loading
        rho1o=rho1i+x*delrho
        rho1=0.5*(rho1i+rho1o)
        mu1o=mu1i+x*delmu
        mu1=0.5*(mu1o+mu1i)
        hst=0.76*k1*(rho1o*(rho1o-rho1i)*9.81/(mu1o*tulo))**(1/3) #stratified
        vt=m1t/(rho1*0.25*3.14*db**2)
        vt1=m1t/(np*rho1i*0.25*3.14*db**2)
        Rea=rho1*vt*db/mu1
        han=0.021*(k1/db)*(Rea**0.8)*(pr1**0.43)*(1+(rho1o/rho1i)**0.5)/2 #annular
        h1=max(han,hst)
        U=(((At/(h1*Ai))+(1/h2v)+(1/F1)+(1/F2)+(At*t/(Ai*2*Kst)))**-1) #Overall
        error=abs((Ua-U)*100/U)
        Ua=U
    i=i+1
    print("U = "+str(U))
```