

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's**. 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]. 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. 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 [2]. 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 [3].

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. 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.

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 3000lbm/hr is heated from 100 to 130 °F in a shell and tube heat exchanger. On the shell side, one pass is used with water as the heating fluid 15000lbm/hr entering exchanger at 200°F. The overall heat transfer coefficient is 250btu/hr-ft² and the average water velocity in the ¾ (0.75)-in-ID tubes is 1.2ft/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 [4]:

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

$$M_H = \frac{Q}{c_{pH} \Delta T_H} \quad (2)$$

The log mean temperature difference (ΔT_{lm}), true mean temperature difference (ΔT_m) and heat transfer area, A were also calculated as [5]:

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

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

$$A = \frac{Q}{U_o \Delta T_m} \quad (5)$$

Where F_t is the correlation factor.

2.1.3 Calculation of geometric parameter

A. For tube-side [6]:

$$\text{Area of one tube, } A_t = \pi \times L \times d_o \quad (6)$$

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

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

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

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

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

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

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

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

B. For shell-side [7]:

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

$$\text{Diameter for shell-side fluid, } D_s = D_b + D_c \quad (16)$$

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

$$\text{Tube pitch, } p_t = S_p \times d_o \quad (18)$$

$$\text{Cross flow area, } A_s = \frac{(p_t - d_o) D_s l_B}{p_t} \quad (19)$$

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

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

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

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

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

The tube side heat transfer coefficient and shell side heat transfer coefficient can be calculated using Eqs. 25 and 26, respectively [7]:

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

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

Also, the tube side pressure drop and shell side pressure drop can be calculated using Eqs. 27 and 28, respectively [7]:

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

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

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 [8]. It is heat because it is the part that deals with temperature related functions of the system. These functions include [9]:

- Basic heat transfer equation
- Overall heat transfer coefficient
- Fouling factor
- Temperature ratio and heat effectiveness
- 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 [10]. 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 [6]:

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

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

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 temperature, and t_2 = outlet low fluid temperature.

Counter current flow is considered for more effective heat transfer.

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 [11, 12]:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o \ln \frac{d_o}{d_i}}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_i} \times \frac{d_o}{d_i} \times \frac{1}{h_{id}} \quad (33)$$

Where U_o = Overall heat coefficient based on outside area of tube ($W/m^2\text{ }^\circ\text{C}$), h_o = outside film coefficient (shell-side) ($W/m^2\text{ }^\circ\text{C}$), h_i = inside film coefficient (tube-side) ($W/m^2\text{ }^\circ\text{C}$), h_{od} = outside dirt coefficient (tube-side) ($W/m^2\text{ }^\circ\text{C}$), h_{id} = inside dirt coefficient ($W/m^2\text{ }^\circ\text{C}$), K_w = thermal conductivity of the tube wall material ($W/m\text{ }^\circ\text{C}$) and d_i = tube inside diameter (m), d_o = tube outside diameter (m).

2.1.7 Shell Heat Transfer Coefficient

The individual heat transfer coefficients for shell-side (h_s) according to Kern is given as [13]:

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

Where h_s = shell-side heat transfer coefficient ($W/m^2\text{C}$), K_f = fluid thermal conductivity ($W/m\text{C}$), d_e = equivalent diameter (m), Re = 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 [13]:

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

where h_i = tube-side heat transfer coefficient ($W/m^2\text{C}$) and j_h = heat transfer factor.

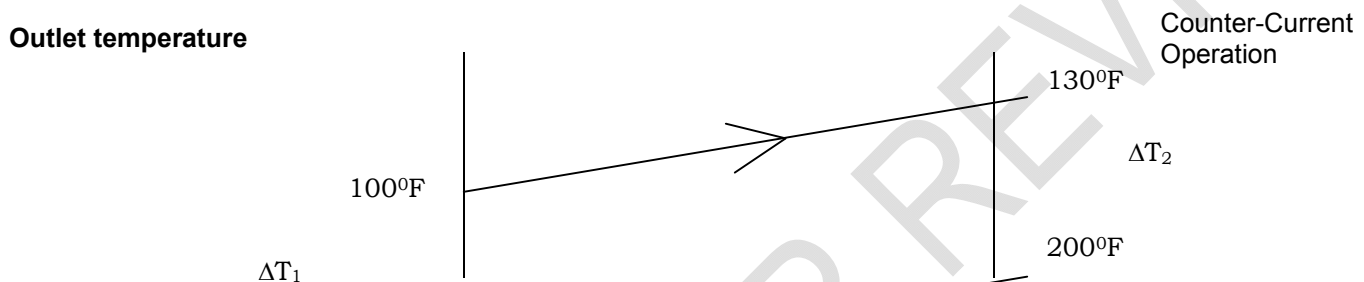
2.2 Solution and Calculation

Step 1: Specification

Calculating the heat transfer rate and the outlet temperature:

$$\begin{aligned} \text{Heat transfer rate, } Q &= MC_p \Delta T \\ &= 30000 \times 1 \times (130 - 100) \\ &= 900,000 \text{ J} \end{aligned}$$

Outlet temperature



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

(Cold water) (Hot water)

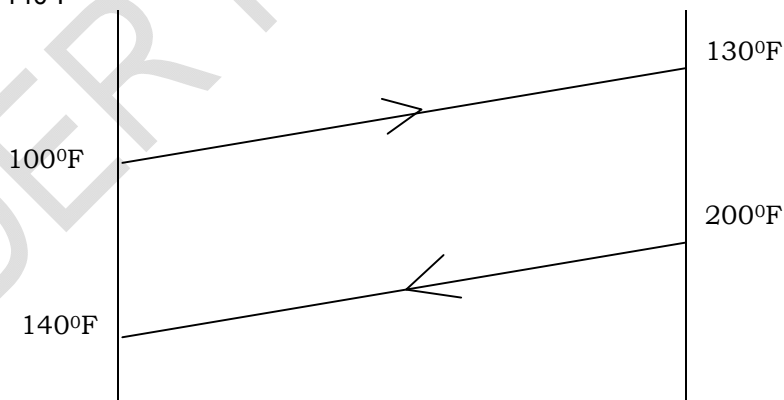
$$M_C = 30000 \text{ lbm/hr}, C_C = 1 \text{ btu/lbm}^\circ\text{F}, T_C = (130 - 100)^\circ\text{F}$$

$$M_H = 150000 \text{ lbm/hr}, C_H = 1 \text{ btu/lbm}^\circ\text{F}, T_H = ?$$

$$\begin{aligned} M_C C_C T_C &= M_H C_H T_H \\ 30000 \times 1 \times (130 - 100) &= 15000 \times 1 \times T_H \\ 900000 &= 15000 T_H \end{aligned}$$

$$T_H = \frac{900000}{15000} = 60^\circ\text{F}$$

$$\Delta T_1 = T_2 - T_H = 200 - 60 = 140^\circ\text{F}$$



$$\text{Mean temperature of water} = \frac{\text{Inlet} + \text{outlet}}{2} = \frac{200 + 130}{2} = 165^\circ\text{F}$$

(Where Inlet temperature = 200°F and outlet temperature = 130°F)

STEP 2: Physical properties of water

Inlet temp.: 200°F, Outlet temp.: 130°F, mean temp.: 165°F, Heat capacity coefficient: 1btu/lbm°F, Thermal conductivity: 0.59 **W/m°C**, Density: 1000kg/m³ = 62.37lbm/ft³ and viscosity of water: **0.8mNs/m²** to be: 0.0008 N.s/m²

STEP 3: Overall coefficient

The overall heat transfer coefficient (μ) is **250 Btu/hr/ft²**

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

- i) $\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{70 - 40}{\ln\left(\frac{70}{40}\right)} = 53.6097^\circ F$
- ii) Correction factor = 0.88
- iii) The temp. difference, $\Delta T_m = \Delta T_{LMTD} \times C_f = 53.6097 \times 0.88 = 47.1765^\circ F$

STEP 5: Heat transfer area

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

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, $p_t = 1.25d_o$ [7]

Where d_o is the tube outside diameter.

Given, $d_o = 0.75\left(\frac{3}{4}\right)$ inch = 19.05mm \cong 0.0625ft

With Triangular pitch, $p_t = 1.25 \times 19.05 = 23.81$ mm pitch \cong 0.0781ft

STEP 7: Number of tubes

From $Q = M_c C_c T_c$, But $M_c = \rho A V$, A = Area, V = Velocity, ρ = density

$\rho_{H_2O} = 1000 \text{ kg/m}^3 = 62.37 \text{ lbm/ft}^3$, $m = 3000 \text{ lbm/hr}$, $r = 1.2 \text{ ft/s}$ and $1 \text{ hr} = 3600 \text{ s}$

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

$$\text{But, } A \text{ (Total flow area)} = \frac{n\pi \delta^2}{4}$$

The Area is the product of the number of tubes and the flow

$A = 0.111$, $\delta = 0.75$ inch = 0.0625ft (since **1.0** inch = 2.54 cm and 1.0 ft = 30.48 cm)

1inch = 2.54 cm

? = 30.48 cm

$$\frac{30.48 \times 1}{2.54} = 12 \text{ inch}$$

$$1 \text{ ft} = 30.48 \text{ cm} = (12 \text{ inch})^2 = 144 \text{ inch}^2$$

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

But the total surface area required for one tube pass exchanger was 67.2ft²

Length of the tube

$$A = \pi \delta L, 67.2 = n \pi \delta L$$

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

The length is greater than the allowable 8 ft so we must use more than one pass

$$q = U A \Delta T_{LMTD} \times \text{correction factor}$$

$$A \text{ (total)} = \frac{q}{U \times \Delta T_{LMTD} \times \text{correction factor}} = \frac{900000}{250 \times 53.6079 \times 0.88} = 76.3 \text{ ft}^2$$

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

$$A = 2n\pi\delta L, L = \frac{A}{2n\pi\delta} = \frac{76.3 \times 12}{2 \times 36 \times 3.142 \times 0.75}$$

Number of tubes per pass = 36, Number of passes = 2

Length of tube per pass = 5.490ft, Area per pass = 36 \times 0.11 = 3.996 ft²

$$\text{Volumetric flow rate} = \text{flow rate} \times \text{Density of } H_2O = 15000 \frac{\text{lbm}}{\text{hr}} \times \frac{1 \text{ ft}^3}{62.7 \text{ lbm}} = 240 \text{ ft}^3/\text{hr}$$

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

STEP 8: Bundle and shell diameter

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

Given, δ_o (tube outside diameter) = 20, $K_1 = 0.249$, $n_i = 2.207$

Using Triangular pitch pattern for two passes

$$N_t = \text{Number of tubes per pass} \times \text{Number of passes} = 36 \times 2 = 72$$

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

For a split-ring floating head exchanger, the typical shell clearance from (from standard value is 52mm)

That is tracing 0.2607 on the graph [7]

$$\text{Shell clearance} = 52 \text{ mm} = 0.17 \text{ ft} \quad (1 \text{ ft} = 30.48 \text{ cm} = 304.8 \text{ mm})$$

$$\text{Shell diameter, } \Delta s = 0.85 + 0.17 = 1.02 \text{ ft}$$

STEP 9: Tube-side heat transfer coefficient

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

$$\rho = 1000 \text{ kg/m}^3, \mu \text{ (from tube side velocity)} = 60.1 \text{ ft/hr} = 0.0051 \text{ m/s}, \delta \text{ (from the question) internal diameter} = 0.75 \text{ inch} =$$

$$0.019 \text{ m and } \mu \text{ (standard viscosity of H}_2\text{O)} = 0.8 \text{ mNs/m}^2 \text{ ????? to be: } \underline{0.0008 \text{ N.s/m}^2}$$

$$Re = \frac{1000 \times 0.0051 \times 0.019}{0.8 \times 10^{-3}} = 121.125 \cong 1.211 \times 10^2$$

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

$$\text{Where } C_p \text{ of H}_2\text{O in S.I. unit} = 4.2 \text{ kJ/kg}^0\text{C}, \mu = 0.8 \times 10^{-3} \text{ and } = \text{Thermal conductivity of H}_2\text{O}, K_f = 0.59 \text{ W/m}^0\text{C}.$$

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

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

$$\text{Where } L = \text{Given length from question: } 8 \text{ ft, } \delta_i = \text{Inside diameter} = 14.88 \text{ mm} \cong 0.0486 \text{ ft}$$

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

Using Reynolds number: 1.2112×10^2 and 164.61 to trace in graph of tube-side heat transfer factor, 9.0×10^{-3} was gotten as standard value in Coulson and Richardson [7] and $j_h = 9.0 \times 10^{-3}$.

(iv.) Nusselt number,

$$Nu = 9.0 \times 10^{-3} \times (Re) \times (Pr)^{0.33} = 9.0 \times 10^{-3} \times (121.125) \times (5.69)^{0.33} = 1.93$$

(v.) Tube side coefficient

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

Step 10: Shell-side heat exchanger coefficient

Calculating for one-shell pass (Triangular pitch)

$$\text{Given, } k_i = 0.319, n_i = 2.142, \delta_o = 20$$

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

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

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

$$\text{Since, } D_s = 560 \text{ mm} \cong 20 \text{ (22.04725) inches} \cong 1.66 \text{ (1.8373) ft}$$

(iii.) Cross-flow Area, A_s

Given, Triangular pitch $\cong 0.0781 \text{ ft}$

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

(iv.) Equivalent diameter, d_e

$$\delta_e = \frac{1.1}{d_o} (P_t^2 - 0.917 d_o^2) = \frac{1.1}{0.0625} (0.0781^2 - 0.917 \times 0.0625^2) = 17.6 (0.00609961 - 0.003582) = 0.044 \text{ ft}$$

(v.) Volumetric flow-rate on shell side

$$\text{Flow rate} \times \text{density of H}_2\text{O (converted)} = 481 \text{ ft}^3/\text{hr}$$

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

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

Given, $\delta = 1000 \text{ kg/m}^3$

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

δ (Equivalent diameter) = 0.044 ft = 0.013 m

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

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

$$h_s = \left(\frac{k_f}{d_e} \right) j_h \times Re \times Pr^{0.33} = \left(\frac{0.59}{0.13} \right) \times 7.5 \times 10^{-3} \times 6048.25 \times 5.69^{0.333} = 3654 \text{ w/ft}^2\text{F} \text{ ??????!!!!}$$

STEP 11: Pressure drop

Tube side (two passes):

(8 and 2.5 are given standard conditions)

$$\Delta P_t = N_p \left[8 j_{ft} \left[\frac{L}{d_i} \right] + 2.5 \right] \frac{\rho_w U_t^2}{2} = 943.03 \text{ lbf/ft}^2 \text{ ??????!!!! to be lbf/ft}^2$$

Shell side:

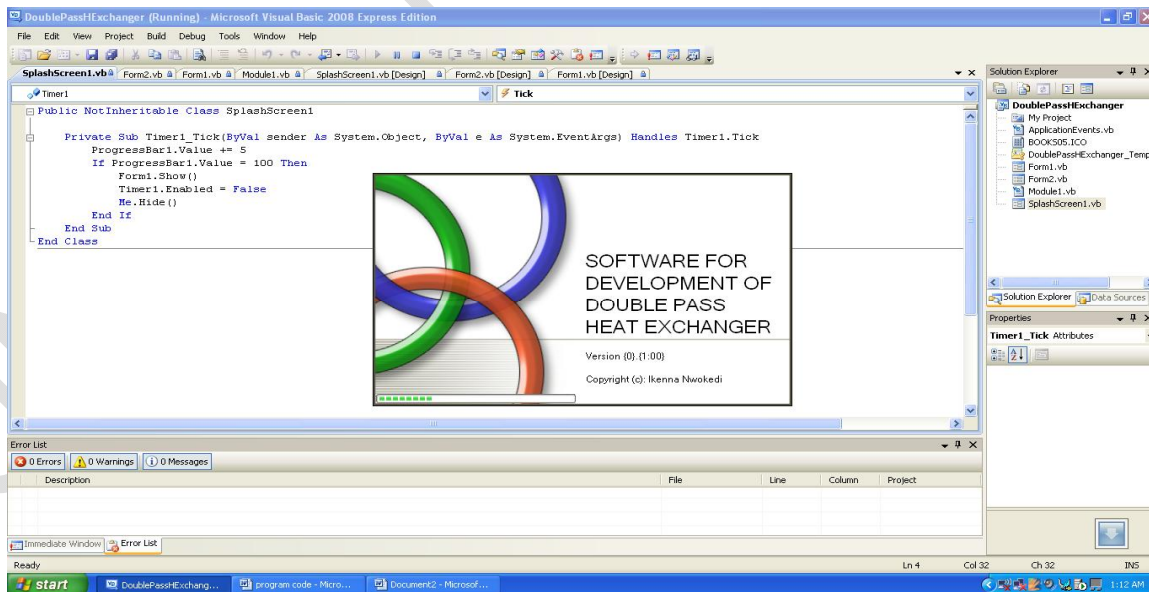
$$\Delta P_s = 8 j_f \left(\frac{D_s}{\delta_e} \right) \left(\frac{L}{L_B} \right) \frac{\rho_m V^2}{2}$$

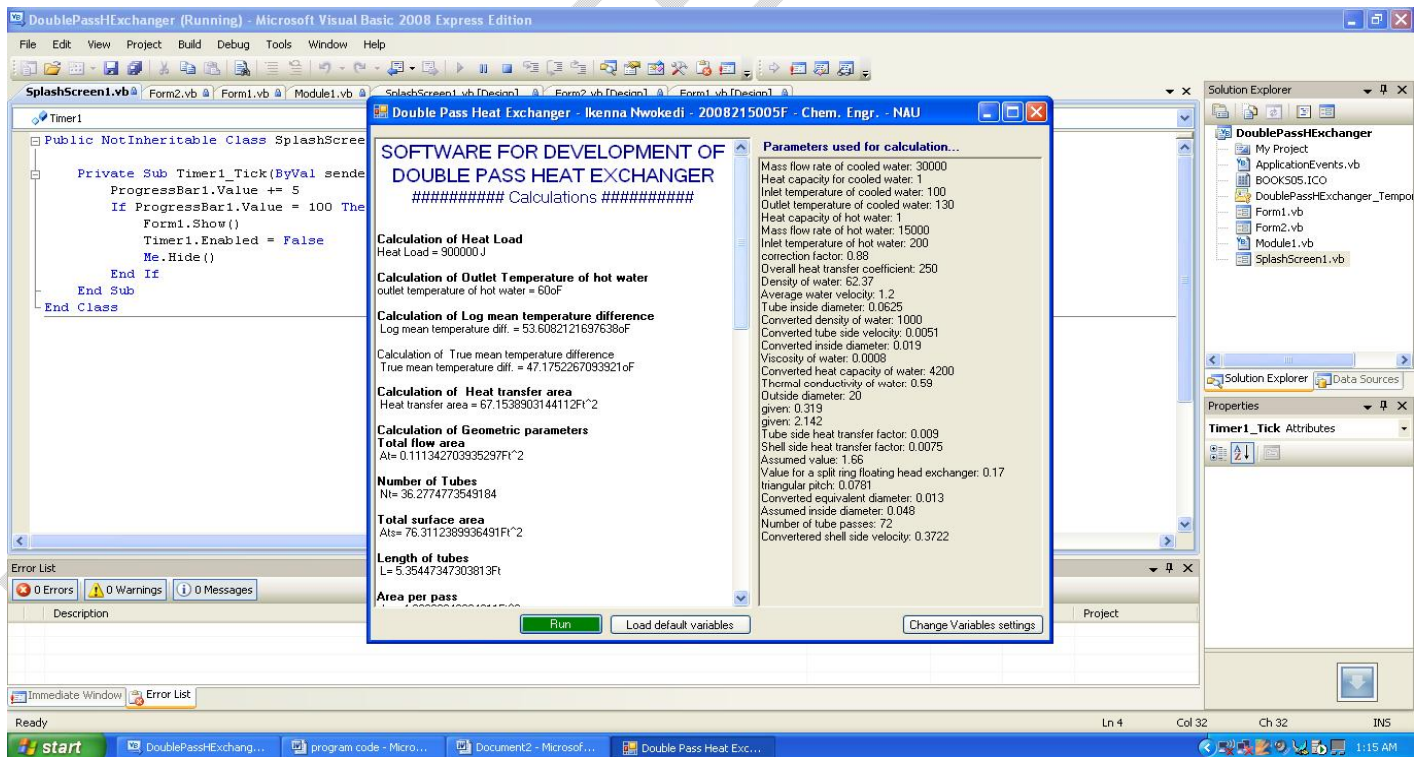
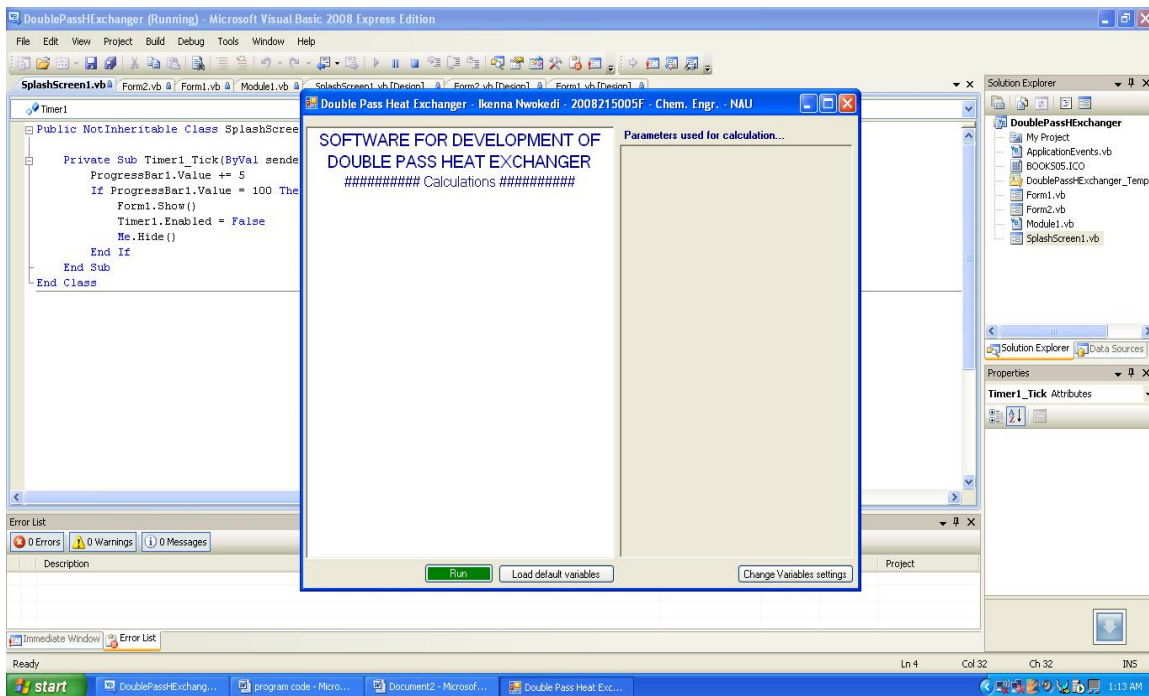
Where D_s = Shell diameter, δ_e = equipment diameter, L_B = Baffle spacing, V = velocity

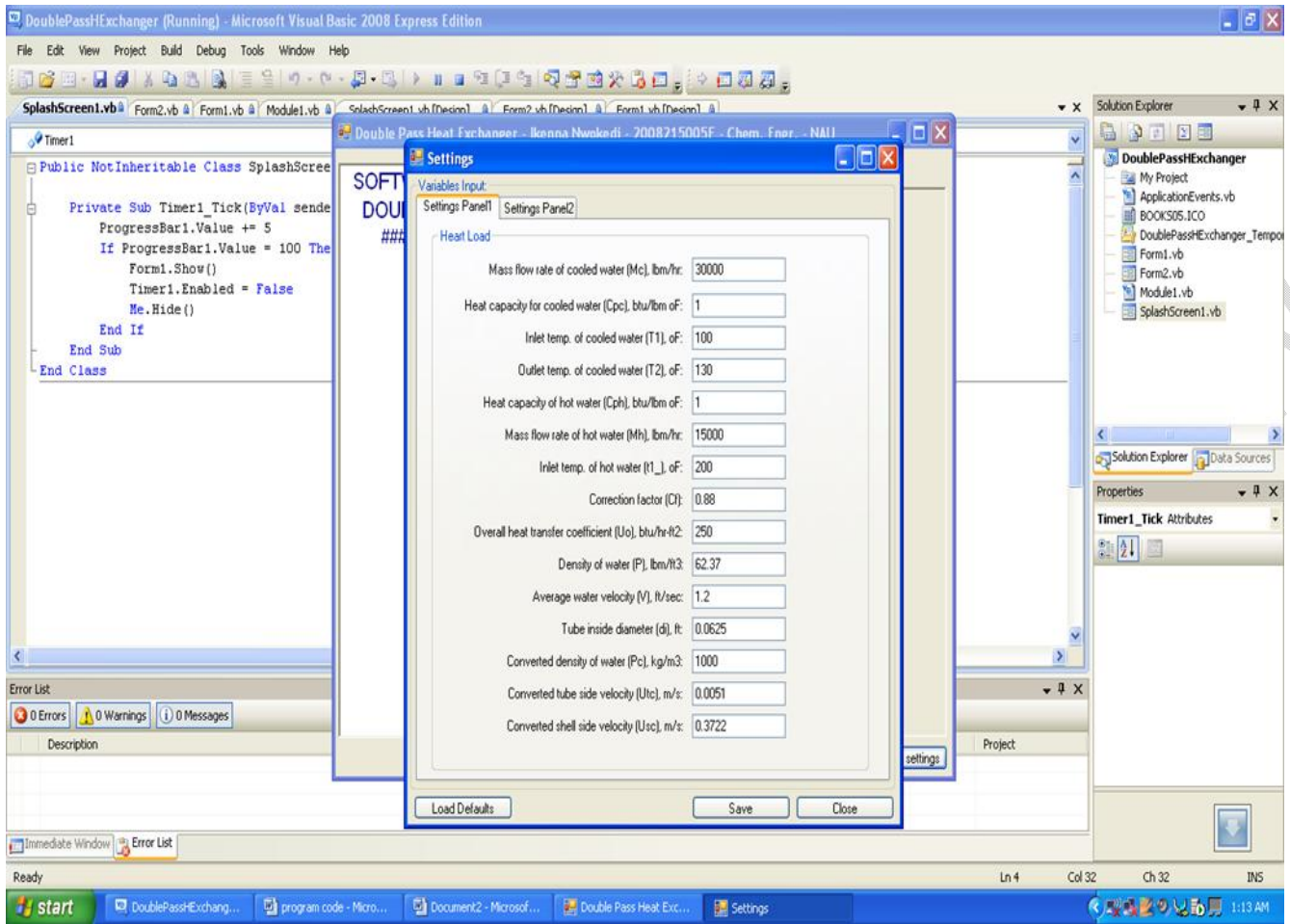
$$\Delta P_s = 8 \times 7.5 \times 10^{-3} \left(\frac{0.99}{0.004} \right) \left(\frac{5.40}{0.33} \right) \frac{62.37 \times 1.2^2}{2} = 1022.08 \text{ lbf/ft}^2 \text{ ??????!!!! to be lbf/ft}^2$$

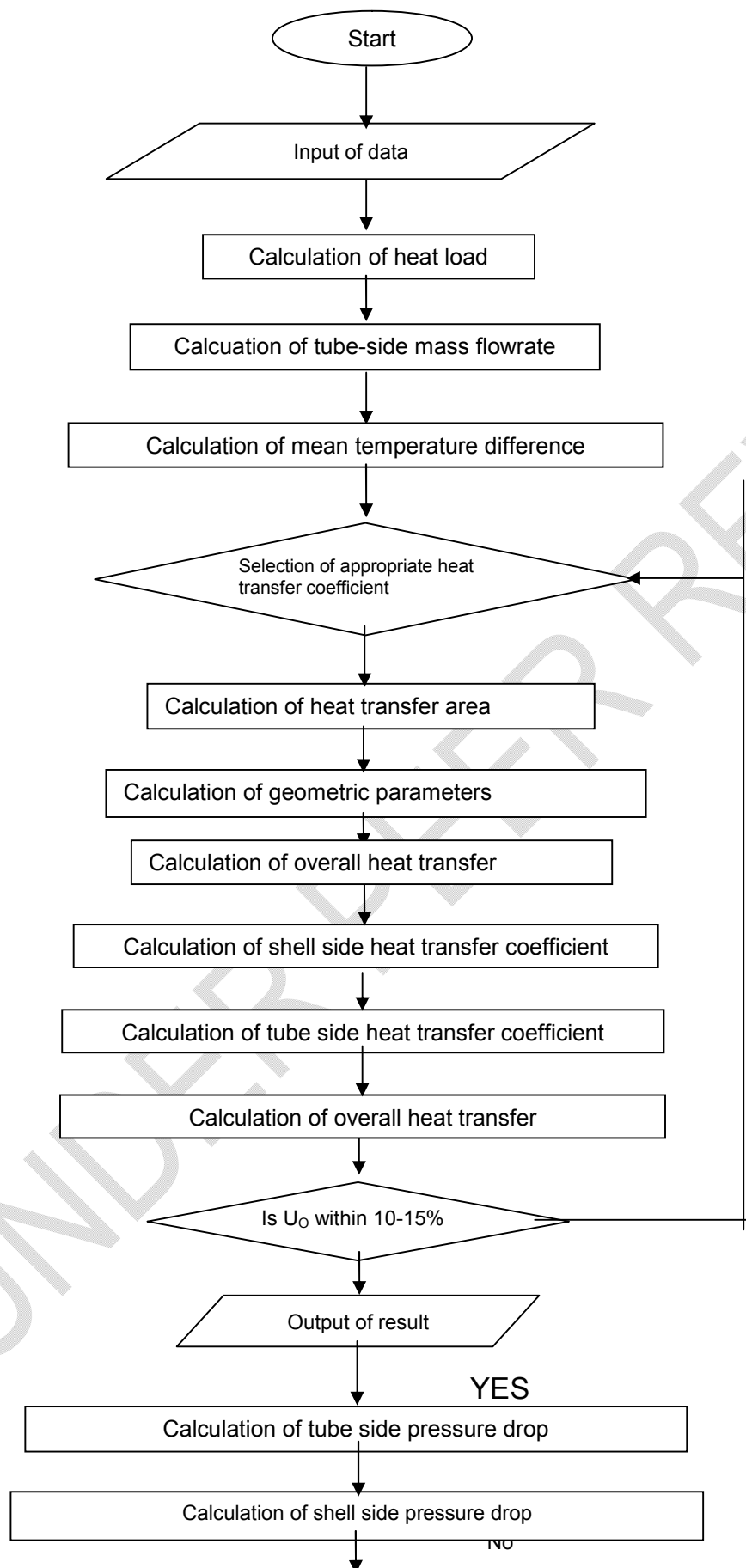
3. RESULTS AND DISCUSSION

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.









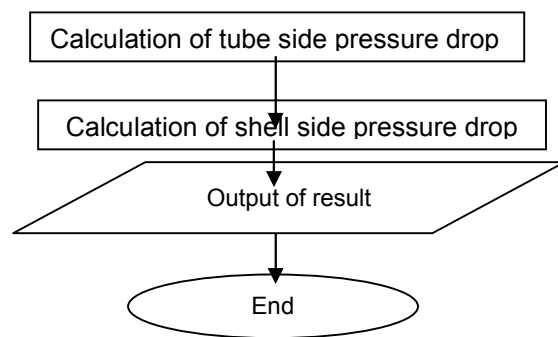


Fig. 1. Flow chart of the design program.

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.

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