COMPUTER AIDED DESIGN MODULE

OF A SHELL AND TUBE HEAT EXCHANGER
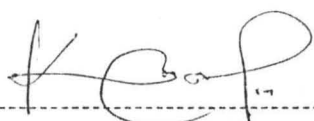
USING KERN'S METHOD.

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A PROJECT SUBMITTED TO THE DEPARTMENT OF CHEMICAL ENGINEERING IN PARTIAL FULFILLMENT OF REQUIREMENTS FOR THE AWARD OF MASTERS OF ENGINEERING (M.ENG.) IN THE SCHOOL OF POSTGRADUATE STUDIES FEDERAL UNIVERSITY OF TECHNOLOGY MINNA. NIGERIA.

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CERTIFICATION.

This is to certify that this project was supervised, moderated and approved by the following underlisted persons on behalf of the Chemical Engineering Department School of Postgraduate studies, Federal University of Technology Minna, Nigeria.

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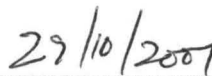External Supervisor   Date.

ii

DECLARATION.

iii

I hereby declare that this project is my original work and has not to my knowledge been ever submitted elsewhere for an award of degree.

Senewo Gbenga.

29/10/2007
Date.

DEDICATION.

THIS WORK IS DEDICATED TO THE GLORY OF GOD WHOSE I AM AND IN WHOM I LIVE AND TO MY LOVING AND WONDERFUL PARENTS MR AND MRS B.F SENEWO.

ACKNOWLEDGEMENT.

ABSTRACT.

Computer Aided Design (CAD) entails the use of a computer system, (a configuration of hardware and software) as a tool for engineering design; the essence being automation leading to increased productivity and product quality. Shell and tube exchangers are the work horses of the process industries, providing a great deal of heat transfer surface in a rugged configuration to meet the requirements of various applications.

In the CAD of a shell tube heat exchanger, a sample problem was taken. The problem was solved using manual calculations and CAD module coded in QBASIC programming language. The major results were found to be identical and listed below:

PARAMETER	CAD	MANUAL
Overall coefficient $W/m^{2o}C$	677	687
Heat Transfer Area (m^2)	311.8	312
Duty (kW)	4338.8	4332
Shell Side Velocity (m/s)	0.5258	0.52
Tube Side Velocity (m/s)	1.3455	1.34
Shell side Pressure Drop (bar)	0.5575	0.55
Tube side pressure drop (bar)	0.6248	0.623
Shell diameter (m)	0.9405	0.944
No of tube passes	4	4
Total number of tubes	1028	1028

Comparison of the results above shows a great similarity between the manual calculation and the CAD program. For this purpose a databank of physical properties was created for the following fluids used in chemical process industries that may be required to undergo a form of heat exchange for certain processes: butane, heptane, hexane, isobutane, methanol, octane, propane, toluene and water.

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ABBREVIATIONS, GLOSSARIES AND SYMBOLS.

a.p.p   Area per pass

BS   Baffle spacing

CAD   Computer aided design.

Cp_c, Cp_h Specific heat at constant pressure for cold and hot streams respectively

De   Equivalent diameter

d_f Diameter under foul condition

di   Inside diameter

F   Temperature correction factor

Fs   safety factor.

j_f Fanning friction factor.

k_f Fouling thermal conductivity

k_t, k_s Thermal conductivities of tube and shell fluids respectively

N_{Gz} Graetz number

N_{Nu} Nusselt number

Np_r Prandtl number

N_{Re} Reynolds number

ΔP_s Shell side pressure drop

ΔP_t tube side pressure drop.

p_T Pitch value

p_w Wetted perimeter

ρ Density of fluid.

R_{fi} Fouling factor based on inside of tube.

R_{fo} fouling factor based on outside of tube.

t.c.s.a   Tube cross sectional area.

TID	Tube inner diameter.
TOD	Tube inner diameter.
ΔT_m	Mean temperature difference.
ΔT_{LM}	Log mean temperature difference.
Tl	Tube length.
ΣR_T	Total thermal resistance along transfer path.
σ	Stefan Boltzman constant.
$U_{ass,}$	Assumed overall coefficient.
U_o	Overall heat transfer coefficient.
U_c	overall heat transfer coefficient under clean conditions.
U_f	overall heat transfer coefficient under foul condition.
u_t, u_s	Velocities of tube and shell fluids.
μ_b, μ_w	Viscosities at mean bulk and wall temperatures respectively.

CHAPTER ONE

1.0   <u>GENERAL INTRODUCTION.</u>

1.1   <u>Definition and Basics of CAD.</u>

Computer aided design (CAD) is a catch phrase for a large variety of technological innovations and applications of computer based technology. It represents a set of methods, procedures, equipment (hardware) and programs (software) involved in design and manufacture of physical objects. A CAD system is a configuration of computer hardware and software used to create a geometric model of a product in computers memory and to display a representation of it on the graphics screen. Computer aided design is the result of applying advanced computer technology to the problems of engineering and production.

The development of CAD has provided the designer with a powerful and flexible tool with which he can, in a matter of hours, prepare and evaluate a preliminary design which otherwise would have taken days or even weeks using the traditional manual methods.

1.2   Fields of Application

CAD has found a wide range of applications in engineering and manufacturing systems. Modern aircrafts and motor vehicles are not presently designed without the aid of the computer neither are VLSI circuits. Artifacts are now being designed at a level of complexity, not previously possible.

CAD systems are moving from large corporations to small and medium-sized firms and a variety of commodities. Electrical, mechanical and nuclear engineering applications of CAD were developed very early. Computer analysis of structures and buildings are quite extensive. In the chemical, petroleum and process industries, computer process control is common. Such industries as shoes, tobacco, textiles, furniture and food processing are now starting to undertake design by CAD.

1.3   HEAT TRANSFER EQUIPMENT.

A device whose primary purpose is the transfer of energy between two fluids is called a heat exchanger. Equipment for transferring heat are used in essentially all the process industries. Heat exchangers are basically classified into 3 categories:

1) **Regenerators:** in which the hot and cold fluids flow alternatively through the same space with as little physical mixing between the two streams as possible.

2) **Open type exchangers:** wherein physical mixing of the two fluids streams actually occurs and

3) **Recuperators:** in which hot and cold fluid streams do not come into direct contact with each other but are separated by a tube wall or surface.

Under this classification are modern heat exchangers ranging from simple concentric pipe exchangers to complex surface condensers. Between these extremes are the conventional shell and tube exchangers which are used extensively in industry.

SHELL AND TUBE EXCHANGERS.

The shell and tube exchangers as the name implies is made up of tube arranged in a shell through which both the process and service fluids pass concurrently or countercurrently. Different configurations like the U-tube, finned tube, fixed tube-sheet, floating head exchangers are available.

The universal popularity of shell and tube heat exchangers is based on their wide range of applicability. i.e.

1. Applications include single phase, boiling and condensation.

2. Any temperature and pressure from vacuum to high (material limits in this case) is possible.

3. Varying sizes from very small to transportation limiting

4. Because of shell type and baffle design, pressure drop can be extremely adjusted within a wide range.

5. Ruggedness.

In contrast to many other types of heat exchangers, shell and tube exchangers are designed more or less on a custom basis with each design consisting of individually selected components. TEMA standards lists a number of shell and tube bundle types that are internationally recognized as representing the basic general design types.

Intelligent selection of heat transfer equipment requires an understanding of the basic theories and principles of heat transfer with methods for design calculations. Determination of appropriate coefficients of heat transfer required for design can either be estimated on basis of past experience or empirical theoretical equations developed by workers in the field.

1.4   DESIGN BY COMPUTER PROGRAMS.

Engineering design deals with structures, objects, devices or machines existing in the physical world. A design process comprises recognizing a problem, having an idea on how to solve it and designing concepts for doing so. As a result of computer technology, it has become possible to provide feedback to the designer about the result of a design decision. Increased automation of manual skills involved in design has eliminated much of the drudgery of tasks, freeing designers' time and mind for the practice of design.

HEAT EXCHANGER DESIGN BY COMPUTERS.

The basic structure is similar to hand based design, the difference being the replacement of the very subtle and complicated human thought process by an algorithm suited to a fast but inflexible computer.

First, the problem must be identified as completely and unambigously as possible: including, stream compositions, flow rates and temperatures as well as likely ranges of variations in these parameters during operation.

The next step is to select a tentative set of exchanger geometrical parameters. The better the starting design the sooner the designer comes to a final design. In either case, initial design will be 'rated' that is the thermal performance and pressure drops for both streams will be calculated for the design. This concept is shown in the Figure 1.1.
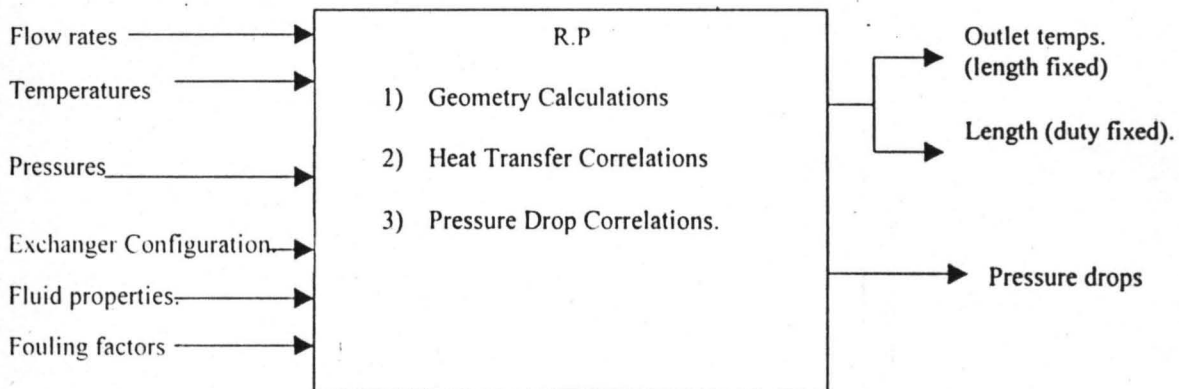
Fig 1.1 Typical Heat Exchanger Rating Program.

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1.5   AIMS & OBJECTIVES

This work is intended to create a CAD module that can be used in the design of a shell and tube heat exchanger. Several correlations, equations and relationships are involved in heat transfer and its associated equipment design. This module aims at coding these equations using the QBasic programming language so as to minimize time as well as the very tedious and iterative calculations involved.

The module is expected to take care of cases where full process specifications are supplied as well as provide guidelines for optimum design where there is a shortfall of specifications required in the design process.

Main results to be obtained are pressure drops and heat transfer coefficients for both streams; overall heat transfer coefficient, duty and the heat transfer area.

1.6   LIMITATIONS

A major limitation encountered in this work is in obtaining adequate and reliable information on the various physical properties of fluids. Also for this work, it has been assumed that the process is a single phase operation hence for cases where there is phase change some correlations may not exactly be applicable. Also charts and figures had to be coded to avoid the necessity of the designer having to read off charts, in some cases, the regressing of such charts proved difficult but so far, correlations have been obtained for virtually all of the relationships encountered in the work. Finally it should be noted that despite the advantages of CAD, design elements specifications, evaluation of results and eventually alternative design decisions are still the exclusive preserve of the engineer.

CHAPTER TWO

LITERATURE SURVEY.

2.1   COMPUTER AIDED DESIGN

2.1.1   THE CAD SYSTEM COMPONENTS

A typical system used for computer aided design application consists of a workstation (hardware a designer uses to interact with a computer) a computer and various peripherals. The workstation includes a graphics device (CRT), an XY input device, a function menu and a keyboard.

Peripheral equipment permits the user a variety of specialized input, output or storage options. The designer can interact with the graphics display by means of devices such as digitizers, menu tablets, light pens. Cursor control can be by joystick, mouse, trackball or thumb wheel. Keyboards are used to enter commands and data. The cursor indicates spatial locations (points). The primary purpose of a CAD system is to create a model of an object which inclusively can be mathematically represented.

2.1.2 ROLES, APPLICATIONS AND BENEFITS.

The main Role of CAD is automation. The replacement of human labor by machine labor as intellectual tasks (comparisons, decisions, reasonings) can now be easily carried out as a result of advances in computer technology. Among the benefits that CAD has brought are:

1. **Increased Productivity:** evident primarily is increased production. More parts can be designed in a specified time. Also because the process leads to fewer errors, less time is spent compensating for mistakes. Overall production costs tend to be lower, and work tends to be of higher quality as people in this environment can do things more quickly thus devoting more time to thinking and improving ideas.

2. **Improved Product Quality:** tendency to produce better products. At the design stage, the engineer has a better understanding of a particular design subjecting it to a variety of analyses, often evaluating it in comparison with several alternatives designs for the same concept.

3. **New Concepts:** new conceptions of what is possible. As a result of computer technology, conceptions of what is possible to do in design are changing, this permits a cost effective design and fabrication of unique products.

2.1.3 CONCEPTS OF MODULARITY.

The sheer complexity of many systems demands an orderly design approach. Modular design is simply the tendency to design in modules. Modular design consists of a set of modules and a set of functions. Modular design is a hierarchal process. To take an example from current computer technology, integrated circuit chips are interconnected to make boards, boards are plugged into black panels to make a system unit and system units are plugged into cabinets to form a system (Siewiorek, 1992).

Some advantages of modular design are:

1) Reduction in development time by allowing the design task to be partitioned making better use of resources (time and manpower)

2) Increased flexibility by allowing the alteration of specifications and redesign of modules.

3) Comprehensibility by allowing the students of the system concentrate on well defined pieces of final object.

4) Maintainability by allowing identification of faulty modules which can be replaced or repaired.

2.1.4 Databases in CAD

A database is a collection of records containing data with associated indexes, stored in a databank containing numeric or factual data (Sieworek, 1992).

Information retrieval is the process of selecting information from a store in response to a query. As design procedures require a regular supply of information, retrieval of such information is becoming increasingly dependent on the computer.

CAD systems retrieve wanted design data from store to feed it into design programs. Hence generally the value of databases in CAD systems is of immense benefit, culminating in using deductive inference to draw conclusions from data retrieval.

2.2   HEAT TRANSFER.

2.2.1   Fundamentals of Heat Transfer.

There are three modes of energy transfer. Conduction, convection and radiation. All heat transfer processes involve one or more of these modes.

Conduction: is primarily a molecular phenomenon. Energy transfer is accomplished by either molecules interaction in which molecules at a higher energy level impart same to adjacent molecules at lower levels or by 'free' electron mechanism significant primarily in pure metallic solids. A general relation for the heat transfer is

$$q_x / A = -kdT / dx \quad \text{------------------- (2.1)}$$

where q_x = heat transfer rate in x direction (W).

A = area normal to direction of heat flow (m^2)

dT / dx = temperature gradient in x direction (K/m)

k = thermal conductivity of material (W/mK).

The negative sign indicates that heat flow is in the direction of a negative temperature gradient.

Convection: Heat transfer due to convection involves energy exchange between a surface and an adjacent fluid. Where an external factor say a pump causes flow of fluid past a solid surface, forced convection is said to occur where as in natural or free convection warmer (or cooler) fluid next to a solid boundary causes circulation due to density difference. The rate equation for convective heat transfer known as the Newton rate equation is

$$dq = h_x \ \Delta T dA \ \text{----------------------------- (2.2)}$$

Where q = rate of convective heat transfer (W)

A = area normal to direction of heat flow (m^2)

ΔT = temperature difference between surface and fluid (K)

h_x = local heat transfer coefficient (W/m^2K).

Radiation: Radiant heat transfer differs from conduction and convection in that no medium is required for propagation and is best infact under a perfect vacuum. The rate of energy emission from a perfect radiator (black body) is given by

$$q / A = \sigma T^4 \ \text{----------------------} \ (2.3)$$

Where q = rate of radiant energy emission (W)

A = area of emitting surface (m^2)

T = absolute temperature (K)

σ = Stefan-Boltzman Constant (5.676×10^{-8} W/m^2K^4)

Combined mechanisms of heat transfer.

It is rare in actual situations for only one mechanism to be involved in the transfer of energy. With respect to the means of calculating heat transfer rates when several modes are involved:

$$q_x = \ \Delta T / \ \Sigma R_T \ \text{---------------------------} \ (2.4)$$

where ΣR_T = total thermal resistance along transfer path (K/W)

Or

$$q_x = U A \Delta T \ \text{-------------------------------}(2.5)$$

Where U = overall heat transfer coefficient.

2.2.2 Basic Equations in Design

From the first law of thermodynamics change in enthalpy of one fluid is an open system under steady state conditions with negligible potential and kinetic energy changes is given by

$$\delta Q = mdi \text{ --------------------(2.6)}$$

> where m = mass flow rate (kg/s)
> i = specific enthalpy (J/kg)
> δQ = heat transfer rate (W).

Integration of (2.6) yields
$$Q = m(i_2 - i_1) \text{ ------------------(2.7)}$$

> Where i_1 and i_2 = initial and final enthalpies of fluid stream.

For negligible heat transfer between exchanger and surroundings (adiabatic process) integration of (2.6) for hot and cold fluid yields

$$Q = m_h(i_{h1} - i_{h2}) \text{ ----------------------- (2.8)}$$

$$Q = m_c(i_{c2} - i_{c1}) \text{ ----------------------- (2.9)}$$

1 and 2 refer to inlet and outlet while c and h refer to cold and hot fluid respectively.

With no phase change and at constant specific heats,

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$$di = CpdT \text{ -----------------------}2.10$$

so that (2.8 & 2.9) become

$$Q = (mCp)(T_{h1} - T_{h2}) \text{ ------------------------}(2.11)$$

$$Q = (mCp)(T_{c2} - T_{c1}) \text{ ------------------------}(2.12)$$

Since temperature difference between hot and cold fluids varies with position in the exchanger, it is convenient to establish an appropriate mean value of the temperature difference between the hot and cold fluids, hence we arrive at

$$Q = UA\Delta Tm \text{ ----------------------------} (2.13)$$

Where A = total heat transfer area (m^2)

U = average Overall heat transfer coefficient (W/m^2K).

ΔTm = function of T_{h1}, T_{h2}, T_{c1}, T_{c2}

2.2.3   The Overall Heat Transfer Coefficient.

The overall heat transfer coefficient for a single smooth and clean plain wall (Kakac, 1991) is written as;

$$UA = 1 / R_t = 1 / [1/h_iA_i + t/kA + 1/h_oA_o] \text{ --------------------} (2.14)$$

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Where R_t = total thermal resistance to heat flow (K/ W)

t = wall thickness (m).

k= wall thermal conductivity (W/mK).

h_i & h_o = heat transfer coefficient for inside and outside flow respectively (W/m^{2}K).

Table 2.1

Typical Overall Coefficients for shell & Tube exchangers (Sinnot, 1993).

Hot Fluid	Cold Fluid	U (W/m^{2o}C)
Water	Water	800 – 1500
Organic Solvents	Organic Solvents	100 – 300
Light Oils (organics)	Light Oils (organics)	100 – 400
Heavy Oils (organics)	Heavy Oils (organics)	50 – 300
Gases	Gases	10 – 50
Organic Solvents	Water	250 – 750
Light Oils (organics)	Water	350 – 900
Heavy Oils	Water	60 – 300
Organic Solvents	Brine	150 – 500
Water	Brine	15 – 250

Table 2.2   Classification of Organics by Viscosity.

Term	Viscosity Pa.s (Ns/m^2)
Light Organics	Less than 0.0005
Medium Organics	0.0005 – 0.001
Heavy Organics	Greater than 0.001

For a clean tubular exchanger, the overall heat transfer coefficient is

$$U_o A_o = U_i A_i = 1/R_t = 1/\left[1/h_i A_i + \ln(r_o r_i)/2\pi kL + 1/h_o A_o\right] \text{---------------(2.15)}$$

Since exchanger walls are of bimetallic nature, coupled with layers of oxidation, crust deposit, additional thermal resistances, in the heat flow path are introduced. The total thermal resistance R_T to heat transfer across a heat exchanger can be expressed as

$$R_T = 1/UA = 1/U_o A_o = 1/U_i A_i = 1/h_i Ai + Rw + R_{fi}/A_i + R_{fo}/A_o + 1/A_o h_o \text{ ---(2.16)}$$

Wall resistance, $Rw = t/kA$ (for a plane wall)---------------(2.17)

$\ln(r_o/r_i)/2\pi kL$ (for a tube wall)-----------------(2.18)

Calculation of an overall heat transfer coefficient depends on whether it is based on the cold or hot side surface area since, $U_o \neq U_i$ if $A_o \neq A_i$. Therefore for the unfinned tubular heat exchanger, the overall heat transfer coefficient is expressed as

$$U_o = 1/[r_o/r_i (1/h_i) + r_o/r_i Rfi + [r_o ln(r_o/r_i)]/k + R_{fo} + 1/h_o] \text{-----------------}(2.19)$$

2.2.3.1 LMTD Method of Heat Exchanger Analysis.

The rate of heat transfer δQ from the hot to the cold fluid across the heat transfer area dA may be expressed as

$$\delta Q = U(T_h - T_c)dA \text{------------------------------------}(2.20)$$

For counterflow;

$$d(T_h - T_c) = dT_h - dT_c = \delta Q(1/C_c - 1/C_h) \text{--------------------- } (2.21)$$

Substituting the value of (2.20) into (2.21);

$$d(T_h - T_c)/(T_h - T_c) = U(1/C_c - 1/C_h)dA \text{-------------------------- } (2.22)$$

Upon integrating 2.22 (with U, C_h & C_c constant along the entire length of the heat exchanger);

$$\ln [T_{h2} - T_{c1}/T_{h1} - T_{c2}] = UA(1/C_c - 1/C_h) \text{ ----------------------- (2.23)}$$

or

$$T_{h2} - T_{c1} = (T_{h1} - T_{c2}) \exp[UA(1/C_c - 1/C_h)] \text{ --------------------- (2.24)}$$

Substituting (2.11& 2.12) into (2.24) and solving for Q;

$$Q = UA[(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})]/\ln[(T_{h1} - T_{c2})/(T_{h2} - T_{c1})] \text{ ------------------ (2.25)}$$

Or

$$Q = UA[(\Delta T_1 - \Delta T_2)/(\Delta T_{1/}\Delta T_2)] \text{ ---------------------------------- (2.26)}$$

Where ΔT_1 is temperature difference between 2 fluids at one end

$\quad \Delta T_2$ is temperature difference between 2 fluids at other end.

Hence over the entire length

$$\Delta T_{LM} = [(\Delta T_1 - \Delta T_2)/\ln(\Delta T_1 / T_2)] \text{ -------------------------------------- (2.27)}$$

This is called the log mean temperature difference (LMTD), so that

$$Q = UA \, \Delta T_{LM} \text{ --- (2.28)}$$

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Accordingly the total heat transfer rate between hot and cold fluids for a single pass flow arrangement is determined from equation (2.28).

For heat transfer analysis of crossflow and multipass exchangers a correction factor, F, is introduced so that (2.28) becomes

$$Q = UAF\Delta T_{LM} = UA\Delta T_{m} \text{----------------------------- (2.29)}$$

F is correction factor and is less than 1 for crossflow and multipass arrangement. It represents the degree of departure of the true mean temperature difference from the LMTD for a counterflow arrangement.

2.2.3.2   Heat Exchanger Pressure Drops.

Pressure drop available to drive fluids through an exchanger may be set up by process conditions while others are set by economic analysis. Fluid friction effects are especially responsible for pressure drops of fluids flowing through a system and hence dictate pumping power or fan work input necessary for upholding flow.

Table 2.3   Relationship Between Fluid Viscosity and Allowable Pressure Drop (Sinnott, 1993).

Viscosity (Ns/m^2)	Allowable Pressure Drop (Bar)
<0.0001	0.35
0.0001 – 0.001	0.5 – 0.7

2.2.3.2.1   Tube Side.

There are two major sources of pressure loss on this side, these are friction loss in tubes and losses due to sudden contraction and expansion as well as flow reversals (Sinnott, 1993). Existing empirical data on the pressure drop within round pipes show a correlation between a fanning friction factor, f, and the Reynolds number for both laminar and fully developed turbulent flows. Graphs are available for the friction factor vs NR_e but correlations which were developed are useful for computer analysis of heat exchangers.

2.2.3.2.2   Shell Side

Flow pattern in the shell of a segmentally baffled heat exchanger is complex making pressure drop determination a little more difficult. Actual flow of the mainstream of fluid is a mixture of crossflow between baffles coupled with axial (parallel) flow in the baffle windows. To obtain the pressure drop on the shell side, Kern's method is used which evaluates the shell-side equivalent diameter (for either square or triangular pitch

arrangements). Friction factors are correlated in a similar manner to those for tube-side flow using the hypothetical shell velocity and diameter

Equivalent diameter, de, for square pitch arrangement, is determined by

$$de = 1.27/d_o \, (Pt^2 - 0.785d_o^2) \, \text{-----------------------(2.30)}$$

For triangular pitch:

$$de = 1.10/d_o \, (Pt^2 - 0.917d_o^2) \, \text{------------------------(2.31)}$$

The different correlations are tabulated in table 2.4

Table 2.4   **Turburlent Flow Isothermal Fanning Friction Factor correlations for smooth circular Ducts (Kakac, 1987)**

Reference	Correlation	Limitation
Blassius	$f = 0.0791 \, Re^{-0.25}$	$4 \times 10^3 < Re < 10^5$
Drew, Koo & McAdams	$f = 0.0014 + 0.125Re^{-0.32}$	$4 \times 10^3 < Re < 5 \times 10^6$
Karman and Nikuradse	$F = (3.64 \log_{10} Re \, 32.28)^{-2}$	
Approximate Karman & Nikuradse	$f = 0.046Re^{-0.2}$	$3 \times 10^4 < Re < 10^6$

Non-Circular Cross Section.

A duct of non-circular cross section does not relate performance, of geometrical shapes (Kakac, 1991). However, f may be evaluated from the data for circular ducts of di replaced by an equivalent diameter de, defined by

$$De = 4A / P_w = 4 \text{ (flow Area) / Wetted perimeter.} \text{-------------}(2.32)$$

The transition Reynolds number for non-circular ducts is also found to be approximately 2300 as for circular ducts.

2.2.4   HEAT EXCHANGER FOULING.

Fouling can be defined as undesirable deposits or accumulations of undesirable substances on a surface resulting in deteriorating performance of the heat exchanger. As a results of the increase in thermal resistance. The heat exchanger is not able to effect the desired change in thermal conditions of the process streams. Sometimes, this degrading performance deteriorates to an extent of withdrawing the equipment from service either for cleaning or replacement.

2.2.4.1.1   Effects on Heat Transfer.

Inside fouling deposit

Tube wall

Outside fouling deposit.
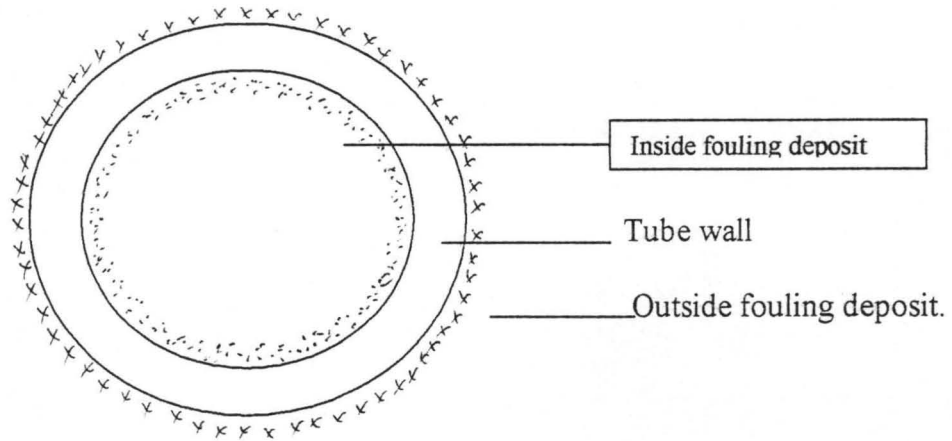
Fig 2.1 Fouling of exchanger surfaces.

As depicted in figure 2.1, fouling adds an insulating layer to heat transfer surface. Heat transfer in the unwanted material takes place by conducting, therefore the fouling factor can be related to the fouling thermal conductivity K_f and fouling thickness t_f by

$$R_f = t_f / k_f \quad - \quad \text{for a plane wall} \text{-----------------------------(2.33)}$$

$$R_f = di \ \ln (di/df)/2k_f \quad - \quad \text{for a cylindrical.------------------(2.34)}$$

The total fouling resistance is given by

$$R_{ft} = A_o R_{fi} / A_i + R_{fo} \text{--}(2.35)$$

And heat transfer rate under fouled conditions expressed as

$$Q_f = U_f A_f \Delta Tm_f \text{--}(2.36)$$

Where subscript f refers to fouled conditions.

2.2.4.1.2   Effects on Pressure Drop

Heat exchangers have been removed from service more for **excessive pressure drop than** for inability to meet heat transfer requirements (Kakac, 1991). **Change resulting in flow** geometry affects the flow field and pressure drop for example in a tubular **heat exchanger** the fouling layer roughens surface. decreases the inside diameter and **increases outside** diameter of tubes; all these result in an increase pressure drop. **Excess surface area is** required to achieve the desired heat transfer under fouled conditions, resulting in an increase in pressure drop.

Table 2.5   Velocity Specification for shell and Tube Heat Exchangers (Sinnott, 1993).

Tube Side Velocity (m/s).	Shell Side (m/s).
Process fluids: 1-2	0.3-1
4.0 max	

2.2.4.1.3   Cost.

Additional cost due to fouling is unavoidable. This comes in the form of

✓ Maintenance cost

✓ Loss of production due to downtime

✓ Energy losses

✓ Increased capital expenditure.

Online and offline cleaning add to the maintenance cost as normal scheduled time incurred in repairs is increased. Loss of production as reduction in capacity, downtime and energy losses due to reduction in heat transfer and increase in pumping power requirements all become a major contribution to fouling cost.

2.2.4.2   Aspects of Fouling

A significant solution to the prediction and mitigation of fouling has been of major concern. Fundamental aspects of fouling are looked into with a view to developing means to predict fouling.

2.2.4.2.1   Categories of Fouling

It is best to classify fouling according to the principal processes that result in it. It is here classified according to :

Particulate fouling:

The accumulation of solid particles suspended in the process stream onto the heat transfer surface. This covers a wide range of materials (organic or inorganic) sizes and shapes.

Crystallization fouling:

Arising primarily from the presence of dissolved inorganic salts in the process stream which exhibit super saturation during heating or cooling. This problem is majorly encountered in cooling-water systems prove to crystal deposition because of presence of salts. This deposits result in a dense well bonded layer referred to as scale or a porous soft layer described as a soft scale, sludge or powdery deposit.

Corrosion fouling:

Exposure to a corrosive fluid may react producing corrosion products which foul the surface provided the P^H value of the fluid is not such that it dissolves corrosion products as they are formed. Corrosion products may also be swept away from the surface where they are produced and transported to other parts of the system.

Biofouling:

Deposition and / or growth of material of biological origin on heat transfer surface including microorganisms (eg. Bacteria, algae, molds) seaweed, water weeds, barnacles from macrobial fouling.

Chemical Reaction Fouling

Chemical reaction fouling is due to chemical reactions within the process stream. The surface does not participate in the reaction although it may act as a catalyst. Polymerization, cracking and coking of hydrocarbons are prime examples.

2.2.4.3 DESIGN OF SUBJECT FOULING

Most process and service fluids will foul the heat transfer surface in an exchanger to a greater or less extent. The deposited material will normally have a relatively low thermal conductivity reducing the overall coefficient. The effect of fouling is allowed for in design by including the inside and outside fouling coefficients.

2.2.4.3.1 Providing a Fouling Allowance

A number of different approaches are used to provide an allowance for fouling all of which result in an excess surface area for heat transfer. These include specifying the fouling resistances, the cleanliness factor or the percentage over surface.

$$CF = U_f/U_c \text{ ------------------------------(2.37)}$$

U_f = overall heat transfer coefficient under fouled condition.

U_c = overall heat transfer coefficient under clean conditions.

and for fouling resistance

$$R_{ft} = (1 - CF)/U_cCF. \text{---------------------------------(2.38)}$$

2.2.4.3.2 Operation of Heat Exchangers Subject to Fouling.

Specification of excess surface area in the heat exchanger because of anticipated fouling leads to operational problems that may accelerate fouling build up.

Operating at design water flow rate, if water flow rate is maintained at the design value, the heat transfer rate will be highest when the surface is clean. Under clean conditions the heat exchanger provides 42% more heat transfer resulting in a water exit temperature 8°C above the design value (Kakac, 1991). As fouling builds up the heat transfer rate as well as the exit water temperature decreases.

Operating at Constant Heat Transfer Rate.

Because of anticipated fouling. the heat exchanger is designed to provide 78% more surface area at the start of operation. Therefore to achieve the design heat transfer when the exchanger is placed in operation under clean conditions, the water flow rate (hence velocity) needs to be adjusted.

2.2.4.3.3 Fouling Control.

A number of strategies are available for controlling fouling. Additives acting as fouling inhibitors can be used during operation. In cases where fouling in unstoppable cleaning becomes inevitable. This can be done on or offline.

Table 2.6 Techniques to control fouling

On-line Techniques	Off –line Techniques
Use and control of appropriate additives	Disassembly & manual cleaning
Inhibitors	Lances
Antiscalants	Liquid jet
Dispersants	Steam
Acids	Airjet
Online cleaning	Mechanical cleaning
Sponge balls	Drills
Brushes	Scrapers
Sonic horns	Chemical cleaning
Soot blowers	
Chain & scrapers	
Thermal shock	
Air bumping	

2.2.5   FLUID ALLOCATION AND PHYSICAL PROPERTIES.

Majors factors involved in determining the best location for fluids in a heat exchanger are the fouling and corrosion characteristics of the fluids; pressure drop across the unit, materials cost and general physical characteristics of the fluids and the exchanger. Of note however is the consideration given to fouling/dirt factors (where all other factors are equal) in which case the dirtier is placed inside the tubes. This is because it is easier to clean inner side of tubes than external.

As for the physical properties of the fluids the major one considered during heat exchanger design/analysis are the viscosity, density, specific heat capacity and thermal conductivities. All these are evaluated at the mean bulk temperature of each stream.

2.2.6   Forced Convection Correlations for Single phase of heat Exchanger.

A large number of experimental and analytical correlations are available for the heat transfer coefficient and the flow friction factor for lamina and turbulent flows through ducts. The correlations representing the Colbourn j factor for the heating and cooling of fluids inside tubes are given below:

Laminar flow:

Low flow is experienced at Reynolds number less than 2100 (Perry, 1997) curves for N_{Re} < 2100 are based on

$$N_{NU} = 1.86 \, N_{az}^{1/3} \, (\mu_b/\mu_w)^{0.14} \text{-----------------------------(2.39)}$$

Where N_{az} = Graetz number.

Transition region:

The transition region lies in the range $2000 < N_{Re} < 10,000$. No simple equation exists for accomplishing a smooth mathematical transition from laminar to turbulent flow.

However, of the relationships proposed, Hausenn's equation (Perry, 1997) fits, hence curves for N_{Re} between 2100 & 10,000 are expressed by

$$N_{NU} = 0.116(N_{Re}^{2/3} - 125)N_{Pr}^{1/3}[1 + (D/L)^{2/3}] \, (\mu_b/\mu_w)^{0.14} \text{-------------(2.40)}$$

D = tube inner diameter

L = tube length

μ_b, μ_w = viscosity evaluated at bulk and wall

temperatures.

Turbulent Flow:

Fully developed turbulent flow exists at $N_{Re} > 10,000$. The curves for N_{Re} above 10,000 are represented by

$$N_{Nu} = 0.023\, N_{Re}^{0.8} N_{pr}^{1/3} (\mu_b/\mu_u)^{0.14}$$

2.2.7 Shell and Tube Heat Exchangers in Process Industries.

Shell and tube heat exchangers play an essential role in chemical processes, generally, they bring feed streams to the proper temperature for reactors, provide for vapor and liquid reflux streams for separation and purification steps and cool products for storage and shipping. Shell and tube heat exchangers come in various shapes and configurations such as the U-tube, fin-tube, fixed tube sheet and floating head exchangers all suited for various purposes.

Shell and tube exchangers permit operation over a wide range of conditions without too much of instability, they're easily maintainable, allowing cleaning of transfer surface and replacement of gasket, tubes and any other components that may fail or deteriorate during service. Shell and tube heat exchangers are the work horses of the process industries because they provide a great deal of heat transfer surface in a mechanically rugged configuration and offer so much designed flexibility to meet the special requirements of a particular application. They are commonly designed to operate at pressures up to 200 atm (20 MPa) or temperatures to 650°C with special designs going higher (Kenneth, 1992).

CHAPTER THREE

METHODOLOGY

3.1 DESIGN PROCEDURES

The overall design of heat exchangers comprises a variety of problems that involves both quantitative and qualitative information and interpretation of results. Design method ensures that the preceding items such as process specifications preliminary problem analysis detailed thermohydraulic design, mechanical design e.t.c. be solved in parallel, that is, the demands of any one discipline being considered with others in an interactive process (Soler, 1986). For example, where heavy fouling is suspected, the result is increased cost through larger surface and possibly more expensive materials as well as costs of cleaning control and production interruption. Therefore all other design aspects become subservient to the solution of minimization of fouling problem. This atimes poses contradictory demands on the overall design. The situation calls however for a weighted compromise frequently requiring the evaluation of alternative designs to optimize the overall effects.

3.2 PROCESS SPECIFICATION

The major roles of the process engineer is to supply information needed for proper design such as:

1) The type of heat exchanger

2) Temperature and pressure of both streams (with acceptable range of seasonal temperature fluctuations if any)

3) Fluid flow rates and compositions.

4) Corrosivity of fluids and environmental hazards if any.

5) The permissible pressure drop (often of great importance)

6) Fouling factors of both fluids.

7) Tube diameter and length.

3.3 Preliminary Problem Analysis

A preliminary analysis of any heat exchanger design problem is always recommended. This includes:

1) Approximate size: To do this, the mean temperature difference should be at hand, while the heat transfer coefficients can be estimated from literature sources. This serves as a guide for subsequent decisions as well as justification for allocation of engineering effort.

2) The determination of distribution of thermal substances.

3) Identification of the most restrictive considerations. This focuses attention on:

   (a) Thermal profile – e.g. Use of counterflow because of close temperature approach or substitution with multitube pass.

   (b) When fouling is a substantial resistance, putting in place defensive measures (flow velocity, material selection etc.) and ease of cleaning.

(c) Pressure distribution and fluid allocation as a result of corrosivity and dirt factors.

4) Identification of probable alternative designs.

5) Inadequate or poor specifications. It is of utmost importance to consider if the given specifications are suspect of being incorrect or at least requiring additional information.

3.4   Thermohydraulic / Mechanical Design Aspects.

The reason for the heat exchanger design is to obtain a global optimum within all demands and restrictions with respect to initial operating cost, reliability and maintenance ease (Kakac, 1991). Some representative items are here presented below:

1) The selection of shell type, baffle design, allocation of fluid to shell and tube are primary decisions. Use is made of the results of the primary design estimates. Shell type and bundle construction will be determined from consideration of:

(a) The lowest cost shell being the U-tube, followed by fixed tube sheet construction. If fouling is no problem or cleaning can be done by chemical means and flow-temperature conditions permit two or more tube passes, the U-tube will be the prime choice. If tube side brush cleaning is required and / or a single tube pass is a consideration because at counterflow, the fixed tube sheet will be preferred unless excluded by restrictions of thermal expansion.

(b) If the stream temperatures and heat transfer coefficients result in large differences between the temperature of shell and tubes, the differential thermal expansivity must be accommodated otherwise the tube-tube joints will be damaged. Again the U-tube is the prime choice here. Fixed tube sheet design can accommodate only minor expansion stresses. For large differences of wall temperature (>50°C approx.) "floating head" construction should be used.

(c) For shell type selection the TEMA J and X shells will produce a greatly decreased pressure drop compared to E shell.

2) Selection of possible alternatives of construction elements such as tube diameter, length, layout, pitch and surfaces.

3) Pressure – drop - under no condition should a design be "pressure drop limited". All previously mentioned design elements may enter as optimization parameters but only few are usually controlled by the computer program logic (e.g. baffle spacing, tube passes) thus requiring human intervention for all others.

4) Problems of maximization of fouling as well as ease of cleaning. These arrangements can include a variety of items ranging from the obvious (flow velocity, baffle design, materials) to the more complex such as water treatment quality and its fluctuations, fluid storage history, possible contaminants, etc.

5) Tube vibration problem.

Mechanical

Some of the most costly errors in heat exchanger design are committed in the area of selection of the mechanical design elements (Yokell,1990) such as:

1) The selection of proper materials for wetted and rewetted surfaces of the heat exchanger. This analysis includes consideration of corrosion, life expectancy, fouling mitigation and numerous other problems. Not always is the least expensive material the most economical. For example, stainless steel tube will increase the initial cost and may help reduce fouling and thus eliminate the need for tube replacements.

2) Compliance with an often bewildering array of national and international codes frequently confusing and contradictory in specifications.

3) General mechanical design integrity. Tube sheet design, tube-tube sheet joints, flanges, welds etc.

4) Aspects of tube vibration are intimately connected with thermal design per se (baffle spacing) and may require special construction design provisions (U- tube support plates, flow distributors etc.)

3.5 Programming Language.

BASIC – the most widely used programming language for microcomputers was developed as an easy to learn language for teaching programming to beginners (Graham,1982). Modern versions such as Q-BASIC, Visual Basic and Turbo have been extended to the point where they are suitable for a wide range of

programming tasks. Q-basic allows build up of complex programmes out of simple parts called modules, each module performing a limited, specific and clearly defined function in basic. These modules are either functions or subroutines with subroutines being more common because of the somewhat limited nature of function definitions in BASIC.

3.6
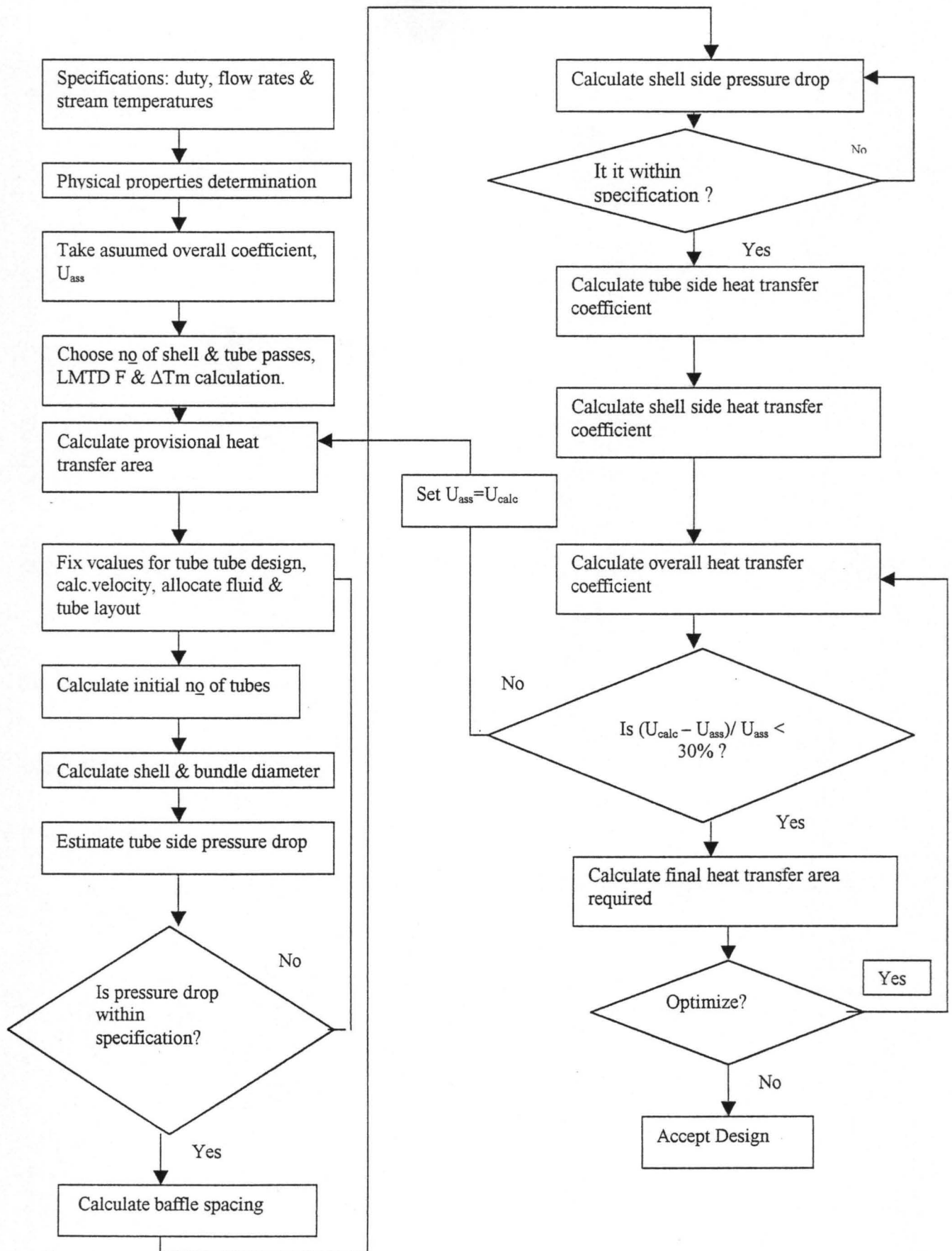
Fig. 3.1   CAD Algorithm (Flowchart).

3.7   Sample Design Problem.

Design an exchanger to sub-cool condensate from a methanol condenser from 95°C to 40°C.

Flow rate of methanol is 100,000 kg/h (i.e 27.78 kg/s). Brackish water is to be used as the coolant with a temperature rise from 25°C to 40°C.

Method for Solution

Kern's Method

This work is based on Kern's method. Kern's method was an experimental work on commercial exchangers with standard tolerances. It gives a reasonably satisfactory prediction of the heat-transfer coefficient for standard designs. The shell-side heat transfer and friction factors (j_h and f) are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and diameter. The shell equivalent diameter is calculated using the flow area between the tubes in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes (Sinnott, 1993).

CHAPTER FOUR

RESULTS, DISCUSSIONS AND CONCLUSION

4.1   RESULTS

A manual calculation of the sample design problem was carried out. The full calculation procedure is given in appendix A. the listing of the program and output from the module are shown in appendix B and C respectively. these results are tabulated below:

Table 4.1   Results of Manual and CAD Procedures

Item Case	Manual	CAD
Tube type:		
Tube length (m)	4.83	4.83
Tube outer and inner diameter (mm)	20,16	20,16
Tube pitch (m)	0.025	0.025
Tube flow velocity (m/s)	1.34	1.3455
Tube fluid pressure drop (bar)	0.623	0.6248
Total number of tubes	1028	1028

Shell		
Shell diameter (m)	0.944	0.9405
Baffle spacing (m)	0.189	0.188
Shell fluid velocity (m/s)	0.52	0.5258
Shell side pressure drop (bar)	0.55	0.5578
Heat Transfer Coefficient:		
Shell Side (W/m^{2o}C)	1253	1206.398
Tube Side (W/m^{2o}C)	5231.9	5480.324
Overall Side (W/m^{2o}C)	687	677
Heat Duty (kW)	4338.8	4332.9
Total Heat Transfer area (m^2)	312	311.8
Initial number of tube passes	2	2
Final number of tube passes	4	4
Area of 1 tube (m^2)	0.3035	0.3033

4.2 DISCUSSION OF RESULTS

From the results tabulated above, it is observed that the solutions given by the manual procedure and the CAD program are in most cases same or very similar.

In shell and tube heat exchanger design, parameters of utmost importance include the pressure drop on both streams, overall heat transfer coefficients as well as flow velocities (Peters 199). Initial flow velocity (tube side) using 2 tubes pass was 0.66m/s from the manual procedure and 0.67 m/s from the CAD program. These values for tube flow velocity were too low (see table 2.5) as it will not aid a high heat transfer coefficient (Sinnott, 1993). Hence the number of tube passes was increased to four, this invariably about doubles the tube flow velocity to a value of 1.345 m/s (CAD) and 1.34 m/s (manual) falling within the limit recommended for tube-side process fluids by Sinnot (1993) to give a high enough heat transfer with reasonable corresponding pressure drop. From the results, a final pressure drop of 0.6249 bar (CAD) and 0.623 bar (manual) was obtained, which is within the range for tube flow allowable pressure drop. From these determinations, the total number of tubes required using four tube passes was estimated to be 1028 tubes for both procedures.

For shell flow velocities, Sinnot (1993) recommended between 0.3 – 1 m/s as vibration induced for high fluid velocities cause premature failure of exchangers. A final value of 0.526 m/s was obtained using the CAD program and 0.52 m/s by manual procedure. The

corresponding pressure drops evaluated gave a final value of 0.5578 bar (CAD) and 0.55 bar (manual). The viscosities of both fluids fell within the range of 1 to 10 mNs/m^2 with a recommended allowable pressure drop of 0.5 − 0.7 bar (see table 2.3) to give near optimum design, so the pressure drops obtained were adjudged all right.

For the tube lengths, inner and outer diameters common specification used in many industrial applications were employed. The heat transfer coefficients based on the outside and inside tube were calculated having satisfied the flow velocities and pressure drops for each stream. The manual procedure gave value of 5231.99 W/m^{2o}C while CAD gave 5480.324 W/m^{2o}C (CAD) for the inside tube heat transfer coefficient. Based on the outside tube (i.e. shell side heat transfer coefficient), 1253 W/m^{2o}C was obtained by the manual procedure and 1206.398 W/m^{2o}C by CAD. All these values compare reasonably well to yield a final overall heat transfer coefficient of 687 W/m^{2o}C (manual) and 677 W/m^{2o}C (CAD). Round off errors in determination of physical properties by manual procedure in contrast to more accurate methods by the CAD program is adjudged to account for differences in the values by CAD and manual calculations. However, an error analysis on the overall heat transfer coefficient showed a mere 1.4% error between the values. A check on table 2.1 shows that the overall heat transfer coefficients calculated fall within the acceptable range for light organics (hot fluid) and water (as cold fluid).

The heat transfer areas and heat duty calculated were also found to be similar 312 m^2 and 311.8m^2 were obtained by manual and CAD respectively for heat transfer area while 4338 kW (manual) and 4333 kW (CAD) were obtained for heat duty.

In many applications, the process engineer supplies the heat exchanger designer all information needed for proper design. These include temperatures (inlet and exit), pressure drops allowable for each stream, flow rates and fluid compositions as well as fouling factors. This program has adequately taken care of situations where some of these parameters are not supplied. For example in many applications the pressure drop available to drive the fluids through the exchanger is set by process conditions. However when the designer is free to select the pressure drop the values suggested in table 2.2, table 2.3 and table 2.5 can also be used as general guides and will give near optimum designs (Sinnott, 1993). This applies to the overall heat transfer coefficients, stream velocities and fouling factors.

4.3   CONCLUSION

Computer aided design is an activity that involves the effective use of the computer and its associated peripherals to create or modify designs. In this CAD program on the design of shell and tube heat exchangers, several correlations, equations and relationships involved in heat transfer and associated equipment have been coded using the **QBASIC** programming language. Where specifications are fully supplied, the designer goes on to input/enter them as required by the program otherwise general guidelines that give optimum design are included to help the designer as in the sample problem. Results obtained from both manual calculations and the CAD program were in most cases similar and values obtained were compared with acceptable limits and found to fall within range. The speed of work (affording the designer time to concentrate on improving and developing an original idea), provision of a databank for a variety of fluids and their associated physical properties, increased productivity as well as product quality are benefits to be derived from the work.

1.10 RECOMMENDATIONS

For further work on this subject, more fluids should be taken into consideration apart from the ones already included in the databank.

Also consideration may be given to situations where there is phase change e.g. in vaporization / condensation etc.

The programming language used in the work may also be changed to align with current development in programming languages used in engineering design.

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APPENDIX A

<u>PROBLEM STATEMENT AND MANUAL SOLUTION.</u>

Design an exchanger to sub-cool condensate from a methanol condenses from 95°C to 40°C.

Flow rate of methanol is 100.000 Kg/h. brackish water will be used as the coolant with a temperature rise from 25°C to 40°C.

SOLUTION

Step One . **Specifications.**

<u>Hot Stream (methanol).</u>

Inlet temp = T_{H1} = 95°C

Outlet temp = T_{H2} = 40 °C

Mean temp = MT_{H} = (95 + 40)/2 = 67.5 °C

Equivalent in Kelvins = (273.15 + 67.5) = 340.5K

<u>Cold Stream (water)</u>

Inlet temp = Tc_1 = 25°C

Outlet temp = T_{C2} = 40 °C

Mean temp = MT_C = (25 + 40)/2 = 32.5 °C

Equivalent in Kelvins = (273.15 + 32.5) = 305.5K

Physical Properties.

	Methanol	Water	Unit
Temperature (mean)	340.5	305.5	K
Specific Heat	2.84	4.2	KJ/Kg K
Thermal Conductivity	0.19	0.59	W/m° K
Density	750	995	Kg / m^3
viscosity	0.34 x 10^{-3}	0.8 x 10^{-3}	Ns /m^2

Heat Duty Calculation (KW)

Using $Q = MCpdt$.

Since flow rate of hot stream is given

$Q = M_hCp_hdT_h$

$Q = (100.000 / 3600)$ x 2.84 x (95-40)

   = 27.78 x 2.84 x 55 = 4338.8 Kw

from the above, we can determine the unspecified flow rate (i.e. of other stream)

$M_C = Q / Cp_cdT_c$ = 4338.8 /4.2 (40-25) = 68.9 Kg/S

Step Three   Assume Overall Coefficient, U_{ass}

From table we take $U_{ass} = 630$

Step Four   Mean Temperature Determination.

log mean temp diff (LMTD)

$$= [(Th_1 - Tc_2) - (Th_2 - Tc_1)]/ \ln (Th_1 - Tc_1)/(Th_2 - Tc_1)$$

$$=[(95 - 40) - (40 - 25)]/\ln [(95-25)/(40-25)] = 30.7$$

Temperature correction factor, F_t

R (heat capacity ratio) = $Th_1 - Th_2 / Tc_2 - Tc_1 = 95 - 40/40-25 = 3.67$

S (thermal effectiveness) = $Tc_2 - Tc_1/ Th_1 - Tc_1 = 40 - 25/95-25 = 0.21$

$Ft = [\sqrt{(R^2 + 1)} \ln ((1 - S)/(1 - RS))]/ (R - 1) \ln [2 - S [R+1 - \sqrt{(R^2 + 1)}/[2 - S(R + 1 + \sqrt{(R^2 + 1)})]$

$Ft = \sqrt{(3.67^2 + 1)} \ln [(1 - 0.21)/ (1 - 3.67 \times 0.21)]/ [(3.67 - 1)\ln[2 - 0.21 [3.67 + 1 - \sqrt{(3.67^2 + 1)}/ 2 - 0.21[3.67 +1 + \sqrt{(3.67^2 + 1)}]]$

$$= 0.75$$

$\therefore \Delta Tm = $ LMTD x Ft = 0.75 x 30.7

$$=23^\circ$$

Step Five   Provisional Heat Transfer Area.

$A = Q/ U_{ass} \Delta Tm = 4338.8 \times 10^3 \, 630 \times 23 = 314 \, m^2$

Step Six   Tube Size, Dimensions, Layout & Fluid Allocation.

Use 1 shell pass, 2 tube passes

Tube outer diameter, TOD = 20×10^{-3}m (20 mm)

Tube inside diameter, TID = 16×10^{-3}m (16 mm)

Tube length diameter, TL = 4.83m

Use split ring floating head type

Triangular pitch arrangement

Tube wall made of Cupro-Nickel.

The dirtier fluid (or more corrosive) should be placed in tubes for easy cleaning

hence water – tube side

methanol – shell side

Step Seven   Number of Tubes calculation

Surface area of 1 tube = π x TOD x TL

$$= \pi \times 20 \times 10^{-3} \times 4.83 = 0.3035 \text{m}^2$$

No of tubes = provisional area/surf. area of 1 tube = 314/0.3035 = 1034

Tube velocity calculation

Vol. Flow rate of tube stream = mass flow/density

$$= 68.9/995 = 0.069 \text{m}^3/\text{s}$$

tube cross-sectional area $= \pi/4 \times (16 \times 10^{-3})^2 = 2.0106 \times 10^{-4} m^2$

tubes per pass $= 1034/2 = 517$

flow area, i.e. area per pass

$\qquad$ a.p.p $= 517 \times$ Tc.s.a

$\qquad\qquad = 517 \times 2.0106 \times 10^{-4} = 0.1039 m^2$

tube side velocity $=$ vol. Flow rate/a.p.p

$\qquad\qquad = 0.069/0.1039 = 0.66 m/s$

Step Eight $\qquad$ Bundle & Shell Diameter

$BD = TOD \, (\, Nt/K_1)^{1/n1}$ Nt $=$ no of tubes

$\qquad\qquad\qquad K_1$ & n_1 are constants

For 2 tube passes, K_1 & n_1 are 0.249 & 2.207 resp.

bundle diam, $BD = 20 \times 10^{-3} \, (1034/.249)^{1/2.207} = 0.875$ m

Shell clearance,

$\qquad SCL = 0.04499 + 0.0271 \, (BD)$

$\qquad = 0.0449 + 0.0271(0.875) + 0.068m$

Shell diameter,

$\qquad SD = 0.875 + 0.068 = 0.944m$

Step Nine   Tube Side Pressure Drop Determination

Using the Blassius eqn. $jf = 0.0791 N_{Re}^{-0.25}$

jf = friction factor   for $4 \times 10^3 < NRe < 10^5$

First determine $N_{Re} = \rho U TID / \mu$

$\quad = 995 \times 0.66 \times 16 \times 10^{-3} / 0.8 \times 10^{-3}$

$\quad = 13134$

$jf = 0.0791 \times 134^{-0.25} = 7.389 \times 10^{-3}$

Tube side pressure drop is given by

$\Delta P_t = Tp[8jf (L/TID) (\mu_b / \mu_w)^{-m} + 2.5] \rho U_t^2 / 2$

$\qquad$ neglecting the viscosity correction term since the fluids are not very

$\qquad$ viscous.

$\Delta P_t = 2[8 \times 7.389 \times 10^3 (4.83/16 \times 10^{-3}) + 2.5] 995 \times 0.66^2 / 2$

$\qquad = 8814 \text{ N/m}^2 \text{ or } 0.08 \text{ bar}$

$\quad$ this value is too low for a desirable pressure drop (check table 2.3).

$\therefore$ double the no of tube passes = 4

velocity doubled as well = $0.66 \times 2 = 1.32$ m/s.

also N_{Re} is doubled = 26268

$jf = 0.0791 (26268)^{-0.25} = 6.213 \times 10^{-3}$

$\Delta P_t = 4[8 \times 6.213 \times 10^{-3} (4.83 / 16 \times 10^{-3}) + 2.5] 995 \times 1.32^2 / 2$

$\qquad = 60696.5 \text{ N/m}^2$

$\qquad = 0.607 \text{ bar} \qquad \text{O.k}$

from this value of p.d actual no of tubes required can be evaluated as

$$= \text{initial no of tubes}/(\text{pd calc.} - \text{allowance}/ \text{p.d calculated})^{0.5}$$

$$= 1034/ (0.61/.607)^{0.5} = 1030$$

use 1028 (to be exactly divisible by no of passes)

Recalculating tube velocity required.

Using 1028 tubes

Tube cross sectional area T.c.sa $= \pi/4 (16 \times 10^{-3})^2$

$$= 2.0106 \times 10^{-4} m^2$$

new tubes per pass $= 1028/4 = 257$

new area per pass $= 257 \times 2.0106 = 0.0517 m^2$

hence new tube velocity $= \text{vol. Flow rate}/ \text{new a.p.p}$

$$0.069/0.05167 = 1.34 m/s$$

hence actual pressure drop =

first determine, $N_{Re} = \rho UTID/\mu$

$$= 995 \times 1.34 \times 16 \times 10^{-3}/0.8 \times 10^{-3} = 2.6666$$

$jf = 0.0791 (26666)^{-0.25} - 6.189 \times 10^{-3}$

$\therefore \Delta P_t = 4[8 \times 6.189 \times 10^{-3} (4.83/16 \times 10^{-3}) + 2.5] 995 \times 1.34^2/2$

$$= 62348.5 N/m^2$$

$$\text{or } 0.623 \text{ bar} \quad \text{O.K}$$

Step Ten   Shell Side Pressure Drop

First determine shell side velocity to get N_{Re}

to get shell velocity, we'll obtain baffle spacing, pitch, and shell equivalent diameter

Baffle spacing = 1/5 of shell diameter

$$= 1/5 \times .944 = 0.189m$$

Pitch value = $1.25 \times TOD \times 20 \times 10^{-3} = 0.025m$

Cross flow area = (Pt – TOD/ Pt) SD x BS

$$= (0.025 – 20 \times 10^{-3}/.025) .944 \times .189 = 0.0356m^2$$

using triangular pitch arrangement,

equivalent diameter = $1.1/20 \times 10^{-3} [(1.25 \times 20 \times 10^{-3})^2 – 0.917 (20 \times 10^{-3})^2]$

$$= 0.0142m$$

shell side vol. Flow rate = mass flow/ density

$$= 27.78/750 = 0.03704 \ m^3s$$

shell side velocity 0.03704/.0356 = 1.038m/s

(may be too high for shell flow)

to get shell N_{Re} = 750 x 1.038 x 0.0142/0.34 x 10^{-3} = 32513

using jf = 0.322 $N_{Re}^{-0.2}$ (for shell side friction factor)

jf = .040313

Shell side pressure drop is given by

ΔP_s= 8jf (SD/de)(TL/BS)ℓsUs^2/2

de = equivalent diameter, S.D=shell diameter, and BS=baffle spacing

$\Delta P_s = 8 \times 0.040313 \times (.944/.0142) \times (4.83/.189) \times 950 \times [1.038]^2/2$

$= 221375 \text{ N/m}^2$

or 2.2 bar (too high) check table 2.3.

Halving the velocity is obtained by doubling the pitch, which in turn reduces ΔP_s by a factor of $(1/2)^2$ or $\frac{1}{4}$

Hence $\Delta P_s \times \frac{1}{4} = 2.2/4 = 0.55$ bar   -acceptable

velocity also becomes $1.038/2 = 0.52$m/s

recalculating for N_{Re} (shell side) $= \rho_s u_s de/\mu_s = 750 \times 0.52 \times 0.0142/0.34 \times 10^{-3}$

$= 16288.$

Step eleven   <u>Tube side heat Transfer Coefficient, hi</u>

Tube side Prandtl no. $N_{pr} = Cp_C \, \mu_t/K_t = 4.2 \times 10^{-3} \times 0.8 \times 10^{-3}/0.59 = 5.7$

Tube side Nusselt no $N_{NU} = 0.023 \, N_{Re}^{0.8} N_{Pr}^{0.33}$

$= 0.023 \, (26666)^{0.8} \, (5.7)^{0.33}$

$= 141.88$

$hi = N_{NU} \times K_t / \text{T.I.D} = 141.88 \times 0.59 / 16 \times 10^{-3} = 5231.9 \text{w/m}^2{}^\circ\text{C}$

<u>Shell side Heat transfer coefficient, ho.</u>

Shell side Prandtl no $= Cp_h \mu_s/K_s = 2.84 \times 10^{-3} \times .34 \times 10^{-3} / 0.19 = 5.1$

for $2000 < N_{Re} < 32,000$, use

$$ho = a_o/ \; Fs \; x \; K_f/de \; . \; N_{Re}^{0.6} \; N_{Pr}^{0.3}$$

$$a_o = .26 \text{ for well lined tubes}$$

$$Fs = \text{safety factor} = 1.6$$

$$ho = 0.26/1.6 \; x \; 0.19/0.0142 \; x \; (16288)^{0.6} \; x \; (5.1)^{0.33} = 1253 \; w/m^{2o}c$$

Overall Heat Transfer Coefficient Required

$$1/U_o = 1/h_o + R_{fo} + TOD \ln (TOD/TID)/2k_w + (TOD/TID)(1/hi + R_{fi})$$

$$R_{fi,} \; R_{fo} = \text{inside and outside fouling factors}$$

$$k_w = \text{tube wall thermal conductivity}$$

$$1/U_o = 1/1253 + 0.000167 + 20 \; x \; 10^{-3} \ln (20/16) \; /2 \; x \; 50 + 20/16 \; x \; (0.000167 + 1/5232)$$

$$1/U_o = 1.45657 \; x \; 10^{-3}$$

$$U_o = (1.45657 \; x \; 10^{-3})^{-1} = 687 \; W/m^{2o}c$$

Heat Transfer Area Required

A   = Total number of tubes x surface area of 1 tube

       = 1028 x 0.3035

       = $312 m^2$

APPENDIX B

<u>CAD PROGRAM LISTING</u>

```
DECLARE SUB CMETALS ()
DECLARE SUB TFFRANGE ()
DECLARE SUB TOC ()
DECLARE SUB VISCO (FLU$, MTEMP)
DECLARE SUB SPECHEAT (FLU$, MTEMP)
DECLARE SUB MYCLEAR ()
DECLARE SUB DENBANK (FLU$, MTEMP)
DECLARE SUB CONDUCT (FLU$, MTEMP)


10 COLOR 14, 1, 8: CLS
LOCATE 5, 10: PRINT " C O M P U T E R    A I D E D    D E S I G N "
LOCATE 8, 10: PRINT "                      O F            "
LOCATE 11, 10: PRINT " S H E L L  A N D  T U B E  H E A T  E X C H A N G E R"
CALL MYCLEAR


30 CLS
  COLOR 9
  COLOR 3, 0
  FOR Y = 4 TO 18
  LOCATE Y, 4
  PRINT STRING$(70, CHR$(176))
  NEXT Y
  COLOR 1
  FOR K = 5 TO 16
  LOCATE K, 20
  PRINT STRING$(40, CHR$(219))
  NEXT K
  'LOCATE 4,4:COLOR 3:PRINT STRING$(70," ")
  LOCATE 25, 4: COLOR 0, 3: PRINT "              PRESS [Esc] TO QUIT          "
  MEN1$ = "  ABOUT THIS PROJECT    "
  MEN2$ = "  DESIGN SPECIFICATION  "
  MEN3$ = "  QUIT THIS PROGRAM     "
  LOCATE 2, 10: COLOR 4: PRINT "[COMPUTER AIDED DESIGN OF SHELL AND TUBE HEAT
EXCHANGER]"
  LOCATE 3, 30: COLOR 15: PRINT "(T H E   M E N U)"
  LOCATE 23, 4: COLOR 0, 3: PRINT "                  .                  "
  LOCATE 23, 15: COLOR 4: PRINT " USE  ": LOCATE 23, 20: PRINT " ": LOCATE 23, 22: PRINT
STRING$(1, 25) + STRING$(1, 24): LOCATE 23, 29: PRINT "KEYS  TO  SELECT  AND  PRESS ";
CHR$(17) + CHR$(196) + CHR$(217)
  COLOR 6, 1
  LOCATE 8, 28: PRINT MEN1$
  LOCATE 10, 28: PRINT MEN2$
  LOCATE 12, 28: PRINT MEN3$
```

```basic
       Y1 = 8
       Y12 = Y1
       COLOR 0, 3: LOCATE Y1, 28: PRINT MEN1$
       MEN22$ = MEN1$
       DO
       A$ = INKEY$
       SELECT CASE A$
       CASE CHR$(0) + CHR$(80):
       'COLOR 0,3:
       Y1 = Y1 + 2
       IF Y1 > 12 THEN Y1 = 8
       IF Y1 = 8 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 8, 28:
PRINT MEN1$
       IF Y1 = 10 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 10, 28:
PRINT MEN2$
       IF Y1 = 12 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 12, 28:
PRINT MEN3$
       Y12 = Y1
       IF Y1 = 8 THEN MEN22$ = MEN1$
       IF Y1 = 10 THEN MEN22$ = MEN2$
       IF Y1 = 12 THEN MEN22$ = MEN3$
       CASE CHR$(0) + CHR$(72):
       Y1 = Y1 - 2
       IF Y1 < 8 THEN Y1 = 12
       IF Y1 = 8 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 8, 28:
PRINT MEN1$
       IF Y1 = 10 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 10, 28:
PRINT MEN2$
       IF Y1 = 12 THEN COLOR 6, 1: LOCATE Y12, 28: PRINT MEN22$: COLOR 0, 3: LOCATE 12, 28:
PRINT MEN3$
       Y12 = Y1
       IF Y1 = 8 THEN MEN22$ = MEN1$
       IF Y1 = 10 THEN MEN22$ = MEN2$
       IF Y1 = 12 THEN MEN22$ = MEN3$
       CASE CHR$(13):
          IF Y1 = 8 THEN GOSUB 40: GOTO 30
          IF Y1 = 10 THEN GOSUB 50: GOTO 30
          IF Y1 = 12 THEN GOSUB 90: GOTO 101
       END SELECT
       COLOR 6, 1
       LOOP UNTIL A$ = CHR$(27)
       A$ = INKEY$: GOSUB 90
       ON KEY(A$ = CHR$(27)) GOSUB 90

       CALL MYCLEAR
       RETURN

50 :   COLOR 15, 1, 0: CLS
       PRINT "HOT STREAM SPECIFICATION"
       PRINT "========================="
       PRINT
       INPUT "ENTER INLET TEMPERATURE OF THE HOT STREAM (øC) = ", Th1
       PRINT
```

```
INPUT "ENTER OUTLET TEMPERATURE OF THE HOT STREAM (øC) = ", Th2
PRINT
MTH = (Th1 + Th2) / 2
PRINT "MEAN TEMPERATURE OF THE HOT STREAM = ", MTH, "øC"
PRINT
MTEMP = MTH + 273.15

INPUT "Name of Fluid"; FLU$
REM     specific heat capacity calculations
CALL SPECHEAT(FLU$, MTEMP)
Cph = spval
CALL MYCLEAR
PRINT "COLD STREAM SPECIFICATION"
PRINT "========================="
PRINT
INPUT "ENTER INLET TEMPERATURE OF THE COLD STREAM (øC) = ", Tc1
PRINT
INPUT "ENTER OUTLET TEMPERATURE OF THE COLD STREAM (øC) = ", Tc2
PRINT
MTC = (Tc1 + Tc2) / 2
PRINT "MEAN TEMPERATURE OF THE COLD STREAM = ", MTC, "øC"
PRINT
MTEMP = MTC + 273.15

INPUT "Name of Fluid"; FLU$
REM     specific heat capacity calculations
CALL SPECHEAT(FLU$, MTEMP)
Cpc = spval
CALL MYCLEAR
PRINT "HEAT DUTY (KW) "
PRINT "==============="
PRINT
PRINT "Enter available stream mass flow rate (Kg/s)";
INPUT "1 - FOR COLD AND 2 - HOT"; MTT
IF MTT = 2 THEN
   INPUT "ENTER MASS FLOW RATE FOR HOT STREAM (Kg/s) "; MFH
   Q = MFH * Cph * (Th1 - Th2)
   PRINT "DUTY = "; Q; "KW"
   MFC = Q / (Cpc * (Tc2 - Tc1))
   PRINT "MASS FLOW RATE FOR COLD STREAM = "; MFC; "Kg/s"
ELSE
   INPUT "ENTER MASS FLOW RATE FOR COLD STREAM (Kg/s) "; MFC
   Q = MFC * Cpc * (Tc2 - Tc1)
   PRINT "DUTY = "; Q; "KW"
   MFH = Q / (Cph * (Th1 - Th2))
   PRINT "MASS FLOW RATE FOR HOT STREAM = "; MFH; "Kg/s"
END IF
CALL MYCLEAR
CALL TOC
INPUT "ENTER AN ASSUMED OVERALL COEFFICIENT FROM THE TABLE ABOVE
(W/mýøC)", UASS
CALL MYCLEAR
PRINT "MEAN TEMPERATURE DETERMINATION"
```

```
PRINT "==============================="
a1 = Th1 - Tc2
a2 = Th2 - Tc1
LMTD = (a1 - a2) / LOG(a1 / a2)
PRINT "LOG MEAN TEMPERATURE DIFFERENCE = ", LMTD, "øC"
PRINT
PRINT "Temperature Correction Factor"
PRINT "------------------------------"
R = (Th1 - Th2) / (Tc2 - Tc1)
s = (Tc2 - Tc1) / (Th1 - Tc1)
B1 = ((R ^ 2 + 1) * (LOG((1 - s) / (1 - (R * s))))) ^ .5
B2 = 2 - (s * ((R + 1) - ((R ^ 2 + 1) ^ .5)))
B3 = 2 - (s * ((R + 1) + ((R ^ 2 + 1) ^ .5)))
FT = B1 / ((R - 1) * LOG(B2 / B3))
PRINT "HEAT CAPACITY RATIO = ", R
PRINT
PRINT "THERMAL EFFECTIVENESS = ", s
PRINT
PRINT "TEMPERATURE CORRECTION FACTOR = ", FT
PRINT
MTD = FT * LMTD
PRINT "MEAN TEMPERATURE DIFFERENCE = ", MTD, "øC"
PRINT
A = (Q * 1000) / (UASS * MTD)
PRINT "HEAT TRANSFER AREA (PROVISIONAL) = ", A, "mý"
PRINT
CALL MYCLEAR
PRINT "TUBE SIZE LAYOUT, DIMENSIONS AND FLUID ALLOCATION"
PRINT "====================================================="
PRINT "USE 1 SHELL PASS, 2 TUBE PASSES"
PRINT
TP = 2
'.02
INPUT "ENTER TUBE OUTER DIAMETER (m) ", TOD
'TID = .016
INPUT "ENTER TUBE INNER DIAMETER (m) ", TID
TL = 4.83
PV = 1.25 * TOD
'PRINT "TUBE OUTER DIAMETER = ", TOD, "m"
PRINT
'PRINT "TUBE INNER DIAMETER = ", TID, "m"
PRINT
'PRINT "TUBE LENGTH = ", TL, "m"
PRINT
INPUT "WHICH OF THE FLUIDS IS DIRTIER (MORE FOULIN) OR CORROSIVE? "; FL$
PRINT
PRINT FL$; "SHOULD BE ALLOCATED TO TUBE SIDE"
PRINT
PRINT "CALCULATION OF NUMBER OF TUBES"
PRINT "==============================="
PI = 3.14
SA1 = PI * TOD * TL
PRINT "SURFACE AREA OF 1 TUBE = ", SA1, "mý"
```

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```
NT = CINT(A / SA1)
NTTEST = NT MOD TP
IF NTTEST <> 0 THEN
  NT = NT - NTTEST
  PRINT "NUMBER OF TUBES = "; NT
ELSE
  PRINT "NUMBER OF TUBES  = "; NT
END IF

PRINT "NUMBER OF TUBES = ", NT
TPP = (NT / TP)
PRINT "TUBE PER PASS = ", TPP
INPUT "ENTER DENSITY OF TUBE FLUID "; DENT
VFRT = MFC / DENT
PRINT "VOLUMENTRIC FLOW RATE(TUBE STREAM)=", VFRT, "m^3/s"
TCA = (PI / 4) * (TID ^ 2)
APP = TPP * TCA
TSV = VFRT / APP
PRINT "TUBE SIDE VELOCITY = ", TSV, "m/s"
CALL MYCLEAR
PRINT "BUNDLE AND SHELL DIAMETER CALCULATION"
PRINT "======================================="
PRINT
INPUT "ENTER 1 - FOR TRIANGULAR PITCH OR 2 - FOR SQUARE PITCH ARRANGEMENT =
", TYPEP
PRINT
IF TYPEP = 1 THEN
  GOSUB TYPE1
ELSE
  GOSUB TYPE2
END IF
BD = TOD * ((NT / K1) ^ (1 / N1))
PRINT "BUNDLE DIAMETER = "; BD; "m"
SCL = .0449 + .0271 * BD
SD = BD + SCL
PRINT
'PRINT "SHELL DIAMETER = "; SD; "m"
PRINT
PRINT "TUBE SIDE PRESSURE DROP"
PRINT "========================"
PRINT
CALL VISCO(FLU$, MTEMP)
Vt = spval
PRINT "VISCOSITY OF TUBE FLUID = "; Vt; "Pa.s"
PRINT
CON1: NRET = (DENT * TSV * TID) / Vt
PRINT "TUBE SIDE REYNOLDS NUMBER = "; NRET
jf = .0791 * (NRET ^ (-.25))
PDT = (TP * (8 * jf * (TL / TID) + 2.5) * DENT * ((TSV ^ 2) / 2)) / (10 ^ 5)
PRINT
PRINT "TUBE SIDE PRESSURE DROP = "; PDT; "bar"
IF (Vt >= .0001) AND (Vt <= .001) THEN
   IF (PDT >= .5) AND (PDT <= .75) THEN
```

```
      PRINT
    ELSE
      TP = TP * 2
      TSV = TSV * 2
      GOTO CON1
    END IF
ELSE
END IF

NT = CINT(NT / (((.624 / PDT) ^ (.5))))
NTTEST = NT MOD TP
IF NTTEST <> 0 THEN
  NT = NT - NTTEST
  'PRINT "ACTUAL NUMBER OF TUBES REQUIRED = "; NT
ELSE
 ' PRINT "ACTUAL NUMBER OF TUBES REQUIRED = "; NT
END IF
CALL MYCLEAR
PRINT "ACTUAL VELOCITY REQUIRED"
PRINT "========================"
PRINT
NTPP = NT / TP
'PRINT "NEW TUBE PER PASS = "; NTPP
NAPP = NTPP * TCA
PRINT
PRINT "NEW AREA PER PASS = "; NAPP, "mý"
FTSV = VFRT / NAPP
PRINT
'PRINT "REQUIRED VELOCITY = "; FTSV; "m/s"
NRET = (DENT * FTSV * TID) / Vt
PRINT
PRINT "TUBE SIDE REYNOLDS NUMBER = "; NRET
jf = .0791 * (NRET ^ (-.25))
PDT = (TP * (8 * jf * (TL / TID) + 2.5) * DENT * ((FTSV ^ 2) / 2)) / (10 ^ 5)
PRINT
'PRINT "ALLOWABLE TUBE SIDE PRESSURE DROP = "; PDT; "bar"
PRINT "SHELL SIDE PRESSURE DROP"
PRINT "========================="
PRINT
BS = (1 / 5) * SD
'PRINT "BAFFLE SPACING = "; BS; "m"
PRINT
PT = 1.25 * TOD
'PRINT "PITCH = "; PT; "m"
PRINT
CFA = ((PT - TOD) / PT) * SD * BS
PRINT "CROSS FLOW AREA = "; CFA; "m"
PRINT
IF TYPEP = 1 THEN
    ED = (1.1 / TOD) * (PT ^ 2 - .917 * TOD ^ 2)
 ELSEIF TYPEP = 2 THEN
    ED = (1.27 / TOD) * (PT ^ 2 - .785 * TOD ^ 2)
END IF
```

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```
PRINT "EQUIVALENT DIAMETER = "; ED; "m"
PRINT
INPUT "ENTER NAME OF THE SHELL FLUID "; FLU$
MTEMP = MTH + 273.15
CALL DENBANK(FLU$, MTEMP)
DENS = ad
CALL VISCO(FLU$, MTEMP)
Vs = spval
VFRS = MFH / DENS
PRINT
PRINT "VOLUMETRIC FLOW RATE FOR SHELL FLUID = "; VFRS; "m^3/s"
PRINT
SSV = VFRS / CFA
PRINT "SHELL SIDE VELOCITY = "; SSV; "m/s"
PRINT
CALL MYCLEAR
NRES = (DENS * SSV * ED) / Vs
PRINT
PRINT "SHELL FLUID REYNOLDS NUMBER = "; NRES
F = .322 * (NRES ^ (-.2))
PDS = (8 * F * (SD / ED) * (TL / BS) * DENS * ((SSV ^ 2) / 2)) / (10 ^ 5)
PRINT
PRINT "PRESSURE DROP FOR SHELL SIDE FLUID = "; PDS; "bar"
PRINT
IF PDS > .85 THEN
    PRINT "PRESSURE DROP IS TOO HIGH, ADJUST THE SHELL SIDE VELOCITY"
    PRINT
    SSV = SSV / 2
    PDS = PDS / 4
    'PRINT "NEW SHELL SIDE VELOCITY = "; SSV; "m/s"
    PRINT
    'PRINT "NEW SHELL SIDE PRESSURE DROP = "; Pds; "bar"
END IF
NRES = (DENS * SSV * ED) / Vs
PRINT
PRINT "NEW SHELL FLUID REYNOLDS NUMBER = "; NRES
CALL MYCLEAR
PRINT "TUBE SIDE HEAT TRANSFER COEFFICIENT"
PRINT "===================================="
PRINT
MTEMP = MTC + 273.15
INPUT "ENTER THE FLUID "; FLU$
CALL CONDUCT(FLU$, MTEMP)
PRINT
Kt = mval
'PRINT "Kt = "; Kt
'PRINT "Vt = "; Vt
'PRINT "Cpc = "; Cpc

NPRT = (Vt * (Cpc * (10 ^ 3))) / Kt

PRINT " TUBE FLUID PRANDTL NUMBER = "; NPRT
PRINT
```

```
NNUT = .023 * (NRET ^ (.8)) * (NPRT ^ (.33))
PRINT "TUBE SIDE NUSSELT NUMBER = "; NNUT
PRINT
hi = (NNUT * Kt) / TID
'PRINT "TUBE SIDE HEAT TRANSFER COEFFICIENT = "; hi; "W/mýøC"
CALL MYCLEAR
PRINT "SHELL SIDE HEAT TRANSFER COEFFICIENT"
PRINT "===================================="
PRINT
MTEMP = MTH + 273.15
INPUT "ENTER THE FLUID "; FLU$
CALL CONDUCT(FLU$, MTEMP)
Ks = mval
'PRINT "Ks = "; Ks
'PRINT "Vs = "; Vs
'PRINT "Cph = "; Cph

NPRS = (Vs * (Cph * 10 ^ 3)) / Ks
PRINT " SHELL FLUID PRANDTL NUMBER = "; NPRS
PRINT
ao = .26
Fs = 1.6
ho = (Ks / ED) * (ao / Fs) * (NRES ^ (.6)) * (NPRS ^ (.3))
'PRINT "SHELL SIDE HEAT TRANSFER COEFFICIENT = "; ho; "W/mýøC"
CALL MYCLEAR
PRINT "OVERALL HEAT TRANSFER COEFFICIENT"
PRINT "===================================="
PRINT
CALL TFFRANGE
PRINT
INPUT "ENTER FOULING FACTOR OUTSIDE (RFO)"; RFO
INPUT "ENTER FOULING FACTOR INSIDE (RFI)"; RFI
PRINT
CALL MYCLEAR
CALL CMETALS
INPUT "ENTER WALL THERMAL CONDUCTIVITY (W/møC) "; Kw
CALL MYCLEAR
X = (1 / ho) + RFO + ((TOD * (LOG(TOD / TID))) / (2 * Kw)) + (TOD / TID) * ((1 / hi) +
RFI)
Uo = 1 / X
' PRINT "OVERALL HEAT TRANSFER COEFFICIENT"
' PRINT "===================================="
' PRINT
'PRINT "OVERALL HEAT TRANSFER COEFFICIENT = "; Uo; "W/mýøC"
'PRINT
PRINT "Actual Heat Transfer Area Required"
'PRINT "--------------------------------"
'PRINT
AA = NT * SA1
'PRINT "ACTUAL HEAT TRANSFER AREA = "; AA; "mý"
'CALL MYCLEAR

OPEN "OUTPUT.DAT" FOR APPEND AS #1
```

```
CLS
PRINT #1, "            CAD OUTPUT            "
PRINT #1, "===================================="
PRINT #1, "Tube inner diameter = ", TID, "m"
PRINT #1, "Tube outer diameter = ", TOD, "m"
PRINT #1, "Tube length =          ", TL, "m"
PRINT #1, "Tube side velocity = ", FTSV, "m/s"
PRINT #1, "Tube side pressure drop = ", PDT, "bar"
PRINT #1, "Number of tube passes = ", TP
PRINT #1, " Total number of tubes = ", NT
PRINT #1, " "
PRINT #1, "Shell diameter = ", SD, "m"
PRINT #1, "Pitch =          ", PT, "m"
PRINT #1, "Baffle spacing = ", BS, "m"
PRINT #1, "Shell side velocity = ", SSV, "m/s"
PRINT #1, "Shell side pressure drop = ", PDS, "bar"
PRINT #1, ""
PRINT #1, "Shell side heat transfer coefficient = ", ho, "W/mýøC"
PRINT #1, "Tube side heat transfer coefficient = ", hi, "W/mýøC"
PRINT #1, "Overall heat transfer coefficient = ", Uo, "W/mýøC"
PRINT #1, "Heat duty =        ", Q, "KW"
PRINT #1, "Heat transfer area = ", AA, "mý"
CLOSE #1


PRINT "            CAD OUTPUT            "
PRINT "===================================="
PRINT "Tube inner diameter = ", TID, "m"
PRINT "Tube outer diameter = ", TOD, "m"
PRINT "Tube length =          ", TL, "m"
PRINT "Tube side velocity = ", FTSV, "m/s"
PRINT "Tube side pressure drop = ", PDT, "bar"
PRINT "Number of tube passes = ", TP
PRINT " Total number of tubes = ", NT
PRINT " "
PRINT "Shell diameter = ", SD, "m"
PRINT "Pitch =          ", PT, "m"
PRINT "Baffle spacing = ", BS, "m"
PRINT "Shell side velocity = ", SSV, "m/s"
PRINT "Shell side pressure drop = ", PDS, "bar"
PRINT ""
PRINT "Shell side heat transfer coefficient = ", ho, "W/mýøC"
PRINT "Tube side heat transfer coefficient = ", hi, "W/mýøC"
PRINT "Overall heat transfer coefficient = ", Uo, "W/mýøC"
PRINT "Heat duty =        ", Q, "KW"
PRINT "Heat transfer area = ", AA, "mý"
CALL MYCLEAR

RETURN


90  COLOR 15, 1, 2: CLS
    LOCATE 6, 10: PRINT "ÞÞÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞÞÞÞ "
    LOCATE 7, 10: PRINT "ÞÞÞ     ÞÞÞ ÞÞÞ     ÞÞÞ ÞÞÞ     ÞÞÞ     ÞÞÞ "
```

```
    LOCATE 8, 10: PRINT "ÞÞÞ        ÞÞÞ   ÞÞÞ ÞÞÞ   ÞÞÞ   ÞÞÞ   ÞÞÞ "
    LOCATE 9, 10: PRINT "ÞÞÞ          ÞÞÞ   ÞÞÞ ÞÞÞ   ÞÞÞ   ÞÞÞ   ÞÞÞ "
    LOCATE 10, 10: PRINT "ÞÞÞ  ÞÞÞÞÞ ÞÞÞ   ÞÞÞ ÞÞÞ `ÞÞÞ   ÞÞÞ . ÞÞÞ "
    LOCATE 11, 10: PRINT "ÞÞÞ    ÞÞÞ ÞÞÞ   ÞÞÞ ÞÞÞ   ÞÞÞ   ÞÞÞ   ÞÞÞ "
    LOCATE 12, 10: PRINT "ÞÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞ ÞÞÞÞÞÞÞÞÞÞÞ "

    LOCATE 14, 18: PRINT "ÞÞÞÞÞÞÞÞÞÞÞ ÞÞÞ   ÞÞÞ ÞÞÞÞÞÞÞÞÞ           "
    LOCATE 15, 18: PRINT "ÞÞÞ     ÞÞÞ ÞÞ ÞÞ ÞÞÞ   ÞÞÞ               "
    LOCATE 16, 18: PRINT "ÞÞÞ     ÞÞÞ ÞÞÞÞÞ  ÞÞÞ                     "
    LOCATE 17, 18: PRINT "ÞÞÞÞÞÞÞÞ   ÞÞÞ   ÞÞÞÞÞÞÞ                   "
    LOCATE 18, 18: PRINT "ÞÞÞ     ÞÞÞ   ÞÞÞ   ÞÞÞ                    "
    LOCATE 19, 18: PRINT "ÞÞÞ     ÞÞÞ   ÞÞÞ   ÞÞÞ   ÞÞÞ              "
    LOCATE 20, 18: PRINT "ÞÞÞÞÞÞÞÞÞÞ   ÞÞÞ   ÞÞÞÞÞÞÞÞÞ               "
    CALL MYCLEAR
    IF A$ = CHR$(27) THEN END ELSE GOTO 100
100  RETURN

101 END




TYPE1:
   IF TP = 1 THEN
       K1 = .319
       N1 = 2.142
      ELSE
      END IF
    IF TP = 2 THEN
      K1 = .249
      N1 = 2.207
     ELSE
    END IF
    IF TP = 4 THEN
     K1 = .175
     N1 = 2.285
     ELSE
    END IF
    IF TP = 6 THEN
     K1 = .0743
     N1 = 2.499
     ELSE
    END IF
    IF TP = 8 THEN
     K1 = .0365
     N1 = 2.675
     ELSE
    END IF
RETURN

TYPE2:
  IF TP = 1 THEN
```

```
        K1 = .215
        N1 = 2.207
      ELSE
  END IF
      IF TP = 2 THEN
        K1 = .156
        N1 = 2.291
       ELSE
       END IF
      IF TP = 4 THEN
        K1 = .158
        N1 = 2.263
       ELSE
       END IF
       IF TP = 6 THEN
        K1 = .0402
        N1 = 2.617
       ELSE
       END IF
       IF TP = 8 THEN
        K1 = .0331
        N1 = 2.643
        ELSE
       END IF
RETURN
```

APPENDIX C
CAD PROGRAM OUTPUT.

CAD OUTPUT

==

Tube inner diameter =	.016	m
Tube outer diameter =	.02	m
Tube length =	4.83	m
Tube side velocity =	1.345541	m/s
Tube side pressure drop =	.6248931	bar
Number of tube passes =	4	
Total number of tubes =	1028	
Shell diameter =	.9405074	m
Pitch =	.025	m
Baffle spacing =	.1881015	m
Shell side velocity =	.5258217	m/s
Shell side pressure drop =	.5575233	bar
Shell side heat transfer coefficient =	1206.398	W/m²°C
Tube side heat transfer coefficient =	5480.324	W/m²°C
Overall heat transfer coefficient =	677.9058	W/m²°C

Heat duty =   4332.895 KW
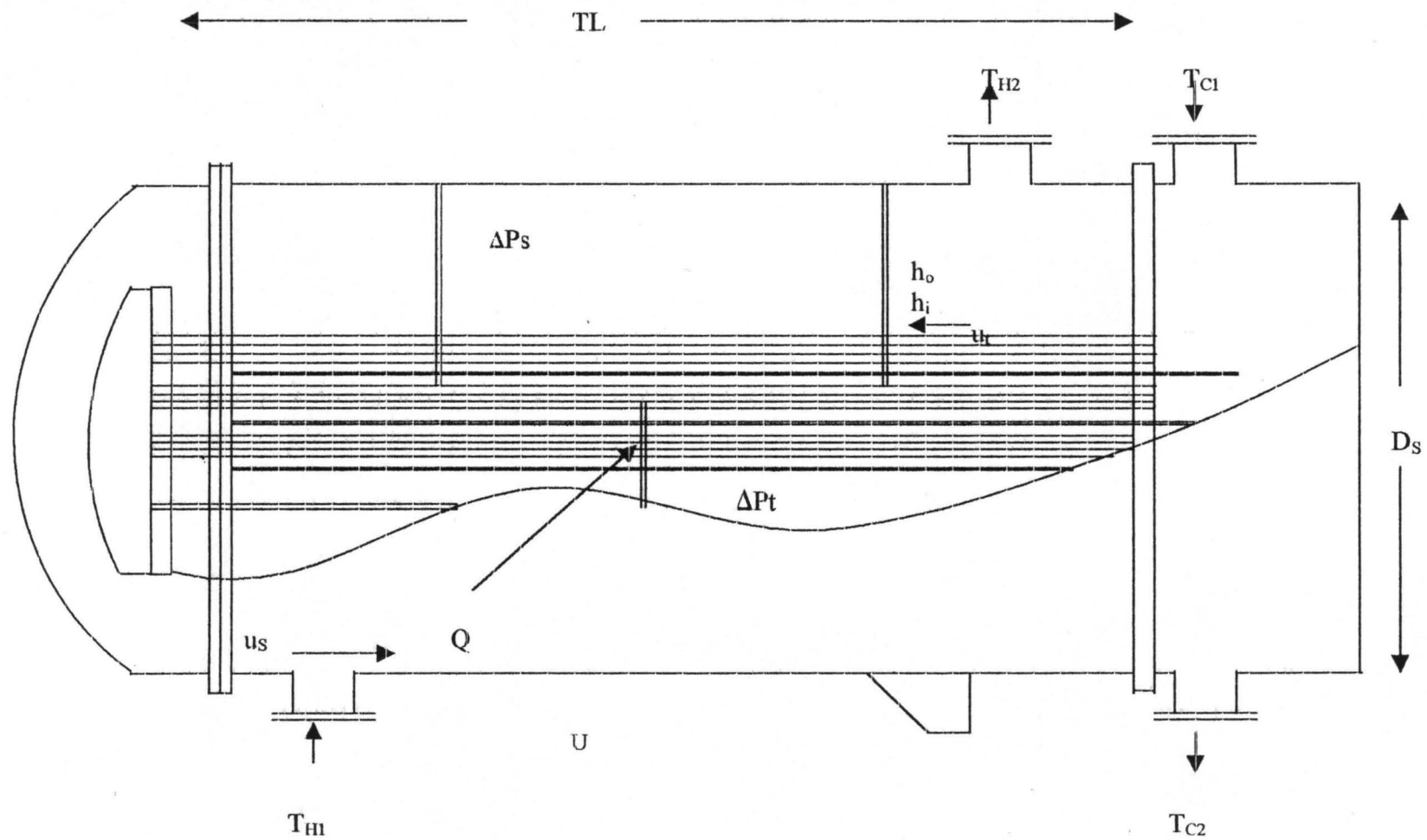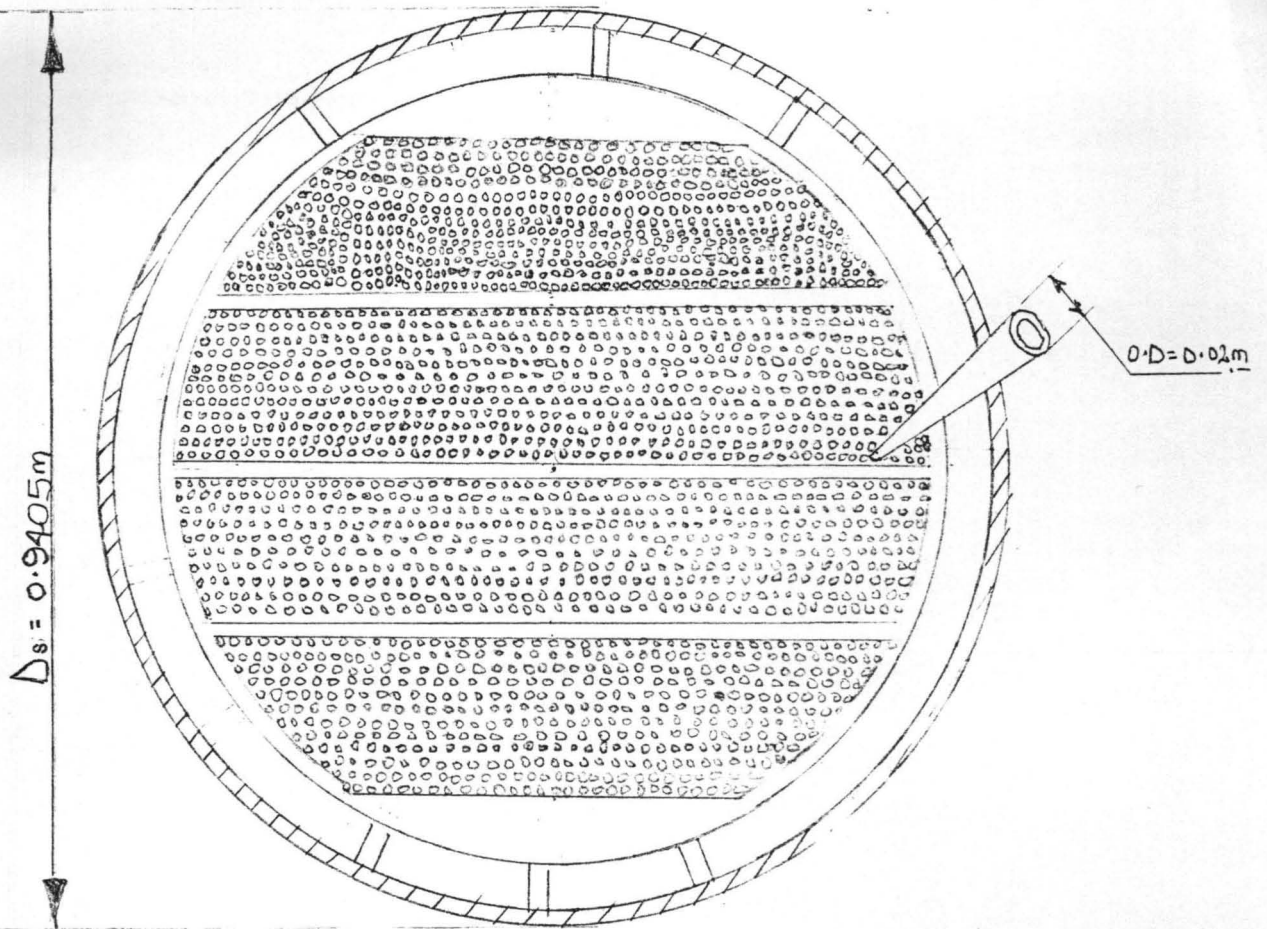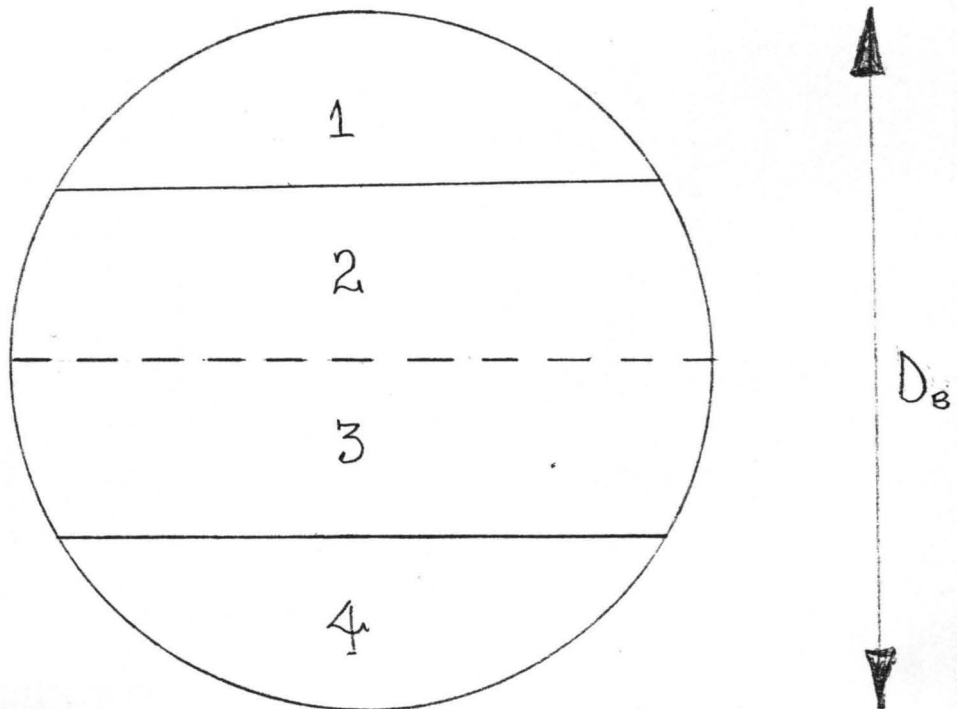
Heat transfer area =   311.87 m^2

Press □ -+

FIG. D1: DIAGRAM OF SHELL AND TUBE HEAT EXCHANGER

CROSS SECTION OF HEAT EXCHANGER.

$D_s = 0.9405\,m$

$O.D = 0.02\,m$

1

2

3

4

D_B

PASS PARTITIONS.