Original Research Article
Design of Shell and Tube Heat Exchanger with Double Passes

ABSTRACT

The exchange of heat is one of the most important processes in the chemical industry. The shell and tube heat exchanger is the major equipment used to transfer heat from one medium to another. This research work on the Computer-Aided Design (CAD) of shell and tube heat exchangers with double passes aims to provide an easy way to design the shell and tube heat exchanger. A case study question was stated and all the necessary calculations in the thermal design were carried out using the Kern's method of heat exchanger design. The thermal design was then used as a guide to the CAD using computer codes. The computer software program, Visual Basic NET (VB.NET) was used because of its numerous advantages over the other software. The result gotten from the computer-aided design was compared to the result from the thermal design. The computer-aided design software was equally used to test other problems on shell and tube heat exchanger. The computer-aided design was found to be more accurate, quicker and more efficient and it is recommended for use in companies and industries.

Keywords: Visual Basic NET, Computer-Aided Design, Heat Exchanger, Log mean temperature difference

1. INTRODUCTION

Heat Exchangers are specialized devices that assist in the transfer of heat from one fluid to the other. In some cases, a solid wall may separate the fluids and prevent them from mixing. In other designs, the fluids may be in direct contact with each other. In most efficient heat exchangers, the surface area of the wall between the fluids is maximized while simultaneously mixing the fluid flow resistance [1, 2]. There are three primary flow arrangements with heat exchangers including counter-flow, parallel flow, and cross flow.

The most common type of heat exchangers used in the process, petroleum, chemical and HVAC industries intended for heating process fluids is the shell and tube heat exchangers [2, 3]. The shell and tube heat exchanger is used when a process requires large amounts of fluids to be heated or cooled. Due to their design, they offer a large heat transfer area and provide high heat transfer efficiency [4]. They consist of tubes and shells. The tubes act as the flow channels for one of the fluids in the heat exchanger, these exchangers are often parallel in order to provide a large surface area for the heat transfer. On the other hand, the shell holds the tube bundle and acts as the conduit for the fluid. The shell assembly houses the shell side connections and is the actual structure in to which the tube bundle is placed. Shell and tube heat exchangers are used in applications where high pressure and temperature is required. They serve in wide range of applications including compressor system, hydraulic system, stationary engines, pain systems, air dryers, lube oil consoles, and several marine applications [5].

In physics and thermodynamics, heat is the process of energy transfer from one body or system due to thermal contact, which in turn is defined as energy transfer to a body in any other way than due to work performed on the body [2]. When an infinitesimal amount of heat, Q is transferred to a body in thermal equilibrium at absolute temperature, T in a reversible way, then it is given by the quantity, TdS, where S is the entropy of the body. A related term is the thermal energy, which is defined as the energy of a body that increases with its temperature. Heat transfer is a path function (process quantity) as opposed to a part function (state quantity). Heat flows between systems that are not in thermal equilibrium with each other; it spontaneously flows from the areas of high temperature to areas of low temperature [2].

The objective of this study is to design and develop a software for solving shell and tube heat exchanger with double passes.

2. MATERIAL AND METHODS

2.1 Mechanism/Methodology of the design of Shell and Tube Heat Exchanger

2.1.1 Question for Design/Problem statement

Water at the rate of 3000 lbm/hr is heated from 100 to 130 ^{o}F in a shell and tube heat exchanger. On the shell side, one pass is used with water as the heating fluid 15000 lbm/hr entering exchanger at 200 ^{o}F. The overall heat transfer coefficient is 250 Btu/hr/ft^2 and the average water velocity in the ¾ (0.75)-in-ID tubes is 1.2 ft/s; because of space limitations, the exchanger must not be longer than 8ft consistent with the restriction. Calculate, the number of tube passes, the number of tube per pass and length of the tubes. The correction factor for exchanger with one shell pass and two tube passes is 0.88.

2.1.2 Methodology

The heat load, Q and tube side mass flow rate, M_H were calculated as follows [6]:

$$Q = m_c C_{pc} (\Delta T)_c \qquad (1)$$

$$M_H = \frac{Q}{C_{pH}\,\Delta T_H} \qquad (2)$$

The log mean temperature difference (ΔT_{lm}), true mean temperature difference (ΔT_m) and heat transfer area, A were also calculated as [7]:

$$\Delta T_{lm} = \frac{(T_1 - t_2) - (T_2 - t_1)}{Ln \frac{T_1 - t_2}{T_2 - t_1}} \qquad (3)$$

$$\Delta T_m = F_t \times \Delta T_{lm} \qquad (4)$$

$$A = \frac{Q}{U_0 \Delta T_m} \qquad (5)$$

Where F_t is the correlation factor.

2.1.3 Calculation of geometric parameter

A. For tube-side [8]:

Area of one tube, $A_t = \pi \times L \times d_0$ (6)

Number of tubes, $N_t = \frac{A}{A_t}$ (7)

Tube cross-sectional area, $A_{ts} = \frac{\pi \times d_i^2}{4}$ (8)

Number of tubes per pass, $T_p = \frac{N_t}{N_p}$ (9)

Total flow rate, $T_A = T_p \times A_{ts}$ (10)

Tube-side mass velocity, $V_t = \frac{m_H}{T_A}$ (11)

Tube-side linear velocity, $U_t = \frac{V_t}{P_H}$ (12)

Reynold's number for tube-side stream, $Re_t = \frac{\rho_H \times U_t \times d_i}{\mu_t}$ (13)

Prandtl number for tube-side stream, $Pr_t = \frac{Cp_H \times \mu_t}{k_{ft}}$ (14)

B. For shell-side [9]:

Bundle diameter, $D_b = d_0 \left[\frac{N_t}{K_1}\right]^{1/n_1}$ (15)

Diameter for shell-side fluid, $D_s = D_b + D_c$ (16)

Baffle spacing, $l_B = \frac{D_s}{5}$ (17)

Tube pitch, $p_t = S_p \times d_0$ (18)

Cross flow area, $A_s = \frac{(P_t - d_0)\, D_s l_B}{P_t}$ (19)

Shell-side mass velocity, $V_s = \frac{m_C}{A_s}$ (20)

Shell-side linear velocity, $U_s = \frac{V_s}{\rho_c}$ (21)

Equivalent diameter, $d_e = \frac{1.1}{d_0}(P_t^2 - 0.917 d_0^2)$ (22)

Reynold's number for shell-side fluid, $Re_s = \frac{V_s \times d_e}{\mu_s}$ (23)

Prandtl's number for shell- side fluid, $Pr_s = \frac{Cp_m \times \mu_m}{k_{fs}}$ (24)

The tube side heat transfer coefficient and shell side heat transfer coefficient can be calculated using Eqs. 25 and 26, respectively [9]:

$$h_i = \frac{k_{ft}}{d_i} j_{ht} Re_t Pr_t^{0.33} \quad (25)$$

$$h_s = \frac{k_{fs}}{d_e} j_{hs} Re_t Pr_t^{0.33} \quad (26)$$

Also, the tube side pressure drop and shell side pressure drop can be calculated using Eqs. 27 and 28, respectively [9]:

$$\Delta P_t = N_p \left[8 j_{ft}\left[\frac{L}{d_i}\right] + 2.5\right]\frac{\rho_w U_t^2}{2} \quad (27)$$

$$\Delta P_s = 8 j_{fs}\left[\frac{D_s}{d_e}\right]\left[\frac{L}{l_B}\right]\frac{\rho_w U_s^2}{2} \quad (28)$$

Where L = calculated length, d_i = inside diameter, ρ_w = density of water and U_t = average water velocity.

2.1.4 Thermal Design

The thermal design as the name implies has to do with the heat aspect of the exchanger [10]. It is heat because it is the part that deals with temperature related functions of the system. These functions include [2, 11]:

i. Basic heat transfer equation
ii. Overall heat transfer coefficient
iii. Fouling factor
iv. Temperature ratio and heat effectiveness
v. Pressure drop

2.1.5 Basic Heat Transfer Equation

For any shell and tube exchanger, the total area of heat transfer is normally based on the outside effective surface area (A) of all the tubes [12]. For plain tubes, the area (A) is equal to the outside surface area between the inner faces of the tube sheets. The rate of heat transfer across a surface is given as [8]:

$$Q = UA\Delta T_m \quad (29)$$

Where Q = Heat transfer per unit time (W), U = Overall heat transfer coefficient (W/m/^{o}C), A = Heat transfer area (m^2) and ΔT_m = Log mean temperature difference, that is, the temperature driving force (^{o}C).

Before the equation can be used to determine the heat transfer rate, an estimate of the mean temperature difference (ΔT) must be made.

$$\Delta T = F_t \Delta T_{lm} \quad (30)$$

Where ΔT = true mean temperature difference, ΔT_{lm} = logarithmic mean temperature difference and F_t = correction factor.

In a double pass heat exchanger,
For counter-current flows:

$$\Delta T_{lm} = (T_1 - t_2) - (T_2 - t_1) \quad (31)$$

And for co-current flow:

$$\Delta T_{lm} = (T_1 - t_1) - (T_2 - t_2) \quad (32)$$

Where T_1 = inlet high temperature, T_2 = outlet high fluid temperature, t_1 = inlet low fluid temperatutre, and t_2 = outlet low fliud temperature.

Counter current flow is considered for more effective heat transer.

2.1.6 Overall Heat Transfer Coefficient

The overall heat transfer coefficient, the inverse of overall resistance to heat transfer has a relationship with the individual coefficients which are reciprocals of individual resistances. This is given as [13, 14]:

$$\frac{1}{U_0} = \frac{1}{h_0} + \frac{1}{h_{0d}} + \frac{d_0 \, Ln\frac{d_0}{di}}{2k_w} + \frac{d_0}{d_i} \times \frac{1}{h_i} \times \frac{d_0}{d_i} \times \frac{1}{h_{id}} \quad (33)$$

Where U_o = Overall heat coefficient based on outside area of tube (W/m/^{o}C), h_o = outside film coefficient (shell-side) (W/m/^{o}C), h_i = inside film coefficient (tube-side) (W/m/^{o}C), h_{od} = outside dirt coefficient (tube-side) (W/m/^{o}C), h_{id} = inside dirt coefficient (W/m/^{0}C), K_w = thermal conductivity of the tube wall material (W/m/^{0}C), and d_i = tube inside diameter (m), d_o = tube outside diameter (m).

2.1.7 Shell Heat Transfer Cofficient

The individual heat transfer coefficients for shell-side (h_s) according to Kern is given as [15]:

$$h_s = (\frac{K_f}{d_e})(j_h Re\ Pr)^{1/3} \quad (34)$$

Where h_s = shell-side heat transfer coefficient (W/m/^{0}C), K_f = fluid thermal conductivity (W/m/^{0}C), d_e = equivalent diameter (m), R_e = Reynolds number, Pr = prandtl number, j_h = heat transfer factor, and U = viscosity (kg/ms).

2.1.8 Tube-Side Heat Transfer Coefficient

The individual coefficient tube-side, h_i can be calculated from Eq. 35 [15]:

$$h_i = K_f j_h Re\ Pr^{0.33} \quad (35)$$

where h_i = tube-side heat transfer coefficient (W/m/^{0}C), and j_h = heat transfer factor.

2.2 Solution and Calculation

Step 1: Specification

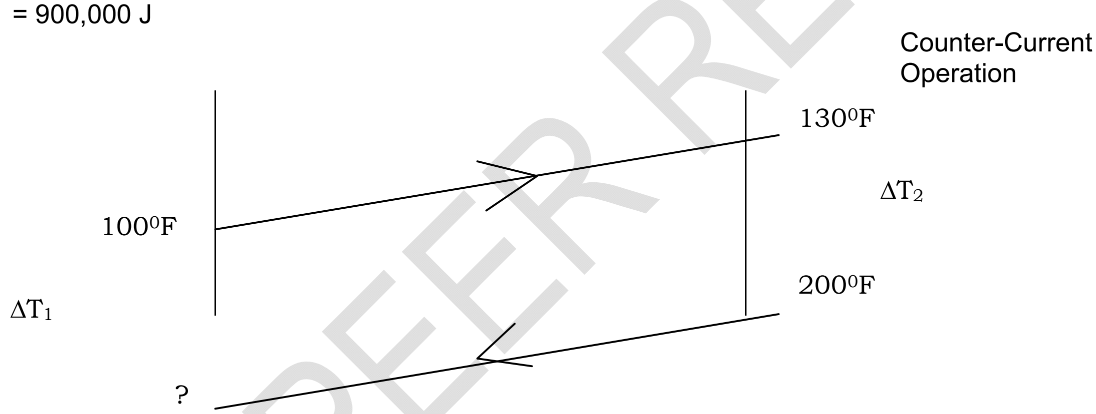
Calculating the heat transfer rate and the outlet temperature:

Heat transfer rate, $Q = MC_p\Delta T$

= 30000 x 1 x (130 – 100)

= 900,000 J

Outlet temperature



$Q = M_C C_C T_C = \quad M_H C_H T_H$

(Cold water) (Hot water)

M_C = 30000 lbm/hr, C_C = 1 btu/lbm^{o}F, T_C = (130 – 100) ^{o}F

M_H = 150000 lbm/hr, C_H = 1 btu/lbm^{o}F, T_H = ?

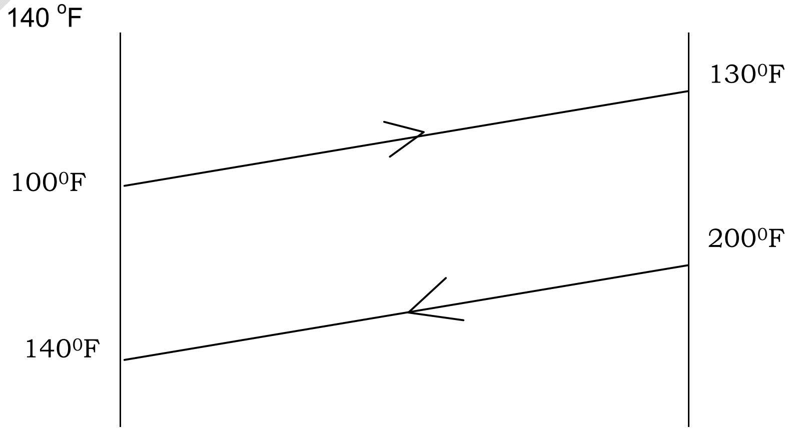
$M_c C_c T_c$ = $M_H C_H T_H$

30000 x 1 x (130 – 100) = 15000 x 1 x T_H

900000 = 15000T_H

$$T_H = \frac{900000}{15000} = 60\ ^oF$$

$\Delta T_1 = T_2 - T_H$ = 200 – 60 = 140 ^{o}F



$Mean\ temperature\ of\ water = \frac{Inlet+outlet}{2} = \frac{200+130}{2} = 165\ ^0\mathrm{F}$
(Where Inlet temperature = 200 ^{0}F and outlet temperature = 180 ^{0}F)

STEP 2: Physical properties of water
Inlet temp.: 200 ^{0}F, Outlet temp.: 130 ^{0}F, mean temp.: 165 ^{0}F, Heat capacity coefficient: 1 Btu/lbm/^{0}F, Thermal conductivity: 0.59 W/m/^{0}C, Density: 1000 kg/m^3 = 62.37 lbm/ft^3 and viscosity of water: 0.0008 N.s/m^2

STEP 3: Overall coefficient
The overall heat transfer coefficient (μ) is 250 Btu/hr/ft^2

STEP 4: Shell and tube heat exchanger with one shell pass and two tube passes

i) $\Delta T_{LMTD} = \dfrac{\Delta T_2 - \Delta T_1}{In\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \dfrac{70-40}{In\left(\frac{70}{40}\right)} = 53.6097\ ^0F$

ii) Correction factor = 0.88

iii) The temp. difference, $\Delta T_m = \Delta T_{LMTD} \times C_f = 53.6097 \times 0.88 = 47.1765\ °\mathrm{F}$

STEP 5: Heat transfer area

$$q = \mu A D T_{LMTD} \rightarrow A = \frac{q}{\mu \Delta T_{LMTD}} = \frac{900{,}000}{250 \text{ x } 53.6097} = 67.15\ \mathrm{ft}^2$$

STEP 6: Layout and tube size
Using a split-ring floating head exchanger for efficiency and ease of cleaning, so plain carbon steel can be used for the shell and tube.
Using Triangular pitch pattern (a constant)
Tube pitch, p_t = 1.25d_o [9]
Where d_o is the tube outside diameter.
Given, d_o = 0.75($^3/_4$) inch =19.05 mm ≅ 0.0625 ft
With Triangular pitch, p_t = 1.25 x 19.05 = 23.81 mm pitch ≅ 0.0781 ft

STEP 7: Number of tubes
From $Q = M_c C_c T_c$, But $M_c = \rho AV$, A = Area, V = Velocity, ρ = density
ρH_20 = 1000 kg/m^3 = 62.37 lbm/ft^3, m = 3000 lbm/h, r = 1.2 ft/s and 1 hr = 3600 s

$$A = \frac{m}{\rho r} = \frac{30000}{62.37 \text{ x } 1.2 \text{ x } 3600} = 0.111\ \mathrm{m}^2$$

But, A (Total flow area) = $\dfrac{n\pi\,\partial^2}{4}$

The Area is the product of the number of tubes and the flow
A = 0.111, δ = 0.75inch = 0.0625ft (since 1 inch = 2.54 cm and 1 ft = 30.48 cm)
1 inch = 2.54 cm
? = 30.48 cm
$\frac{30.48 \text{ x } 1}{2.54} = 12 \text{ inch}$
1 ft = 30.48 cm = (12 inch)^2 = 144 inch^2

$n = \dfrac{4A}{\delta^2 \pi} = \dfrac{4 \text{ x } 0.111 \text{ x } 144}{(0.75)^2 \text{ x } 3.142}$ = 36.17, n = 36 tubes (Number of tubes per pass)

But the total surface area required for one tube pass exchanger was 67.2 ft^2

Length of the tube
$A = \pi\delta L$, $67.2 = n\pi\delta L$

$L = \dfrac{67.2}{n\pi\delta} = \dfrac{67.2 \text{ x } 12}{36 \text{ x } 3.142 \text{ x } 0.75}$ = 9.51ft

The length is greater than the allowable 8ft so we must use more than one pass
$\mathrm{q = UADLMTD \text{ x } correction\ factor}$

A (total) = $\dfrac{q}{U \times \Delta_{\mathrm{LMTD}} \times \text{correction factor}} = \dfrac{900000}{250 \text{ x } 53.6079 \text{ x } 0.88}$ = 76.3ft^2

For two tube pass system, the total surface area is now related to the length by

$A = 2n\pi\delta l$, $L = \dfrac{A}{2n\pi\delta} = \dfrac{76.3 \text{ x } 12}{2 \text{ x } 36 \text{ x } 3.142 \text{ x } 0.75}$

Number of tubes per pass = 36, Number of passes = 2
Length of tube per pass = 5.490ft, Area per pass = 36 x 0.11 = 3.996 ft^2

Volumetric flow rate = flow rate x Density of H_20 = 15000 $\frac{lbm}{hr} \times \frac{1\, ft^3}{62.7\, lbm}$ = 240 ft^3/hr

Tube side velocity $= \frac{\text{volumetric flow rate}}{\text{Area per pass}} = \frac{240}{3.996}$ = 60.1 ft/hr

STEP 8: Bundle and shell diameter

Bundle diameter, $D_b = \delta_0 \left(\frac{N_t}{K_1}\right)^{1/n_i}$

Given, δ_o (tube outside diameter) = 20, K_1 = 0.249, n_i = 2.207
Using Triangular pitch pattern for two passes
N_t = Number of tubes per pass x Number of passes = 36 x 2 = 72

$D_b = 20\left(\frac{72}{0.249}\right)^{1/2.207}$ = 260.72 mm = 0.2607 m ≅ 0.8 ft

For a split-ring floating head exchanger, the typical shell clearance from (from standard value is 52 mm)
That is tracing 0.2607 on the graph [9]
Shell clearance = 52 mm = 0.17 ft (1 ft = 30.48 cm = 304.8 mm)
Shell diameter, ∆s = 0.85 + 0.17 = 1.02 ft

STEP 9: Tube-side heat transfer coefficient

(i.) Reynolds number, $Re = \frac{\rho\mu\delta}{\mu}$

ρ = 1000 kg/m^3, μ (from tube side velocity) = 60.1 ft/hr = 0.0051 m/s, δ (from the question) internal diameter = 0.75 inch = 0.019 m and μ (standard viscosity of H_20) = 0.8 mNs/m^2

$Re = \frac{1000 \text{ x } 0.0051 \text{ x } 0.019}{0.8 \text{ x } 10^{-3}}$ = 121.125 ≅ 1.211 x 10^2

(ii.) Prandlt number, $Pr = \frac{C_p\mu}{K_f}$

Where C_p of H_20 in S.I. unit = 4.2 kJ/kg/^{0}C, μ = 0.8 x 10^{-3}, and thermal conductivity of H_20, k_f = 0.59 W/m/^{0}C.

$Pr = \frac{4.2 \text{ x } 10^3 \text{ x } 0.8 \text{ x } 10^{-3}}{0.59}$ = 5.69

(iii.) $\frac{L}{\delta_i}$

Where L = Given length from question: 8ft, δ_i = Inside diameter =14.88mm ≅ 0.0486 ft

$\frac{L}{\delta_i} = \frac{8}{0.0486} = 164.61$

Using Reynolds number, R_e: 1.2112 x 10^2 and 164.61 to trace in graph of tube-side heat transfer factor, 9.0 x 10^{-3} was gotten as standard value in Coulson and Richardson [9] and j_h = 9.0 x 10^{-3}.
(iv.) Nusselt number,
Nu = 9.0 x 10^{-3} x (R_e) x $(Pr)^{0.33}$ = 9.0 x 10^{-3} x (121.125) x $(5.69)^{0.33}$ = 1.93
(v.) Tube side coefficient

$Nu = \frac{h_i\delta_i}{k_f} \rightarrow h_i = \frac{N_i \times k_f}{\delta_i} = \frac{1.93 \text{ x } 0.59}{0.0486}$ = 23.43 W/ft^2

Step 10: Shell-side heat exchanger coefficient

Calculating for one-shell pass (Triangular pitch)
Given, k_i = 0.319, n_i = 2.142, δ_o = 20

(i.) Bundle Diameter, $D_b = \delta_o \left(\frac{N_t}{k_i}\right)^{1/n_i}$

$D_b = 20\left(\frac{72}{0.319}\right)^{1/2.142}$ = 250.99 mm ≅ 0.82 ft

(ii.) Baffle spacing, $l_B = \frac{D_s}{5} = \frac{1.66}{5} = 0.33$ ft

Since, D_s = 560 mm = 22.04725 inch = 1.8373 ft

(iii.) Cross-flow Area, A_s:
Given, Triangular pitch ≅ 0.0781 ft

$A_s = \frac{0.0781 - 0.0625}{0.0781} \times 1.66 \times 0.33 = 0.1094\ ft^2$

(iv.) Equivalent diameter, de:

$\delta_e = \frac{1.1}{d_0}\left(P_t^2 - 0.917 d_0^2\right) = \frac{1.1}{0.0625}\left(0.0781^2 - 0.917 \times 0.0625^2\right)$

$= 17.6\ (0.00609961 - 0.003582) = 0.044\ ft$

(v.) Volumetric flow-rate on shell side

Flow rate x density of H_2O (converted) = 481 ft^3/hr

(vi.) Shell-side velocity = $\frac{\text{volumetric flow rate}}{\text{cross-flow rate}} = \frac{481\ ft^3/hr}{0.1094\ ft^2}$ = 4396.71 ft/hr

(vii.) Reynolds number, $R_e = \frac{\rho \mu \delta}{\mu}$

Given, δ = 1000 kg/m^3

μ (shell side velocity) = 4396.71 ft/hr = 0.3722 m/s

δ (Equivalent diameter) = 0.044 ft = 0.013m

$R_e = \frac{1000 \times 0.3722 \times 0.013}{0.8 \times 10^{-3}} = 6048.25 \cong 6.04825 \times 10^3$

(viii.) Prandtl number, $Pr = \frac{C_p \mu}{k_f} = \frac{4.2 \times 10^3 \times 0.8 \times 10^{-3}}{0.59} = 5.69$

Using the Reynolds number, R_e of 6.04825×10^3, trace on graph (shell-side heat transfer factor, use segmental baffles with a 25% cut) [7].

$h_s = \left(\frac{k_f}{d_e}\right) jh \times Re \times Pr0.33 = \left(\frac{0.59}{0.13}\right) \times 7.5 \times 10^{-3} \times 6048.25 \times 5.69^{0.333} = 367.326\ W/ft/^0F$

STEP 11: Pressure drop

Tube side (two passes):

(8 and 2.5 are given standard conditions)

$\Delta P_t = N_p \left[8 j_{ft} \left[\frac{L}{d_i}\right] + 2.5\right] \frac{\rho_w U_t^2}{2}$ = 943.03 lbm/ft^2

Shell side:

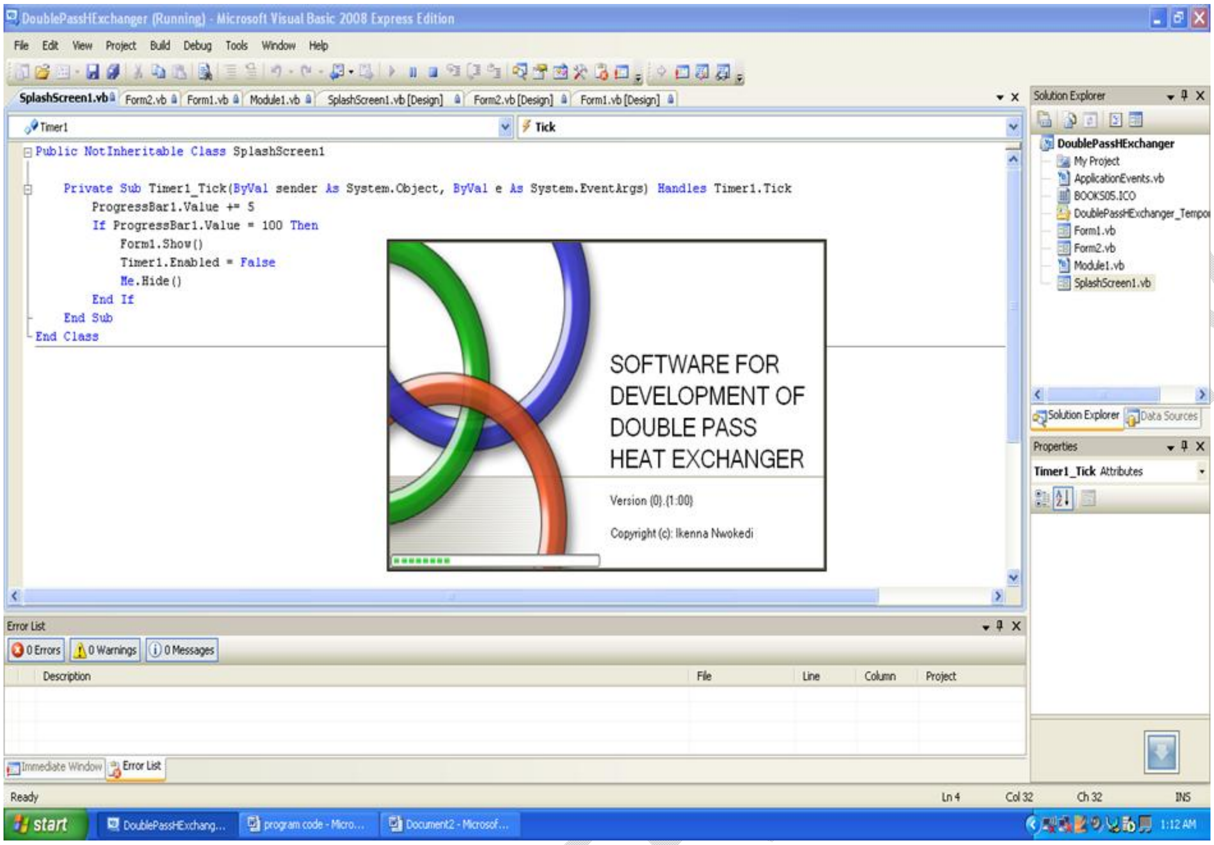
$\Delta P_s = 8 J_f \left(\frac{D_s}{\delta_e}\right)\left(\frac{L}{L_B}\right)\frac{\rho_m V^2}{2}$

Where D_s = Shell diameter, δ_e = equipment diameter, L_B = Baffle spacing, V = velocity

$\Delta P_s = 8 \times 7.5 \times 10^{-3} \left(\frac{1.8373}{0.044}\right)\left(\frac{5.40}{0.33}\right)\frac{62.37 \times 1.2^2}{2}$ **=** 1841 lbf/ft^2 = 12.8 psi

3. RESULTS AND DISCUSSION

The results obtained from the thermal design and the computer-aided design is shown below. For the computer-aided design, the values are used as the results which are the output. There are five interfaces; interface 1 shows the beginning of the program, interface 2 shows default (before the values are loaded). Interfaces 3, 4 and 5 show the calculations made. Also, the flow chart of the design program is shown in Fig. 1.


DoublePassHExchanger (Running) - Microsoft Visual Basic 2008 Express Edition
File Edit View Project Build Debug Tools Window Help
SplashScreen1.vb Form2.vb Form1.vb Module1.vb SplashScreen1.vb [Design] Form2.vb [Design] Form1.vb [Design]
Timer1
Tick
Public NotInheritable Class SplashScreen1
Private Sub Timer1_Tick(ByVal sender As System.Object, ByVal e As System.EventArgs) Handles Timer1.Tick
ProgressBar1.Value += 5
If ProgressBar1.Value = 100 Then
Form1.Show()
Timer1.Enabled = False
Me.Hide()
End If
End Sub
End Class
SOFTWARE FOR DEVELOPMENT OF DOUBLE PASS HEAT EXCHANGER
Version {0}.{1:00}
Copyright (c): Ikenna Nwokedi
Solution Explorer
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My Project
ApplicationEvents.vb
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Form1.vb
Form2.vb
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SplashScreen1.vb
Properties
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0 Messages
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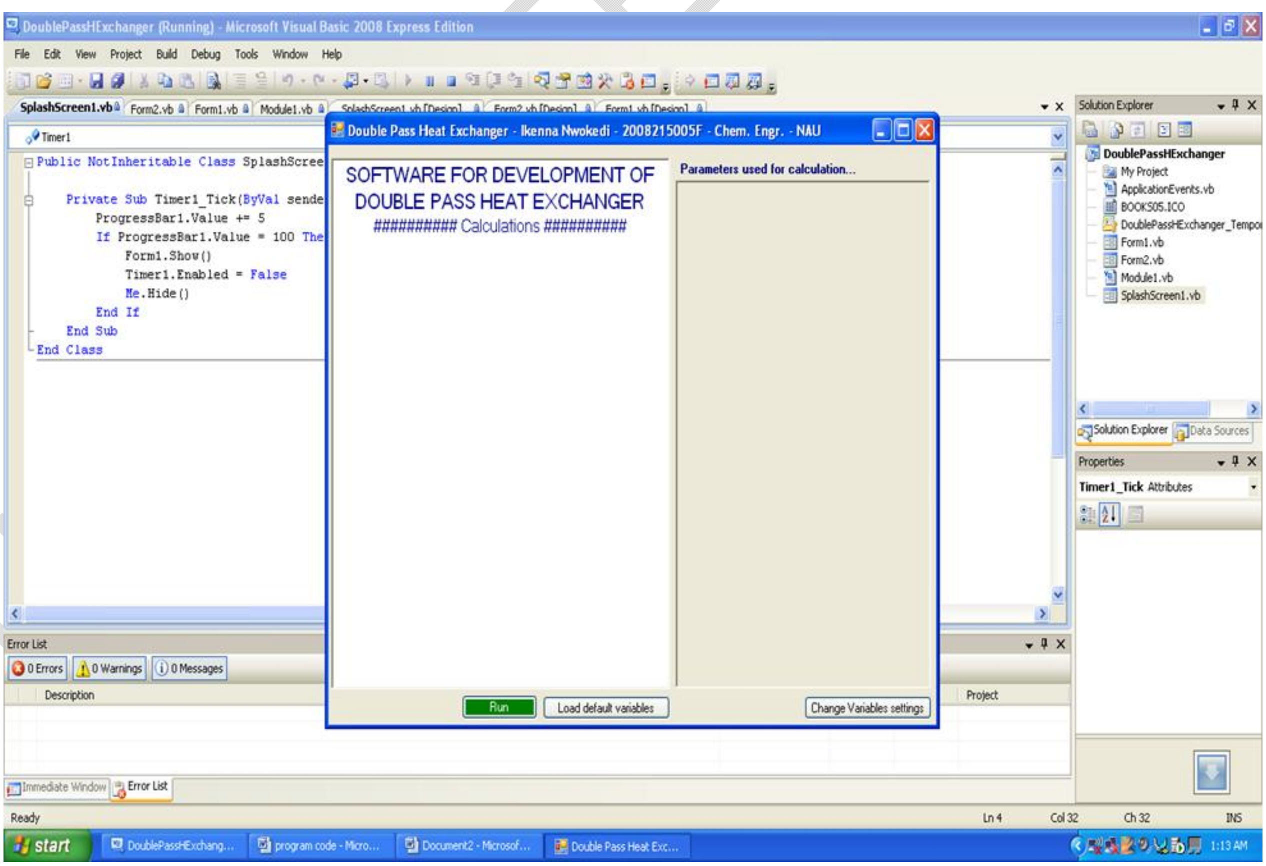
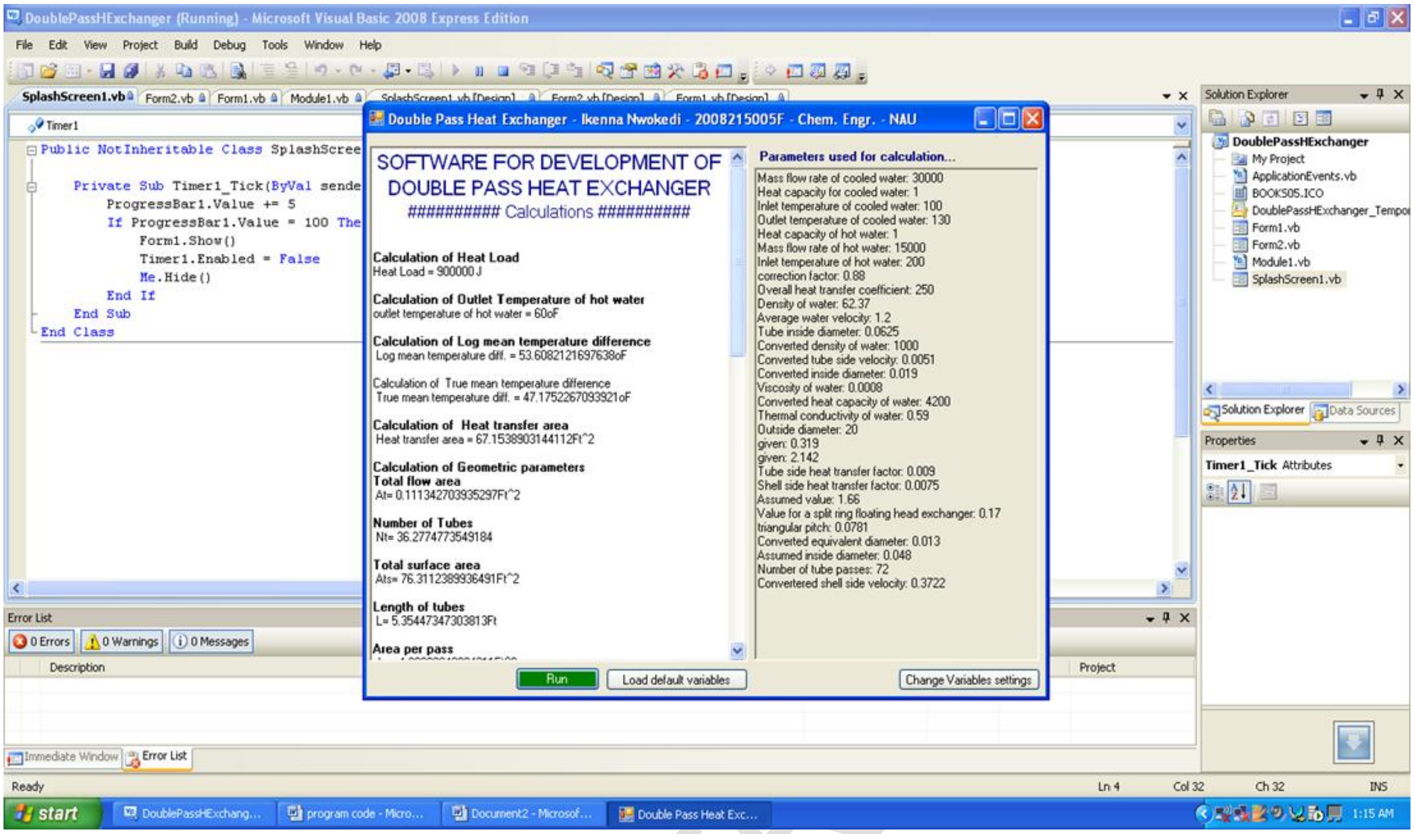

DoublePassHExchanger (Running) - Microsoft Visual Basic 2008 Express Edition
Double Pass Heat Exchanger - Ikenna Nwokedi - 2008215005F - Chem. Engr. - NAU
SOFTWARE FOR DEVELOPMENT OF DOUBLE PASS HEAT EXCHANGER
########## Calculations ##########
Parameters used for calculation...
Run
Load default variables
Change Variables settings
Public NotInheritable Class SplashScreen1
Private Sub Timer1_Tick(ByVal sender
ProgressBar1.Value += 5
If ProgressBar1.Value = 100 Then
Form1.Show()
Timer1.Enabled = False
Me.Hide()
End If
End Sub
End Class
Solution Explorer
DoublePassHExchanger
My Project
ApplicationEvents.vb
BOOKS05.ICO
Form1.vb
Form2.vb
Module1.vb
SplashScreen1.vb
Properties
Timer1_Tick Attributes
Error List
0 Errors
0 Warnings
0 Messages
Ready
Ln 4
Col 32
Ch 32
INS
start
1:13 AM



DoublePassHExchanger (Running) - Microsoft Visual Basic 2008 Express Edition
Public NotInheritable Class SplashScree
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ProgressBar1.Value += 5
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Form1.Show()
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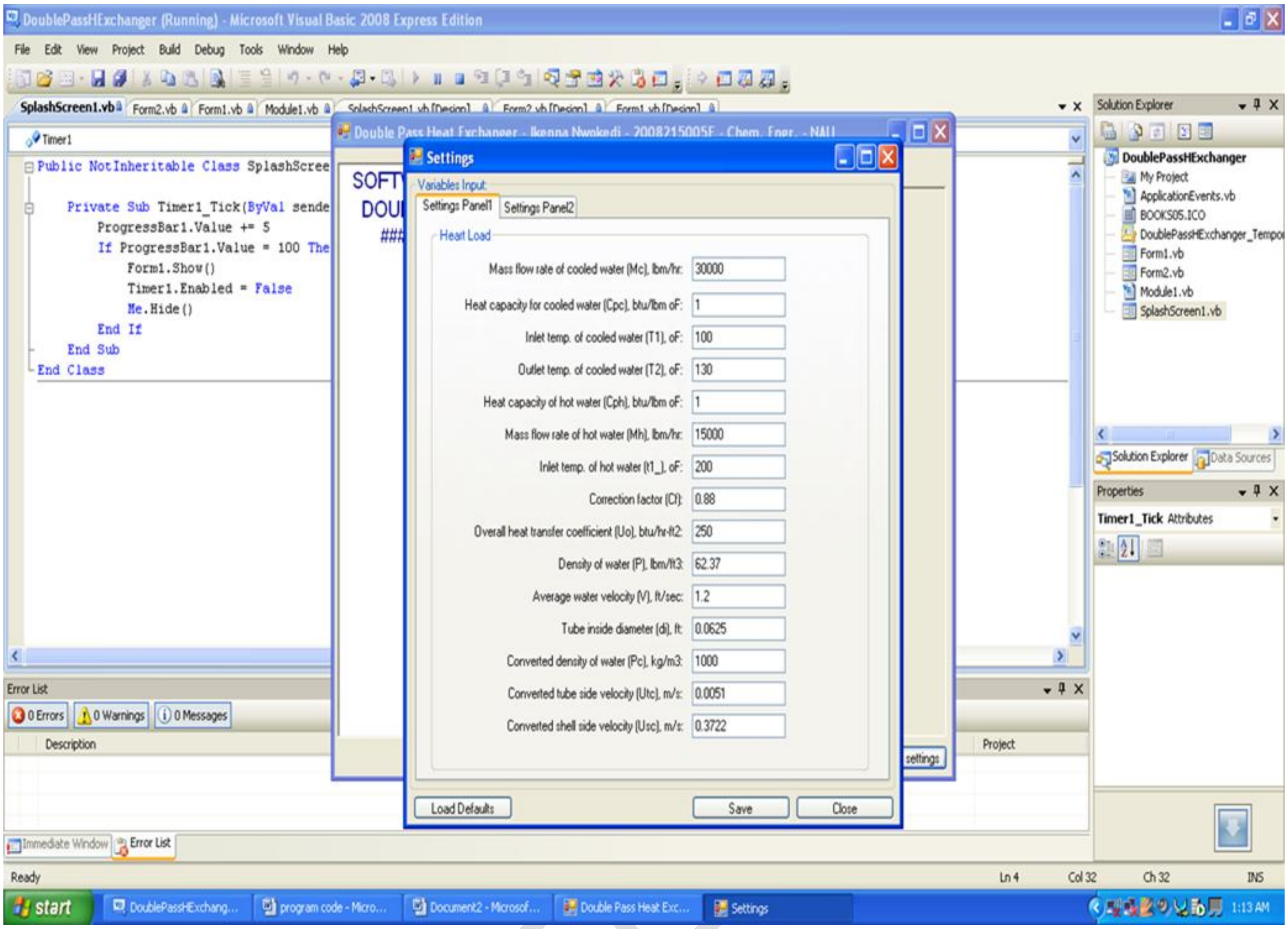
End Class
Double Pass Heat Exchanger - Ikenna Nwokedi - 2008215005F - Chem. Engr. - NAU
SOFTWARE FOR DEVELOPMENT OF
DOUBLE PASS HEAT EXCHANGER
########## Calculations ##########
Calculation of Heat Load
Heat Load = 900000 J
Calculation of Outlet Temperature of hot water
outlet temperature of hot water = 60oF
Calculation of Log mean temperature difference
Log mean temperature diff. = 53.6082121697638oF
Calculation of True mean temperature difference
True mean temperature diff. = 47.1752267093921oF
Calculation of Heat transfer area
Heat transfer area = 67.1538903144112Ft^2
Calculation of Geometric parameters
Total flow area
At= 0.111342703935297Ft^2
Number of Tubes
Nt= 36.2774773549184
Total surface area
Ats= 76.3112389936491Ft^2
Length of tubes
L= 5.35447347303813Ft
Area per pass
Parameters used for calculation...
Mass flow rate of cooled water: 30000
Heat capacity for cooled water: 1
Inlet temperature of cooled water: 100
Outlet temperature of cooled water: 130
Heat capacity of hot water: 1
Mass flow rate of hot water: 15000
Inlet temperature of hot water: 200
correction factor: 0.88
Overall heat transfer coefficient: 250
Density of water: 62.37
Average water velocity: 1.2
Tube inside diameter: 0.0625
Converted density of water: 1000
Converted tube side velocity: 0.0051
Converted inside diameter: 0.019
Viscosity of water: 0.0008
Converted heat capacity of water: 4200
Thermal conductivity of water: 0.59
Outside diameter: 20
given: 0.319
given: 2.142
Tube side heat transfer factor: 0.009
Shell side heat transfer factor: 0.0075
Assumed value: 1.66
Value for a split ring floating head exchanger: 0.17
triangular pitch: 0.0781
Converted equivalent diameter: 0.013
Assumed inside diameter: 0.048
Number of tube passes: 72
Convertered shell side velocity: 0.3722
Run
Load default variables
Change Variables settings


DoublePassHExchanger (Running) - Microsoft Visual Basic 2008 Express Edition
File Edit View Project Build Debug Tools Window Help
SplashScreen1.vb Form2.vb Form1.vb Module1.vb
Timer1
Public NotInheritable Class SplashScree
Private Sub Timer1_Tick(ByVal sende
ProgressBar1.Value += 5
If ProgressBar1.Value = 100 The
Form1.Show()
Timer1.Enabled = False
Me.Hide()
End If
End Sub
End Class
Settings
Variables Input
Settings Panel1
Settings Panel2
Heat Load
Mass flow rate of cooled water (Mc), lbm/hr: 30000
Heat capacity for cooled water (Cpc), btu/lbm oF: 1
Inlet temp. of cooled water (T1), oF: 100
Outlet temp. of cooled water (T2), oF: 130
Heat capacity of hot water (Cph), btu/lbm oF: 1
Mass flow rate of hot water (Mh), lbm/hr: 15000
Inlet temp. of hot water (t1_), oF: 200
Correction factor (Cf): 0.88
Overall heat transfer coefficient (Uo), btu/hr-ft2: 250
Density of water (P), lbm/ft3: 62.37
Average water velocity (V), ft/sec: 1.2
Tube inside diameter (di), ft: 0.0625
Converted density of water (Pc), kg/m3: 1000
Converted tube side velocity (Utc), m/s: 0.0051
Converted shell side velocity (Usc), m/s: 0.3722
Load Defaults
Save
Close
Solution Explorer
DoublePassHExchanger
My Project
ApplicationEvents.vb
BOOKS05.ICO
Form1.vb
Form2.vb
Module1.vb
SplashScreen1.vb
Properties
Timer1_Tick Attributes
Error List
0 Errors
0 Warnings
0 Messages
Description
Project
Immediate Window
Error List
Ready
Ln 4
Col 32
Ch 32
INS
start
Settings
1:13 AM

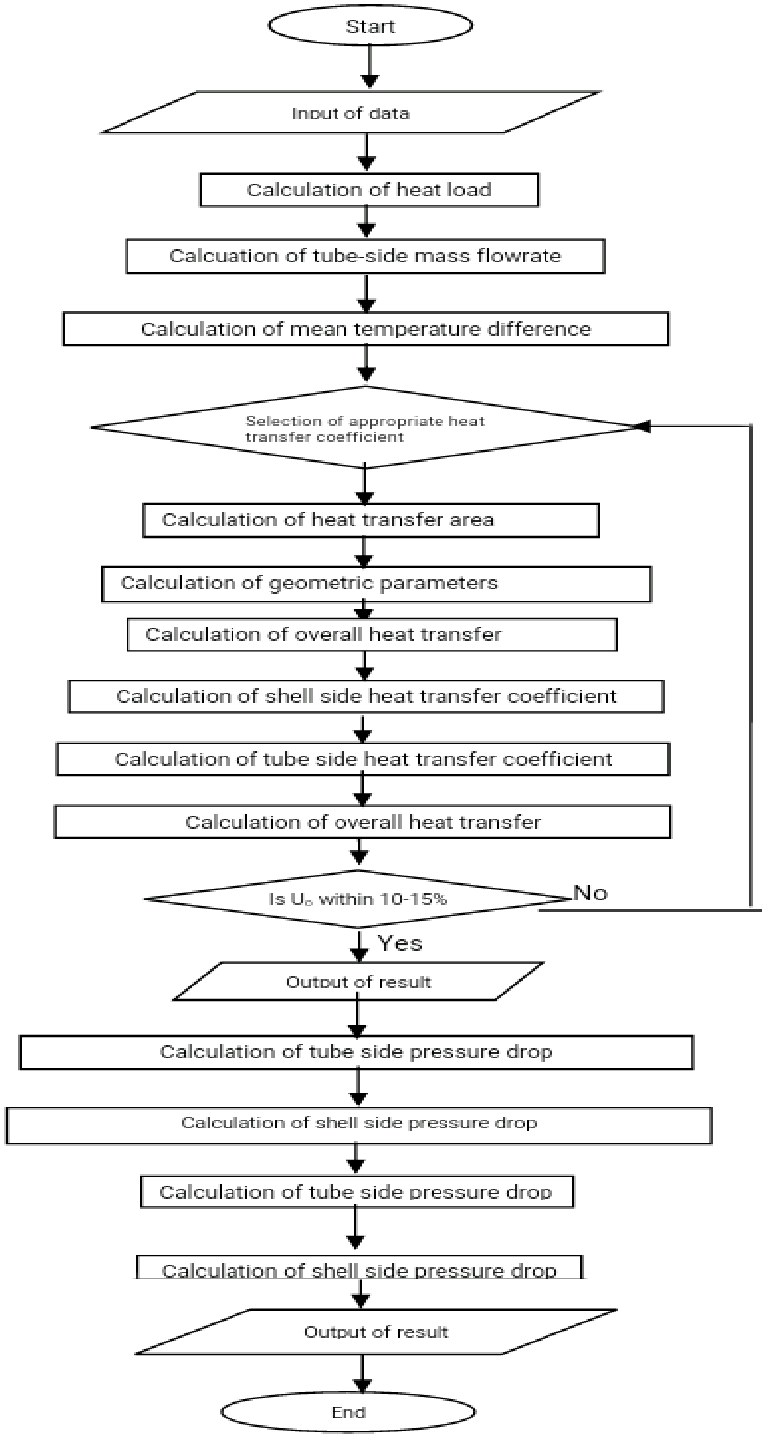


Fig. 1. Flow chart of the design program.

4. CONCLUSION

From this work, slight variations between the results from the computer design were observed. Some of the variations were seen in the pressure drop calculated for both shell and tube side. It was discovered that the software (VB.NET) computes values mostly between 8-16 digit decimal places while the manual design computes the values with 4 digit decimal places. So this makes the computer software more accurate and reliable than the manual design. The software also takes lesser time in the computation of the design value compared to the manual design. This shows that the software can accurately and effectively design any shell and tube heat exchanger and any other form of exchanger as long as the Kern's method of heat exchanger design is used, with the input parameters fed correctly at the specified unit.

This program should be developed to increase its accuracy and to build up an available database so as to reduce the number of required input data.

DECLARATION

Authors declare that this work is part of the thesis of by Nwokedi Ikenna Chukwudi, A Research Thesis Submitted to the Department of Chemical Engineering, Faculty of Engineering, Nnamdi Azikiwe University, University, Awka, Nigeria in Partial Fulfillment of the Requirement for the Award of Post-Graduate Diploma (PGD) in Chemical Engineering, 2010 (https://unizik.edu.ng/oer/download/design-development-software-solving-shell-tube-heat-exchanger-double-passes/).

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COMPETING INTERESTS

Authors have declared that no competing interests exist.

AUTHORS' CONTRIBUTIONS

'Author ICN' designed the study, performed the design and analysis, wrote the protocol, and wrote the first draft of the manuscript. 'Author CAI' managed the literature searches. All authors read and approved the final manuscript.

REFERENCES

1. Saunders EAD. Heat Exchangers: selection Design and Construction, Scientific and Technical, Longman press Inc., N.T.;1988:421-493.

2. Nwokedi IC. Design and development of software for solving shell and tube heat exchanger with double passes. Post-Graduate Diploma (PGD), Department of Chemical Engineering, Nnamdi Azikiwe University, Awka, Nigeria; 2010.

3. Narayanan CM, Bhadra S, Das MK. Computer Aided Design of Shell and Tube Heat Exchangers (Incorporating Most Recent Developments), Advances in Chemical Engineering, Chapter 4. http://openaccessebooks.com/chemical-engineering/computer-aided-design-of-shell-and-tube-heat-exchangers-incorporating-most-recent-developments.pdf

4. Holman JP. Heat Transfer. 9th ed. McGraw-Hill, New York; 2002:184-186.

5. Reif F. Fundamentals of statistical and thermal physics. MCGraw-Hill, Inc., Singapore, 2000:66.

6. Perry RH, Green DW. Perry's Chemical Engineers' Handbook. 6th Ed. McGraw-Hill;1984.

7. Peters MS, Timmerhaus KD, West RE. Plant Design and Economics for Chemical Engineers. McGraw-Hill, New York. 2003;331-352.

8. Coulson J, Richardson J. Chemical Engineering – Design (SI units). Pergamon Press, Oxford; 1983(6):273-280.

9. Coulson J, Richardson J. Chemical Engineering – Fluid flow. Heat Transfer and Mass, Pergamon Press, Oxford,1999;242-252.

10. Schroeder VN, Daniel V. An introduction to thermal physics. Addison-Wesley, San Francisco, California, 2000:18.

11. Bird S, Light F. Transport Phenomena. John Wiley and Sons, Inc., New York. 1960:399.

12. Perrot A, Pierre D. A to Z of Thermodynamics. Oxford University Press; 1998:75 – 77.

13. Smith JM, Van-Ness HC, Abbot MM. Introduction to Chemical Engineering Thermodynamics. McGraw-Hill, New York. 2005:96 – 105.

14. Rajput RK. Heat and mass transfer in SI units. 2nd ed. S. Chand and Company Limited; 2003:529-626.

15. Cengel YA, Boles M. Thermodynamics – An Engineering Approach. 4th ed. McGraw-Hill; 2002:17-18.