HEXSIM: A PRELIMINARY SHELL AND TUBE AND AIR-COOLED HEAT EXCHANGER DESIGN/RATING PROGRAM

By

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PREFACE

The HEXSIM simulator was debugged and modified to increase user friendliness. Then a user's manual and technical documentation were written. HEXSIM is capable of calculating size and configuration for both shell and tube and air-cooled heat exchangers for sensible heat transfer problems. The methods used are given by Bell (4) for shell and tube exchangers and by the GPSA Engineering Data Book (6) for air-coolers.

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NOMENCLATURE

A	factor used to calculate CLMTD1
A_{f1}, A_{f2}, A_{f3}, A_{r1}	factors used in calculating APF, APSFPR, AR
Airfar	factor used to calculate air velocity
A_i'	inside area, ft^2
AMTD	arithmetic mean temperature difference
A_m'	mean area for heat transfer, ft^2
A_o'	outside area for heat transfer, ft^2
A_o	total heat transfer area
APF	total external area per foot of fin tube, ft^2/ft
APSFPR	external area in ft^2/ft^2 of bundle face area per row
AR	exterior area of bare tube, ft^2
A_{rfa1}, A_{rfa2}	factors used in calculating air velocity
A_{root}	bare tube area, in.
AVFF	air velocity at the fan face, ft/sec
AVPF	air volume per fan, ft^3/min
AVTF	air velocity at tube face, ft/min
B	factor used to calculate CLMTD1
BHP	fan requirement, hp
C	minimum clearance between the outermost tubes and the inside of the shell, in. or mm
C_1	tube count for single-pass exchanger
C_2	tube count for 2-pass exchanger

C_4	tube count for 4-pass exchanger
C_6	tube count for 6-pass exchanger
C_8	tube count for 8-pass exchanger
CLMTD1, CLMTD	correction factor times logarithmic mean temperature difference for air-cooled heat exchangers
C_p, c_p	specific heat of hot fluid and cold fluid, respectively
C_{pshell}	specific heat of shell side fluid
C_x	number of tubes along horizontal axis
C_y	number of tubes along vertical axis
C_{pa}	air side specific heat
D_i	shell inside diameter
D_o	diameter over fins, in.
D_{otl}	diameter of the outer tube limit
D_r	root diameter, in.
D_w	equivalent diameter of the window
d_i	inside tube diameter
d_m	mean wall heat transfer diameter
d_o	tube outside diameter
e	dimensionless constant; 0.265 for 6-pass arrangement and 0.404 for 8-pass arrangement
eff	fan efficiency
F_a	face area
FAPF	face area per fan
F_c	fraction of the total tubes that are in crossflow

F_{sbp} fraction of total crossflow area that is available for bypass flow around the tube bundle and through pass partition lanes

F_t configuration correction factor on LMTD

f_f friction factor for tube side fluid

f_i friction factor for flow across an ideal tube bank

f_r air-side friction factor

g_c gravitational constant, 4.17×10^8 lbm-ft/lbf-hr^2

H height of fin

h_{air} air-side individual heat transfer coefficient, Btu/ft^2/ hr°F

h_i tube side heat transfer coefficient

h_{ideal} shell side heat transfer coefficient for an ideal tube bank

h_o shell side heat transfer coefficient

J_b correction factor or the shell side heat transfer coefficient for bundle by pass effects

J_c correction factor on the shell-side heat transfer coefficient to account for baffle configuration effects

J_l correction factor on the shell-side heat transfer coefficient to account for baffle leakage effects

J_r^* base correction factor on the shell-side heat transfer coefficient to account for buildup of adverse temperature gradient

J_s correction factor on the shell-side heat transfer coefficient to account for unequal baffle spacing

j_i Colburn factor for an ideal tube bank

k_a air thermal conductivity, Btu/ft hr°F

k thermal conductivity of fluid

k_f fin thermal conductivity

k_{shell} shell side thermal conductivity

k_w thermal conductivity of tube wall

L effective tube length

LMTD logarithmic mean temperature difference for countercurrent flow

l_c baffle cut distance from baffle tip to shell inside diameter

l_s baffle spacing, center-to-center of consecutive baffles

$l_{s,I}$ - $l_{s,o}$ baffle spacing at inlet and exit of the exchanger, respectively. $l^*_{s,I}$ and $l^*_{s,o}$ are the corresponding dimensionless values

MTD mean temperature difference

m factor used to calculate R_{fin}

N_b number of baffles in exchanger

N_c number of tube rows crossed during flow through one crossflow section

N_{cw}	number of effective cross flow rows in each window section
N_f	number of fins per inch
N_p	number of pass partition lanes through the tube field parallel to the direction of the crossflow
N_R	number of rows in an air-cooled exchanger
N_{ss}	number of sealing strips or equivalent obstructions to bypass flow encountered by the stream in one crossflow section
N_t	total number of tubes in the exchanger
N_{tp}	number of tube passes
Nu	Nusselt number
n, n'	exponents for the relationship between j_i and Re_s and f_i and Re_s, respectively
n_a	factor used to calculate R_{fin}
$\Delta P_{b,i}$	pressure drop during flow across one ideal crossflow section
ΔP_e	pressure drop due to entrance effects
Pr	Prandtl number
ΔP_t	tube side pressure drop
$\Delta P_{w,i}$	pressure drop through one ideal window section
p	tube pitch; distance between centers of nearest tubes in tube layout
p_n	tube pitch normal to flow: distance between centers of adjacent tube rows normal to the flow

p_p	tube pitch parallel to flow: distance between centers of adjacent tube rows in the direction of flow
Q	total heat duty of exchanger
Q_{t1}	intermediate heat transferred in multiple shells in series
r	dimensionless radial span or the distance at which the center of the farthest tube may be located from the center of the shell, in order to maintain the minimum clearance
R_b	correction factor for effect of bundle bypass on pressure drop
Re	Reynolds number
Re_s	Reynolds number, shell side of exchanger
R_{fa}	fouling resistance for air
R_{fi}	fouling resistance to heat transfer on tube side
R_{fin}	fin resistance to heat transfer
R_{fo}	fouling resistance to heat transfer on tube side and shell side, respectively
R_l	correction factor for effect of baffle leakage on pressure
R_s	correction factor for effect of unequal baffle spacing
S	space between fins, in.
S_m	crossflow area at or near centerline for one crossflow section
S_{sb}	shell-to-baffle leakage area for one baffle

S_{tb}	tube-to-baffle leakage for one baffle
S_w	area for flow through window
S_{wg}	window gross area
S_{wt}	window area occupied by tubes
T_1, T_2	inlet and outlet temperature, hot fluid
t_1, t_2	inlet and outlet temperature, cold fluid
t_{c1}, t_{c2}	cold fluid intermediate temperatures in multiple shells in series
t_{h_1}	intermediate hot temperature in multiple shells in series
U_o	overall heat transfer coefficient based on shell-side heat transfer area
u, u_1, u_2	dimensionless dummy variables
V	velocity of fluid
V_a	air velocity, ft/hr
v	dimensionless dummy variable
W, w	mass flow rate of hot fluid and cold fluid, respectively
W_{shell}	mass flow rate of shell side
w_1, w_2	dimensionless dummy variable
w_p	width of pass partition clearance in tube field
Δx	wall thickness
Y	mean fin thickness, in.
z, z^*, Z_1, Z_2	dimensionless dummy variables
α	1.7 for air-coolers CLMTD1 calculation
β	1.7 for air-coolers CLMTD1 calculation
δ_{sb}	diametral clearance between shell and baffle

δ_{tb}	diametral clearance between tube and baffle
μ	viscosity of fluid
μ_a	viscosity of air, lb/ft hr
μ_{shell}	viscosity of shell-side fluid at bulk stream temperature
$\mu_{shell, w}$	viscosity of shell-side fluid at wall temperature
ρ	density of fluid
ρ_a	air density, lb/ft^3
ρ_{shell}	density of shell-side fluid
Θ	baffle cut angle, radians

CHAPTER I

INTRODUCTION

Many times for preliminary design purposes a general idea of the size and configuration of the heat exchanger is all that is required. The HEXSIM simulator is designed to meet this need. HEXSIM was written to fill the gap between hand calculations and the massive detailed heat exchanger design simulators. HEXSIM provides basic information on the size and configuration of shell and tube and air-cooled heat exchangers for sensible heat transfer problems. HEXSIM is implemented in an interactive user-friendly mode to allow any input parameter modifications from the screen.

For shell and tube heat exchangers, HEXSIM has the following capabilities:

1. Calculates and checks heat balance
2. Calculates log mean temperature difference and correction factor
3. Calculates overall area
4. Calculates feasible inside shell diameters, outer tube bundle limits, tube lengths, tube counts, and length to diameter ratios for a given area.
5. Calculates individual and overall heat transfer coefficients
6. Calculates pressure drops on both the shell side and tube side.

For air-cooled heat exchangers, HEXSIM is capable of calculating the following items:

1. Heat balance
2. Air temperature and pressure at exit
3. Pressure drop of both air side and tube side
4. Individual and overall heat transfer coefficients
5. Log mean temperature difference and configuration correction factor
6. Area of the air cooler
7. Number of tubes required
8. Bay width and length
9. Number, diameter, and power requirements of each fan
10. Air volume per fan
11. Air velocity at fan face and tube face

A. Methods of Calculation

A.1. Shell and Tube Exchangers

Standard heat transfer equations were used to calculate the overall heat transfer coefficient, log mean temperature difference, pressure drop, heat balance, and the overall area. The shell diameter, outer tube limit, and length to diameter ratios were determined using the method described by Bell (1). The Delaware method (2,3,4) is the solution of shell side flow pressure drop and heat transfer coefficient. The tube side heat transfer coefficient is calculated using the Hausen equation for laminar flow (Re<2000) and the Sieder-Tate equation for turbulent flow (Re>10,000). Interpolation is used in the transition region (2000<Re<10,000).

A.2. Air-Cooled Exchangers

The fin dimensions and areas are calculated using methods given in Bell's Process Heat Transfer Notes (4). The heat duty, and pressure drop for the tube side fluid and log mean temperature difference are obtained using standard heat transfer methods. The outlet air temperature, exchanger area, exchanger dimensions, number of tubes, number and power of each fan are calculated using methods given in the GPSA manual (6). The tube side heat transfer coefficient was calculated using the Hausen equation for laminar flow, and the Sieder-Tate equation for turbulent flow. Interpolation was used for transition region. The air-side heat transfer coefficient and pressure drop were calculated using the methods given in the GPSA Engineering Data Book (6).

CHAPTER II

PROGRAM HISTORY

The HEXSIM simulator was programemd by Dr. John Erbar between 1981 and 1984 using the methods given by Bell in his Process Heat Transfer notes (4) for shell and tube exchangers and using the GPSA Engineering Data Book for the air-coolers (13). At this time the program was still not operational due to errors in the program. In January of 1986, work was begun to remove all compiler errors. These errors included such things as misspelled variable names, missing statements, data statements with an incorrect number of initializers, and missing equivalence statements, etc. Once this was accomplished comment statements were added for program documentation. The program documentation included adding a short description of the purpose of each subroutine, the equation number for most calculations such as for Delaware method parameters, and in some cases the source used for method of calculation (i.e. Bell, K. J., Process Heat Transfer Notes at the top of the Delaware calculations subroutine).

The next step that was accomplished was to add default values. Default values for the baffle spacing, tube-to-baffle clearance, shell-to-baffle clearance, percent baffle cut, and length of baffle cut were added to the program.

Also, warning statements were added for the baffle cut and baffle spacing to follow TEMA class R construction (13). The diagnostic statement for "error in heat balance > 1%" did not originally repeat the inputs given so diagnosing the cause of the problem was difficult. As part of this work, statements were added so that the inputs were repeated. A warning statement and corrective action for a low configuration correction factor for air-coolers was programmed. Also, a check and warning statement were added for the face velocity of an air-cooler.

Several major changes were made to the shell and tube exchanger calculations. As originally programmed, if multiple shells in series were required the output was given in terms of one exchanger for the preliminary calculations. The program was modified so that the output is given in terms of each identical individual shell. The order of subroutine calls in the preliminary calculations program caused the input to be requested when the program was being restarted. This problem was corrected. There was a misspelled variable name in the main Delaware calculation which caused many problems. The subroutine which calculates the shell side heat transfer coefficient had some equations which were incorrect. In the original version the outlet temperatures were always adjusted to meet the length of the exchanger in the Delaware calculations. The program was modified so that the user was given the choice of having the outlet temperature set and the length calculated or the length set and the outlet temperature calculated. This modification was very complicated due to the way in which the program was initially programmed. The curve fit for f_i, the ideal tube bank friction factor was replaced since the original fit was not as close as possible. The curve fit for, j_i the ideal heat transfer coefficient also was replaced since the original fit was not stable. The tube side friction factor function routine was corrected so that a friction factor was calculated for laminar flow. Shell diameters and outer tube limit diameters were added. Checks were added so that negative logarithmic mean temperature differences are not allowed. The tube count subroutine had to be replaced since the original routine was set up only to calculate tube counts for 3/4 inch tubes and 1 inch tubes with only a few tube pitches. The tube count routine was programmed using the method given by Phadke (10) which allows any tube diameter and pitch. The calculation of the tube side and shell side velocity was added to the program.

Changes were made to the EDIT section in order to increase the ease of usage. A subroutine was completed so that all intermediate Delaware parameters could be printed every time the shell and tube exchangers are run. EDIT commands were added to start a new problem calculation, and so that summary sheets for each calculation could be repeated. Several EDIT commands were missing which were added to the program.

The major addition to the air-cooled heat exchanger calculations was a subroutine to calculate a configuration correction factor. The method was given by Pigorini (11). In the main air cooler program, the pressure drop is calculated twice: the first is an estimate and the second is calculated using all the exchanger geometry. In the original program the first pressure drop calculated was printed out which made the output inconsistent.

The program units routine also required some modifications. The subroutine which checks the units system after a restart was not properly functioning since all the formats and write statements were missing. The viscosity units were originally only lb/ft-hr and kg/m-hr. This was changed so that the user could use viscosity units of lb/ft-hr, kg/m-hr, and centipoise.

The main program was mofified so that after each calculation the user was prompted to the EDIT mode. The program was also modified so that files could be stored and retrieved on the IBM/TSO system.

The most important step taken in the programing phase of this research was to thoroughly check the equations and logic of HEXSIM. The program was checked by inspection of the source code and by running and checking test problems as well as hand calculations.

CHAPTER III

LITERATURE REVIEW

There are many types of programs written for heat exchanger design at various levels of sophistication. The programs that are relatively simple and those that are the most sophisticated seem to be of general interest. Simple programs such as HEXSIM are typically used for an overall process simulation or an economic study. These types of programs typically run in a fraction of a second on a large-scale computer, therefore, general trends and the results can be used to select the cases to be run on more sophisticated programs.

At the other extreme are highly sophisticated programs that model the actual geometric configuration as precisely as possible. This type of program requires many more inputs and takes much longer to run than a simple program. These programs are able to simulate accurately the influence of independent factors so that parametric investigations can be made.

Heat exchanger design programs whether simple or complex fall into four major categories. These are thermal and hydraulic design programs, thermal and hydraulic rating programs, mechanical design programs, and economic comparison (8).

A. Thermal and Hydraulic Design Programs

These programs are the basic tools of heat exchanger design. They are used to determine geometry and performance for a given process condition. Design programs typically determine the minimum area that will provide the surface required to transfer a given heat duty and satisfy the pressure drop specifications.

B. Thermal and Hydraulic Rating Programs

These programs evaluate the thermal and hydraulic performance of an exchanger of a given geometry, assuming either a known or unknown heat duty. These programs are used to predict the performance of an existing or specified unit and evaluate them at alternative design conditions.

C. Mechanical Design Programs

A separate program usually performs the actual mechanical design on a heat exchanger. The sizing information from the thermal and hydrualic design, as well as material properties, codes and standards specifications form the input to the program. The program then determines the construction details, tube-field layouts, tabulates material take-offs, etc.

D. Cost-Evaluation Programs

These programs evaluate the economic problems associated with the initial cost, operation, and maintenance of shell- and tube-exchangers. These programs typically use some form of optimization technique to seek a lowest cost solution. They usually require the input of many economic factors to reflect the current and future situation of the use the exchanger. The formulation of the desired optimizing function is usually very complex and specific to each particular problem.

There are many programs written to perform the calculation required for thermal and hydraulic design, and thermal and hydraulic rating for sensible heat transfer. The following paragraphs contain a short description for some of the programs available. A more complete listing is given by Peterson (9) in which all major categories of exchanger design programs are listed, as well as, most other unit operations.

E. ACOL by AERE Harwell

ACOL models the performance of air-cooled heat exchangers (9). The program computes the performance of air-cooled heat exchangers used to: 1) cool single phase liquids, 2) cool single phase gases, 3) condense vapors with or without noncondensables. Subcooling and desuperheating can be handled by the program although the predictions are on the safe side.

F. Air-Fin (No Phase Change) Program 9286 by Phillips Petroleum Company

Heat exchanger program 9286 will design or rate either induced- or forced-draft air-fin exchangers (9). Program 9286 can be used only with single phase streams. Using the rating option, the program solves for the following (for an otherwise fully-described exchanger):

1. Minimum air volume required for a given-sized unit to perform the given heat duty.
2. Maximum process flow rate that can be used for each of several ambient air-temperature and air-flow rate combinations.

3. Minimum process-outlet temperature and corresponding air-outlet temperature that can be obtained in a given size unit with specified flow rates.

G. HTEX2 by A. M. Kinney Inc.

HTEX2 computes design parameters of heat exchangers for three cases (9): 1) no phase change, 2) condensing a vapor from a noncondensable gas, 3) pure condensation. The program is applicable for both horizontal and vertical condensers. The input required will depend on the type of system specified. The common inputs to the three sytems are ratio of pitch/outside diameter, maximum shell inside diameter, percentage baffle cut, shell side and tube side fouling factors, number of tube passes, outside tube diameter, tube length, tube thickness, and maximum shell side and tube side pressure drops. The output consists of shell surface area, log mean temperature difference, heat transfer coefficients, shell side and tube side pressure drops, velocities, channel side and shell side factors, and shell and tube side characteristics.

H. Shell and Tube Heat Exchange (No Phase Change)
Program 9282 by Phillips Petroleum Company

Program 9282 designs or rates a shell and tube exchanger used in heating or cooling services with no phase change (9). Program 9282 accepts plain tubes or finned tubes, and the use of turbulency promoters inside tubes. It also evaluates either conventional-flow or split-flow on the shell-side. Design solves for size, number and arrangement of the shell, and all essential construction details subject to limitations imposed by input data. The criteria used in design are to first minimize the number of shells, then minimize shell diameter, then minimize the length. Rating solves for any one of the following: 1) both flow rates, 2) both outlet temperatures, 3) certain combinations of flow rates and outlet temperatures, 4) total area required for a given duty, and 5) maximum allowable fouling for a given duty.

I. SHELLI by PFR Engineering, Inc.

SHELLI evaluates shell and tube heat exchangers (9). SHELLI includes accurate methods for shell and tube side calculations for sensible heaters or coolers, condensers, or reboilers. It may be used for the design of exchangers or for the evauluation of performance of existing or proposed exchangers. A heat release or enthalpy profile can be input to describe either or both hot and cold fluids.

J. STEP: Performance and Design of Shell and Tube Heat Exchangers with Single Phase Flow by AERE Harwell

The objective of STEP is to provide an inexpensive, though comprehensive, design tool for optimized exchangers with known geometry (9). The program's STEP3 and STEP4 carry out a systematic investigation of solutions to the specified heat transfer performance, within the limits of specified tube side and shell side pressure drop, by examining cases in which the following variables are treated sequentially:

1. tube length
2. numbers of shell in series and/or parallel
3. number of tube side passes

For design, the following types of input information are required:

1. physical property data
2. costing data
3. dimensional data
4. thermal data

K. HEXNET: Heat Exchanger Network by Profimatics, Inc.

HEXNET is used for the solution of heat-transfer problems that involve a complex arrangement of heat exchangers (9). Four exchanger services (liquid-liquid, boiler, condensers, and reboilers) are provided. The exchanger pass configuration is specified as 1-1,1-2, or 1-4. Changes in heat-transfer coefficients resulting from flow changes are accounted for. Input data includes inlet flows and properties, exchanger parameters, and equipment configuration.

L. ST-4 by HTRI

ST-4 program primarily handles no phase change fluids on both the tube side and shell side flowing in laminar, transitional, and turbulent regimes on both shell side and tube side (9). In the design mode, ST-4 program designs the minimum number of shells in series and/or parallel of the smallest diameter that will satisfy the process conditions while respecting an optional set of constraints given by the user.

ST-4 also handles known heat duty rating cases. The inputs required are shell diameter, central baffle spacing, and number of tube passes. All other geometry is calculated or set according to TEMA (13) standards. The final result is the percent over design and the differential resistance based on the difference between the actual heat duty and the required heat duty satisfying the given process conditions.

For unknown heat duty rating cases, the program calculates the two missing process conditions so that the expected performance of the exchanger will match exactly the required heat duty.

CHAPTER IV

DISCUSSION

In this chapter, the design methods for a process heat exchanger are described. The HEXSIM simulator uses the methods described in this chapter to make an estimate of the size and configuration of an exchanger quickly with a limited amount of information. The purpose of this chapter is to document the methods used in the simulation. The first half of the discussion section deals with shell and tube exchangers while the last half discusses air-cooled exchangers.

A. Shell and Tube Preliminary Calculation Methods

A.1. Basic Structure of Preliminary Calculations

The need for a preliminary estimate of the heat exchanger size is often a very useful first step in obtaining an exact design. The information required to make a preliminary estimate of the size of the exchanger is:

1. Flow rates of shell side and tube side fluids
2. Inlet and outlet temperatures for shell side and tube side fluids
3. Specific heat of each fluid
4. Number of tube and shell passes
5. Tube outside diameter, inside diameter, and pitch
6. Thermal conductivity of tube metal

7. Tube arrangement and type of bundle construction

8. Estimated individual heat transfer coefficients and fouling factors

Using the above information the area, length, shell diameter, and tube count can be calculated.

A.2. The Basic Design Equation

The basic design equation to be used in this section is

$$A_o = \frac{Q}{U_o\ (MTD)} = \frac{Q}{U_o\ F_t\ LMTD} \tag{1}$$

where

A_o = is the total heat transfer area required in the exchanger

Q = is the total amount of heat transferred by the exchanger

U_o = is the overall heat transfer coefficient

LMTD = is the logarithmic mean temperature difference calculated for countercurrent flow

F_t = is the configuration correction factor

It should be noted that the validity of equation 1 is dependent upon a number of assumptions. The assumptions are as follows:

1. All elements of a given stream have the same thermal history.
2. The heat exchanger is at steady state.
3. Each stream has a constant specific heat.
4. The overall heat transfer coefficient is constant.
5. The flow is either entirely cocurrent or entirely countercurrent.

6. The heat exchanger does not exchange heat with the surroundings.

These conditions are often not completely met but most of the departures from the above assumptions introduce smaller errors than the probable error in the other approximations inherent in the method.

A.3. Estimation of Heat Load

The heat load can be quickly calculated for the sensible heat transfer case from:

$$Q = W * C_p * (T_1 - T_2) = w * c_p * (t_2 - t_1) \tag{2}$$

where W and w are the mass flow rates of hot and cold fluids, respectively, C_p and c_p the specific heats hot and cold fluid, T_1 and T_2 the inlet and outlet temperatures of the hot stream, and t_1 and t_2 the inlet and outlet temperatures of the cold stream.

A.4. Estimation of the Mean Temperature Difference

The first step in calculating the mean temperature difference (MTD) is to find the logarithmic mean temperature difference (LMTD) for counter-current flow. The LMTD is calculated from the following equation.

$$\mathrm{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\frac{T_1 - t_2}{T_2 - t_1}\right)} \tag{3}$$

where

T_1 = inlet temperature of the hot stream

T_2 = outlet temperature of the hot stream

t_1 = inlet temperature of the cold stream

t_2 = outlet temperature of the cold stream

If the term $|T_1-t_2|/|T_2-t_1|$ is equal to or less than 0.2, the arithmetic mean temperature difference (AMTD) is used. The arithmetic mean temperature difference is

$$AMTD = 1/2\ [(T_1 - t_2) + (T_2 - t_1)] \tag{4}$$

The LMTD is always equal to or less than the AMTD. The difference between the LMTD and AMTD increases with an increasing ratio of $|T_1-t_2|$ and $|T_2-t_1|$.

The calculation of F_t,the configuration correction factor, is more complicated, since it requires the use of charts or curve fits. To find F_t use Figure 1 or see Appendix B for the curve fit. The method used for calculating F_t is that given by Bowman (5). F_t should be lower than 0.8. If the value of F_t is lower than 0.8, this is an indication that the thermodynamic feasibility of the design should be checked before proceeding further. One way to increase F_t is to use multiple shells in series.

A.5. Estimation of Number of Shells in Series

There is a rapid graphical technique for estimating a sufficient number of shells in series. This method can be easily programmed. The procedure is shown in Figure 2. The method goes as follows (1).

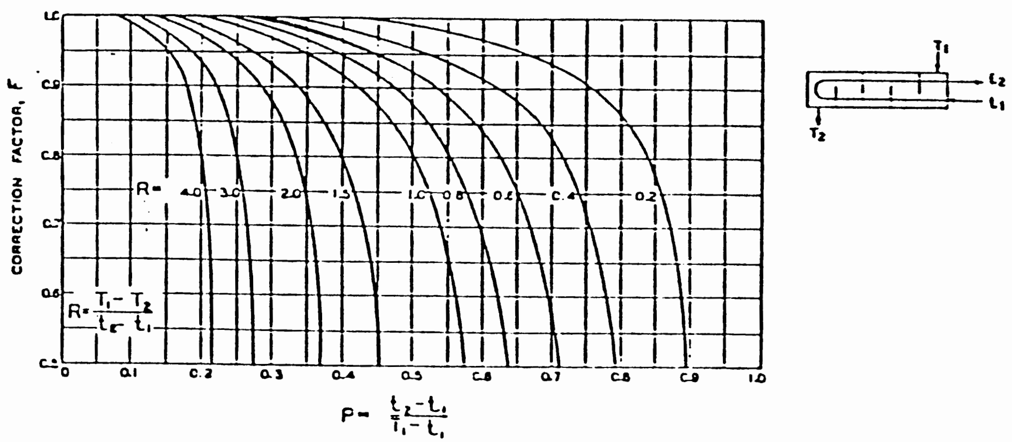


Figure 1. LMTD Correction Factor For an Even Number of Tube Passes (5)

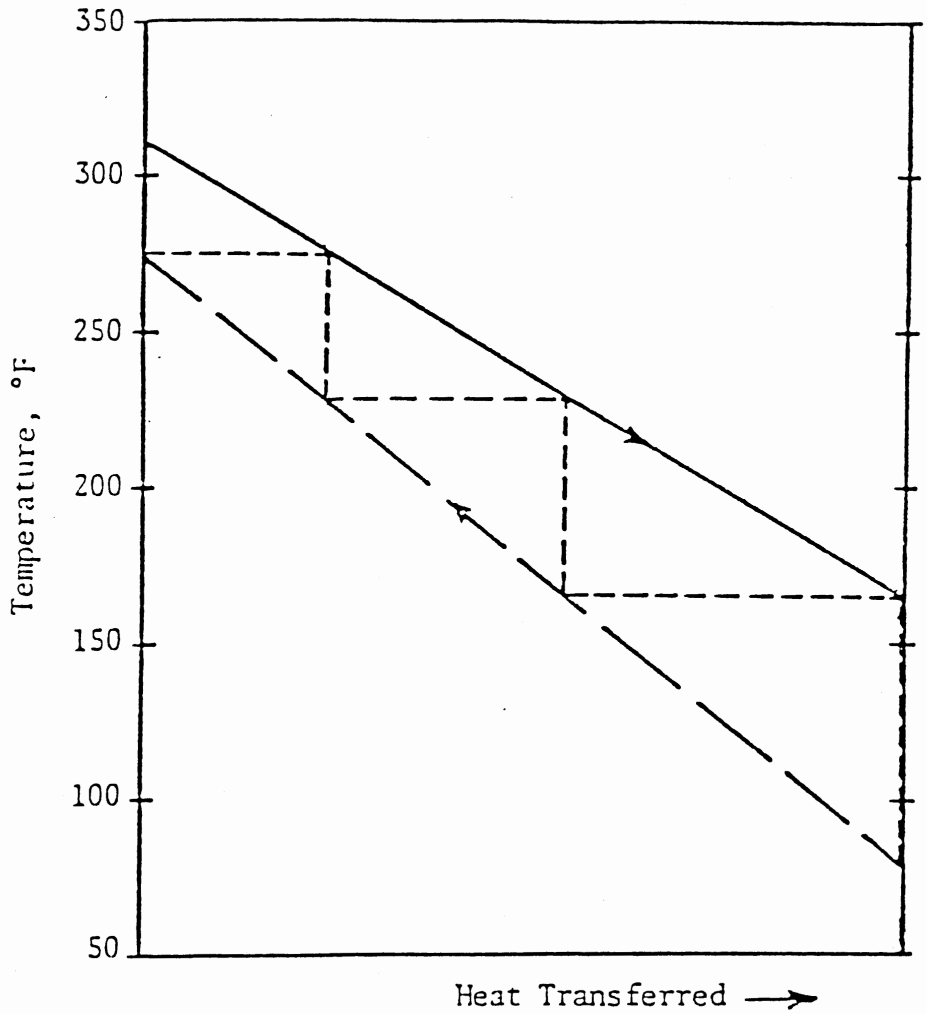


Figure 2. Estimation of Required Number of Shells in Series (4)

a. The terminal temperatures of the two streams are plotted on the ordinates of ordinary arithmetic graph paper, the hot fluid inlet temperature and the cold fluid outlet temperature on the left-hand ordinate and the hot fluid outlet and cold fluid inlet on the right-hand ordinate. The distance between them is arbitrary, but should correspond to the total amount of heat transferred.

b. For a constant specific heat, straight lines are drawn from the inlet to the outlet temperature for each stream.

c. Starting with the cold fluid outlet temperature, a horizontal line is laid off until it intersects the hot fluid line. From that point a vertical line is dropped to the cold fluid line. This operation defines a heat exchanger in which the hot fluid temperature is never less than any temperature reached by the cold fluid; therefore, there can be no temperature cross.

d. The process is repeated until a vertical line intercepts the cold fluid operating line at or below the cold fluid inlet temperature.

e. The number of horizontal lines is equal to the number of shells in series that is clearly sufficient to perform the duty. In the example shown the number is 3.

To program the graphical method shown in Figure 2 the following steps are taken. First the horizontal line is found by the following equation.

$$Q_{t_1} = W * C_p * (T_{h_1} - t_{c_2}) \tag{5}$$

and cold fluid temperature (or vertical line down) is found from the equation (6).

$$t_{c_1} = t_{c_2} - Q_{t_1}/(c_p * w) \tag{6}$$

Then the number of shells in series is increased by one. This process is continued until t_{c_1} is less than or equal to the cold fluid inlet temperature. If t_{c_1} is greater than the cold fluid inlet temperature t_{h_1} is set equal to t_{c_2}, t_{c_2} is set equal to t_{c_1} and the whole process is repeated.

A.6. Calculation of the Overall Heat Transfer Coefficient

The overall heat transfer coefficient, U_o, is built up from the individual resistances to heat transfer. U_o is calculated from the following equation. For the preliminary design calculations estimated heat transfer coefficients are used.

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + \frac{\Delta x}{k_w}\left(\frac{d_o}{d_m}\right) + \left(R_{fi} + \frac{1}{h_i}\right)\frac{d_o}{d_i} \tag{7}$$

where

h_o = the outside (shell side) heat transfer coefficient

R_{fo} = the shell side fouling resistance

Δx = the tube wall thickness

k_w = the tube metal thermal conductivity

d_o = the outside tube diameter

d_i = the inside tube diameter

d_m = the mean wall heat transfer area, usually taken to be the arithmetic mean, $1/2\ (d_o + d_i)$

R_{fi} = the tube side fouling resistance

h_i = the tube side heat transfer coefficient

A.7. Calculation of Individual Heat Transfer Coefficients

The outside (shell side) heat transfer coefficient is calculated using methods given in section B.5.1. It is calculated using the Delaware method for calculating heat transfer coefficient based on stream analysis (2,3,4).

The prediction of an inside (tube side) heat transfer coefficient is strongly dependent upon the flow regime. The flow regime that exists in a given flow situation is ordinarily characterized by the Reynolds number. The Reynolds number is generally defined as in the following equation.

$$Re = \frac{d_i\ \rho\ V}{\mu} \tag{8}$$

where

d_i = the inside tube diameter

ρ = the density of the fluid

μ = the viscosity of the fluid

V = the tube side fluid velocity

Reynolds numbers below 2,100 results in stable laminar flow. Reynolds numbers above 10,000 give turbulent flow for heat transfer. The range between 2,100 and 10,000 is generally referred to as transition flow.

In laminar flow, the heat transfer coefficient is calculated from the Hausen equation (4). The Hausen equation is:

$$\frac{h_i\ d_i}{k} = [3.65 + \frac{0.0668\ Re\ Pr\ (d_i/L)}{1 + 0.04\ [Re\ Pr\ (d_i/L)]^{2/3}}]\ (\frac{\mu}{\mu_w})^{0.14} \tag{9}$$

where

h_i = inside (tube side) heat transfer coefficient

d_i = inside tube diameter

k = thermal conductivity of tube side fluid

Re = Reynolds number

Pr = Prandtl number = $\frac{c_p\ \mu}{k}$

c_p = is specific heat

μ = viscosity

k = thermal conductivity of the fluid

L = tube length

The Sieder-Tate equation is used to calculate the heat transfer coefficient in turbulent flow. The Seider-Tate equation has the form

$$\frac{h_i\ d_i}{k} = 0.023\ Re^{0.8}\ Pr^{1/3}\ (\frac{\mu}{\mu_w})^{0.14} \tag{10}$$

where

h_i = inside heat transfer coefficient

d_i = inside tube diameter

k = thermal conductivity

Re = Reynolds number

Pr = Prandtl number

μ = bulk viscosity

μ_w = viscosity at the tube wall

Interpolation is used in the transition region (2100 <Re<10,000). The interpolation is carried out as follows.

$$\overline{Nu}_{laminar} = \frac{h_i\ d_i}{k} = [3.65 + \frac{0.0668\ (2100)\ Pr\ (d_i/L)}{1 + 0.04\ [(2100)\ Pr\ (d_i/L)]^{2/3}}] \tag{11}$$

$$\overline{Nu}_{turb} = \frac{h_i\ d_i}{k} = 0.023\ (10,000)^{0.8}\ Pr^{1/3}\ (\frac{\mu}{\mu_w})^{0.14} \tag{12}$$

$$\overline{Nu}_{trans} = \overline{Nu}_{laminar} + [(Re - 2100)/7900] * (Nu_{turb} - Nu_{laminar}) \tag{13}$$

A.8. Calculation of Heat Transfer Area, Ao

Once Q, MTD, and U_o are known the total outside area can be calculated from equation (1).

A.9. Estimation of Major Exchanger Parameters

A.9.1 Heat Transfer Area for a Given Shell Diameter and Length. Once the area required is known then combinations of tube length and shell diameter can be calculated. First the shell diameters and diameter of the outer tube limit are given in a data statement. The inside shell diameters range from 8 to 120 inches. The shell diameter and outer tube limit are those for a conventional split ring floating head design fully tubed out. Shell ouside diameters and outer tube limits are given in Table I. Next the tube count possible for a given

TABLE I

SHELL DIAMETER AND OUTER TUBE DIAMETER
LIMITS USED IN HEXSIM (4)

Nominal Shell Inside Diameter, D_i, in	Outer Tube Diameter Limit, D_{otl}, in.
8.071	6.821
10.020	8.770
12.000	10.750
13.250	12.000
15.250	14.000
17.250	16.000
19.250	18.000
21.000	19.250
23.250	21.500
25.000	23.375
27.000	25.375
29.000	27.375
31.000	29.375
33.000	31.375
35.000	33.375
37.000	35.250
39.000	37.250
42.000	40.250
44.000	42.250
48.000	46.000
52.000	50.000
56.000	54.000
60.000	58.000
66.000	64.000
72.000	70.000
78.000	76.000
84.000	82.000
90.000	88.000
96.000	94.000
108.000	106.000
120.000	118.000

inside shell diameter, outer tube limit, tube arrangement, tube outside diameter, tube pitch, and number of tube passes is calculated. The method used to calculate the number of tubes is given in the next section. With the number of tubes known the length of the exchanger can be calculated from the following equation.

$$L = \frac{A_o}{\pi \, d_o \, N_t} \tag{14}$$

where

L = tube length

A_o = the total heat transfer area required in the exchanger

d_o = the tube outside diameter

N_t = the total number of tubes

Next the length to diameter ratio is calculated. The length to diameter ratio is the ratio of the length of the exchanger to the inside shell diameter. Shells shorter than three times the shell diameter may suffer from poor fluid distribution and excessive entry and exit losses, and are to be more likely expensive than a longer, smaller diameter unit. Shells longer than 15 times the shell diameter are likely to be difficult to handle mechanically, require a large clearway for bundle removal and are not as cost effective as shorter exchangers. For this reason only shell diameters which fall between 3:1 and 15:1 are ordinarily considered.

A.9.2 Estimation of Tube Count. The number of tubes that can be accommodated in a shell of a given inside diameter is known as the tube count. The tube count depends on the inside shell diameter, outer tube diameter limit, tube outside diameter, tube pitch, tube arrangement, and the number of exchanger passes.

The technique used to calculate the tube count is given by Phadke (10). The technique is based on number theory. The tube counts obtained for single-pass exchangers are exactly as calculated by the theory. However for 2, 4, 6, and 8 pass arrangements, a certain number of tubes will have to be removed to accommodate the pass partition plates. For these cases, the following assumptions have been made:

1. The pass partition plate is located where a row of tubes would have been, and only one row, or at the most two, will be affected.
2. The thickness of the partition plate is less than 70% of the tube outside diameter.
3. The distance between the centerline of the partition plate and the center line of the nearest row of tubes is equal to the pitch.

For a single pass exchanger, first calculate the two basic dimensionless quantities:

$$r = \frac{0.5\ (D_i - d_o) - c}{p} \tag{15}$$

where

D_i = shell inside diameter, in or mm

d_o = tube diameter, in or mm

c = minimum clearance between the outermost tubes and the inside of the shell, in or mm

p = tube pitch, in or mm

$$s = r^2 \tag{16}$$

Then determine N_r and N_s, the largest integers equal to or less than r and s, respectively. Locate in the N_s column of the appropriate table (Table II or III) the corresponding C_1 value which is the tube count for a single tube pass.

Triangular layouts: 2- or 4-pass

First calculate the following

$$w = \frac{2r}{\sqrt{3}} \tag{17}$$

Then calculate the corresponding integer N_w as well as C_x and C_y by the following equations:

$$C_x = 2\ N_r + 1 \tag{18}$$

$$C_y = 2\ N_w \qquad \text{if } N_w \text{ is even} \tag{19a}$$

$$C_y = 2\ N_w + 1 \qquad \text{if } N_w \text{ is odd} \tag{19b}$$

For triangular 2-pass layouts:

$$C_2 = C_1 - C_x \tag{20}$$

For triangular 4-pass layouts:

TABLE II

TRIANGULAR LAYOUTS TUBE COUNT TABLE (10)

N_s	C_1	N_s	C_1	N_s	C_1
1	7	45	165	89	328
2	10	46	166	90	331
3	13	47	167	91	337
4	19	48	169	92	343
5	23	49	187	93	349
6	27	50	191	94	352
7	31	51	195	95	355
8	34	52	199	96	358
9	37	53	201	97	361
10	39	54	204	98	363
11	41	55	206	99	365
12	43	56	208	100	367
13	55	57	211	101	371
14	57	58	214	102	375
15	59	59	217	103	379
16	61	60	220	104	380
17	65	61	223	105	381
18	69	62	229	106	382
19	73	63	235	107	384
20	79	64	241	108	385
21	85	65	245	109	397
22	87	66	249	110	403
23	88	67	253	111	409
24	90	68	255	112	421
25	91	69	257	113	423
26	94	70	259	114	426
27	97	71	261	115	428
28	109	72	263	116	431
29	113	73	265	117	433
30	117	74	268	118	435
31	121	75	271	119	436
32	122	76	283	120	438
33	123	77	287	121	439
34	124	78	291	122	443
35	125	79	295	123	447
36	127	80	298	124	451
37	139	81	301	125	455
38	145	82	305	126	459
39	151	83	309	127	463
40	154	84	313	128	469
41	157	85	316	129	475
42	160	86	319	130	481
43	163	87	322	131	487
44	164	88	325	132	493

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
133	499	177	653	221	811
134	501	178	655	222	814
135	503	179	657	223	817
136	505	180	659	224	820
137	507	181	661	225	823
138	509	182	667	226	827
139	511	183	673	227	831
140	512	184	675	228	835
141	513	185	677	229	847
142	515	186	679	230	849
143	516	187	681	231	850
144	517	188	683	232	851
145	523	189	685	233	853
146	529	190	687	234	854
147	535	191	689	235	856
148	547	192	691	236	858
149	551	193	703	237	859
150	555	194	709	238	862
151	559	195	715	239	865
152	561	196	721	240	868
153	564	197	725	241	871
154	566	198	729	242	874
155	569	199	733	243	877
156	571	200	739	244	889
157	583	201	745	245	897
158	585	202	747	246	905
159	587	203	748	247	913
160	589	204	750	248	915
161	591	205	752	249	918
162	593	206	754	250	920
163	595	207	755	251	923
164	598	208	757	252	925
165	601	209	761	253	927
166	604	210	765	254	928
167	607	211	769	255	930
168	610	212	773	256	931
169	613	213	777	257	939
170	619	214	781	258	947
171	625	215	785	259	955
172	637	216	789	260	956
173	641	217	793	261	958
174	645	218	799	262	959
175	649	219	805	263	960
176	651	220	808	264	961

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
265	962	309	1135	353	1288
266	964	310	1138	354	1290
267	965	311	1141	355	1292
268	967	312	1144	356	1294
269	971	313	1147	357	1295
270	975	314	1151	358	1297
271	979	315	1155	359	1299
272	1003	316	1159	360	1301
273	1005	317	1159	361	1303
274	1008	318	1160	362	1306
275	1010	319	1161	363	1309
276	1013	320	1162	364	1333
277	1015	321	1162	365	1337
278	1021	322	1163	366	1341
279	1027	323	1164	367	1345
280	1030	324	1165	368	1347
281	1033	325	1177	369	1350
282	1036	326	1183	370	1352
283	1039	327	1189	371	1354
284	1040	328	1192	372	1357
285	1041	329	1195	373	1369
286	1042	330	1198	374	1371
287	1043	331	1201	375	1373
288	1044	332	1207	376	1375
289	1045	333	1213	377	1377
290	1051	334	1217	378	1379
291	1057	335	1221	379	1381
292	1069	336	1225	380	1387
293	1070	337	1237	381	1393
294	1071	338	1241	382	1395
295	1072	339	1245	383	1397
296	1073	340	1249	384	1399
297	1073	341	1253	385	1401
298	1074	342	1257	386	1403
299	1074	343	1261	387	1405
300	1075	344	1263	388	1417
301	1099	345	1265	389	1418
302	1103	346	1267	390	1420
303	1107	347	1269	391	1421
304	1111	348	1271	392	1422
305	1115	349	1273	393	1424
306	1119	350	1279	394	1425
307	1123	351	1285	395	1426
308	1129	352	1286	396	1428

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
397	1429	441	1615	485	1769
398	1441	442	1619	486	1773
399	1453	443	1623	487	1777
400	1459	444	1627	488	1783
401	1467	445	1630	489	1789
402	1475	446	1633	490	1791
403	1483	447	1636	491	1792
404	1485	448	1639	492	1794
405	1487	449	1641	493	1796
406	1489	450	1644	494	1798
407	1491	451	1646	495	1799
408	1493	452	1649	496	1801
409	1495	453	1651	497	1805
410	1499	454	1654	498	1809
411	1503	455	1657	499	1813
412	1507	456	1660	500	1815
413	1509	457	1663	501	1818
414	1512	458	1665	502	1820
415	1514	459	1667	503	1822
416	1517	460	1669	504	1825
417	1519	461	1671	505	1827
418	1522	462	1673	506	1829
419	1525	463	1675	507	1831
420	1529	464	1677	508	1843
421	1531	465	1680	509	1851
422	1535	466	1682	510	1859
423	1539	467	1685	511	1867
424	1543	468	1687	512	1873
425	1547	469	1711	513	1879
426	1551	470	1717	514	1883
427	1555	471	1723	515	1889
428	1556	472	1726	516	1891
429	1557	473	1729	517	1893
430	1558	474	1732	518	1894
431	1559	475	1735	519	1896
432	1561	476	1739	520	1898
433	1567	477	1743	521	1900
434	1573	478	1747	522	1901
435	1579	479	1751	523	1903
436	1585	480	1755	524	1909
437	1589	481	1759	525	1915
438	1593	482	1761	526	1917
439	1597	483	1763	527	1918
440	1606	484	1765	528	1920

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
529	1921	573	2079	617	2241
530	1929	574	2080	618	2243
531	1937	575	2082	619	2245
532	1945	576	2083	620	2247
533	1946	577	2095	621	2250
534	1948	578	2101	622	2252
535	1949	579	2107	623	2256
536	1950	580	2109	624	2257
537	1951	581	2111	625	2263
538	1952	582	2113	626	2267
539	1954	583	2115	627	2271
540	1955	584	2117	628	2275
541	1957	585	2119	629	2279
542	1963	586	2121	630	2283
543	1969	587	2123	631	2287
544	1972	588	2125	632	2293
545	1975	589	2149	633	2299
546	1978	590	2153	634	2308
547	1981	591	2157	635	2317
548	1987	592	2161	636	2326
549	1993	593	2163	637	2335
550	1999	594	2166	638	2337
551	2005	595	2168	639	2339
552	2011	596	2171	640	2341
553	2017	597	2173	641	2343
554	2021	598	2176	642	2345
555	2025	599	2179	643	2347
556	2029	600	2182	644	2350
557	2037	601	2185	645	2353
558	2045	602	2191	646	2356
559	2053	603	2197	647	2359
560	2054	604	2209	648	2362
561	2056	605	2213	649	2365
562	2057	606	2217	650	2368
563	2059	607	2221	651	2371
564	2060	608	2223	652	2383
565	2062	609	2225	653	2385
566	2063	610	2227	654	2388
567	2065	611	2229	655	2390
568	2068	612	2231	656	2393
569	2071	613	2233	657	2395
570	2074	614	2235	658	2398
571	2077	615	2237	659	2401
572	2078	616	2239	660	2404

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
661	2407	705	2567	749	2723
662	2408	706	2569	750	2724
663	2410	707	2571	751	2725
664	2411	708	2573	752	2727
665	2413	709	2575	753	2730
666	2414	710	2581	754	2732
667	2416	711	2587	755	2735
668	2418	712	2589	756	2737
669	2419	713	2592	757	2749
670	2422	714	2594	758	2753
671	2425	715	2597	759	2757
672	2428	716	2599	760	2761
673	2431	717	2601	761	2765
674	2434	718	2604	762	2769
675	2437	719	2606	763	2773
676	2455	720	2609	764	2774
677	2463	721	2611	765	2775
678	2471	722	2617	766	2777
679	2479	723	2623	767	2778
680	2481	724	2635	768	2779
681	2484	725	2639	769	2791
682	2486	726	2643	770	2795
683	2489	727	2647	771	2799
684	2491	728	2650	772	2803
685	2495	729	2653	773	2807
686	2499	730	2657	774	2811
687	2503	731	2661	775	2814
688	2515	732	2665	776	2827
689	2519	733	2677	777	2839
690	2523	734	2679	778	2841
691	2527	735	2681	779	2844
692	2528	736	2682	780	2846
693	2530	737	2684	781	2848
694	2531	738	2686	782	2850
695	2533	739	2689	783	2852
696	2534	740	2701	784	2857
697	2535	741	2713	785	2861
698	2537	742	2714	786	2865
699	2538	743	2715	787	2869
700	2539	744	2717	788	2873
701	2547	745	2718	789	2877
702	2555	746	2719	790	2881
703	2563	747	2720	791	2885
704	2565	748	2721	792	2889

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
793	2893	837	3049	881	3213
794	2897	838	3050	882	3215
795	2901	839	3052	883	3217
796	2905	840	3053	884	3221
797	2906	841	3055	885	3225
798	2908	842	3059	886	3229
799	2909	843	3063	887	3233
800	2911	844	3067	888	3237
801	2912	845	3071	889	3241
802	2914	846	3075	890	3245
803	2915	847	3079	891	3249
804	2917	848	3085	892	3253
805	2918	849	3091	893	3253
806	2920	850	3094	894	3254
807	2922	851	3097	895	3255
808	2923	852	3100	896	3256
809	2926	853	3103	897	3256
810	2927	854	3105	898	3257
811	2929	855	3107	899	3258
812	2935	856	3109	900	3259
813	2941	857	3111	901	3267
814	2947	858	3113	902	3275
815	2953	859	3115	903	3283
816	2959	860	3115	904	3286
817	2965	861	3116	905	3289
818	2971	862	3117	906	3292
819	2989	863	3118	907	3295
820	2992	864	3119	908	3297
821	2995	865	3119	909	3299
822	2998	866	3120	910	3302
823	3001	867	3121	911	3305
824	3003	868	3145	912	3307
825	3005	869	3153	913	3310
826	3007	870	3161	914	3313
827	3009	871	3169	915	3316
828	3011	872	3175	916	3319
829	3013	873	3181	917	3323
830	3019	874	3185	918	3327
831	3025	875	3189	919	3331
832	3037	876	3193	920	3337
833	3039	877	3205	921	3343
834	3041	878	3207	922	3346
835	3044	879	3209	923	3349
836	3047	880	3211	924	3352

TABLE II (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
925	3353	969	3507		
926	3361	970	3509		
927	3367	971	3510		
928	3376	972	3511		
929	3385	973	3535		
930	3394	974	3541		
931	3403	975	3547		
932	3405	976	3559		
933	3407	977	3561		
934	3409	978	3564		
935	3411	979	3566		
936	3413	980	3569		
937	3415	981	3571		
938	3421	982	3574		
939	3427	983	3578		
940	3428	984	3585		
941	3429	986	3588		
942	3431	987	3591		
943	3432	988	3595		
944	3434	989	3599		
945	3435	990	3603		
946	3436	991	3607		
947	3438	992	3613		
948	3439	993	3619		
949	3463	994	3622		
950	3464	995	3625		
951	3466	996	3628		
952	3467	997	3631		
953	3469	998	3637		
954	3470	999	3643		
955	3472				
956	3473				
957	3475				
958	3476				
959	3478				
960	3479				
961	3481				
962	3485				
963	3489				
964	3493				
965	3497				
966	3501				
967	3505				
968	3506				

TABLE III

SQUARE LAYOUTS TUBE COUNT TABLE (10)

N_s	C_1	N_s	C_1	N_s	C_1
1	5	45	145	89	285
2	9	46	146	90	293
3	11	47	147	91	294
4	13	48	148	92	295
5	21	49	149	93	296
6	22	50	161	94	298
7	24	51	165	95	299
8	25	52	169	96	300
9	29	53	177	97	301
10	37	54	179	98	305
11	40	55	180	99	311
12	43	56	182	100	317
13	45	57	183	101	325
14	46	58	185	102	328
15	48	59	188	103	330
16	49	60	190	104	333
17	57	61	193	105	337
18	61	62	194	106	341
19	65	63	196	107	344
20	69	64	197	108	347
21	71	65	213	109	349
22	74	66	216	110	351
23	76	67	218	111	353
24	79	68	221	112	355
25	81	69	222	113	357
26	89	70	223	114	359
27	92	71	224	115	362
28	94	72	225	116	365
29	97	73	233	117	373
30	98	74	241	118	374
31	100	75	242	119	375
32	101	76	244	120	376
33	105	77	245	121	377
34	109	78	246	122	385
35	111	79	248	123	390
36	113	80	249	124	396
37	121	81	253	125	401
38	124	82	261	126	402
39	127	83	266	127	404
40	129	84	272	128	405
41	137	85	277	129	413
42	139	86	279	130	421
43	141	87	281	131	422
44	143	88	283	132	424

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
133	425	177	559	221	697
134	426	178	561	222	700
135	428	179	565	223	703
136	429	180	569	224	706
137	437	181	577	225	709
138	437	182	581	226	717
139	438	183	585	227	719
140	439	184	589	228	722
141	439	185	593	229	725
142	440	186	594	230	727
143	440	187	595	231	730
144	441	188	596	232	733
145	457	189	597	233	741
146	465	190	598	234	749
147	469	191	599	235	750
148	473	192	600	236	751
149	481	193	601	237	752
150	483	194	609	238	754
151	485	195	611	239	755
152	487	196	613	240	756
153	489	197	621	241	757
154	491	198	625	242	761
155	493	199	629	243	765
156	495	200	633	244	769
157	497	201	637	245	777
158	500	202	641	246	780
159	502	203	646	247	783
160	505	204	652	248	786
161	507	205	657	249	788
162	509	206	659	250	793
163	513	207	662	251	794
164	517	208	665	252	794
165	519	209	667	253	795
166	522	210	669	254	796
167	524	211	671	255	796
168	527	212	673	256	797
169	529	213	674	257	805
170	545	214	676	258	810
171	548	215	677	259	816
172	550	216	678	260	821
173	553	217	680	261	829
174	555	218	681	262	833
175	556	219	686	263	837
176	558	220	692	264	841

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
265	845	309	976	353	1109
266	847	310	978	354	1112
267	849	311	979	355	1114
268	851	312	980	356	1117
269	853	313	981	357	1119
270	856	314	989	358	1121
271	858	315	992	359	1123
272	861	316	994	360	1125
273	865	317	997	361	1129
274	869	318	1000	362	1137
275	872	319	1002	363	1142
276	874	320	1005	364	1148
277	877	321	1006	365	1153
278	879	322	1007	366	1155
279	881	323	1008	367	1157
280	883	324	1009	368	1159
281	885	325	1030	369	1161
282	885	326	1036	370	1177
283	886	327	1039	371	1179
284	887	328	1041	372	1182
285	887	329	1043	373	1185
286	888	330	1044	374	1189
287	888	331	1046	375	1193
288	889	332	1047	376	1197
289	901	333	1049	377	1201
290	917	334	1051	378	1202
291	921	335	1053	379	1203
292	925	336	1055	380	1204
293	933	337	1057	381	1204
294	936	338	1069	382	1205
295	938	339	1077	383	1206
296	941	340	1085	384	1207
297	945	341	1086	385	1208
298	949	342	1088	386	1209
299	951	343	1089	387	1213
300	953	344	1090	388	1217
301	956	345	1092	389	1225
302	957	346	1093	390	1226
303	960	347	1096	391	1228
304	962	348	1098	392	1229
305	965	349	1101	393	1233
306	973	350	1103	394	1237
307	974	351	1105	395	1239
308	975	352	1107	396	1241

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
397	1245	441	1373	485	1533
398	1249	442	1389	486	1535
399	1253	443	1397	487	1538
400	1257	444	1399	488	1541
401	1265	445	1405	489	1545
402	1268	446	1407	490	1549
403	1270	447	1409	491	1554
404	1273	448	1411	492	1559
405	1281	449	1413	493	1565
406	1283	450	1425	494	1567
407	1285	451	1427	495	1569
408	1287	452	1430	496	1571
409	1289	453	1433	497	1574
410	1305	454	1435	498	1576
411	1306	455	1437	499	1578
412	1308	456	1439	500	1581
413	1309	457	1441	501	1584
414	1310	458	1449	502	1587
415	1312	459	1451	503	1591
416	1313	460	1454	504	1594
417	1315	461	1457	505	1597
418	1316	462	1460	506	1599
419	1318	463	1462	507	1601
420	1317	464	1465	508	1603
421	1321	465	1469	509	1605
422	1324	466	1473	510	1606
423	1326	467	1477	511	1608
424	1329	468	1480	512	1609
425	1353	469	1481	513	1613
426	1354	470	1483	514	1617
427	1355	471	1484	515	1620
428	1356	472	1484	516	1622
429	1357	473	1485	517	1625
430	1358	474	1486	518	1628
431	1359	475	1487	519	1630
432	1360	476	1488	520	1633
433	1361	477	1489	521	1641
434	1363	478	1493	522	1649
435	1366	479	1497	523	1649
436	1369	480	1501	524	1650
437	1370	481	1505	525	1651
438	1370	482	1513	526	1651
439	1371	483	1515	527	1652
440	1372	484	1517	528	1652

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
529	1653	573	1791	617	1941
530	1669	574	1792	618	1943
531	1674	575	1792	619	1946
532	1680	576	1793	620	1948
533	1685	577	1801	621	1951
534	1687	578	1813	622	1953
535	1688	579	1821	623	1956
536	1690	580	1829	624	1958
537	1691	581	1831	625	1961
538	1693	582	1833	626	1969
539	1696	583	1835	627	1973
540	1698	584	1837	628	1977
541	1701	585	1853	629	1993
542	1704	586	1861	630	1995
543	1706	587	1862	631	1996
544	1709	588	1864	632	1997
545	1725	589	1865	633	1999
546	1728	590	1866	634	2001
547	1730	591	1868	635	2004
548	1733	592	1869	636	2006
549	1741	593	1877	637	2009
550	1743	594	1880	638	2012
551	1744	595	1882	639	2014
552	1746	596	1885	640	2017
553	1747	597	1887	641	2025
554	1749	598	1888	642	2025
555	1752	599	1890	643	2026
556	1754	600	1891	644	2027
557	1757	601	1893	645	2027
558	1759	602	1895	646	2028
559	1760	603	1897	647	2028
560	1762	604	1899	648	2029
561	1764	605	1901	649	2041
562	1765	606	1904	650	2053
563	1770	607	1907	651	2057
564	1776	608	1910	652	2058
565	1781	609	1914	653	2061
566	1783	610	1917	654	2064
567	1785	611	1921	655	2066
568	1787	612	1925	656	2069
569	1789	613	1933	657	2077
570	1789	614	1935	658	2079
571	1790	615	1937	659	2081
572	1791	616	1939	660	2083

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
661	2085	705	2223	749	2367
662	2087	706	2225	750	2369
663	2088	707	2228	751	2371
664	2089	708	2230	752	2373
665	2091	709	2233	753	2375
666	2093	710	2235	754	2377
667	2094	711	2238	755	2380
668	2095	712	2241	756	2382
669	2096	713	2242	757	2385
670	2098	714	2243	758	2387
671	2099	715	2244	759	2389
672	2100	716	2245	760	2391
673	2101	717	2246	761	2393
674	2109	718	2247	762	2397
675	2115	719	2248	763	2401
676	2121	720	2249	764	2405
677	2129	721	2251	765	2409
678	2134	722	2253	766	2411
679	2140	723	2257	767	2413
680	2145	724	2261	768	2415
681	2148	725	2285	769	2417
682	2151	726	2286	770	2420
683	2155	727	2287	771	2422
684	2158	728	2288	772	2425
685	2161	729	2289	773	2433
686	2165	730	2305	774	2436
687	2169	731	2308	775	2438
688	2173	732	2310	776	2441
689	2177	733	2313	777	2445
690	2180	734	2315	778	2449
691	2182	735	2331	779	2450
692	2185	736	2332	780	2450
693	2188	737	2333	781	2451
694	2191	738	2334	782	2452
695	2195	739	2335	783	2453
696	2198	740	2337	784	2453
697	2201	741	2340	785	2459
698	2209	742	2343	786	2460
699	2212	743	2346	787	2471
700	2214	744	2349	788	2477
701	2217	745	2353	789	2480
702	2219	746	2361	790	2483
703	2220	747	2363	791	2487
704	2222	748	2365	792	2490

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
793	2493	837	2623	881	2765
794	2501	838	2624	882	2769
795	2504	839	2626	883	2770
796	2506	840	2627	884	2785
797	2509	841	2629	885	2788
798	2513	842	2637	886	2790
799	2517	843	2645	887	2793
800	2521	844	2653	888	2796
801	2529	845	2661	889	2798
802	2537	846	2664	890	2801
803	2538	847	2666	891	2802
804	2540	848	2669	892	2803
805	2541	849	2681	893	2804
806	2542	850	2693	894	2805
807	2544	851	2696	895	2806
808	2545	852	2698	896	2807
809	2553	853	2701	897	2808
810	2561	854	2703	898	2809
811	2562	855	2705	899	2811
812	2563	856	2707	900	2812
813	2564	857	2709	901	2837
814	2565	858	2711	902	2840
815	2566	859	2713	903	2842
816	2567	860	2715	904	2845
817	2568	861	2717	905	2861
818	2569	862	2719	906	2863
819	2577	863	2721	907	2865
820	2585	864	2723	908	2867
821	2593	865	2725	909	2869
822	2594	866	2733	910	2871
823	2595	867	2734	911	2872
824	2596	868	2735	912	2874
825	2597	869	2736	913	2875
826	2598	870	2737	914	2877
827	2599	871	2738	915	2881
828	2600	872	2741	916	2885
829	2601	873	2749	917	2886
830	2604	874	2751	918	2888
831	2606	875	2759	919	2889
832	2609	876	2755	920	2890
833	2617	877	2757	921	2892
834	2618	878	2759	922	2893
835	2620	879	2761	923	2901
836	2621	880	2763	924	2909

TABLE III (CONTINUED)

N_s	C_1	N_s	C_1	N_s	C_1
925	2917	969	3053		
926	2920	970	3061		
927	2922	971	3062		
928	2925	972	3064		
929	2933	973	3065		
930	2936	974	3067		
931	2938	975	3068		
932	2941	976	3069		
933	2943	977	3077		
934	2945	978	3080		
935	2947	979	3082		
936	2949	980	3085		
937	2957	981	3093		
938	2959	982	3097		
939	2961	983	3101		
940	2963	984	3105		
941	2965	985	3109		
942	2967	986	3125		
943	2969	987	3126		
944	2971	988	3126		
945	2973	989	3127		
946	2975	990	3128		
947	2977	991	3128		
948	2979	992	3129		
949	2981	993	3130		
950	2983	994	3130		
951	2985	995	3131		
952	2987	996	3132		
953	2989	997	3133		
954	2997	998	3138		
955	2997	999	3144		
956	2998	1000	3149		
957	2999				
958	2999				
959	3000				
960	3000				
961	3001				
962	3017				
963	3021				
964	3025				
965	3041				
966	3042				
967	3044				
968	3045				

$$C_4 = C_1 - C_x - C_y \tag{21}$$

Triangular layouts: 6- or 8-pass

For triangular configurations, calculate:

$$V = \frac{2\ e*r}{\sqrt{3}} + 0.5 \tag{22}$$

where

e = dimensionless constant; 0.265 for 6-pass arrangement and 0.404 for 8-pass arrangements.

Then calculate:

$$u = \frac{\sqrt{3}\ N_v}{2} \tag{23}$$

$$Z = \sqrt{s - u^2} \qquad \text{if } N_v \text{ is even} \tag{24a}$$

$$Z = \sqrt{s - u^2} - 0.5 \qquad \text{if } N_v \text{ is odd} \tag{24b}$$

For a 6-pass layout:

$$C_6 = C_1 - C_y - 4 * N_z - 1 \tag{25}$$

For an 8-pass layout:

$$C_8 = C_1 - C_x - C_y - 4 * N_z \tag{26}$$

Inline Square layouts: 2- or 4-pass

For inline square layouts, first calculate

$$w = \frac{2r}{\sqrt{3}} \tag{27}$$

Then calculate the corresponding interger, N_w; calculate C_x and C_y by the following equations.

$$C_x = 2\ N_r + 1 \tag{28}$$

$$C_y = C_x - 1 \tag{29}$$

For inline square 2-pass layouts:

$$C_2 = C_1 - C_x \tag{30}$$

For inline square 4-pass layouts:

$$C_4 = C_1 - C_x - C_y \tag{31}$$

Inline Square Layouts: 6- or 8- Pass

For inline square layouts, calculate:

$$V = e * r + 0.05 \tag{32}$$

where

e = dimensionless constant: 0.265 for 6 pass arrangements and 0.404 for 8-pass arrangements

$$Z = \sqrt{s - N_v^2} \tag{33}$$

For 6-pass layouts:

$$C_6 = C_1 - C_y - 4 * N_z - 1 \tag{34}$$

For an 8-pass layout:

$$C_8 = C_1 - C_x - C_y - 4 * N_z \tag{35}$$

Rotated Square Layouts: 2-or 4-pass

For rotated square or diamond layouts, let:

$$w = \frac{r}{\sqrt{2}} \tag{36}$$

Then calculate:

$$C_x = 2 N_w + 1 \tag{37}$$

$$C_y = C_x - 1 \tag{38}$$

For rotated square 2-pass layouts:

$$C_2 = C_1 - C_x \tag{39}$$

For rotated square 4-pass layouts:

$$C_4 = C_1 - C_x - C_y \tag{40}$$

Rotated Square: 6- or 8-pass

For rotated square configurations, calculate:

$$V = \sqrt{2}\, e * r \tag{41}$$

where

e = dimensionless constant: 0.265 for 6-pass arrangements and 0.404 for 8-pass arrangements

Then find:

$$u_1 = \frac{N_v}{\sqrt{2}} \tag{42}$$

$$Z = \sqrt{s - u_1^{\,2}} \tag{43}$$

$$w_1 = \sqrt{2}\, z \tag{44}$$

$$u_2 = \frac{N_v + 1}{\sqrt{2}} \tag{45}$$

$$Z^* = \sqrt{s - u_2^2} \tag{46}$$

$$w_2 = \sqrt{2}\ z^* \tag{47}$$

$$Z_1 = 0.5\ (w_1 + 1) \qquad \text{if } N_v \text{ is odd} \tag{48a}$$

$$Z_1 = 0.5\ w_1 \qquad \text{if } N_v \text{ is even} \tag{48b}$$

$$Z_2 = 0.5\ w_2 \qquad \text{if } N_v \text{ is odd} \tag{49a}$$

$$Z_2 = 0.5\ (w_2 + 1) \qquad \text{if } N_v \text{ is even} \tag{49b}$$

For a 6-pass rotated square layout:

$$C_6 = C_1 - C_x - 4\ (N_{z_1} + N_{z_2}) \tag{50}$$

For an 8-pass rotated square layout:

$$C_8 = C_1 - C_x - C_y - 4\ (N_{z_1} + N_{z_2}) \tag{51}$$

A.10. Estimation of Tube-Side Pressure Drop

Pressure drop during fully developed flow in a cylindrical conduit is best characterized using a friction factor. The friction factor is dependent on the flow regime as was the heat transfer coefficient. When the flow is laminar (Re<1000), the friction factor is approximated by the following equation:

$$f_f = 16/Re \tag{52}$$

When the flow is turbulent (Re>4000), the friction factor is approximated by the following:

$$f_f = 0.04/Re^{0.194} \tag{53}$$

Equation (53) is for flow inside smooth tubing. When in the transition region (1000<Re<4000) the friction factor is obtained from the following equation.

$$f_f = \ln (4000/Re)/1.38629 \tag{54}$$

Once the friction factor is obtained, the pressure drop in the tubes is calculated by the following equation.

$$\Delta P_t = \frac{2\ \rho\ V^2\ L\ f_f}{di\ gc} \tag{55}$$

The entrance and exit losses are calculated from the following equation.

$$\Delta\ P_e = 2\ (N_{tp})\ (\frac{\rho V^2}{2gc}) \tag{56}$$

where

N_{tp} = number of tube passes

ρ = density of tube side fluid

V = velocity of the tube side fluid

g_c = gravitational conversion constant

The total pressure drop for the tube side is

$$\Delta P = \Delta P_t + \Delta P_e \tag{57}$$

B. Delaware Method

B.1. Simplified Mechanisms of Shell-Side Flow

Shown in Figure 3 is a diagram of the shell-side flow mechanism in a highly idealized form. In this figure, basically five different streams can be identified on the shell-side. Stream B is the desired flow path for fluid on the shell-side of the exchanger. Stream B is the main crossflow stream traveling through one window across the tubes and out through the opposite window (4).

However, due to the mechanical clearances required in a shell and tube exchanger, not all of the fluid in the shell follows the path of stream B. There is the A stream which leaks through the tube to baffle clearance from one baffle compartment to the next. The C stream, the bundle bypass stream, flows around the tube bundle between the outermost tubes and the shell. The E stream is the shell to baffle leakage stream flowing through the clearance between the baffles and inside shell diameter. The last of the major streams identified is the F stream, which flows through any channel within the tube bundle caused by the provision of pass dividers in the exchanger header (i.e. only in multiple tube pass configurations). These streams do not exist as neatly and precisely defined as the streams shown in Figure 3. Figure 3

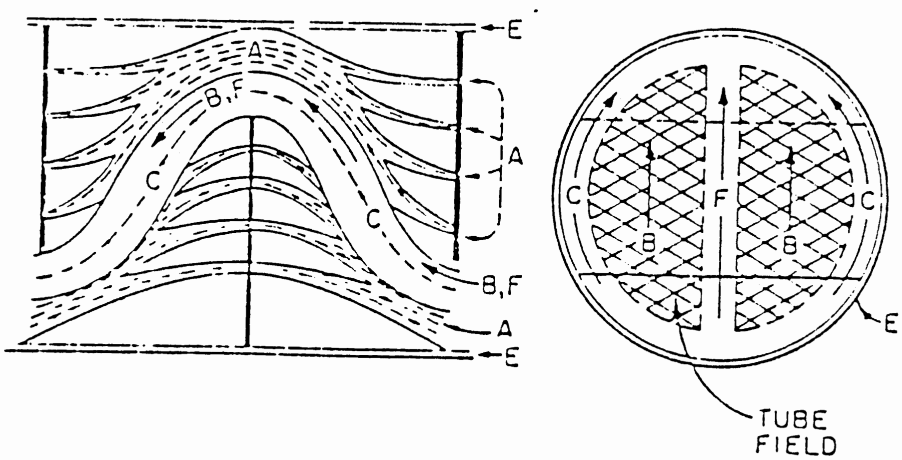


Figure 3. Idealized Diagram of Shell-Side Flow (4)

is an idealized representation but does allow a discussion of major modifying effects on the desired flow stream.

In the Delaware method (2,3,4), the B stream is the main flow stream with the other flow streams exerting various modifying effects upon the performance of the exchanger assuming the B stream alone exists. The leakage and bypass streams have basically two effects on the exchanger performance. The first is to reduce the local heat transfer coefficient because stream B is reduced. The second is the shell-side temperature profile is altered.

The various leakage and bypass streams have different degrees of effect on the shell-side performance. For example, stream A (tube-to-baffle leakage) has only a relatively small effect upon both the heat transfer coefficient and pressure drop. Stream C has a large effect but can be partially blocked. Stream E (shell-to baffle leakage) has extremely serious effect but little can be done to improve this problem. Stream F (the pass divider leakage) has a moderate effect and can sometimes be improved using mechanical means (4).

B.2. Basic Structure of the Delaware Method

The Delaware method is a rating method as are most "design" methods. A rating method is one in which the heat exchanger under consideration is fairly completely described geometrically and the process specifications for the streams are given. The information required to use this method is:

1. Both tube side and shell side flow rates, inlet and outlet temperatures, physical properties, and fouling characteristics
2. Shell inside diameter

3. Outer tube limit
4. Tube diameter and layout
5. Baffle spacing and cut
6. Length or Duty

Using the above information either the length of the exchanger or the duty of the exchanger is calculated. Also, the pressure drop for both the tube side and shell side can be calculated.

B.3. Specification/Calculation of Shell Side Geometry Parameters

B.3.1 Input Data Requirements for Delaware Method. The Delaware method assumes that the shell-side properties, flow rates, and temperatures are known or can be reasonably estimated. The method also assumes that the following minimum set of shell-side geometry data are known or specified:

1. Tube outside diameter, d_o
2. Tube pitch, p
3. Tube geometrical arrangement
4. Shell inside diameter, D_i
5. Shell outer tube limit, D_{otl}
6. Effective tube length, L
7. Baffle cut, l_c
8. Baffle spacing, l_s (also the inlet and outlet baffle spacing, $l_{s,I}$, and $l_{s,o}$, respectively)
9. Number of sealing strips/side, N_{ss}

From this basic geometrical information all remaining parameters needed in the shell-side calculations can be calculated or estimated by methods given in this chapter.

B.4. Calculation of Shell-Side Geometrical Parameters

B.4.1 Total number of tubes in the exchanger, N_t. N_t can be found by a direct tube count, tube count table, or by the method given by Phadke (10) discussed earlier in this chapter.

B.4.2 Tube pitch parallel to flow p_p, and normal to flow p_n. These quantities are needed only for the purpose of estimating other parameters. The quantities are dependent on both the layout and the tube pitch. The following equations were used to calculate p_p and p_n for a triangular layout.

$$p_p = \frac{\frac{1}{2}}{\tan\,(30^0)} * p \tag{58}$$

$$p_n = 1/2 * p \tag{59}$$

where

p = tube pitch

The equations used for an inline-square layout are the following;

$$p_p = p \tag{60}$$

$$p_n = p \tag{61}$$

The equations used to calculate these parameters for a rotated square layout are;

$$p_p = \cos(45^0) * p \tag{62}$$

$$p_n = \sin(45^0) * p \tag{63}$$

Figure 4 shows the tube pitches parallel and normal to flow for a triangular layout.

B.4.3 Number of tube rows crossed in one crossflow section area between baffle tips), N_c. N_c can be estimated from the following equation.

$$N_c = \frac{D_i\,[\,1 - 2\,(l_c/D_i)]}{p_p} \tag{64}$$

B.4.4 Fraction of total tubes in crossflow, F_c. F_c can be calculated from the following equation.

$$F_c = \frac{1}{\pi}\,\{\pi + 2\,(\frac{D_i - 2\,l_c}{D_{otl}})\,\sin[\cos^{-1}\,(\frac{D_i - 2\,l_c}{D_{otl}})]$$

$$- 2\,\cos^{-1}\,(\frac{D_i - 2\,l_c}{D_{otl}})\} \tag{65}$$

where all the angles are read in radians.

B.4.5 Number of effective crossflow rows in each window, N_{cw}. N_{cw} can be calculated using the following equation.

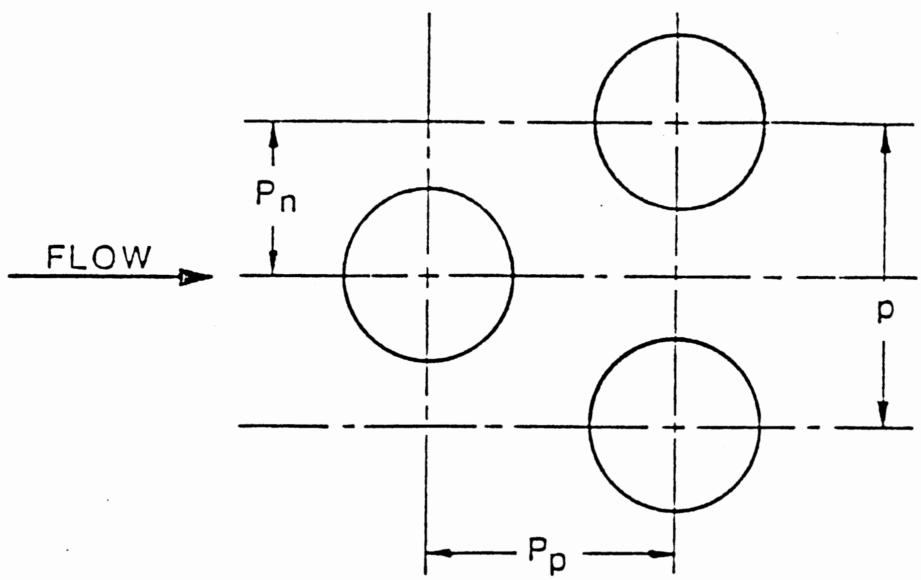


Figure 4. Tube Pitches Parallel and Normal to Flow (4)

$$N_{cw} = \frac{0.8\ l_c}{p_p} \tag{66}$$

This equation assumes that the shell-side fluid on the average crosses about one half of the tube rows in the window and the tubes extend about 80 % of the distance from the baffle tip to the shell inside diameter.

B.4.6 Number of baffles, N_b. Calculate N_b from:

$$N_b = \frac{L - l_{s,I} - l_{s,o}}{l_s} + 1 \tag{67}$$

This equation takes into account that the entrance and/or exit baffle spacing may be different from the central baffle spacing. If the length of the exchanger is being calculated N_b is calculated after the shell-side heat transfer coefficient is calculated.

B.4.7 Crossflow area at or near centerline for one crossflow section S_m. S_m is calculated from the following equation.

$$S_m = l_s\ [D_i - D_{otl} + (\frac{D_{otl} - d_o}{p_n})\ (p - d_o)] \tag{68a}$$

for square layouts, both inline and rotated, and

$$S_m = l_s\ [D_i - D_{otl} + (\frac{D_{otl} - d_o}{p})\ (p - d_o)] \tag{68b}$$

for triangular layouts.

These equations assume a nearly uniform tube field. The clearances for the tube pass partition lanes, the difference between the shell inside diameter and outer tube limit, are corrected for separately.

B.4.8 Fraction of crossflow area available for bypass flow, F_{sbp}. This parameter can be calculated by the following equation.

$$F_{sbp} = \frac{[D_i - D_{otl} + 1/2\ (N_p\ W_p)]\ l_s}{S_m} \qquad (69)$$

where,

N_p = the number of pass partition lanes through the tube field parallel to the direction of the crossflow stream

W_p = the width of the pass partition lanes

B.4.9 Tube-to-baffle leakage area for one baffle, S_{tb}. S_{tb} can be estimated from equation (70).

$$S_{tb} = \pi\ d_o\ \delta_{tb}\ (1/2)\ (1 + F_c)\ N_t \qquad (70)$$

where,

δ_{tb} = the diametral clearance between the tube and the baffle. Values for δ_{tb} can be found in Appendix A on page 125.

B.4.10 Baffle Cut Angle, Θ. The baffle cut angle, Θ, is the angle opposite of the intersection of the cut edge of the baffle with the inside surface of the shell. The baffle cut angle in radians can be calculated by:

$$\Theta = 2 \cos^{-1} \left(1 - \frac{2\ l_c}{D_i}\right) \qquad (71)$$

<u>B.4.11 Shell-to-baffle leakage area for one baffle, S_{sb}.</u> S_{sb} can be calculated from;

$$S_{sb} = \frac{\pi \, D_i \, \delta_{sb}}{2} \left[1 - \frac{\Theta}{2\,\pi}\right] \tag{72}$$

where

δ_{sb} = shell-to-baffle clearance

Θ = in radians and is between 0 and π

TEMA Standards for class R construction for the shell-to-baffle clearance are shown in Table IV.

<u>B.4.12 Area for flow through the window, S_w.</u> This area is obtained as the difference between the gross window area, S_{wg} and the window area occupied by the tubes, S_{wt}. The window area is indicated by the cross hatched regions in Figure 5.

$$S_w = S_{wg} - S_{wt} \tag{73}$$

The value of S_{wg} can be calculated from:

$$S_{wg} = \frac{D_i^{\,2}}{4} \left\{\frac{\Theta}{2} - \left[1 - 2 \left(\frac{l_c}{D_i}\right)\right] \sin\left(\frac{\Theta}{2}\right)\right\} \tag{74}$$

The window area occupied by the tubes, S_{wt}, can be calculated from:

$$S_{wt} = \frac{N_t}{8} \left(1 - F_c\right) \pi \, d_o^{\,2} \tag{75}$$

TABLE IV

TEMA STANDARDS FOR SHELL TO BAFFLE CLEARANCE (12)

Nominal Shell Inside Diameter, D_i, in	Diametral Shell-to-Baffle Clearance, δ_{sb}, in.
8-13	0.100
14-17	0.125
18-23	0.150
24-39	0.175
40-54	0.225
55-60	0.300

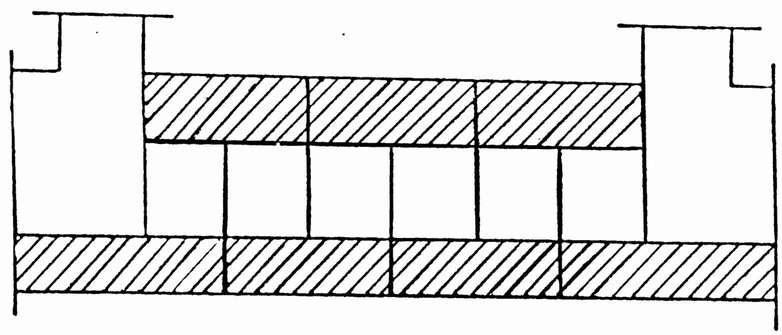


Figure 5. Window Sections in an E-Shell
Exchanger (4)

B.4.13 Equivalent diameter of window, D_w. This parameter is required only if the shell-side Reynolds number is less than or equal to 100. D_w can be calculated from:

$$D_w = \frac{4\ S_w}{\frac{\pi}{2}\ N_t\ (1 - F_c)\ d_o + D_i\ \Theta} \tag{76}$$

B.5. Calculation of Shell-Side Heat Transfer Coefficient

B.5.1. Calculate shell-side Reynolds number, Re_s. The shell-side Reynolds number can be calculated from the following equation.

$$Re_s = \frac{d_o\ W_s}{\mu_s\ S_m} \tag{77}$$

For most cases it is adequate to use the arithmetic mean bulk shell-side fluid temperature to evaluate all bulk properties of the shell-side fluid.

B.5.2. Calculate Colburn j-factor for an ideal tube bank, j_i. j_i can be found for a given tube layout at the calculated value of Re_s using Figure 6. A curve fit for Figure 6 is given in Appendix B.

B.5.3. Calculate the shell-side heat transfer coefficient for an ideal tube bank.

$$h_{ideal} = j_i\ C_{pshell}\ \left(\frac{W_{shell}}{S_m}\right)\ \left(\frac{K_{shell}}{C_{pshell}\ \mu_{shell}}\right)\ \left(\frac{\mu_{shell}}{\mu_{shell,\ w}}\right)^{0.14} \tag{78}$$

B.5.4. Find the correction factor for baffle configuration effects, J_c. J_c can be calculated from the following equation.

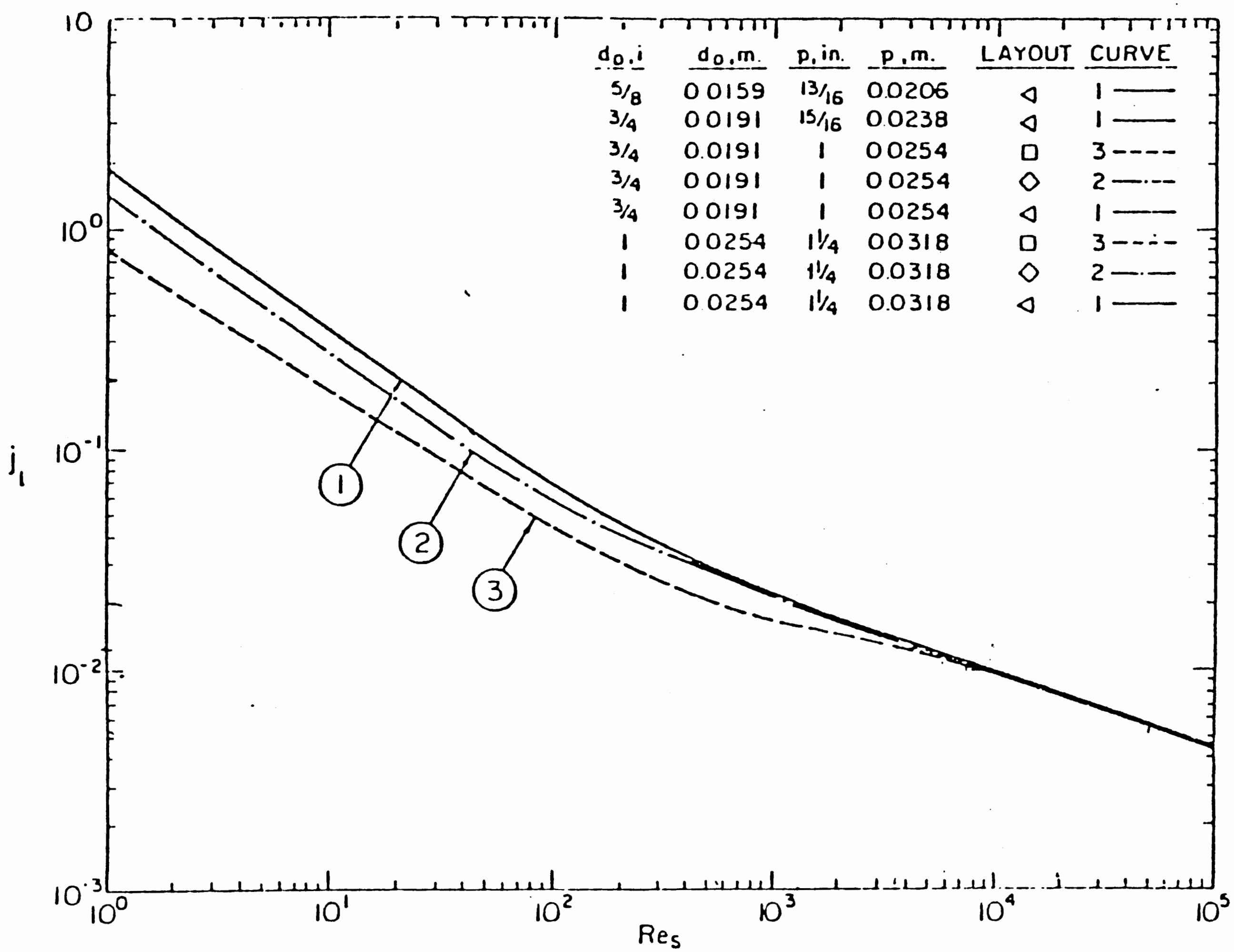


Figure 6. Correlation of j_i for Ideal Tube Banks (4)

$$J_c = F_c + 0.524\ (1.0 - F_c)^{0.32} \tag{79}$$

J_c is the correction factor for baffle cut and spacing. This factor takes into account the heat transfer in the window and calculates the overall average for the entire heat exchanger. This correction factor is essentially a function of the fraction of the total tubes in the heat exchanger that are in crossflow. A typical value for a well-designed heat exchanger with liquid on the shell-side is near 1.0.

<u>B.5.5. Find the correction factor for baffle leakage effects, J_l.</u> J_l is a function of the ratio of the total baffle leakage area, $(S_{tb}+S_{sb})$, to the cross flow area, S_m, and of the ratio of the shell-to baffle leakage area S_{sb} to the total baffle leakage area S_{tb}. J_l can be read from Figure 7. A curve fit for Figure 7 is given in Appendix B.

J_l is the correction factor for baffle leakage effects, including both shell-to-baffle and tube-to-baffle leakage. This correction factor is a function of the ratio of total leakage area per baffle and also the ratio of the shell-to-baffle leakage area. J_l weights the shell-to baffle leakage more heavily than the tube-to-baffle leakage. J_l is a function of the clearance between tube to baffle and shell to baffle so that credit is given for tighter construction practices. Also, if the baffles are too close together, J_l penalizes the heat transfer coefficient.

<u>B.5.6. Find the correction factor for bundle bypass effects, J_b.</u> J_b is a function of F_{sbp} and N_{ss}/N_c (the ratio of the number is sealing strips per side to the number of rows crossed in one baffle crossflow section). J_b can be found by using Figure 8 or the curve fit given in Appendix B.

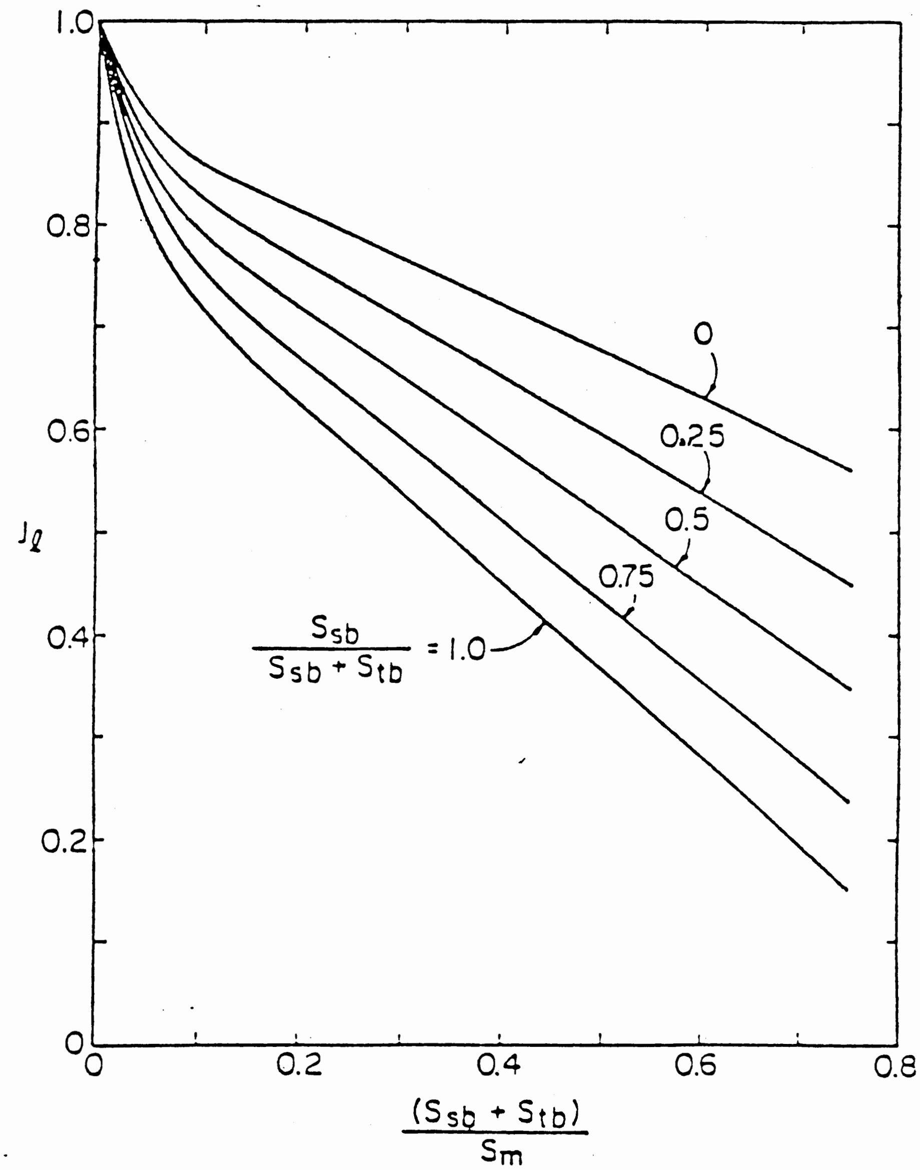


Figure 7. Correction Factor for Baffle Leakage (4)

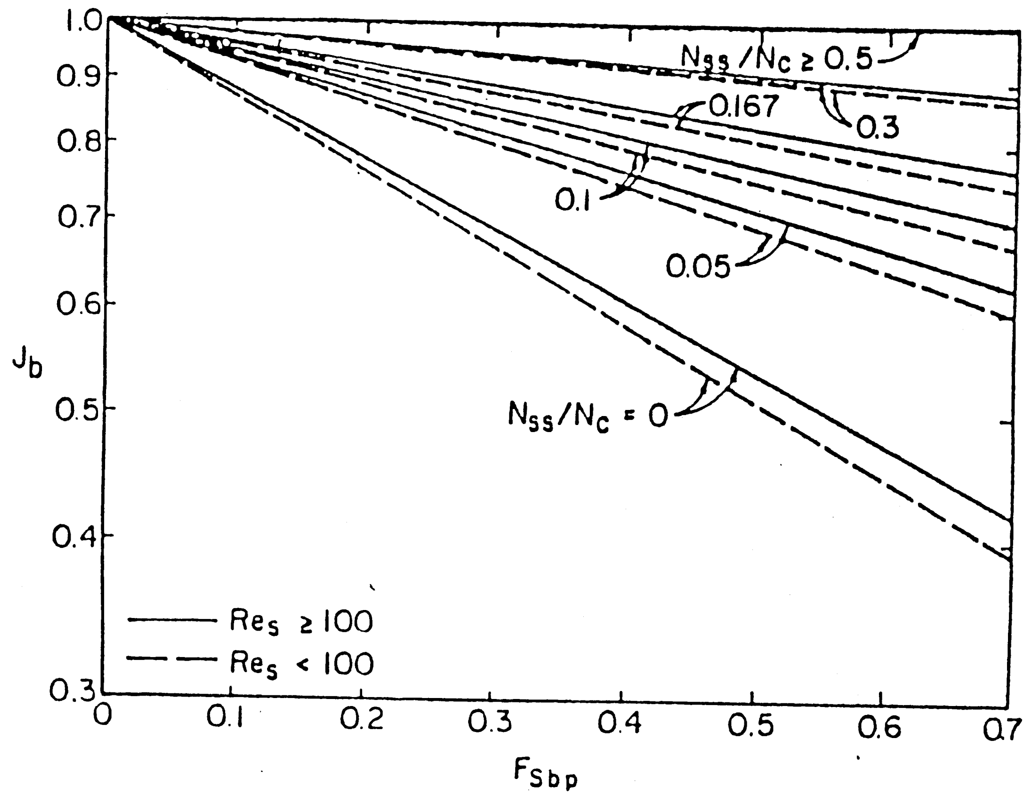


Figure 8. Correction Factor for Bypass Flow (4)

J_b is the correction factor for the bundle bypass flow (C and F streams). J_b accounts for differences in construction. For example in fixed tube sheets and pull-through floating head construction the clearances between the tubes and the inside shell diameter are quite different. J_b also considers the improvement made by sealing strips. Sealing strips are typically longitudinal strips of metal between the outside of the bundle and the shell and fastened at the baffle. These strips force the bypass flow periodically back into the tube field.

B.5.7. Find the correction factor for adverse temperature gradient buildup at low Reynolds numbers, J_r. This factor is equal to 1.00 if the shell side Reynolds number is greater than 100. For Re_s less than 20, the factor is only a function of the total number of tube rows crossed. For Re_s between 20 to 100, a linear proportion rule is used. Therefore:

A. If $Re_s < 100$ find J_r* from Figure 9 knowing N_b and (N_c+N_{cw})

B. If $Re_s <= 20$, $J_r = J_r^*$

C. If $20 < Re_s < 100$ find J_r from Figure 10.

The curve fits for Figures 9 and 10 are given in Appendix B.

J_r is the correction factor for adverse temperature gradient build-up. In laminar flow, the heat transfer coefficient decreases with increasing distance from the start of heating due to the development of an adverse temperature gradient. This gradient resists further heat transfer which therefore lowers the local and average heat transfer coefficients.

B.5.8. Find the correction factor for unequal baffle spacing at inlet and/or outlet, J_s. The equation for J_s is:

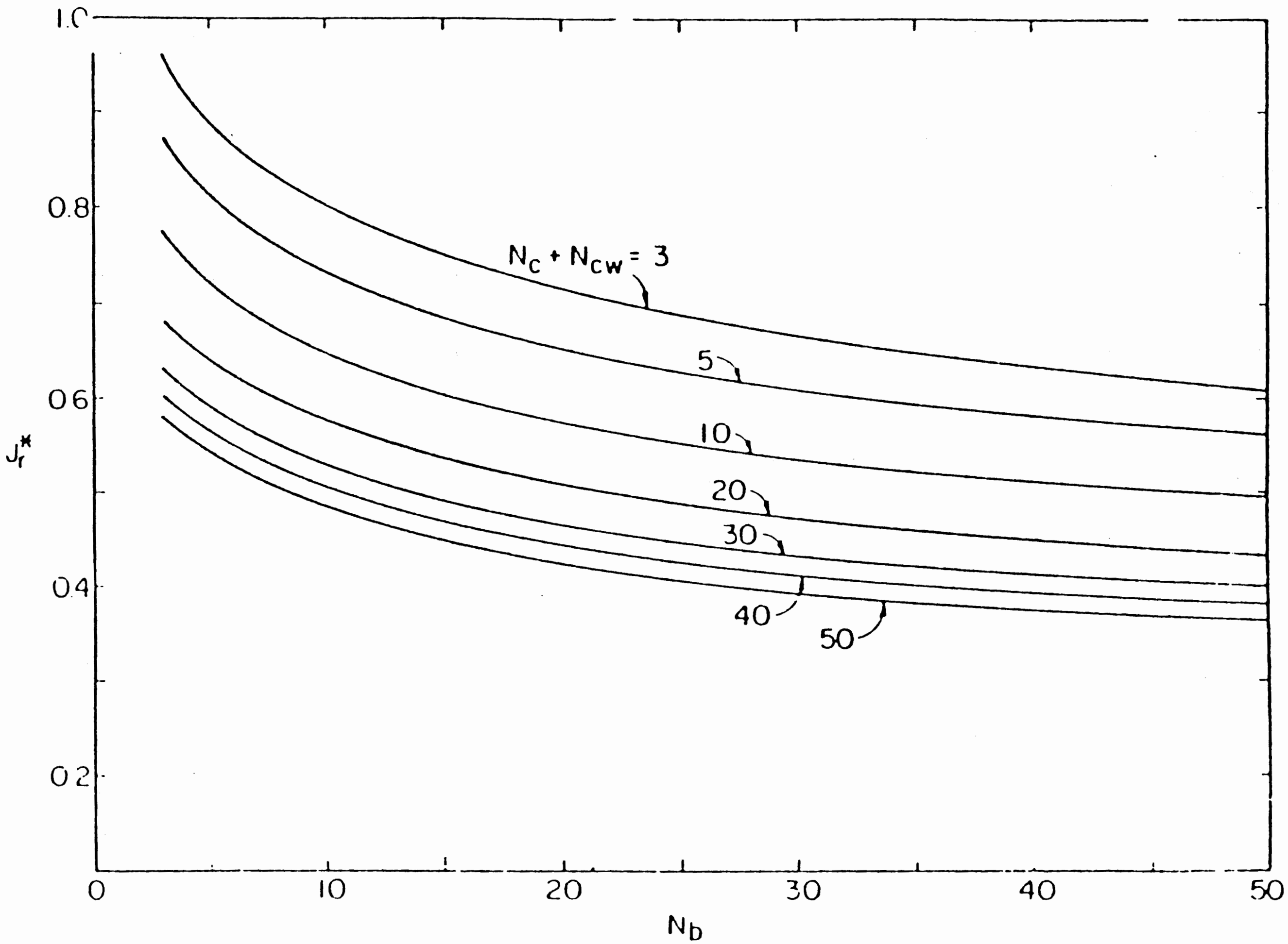


Figure 9. Correction Factor For Adverse Temperature Gradient at Low Reynolds Number (4)

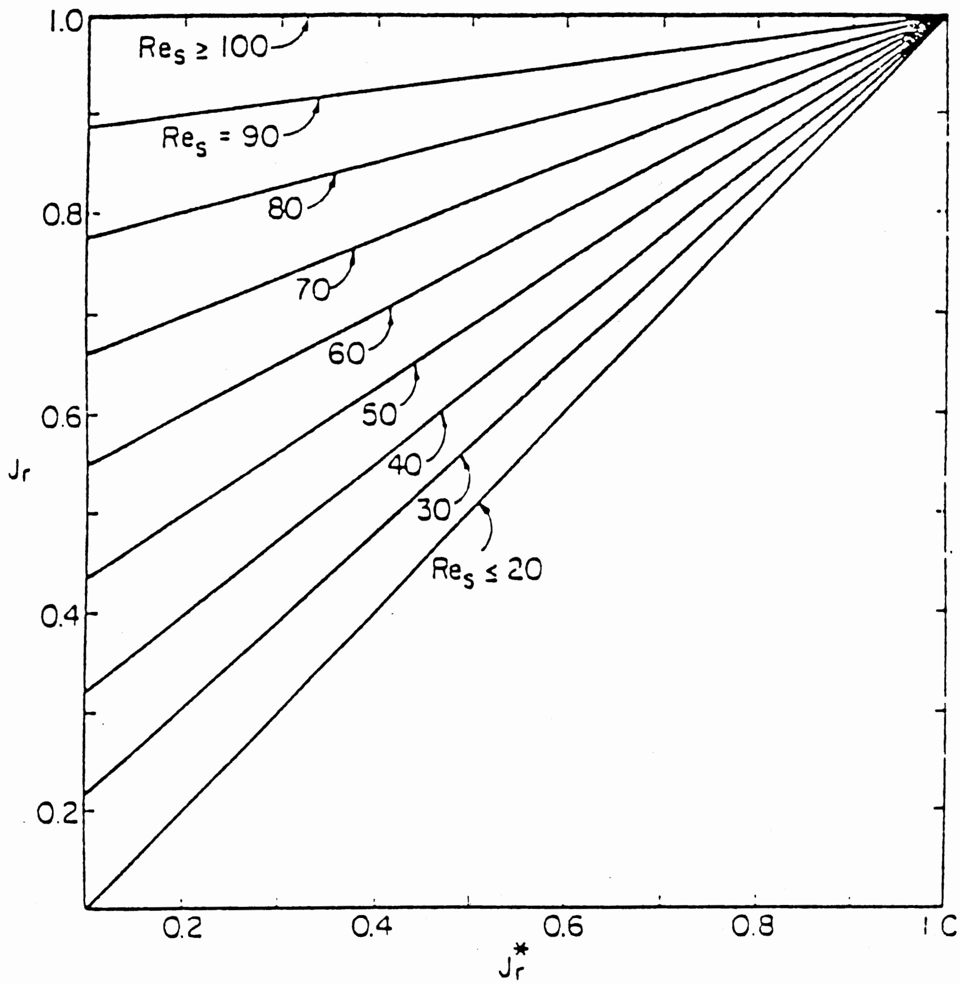


Figure 10. Correction Factor For Adverse Temperature Gradient at Intermediate Reynolds Number (4)

$$J_s = \frac{(N_b - 1) + (l_{s,I}^*)^{1-n} + (l_{s,o}^*)^{1-n}}{(N_b - 1) + l_{s,I}^* + l_{s,o}^*} \tag{80}$$

where,

N_b = number of baffles

$l_{s,I}^* = l_{s,I}/l_s$

$l_{s,o}^* = l_{s,o}/l_s$

l_s = internal baffle spacing

$l_{s,I}$ = entrance baffle spacing

$l_{s,o}$ = exit baffle spacing

$n =$ 0.6 for turbulent flow (Re_s>=100)
1/3 for laminar flow (Re_s<100)

J_s is the correction factor for variable baffle spacing in the inlet and outlet sections. The correction factor allows for the change in the average shell-side coefficient caused by these locally lower velocities.

B.5.9. Calculate the shell-side heat transfer coefficient for the exchanger, h_o, from the equation:

$$h_o = h_{ideal}\, J_c\, J_l\, J_b\, J_r\, J_s \tag{81}$$

The combined effect of J_c, J_l, J_b, J_r, and J_s for a well designed heat exchanger is typically about 0.6.

B.6 Calculation of Shell-Side Pressure Drop

B.6.1. Find f_i from the ideal tube bank friction factor curve for the given tube layout at the calculated value of Re_s using Figures 11 and 12. Curve fits for these curves are given in Appendix B.

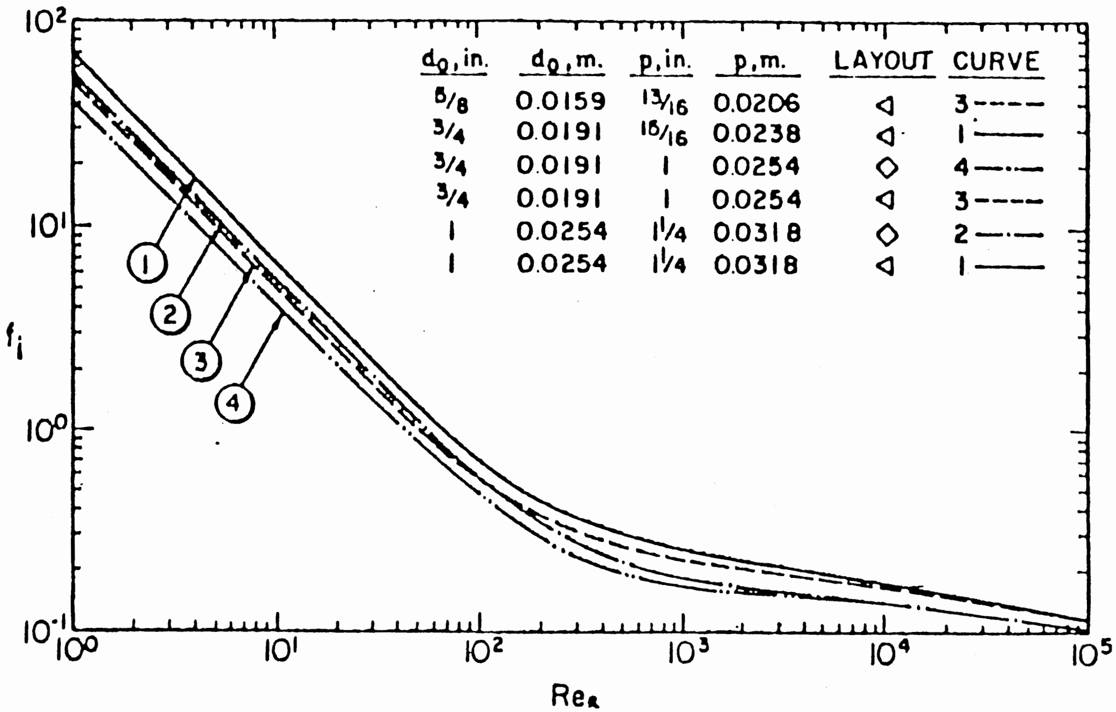


Figure 11. Correction of Friction Factors For Ideal Tube Banks (4)

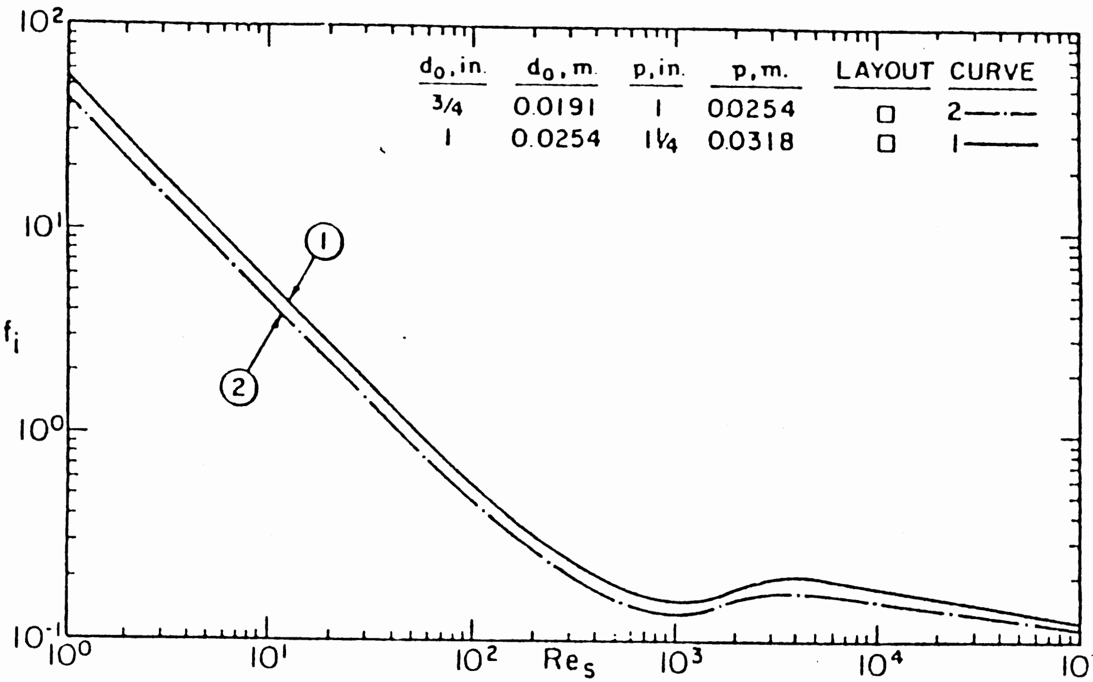


Figure 12. Correction of Friction Factors For Ideal Tube Banks (4)

B.6.2. Calculate the pressure drop for an ideal crossflow section, $\Delta P_{b,i}$:

$$\Delta P_{b,i} = \frac{4\ f_i\ W_{shell}^2\ N_c}{2\ \rho_{shell}\ g_c\ S_m^2} \left(\frac{\mu_{shell,\ w}}{\mu_{shell}}\right)^{0.14} \tag{82}$$

B.6.3. Calculate the pressure drop for an ideal window section $\Delta P_{w,i}$:

A. If $Re_s >= 100$:

$$\Delta P_{w,i} = \frac{(W_{shell})^2\ (2 + 0.6\ N_{cw})}{2\ g_c\ S_m\ S_w\ \rho_{shell}} \tag{83a}$$

B. If $Re_s < 100$:

$$\Delta P_{wi} = 26\ \frac{\mu_{shell}\ W_{shell}}{\sqrt{S_m\ S_w}\ \rho_{shell}} \left[\frac{N_{cw}}{p - d_o} + \frac{l_s}{D_w^2}\right] + \frac{(W_{shell})^2}{g_c\ S_m\ S_w\ \rho_{shell}} \tag{83b}$$

B.6.4. Calculate the correction factor for effect of baffle leakage on pressure drop, R_l. R_l can be read from Figure 13. Curves shown are not to be extrapolated beyond the points shown. Curve fits for R_l vs $(S_{sb}+S_{tb})/S_m$ are shown in Appendix B.

B.6.5. Find the correction factor for bundle bypass, Rb. Rb can be read from Figure 14 as a function of F_{sbp} and N_{ss}/N_c. The solid lines are for $Re_s >= 100$; the dashed lines are for $Re_s < 100$.

B.6.6. Find the correction factor for unequal baffle spacing, R_s. R_s can be calculated from:

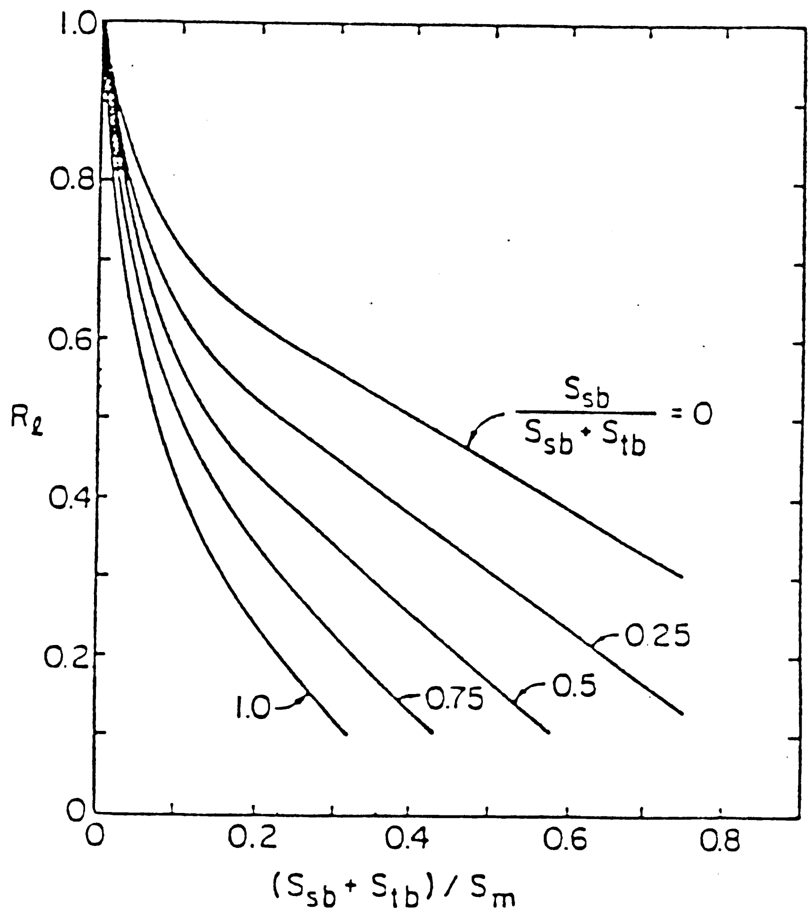


Figure 13. Correction Factor For Baffle Leakage Effect on Pressure Drop (4)

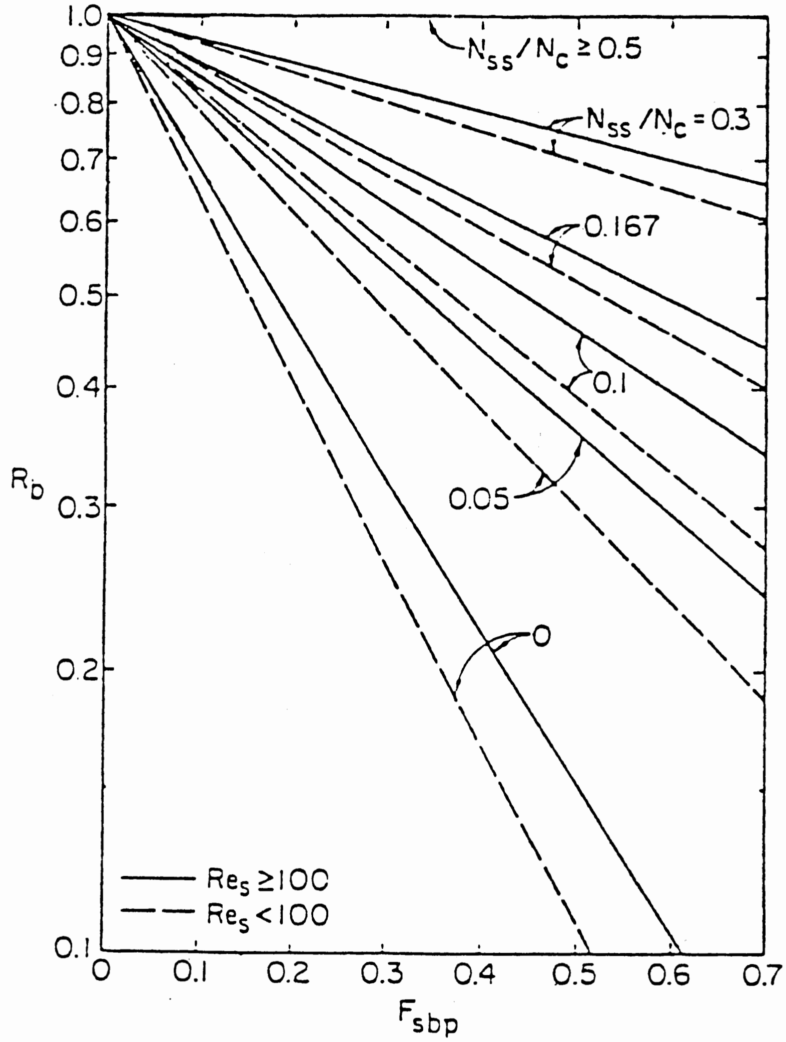


Figure 14. Correction Factor on Pressure Drop for Bypass Flow (4)

$$R_s = 1/2\ [(l^*_{s,I})^{-n'} + (l^*_{s,o})^{-n'}] \tag{84}$$

where

$l^*_{s,I} = l_{s,I}/l_s$

$l^*_{s,o} = l_{s,o}/l_s$

$$n' = \begin{cases} 1.6 \text{ for turbulent flow } (Re_s > 100) \\ 1 \text{ for laminar flow } (Re_s < 100) \end{cases}$$

B.6.7. Calculate the pressure drop across the shell-side (excluding nozzles), from:

$$\Delta P_s = [(N_b - 1)\ (\Delta P_{b,i})\ R_b + N_b\ \Delta P_{w,i}]\ R_l$$

$$+ 2\ \Delta P_{b,i}\ R_b\ (1 + \frac{N_{cw}}{N_c})\ R_s \tag{85}$$

C. Air-Cooled Heat Exchanger Calculation Method

C.1. Basic Information Required for Air-Cooler Calculations

The information required to make a preliminary estimate of the size of the exchanger and fan requirements is:

1. Tube mass flow rate, inlet and outlet temperatures, and physical properties
2. The number of tube passes
3. Tube outside diameter, inside diameter, and pitch
4. Thermal conductivity of tube metal
5. Tube arrangement
6. Fouling resistances for air side and tube side

7. Fin height
8. Fin thickness
9. Number of fins per inch
10. Fin thermal conductivity
11. Altitude
12. Number of tube rows
13. Inlet air temperature
14. Type of draft

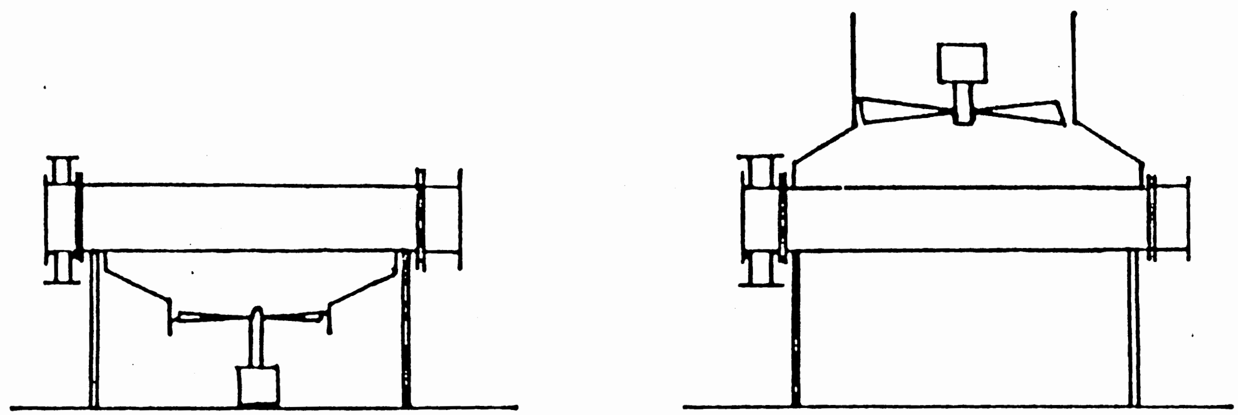
C.2. Air-Cooled Heat Exchanger Arrangement

There are two basic air-cooled heat exchanger arrangements. One is forced draft and induced draft. This is shown in Figure 15. The forced draft arrangement has the fan underneath the tube bank pushing air across the bank of finned tubes and discharging into the atmosphere above. Induced-draft has the fan located above the bank of tubes, drawing air across the tube bank and then discharging it through an upper plenum back into the atmosphere.

The forced-draft arrangement has the advantage of putting the fan and driver underneath the tube bank where it is more accessible. Also, the driver and fan are operated at atmospheric temperatures which reduces maintenance problems, and allows for a wider choice of materials of construction. The disadvantages of forced draft construction are:

a) The fan is exposed to places where operating personnel may walk.

b) The fan is more likely to ingest trash or debris.



Forced Draft

Induced Draft

Figure 15. Forced Draft and Induced Draft Arrangements of Air-Coolers (4)

c) The discharge velocity from the fan is not very uniform which leads to a wide range of local air flow velocities across the tube bank, which reduces thermal efficiency.

d) The air is discharged from the top of the exchanger at fairly low velocities, therefore, it can mix with incoming air giving rise to partial recirculation

e) The top of the exchanger is exposed which means it could be easily damaged by hail and may suffer more degradation and corrosion.

f) A sudden rain shower can lower the air temperature and increasing cooling rapidly which causes control problems.

The induced draft design has the disadvantage of putting the fan and driver overhead where it is hard to service and exposed to the hot air coming off of the heat exchanger, requiring selection of special materials of construction to satisfy the operating temperature limits. The advantages of induced draft construction are;

a) The fan is not located near personnel work areas.

b) Uniform air flow across the tube bank.

c) The discharge plenum can produce a reasonably high velocity, therefore, recirculation is not often a problem.

C.3. The Basic Design Equation

The basic design equation to be used for air-cooled heat exchanger is,

$$A = \frac{Q}{U_o\,(MTD)} = \frac{Q}{U_o\,F_t\,LMTD} \tag{86}$$

where

A_o = the total heat transfer area required

Q = the total heat transferred

U_o = the overall heat transfer coefficient

LMTD = the configuration correction factor

F_t = the logarithmic mean temperature difference

C.4. Estimation of Heat Load

The heat load for the sensible heat transfer case can be calculated from the following:

$$Q = W\,C_p\,(T_2 - T_1) \tag{87}$$

where

Q = amount of heat transferred

W = mass flow rate of tube side fluid

C_p = specific heat of tube side fluid

T_2 = tube side fluid inlet temperature

T_1 = tube side fluid outlet temperature

C.5. Estimation of Mean Temperature Difference

The first step in calculating the mean temperature difference is to find the logarithmic mean temperature difference (LMTD) for counter-current flow. The LMTD is calculated from the following equation

$$\text{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\frac{T_1 - t_2}{T_2 - t_1}\right)} \tag{88}$$

where

T_1 = the tube side inlet temperature

T_2 = the tube side outlet temperature

t_1 = the air inlet temperature

t_2 = the air outlet temperature

One difficulty which arises is that the outlet air temperature is not known at this point but may be estimated form the following equation.

$$t_2 = t_1 + [\frac{(U_o + 1)}{10.0} * (\frac{(T_1 + T_2)}{2} - t_1)] \tag{89}$$

where the overall heat transfer coefficient, U_o, is estimated from Table V. Then the calculations proceed until U_o that is guessed is equal to U_o calculated.

Next the configuration correction factor is calculated. To find F_t, the correction factor for the logarithmic mean temperature difference, the following equations are used as given by Pigorini (11).

$$A = [(T_1 - T_2)^2 + (t_1 - t_2)^2]^{0.5} \tag{90}$$

$$B = [(T_1 - t_2)^{1/2} + (T_2 - t_1)^{1/\alpha}]^{\alpha} \tag{91}$$

$$CLMTD1 = \frac{A}{\beta\ (\ln(\frac{B + A}{B - A}))} \tag{92}$$

where

$\alpha = \beta = 1.7$

For a single pass exchanger

$$F_t = \frac{CLMTD1}{LMTD} \tag{93}$$

TABLE V

TYPICAL OVERALL DESIGN COEFFICIENTS FOR AIR-COOLED EXCHANGERS (6)

FLUID 1	FLUID 2	TOTAL FOULING RESISTANCE (1) HR FT2 F/BTU	Uo,(1) BTU/HR FT2 F
AIR	WATER	0.001	7.5 - 4.4
AIR	LIQUID (2) HYDROCARBONS	0.0005	5.9 - 0.6
AIR	GAS, 10 PSIG	0	2.1 - 0.6
AIR	GAS, 100 PSIG	0	2.4 - 1.9
AIR	GAS, 1000 PSIG	0	5.2 - 4.2

1. The total fouling resistance and the overall heat transfer coefficient are based on the total outside tube area.

2. Viscosity is 0.2 to 10.0 cP.

For a two pass exchanger

$$CLMTD = 0.6\ (CLMTD1) + 0.4\ (LMTD) \tag{94}$$

$$F_t = \frac{CLMTD}{LMTD} \tag{95}$$

When the number of tube passes is greater than or equal to three, the configuration correction factor is equal to 1.0.

C.6. Calculation of Geometrical Parameters

The basic geometrical parameters for high finned tubes are shown in Figure 16. The root diameter is equal to the bare tube outside diameter. The total external area/ft of fintube in sq. ft/ft (APF) is found from the following equations

$$A_{root} = \pi\ D_r \tag{96}$$

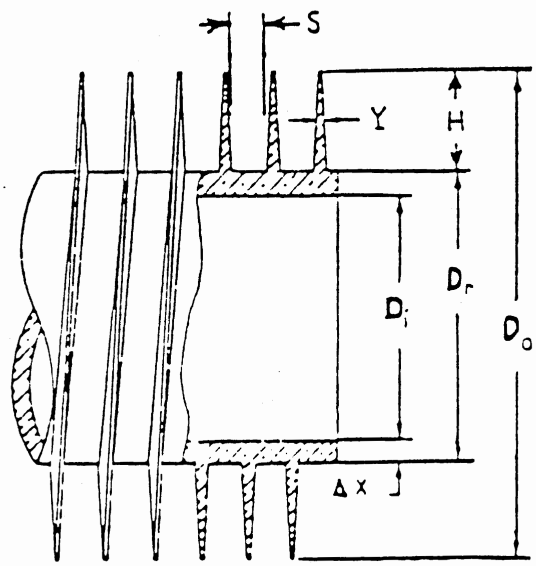
$$D_o = D_r + 2 * H \tag{97}$$

$$A_{f1} = \pi\ (D_o^2 - D_r^2)/2.0 \tag{98}$$

$$A_{f2} = N_f * A_{f1} \tag{99}$$

$$A_{r1} = (1 - N_f * Y) * A_{root} \tag{100}$$

$$A_{f3} = (\pi\ N_f\ Y\ D_o)/2 \tag{101}$$



D_r - Root diameter

D_o - Diameter over fins

D_i - Inside diameter at finned section

Δx - Wall thickness of finned section

Y - Mean fin thickness

H - Height of fin

S - Space between fins

Figure 16. High Finned Tubes Geometrical Parameters (4)

$$APF = (A_{f2} + A_{r1} + A_{f3})/12.0 \tag{102}$$

where

D_o = diameter over fins, in.

D_r = root diameter of finned section, in.

H = height of fins, in.

Y = mean fin thickness, in.

N_f = number of fins per inch

Next, the area per sq. ft of bundle face area per row is calculated in sq. ft/sq. ft (APSFPR).

$$APSFPR = APF * (12.0/p) \tag{103}$$

where

P = tube pitch, in.

The last geometrical quantity required is the area ratio of fin tube compared to the exterior area of a bare tube (AR).

$$AR = (A_{f2} + A_{r1} + A_{f3}/12.0)/(A_{root}/12.0) \tag{104}$$

C.7. Calculation of Overall Heat Transfer Coefficient, U_o

The overall heat transfer coefficient is calculated from the following equation.

$$\frac{1}{U_o} = \frac{1}{h_{air}} + R_{fa} + R_{fin} + \frac{\Delta x}{k_w}\frac{A_o'}{A_m'} + (R_{fi} + \frac{1}{h_i})\frac{A_o'}{A_i'} \tag{105}$$

h_{air} = air-side individual heat transfer coefficient

R_{fo} = the air-side fouling resistance

R_{fin} = the fin resistance

Δx = the wall thickness

k_w = the bare tube metal thermal conductivity

A_o' = the outside tube area

A_m' = the mean outside tube area

R_{fi} = the tube side fouling factor

R_i = the tube side heat transfer coefficient

A_i' = the inside tube area

C.8. Individual Heat Transfer Coefficents

The tube side heat transfer coefficient is calculated according to the method given in Section A.7.

The air-side heat transfer is based on methods given by Bell (4) and modified by Gianolo and Cuti (7). The equation used to calculate the airside coefficient depends on the type of draft (forced or induced).

$$Re = \frac{(D_r/12.0)\ \rho_a\ V_a}{\mu_a\ (3600)} \tag{106}$$

where

Re = Reynolds number

D_r = root diameter, in.

ρ_a = density of air, lb/ft^3

V_a = air velocity, ft/sec

μ_a = viscosity of air, lb/ft hr

The air velocity is calculated by the following equations.

The air velocity is calculated by the following equations.

$$A_{rfa1} = \{(p - D_o)/12.0\} * L * (\frac{N_t}{N_R} - 1.0) \qquad (107)$$

$$A_{rfa2} = 2 * ((H * S * N_f)/12.0 * L * (N_t/N_R) \qquad (108)$$

$$Air_{far} = A_{rfa1} + A_{rfa2} \qquad (109)$$

$$V_a = (w/\rho_a)/Air_{far} \qquad (110)$$

where

p = tube pitch, in.

D_o = diameter over fins, in.

L = length of the exchanger, ft.

N_t = total number of tubes

N_R = total number of rows

H = height of fin, in.

S = space between fins, in.

N_f = number of fins per inch

ρ_a = density of air, lb/ft^3

w = air mass flow rate, lb/fr

The Prandtl number is then calculated.

$$Pr = \frac{c_{pa}\ \mu_a}{k_a} \qquad (111)$$

Finally the air-side heat transfer coefficient for forced draft is

$$\frac{h_{air}\ k_a}{(D_r/12.0)} = 0.134\ Re^{0.681}\ Pr^{1/3}\ (H/S)^{-0.2}$$

$$(1 + V/11811.02 * N_R^{\ 2})^{-0.14}\ (Y/S)^{-0.1143} \qquad (112)$$

where

h_{air} = air side individual heat transfer coefficient, Btu/ft^2 hr°F

k_a = air-side thermal conductivity, Btu/ft hr°F

D_r = root diameter, in.

Re = Reynolds number

Pr = Prandtl number

H = height of fin, in.

S = space between fins, in.

V = velocity of the fluid, ft/hr

N_R = number of tube rows

Y = mean fin thickness, in.

The equation used for induced draft is calculated from the following equation.

$$\frac{h_{air}\ k_a}{D_r} = 0.287\ Re^{0.685}\ Pr^{1/3}\ AR^{-0.311}\ (N_R/6)^{-0.138} \qquad (113)$$

where

h_{air} = air-side individual heat transfer coefficient, Btu/ft^2 hr°F

k_a = air thermal conductivity, Btu/ft hr°F

D_r = root diameter, in.

Re = Reynolds number

Pr = Prandtl number

AR = is the area ratio of fin tube compared to the exterior area of bare tube, ft^2/ft

N_R = number of tube rows

C.9. Fin Resistance and Tube Area Calculations, R_{fin}, A_i', A_o', and A_m'

The fin resistance, R_{fin}, is calculated by the following method. The fin resistance is a function of the outside heat transfer coefficient and the fouling resistance.

$$m = \left(\frac{H}{12.0}\right) \sqrt{\frac{2}{\left(\frac{1}{h_{air}} + R_{fa}\right) k_f \left(\frac{Y}{12.0}\right)}} \tag{114}$$

$$n_a = \frac{1}{1 + \frac{m^2}{3} \sqrt{\frac{D_o}{D_r}}} \tag{115}$$

$$R_{fin} = \left[\frac{1 - n_a}{\frac{A_{root}}{APF} + n_a}\right] \left(\frac{1}{h_{air}} + R_{fa}\right) \tag{116}$$

where

H = fin height, in.

D_o = diameter over fins, in.

D_r = root diameter of finned section, in.

h_{air} = air-side heat transfer coefficient, Btu/ft^2 hr°R

R_{fa} = air-side fouling factor, ft^2 hr°R/Btu

k_f = fin metal thermal conductivity, Btu/ft hr°R

Y = mean fin thickness, in.

APF = quantities defined in Section C.6, ft^2/ft

The outside tube area, A_o', is calculated from the following equation.

$$A_o' = APF + \left(\frac{A_{root}}{12.0}\right) * (1 - Y * N_f) \tag{117}$$

where

Y = mean fin thickness, in.

N_f = number of fins per inch.

The total external area per foot of fin tube is calculated in section C.6. The mean outside tube area is calculated from the following equation.

$$A_m' = (\pi D_r/12.0 + \pi di/12.0)/2.0 \tag{118}$$

where

D_r = root diameter, inches

d_i = inside diameter, inches

The inside tube area is calculated by;

$$A_i' = \pi (d_i/12.0) \tag{119}$$

C.10. Pressure Drop Calculations

The tube side pressure drop is calculated according to the method given in section A.10.

The airside pressure drop is calculated using the following equations.

$$\Delta P = \frac{f_r \; N_R \; \rho_a \; V_a^2}{[g_c * 144]} \tag{120}$$

where

f_r = is the air-side friction factor

N_R = is the number of tube rows

ρ_a = is the air density, lb/ft^3

V_a = is the velocity, ft/sec

g_c = 32.174 lb-ft/s^2-lbf

Δp = pressure drop, psi

The air-side friction factor is calculated by the methods given by Robinson and Briggs (12). The friction factor for a triangular layout is:

$$f_r = 18.93 \left(\frac{\left(\frac{D_r}{12.0}\right) \rho_a \; V_a}{\mu_a} \right)^{-0.316} * \left(\frac{p}{D_r}\right)^{-0.927} \tag{121}$$

where

D_r = root diameter, in.

ρ_a = density of air, lb/ft^3

μ_a = viscosity of air, lb/ft hr

V_a = velocity as defined in equation (110), ft/hr

p = tube pitch , in.

For tubes laid out in rotated square arrangements, the friction factor is

$$f_r = 18.93 \left(\frac{\left(\frac{D_r}{12.0}\right) \rho_a \; V_a}{\mu_a} \right)^{-0.316} \left(\frac{p}{D_r}\right)^{-0.927} \left(\frac{p}{p_1}\right)^{0.52} \tag{122}$$

where

p_l = is the longitudinal pitch between centers of adjacent tubes in different rows measured along the diagonal,

$p_l = p/0.7071$

The pressure drop due to entrance effects is given by the following equation.

$$\Delta P_e = \frac{(AVFF)^2 \rho_a}{2 g_c (144)} \tag{123}$$

where

AVFF = air velocity at fan face, ft/sec

ρ_a = density, lb/ft^3

g_c = 32.174 $lb\text{-}ft/lbf\text{-}sec^2$

This quantity is calculated after the horse power, number of fans, and fan diameters. Then to get the total pressure drop the pressure drop due to entrance effects is added to the pressure drop found in equation (120) and the horsepower is recalculated.

C.11. Calculation of Heat Exchanger Geometry

First the face area, F_a, of the exchanger is calculated using APSFPR factor calculated in the section C.6. The face area is found from the following equation.

$$F_a = \frac{A_o}{N_R \ APSFPR} \tag{124}$$

where

N_R = the number of tube rows

The bay width is calculated using the following equation.

$$\text{Width} = F_a/L \tag{125}$$

where

L = length of the exchanger, ft

The tube count for an air-cooled exchanger is calculated from the following.

$$N_t = \frac{A_o}{(APF)\ L} \tag{126}$$

C.12. Calculation of Fan Power Requirements And Estimation of Number of Fans

The fan power requirements are calculated using the following equations.

$$BHP = \frac{\Delta P * w}{(eff)\ \rho_a\ (13750.0)} \tag{127}$$

where

BHP = the fan power requirements, hp

ΔP = the total fan pressure drop, psi

ρ_a = the air density, lb/ft^3

w = the air mass flow rate, lb/hr

eff = the fractional fan efficiency

The number of fans required can be estimated by dividing the horse power requirements by 25.

C.13. Calculation of Fan Diameter

The fan diameter was determined by the method given in the GPSA Engineering Data Book. To calculate the fan diameter first the fan area/fan (FAPF) must be calculated.

$$FAPF = \frac{0.4\ F_a}{(\text{number of fans})} \tag{128}$$

Then the fan diameter is calculated.

$$\text{Fan Diameter} = \sqrt{FAPF/(\tfrac{\pi}{4})} \tag{129}$$

C.14. Calculation of Air Volume and Velocity

The air volume per fan (AVPF) is calculated from the following equation.

$$AVPF = \frac{w}{(\text{number of fans})\ \rho_a} \tag{130}$$

The air velocity at the fan face (AVFF) is

$$AVFF = \frac{w}{(\text{number of fans})\ (\rho_a)\ (\frac{\pi}{4})\ (\text{fan diameter})^2} \tag{131}$$

The air velocity at the tube face (AVTF) is

$$AVTF = w/\rho_a F_a \tag{132}$$

C.15. Procedure for Air-Cooled Heat Exchanger Calculations

The procedure given starts with a step for approximating air-temperature rise. After the air outlet temperature has been determined, the corrected log mean temperature difference is calculated. A typical overall heat transfer coefficient is used to approximate the heat transfer area required. Then an overall heat transfer coefficient is calculated. If the heat transfer coefficient is not the same as guessed, then the procedure is repeated.

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

The purpose of this study was to modify and document a preliminary heat exchanger design simulator and to write a user's manual for this simulator. HEXSIM is designed to give a preliminary estimate of the size and configuration of a shell and tube and air-cooled heat exchanger for sensible heat transfer. HEXSIM can be used as a useful teaching tool for senior level design students and returning graduate students in order to give them a feel for the size and geometry of a heat exchanger.

From the work on HEXSIM, the following recommendations are made as guidelines for future work:

1. The simulator should be expanded to handle cases where there is a phase change, i.e. condensers and boilers.
2. The accuracy of the air-cooled heat exchanger method should be checked by testing the results obtained using HEXSIM against that actually built. If the current method is unsatisfactory, it should be changed to another method available in the open literature.

BIBLIOGRAPHY

1. Bell, K. J., in Schlunder, E. U., ed., Heat Exchanger Design Handbook, Vol. 3, Sec. 3.1.4, Hemisphere Publ. Corp., Washington, 1983.

2. Bell, K. J., "Exchanger Design Based on the Delaware Research Program", Petro/Chem Engineer, 32, Oct. 1969, pp. c-26 - c-40c.

3. Bell, K. J., "Final Report of the Cooperative Research Program on Shell and Tube Exchangers", Bulletin No. 5, University of Delaware Experiment Station, Newark, Delaware, 1963.

4. Bell, K. J., Process Heat Transfer, class notes at Oklahoma State University, Unpublished, 1984.

5. Bowman, R. A., Mueller, A. C., and Nagle, W. M., "Mean Temperature Difference in Design", Journal of Heat Transfer, 1940, p. 283.

6. Engineering Data Book, Gas Processors Suppliers Association, Tulsa, 1972.

7. Gianolio, E., and Cuti, F., "Heat Transfer Coefficients and Pressure Drop for Air Coolers with Different Number of Rows Under Forced Draft and Induced Draft", Heat Transfer Engineering, Vol. 3, No. 1, July - September 1981, p. 38.

8. Leesley, M. E., ed., Computer-Aided Process Plant Design, Gulf Publishing Company, Houston, 1982.

9. Peterson, J. N., Chen, C. C., and Evans, L. B., "Computer Programs for Chemical Engineers: 1978", Chemical Engineering, June-August, 1978.

10. Phadke, P. S., "Determining Tube Counts for Shell and Tube Exchangers", Chemical Engineering, p. 65, September 3, 1984.

11. Pigorini, A., De Pascale, T., and Milanesi, F., "Program Simplifies CLMTD Calculations", Hydrocarbon Processing, November 1982, p. 205.

12. Robinson, K. K., and Briggs, D. E., Chemical Engineering Symposium Series, No. 64, "Heat Transfer - Los Angeles", 62, 177 (1967).

13. Tubular Exchanger Manufacturers Association, Standard, 6th Ed., New York, 1978.

APPENDIX A

HEXSIM: A PRELIMINARY SHELL AND TUBE AND AIR-COOLED HEAT EXCHANGER DESIGN/RATING SIMULATOR

USER'S MANUAL

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CHAPTER I

INTRODUCTION

A. Overview

The HEXSIM program is designed to fill the gap between hand calculations and the massive detailed heat exchanger design simulators. Many times for preliminary design purposes a general idea of the size and configuration of the heat exchanger is all that is required. This would take several hours by hand, but is accomplished very quickly by computer. The HEXSIM program was designed to provide basic information on the size and configuration of shell and tube and air-cooled heat exchangers for sensible heat transfer. HEXSIM is only for Newtonian fluids. HEXSIM is interactive to allow the user to easily change the input parameters from the screen. HEXSIM has the following capabilities:

1. Calculates and checks heat balance
2. Calculates log mean temperature difference and correction factor
3. Calculates overall area
4. For a given area feasible inside shell diameters, outer tube bundle limits, tube lengths, tube counts, and length to diameter ratios are calculated
5. Calculates individual and overall heat transfer coefficients
6. Calculates pressure drop on both the tube side and shell side

7. Calculates bay width and length for air-coolers
8. Calculates number, diameter, and power requirements of each fan
9. Calculates air volume per fan
10. Calculates average air temperature and pressure at exit of each exchanger
11. Calculates air velocity at fan face and tube face
12. Allows modification of the problem or calculational conditions

Most of the operations can be accomplished using one or two simple mnemonic commands available in the EDIT section.

B. Program Data Requirements

There are two options available in the shell and tube exchanger calculations. The first is a preliminary calculation, and the second is the more detailed Delaware method (2,3,5) of calculation for shell side analysis. The data required for the preliminary calculations are the following.

1. Tube side and shell side mass flow rates
2. All inlet and outlet temperatures
3. The specific heat of the tube side and shell side fluids
4. Number of tube and shell passes
5. Tube outside diameter, inside diameter, and pitch
6. Thermal conductivity of tube metal
7. Tube arrangement
8. Type of tube bundle construction
9. Estimated individual heat transfer coefficients
10. Estimated fouling factor for both tube and shell side

The additional data required for the Delaware calculations are as follows.

1. The fluid density and phase of both shell and tube side fluids
2. Viscosity at the wall of the tube side and shell side fluids
3. Bulk viscosity of the tube side and shell side fluids
4. Thermal conductivity of each fluid
5. Baffle spacing
6. Percent baffle cut
7. Length of baffle cut
8. Tube to baffle clearance
9. Shell to baffle clearance
10. Number of sealing strips
11. Number of pass partition lanes
12. Width of pass partition lanes
13. Number of tubes
14. Shell diameter
15. Outer tube limit
16. Tube length

Mnemonic commands can then be used to adjust the problem parameters to any desired configuration.

The data required for air-cooled heat exchangers are the following.

1. Tube side mass flow rates
2. Inlet and outlet temperatures for the tube side
3. The specific heat, density, viscosity, and thermal conductivity of the tube side fluid
4. Number of tube passes
5. Tube outside diameter, inside diameter, and pitch

6. Thermal conductivity of tube metal
7. Tube arrangement
8. Tube side and air side fouling factor
9. Fin height and thickness
10. Mean fin thickness
11. Number of fins per inch
12. Fin thermal conductivity
13. Altitude of the exchanger location
14. Number of tube rows
15. Inlet air temperature
16. Type of draft
17. Tube length
18. Number, diameter, and driver efficiency per fan
19. Type of fan draft

Mnemonic commands can then be used to adjust the parameters of the problem.

C. Methods of Calculation

C.1. Shell and Tube Exchangers

Standard heat transfer equations were used to calculate the overall heat transfer coefficient, log mean temperature difference, heat balance, and the overall area. The shell diameter, outer tube limit, and length to diameter ratio's were determined using the method outlined by Bell (1,4). The shell side heat transfer coefficient was obtained using the Delaware method. The Delaware method is the solution of shell side flow pressure drop and heat transfer coefficient by stream analysis method (2,3,5). The tube side heat transfer coefficient is calculated using the Hausen equation for laminar flow ($Re<2000$) and the Sieder-Tate equation for turbulent flow ($Re>10,000$). Interpolation is used in the transition region ($2000<Re<10,000$).

C.2. Air-Cooled Exchangers

The fin dimensions and areas are calculated using methods given in Bell's notes (5). The heat duty and log mean temperature difference are calculated using standard methods. The outlet air temperature, exchanger area, dimensions, number of tubes, number and power of each fan are calculated using methods given in the GPSA manual (7). The tube side heat transfer coefficient and pressure drop were calculated using the same methods as for the shell and tube exchangers. The air side heat transfer coefficient and the pressure drop were calculated using the methods given in the GPSA manual (7).

D. Overall Program Flow

The input/calculational procedure followed by the program is shown in Figure 1. The first step is to design the problem to be solved. The inputs for the designated type of problem (i.e. air-cooled or shell and tube exchangers) are then requested. The designated heat exchanger design calculation is performed. Stream temperatures, properties, etc. can be changed in the EDIT sections. The program is equipped to store and retrieve information from previous runs.

E. Summary

A brief discription of the capabilities of HEXSIM has been given. The details of using the program-data input, use of EDIT commands,

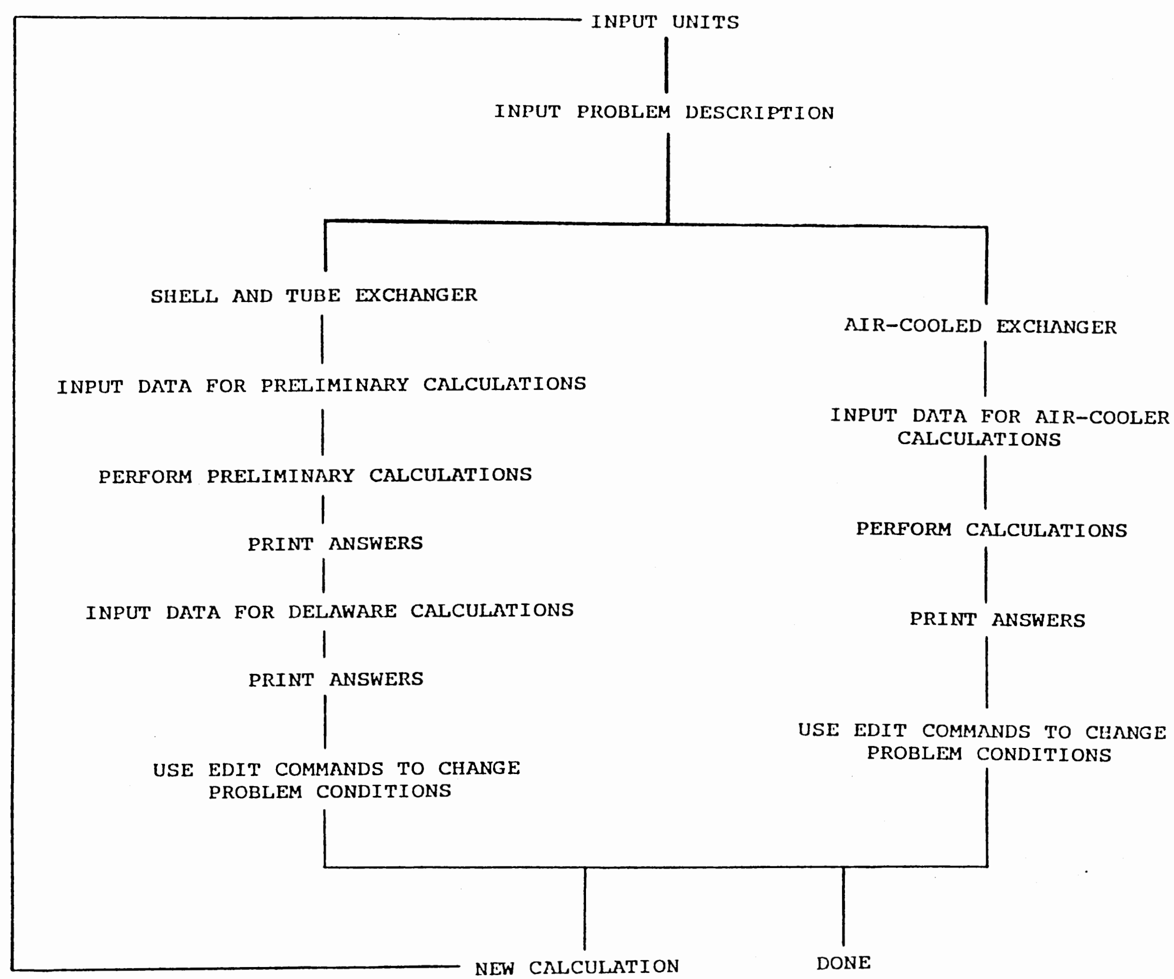


Figure 1. Logic Diagram for HEXSIM

selection of the type of exchanger, interpretation of output are covered in later sections of the documentation.

CHAPTER II

HEXSIM PROGRAM INPUT

A. Overview

There are basically two different modes which can be used to input data in HEXSIM. These are the basic problem input data and the EDIT section. This chapter reviews both forms of data input. If fractional inputs are required, they must have a decimal point. If an input must be an interger, this will be indicated.

Samples for each input are provided for your guidance. All entries are terminated by a carriage return.

B. Basic Problem Input

The basic problem input data are required to initiate HEXSIM calculations or to change from the current system to a completely different system. The basic input data requirements depend to a certain extent on the type of calculation to be performed. Only the inputs required for a given calculation type are requested; therefore there is no set input for all problems. The following sections give the inputs required for each general type of problem.

C. Inputs

HEXSIM first requests information about the nature of the problem by printing:

HEXSIM CLIST

TO RUN HEXSIM ON IBM/TSO SYSTEMS INPUT/OUTPUT FILES NEED TO BE ALLOCATED WITH THEIR DD STATEMENTS, THIS CLIST (COMMAND LIST) ALLOCATES THOSE FILES INTERACTIVELY.

RUNNING HEXSIM VERSION 1.0 INTERACTIVELY

DO YOU NEED TO ALLOCATE I/O FILES FOR THIS RUN (Y OR N):

Respond with either a Y for yes or an N for no. If you do not plan to save the input data (file) from this run answer N for no. If you think you would like to save data or create a file answer Y for yes.

If you answer yes the CLIST execution will continue by printing the following statement.

ALLOCATING DATAFILES

WAS THE I/O FILE PREVIOUSLY ALLOCATED (OLD FILE) ? (Y OR N):

Respond with a Y or N. If you have previously allocated a file then answer Y for yes if you have not allocated a file answer N for no. If you answered yes the following will be printed.

ALLOCATING FILE AND FILE UNIT NUMBER

ENTER FILE NUMBER (NO EXTENSION):

Respond with the file name.

ENTER FILE UNIT NUMBER (TWO DIGITS .... 08-99):

Respond with a two digit number between 08 and 99. This is the file unit number for this problem. When the file has been created the following message will be printed.

ALLOCATING DATAFILE * U11111A.FILENAME * AS DD FILE * FT08F001 * (SHR)

CALLING THE PROGRAM HEXSIM

If you answered no to the question "WAS THE I/O FILE PREVIOUSLY ALLOCATED (OLD FILE)?", it will be assumed that the file to be allocated is new. Thus a request for the file name and unit number will be made. The following will then be printed.

ALLOCATING DATAFILE * U11111A.FILENAME* AS DD FILE* (NEW)

CALLING THE PROGRAM HEXSIM

A sample input session is shown in Table I. Next the program HEXSIM is entered. HEXSIM then prints the following.

IS THIS A RESTART: YES OR NO?

Respond with either a Y or N. If you have saved a file using the FL feature in the EDIT command system, you can restart the calculations from that file without going through the normal input dialogue by answering Y. If you respond with a Y, HEXSIM will ask for a logical unit number by printing:

ENTER LOGIC UNIT #?

Respond with the value that was entered for the FL command which was used to create the file. This number must also agree with file number allocated. HEXSIM will then go directly to the EDIT command system. Note: Use of this command and the exact inputs will be installation dependent.

DO YOU WANT TO CHANGE THE UNIT BASIS: YES OR NO?

Respond with a Y or N. If you enter a Y, HEXSIM will request the new unit basis according to the standard units dialogue. If you respond with an N, HEXSIM will proceed directly to the EDIT segment of the program.

If you responded to the RESTART question with an N, you will enter the normal problem input system and HEXSIM will request the problem units system by printing:

ENTER TEMPERATURE UNITS: 0-F,1-R,2-C,3-K?

TABLE I

FILE ALLOCATION DIALOGUE

HEXSIM CLIST

TO RUN HEXSIM ON THE IBM/TSO SYSTEMS INPUT/OUTPUT FILES NEED TO BE ALLOCATED WITH THEIR DD STATEMENTS, THIS CLIST (COMMAND LIST) ALLOCATES THOSE FILES INTERACTIVELY.

RUNNING HEXSIM VERSION 1.0 INTERACTIVELY

DO YOU NEED TO ALLOCATE I/O FILES FOR THIS RUN ? (Y OR N): Y

ALLOCATING DATAFILES

WAS THE I/O FILE PREVIOUSLY ALLOCATED (OLD FILE)? (Y OR N): N

ALLOCATING FILE AND FILE UNIT NUMBER

ENTER FILE NAME (NO EXTENSION): TEST

ENTER FILE UNIT NUMBER (TWO DIGITS .... 08-99): 08

ALLOCATING DATAFILE * U11111A.TEST * AS DD FILE * FT08F001 *
(NEW)

CALLING THE PROGRAM HEXSIM

Respond with 0, 1, 2, or 3 to indicate your desired input/output units for temperature. Enter an integer only for all units input.

ENTER PRESSURE UNITS: 0-PSIA,1-ATM-2-KPA,3-BAR,4-MPA,5-ATA?

Respond with an integer (0, 1, 2, 3, 4, or 5) to define your pressure units. Note: ATA means absolute technical atmosphere, kg/cm2.

ENTER VISCOSITY UNITS: 0-LB/FT-HR,1-CP,2-KG/M-HR?

Respond with 0, 1, or 2 to indicate viscosity units in which to work.

ENTER UNITS FOR CP,TC,ECT: 0-US,1-METRIC,2-SI?

Respond with 0, 1, or 2 to indicate which system of units in which you would like to work.

IS THIS AN AIR-COOLED OR SHELL & TUBE EXCHANGER; AC OR ST?

If design of an air-cooled heat exchanger is to be performed enter AC or if a shell and tube exchanger design is desired enter a ST.

Sample input dialogue is given in Table II.

D. Shell and Tube Exchanger Input

GO THRU PRELIMINARY DESIGN CALC: YES OR NO?

Respond with either Y for yes or N for no. A yes response will start the requests for the data required for preliminary calculations. The preliminary calculations result in a heat balance within 1%, overall heat transfer coefficient from estimated individual coefficients, log mean temperature difference, area per shell, and a list of acceptable inside shell diameters, outer tube limits, number of tubes, length of shell, length of tubes, and length to diameter ratios. A no response will take you directly to the Delaware calculations. Note: It is preferable to first run a preliminary calculation before proceeding to

TABLE II

UNITS INPUT DIALOGUE

```
IS THIS A RESTART: YES OR NO? 0

ENTER TEMPERATURE UNITS:0-F,1-R,2-C,3-K? 0

ENTER PRESSURE UNITS:0-PSIA,1-ATM,2-KPA,3-BAR,4-MPA,5-ATA? 0

ENTER VISCOSITY UNITS:0-LB/FT-HR,1-CP,2-KG/M-HR? 0

ENTER UNITS FOR CP,TC,ECT:0-US,1-METRIC,2-SI? 0

IS THIS AN AIR-COOLED OR SHELL & TUBE EXCHANGER; AC OR ST? ST
```

the Delaware calculations in order to get an estimate of shell diameter, tube length, etc.

<u>D.1. Preliminary Shell and Tube Calculation Input</u>

ENTER PROBLEM DESCRIPTION?

Any valid keyboard character can be used. Up to 60 characters will be retained for subsequent problem identification. The summary sheets will print the title in 3 lines with 24 characters on each of the first two lines and 12 characters on the third line.

ENTER TUBE MASS FLOW RATE,LB/HR?

Respond with the mass flow rate of the fluid in the tube side of the exchanger. Note: For the purpose of explanantion U.S. units are used in this section.

ENTER TUBE INLET TEMPERATURE,DEG F?

Respond with the inlet temperature to the tube side in degrees Fahrenheit.

ENTER TUBE OUTLET TEMPERATURE,DEG F?

Respond with the outlet temperature from the tube side.

ENTER TUBE SPECIFIC HEAT, BTU/LB-R?

Respond with the specific heat of the fluid on the tube side.

ENTER SHLL MASS FLOW RATE,LB/HR?

Respond with the mass flow rate of the fluid in the shell side of the exchanger.

ENTER SHLL INLET TEMP,DEG F?

Respond with the inlet temperature to the shell side.

ENTER SHLL OUTLET TEMP,DEG F?

Respond with the outlet temperature from the shell side.

ENTER SHLL SPECIFIC HEAT, BTU/LB-R?

Respond with the specific heat of the shell side fluid.

ENTER NUMBER OF SHLL PASSES?

Respond with the number of shells in series. This number must be an integer.

ENTER NUMBER OF TUBE PASSES?

Respond with the total number of tube passes. This number must be an integer. HEXSIM accepts only 1, 2, 4, 6, or 8 tube passes per shell. The number entered us number of tube passes per shell multiplied by the number of shells.

ENTER TUBE OD,IN?

Respond with the tube outside diameter in inches.

ENTER TUBE ID,IN?

Respond with the tube inside diameter.

ENTER TUBE PITCH,IN?

Respond with the tube pitch. HEXSIM only accepts a pitch to diameter ratio of 1.25 or 1.3.

ENTER THERM COND OF TUBE METAL,BTU/HR-FT-R?

Respond with the thermal conductivity of the tube metal.

ENTER TUBE ARRANGEMENT: T-TRIANGULAR, S-INLINE SQUARE, R-ROTATED SQUARE?

Respond with T, S, or R for the tube arrangement. Figure 2 shows a drawing of these tube arrangements.

**ENTER TYPE OF TUBE BUNDLE CONSTRUCTION:
S-SPLIT BACKING RING,P-OUTSIDE PACKED FLOATING HEAD,
U-U TUBE,T-PULL THRU FLOAT HEAD,F-FIXED TUBE SHEET?**

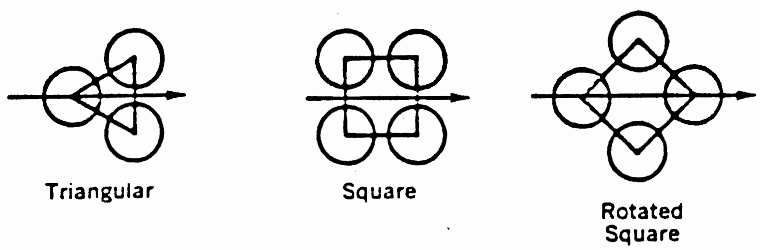


Figure 2. Tube Arrangement (14)

Respond with S, P, U, T, or F for the type of exchanger construction type desired. Figure 3 shows these exchanger configurations.

ENTER ESTIMATED TUBE SIDE H,BTU/FT2-HR-R?

Respond with an estimated value for the tube side individual heat transfer coefficient. Guidelines for estimating individual heat transfer coefficients and fouling factors are given in Chapter III.

ENTER ESTIMATED SHLL SIDE H,BTU/FT2-HR-R?

Respond with an estimated value for the shell side individual heat transfer coefficient.

ENTER ESTIMATED TUBE SIDE FOULING FACTOR,(BTU/FT2-HR-R)-1?**

Respond with an estimated fouling factor for the tube side fluid.

ENTER ESTIMATED SHLL SIDE FOULING FACTOR,(BTU/FT2-HR-R)-1?**

Respond with an estimated value for the shell side fouling factor.

ARE YOUR INPUT DATA ALL OK; YES OR NO?

If you respond with a Y for yes, the calculation will be performed and the summary sheet printed for the preliminary calculations. If you respond with N for no, HEXSIM will proceed to the EDIT mode. The EDIT session is started by asking the following.

ENTER NEXT COMMAND?

One of the responses from the EDIT section should be given.

A sample input session for the preliminary calculations is shown in Table III.

D.2. Delaware Calculation Inputs

GO THRU DELAWARE CALCS; YES OR NO?

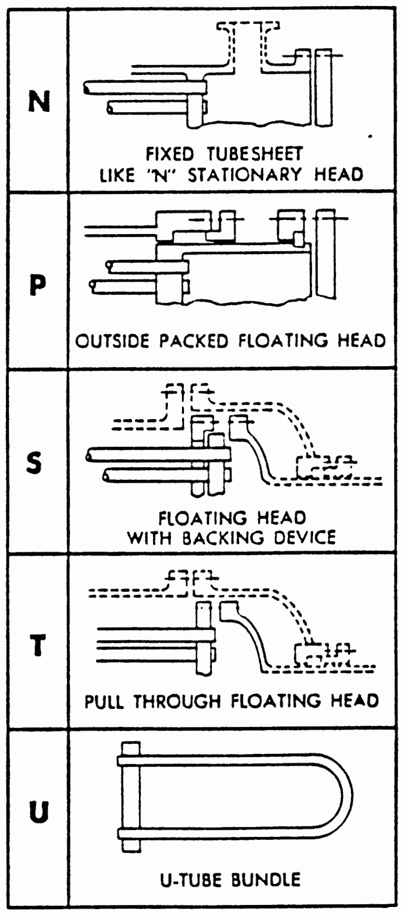


Figure 3. Tube Bundle Construction (5,14)

TABLE III

SAMPLE INPUT FOR PRELIMINARY CALCULATIONS

```
ENTER PROBLEM DESCRIPTION ? EXAMPLE

ENTER TUBE MASS FLOW RATE,LB/HR? 465000.

ENTER TUBE INLET TEMP,DEG F? 80.

ENTER TUBE OUTLET TEMP,DEG F? 115.

ENTER TUBE SPECIFIC HEAT,BTU/LB-R? 1.0

ENTER SHLL MASS FLOW RATE,LB/HR? 180000.

ENTER SHLL INLET TEMP,DEG F? 235.

ENTER SHLL OUTLET TEMP,DEG F? 100.

ENTER SHLL SPECIFIC HEAT,BTU/LB-R? 0.667

ENTER NUMBER OF SHLL PASSES? 2

ENTER NUMBER OF TUBE PASSES? 4

ENTER TUBE OD,IN? 0.75

ENTER TUBE ID,IN? 0.68

ENTER TUBE PITCH,IN? 0.9375

ENTER THERM COND OF TUBE METAL,BTU/HR-FT-R? 26.

ENTER TUBE ARRANGEMENT: T-TRIANGULAR, S-INLINE SQUARE,R-ROTATED
SQUARE? T

ENTER TYPE OF BUNDLE CONSTRUCTION:
S-SPLIT BACKING RING,P-OUTSIDE PACKED FLOATING HEAD,
U-U TUBE,T-PULL THRU FLOAT HEAD,F-FIXED TUBE SHEET? S

ENTER ESTIMATED TUBE SIDE H,BTU/FT2-HR-R? 1000.

ENTER ESTIMATED SHLL SIDE H,BTU/FT2-HR-R? 350.

ENTER ESTIMATED TUBE SIDE FOULING FACTOR,(BTU/FT2-HR-R)**-1? .001

ENTER ESTIMATED SHLL SIDE FOULING FACTOR,(BTU/FT2-HR-R)**-1? .005

ARE YOUR DATA ALL OK; YES OR NO? Y
```

If you respond with N for no the program will ask "ENTER NEXT COMMAND?". If you respond with Y for yes HEXSIM will proceed with the Delaware calculations. The Delaware calculations checks the heat balance, calculates individual and overall heat transfer coefficients, pressure drop on both shell and tube side, and total area. The default values used in this section are for class R construction. Class R is for petroleum refinery use as given in TEMA standards (14).

DO YOU WANT TO ENTER MASS FLOWS, TEMPS, ETC;YES OR NO?

If you went through the preliminary calculations, it is not necessary to re-enter mass flow rates, temperatures, ect., but if you wish to make changes they can be modified at this point. If you respond with Y for yes, HEXSIM will repeat the input questions for preliminary calculations shown in Table III. The preliminary calculations will not be repeated. If you respond with N, HEXSIM will proceed to ask for the data necessary for the Delaware calculations.

DO YOU WANT THE OUTLET TEMPERATURES THE SAME; YES OR NO?

If you want to keep the outlet temperatures the same as what you entered then respond with Y for yes. If you want the outlet temperatures adjusted to correspond to the length specified respond with N for no.

ENTER PHASE OF SHLL SIDE FLUID; 1-LIQ OR 2-GAS?

Respond with a 1 or 2 for the phase of the shell side fluid; where a 1 is for a liquid and 2 is for a gas.

ENTER PHASE OF TUBE SIDE FLUID; 1-LIQ OR 2-GAS?

Respond with a 1 or 2 for the phase of the tube side fluid; where 1 is for a liquid and 2 is for a gas.

ENTER MAX TUBE SIDE PRESSURE DROP, PSI?

Respond with the maximum tube side pressure drop. The default value for the maximum tube side pressure drop is 10 psi.

ENTER MAX SHLL SIDE PRESSURE DROP, PSI?

Respond with the maximum shell side pressure drop. The default value for the maximum shell side pressure drop is 10 psi.

ENTER TUBE SIDE FLUID DENSITY,LB/FT3?

Respond with the density of the tube side fluid.

ENTER SHLL SIDE FLUID DENSITY,LB/FT3?

Respond with the density of the shell side fluid.

ENTER TUBE SIDE FLUID VISCOSITY AT WALL,LB/FT-HR?

Respond with the tube side fluid viscosity at the tube wall.

ENTER SHLL SIDE FLUID VISCOSITY AT WALL,LB/FT-HR?

Respond with the shell side fluid viscosity at the wall.

ENTER TUBE SIDE FLUID BULK VISCOSITY,LB/FT-HR?

Respond with the bulk viscosity of the tube side fluid.

ENTER SHLL SIDE FLUID BULK VISCOSITY,LB/FT-HR?

Respond with the bulk viscosity of the shell side fluid.

ENTER TUBE SIDE FLUID THERMAL CONDUCTIVITY,BTU/HR-FT-R?

Respond with the thermal conductivity of the tube side fluid.

ENTER SHLL SIDE FLUID THERMAL CONDUCTIVITY,BTU/HR-FT-R?

Respond with the thermal conductivity of the shell fluid.

ENTER BAFFLE SPACING,IN?

Respond with the baffle spacing in the interior of the exchanger. The default value for the baffle spacing is one half the inside shell diameter for a liquid and equal to the shell diameter for a gas.

ENTER INLET BAFFLE SPACING,IN?

Respond with the baffle spacing at the inlet of the exchanger. The default value for the inlet baffle spacing is one half the inside shell diameter for a liquid and equal to the shell diameter for a gas.

ENTER OUTLET BAFFLE SPACING,IN?

Respond with the baffle spacing at the outlet of the exchanger. The default value for the outlet baffle spacing is one half the inside shell diameter for a liquid and equal to the shell diameter for a gas.

ENTER % BAFFLE CUT?

Respond with the baffle cut expressed in percent. The percent baffle cut is equal to the baffle cut distance from baffle tip to shell inside diameter per shell inside diameter times one hundred. The percent baffle cut should range from 9-49 %. If the length baffle cut is entered this value is calculated. The default value for the baffle cut is 25% for a liquid and 45% for a gas.

ENTER LENGTH BAFFLE CUT,IN?

Respond with the baffle cut distance from tip to shell inside diameter. This number is best estimated by deciding on a baffle cut (percent baffle cut/100) and multiplying this number by the shell inside diameter. If the percent baffle cut is specified then the length is calculated.

ENTER TUBE TO BAFFLE CLEARANCE,IN?

Respond with clearance between the tube and the baffle holes, where the clearance is the difference between the tube outside diameter and the tube holes cut in the baffle. TEMA class R (14) construction specifies a tube to baffle clearance of 1/32 inches where the maximum unsupported tube length does not exceed 36 inches and specifies a tube to baffle clearance of 1/64 inches otherwise. These values are default values but if extra tight construction is desired other values can be used.

ENTER SHLL TO BAFFLE CLEARANCE,IN?

Respond with the clearance between the baffle and the inside shell diameter where the clearance is the difference between the shell inside diameter and the outside diameter of the baffle. TEMA standards (14) for shell to baffle clearances given in Table IV. The values given in Table IV are default values.

ENTER NUMBER OF SEALING STRIPS/SIDE?

Respond with the number of sealing strips per side.

ENTER NUMBER OF PASS PARTITION LANES?

Respond with the number of pass partition lanes. A pass partition lane is a divider through the tube field parallel to the direction of the crossflow.

ENTER WIDTH OF PASS PARTITION LANES,IN?

Respond with the width of the pass partition lane.

ENTER NUMBER OF TUBES?

Respond with the total number of tubes in the exchanger.

ENTER SHLL DIAMETER,IN?

Respond with the inside shell diameter.

ENTER DIAMETER OF OUTER TUBE LIMIT,IN?

Respond with the diameter of outer tube limit.

ENTER TUBE LENGTH,FT?

Respond with the tube length.

DO YOU WANT INTERMEDIATE OUTPUT; YES OR NO?

TABLE IV

TEMA STANDARDS FOR SHELL TO BAFFLE CLEARANCES (5,14)

Di, in	Diametral shell-to-baffle clearance, in.
8-13	0.100*
14-17	0.125
18-23	0.150
24-39	0.175
40-54	0.225
55-60	0.300

*These values are for pipe shells; if rolled shells are used, add 0.125. Default values are for pipe shells.

If you respond with Y for yes, HEXSIM will print values for the Delaware parameters. If you respond with N, HEXSIM will not print the Delaware parameters.

A sample input session for the Delaware calculation is shown in Table V.

E. Air-Cooled Exchanger Input

ENTER PROBLEM DESCRIPTION?

Any valid keyboard character can be used. Up to 60 characters will be retained for subsequent problem identification. The summary sheets will print the title in three lines with 24 characters on each of the first two lines and 12 characters on the third.

ENTER TUBE MASS FLOW RATE,LB/HR?

Respond with the mass flow rate of the fluid in the tube side of the exchanger.

ENTER TUBE INLET TEMP, DEG F?

Respond with the inlet temperature to the tube side in degrees fahrenheit.

ENTER TUBE OUTLET TEMP, DEG F?

Respond with the outlet temperature from the tube side.

ENTER TUBE SPECIFIC HEAT,BTU/LB-R?

Respond with the specific heat of the tube side fluid.

ENTER TUBE SIDE FLUID DENSITY,LB/FT3?

Respond with the density of the tube side fluid.

ENTER TUBE SIDE FLUID VISCOSITY AT THE WALL, LB/FT-HR?

Respond with the tube side fluid viscosity at the tube wall.

ENTER TUBE SIDE BULK VISCOSITY,LB/FT-HR?

TABLE V

SAMPLE INPUT SESSION FOR DELAWARE CALCULATIONS

```
DO YOU WANT ENTER MASS FLOW,TEMPS,ETC; YES OR NO? N
DO YOU WANT THE OUTLET TEMPERATURES THE SAME; YES OR NO? N
ENTER PHASE OF SHLL SIDE FLUID; 1-LIQ OR 2-GAS? 1
ENTER PHASE OF TUBE SIDE FLUID; 1-LIQ OR 2-GAS? 1
ENTER MAX TUBE SIDE PRESSURE DROP, PSI? 10.0
ENTER MAX SHLL SIDE PRESSURE DROP, PSI? 10.0
ENTER TUBE SIDE FLUID DENSITY,LB/FT3? 62.
ENTER SHLL SIDE FLUID DENSITY,LB/FT3? 31.8
ENTER TUBE SIDE FLUID VISCOSITY AT WALL,LB/FT-HR? 1.6498
ENTER SHLL SIDE FLUID VISCOSITY AT WALL,LB/FT-HR? .344
ENTER TUBE SIDE BULK VISCOSITY,LB/FT-HR? 1.6828
ENTER SHLL SIDE BULK VISCOSITY,LB/FT-HR? 0.342
ENTER TUBE SIDE THERMAL CONDUCTIVITY,BTU/HR-FT-R? 0.364
ENTER SHLL SIDE THERMAL CONDUCTIVITY,BTU/HR-FT-R? 0.0685
ENTER BAFFLE SPACING,IN? 12.
ENTER INLET BAFFLE SPACING,IN? 12.
ENTER OUTLET BAFFLE SPACING,IN? 12.
ENTER % BAFFLE CUT? 25.
ENTER LENGTH TO BAFFLE CUT,IN? 5.25
ENTER TUBE TO BAFFLE CLEARANCE,IN? 0.03215
ENTER SHLL TO BAFFLE CLEARANCE,IN? 0.15
ENTER NUMBER OF SEALING STRIPS/SIDE? 2.
ENTER NUMBER OF PASS PARTITION LANES? 0
ENTER WIDTH OF PASS PARTITION LANES? 0
```

TABLE V (CONTINUED)

```
ENTER NUMBER OF TUBES? 342.

ENTER SHLL DIAMETER,IN? 21.

ENTER DIAMETER OF OUTER TUBE LIMIT,IN? 19.5

ENTER TUBE LENGTH,FT? 14.74

ARE YOUR INPUT ALL OK; YES OR NO? Y

DO YOU WANT INTERMEDIATE OUTPUT; YES OR NO? Y
```

Respond with the tube side fluid average viscosity.

ENTER TUBE SIDE FLUID THERMAL CONDUCTIVITY, BTU/HT-FT-R?

Respond with the thermal conductivity of the fluid in the tubes.

ENTER NUMBER OF TUBE PASSES?

Respond with the number of tube passes. This number must be an integer.

ENTER TUBE OD,IN?

Respond with the tube outside diameter. Base tube diameters are usually 5/8 to 1 1/2 inches (7).

ENTER TUBE ID,IN?

Respond with the tube inside diameter.

ENTER TUBE PITCH,IN?

Respond with the tube pitch in inches. The tube pitch should be arranged such that the fin tips of adjacent tubes are touching or separated by 1/16 to 1/4 of an inch (7).

ENTER THERMAL CONDUCTIVITY OF TUBE METAL, BTU/HR-FT-R?

Respond with the thermal conductivity of the base tube metal.

ENTER TUBE ARRANGEMENT; T-TRIANGULAR,S-INLINE SQUARE,R-ROTATED SQUARE?

Respond with T, S, or R for the tube arrangement. For air-cooled heat exchangers triangular layout is the most common arrangement (7). Inline-square arrangements are never used with high-finned tubes. It has been demonstrated that the air will tend to flow in the channels between the fin tips from the inlet to the tube bank exit. The net result is a very severe deterioration in the apparent heat transfer coefficient (5).

ENTER ESTIMATED TUBE SIDE FOULING FACTOR,(BTU/FT2-HR-R)-1?**

Respond with the tube side fouling factor. See Chapter III for guidelines.

ENTER AIR SIDE FOULING FACTOR (BTU/HR-FT2-R)-1?**

Respond with the air side fouling factor. The default value for the air-side fouling factor is 1×10^{-15} ((hr-ft^2-R)/Btu).

ENTER FIN HEIGHT, IN?

Respond with the fin height in inches. Fin heights range from 1/2 to 1 inch. See Figure 4 for definition of geometerical parameters for high-finned tubes (5). The default value for the fin height is 0.625 inches.

ENTER MEAN FIN THICKNESS, IN?

Respond with the mean fin thickness. Fins are usually aluminum, with an average thickness of 0.012 to 0.02 inches. The default value for the fin thickness is 0.015 inches.

ENTER NUMBER OF FINS/IN?

Respond with the number of fins per inch. Fins are usually spaced from 7 to 11 per inch (7). The default value for the number of fins per inch is 10.0.

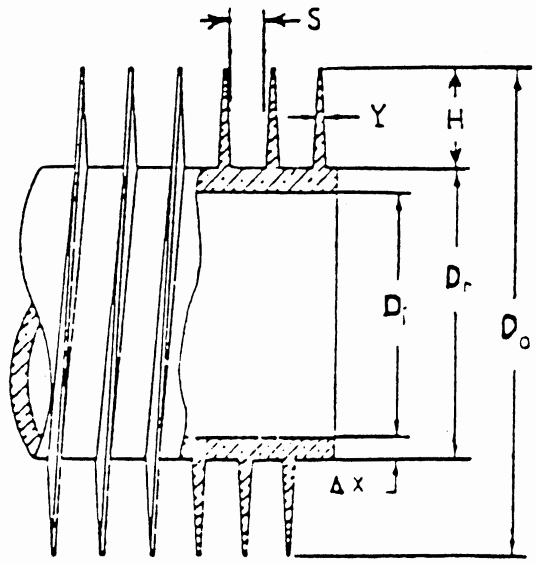
ENTER FIN THERMAL CONDUCTIVITY, BTU/HR-FT-R?

Respond with the fin thermal conductivity. The fins are typically made of aluminum, while the base tube is low carbon steel (5). The default value for the fin thermal conductivity is 119 Btu/hr ft°R for aluminum fins.

ENTER ALTITUDE, FT?

Respond with the altitude of the exchanger location.

ENTER NUMBER OF TUBE ROWS?



D_r - Root diameter of finned section

D_o - Diameter over fins

D_i - Inside diameter of finned section

Δx - Wall thickness of finned section

Y - Mean fin thickness

H - Height of fin

S - Space between fins

Figure 4. Geometrical Parameters For High Finned Tubes (5)

Respond with the number of tube rows. The number of tube rows or bundle depth usually ranges from 3 to 8 rows with 4 rows being the most common (7). Bundles may be stacked, resulting in a total depth of up to 30 rows of tubes.

ENTER TUBE LENGTH, FT?

Respond with the desired tube length if known. If the tube length is not known press the carriage return, HEXSIM will then calculate the unit width for several standard tube lengths. The standard tube lengths are 8, 10, 12, 20, 24, and 30 feet.

ENTER INLET AIR TEMPERATURE, DEG F?

Respond with the inlet air temperature. This air temperature is typically a temperature not exceeded more than 5% of the time during the hottest months of the year or the average daily maximum temperature for the hottest month of the year (12).

ENTER NUMBER OF FANS?

Respond with the number of fans desired. If the number of fans needed is not known HEXSIM will calculate the number of fans required.

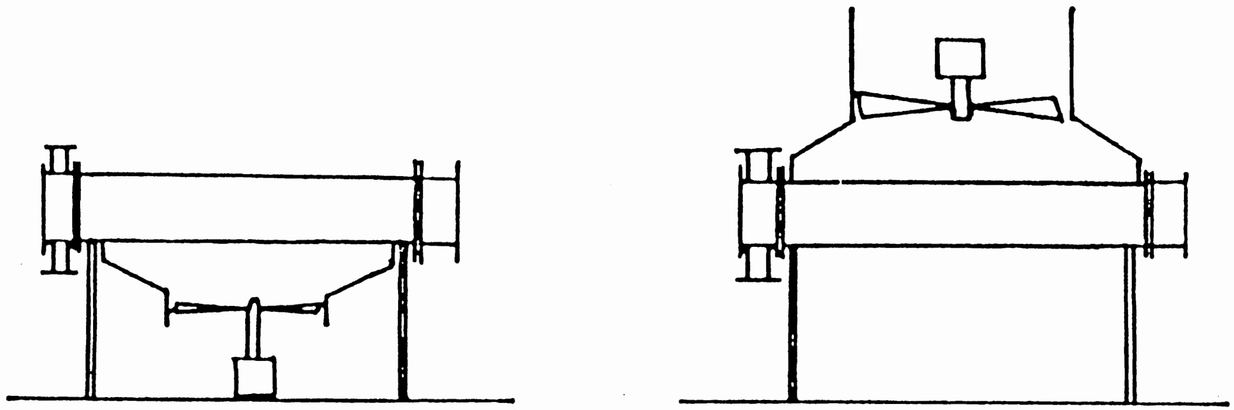
ENTER FAN DRIVER EFFICIENCY, %?

Respond with the percent fan driver efficiency. The default value is 70%.

ENTER TYPE OF DRAFT; FORCED=F, INDUCED=I?

Respond with the type of fan draft desired. Both types are illustrated in Figure 5.

A sample input session for air-cooled heat exchangers is shown in Table VI.



Forced Draft

Induced Draft

Figure 5. Type of Draft for Air-Cooler (5)

TABLE VI

SAMPLE INPUT SESSION FOR AIR-COOLED EXCHANGERS

```
ENTER PROBLEM DESCRIPTION ? EXAMPLE

ENTER TUBE MASS FLOW RATE,LB/HR? 38000.

ENTER TUBE INLET TEMP, DEG F? 170.

ENTER TUBE OUTLET TEMP, DEG F? 124.

ENTER TUBE SPECIFIC HEAT, BTU/LB-R? .609

ENTER TUBE SIDE FLUID DENSITY, LB/FT3? 2.5

ENTER TUBE SIDE VISCOSITY AT WALL,LB/FT-HR? .02

ENTER TUBE SIDE FLUID BULK VISCOSITY,LB/FT-HR? 0.021

ENTER TUBE SIDE FLUID THERMAL CONDUCTIVITY,BTU/HR-FT-R? 0.013

ENTER NUMBER OF TUBE PASSES? 2

ENTER TUBE OD,IN? 1.

ENTER TUBE ID,IN? 0.93

ENTER TUBE PITCH,IN? 2.25

ENTER THERMAL CONDUCTIVITY OF TUBE METAL,BTU/HR-FT-R? 26.

ENTER TUBE ARRANGEMENT;T-TRIANGULAR,S-INLINE SQUARE,R-ROTATED
SQUARE? T

ENTER ESTIMATED TUBE SIDE FOULING FACTOR,(BTU/FT2-HR-R)**-1? .005

ENTER ESTIMATED AIR SIDE FOULING FACTOR,(BTU/FT2-HR-R)**-1? .001

ENTER FIN HEIGHT,IN? .625

ENTER MEAN FIN THICKNESS,IN? 0.015

ENTER NUMBER OF FINS/IN? 11.

ENTER FIN THERMAL CONDUCTIVITY,BTU/HR-FT-R? 119.

ENTER ALTITUDE,FT? 0

ENTER NUMBER OF TUBE ROWS? 4.
```

TABLE VI (CONTINUED)

```
ENTER TUBE LENGTH,FT?
ENTER INLET AIR TEMPERATURE, DEG F? 92.
ENTER NUMBER OF FANS?
ENTER FAN DIAMETER, FT?
ENTER FAN DRIVER EFFICIENCY, %?
ENTER TYPE OF DRAFT; FORCED=F, INDUCED=I? F
ARE YOUR INPUT DATA ALL OK; YES OR NO? Y
```

F. Exchanger Program Data Input/Transfer

This section of the input data system allows the user to manipulate the mass flow rate, temperature, physical properties, etc. The basic guidelines for using the EDIT feature are discussed in this section. From a program use standpoint, the EDIT commands can be issued in any order.

G. Effective Use Of Edit/Hexsim

Experience with HEXSIM has allowed us to evolve a strategy for using EDIT/HEXSIM to minimize problems with the EDIT commands and data input. These guidelines are:

1. For the first problem, select a simple calculation such as a single shell exchanger. Carefully check input data, i.e. the temperature, mass flow rates, specific heats, etc. If any errors are found, correct them using the appropriate EDIT commands and run the preliminary design to verify that the problem is now correctly formulated.
2. After the problem is correctly formulated, run the problem as specified.

TABLE VII

PRELIMINARY CALCULATION ENTRY/REVISION COMMANDS

MNEMONIC	FUNCTION
MT	ENTER TUBE MASS FLOW RATE
IT	ENTER TUBE INLET TEMPERATURE
OT	ENTER TUBE OUTLET TEMPERATURE
TP	ENTER TUBE SPECIFIC HEAT
MS	ENTER SHELL MASS FLOW RATE
IS	ENTER SHELL INLET TEMPERATURE
OS	ENTER SHELL OUTLET TEMPERATURE
SP	ENTER SHELL SPECIFIC HEAT
PS	ENTER NUMBER OF SHELL PASSES
PT	ENTER NUMBER OF TUBE PASSES
OD	ENTER TUBE OUTSIDE DIAMETER
ID	ENTER TUBE INSIDE DIAMETER
PI	ENTER TUBE PITCH
KT	ENTER THERMAL CONDUCTIVITY OF TUBE METAL
TA	ENTER TUBE ARRANGEMENT:T-TRIANGULAR, S-INLINE SQUARE,R-ROTATED SQUARE
ST	ENTER TYPE OF BUNDLE CONSTRUCTION: S-SPLIT BACKING RING,P-OUTSIDE PACKED FLOATING HEAD,U-U TUBE,T-PULL THRU FLOAT HEAD,F-FIXED TUBE SHEET
HT	ENTER ESTIMATED TUBE SIDE HEAT TRANSFER COEFFICIENT
HS	ENTER ESTIMATED SHELL SIDE HEAT TRANSFER COEFFICIENT
TF	ENTER ESTIMATED TUBE SIDE FOULING FACTOR
SF	ENTER ESTIMATED SHELL SIDE FOULING FACTOR

TABLE VIII

DELAWARE CALCULATION ENTRY/REVISION COMMANDS

MNEMONIC	FUNCTION
TD	ENTER MAX TUBE SIDE PRESSURE DROP
SE	ENTER MAX SHLL SIDE PRESSURE DROP
RT	ENTER TUBE SIDE DENSITY
RS	ENTER SHELL SIDE DENSITY
WT	ENTER TUBE SIDE VISCOSITY AT THE WALL
WS	ENTER SHELL SIDE VISCOSITY AT THE WALL
VT	ENTER BULK VISCOSITY OF TUBE SIDE FLUID
VS	ENTER BULK VISCOSITY OF SHELL SIDE FLUID
TK	ENTER TUBE SIDE FLUID THERMAL CONDUCTIVITY
SK	ENTER SHELL SIDE FLUID THERMAL CONDUCTIVITY
BS	ENTER BAFFLE SPACING
BI	ENTER INLET BAFFLE SPACING
BO	ENTER OUTLET BAFFLE SPACING
BC	ENTER % BAFFLE CUT
BL	ENTER LENGTH TO BAFFLE CUT
CT	ENTER TUBE TO BAFFLE CLEARANCE
CS	ENTER SHELL TO BAFFLE CLEARANCE
NS	ENTER NUMBER OF SEALING STRIPS/SIDE
NP	ENTER NUMBER OF PASS PARTITION LANES
WP	ENTER WIDTH OF PASS PARTITION LANES
NT	ENTER NUMBER OF TUBES
SD	ENTER INSIDE DIAMETER

TABLE VIII (CONTINUED)

MNEMONIC	FUNCTION
DO	ENTER DIAMETER OF OUTER TUBE LIMIT
LT	ENTER TUBE LENGTH

TABLE IX

AIR-COOLED HEAT EXCHANGER ENTRY/REVISION COMMANDS

MNEMONIC	FUNCTION
MT	ENTER TUBE MASS FLOW RATE
IT	ENTER TUBE INLET TEMPERATURE
OT	ENTER TUBE OUTLET TEMPERATURE
TP	ENTER TUBE SIDE FLUID SPECIFIC HEAT
RT	ENTER TUBE SIDE DENSITY
WT	ENTER TUBE SIDE VISCOSITY AT THE WALL
VT	ENTER BULK VISCOSITY OF THE TUBE SIDE FLUID
TK	ENTER THERMAL CONDUCTIVITY OF THE TUBE SIDE FLUID
PT	ENTER NUMBER OF TUBE PASSES
OD	ENTER TUBE OUTSIDE DIAMETER
ID	ENTER TUBE INSIDE DIAMETER
PI	ENTER TUBE PITCH
KT	ENTER THERMAL CONDUCTIVITY OF TUBE METAL
TA	ENTER TUBE ARRANGEMENT;T-TRIANGULAR,S-INLINE SQUARE, R-ROTATED SQUARE
TF	ENTER ESTIMATED TUBE SIDE FOULING FACTOR
AF	ENTER ESTIMATED AIR SIDE FOULING FACTOR
FH	ENTER FIN HEIGHT
FT	ENTER MEAN FIN THICKNESS
NF	ENTER NUMBER OF FINS/UNIT LENGTH
FK	ENTER FIN THERMAL CONDUCTIVITY

TABLE IX (CONTINUED)

AT	ENTER ALTITUDE
AR	ENTER NUMBER OF TUBE ROWS
AI	ENTER INLET AIR TEMPERATURE
FN	ENTER NUMBER OF FANS
FD	ENTER FAN DIAMETER
FE	ENTER FAN DRIVER EFFICIENCY
DF	ENTER TYPE OF DRAFT; FORCED=F, INDUCED=I?

TABLE X

MISCELLANOUS ENTRY REVISION COMMANDS

MNEMONIC	I	FUNCTION
DL	-	PROCEED TO DELAWARE CALCULATIONS
DN	-	ALL CALCULATIONS ARE COMPLETE; STOP THE PROGRAM. ALL CALCULATIONS WILL BE LOST UNLESS THE FL OPTION IS EXERCISED.
FL	8-99	SAVE THE NECESSARY PROBLEM DATA IN A FILE SO THAT THE PROBLEM CAN BE RESTARTED AT SOME FUTURE TIME. I IS THE LOGICAL UNIT NUMBER.
NW	-	A NEW PROBLEM WILL BE STARTED. ALL PREVIOUS CALCULATIONS WILL BE LOST UNLESS THE FL OPTION IS EXERCISED.
PA	-	PRINTS AIR-COOLER SUMMARY OUTPUT
PD	-	PRINTS DELAWARE SUMMARY OUTPUT
PL	-	PRINTS SUMMARY OF PRELIMINARY CALCULATIONS
PP	-	PRINTS SPECIFIED FLUID PROPERTIES
RN	-	THE PROBLEM AS NOW DEFINED WILL BE RUN WITHOUT GOING THROUGH THE PREVIOUS DATA INPUT.
TI	-	ENTER A NEW PROBLEM TITLE

TABLE XI

ALPHABETICAL LISTING OF ENTRY/REVISION COMMANDS

MNEMONIC	I	FUNCTION
AF	-	ENTER AIR SIDE FOULING FACTOR
AI	-	ENTER INLET AIR TEMPERATURE
AL	-	ENTER TUBE LENGTH FOR AIR COOLERS
AR	-	ENTER NUMBER OF TUBE ROWS IN AIR COOLERS
AT	-	ENTER ALTITUDE
BC	-	ENTER % BAFFLE CUT
BI	-	ENTER INLET BAFFLE SPACING
BL	-	ENTER LENGTH OF BAFFLE CUT
BO	-	ENTER OUTLET BAFFLE SPACING
BS	-	ENTER BAFFLE SPACING
CS	-	ENTER SHELL TO BAFFLE CLEARANCE
CT	-	ENTER TUBE TO BAFFLE CLEARANCE
DF	-	ENTER TYPE OF DRAFT;FORCED=F,INDUCED=I
DL	-	PROCEED TO DELAWARE CALCULATIONS
DN	-	ALL CALCULATIONS ARE COMPLETE; STOP THE PROGRAM. ALL CALCULATIONS WILL BE LOST UNLESS THE FL OPTION WAS EXECUTED
DO	-	ENTER OUTER TUBE LIMIT DIAMETER
FD	-	ENTER FAN DIAMETER FOR AIR COOLER
FE	-	ENTER FAN DRIVER EFFICIENCY
FH	-	ENTER FIN HEIGHT
FK	-	ENTER FIN THERMAL CONDUCTIVITY
FL	8-99	SAVE THE NECESSARY PROBLEM DATA IN A FILE SO THAT THE PROBLEM CAN BE RESTARTED AT SOME FUTURE TIME. I IS THE LOGICAL UNIT NUMBER.

TABLE XI (CONTINUED)

MNEMONIC	I	FUNCTION
FN	-	ENTER NUMBER OF FANS ON AIR COOLER
FT	-	ENTER FIN THICKNESS FOR AIR COOLER
HS	-	ENTER ESTIMATED SHELL SIDE HEAT TRANSFER COEFFICIENT
HT	-	ENTER TUBE SIDE HEAT TRANSFER COEFFICIENT
ID	-	ENTER TUBE INSIDE DIAMETER
IS	-	ENTER SHELL SIDE INLET TEMPERATURE
IT	-	ENTER TUBE SIDE INLET TEMPERATURE
KT	-	ENTER THERMAL CONDUCTIVITY OF TUBE METAL
LT	-	ENTER TUBE LENGTH
MS	-	ENTER SHELL MASS FLOW RATE
MT	-	ENTER TUBE MASS FLOW RATE
NF	-	ENTER NUMBER OF FINS PER UNIT LENGTH
NP	-	ENTER NUMBER OF PASS PARTITION LANES
NS	-	ENTER NUMBER OF SEALING STRIPS PER SIDE
NT	-	ENTER NUMBER OF TUBES
NW	-	A NEW PROBLEM WILL BE STARTED. ALL PREVIOUS CALCULATIONS WILL BE LOST UNLESS THE FL OPTION IS EXERCISED.
OD	-	ENTER TUBE OUTSIDE DIAMETER
OS	-	ENTER SHELL OUTLET TEMPERATURE
OT	-	ENTER TUBE OUTLET TEMPERATURE
PA	-	PRINTS AIR COOLER SUMMARY OUTPUT
PD	-	PRINTS DELAWARE SUMMARY OUTPUT
PI	-	ENTER TUBE PITCH

TABLE XI (CONTINUED)

MNEMONIC	I	FUNCTION
PL	-	PRINTS SUMMARY OF PRELIMINARY CALCULATIONS
PP	-	PRINTS SPECIFIED FLUID PROPERTIES FOR DELAWARE METHOD
PS	-	ENTER NUMBER OF PASSES ON SHELL SIDE
PT	-	ENTER NUMBER OF PASSES ON TUBE SIDE
RN	-	THE PROBLEM AS NOW DEFINED WILL BE RUN WITHOUT GOING THROUGH THE PREVIOUS DATA INPUT.
RS	-	ENTER SHELL SIDE DENSITY
RT	-	ENTER TUBE SIDE DENSITY
SD	-	ENTER SHELL DIAMETER
SE	-	ENTER MAX SHELL SIDE PRESSURE DROP
SF	-	ENTER SHELL SIDE FOULING FACTOR
SK	-	ENTER SHELL SIDE THERMAL CONDUCTIVITY
SP	-	ENTER SHELL SIDE SPECIFIC HEAT
ST	-	ENTER TYPE OF BUNDLE CONSTRUCTION
TA	-	ENTER TUBE ARRANGEMENT
TD	-	ENTER MAX TUBE SIDE PRESSURE DROP
TF	-	ENTER TUBE SIDE FOULING FACTOR
TI	-	ENTER PROBLEM TITLE
TK	-	ENTER TUBE SIDE THERMAL CONDUCTIVITY
TP	-	ENTER TUBE SIDE SPECIFIC HEAT
VS	-	ENTER SHELL SIDE BULK VISCOSITY
VT	-	ENTER TUBE SIDE BULK VISCOSITY
WP	-	ENTER WIDTH OF PASS PARTITION LANE

TABLE XI (CONTINUED)

MNEMONIC	I	FUNCTION
WS	-	ENTER SHELL SIDE VISCOSITY AT THE WALL
WT	-	ENTER TUBE SIDE VISCOSITY AT THE WALL

CHAPTER III

USING HEXSIM

A. Overview

Effective use of any program, HEXSIM included, requires that the user be able to interpret the results, decide if the results are faulty and if so take corrective action. In this section you are given: (1) a brief discussion of the output format and its interpretation, (2) some check points which can be used to assess the validity of the results, (3) some suggestions about how you can improve the quality of your results, and (4) HEXSIM diagnostic comments their interpretation and possible corrective action.

B. Output Interpretation

B.1. Shell and Tube Exchangers

For each calculation performed a summary output is printed. An example of the preliminary shell and tube design output is shown in Table XII. The output consists of

1. Inlet and outlet temperatures
2. Mass flow rate of both shell and tube passes
3. Heat transferred
4. Specific heat of each fluid
5. Number of shell and tube passes

TABLE XII

PRELIMINARY CALCULATION OUTPUT

```
MAXISIM-HEXSIM
PAGE       4

EXAMPLE


SHELL & TUBE INLET AND OUTLET TEMPERATURES ,
DUTIES & RATES
         IN      OUT          Q              RATE
           DEG F            BTU/HR          LB/HR
  TUBE   80.00 115.00     1.6275E+07     465000.0
  SHELL 235.00 100.00    -1.6208E+07     180000.0

C SUB P TUBE SIDE FLUID   1.000 BTU/LB-R
C SUB P SHELL SIDE FLUID  0.667 BTU/LB-R



NUMBER OF TUBE PASSES    4
NUMBER OF SHELL PASSES   2
TUBE OD  0.7500 IN
TUBE ID  0.6800 IN
TUBE PITCH     0.9375 IN
TUBE METAL THERM COND  26.00000 BTU/HR-FT-R
TUBE ARRANGEMENT T
BUNDLE CONSTRUCTION, TEMA CLASS=S


SHELL SIDE H     350.00 BTU/FT2-HR-R
TUBE SIDE H     1000.00 BTU/FT2-HR-R

SHELL SIDE FOULING FACTOR 0.00500 (BTU/FT2-HR-R)**-1
TUBE SIDE FOULING FACTOR  0.00100 (BTU/FT2-HR-R)**-1

OVERALL U      97.10 BTU/FT2-HR-R
F SUB T    0.93019
LOG MEAN TEMP DIFFERENCE     55.81 DEG F

AREA PER SHELL PASS     1610.92 SQFT

EXCHANGER CONFIG. FOR   2 SHELLS WHICH ARE IDENTICAL
```

SHELL DIA IN	BUNDLE DOTL IN	SHELL LENGTH FT	TUBE LENGTH FT	TUBE COUNT	BUNDLE L/D
21.00	19.25	25.62	23.99	342.	13.71
23.25	21.50	21.32	19.53	420.	10.08
25.00	23.38	18.38	16.47	498.	7.91
27.00	25.38	15.48	13.45	610.	5.98
29.00	27.38	13.62	11.46	716.	4.74
31.00	29.38	12.28	9.98	822.	3.86
33.00	31.38	11.24	8.81	931.	3.20

6. Tube outside and inside diameter
7. Tube pitch
8. Thermal conductivity of tube metal
9. Tube arrangement
10. Tube bundle construction
11. Estimated individual heat transfer coefficients for both shell and tube side
12. Estimated fouling factors for shell and tube side
13. Overall heat transfer coefficients based on estimated individual coefficients and fouling factors
14. Log mean temperature difference and correction factor
15. Area per shell pass
16. Feasible inside shell diameters, diameter of the outer tube limit, shell length, tube length, tube count, and bundle length to diameter ratio

There are two forms for the Delaware calculation output. The difference between the two is that one includes detailed Delaware parameter output and the other does not. An example of output for each type is shown in Table XIII-XIV. The output without the detailed Delaware parameter output consists of:

1. Inlet and outlet temperatures
2. Mass flow rates for shell and tube fluids
3. Heat transferred
4. Fluid properties such as specific heat, density, bulk viscosity, viscosity at walls, thermal conductivity
5. Tube metal thermal conductivity

TABLE XIII

DELAWARE CALCULATION OUTPUT INCLUDING

DELAWARE PARAMETER OUTPUT

MAXISIM-HEXSIM
PAGE 1

EXAMPLE

DETAILED DELAWARE PARAMETER OUTPUT

NC	12.00	FC	0.65085
NCW	5.00	NB	13.00
SM	63.00	FSBP	0.28571
STB	21.38	THETA	2.09
SSB	3.30	SWG	67.71
SWT	26.38	SW	41.34
RESHL	75187.94	JSUBI	0.0049
HIDEAL	596.63	JSUBC	1.0250
JSUBL	0.6882	JSUBB	0.8972
JSUBR*	1.0000	JSUBR	1.0000
JSUBS	1.0000	HOSHL	377.62
FSUBI	0.1149	RSUBL	0.4437
RSUBB	0.7311	RSUBS	1.0000
DPBI	35.23	DPWI	48.64
DPS	6.8155		
RETUB	36308.29	FSUBIT	0.0052
HITUB	1097.37	DPT	4.7815
VELTUB	4.83		

TABLE XIII (CONTINUED)

MAXISIM-HEXSIM
PAGE 2

EXAMPLE

SHELL & TUBE INLET AND OUTLET TEMPERATURES,
DUTIES AND RATES

	IN	OUT	Q	RATE
	DEG F		BTU/HR	LB/HR
TUBE	80.00	109.84	1.3875E+07	465000.0
SHELL	235.00	119.43	-1.3875E+07	180000.0

SPECIFIED FLUID PROPERTIES

PROPERTY	UNITS	SHELL SIDE	TUBE SIDE
SPECIFIC HEAT ,	BTU/LB-R	0.6670	1.0000
DENSITY ,	LB/FT3	31.8000	62.0000
BULK VISCOSITY ,	LB/FT-HR	0.3420	1.6828
VISCOS AT WALL ,	LB/FT-HR	0.3440	1.6498
THERMAL CONDUCT,	BTU/HR-FT-R	0.0685	0.3640

TUBE METAL THERMAL CONDUCT 26.0000 BTU/HR-FT-R

HEAT EXCHANGER GEOMETRY

TUBE ARRANGEMENT	T	BUNDLE CONSTRUCTION TEMA CLASS=	S
NUMBER OF TUBES	342	NUMBER OF BAFFLES	13
TUBE LENGTH	14.00 FT	BAFFLE CUT	25.00 %
TUBE OD	0.7500 IN	BAFFLE TIP TO SHELL	5.25 IN
TUBE ID	0.6800 IN		
TUBE PITCH	0.9375 IN	BAFFLE SPACING(CENTER)	12.00 IN
NUMBER OF TUBE PASSES	4	(INLET)	12.00 IN
NUMBER OF SHELL PASSES	2	(OUTLET)	12.00 IN
NUMBER OF OPEN PASS		BAFFLE TO TUBE CLEARANCE	0.03215 IN
PARTITION LANES	0	BAFFLE TO SHELL CLEARANCE	0.15000 IN
WIDTH PASS PART LANES	0.00 IN	NUMBER OF SEALING STRIPS	2
DIA. OUTER TUBE LIMIT	19.50 IN	SHELL DIAMETER	21.00 IN

HEAT TRANSFER/PRESSURE DROP CALCULATION RESULTS

ITEM	UNITS	SHELL SIDE	TUBE SIDE
FILM COEFFICIENT,	BTU/FT2-HR-R	377.62	1097.37
FOULING FACTOR,	1/(BTU/FT2-HR-R)	0.0050	0.0010
VELOCITY,	FT/SEC	3.5939	4.8308
PRESSURE DROP(TOTAL),	PSI	6.816	4.782

OVERALL HEAT TRANSFER COEFFICIENT 100.09 BTU/FT2-HR-R
F SUB T 0.9934
LOG MEAN TEMPERATURE DIFFERENCE 74.22 DEG F

AREA/SHELL PASS 940.10 SQFT

TABLE XIV

DELAWARE CALCULATION OUTPUT WITHOUT DETAILED DELAWARE PARAMETER OUTPUT

```
MAXISIM-HEXSIM
PAGE       2

EXAMPLE


SHELL & TUBE INLET AND OUTLET TEMPERATURES,
DUTIES AND RATES
        IN      OUT          Q             RATE
          DEG F            BTU/HR          LB/HR
  TUBE    80.00 109.84    1.3875E+07     465000.0
  SHELL 235.00 119.43    -1.3875E+07     180000.0


SPECIFIED FLUID PROPERTIES

    PROPERTY            UNITS     SHELL  SIDE   TUBE  SIDE
    SPECIFIC HEAT   ,  BTU/LB-R       0.6670       1.0000
    DENSITY         ,  LB/FT3        31.8000      62.0000
    BULK VISCOSITY  ,  LB/FT-HR       0.3420       1.6828
    VISCOS AT WALL  ,  LB/FT-HR       0.3440       1.6498
    THERMAL CONDUCT,BTU/HR-FT-R       0.0685       0.3640


TUBE METAL THERMAL CONDUCT    26.0000 BTU/HR-FT-R


HEAT EXCHANGER GEOMETRY

    TUBE ARRANGEMENT          T       BUNDLE CONSTRUCTION TEMA CLASS= S
    NUMBER OF TUBES         342       NUMBER OF BAFFLES             13
    TUBE LENGTH           14.00 FT    BAFFLE CUT                 25.00 %
    TUBE OD              0.7500 IN    BAFFLE TIP TO SHELL         5.25 IN
    TUBE ID              0.6800 IN
    TUBE PITCH           0.9375 IN    BAFFLE SPACING(CENTER)     12.00 IN
    NUMBER OF TUBE PASSES    4                      (INLET )     12.00 IN
    NUMBER OF SHELL PASSES   2                      (OUTLET)     12.00 IN
    NUMBER OF OPEN PASS               BAFFLE TO TUBE CLEARANCE  0.03215 IN
       PARTITION LANES       0        BAFFLE TO SHELL CLEARANCE 0.15000 IN
    WIDTH PASS PART LANES  0.00 IN    NUMBER OF SEALING STRIPS      2
    DIA. OUTER TUBE LIMIT 19.50 IN    SHELL DIAMETER             21.00 IN


HEAT TRANSFER/PRESSURE DROP CALCULATION RESULTS

    ITEM                 UNITS       SHELL  SIDE TUBE  SIDE
    FILM COEFFICIENT, BTU/FT2-HR-R       377.62    1097.37
    FOULING FACTOR,1/(BTU/FT2-HR-R)      0.0050     0.0010
    VELOCITY,  FT/SEC                    3.5939     4.8308

    PRESSURE DROP(TOTAL), PSI             6.816      4.782

OVERALL HEAT TRANSFER COEFFICIENT    100.09 BTU/FT2-HR-R
F SUB T                              0.9934
LOG MEAN TEMPERATURE DIFFERENCE       74.22 DEG F

AREA/SHELL PASS                 940.10 SQFT
```

6. Heat exchanger geometery which includes tube arrangement, tube bundle construction, number of tubes, tube length, tube OD, tube ID, tube pitch, number of baffles, baffle cut, baffle spacing, baffle to tube clearance, baffle to shell clearance, number of sealing strips, number of tube passes, number of open pass partition lanes, diameter outer tube limit, shell diameter
7. Calculated individual heat transfer coefficients
8. Estimated fouling factors
9. Calculated velocity for both shell side and tube side
10. Calculated pressure drop for shell side and tube side
11. Overall heat transfer coefficient
12. Log mean temperature difference and correction factor
13. Area/shell pass

The key output results to check are

1. Heat balance and inputs
2. Exchanger configuration
3. The velocity
4. The pressure drop
5. Intermediate Delaware parameters

The rate of heat transfer from the cold to the hot fluid for all problems should agree to within one percent. If for some reason, the heat rates do not agree within one percent a warning comment is printed. The correction factor for the log mean temperature difference should be greater than 0.8. The reason is that below this value even a small failure in the basic assumption of this method can easily render the exchanger thermodynamically incapable of meeting the specified performance.* If the LMTD correction factor is below 0.8 a warning will be printed. Also, a temperature cross should be avoided since it is indicative of a relatively small temperature potential between the fluids. This requires a large area for heat exchange. For a counterflow heat exchanger a temperature cross occurs when the exit temperature of the cold fluid is higher than the exit temperature of the hot fluid. Oftentimes the cause of these problems is incorrectly specified inputs. The exchanger configuration should be checked to ensure that it meets design requirements (15). The following paragraphs give guidelines which should be followed when designing shell and tube exchangers.

The fluid velocity is set by the fluid flow rates and cross sectional area for flow. Liquid velocities are ordinarly kept between 2 and 15 ft/sec and gas velocities between 10 to 100 ft/sec (with each usually near the middle of the range given.) It is sometimes necessary to restrict the fluid velocity to avoid difficulties with such problems as erosion, tube vibration or noise (9).

The pressure drop through the exchanger should also be checked. For most heat exchanger applications, the pressure drop in each of the two fluid streams is limited to between 5 to 50 psi to avoid both excessive pumping power losses and excessive pressures in the shells and piping systems (9).

---

*Four of the primary assumptions for validity of the F-correction factor method are constant overall heat transfer coefficients, constant flow rate, constant specific heat of the two streams, and no bypass (13).

The detailed Delaware parameter output can also be used to check the validity of the exchanger design. The Delaware ideal tube bank data were obtained for a variety of geometries of industrial interest (3).

J_c is the correction factor for baffle cut and spacing. This factor is equal to 1.0 for an exchanger in which there are no tubes in the windows, increases to a value as high as 1.15 for a design value in which the windows are relatively small and the window velocity is very high, and decreases to a value of 0.52 for very large baffle cuts (5).

J_l is the correction factor for baffle leakage effects, including both shell-to-baffle and tube-to-baffle leakage. A typical value is in the range of 0.7 to 0.8 (5).

J_b is the correction factor for the bundle bypss flow. For the relatively small clearance between the outermost tubes and the shell for a fixed tube sheet construction J_b is approximately 0.9; whereas for the larger clearances required for the pull-through floating head construction J_b is approximately 0.7, but the value of J_b can be increased to 0.9 or better by the use of sealing strips (5).

J_s is the correction factor for variable baffle spacing in the inlet and outlet sections. J_s is usually between 0.85 and 1.0 (5).

If all checks for answer quality are satisfactory, you probably have a good solution. We strongly suggest you develop the habit of checking your results carefully.

B.1.1. Improving Your Results. The key to getting good results from any heat exchanger design routine is to supply it with good information about the system being considered. In this section guidelines or "rules of thumb" will be given for good exchanger design. The choice of which fluid to pass on the shell side and which on the tube side involves a number of factors and is generally a compromise amoung these requirements. According to Walker (15) the main considerations are:

1. Pressure. The wall thickness required (and hence the weight and cost of the material) to contain a given pressure increases directly with the diameter. Therefore high-pressure fluids should be contained in the tubes.

2. Temperature. High temperatures reduce the permissible stress levels of materials so that a greater wall thickness is required. Therefore high temperature fluid should be on the tube side. Also, if a high-temperature fluid is on the shell side additional insulation may be required to conserve energy or for safety.

3. Corrosiveness of the Fluid. Corrosive fluids require the use of special (and usually expensive) alloys or other materials. If only one fluid is corrosive passing it in the tubes will avoid the need for an expensive alloy shell.

4. Cleanliness of the Fluid. Processes with above-average requirements for cleanliness may require the use of special materials.

5. Hazardous or Expensive Fluids. Leakage of fluid is less likely from the tube side than from the shell side in most types of exchangers.

6. Pressure Drop. The pressure drop inside the tubes can be more accurately predicted than that in the shell. Where the fluid pressure drop is critical and must be accurately predicted, fluids should pass through the tubes.

7. Fluid Viscosity. To maximize heat transfer, both fluids should be in turbulent flow. Fluids of high viscosity may be laminar in

the tubes but turbulent in the shell depending on the tube side clearance. If the flow is laminar in both the shell and the tubes, the viscous fluid should in the tubes because more reliable heat-transfer and flow distribution predicitions can be made.

8. Mass Flow. In general it is better to put the fluid having the lower mass flow rate on shell side. Turbulent flow is obtained at lower Reynolds numbers on the shell side.

9. Cleaning. The tube outer surfaces are more difficult to clean than the tube internal surface. Therefore, the cleaner fluid should pass through the shell.

Frank (8) has given some useful "rules of thumb" about temperature differences and temperature approaches (where a temperature approach is the minimum temperature difference between fluids).

1. The temperature approach should normally be at least 41°F to 45°F for a refrigeration systems.

2. In recuperative heat exchangers*, a temperature approach of at least 68°F should be maintained.

3. Cooling water rise may be 50 to 68°F at high mean temperature differences and less for low mean temperature differences.

4. Cooling water outlet temperatures should be below 130°F.

Oftentimes more than one shell is needed in order to perform the required service. There is a rapid graphical technique for estimation of the the number of shells in series for a given heat duty (1,5). The procedure is as follows:

---

*Assumed to be exhaust gas/air preheater.

1. The terminal temperatures of the two streams are plotted on the ordinates of ordinary arithmetic graph paper with the hot fluid inlet temperature and the cold fluid outlet temperature on the left hand side and the hot fluid outlet and the cold fluid inlet on the right hand ordinate. The distance between the two is arbitary.

2. If the specific heat of each stream is constant, straight lines are drawn from the inlet to the outlet temperature point for each stream.

3. Starting with the cold fluid outlet temperature a horizontal line is laid off until it reaches the hot fluid line. From that point a vertical line is dropped to the cold fluid line. This operation is continued until a vertical line intercepts the cold fluid operating line at or below the cold fluid inlet temperature.

4. The number of horizontal lines is equal to the number of shells in series that is required to perform the duty. This technique is shown in Figure 6.

Another consideration is tube diameter, tube length, layout pattern, and pitch. Thermohydraulic considerations favor small tube diameter. Also, greater surface density within a given shell is possible with small diameter tubes. In general, the longer the tube the lower the cost of the resulting smaller shell diameter, thinner tubesheets and flanges, fewer pieces to handle, and fewer holes to drill. The limiting factors are accomodating shell-side flow area with reasonable baffle spacing and practical design considerations. Usual length-to-shell diameter ratios range from 5 to 10 for best performance. A good practice for tube layout calls for a minimum pitch of 1.25 times the tube diameter. Generally the smallest pitch in a

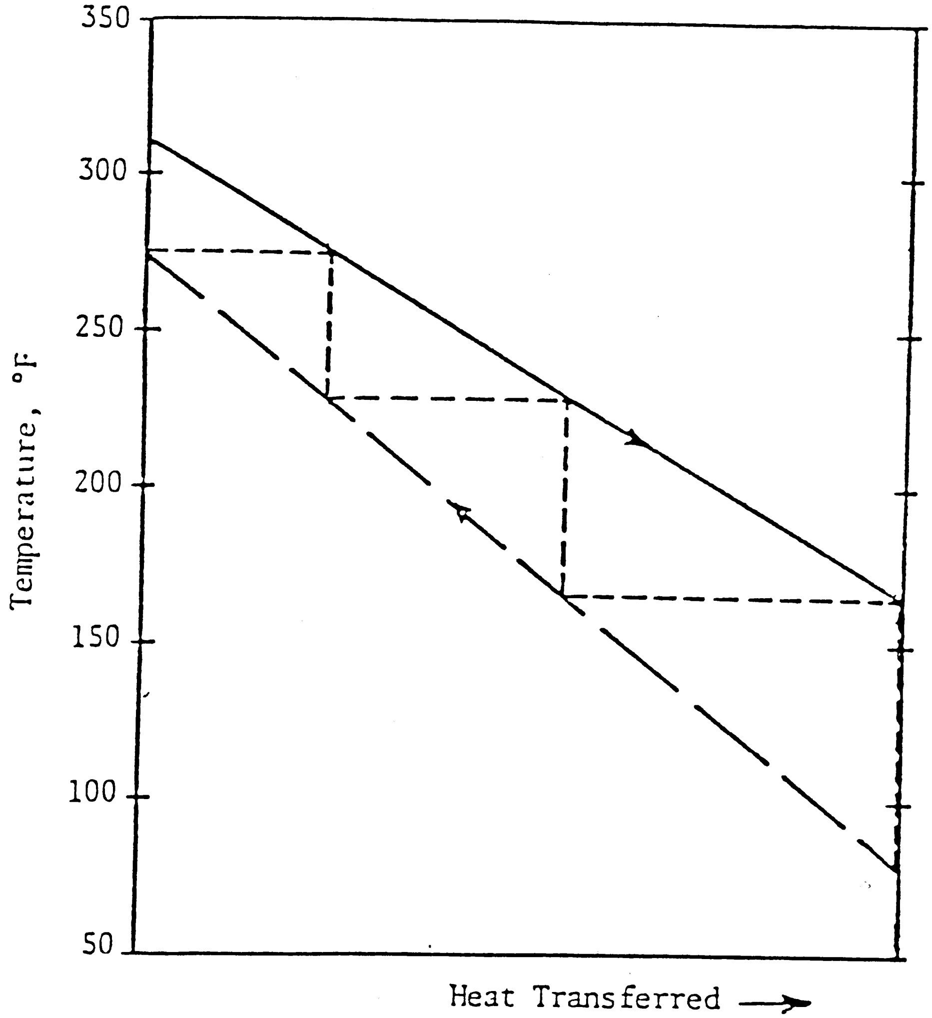


Figure 6. Estimation of Required Number of Shells in Series (5)

triangular layout is preferred for flow in clean service and square layout for cases where mechanical cleaning is required (13).

Tube bundle configuration is usually determined by thermal expansion considerations and cleaning considerations. Each tube bundle configuration has advantages and disadvantages. The following paragraphs give a short summary of the exchanger advantages and disadvantages.

The fixed tube sheet exchanger has no provision for accommodating differential expansion, and therefore is very vulnerable to thermal stress damage. Its advantages are that individual tubes are easily replaced and it can be mechanically cleaned (5). It is also cheaper to build in a given shell diameter, and the chances for bypass are minimized.

The U-tube configuration allows independent expansion of tubes and shell. The major disadvantages are that individual tubes cannot be replaced (except in the outer tube row), the tube side cannot be mechanically cleaned, and erosion can occur inside the tubes U-bend (5).

The simplest floating head design is the "pull-through bundle" type. It does allow for thermal stress but many tubes must be omitted from the edge of the full bundle to allow for the bonnet flange and bolt circle. Another floating head design that partially offsets the above disadvantage is the "split-backing ring" floating head type. In this type of exchanger the floating head bonnet is bolted to a split backing ring rather than the tube sheet. Therefore, no bolt circle has to be provided on the tube sheet. However, this makes the exchanger more complicated mechanically. The "outside packed floating head" is less postively sealed against leakage to the atmosphere than the previous types but have the advantage of allowing single tube pass construction (5).

For preliminary design purposes, there is a need for experience based fouling factors and individual heat transfer coefficients. An extensive listing is contained in the Heat Exchanger Design Handbook (4). This listing is shown in Table XV. There is a large uncertainity in these values but they are sufficient in most cases for estimation purposes.

In this section a discussion of modifications which might be necessary to improve exchanger performance are discussed. The suggestions contained in the following paragraphs are intended to be a guideline to help the inexperienced designer to correct some common problems encountered in shell and tube exchanger design.

If the heat exchanger is limited by the amount of heat that it can transfer, you should either increase the area of the exchanger or increase the heat transfer coefficient. To increase the tube side heat transfer coefficient the number of tube passes can be increased therefore increasing the log mean temperature difference or you can add multiple shells in series (5).

If the exchanger is limited by pressure drop, you can decrease the number of tube passes or increase the tube diameter. Also, you can decrease the tube length and increase the shell diameter and number of tubes. If the exchanger is limited by shell side pressure drop. You can increase the baffle cut, increase the baffle spacing, or increase the tube pitch (5).

TABLE XV

TYPICAL FILM HEAT TRANSFER COEFFICIENTS FOR SHELL AND TUBE EXCHANGERS (4)

FLUID CONDITIONS		H,(BTU/H-FT2)[a,b]	FOULING FACTOR (HR-FT2-F/BTU)[a]
SENSIBLE HEAT TRANSFER			
WATER[c]	LIQUID	880-1320	0.00056-0.0014
AMMONIA	LIQUID	1056-1408	0.0 - 0.00056
LIGHT ORGANICS[d]	LIQUID	264-352	0.00056-0.0011
MEDIUM ORGANICS[e]	LIQUID	132-264	0.00085-0.00227
HEAVY ORGANICS[f]	LIQUID		
	HEATING,	44-132	0.0011-0.0056
	COOLING,	26-70	0.0011-0.0056
VERY HEAVY ORGANICS[g]	LIQUID,		
	HEATING,	18-53	0.0023-0.017
	COOLING,	11-26	0.0023-0.017
GAS[h]	14.5-29 PSIA	14-22	0.0-0.00056
GAS[h]	145 PSIA	44-70	0.0-0.00056
GAS[h]	1450 PSIA	88-141	0.0-0.00056

a. Heat transfer coefficients and fouling resistances are based on area in contact with fluid. Ranges shown are typical, not all encompassing. Temperatures are assumed to be in normal processing range; allowances should be made for very high or low temperatures.

b. Allowable pressure drops on each side are assumed to be about 7- 20 psia except for (1) low-pressure gas where the pressure drop is assumed to be about 5% of the absolute pressure; and (2) very viscous organics, where the allowable pressure drop is assumed to be about 21 to 45 psia.

TABLE XV (FOOTNOTES CONTINUED)

c. Aqueous solutions give appoximately the same coefficients as water.

d. "Light organics" include fluids with liquid viscosities less than about .5 cP, such as hydrocarbons through C8, gasoline, light alcohols and ketones, etc.

e. "Medium organics" include fluids with liquid viscosities between about .5 cP and 2.5 cP, such as kerosene, straw oil, hot gas oil, absorber oil, and light crudes.

f. "Heavy organics" include fluids with liquid viscosities greater than 2.5 cP, but not more than 50 cP, such as cold gas oil, lube oils, and heavy and reduced crudes.

g. "Very heavy organics" include tars, asphalts, polymer melts, greases, etc., having liquid viscosities greater than about 50 cP. Estimation of coefficients for these materials is very uncertain and depends strongly on the temperature difference, because natural convection is often a significant contribution to heat transfer in heating, whereas congelation on the surface and particularly between fins can occur in cooling. Since many of these materials are thermally unstable, high surface temperatures can lead to extremely severe fouling.

h. Values given for gases apply to such substances as air, nitrogen, carbon dioxide, light hydrocarbon mixtures (no condensates), etc. Because of the very high thermal conductivities and specific heats of hydrogen and helium, gas mixtures containing appreciable fractions of these components will generally have substantially higher heat transfer coefficients.

B.2. Air-Cooled Heat Exchangers

For each calculation performed a summary output is printed. An example of the output for an air-cooled heat exchanger is shown in Table XVI. If the tube length is not specified widths are calculated for standard tube lengths. Then a request for the desired tube length is issued. This is shown in Table XVII. The air-cooled heat exchanger output consists of the following.

1. Inlet and outlet temperatures
2. Mass flow rate of air and tube side fluids
3. Heat transferred
4. Air and tube side specific heat, density, bulk viscosity, viscosity at the wall, thermal conductivity
5. Tube metal thermal conductivity and fin metal thermal conductivity
6. Heat exchanger and fan geometery which includes tube arrangement, number of tubes, tube length, tube OD, tube ID, tube pitch, number of tube rows, number of tube passes, number of fins per unit length, fin thickness, fin spacing, bay width, type of draft, number of fans, fan efficiency, fan diameter, power per fan, air volume per fan, outlet air temperature, outlet air pressure, air velocity at fan face, air velocity at tube face
7. Calculated individual heat transfer coefficients for air side and tube side
8. Estimated fouling factors for air side and tube side
9. Calculated fin resistance
10. Pressure drop both air side and tube side

TABLE XVI

AIR-COOLED HEAT EXCHANGER OUTPUT

MAXISIM-HEXSIM
PAGE 3

EXAMPLE

AERIAL COOLER INLET AND OUTLET TEMPERATURES AND DUTIES & RATES

	IN	OUT	Q	RATE
	DEG F		BTU/HR	LB/HR
TUBE	170.00	124.00	-1.0645E+06	38000.0
AIR	92.00	107.42	1.0645E+06	285772.6

CALCULATED & SPECIFIED FLUID PROPERTIES

PROPERTY	UNITS	AIR	TUBE SIDE
SPECIFIC HEAT ,	BTU/LB-R	0.2416	0.6090
DENSITY ,	LB/FT3	0.0719	2.5000
BULK VISCOSITY ,	LB/FT-HR	0.0458	0.0210
VISCOS AT WALL ,	LB/FT-HR	0.0000	0.0200
THERMAL CONDUCT ,	BTU/HR-FT-R	0.0156	0.0130

TUBE METAL THERMAL CONDUCT 26.0000 BTU/HR-FT-R
FIN METAL THERMAL CONDUCT 119.0000 BTU/HR-FT-R

HEAT EXCHANGER & FAN GEOMETRY

TUBE ARRANGEMENT	T	TYPE OF DRAFT	F
NUMBER OF TUBES	220	NUMBER OF FANS	1
TUBE LENGTH	10.00 FT	FAN EFFIC	70.00 %
TUBE OD	1.0000 IN	FAN DIAMETER	7.50 FT
TUBE ID	0.9300 IN	POWER/FAN	7.57 HP
TUBE PITCH	2.2500 IN		
NUMBER OF TUBE ROWS	4		
NUMBER OF TUBE PASSES	2	AIR VOLUME/FAN	66234.37 CUFT/MIN
FINS/IN	11.00	(TEMP)	92.00 DEG F
FIN THICKNESS	0.0150 IN	(PRESS)	14.69 PSI
FIN SPACING	2.2500 IN	AIR VELOCITY AT FAN FACE	24.99 FT/SEC
BAY WIDTH	10.33 FT	AIR VELOCITY AT TUBE FACE	641.21 FT/MIN

HEAT TRANSFER/PRESSURE DROP CALCULATION RESULTS

ITEM	UNITS	AIR	TUBE SIDE
FILM COEFFICIENT,	BTU/FT2-HR-R	10.36	85.47
FOULING FACTOR,	1/(BTU/FT2-HR-R)	0.0010	0.0050
FIN RESISTANCE,	1/(BTU/FT2-HR-R)	0.0174	
PRESSURE DROP(TOTAL),	PSI	0.018	0.172
	IN H2O	0.507	

OVERALL HEAT TRANSFER COEFFICIENT 1.80 BTU/FT2-HR-R
F SUB T 0.9607
LOG MEAN TEMPERATURE DIFFERENCE 45.60 DEG F

EXTENDED AREA OF COOLER 13479.28 SQFT

TABLE XVII

AIR-COOLED HEAT EXCHANGER TUBE LENGTH

CALCULATION OUTPUT

STANDARD TUBE LENGTH FT	UNIT WIDTH FT
8.00	5.73
10.00	4.58
12.00	3.82
20.00	2.29
24.00	1.91
30.00	1.53

ENTER DESIRED TUBE LENGTH,FT?
10.

11. Calculated overall heat transfer coefficient
12. Calculated log mean temperature difference and correction factor
13. Overall area of air-cooled heat exchanger

The key output results to check are

1. Heat balance and inputs
2. Velocities
3. Pressure drop

The rate of heat transfer from the tube side fluid to the air for all problems should agree to within one percent. The heat must be transferred from the tube side fluid to the air otherwise a warning comment will be printed. The correction factor for the log mean temperature difference should be greater than 0.8. Below this value the exchanger may be thermodynamically incapable of meeting the performance. If the LMTD correction factor is below 0.8 a warning will be printed. Oftentimes the cause of these problems is incorrectly specified inputs. The exchanger configuration should be checked to ensure that it meets design requirements.

The fluid velocity is set by the fluid flow rate and cross sectional area for flow. Tube side velocities are ordinarily kept between 2 and 15 ft/sec for liquids and gas velocities between 10 to 100 ft/sec (with each usually near the middle of the range given). It is sometimes necessary to restrict the fluid velocity to avoid difficulties with such problems as erosion (9). Air side face velocities (that is the approach velocity of the air to the face of the heat exchanger, assuming uniform velocity) is about 300 to 900 ft/min. These air velocities are limited by the frictional pressure drop across the tube bank (5).

The pressure drop both on the air side and the tube side is another point which should be checked. The pressure drop in the tubes is limited to between 5 to 50 psi to avoid both excessive pumping power losses and excessive pressure loss (9). Most air cooled heat exchangers now in service are designed for fan static pressure of 0.45 to 0.75 in H2O (0.01 to 0.04 psi) (6). A typical design value is on the order of 1/2 in H2O frictional pressure drop across the tube bank (5).

B.2.1. Improving Your Results. Another important choice in the design of air-coolers is the selection of the type of draft used forced or induced. Each design has relative good and bad points, but often cost is the largest influencing factor.

Forced draft usually has a power advantage, especially if the temperature rise of the air is comparatively high. Forced-draft design permits a more covenient and economical mounting arrangement where a number of bundles and services are to be combined in a single unit.

Induced-draft design provides a more even distribution of air across the bundles, and for a given bundle elevation, affords more space for location of additional plant equipment beneath the unit. This design is more adaptable to suspension of the mechanical equipment from the unit itself, therefore, making more suitable mounting the unit above a pipe rack or above shell and tube exchangers.

Induced-draft units are much less likely to recirculate hot exhaust air, since the exit air velocity is from two to three times that of a forced-draft unit. This fact becomes increasingly important in the case of a large heat exchanger installation. In most cases the advantages of the induced-draft design outweigh disadvantages, but the problem should be studied for each case (12).

The air side fouling factor usually is negligible depending on the cleanliness of the atmosphere. In most cases the fouling factor is approximately 0.00045 (6).

C. A Comparison Of Air-Cooled And Shell And Tube Exchangers

Chemical process plants as well as steam power plants require the rejection of large quantities of heat. Water-cooled exchangers are often used but air-cooled units can also be used in this service. Air-cooled exchangers often compare favorably with water-cooling for many services in heat rejection service. Table XVIII contains a list of air-cooler advantages and disadvantages as compared to a water-cooled exchanger. Table XVIII is provided as a guide to help the user select the type of exchanger for the problem at hand. Depending upon the service, not all of the points will apply, while others may be of minor importance (6).

D. Diagonostic Comments

HEXSIM contains only a few diagnostic comments which pertain to the feasibility of the heat exchanger design. This section gives the diagnostic comments in HEXSIM, the possible conditions causing the comment, and some guidelines for eliminating the problem.

The presence or absence of a diagnostic comment is not always a postive indication of the quality of the results. Careful checking of

TABLE XVIII

A COMPARISON OF AIR-COOLED AND SHELL AND TUBE EXCHANGERS (6)

ADVANTAGES

1. Air is always available. Its use saves water for more critical service. Even with cooling tower recycle systems, up to 3% or more of the total recycle stream is required for makeup.

2. Heating of surface water is eliminated when it would be the coolant source for a once-through system.

3. Air eliminates the cost and space required for water facilities, pumps, treating equipment, cooling towers or filters, blowdown disposal area, and distribution and disposal lines.

4. Plant-site location is not as restricted as with water cooling.

5. Choice of exchanger materials is simpler and cheaper because, unlike water, air is seldom corrosive.

6. Tube-wall temperature is limited only by materials of construction, although special fin-to-tube joining is required for high process and air temperatures. Water outlet temperatures usually have to be held to 120 to 130°F maximum to prevent excessive scale formation.

7. Air side fouling is negligible. This is in contrast to water side formation of hard, scaly corrosion and depostion of dirt.

8. Less shutdown is required because fin-side fouling is negligible.

9. Operating costs are lower and can be further reduced with designs that use lower fan horsepower (at higher initial cost) and controlled fan operation to take advantage of the fact that air is predominately well below its design temperature.

10. Capacity can usually be easily changed by changing a motor horsepower or fan, or both.

11. Operating and maintenance labor requirements are on average about one-quarter of the costs for water coolers. The maintenance requirements consist mainly of periodic, routine checking and lubricating; regular checking of fan blade pitch; and cleaning of fins once a year.

TABLE XVIII (CONTINUED)

ADVANTAGES

12. Cleaning is simpler and cheaper and can be done with inexpensive equipment because the process fluid is inside the tubes.

13. Bundle pulling is not required. Therefore longer tube lengths are practical.

14. Possible contamination of process streams by water is eliminated.

15. Leaks from an air cooler will not remain undetected for long.

16. Capacity can be increased to a much greater extent than is possible with other exchangers by increasing air flow. Normally this will require minor equipment cost. However operating cost would be higher.

DISADVANTAGES

1. Air-cooled heat exchangers generally have a much higher initial cost - two to four times is common - when comparing only the exchangers themselves. Even though substantial initial savings have been made on some projects by using air-coolers, it is necessary to include operating costs and a reasonable payout period to justify air cooling.

2. Air-coolers require more space when comparing just the exchangers. However, when tower space requirements are considered, the area requirements are equal. Also, air coolers are often placed in unused spaces such as over pipe racks.

3. Location of air coolers is more of a problem because size. This also sometimes increases the process piping. Air coolers should not be located near large obstructions, especially not down wind of prevailing breezes. This is because down drafts can cause the warm air to recirculate.

4. Design inlet temperatures are typically up to 20° Fahrenheit higher for air than water.

5. Water temperatures from towers or rivers, etc., are relatively slow to change. Air fluctuates widely and more rapidly. This helps the power consumption but complicates control. Rain showers can rapidly drop process temperatures while radiant heat from the sun must also be considered.

TABLE XVIII (CONTINUED)

DISADVANTAGES
6. Hail screens are recommended in areas where hail occurs frequently because unprotected fins can be bent down in hail storms.
7. Atmospheric corrosion is a problem at some plants, and fins cannot be easily protected from this kind of damage.
8. Winter air temperatures in most of the U.S. are low enough to present a potential danger of freezing some process fluids unless the control system is able to prevent it.
9. Performance tests are difficult to make at times because of the changes in the weather.
10. Large numbers of air-cooled exchangers in a single plant may cause local increases in plant air temperatures.
11. The air-cooler industry uses 1-inch OD tubes as the standard. Other sizes are not common with all manufacturers because of the engineering and tooling expense involved.
12. The fans of air coolers are quite noisy.
13. Fire or other hazards are sometimes considered to be more serious with air coolers because of possible leaks and the fanning of fires that might result.
14. Tube storage and stocking are more complicated with finned tubes.

the results is always in order; results with diagnostic comments may be in some cases adequate.

The following diagnostic comments may be printed by HEXSIM.

D.1. Shell and Tube Diagnostic Comments

INCORRECT TUBE ARRANGEMENT

This comment will appear only if you specified something other than triangular, inline square, or rotated square. The following question will be repeated.

"ENTER TUBE ARRANGEMENT;T-TRIANGULAR,S-INLINE SQUARE,R-ROTATED SQUARE ?"

INCORRECT BUNDLE CONSTRUCTION SPECIFIED

This comment will appear only if you specified a tube bundle construction other than split backing ring, outside packed floating head, U-tube, pull thru floating head, or fixed tube sheet. The following question will be repeated.

"ENTER TYPE OF TUBE BUNDLE CONSTRUCTION:
S-SPLIT BACKING RING,P-OUTSIDE PACKED FLOAT HEAD,
U-U TUBE,T-PULL THRU FLOAT HEAD,F-FIXED TUBE SHEET?"

U TUBE BUNDLES REQUIRE AN EVEN NUMBER OF TUBE PASSES.

For a U tube bundle only an even number of tube passes may be specified. If an odd number is specified this warning statement is printed. The simulator will then request for a bundle specification.

THE MAX (TUBE OR SHELL) SIDE PRESSURE DROP HAS BEEN EXCEEDED
THE MAX PRESSURE DROP IS --- PSI
THE CALC PRESSURE DROP IS --- PSI
ENTER NEXT COMMAND?

This statement is printed when either the shell side or tube side pressure drop is exceeded. HEXSIM then returns the user to the EDIT mode. For possible solutions to this problem see page 163.

THE TUBE SIDE VELOCITY IS LOW, V = -- FT/SEC

This comment is printed when the tube side velocity is below 5 ft/sec for a liquid and 10 ft/sec for a gas. HEXSIM then returns the user to the EDIT mode.

THE TUBE SIDE VELOCITY IS HIGH, V = -- FT/SEC

This comment is printed when the tube side velocity is above 15 ft/sec for a liquid and 100 ft/sec for a gas. HEXSIM then returns to the EDIT mode.

FT = WHICH IS LESS THAN RECOMMENED MINIMUM VALUE
DO YOU WANT TO CONSIDER MULTIPLE SHELLS IN SERIES;YES OR NO?

This means that the value of the correction factor for the log mean temperature difference is below 0.8 for the problem being calculated. When the value for the correction factor is below 0.8, it means that even a small failure in the assumptions of this method render the exchanger thermodynamically incapable of meeting the specified performance. The simplist way to correct this problem is to use multiple shells in series in order to raise the correction factor to 0.8 or greater.

HEXSIM will automatically calculate the number of shells in series in order to raise the correction factor to above 0.8. HEXSIM will print the dialogue shown in Table XIX.

BOTH OUTLET TEMPS ARE ZERO, CAN'T SOLVE FOR Q REENTER DATA; YES OR NO?

This means both the shell side and tube side outlet temperatures are zero.

ERROR IN HEAT BALANCE > 1%

This comment is printed when the heat balance does not agree to within 1% or less. After this comment the following information is

TABLE XIX

DIALOGUE FOR CORRECTION OF LOW LOG MEAN TEMPERATURE DIFFERENCE CORRECTION FACTOR

----SHELLS IN SERIES WILL GIVE A PROCESS WITH NO TEMPERATURE CROSSES

DO YOU WANT TO USE THIS NUMBER OF SHELLS; YES OR NO?

ENTER NUMBER OF SHELLS YOU WANT TO USE?

printed; the inlet and outlet temperatures, mass flow rates for each stream, and the heat transfered by each stream. HEXSIM will then ask

"DO YOU WANT TO CHANGE ANYTHING;YES OR NO?"

% BAFFLE CUT TOO SMALL

This comment is printed when the percent baffle cut is less than nine percent. This is a limit imposed due to mechanical constraints. It is recommended that you enter a larger value.

% BAFFLE CUT TOO LARGE

This comment is printed when the percent baffle cut is larger than forty-nine percent. The baffle cut cannot be any larger due to physical constraints.

BAFFLE SPACING TOO CLOSE

This means that the baffle spacing is smaller than TEMA class R specifications allow. The minimum baffle spacing allowed is one fifth of the shell diameter (14).

BAFFLE SPACING TOO LARGE

This comment indicates that the baffle spacing is further apart than is allowed by TEMA class R specifications. The maximum allowable baffle spacing is 52 inches. This value is the most limiting case (14).

SHELL AND TUBE INPUTS ARE NOT CONSISTENT

This comment is printed when the shell inlet temperature is equal to the tube outlet temperature and/or when the shell outlet temperature is greater than or equal to the tube inlet temperature. This is indicative of a temperature cross. The outlet temperatures need to be adjusted.

CANNOT GET SOLUTION FOR YOUR SPECIFIC FLOWS & TEMPS

CHECK OUTPUT & RERUN WITH NEW PARAMETERS

This is printed when no acceptable length for the problem was found as orginally specified. You should review your problem specifications.

CANNOT FIND SHELL DIAMETER TO MEET SPECIFIED VALUE

This message is printed when the diameter calculated for the specified inputs is not in the range of 8 to 120 inches. One way to solve this problem is to go to multiple shells in series. HEXSIM will then enter the EDIT mode.

NUMBER OF TUBE PASSES OTHER THAN 1, 2, 4, 6, OR 8 PER SHELL

This message is printed when the number of tube passes entered is other than 1, 2, 4, 6, or 8. HEXSIM will proceed to the EDIT mode.

PITCH TO DIAMETER RATIO OTHER THAN 1.25 OR 1.333

This message is printed when the pitch to diameter ratio is other than 1.25 or 1.333. This is usually the best range for good design.

D.2. Air-Cooled Heat Exchanger

Diagnostic Comments

FT= -----WHICH IS LESS THAN THE RECOMMENDED MINIMUM VALUE

This statement means that the value of the correction factor for the log mean temperature difference is below 0.8 for the problem being calculated. When the value for the correction factor is below 0.8 the exchanger design may be thermodynamically incapable of meeting the specified performance. The simplest way to correct this problem is the use multiple tube passes in order to raise the correction factor to 0.8 or greater. HEXSIM will ask the following question.

DO YOU WANT THE NUMBER OF TUBE PASSES INCREASED; YES OR NO?

Respond with a Y or N. If you respond with a Y the number of tube passes will be increased to 4 and the calculations will continue.

CHECK YOUR INPUT DATA; DUTY HAS WRONG SIGN
CHANGE SOMETHING AND RERUN

This comment is printed when the air is being cooled in other words it is being used to heat the process stream not cool it. The most probable cause of this comment is an error in the inlet and/or outlet tube temperatures.

CANNOT FIND ACCEPTABLE FACE VELOCITY/HEAT TRANSFER FACE AREA COMBINATION ****WILL USE 8 TUBE ROWS AND CONTINUE

This comment is printed when the air face velocity is less than 500 ft/min or greater than 900 ft/min. The maximum design pressure drop obtainable from a typical fan used for air-cooled heat exchangers is on the order of 1 in H20 or 0.04 psi. A commonly used design value is on the order of 1/2 in H20 or 0.02 psi. This pressure drop corresponds to a set of values of design face velocities and number of rows of tubes as given in Table XX. This comment will only be printed if the face area was not specified (5).

The values in this table are only approximate but if the face velocity is significantly above the values shown the result will be that the allowable pressure drop developed by the fan will be exceeded. While a lower velocity will result in an excessively large heat exchanger face area.

TABLE XX

TYPICAL COMBINATIONS OF NUMBER OF TUBE ROWS AND DESIGN FACE VELOCITIES IN AIR-COOLED HEAT EXCHANGERS (5)

NUMBER OF TUBE ROWS	FACE VELOCITY,FT/MIN
3	900
4	800
5	700
6	600
8	500
10	400
12	300

CHAPTER IV

PROGRAM LIMITATIONS AND ACCURACY

A. Overview

In this chapter, the known program limitations and some estimates of the accuracy of the methods used are given. Recognize that is completely impossible to state definitive program limitations or accuracies; these are too dependent on how you, the user, characterize the system, i.e. the accuracy of the information you give the program. Thus, the comments that follow should be regarded as guidelines.

B. LMTD, Ft Method

The logarithmic mean temperature difference method of heat exchanger design is very precise when the assumptions are completely valid. The set of assumptions for this method are (5);

1. All elements of a given stream have the same thermal history.
2. The heat exchanger is at steady state.
3. Each stream has a constant specific heat.
4. The overall heat transfer coefficient is constant.
5. The flow is either entirely cocurrent or entirely countercurrent.
6. No bypass flow
7. The heat exchanger does not exchange heat with the surroundings.

C. Tube Side Methods

For turbulent flow the individual heat transfer equation used is the Seider-Tate equation. This equation gave a maximum mean deviation of approximately +15 to -10 percent for Reynolds numbers above 10,000. The Hausen equation for laminar flow is very accurate as long as the Reynolds number is below 2,100.

D. Delaware Method

The Delaware method, though probably the best in the open literature is not extremely accurate. An exhaustive study by Palen and Taborek (11) tested various literature methods against 972 heat transfer data points and 1332 pressure drop points covering a wide range of fluids and geometrical parameters. This study showed that the shell-side heat transfer coefficients ranged from about 50% low to 100% high, while the pressure drop range was from 50% low to 200% high. The mean error for heat transfer was about 15% low for all Reynolds numbers, while the mean error for pressure drop was from about 5% low (unsafe) at Reynolds numbers above 1000 to about 100% high at Reynolds number below 10.

E. Air-Cooler Methods

The methods used to calculate the exchanger width, number of tubes, number and power requirements of the fans is only an approximate method and is based on the LMTD, FT method of design. The average accuracy of the methods given in the GPSA Data Book is for most problems 10 to 15 percent when compared with the best design procedures as suggested by engineers at the Gas Processors Suppliers Assocation.

BIBLIOGRAPHY

1. Bell, K. J., "Estimate S & T Exchanger Design Fast", Oil and Gas Journal, Dec. 4, 1978, pp. 59-68.

2. Bell, K. J.,"Exchanger Design Based on The Delaware Research Program", Petro/Chem Engineer, 32, Oct. 1969, pp.C-26-C-40c.

3. Bell, K. J., "Final Report of the Cooperative Research Program on Shell and Tube Heat Exchangers", Bullentin No. 5, University of Delaware Engineering Experiment Station, Newark, Delaware, 1963.

4. Bell, K. J., in Schlunder, E. U., ed., Heat Exchanger Design Handbook, Vol. 3, Sec. 3.1.4, Hemisphere Publ. Corp., Washington, 1983.

5. Bell, K. J., Process Heat Transfer, class notes at Oklahoma State University, Unpublished, 1984.

6. Cook, E. M., "Air-Cooled Heat Exchangers", Chemical Engineering, May 25, 1964, p.137; July 6, 1964, p. 131; August 3, 1964, p 97.

7. Engineering Data Book, Gas Processors Suppliers Association, Tulsa, 1972.

8. Frank, O., "Simplified Design Procedures for Tubular Heat Exchangers", Practical Aspects of Heat Transfer, American Institute of Chemical Engineers, New York, 1978, pp. 1-25.

9. Fraas, A. P., and Ozisik, M. N., Heat Exchanger Design, John Wiley and Sons Inc., New York, 1965.

10. Kern, D. Q., Process Heat Transfer, McGraw-Hill Book Co., New York, 1948.

11. Palen, J. W., and Taborek, J., "Solution of Shell-Side Flow Pressure Drop and Heat Transfer by Stream Analysis Method", CEP Symp. Ser. No. 92, 65, "Heat Transfer-Philadelphia", 1969, pp. 53-63.

12. Smith, E. C., "Air-Cooled Heat Exchangers", Chemical Engineering, November 17, 1958.

13. Taborek, J., in Schlunder, E. U., ed., Heat Exchanger Design Handbook, Vol. 3, Sec. 3.3.4, Hemisphere Publ. Corp., Washington, 1983.

14. Tubular Exchanger Manufacturers Association, Standard, 6th Ed., New York, 1978.

15. Walker, G., Industrial Heat Exchangers, McGraw-Hill International Book Co., New York, 1982.

APPENDIX B

CURVE FITS

APPENDIX B

CURVE FITS

Fit for Figure 1: F_t vs. P for an exchanger having one pass shell side, two passes tube side:

$$P = (t_2 - t_1)/(T_1 - t_1)$$

$$R = (T_1 - T_2)/(t_2 - t_1)$$

$$F_t = \frac{\frac{\sqrt{R^2 + 1}}{R - 1} \log_{10} \frac{1 - P}{1 - PR}}{\log_{10} \frac{(2/P) - 1 - R + \sqrt{R^2 + 1}}{(2/P) - 1 - R - \sqrt{R^2 + 1}}}$$

If R = 1, then $1/(R - 1) \log_{10} (1 - P)/(1 - PR)$ becomes indeterminate, the expression is then replaced with

$$\frac{P}{1.3\ (1 - P)}$$

<u>One Pass Shell Side; Infinite Passes Tube Side</u>. It has been pointed out by Bowman (5) that the correction factor F for a multipass heat exchanger, having one shell-side pass and a large number of tube side passes approaches ideal crossflow. Even at this limit the value

for F_t is generally only 1 to 2 percent less than that of the one-two exchangers.

For multiple shell side passes the following equations are used:

$$P_{1,2} = (t_2 - t_1)/(T_1 - t_1)$$

$$R = (T_1 - T_2)/(t_2 - t_1)$$

Then P is recalculated

$$P_{N,2N} = \left(\frac{1 - \left(\frac{1 - P_{1,2} R}{1 - P_{1,2}}\right)^{1/N}}{R - \left(\frac{1 - P_{1,2} R}{1 - P_{1,2}}\right)^{1/N}}\right)$$

when R = 1

$$P_{N,N2} = \frac{P_{1,2} * N}{P_{1,2N} - P_{1,2} + 1}$$

and the term 1/R - 1 $\log_{10}$ 1 - P/1 + PR is set as (1.0 - P)/P

where

N = number of shell side passes

$$F = \frac{\frac{\sqrt{R^2 + 1}}{R - 1} \log_{10} \frac{1 - P}{1 - P}}{\log_{10} \frac{(2/P) - 1 - R + \sqrt{R^2 + 1}}{(2/P) - 1 - R + \sqrt{R^2 + 1}}}$$

Fit for Figure 6: j_i vs. Re_s

Curve 1: for triangular layouts

$$j_i = 1.73\ Re_s^{-0.694} \qquad 1.0 \leqslant Re_s \leqslant 100$$

$$j_i = 0.717\ Re_s^{-0.507} \qquad 100 \leqslant Re_s \leqslant 1000$$

$$j_i = 0.236\ Re_s^{-0.346} \qquad 1000 \leqslant Re_s$$

Curve 2: for rotated square layouts

$$j_i = 1.39\ Re_s^{-0.691} \qquad 1.0 \leqslant Re_s \leqslant 100$$

$$j_i = 0.414\ Re_s^{-0.425} \qquad 100 \leqslant Re_s \leqslant 1000$$

$$j_i = 0.257\ Re_s^{-0.357} \qquad 1000 \leqslant Re_s$$

Curve 3: inline square layouts

$$j_i = 0.817\ Re_s^{0.632} \qquad 1.0 \leqslant Re_s \leqslant 100$$

$$j_i = 0.290\ Re_s^{-0.418} \qquad 100 \leqslant Re_s \leqslant 700$$

$$j_i = 0.059\ Re_s^{-0.181} \qquad 700 \leqslant Re_s \leqslant 4000$$

$$j_i = 0.185\ Re_s^{-0.324} \qquad 4000 \leqslant Re_s$$

Curve fit for Figure 7: J_1 vs $(\frac{S_{sb} + S + b}{S_m})$

For $(\frac{S_{tb} + S_{sb}}{S_m}) < 0.15$

$$Slope = -0.46617 - -0.40601 \left(\frac{S_{sb}}{(S_{sb} + S_{tb})}\right)$$

$$Y_1 = 1.03 + Slope * 0.415$$

$$A_1 = (Y_1 - 1.0 - 0.075 * Slope)/0.075$$

$$A_2 = (Slope - A_1)/0.3$$

$$J_1 = 1.0 + (A_1 + A_2 * (\frac{S_{sb} + S_{tb}}{S_m})) * (\frac{S_{sb} + S_{tb}}{S_m})$$

For $(\frac{S_{tb} + S_{sb}}{S_m}) > 0.15$

$$Slope = 0.46617 - 0.40601 * (\frac{S_{sb}}{S_{sb} + S_{tb}})$$

$$J_1 = 1.03 + Slope * ((\frac{S_{sb} + S_{tb}}{S_m}) + 0.265)$$

Curve fit for Figure 8: J_b vs F_{sbp}

For $10 \leqslant Re_s \leqslant 100$

$$Factor = ((0.5 - \frac{N_{ss}}{N_c})/0.5)^4$$

For $N_{ss}/N_c < 0.05$ $\quad$ $Slope = -1.2654 + 0.81068 * (\ln(1 + 20 * \frac{N_{ss}}{N_c}))$

For $N_{ss}/N_c \geq 0.05$ Slope = $-0.97686 + 0.40738 (\ln(1 + 20 * \frac{N_{ss}}{N_c}))$

$J_b = \exp(Slope * F_{sbp} * Factor)$

For $Re_s > 100$

For $N_{ss}/N_c < 0.05$ Slope = $-1.26540 + 0.81068 * \ln(1 + 20 * \frac{N_{ss}}{N_c}))$

For $N_{ss}/N_c \geq 0.05$ Slope = $-0.97686 + 0.40738 \ln(1 + 20 * \frac{N_{ss}}{N_c})$

$J_b = \exp(Slope * F_{sbp})$

Curve fit for Figure 9: J_r^* vs N_b

$J_r^* = ((N_b + 1.0) * (N_c + N_{cw})/10.0)^{-0.18}$

Curve fit for Figure 10: J_r vs J_r^*

For $20 \leq Re_s < 100$

$J_r = (J_r^* - 0.2) * (1.25 - Re_s/80.0) + Re_s/100.0$

For $Re_s < 20$

$J_r = J_r^*$

Curve fit for Figure 11: f_i vs Re_s

Curve 1: triangular pitch p/do = pitch to diameter ratio = 1.25

$$f_i = 68\ Re_s^{-1.0} + 1.6 \qquad 1 \leqslant Re_s \leqslant 500$$

$$f_i = 0.97\ Re_s^{-0.19} \qquad 500 < Re_s$$

Curve 2: rotated square p/do = 1.25

$$f_i = 56\ Re_s^{-1.0} + 0.13 \qquad 1 \leqslant Re_s \leqslant 600$$

$$f_i = 0.64\ Re_s^{-0.17} \qquad 600 \leqslant Re_s$$

Curve 3: triangular pitch p/do = 1.3

$$f_i = 52\ Re_s^{-1.0} + 0.17 \qquad 1 \leqslant Re_s \leqslant 500$$

$$f_i = 0.56\ Re_s^{-0.14} \qquad 500 \leqslant Re_s$$

Curve 4: rotated square pitch p/d = 1.3

$$f_i = 42\ Re_s^{-1.0} + 0.11 \qquad 1 \leqslant Re_s \leqslant 600$$

$$f_i = 0.37\ Re_s^{-0.11} \qquad 600 \leqslant Re_s$$

Curve Fit for Figure 12: f_i vs Re_s

Curve 1: inline square pitch p/do = 1.3

$$f_i = 56\ Re_s^{-1.0} + 0.09 \qquad 1 \leqslant Re_s \leqslant 1000$$

$$f_i = 0.65\ Re_s^{-0.14} \qquad 4000 \leqslant Re_s$$

$$f_i = f_{i,1000} + (f_{i,4000} - f_{i,1000})\ (\frac{Re_s - 1000}{3000})$$

Curve 2: square pitch p/do = 1.25

$$f_i = 45\ Re_s^{-1.0} + 0.09 \qquad 1 \leqslant Re_s \leqslant 1000$$

$$f_i = 0.53\ Re_s^{-0.14} \qquad 4000 \leqslant Re_s$$

$$f_i = f_{i,1000} + (f_{i,4000} - f_{i,1000})\ (\frac{Re_s - 1000}{3000})$$

Curve fit for Figure 13: Re vs $\frac{S_{sb} + S_{tb}}{S_m}$

For $\frac{S_{sb} + S_{tb}}{S_m} \leqslant 0.2$

$$Re = 1.03 + (-0.5833 - 0.55417 * (\frac{S_{sb}}{S_{sb} + S_{tb}}))$$

$$* ((S_{sb} + S + b/S_m) + 0.500)$$

For $\frac{S_{sb} + S + b}{S_m} < 0.2$

$$Slope = -0.58333 - 0.55417 * \frac{S_{sb}}{S_{sb} + S + b}$$

$$Y_2 = 1.03 + Slope\ 0.7$$

$$A_1 = (Y_2 - 1.0 - 0.1 * Slope)/0.1$$

$$A_2 = (Slope - A_1)/0.4$$

$$R_1 = 1.0 + (A_1 + A_2 * (\frac{S_{sb} + S_{tb}}{S_m})) * (\frac{S_{sb} + S_{tb}}{S_m})$$

Curve fit for Figure 14: R_b vs F_{sbp}

For $N_{ss}/N_c < 0.05$

$$Re_s < 100\ Factor = ((0.05 - N_{ss}/N_c)/0.05)^4$$

$$Re_s > 100\ Factor = 1.0$$

$$Slope = -3.67879 + 2.41243 * (\ln(1.0 + 20.0\ \frac{N_{ss}}{N_c}))$$

$$R_b = \exp(Slope * Factor * F_{sbp})$$

For $N_{ss}/N_c \leqslant 0.05$

$$\text{if } Re_s < 100 \quad Factor = ((0.50 - N_{ss}/N_c)/0.5)^4$$

if Re_s > 100 Factor = 1.0

$$Slope = -2.82192 + 1.17683 * (\ln(1.0 + 20 * N_{ss}/N_c))$$

$$R_b = \exp(slope * factor * F_{sbp})$$

VITA

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