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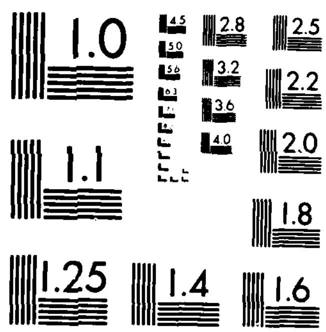

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Classification, nomenclature, utilization and cost estimating of shell and tube heat exchangers are presented along with an historical overview of various methods currently employed in their design.

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COMPUTER-AIDED THERMODYNAMICS DESIGN OF TEMA TYPE E SHELL AND TUBE HEAT EXCHANGERS FOR USE IN LOW PRESSURE, LIQUID-TO-LIQUID, SINGLE PHASE APPLICATIONS

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COMPUTER-AIDED THERMOHYDRAULIC DESIGN OF TEMA TYPE E SHELL AND TUBE HEAT EXCHANGERS FOR USE IN LOW PRESSURE, LIQUID-TO-LIQUID, SINGLE PHASE APPLICATIONS.

BY

NICHOLAS JOHN KOLAR

PRESENTED TO THE FACULTY OF

BUCKNELL UNIVERSITY

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

MASTERS OF SCIENCE IN MECHANICAL ENGINEERING

APPROVED: J. Ben Austin
Advisor

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ABSTRACT

Classification, nomenclature, utilization and cost estimating of shell and tube heat exchangers are presented along with an historical overview of various methods currently employed in their design.

A procedure for providing preliminary estimates of shell and tube heat exchanger design is developed in detail. The author formulates a computer program which employs this sizing algorithm for low pressure liquid-to-liquid heat exchanger applications.

Additionally, problems encountered in the design and manufacture of shell and tube heat exchangers are described along with present methods of solution for each.

I. INTRODUCTION

In the most basic terms, a heat exchanger is a device which provides for the transfer of thermal energy between fluids at different temperatures. Since high energy costs reduce profit margins of industrial products, any means of recovering by-product thermal energy, may result in an increase in corporate gains. For this reason, heat exchangers of all shapes, sizes, and configurations can be found throughout a wide range of industries.

Because of several factors which include: large ratios of heat transfer area to volume and weight, ease of construction and maintenance, design flexibility, and availability of proven design methodology, shell and tube heat exchangers are the most widespread and commonly used basic heat exchanger configuration in the process industries.[1]* It is with this in mind that shell and tube heat exchangers were chosen as the subject for this computer-aided design.

It should be noted that the design of a shell and tube exchanger consists of two basic areas. The first is the thermal-hydraulic design, which considers process fluids, flow arrangements, fluid flow rates, inlet and outlet fluid temperatures, fouling factors, pressure drops, construction materials and overall dimensions to ensure that the design allows for the thermal energy transfer required in the problem. The second is the mechanical design of the heat exchanger which ensures the mechanical integrity of the exchanger components during its operation. Pressure stresses and thermal stresses must be evaluated for each integral component, along with other considerations such as methods or

* Numbers in the [ ] bracket indicate sitings in the References.

B. Delaware Method

Probably the most widely publicized research program in the area of shell and tube heat exchanger design was an ASME sponsored project undertaken at the University of Delaware from 1947 to 1963. Under the auspices of Professors Olaf Bergelin and Allan Colburn, the Department of Chemical Engineering did extensive experimental measurements of shell-side heat transfer and pressure drop for different configurations of shell and tube heat exchangers. Systematically, baffle spacing, baffle-to-shell clearances, and tube-to-baffle clearances were adjusted to measure their effect on the shell-side performance characteristics. The Delaware Project produced several technical papers, and provided invaluable experimental data. One quantitative result of the project was an equation for the heat transfer coefficient which includes five correction factors to the ideal cross-flow heat transfer coefficient. Another, was an equation for the shell-side pressure drop which includes three correction factors to the ideal pressure loss of a shell with no leakage or bypass.

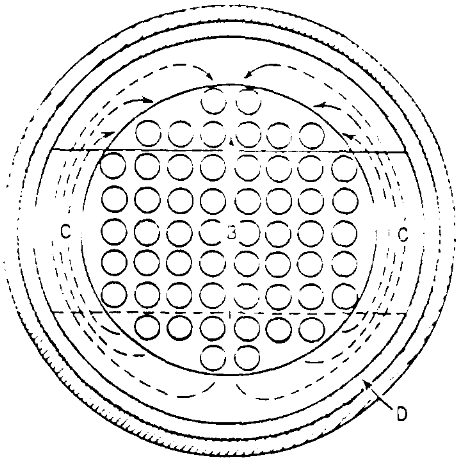
It must be noted, that although this method is generally accepted to be the best available in the open literature, it is not extremely accurate. Tests have shown that this method predicted shell-side heat transfer coefficients from about 50 percent low to 100 percent high, and the predicted shell-side pressure loss was from about 50 percent low to 200 percent high.[10]

C. Bell Method

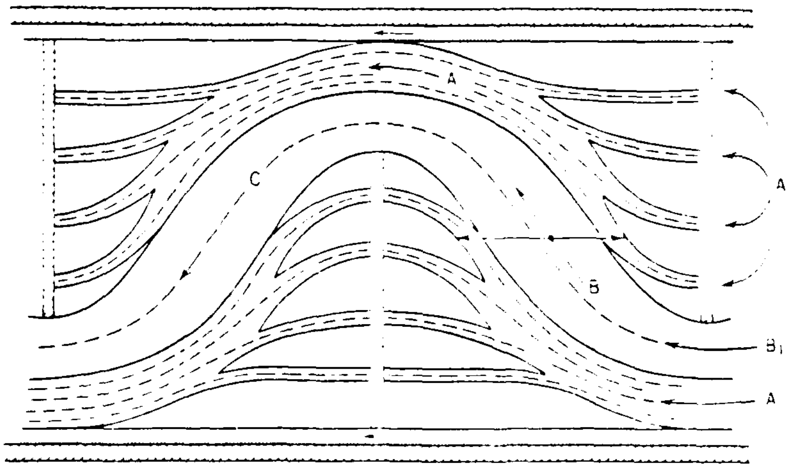
Professor Kenneth J. Bell from Oklahoma State University devised a method for preliminary design of shell and tube heat exchangers which

FIGURE 5

LEAKAGE PATHS FOR FLOW BYPASSING THE TUBE MATRIX



FRONT VIEW



SIDE VIEW

IV. DESIGN TECHNIQUES

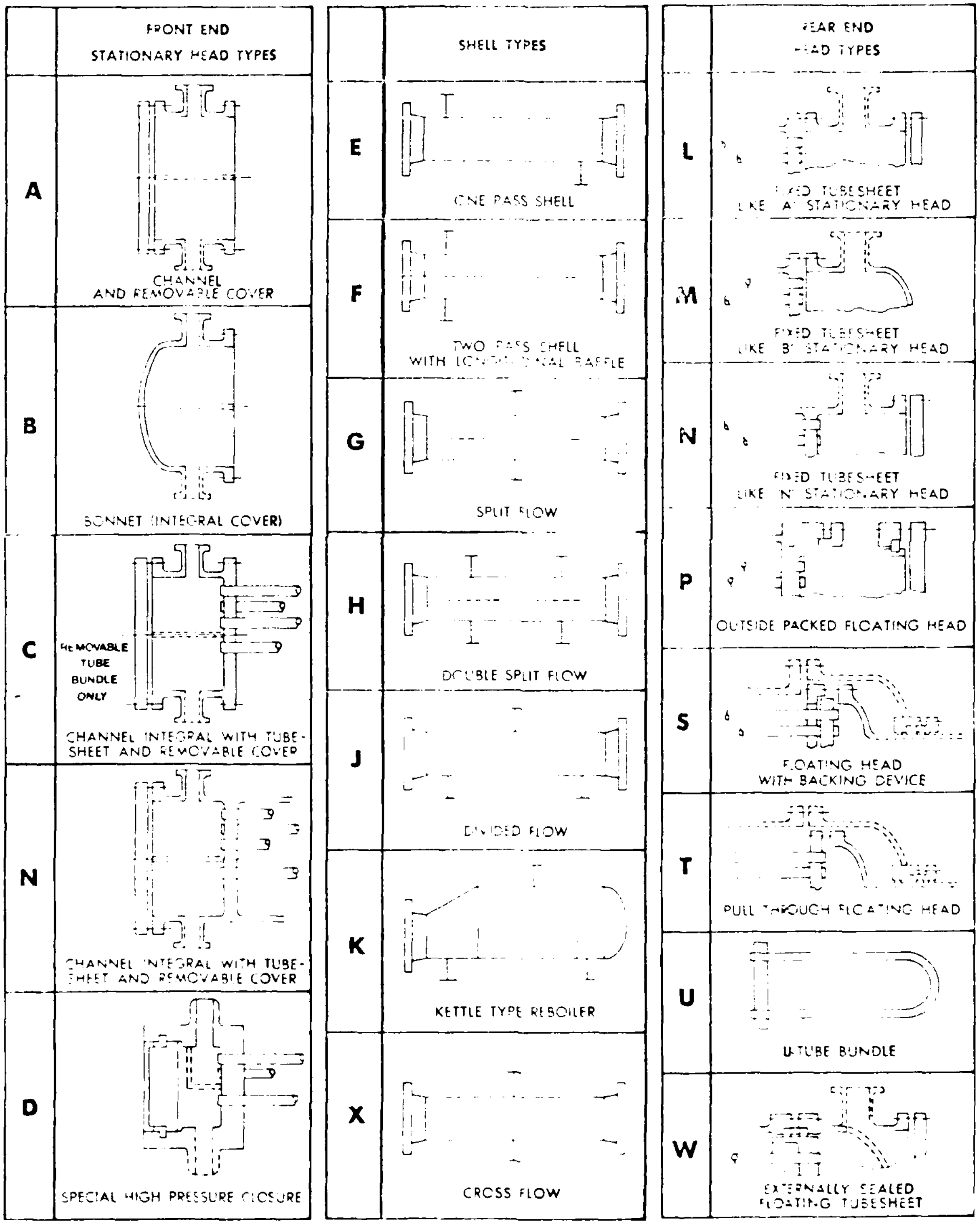
A. General

The most dynamic period of development in heat transfer occurred between 1920 and 1950. During this period almost all of the principles of thermodynamic behavior of heat exchangers became understood. One book, which was published after this important technological period, stands alone in the design of shell and tube heat exchangers. The book, Process Heat Transfer, by Donald Q. Kern, a professor from Case Institute of Technology, has provided the methods and procedures for the design of more shell and tube heat exchangers than any other single source.[8]

Probably the most difficult design parameters to quantify are the shell-side performance characteristics. Ideally, the shell-side flow can be viewed as a cross flow of fluid across a bank of tubes. But in reality, the flow goes both across the tubes and parallel to the tubes due to the serpentine motion of the fluid caused by the internal baffles of the exchanger. This is further complicated by leakage which occurs due to gaps between the baffle and the shell and spaces between the baffle and the tubes. This leakage or bypass flow causes a decrease in the heat transfer coefficient and uncertainty in the shell-side pressure drop. A diagram of the leakage paths taken from Reference 9 is shown in Figure 5. The elusive shell-side performance characteristics have been the cause for many independent studies, ultimately resulting in innovative methods for design.

FIGURE 4

TEMA TYPE DESIGNATIONS



and abroad. It has proven successful over the years because it is based on sound engineering practice. Probably most notable is the standardization of size designation and nomenclature which is used throughout the industry. Figure 4 shows the standard letter designations for the stationary head types, shell types, and rear-end head types. The program presented in this thesis designs TEMA "E" shell configurations.

TEMA standards require that ASME codes are followed. TEMA does cover internal components where ASME code rules are not available or are not applicable. In essence, TEMA provides good solid design and construction rules for heat exchangers considered as a special type of pressure vessel.

Three different classes of heat exchangers are developed in detail.

Class "R" - Shell and tube heat exchanges which must meet the severe requirements of the petroleum industry.

Class "C" - Shell and tube heat exchangers which must meet the moderate requirements of commercial and general process applications.

Class "B" - Shell and tube heat exchangers which must meet the requirements of chemical process service.

For each classification, TEMA sets the standards for design for each of the integral parts of the heat exchanger. These include: tubing, shells, baffles, heads, gaskets, tubesheets, channels, nozzles, end covers and bolting.

Additionally, TEMA provides information on generally used materials, fouling resistances and physical properties of fluids. General information such as characteristics of tubing, metal properties and conversion factors are also provided.

There is presently international interest in developing a code which will provide rules for use on an international basis. There are several major obstacles to its development including safety factors and material specifications.

D. American National Standards Institute (ANSI) Standards

ANSI is a Federal organization which manages standards throughout the United States. The ASME Code is an ANSI standard. A particular ANSI standard that is not covered by any other code is on that deals specifically with chemical processes.

E. American Petroleum Institute (API) Standards

API standards deal specifically with shell and tube heat exchangers which are used in connection with oil refinery services. The standards have been developed over the years from the accumulated knowledge and experiences of petroleum refiners nationwide.

F. Heat Exchange Institute (HEI) Standards

The HEI organization has developed standards for heat transfer equipment for use in steam power plants. Particular areas include: standards for closed feedwater heaters, standards for power plant heat exchangers (electric generating plants), and standards for steam surface condensers.

G. Tubular Exchanger Manufacturer Association (TEMA) Standards

This standard was established in 1939 in order to provide a measure of quality as well as uniformity in the design and construction of shell and tube heat exchangers. It is used extensively in the United States

III. CODES AND STANDARDS

A. General

Shell and tube heat exchangers are the most versatile and most widely used type of heat transfer equipment. Units can be designed to handle sensible heating, condensing and boiling. Operating pressures range from near vacuum to above 5800 psi. They may weigh as much as 200 metric tons and encompass as much as 50,000 ft^2 of heat transfer surface area.[7] To find codes and standards for its complex design, one needs to go to several different sources.

B. American Society of Mechanial Engineers (ASME) Boiler and Pressure Vessel Code

A shell and tube heat exchanger is indeed a pressure vessel and therefore must comply with several sections of this code which apply to: power boilers, material specifications, nuclear power plant components, nondestructive examination, pressure vessels, and welding and brazing qualifications.

In the United States and Canada, the ASME code is recognized as the means by which control over the design, fabrication and testing of boilers and pressure vessels is achieved.

C. Foreign Pressure Vessel Codes

Most major industrial countries such as France, Great Britain, and West Germany have their own boiler and pressure vessel codes. Normally, if a unit is to be placed in service in one of these countries, their standards and limitations must be adhered to.

FIGURE 3

BAFFLE CONFIGURATIONS

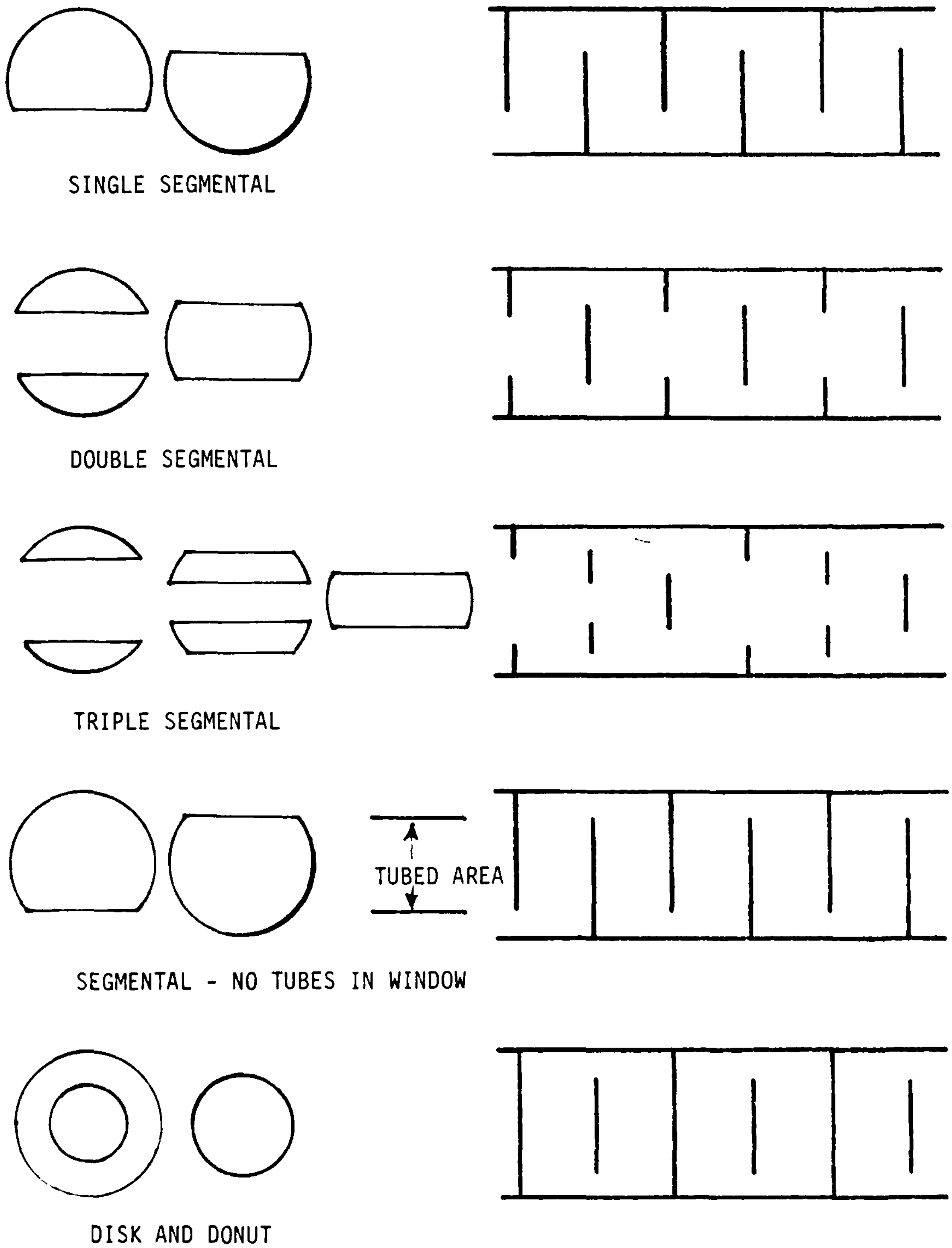
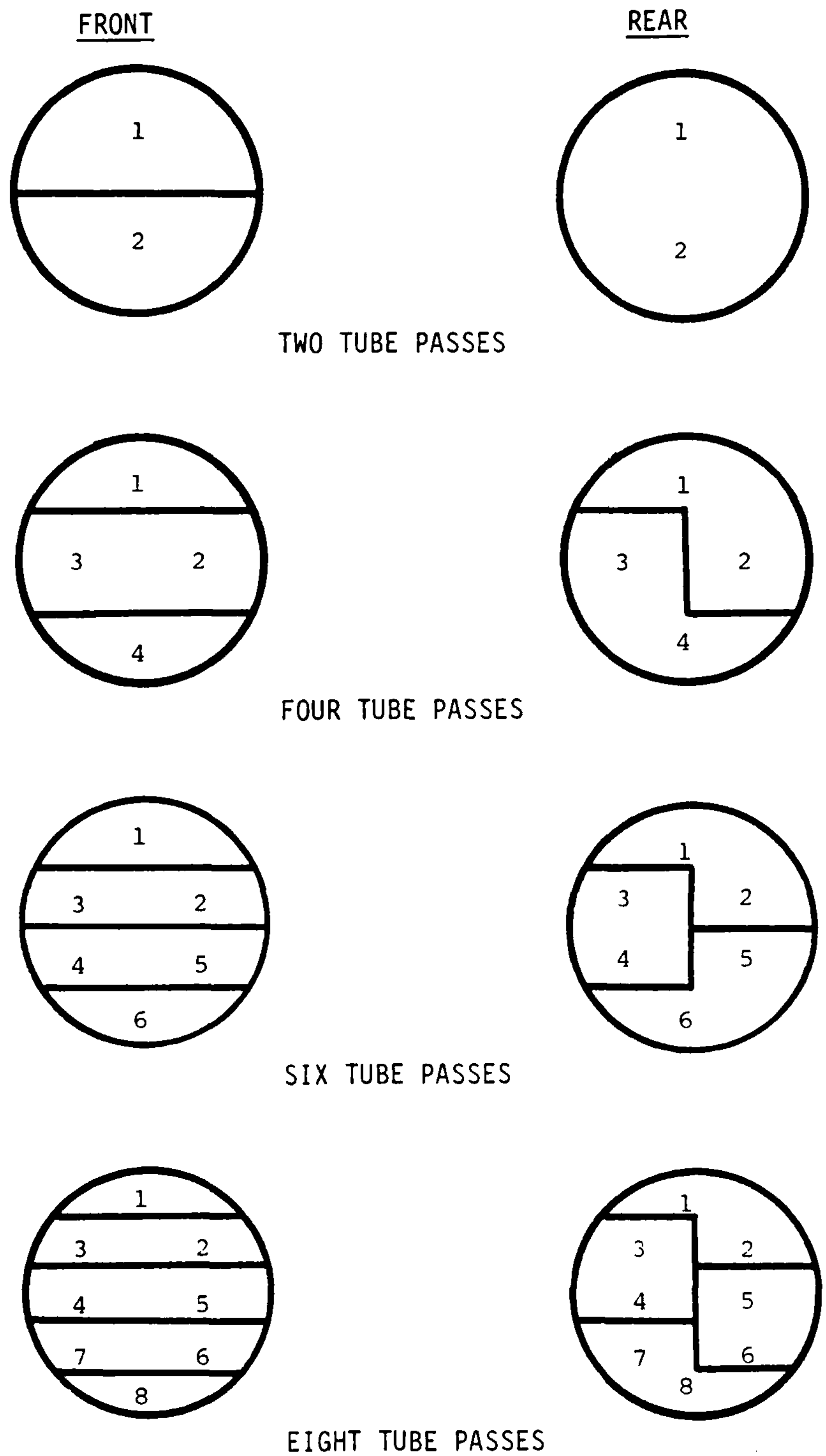


FIGURE 2

PARTITION PLATE CONFIGURATIONS



The most common configurations are: one shell pass two tube passes, one shell pass four tube passes, one shell pass six tube passes, and one shell pass eight tube passes. In these multiple-tube pass heat exchangers, pass partitions are required in the tube-side chambers of the heat exchanger to direct the flow of the tube fluid. The arrangement of the tube-side pass partitions affects the heat transfer behavior of the heat exchanger, and obviously affects the placement of the tube fluid entrance and exit nozzles. The most common tube-side pass partitions are shown in Figure 2.[3,4]

I. Baffles

The interior baffles of the heat exchanger serve two important functions. First, they support the tube bundle keeping the proper tube spacing during both construction and operation and at the same time prevent unwanted vibration in the tube bundle. Secondly, they provide a path for the shell-side fluid to move back and forth across the tube bundle increasing the shell-side fluid velocity and also increasing the heat transfer coefficient.

Several of the baffle configurations that are in use today are shown in Figure 3. The most common (and the one that this computer-aided design program is based upon) is the single segmental. The segment sheared off must be less than 50 percent and for liquid to liquid applications, a baffle cut of 20 - 25% is common.[6]

are: roller expansion of the tube into ungrooved tubesheets, hydraulic expansion of the tubes into grooved tubesheets, and welding the tube ends directly to the tubesheets.

D. Shell

The shell is a cylindrical vessel which holds the shell-side fluid. In small diameter shells (less than 24 inches), the shell is cut from standard pipe. For larger diameter shells, the shell is fabricated from rolling metal sheets of the desired dimensions and longitudinally welding the butt joint.

E. Shell Nozzle

The shell has two nozzles which allow for the entrance and exit of the shell-side fluid. The inlet nozzle normally also has an impingement plate which prevents the incoming fluid stream from impacting directly upon the exposed tubes causing unwanted erosion and vibration.

F. Tube-side Channels and Nozzles

The tube-side channels and nozzles direct and control the flow of the tube-side fluid when it is not in the tubes of the heat exchanger.

G. End Covers

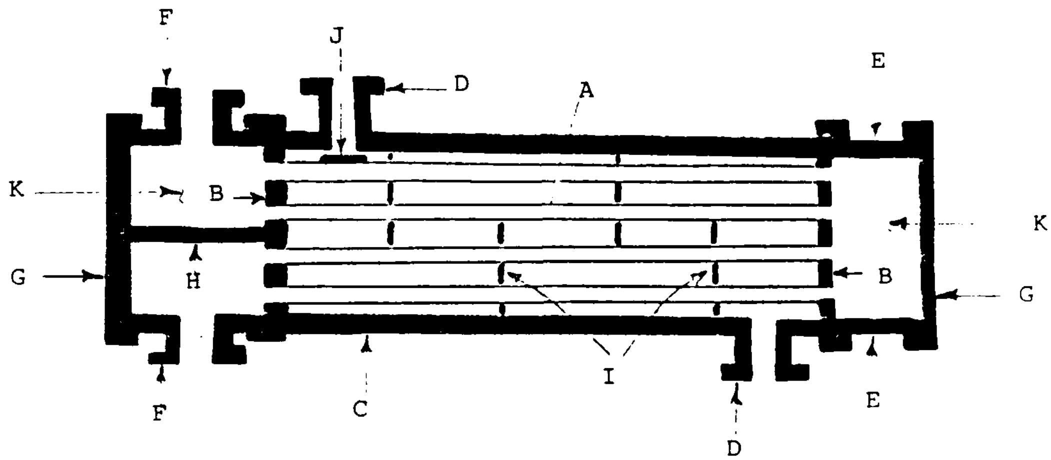
The end covers are metal plates that normally are bolted onto the ends of the shell. They are bolted rather than welded to facilitate maintenance and inspection of the tubes and channels in the heat exchanger.

H. Pass Partition Plate

The fluid in the tubes may travel the length of the exchanger more than one time. These are heat exchangers with multiple tube passes.

FIGURE 1

DIAGRAM OF A TYPICAL FIXED TUBE SHEET SHELL AND TUBE HEAT EXCHANGER



A. TUBES

B. TUBESHEETS

C. SHELL

D. SHELL-SIDE NOZZLE

E. TUBE-SIDE CHANNEL

F. TUBE-SIDE NOZZLE

G. END COVER

H. PASS PARTITION PLATE

I. BAFFLE

J. IMPINGEMENT PLATE

K. TUBE-SIDE CHAMBER

II. BASIC COMPONENTS OF SHELL AND TUBE HEAT EXCHANGERS

A. General

For the purpose of establishing standard terminology, Figure 1 illustrates several basic components of shell and tube heat exchangers which will be described later. Even though there are thousands of specific design features that can be used, these specific components will invariably be present in all shell and tube heat exchangers.

B. Tubes

The tubes are the most basic component of the shell and tube heat exchanger. The tube wall provides the heat transfer surface between the fluid flowing inside the tube and the fluid flowing inside the shell. The tubes can be either seamless or welded. The metals used are carbon steel, low alloy steel, stainless steel, copper and copper alloy, nickel and nickel alloy, aluminum and aluminum alloy, titanium and zirconium, although special metals may be used to meet specific applications.[2]

The tubes may be bare or have integral circumferential finning. The computer-aided design program formulated in this thesis assumes that the tubes used are bare. The assemblage of all the tubes in the heat exchanger is referred to as the tube bundle.

C. Tubesheets

The tubes of the exchanger are held in place by being inserted into holes in metal sheets at either end of the shell. They are fixed in place by a variety of methods. The tube to tubesheet joint has the single objective of sealing against interleakage between the tube-side and shell-side fluids. Common methods of joining the two components bonding components, types and sizes of gaskets and seals, and minimization of flow induced vibrations. This thesis covers only the thermal-hydraulic design of the heat exchanger. Although equally important as the thermal-hydraulic design, the mechanical design is much more intricate and has been left to those who are most capable of accomplishing the task - namely, the shell and tube heat exchanger manufacturers of the United States and abroad. For it is their decades of experience, experimentation, and research that has developed the standards of construction and manufacture that are in use today.

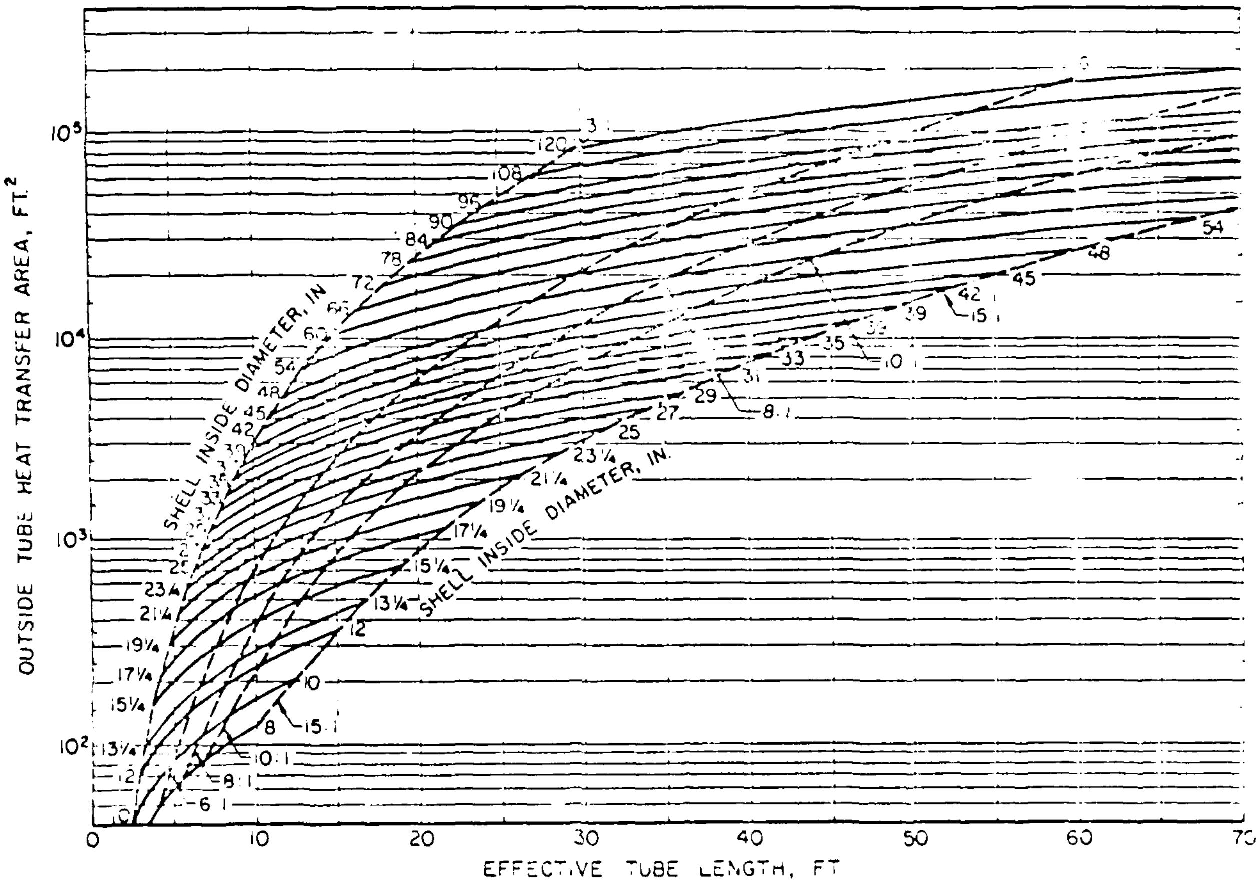
also takes into account deviations from ideality. His method is unlike the Delaware method in that he does not directly calculate the shell-side heat transfer coefficient. He uses typical overall design coefficients (U-valves) for shell and tube heat exchangers depending upon the type of fluids that are used. The data has been collected from years of experimental performance and commercial usage. Using this overall design coefficient, Bell calculates the required outside tube heat transfer area. The key to the Bell Method is the graph shown in Figure 6. Bell has depicted on this graph the possible configurations of shell and tube exchangers using standard tube counts for 3/4 inch O.D. tubes on a 15/16 inch triangular pitch for a fixed tube sheet exchanger with one tube-side pass. Also shown are the effective tube length to shell diameter ratios (3:1, 5:1, etc.) available. The user simply enters the ordinate of the graph with the required area and immediately knows the combinations of tube length and shell diameter that will provide that area in a single shell for an exchanger of Bell's standard tube size and layout. For those exchangers that do not conform to Bell's standard, three correction factors to the area calculation are utilized; one for different tube diameters and layouts, one for different numbers of tube-side passes, and one for different TEMA tube bundle construction types. The Bell Method is used solely for preliminary estimates of the heat exchanger size, and is obviously not a complete design methodology.

D. Tinker Method

In 1951 Townsend Tinker, vice-president for engineering of Ross Heat Exchanger Division, American Radiator and Standard Sanitary Corporation

FIGURE 6

BELL METHOD DESIGN CHART



was the first to develop a complex but effective methodology for including the effects of deviations from ideality. The fluid streams in the shell depicted in Figure 5 was taken from his original work. Tinker uses graphs of modified shell-side Reynolds numbers vs. shell-side fluid friction factors and modified shell-side Reynolds numbers vs. shell-side heat transfer coefficients for three different tube-hole patterns: triangular pitch, square pitch and staggered square pitch. The experimental data he uses comes from the research done by the Delaware Project.

While Tinker's basic derivations are general, they are cumbersome unless the range of possible combinations of his geometric parameters is reduced. He does this by adopting representative values for three geometric ratios: the ratio of the clearance between the tubes and holes in the baffles to the tube outside diameter, the ratio of the clearance between the baffle and the shell to the shell inside diameter, and the ratio of the shell inside diameter to the outside diameter of the tube bundle. These ratios are closely related to the various bypass flow streams shown in Figure 5 and are kept as small as possible, consistent with manufacturing tolerances and the clearances required for assembly.

Once these ratios are fixed, Tinker showed that the various bypass flow rates become functions of five more geometric ratios: the ratio of the shell diameter to the tube length between baffles, the ratio of the tube pitch to the tube length between baffles, the ratio of the tube pitch to the tube outside diameter, the ratio of the baffle height to the shell diameter, and the ratio of the number of rows of tubes traversed per pass to the shell inside diameter divided by the tube pitch.

The modifications to the Reynolds number are made using correction factors which depend upon the various clearance ratios previously cited. Once the modified Reynolds number is calculated, the heat transfer coefficient and shell-fluid friction factor can be determined from the appropriate chart.

As one would suspect, Tinker's calculated values, with their many complex relationships, have been proven to be as much as 50 percent deviant from the experimentally determined values. Tinker was however, the forerunner in this area, and his published works have stimulated many advances in this area.[11,12]

E. Devore Method

Anthony Devore used the basic method of Tinker but made several simplifications. First, he used standard tolerances for commercial exchangers in lieu of certain of Tinker's geometric ratios. Second, he limited the number of baffle cut ratios. Lastly, he provided several charts and tables for easy determination of correction factors. The results are even more conservative than those obtained via Tinker's original method.[13]

F. Numerical Methods

The most promising method of calculating the design parameters for a shell and tube heat exchanger stems directly from sophisticated computer-based numerical procedures for calculating fluid-flow distributions. Shell and tube exchanger geometries necessitate a three-dimensional procedure. Three velocity components have to be computed for the shell fluid at each point. Furthermore, the lack of any axial symmetry causes these velocities, as well as the temperatures and pressures, to vary throughout all three space dimensions. The solution to the differential equations which govern the flow of a continuous fluid can be accomplished by using a finite-difference approach. Alternately, the space within the shell can be regarded as uniformly filled with fluid, through which is distributed, on a fine scale, a resistance to fluid motion.

This alternate continuum approach has two advantages. First, it allows the use of a grid which is less fine than the finite-difference approach, with a savings in computer time. Second, it allows for the use of the extensive experimental data that is available on the pressure-drop and heat transfer performance of fluid flow through tube banks. The results of this type of computer-based numerical method is an increased realism of the predictions of the pressure drop and heat transfer coefficient.[14]

G. Proprietary Programs

During the boom years of heat transfer development, universities and industries worked hand in hand to solve heat transfer problems. Since that time, the emergence of the new fields of nuclear energy and aerospace flight coupled with the revolutionary computational power made available by high-speed digital computers caused a partial disjunction between this university-industry fellowship. Professor Thomas Sherwood from the Massachusetts Institute of Technology describes it in these words:

> ...Prior to World War II the emphasis at universities was on the collection and correlation of data intended to be of direct use by the practicing design engineer.

> Industry had few such data and published little, so schools felt a responsibility to fill the need. This urge to be immediately helpful to industry has largely disappeared today; research in schools is now along more scientific and theoretical lines, hopefully of value to industry a generation hence. Our rapport with industry has suffered.[15]

The new fields of nuclear energy and aerospace flight transfused research funds of unprecedented magnitude into the laboratories and research institutes. The results of this expanded research had, unfortunately, little applicability to industrial needs.

The computer's mammoth capabilities enabled theorists and design engineers alike to solve the heat transfer problems using numerical methods that were abandoned in the boom years due to their computational sophistication. Computer programs were rapidly developed and the need for the university's experimental data ebbed.

What evolved to take the place of university research were organizations that did private research for the benefit of the members of their consortium. Their experimental data along with their computer source-codes were proprietary. The suppression of publicly available information and experimental data was widespread, and today the effects of this lack of scholarly interchange cannot really be discerned.

In the area of heat transfer, one organization in particular stands out as being the most knowledgeable in the field. Heat Transfer Research Incorporated (HTRI) in Alnambra, California, has developed the most sophisticated software in use today in the design of shell and tube heat exchangers. The majority of the shell and tube heat exchanger manufacturers use their programs, such as ST-4 to compute the heat transfer characteristics and pressure drops in their heat exchangers.[16,17]

V. LIQUID-LIQUID SHELL AND TUBE HEAT EXCHANGER UTILIZATION

Because shell and tube heat exchangers can be constructed with very large heat transfer surfaces in relatively small volume, can be fabricated from alloy metals to resist corrosion and can be used for heating and cooling virtually all fluids, they are universally accepted as the most important class of heat transfer equipment.[18]

From cooling chemical solutions in world-class chemical processing plants to heating water for a backyard swimming pool, applications for liquid-liquid shell and tube heat exchangers abound. The nuclear industry uses them to cool heavy water in atomic reactor installations. The metallurgical industry uses them to cool quench oil. The Food Processing industry uses them for vegetable oil heaters and dairy product coolers. The petroleum industry uses them to cool oil used in machine lubrication on offshore oil platforms by using sea water as the cooling medium. Local schools may use them to preheat low-grade fuel oil. The list of applications will be as long as the imagination of the design and process engineer is broad.[19,20]

Virtually all industries that produce a waste or by-product fluid stream that has an exit temperature higher than the inlet fluid temperature can use a liquid-liquid shell and tube heat exchanger to recover some of the thermal energy that would otherwise be lost. Particularly during the present period of high energy costs, the capital outlay for a heat exchanger is recouperated via lowered heating costs allowed by the thermal energy retrieved from the waste stream.

VI. PROBLEMS ENCOUNTERED IN SHELL AND TUBE HEAT EXCHANGER DESIGN

A. General

The designer of shell and tube heat exchangers is plagued with an enormous amount of uncertainties. Two important problems that must be dealt with, that are particularly troublesome in shell and tube heat exchangers designed for liquid-liquid applications are flow induced vibrations and fouling. Each of these problems will be discussed and possible solutions will be provided.

B. Flow Induced Vibrations

The cross-flow component of the shell-side fluid stream has been held primarily responsible for the widespread failure of tubes that occurred in the late 1960's. Other factors, such as baffle configurations and unsupported tube spans, were also defined as having at least a part in the destructive vibratory action. Present-day understanding of this complex phenomenon is limited. In fact, TEMA standards allow a disclaimer that the manufacturer is not responsible or liable for any direct, indirect, or consequential damages resulting from vibration.[21]

Mechanical failure due to flow-induced vibrations can occur in several different ways. The amplitude of the vibration may be large enough that two tubes collide with each other or a peripheral tube will strike the shell inner wall. In either case, the tube wall is worn down, and the tube eventually leaks. The tube may repeatedly impact with the inside of the baffle hole causing the same type of abrasion. Failure can also be caused by vibration-induced stress fluctuations in the tube when macroscopic flaws on the tube surface propagate to serious cracks.[22] In any case, the result is the same. The heat exchanger's efficiency is degraded and eventually must be shut down for costly repair or retubing. Without necessary design modifications however, the repaired heat exchanger will be put back into service and will continue to have serious problems caused by the unwanted vibrations.

Presently, world-wide research into this problem has produced only a few experimentally proven methods to reduce vibrations. Since field experience has shown that tubes in the area of the baffle cuts are the most prone to vibrate, the utilization of the "no tubes in window" baffle configuration shown previously in Figure 3 obviously eliminates these vibration-proven tubes. The draw-back here is that in order to accommodate the reduced tube area, the shell diameter must be increased. Another method widely acclaimed is the insertion of helical rods as tube spacers at the midpoint of unsupported tube spans. Stability is added and vibration is reduced, but the trade-off is flow diversion and consequent increased fluid friction caused by these obstructions. Simple remedies such as decreasing the hardness of the baffle material and increasing the lightness of the tube-to-tube bundle fit have had limited success. No doubt, the present day research in this area will lead to a better understanding of this phenomenon and result in newer methods of design which will reduce or eliminate the likelihood of vibration.[23]

C. Fouling

Undesirable buildup of deposits on tubes have plagued shell and tube heat exchanger manufacturers and users since the first unit was placed into operation. Performance degradation of up to 80%, and complete failures were commonplace. The mechanisms of fouling have been categorically identified as: corrosion fouling, precipitation fouling, particulate fouling, chemical reaction fouling and biological fouling. The identification of the mechanisms was relatively easy, the difficulty has been the development of empirical models to characterize and predict each of the mechanisms. Inspite of enourmous research in this area, good prediction methodology is lacking. Designers still add typical fouling resistances based on past experience to their calculations in determining the overall heat transfer coefficient for the exchanger. This intentional oversizing of the exchanger considerably increases the capital cost of the exchanger, yet does nothing to deter the fouling from taking place. In fact, this design practice can lead to catastrophic results, in that a unit designed as a sensible heat exchanger may initially operates as a boiler.[24,25]

Due to our lack of understanding of fouling, techniques to reduce its damaging effects are limited. The basic rule in design for fouling is to place the fluid which causes the greatest amount of fouling inside the tubes. There, stagnant regions, which cause the worst fouling, are less likely to occur. Furthermore, when the inevitable maintenance due to fouling comes about, the mechanical cleaning of the insides of the tubes is usually much easier than removing the scale on the outside of tubes in a tube bundle.[26] The use of exotic metals such as zirconium and titanium in the manufacture of the tubes has proven to be effective in decreasing the rate of fouling build-up.[27] Finally, the Heresite-Saekaphen Company in Wisconsin has patented a tube coating system which prevents fouling, and is resistant to erosion, corrosive chemicals and solvents. Experimental and real-life results have shown that the service life of tube bundles has increased more than five-fold. The cost for this .006 inch phenolformaldehyde coating is cost-effective since the coating eliminates the need to clean the tubes, and allows for an initial exchanger design that is not oversized.[28] Just as with flow induced vibrations, the ongoing research in this area will bring about new innovative methods to combat the fouling problem.

VII. COMPUTER-AIDED DESIGN PROCEDURE

A. General

In order to provide prospective shell and tube heat exchanger buyers a preliminary size and cost of a shell and tube heat exchanger that will accomplish the required sensible heat transfer, a computer program was developed. The program designs exchangers with TEMA Type E shells from six to sixty inches in two inch increments. One shell-side pass is provided with either two, four, six, or eight tube-side passes. The program is limited to liquid-to-liquid applications where pressures are less than 300 pounds per square inch.

The program is user-friendly in that instructions for the use of the program are initially displayed, and the user needs to respond only to questions asked. Examples of input are provided, and required units are clearly established. The input data can be modified and final design configurations can be compared. The nomenclature used throughout this chapter can be found in Appendix A. The main program source code is found in Appendix B.

B. Instructions

The initial instructions include remarks about the program's capabilities, limitations, and required input data. The required inputs are: tube-fluid entrance temperature (TT1), tube-fluid exit temperature (TT2), shell-fluid entrance temperature (TS1), tube-fluid flow rate (MT), shell-fluid flow rate (MS), maximum tube length (LRESTR), maximum allowable shell-side pressure loss (AMAXSP), and maximum allowable tube-side pressure loss (AMAXTP). The temperature units are in °F, the flow rates are in gallons per minute, lengths are in feet and pressure losses are in pounds per square inch. The initial instructions also provide a listing of the major works that were used as sources for the program's development.

C. Input Data

The user is asked to make a choice of tubing to be used in the exchanger from a listing of thirty-six common tube sizes and Birmingham Wire Gages (BWG). Tube outside diameters (DTOT) range from 0.75 to 1.5 inches and the range of BWG is from seven to twenty. A choice of tube material must also be made from a selection of eighteen different metals and alloys ranging from common carbon steel to the more exotic metals of titanium and zirconium.

The next step is to choose the fluids that are to be used. Seven fluids are available for selection, they are: water, methyl alcohol, gasoline, Dowtherm A, kerosene, SAE 10 lubricating oil, and ethylene glycol. In keeping with design procedures, the fluid that will cause the worst fouling should be placed inside the tubes. Any combination of these seven fluids are design possibilities.

The input data previously discussed is input to the computer as prompted for by the program. The last required input is whether or not the addition of fouling factors is desired to be used in the calculations. A negative response to this question will result in the heat exchanger being sized utilizing the overall heat transfer coefficient calculated assuming clean tubes. This may result in the heat exchanger not meeting its design performance sometime in the future if no precautions are taken to guard against fouling.

exchanger flow arrangement. These parameters are useful in allowing for compact graphical presentation of exchanger performance. Figure 8 graphically shows the relationship between these parameters for a heat exchanger with one shell pass and 2-, 4-, 6-, or 8-tube passes. The algebraic relationship between the number of transfer units, the capacity rate ratio and the heat exchanger effectiveness is:[39]

$$NTU = -(1+C^2)^{-1/2} \ln \left[\frac{\frac{2}{E} - 1 - C - (1+C^2)^{1/2}}{\frac{2}{E} - 1 + C + (1+C^2)^{1/2}} \right]$$

where $C = \frac{C_{MIN}}{C_{MAX}}$

Although this equation was derived for a heat exchanger with one shell pass and two tube passes, it is used as the basic NTU-Relationship. The equations that result from the study of 4-, 6-, or 8-pass exchangers yield results that are numerically so close to the two-tube-pass situation that nothing is to be gained by the utilization of these more complex equations.[40]

M. Calculation of Estimated Overall U-Value

Tables of U-values typical of various applications of shell and tube exchangers are widespread. They normally are tabulated giving a range of U-values that can be expected for two particular fluid streams. For example, a water-to-water shell and tube heat exchanger will have a U-value of between 250 - 300 BTU/hr-ft^2-°F. This is in contrast to a water-to-heavy organic liquid shell and tube heat exchanger which would

The fluid with the lowest capacity rate is called the minimum fluid (CMIN). The fluid with the greatest capacity rate is called the maximum fluid (CMAX). The capacity rate ratio is defined as CMIN/CMAX.

K. Heat Transfer Effectiveness Calculation

Heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the maximum possible heat transfer. Kays and London use this equation for effectiveness:[37]

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_h(t_{h,in} - t_{h,out})}{C_{min}(t_{h,in} - t_{c,in})} = \frac{C_c(t_{c,out} - t_{c,in})}{C_{min}(t_{h,in} - t_{c,in})}$$

where C_{min} is the smaller of the C_h and C_c magnitudes.

The thermodynamically limited maximum heat transfer rate (q_{max}), would only be realized in a counterflow heat exchanger of infinite heat transfer area. Although this effectiveness is in reality a heat transfer effectiveness, the above equation reduces to a temperature ratio. Using nomenclature from Appendix A:

$$EFF = \frac{|T2MIN - TIMIN|}{|TIMAX - TIMIN|}$$

L. Determination of the Number of Transfer Units

The number of heat transfer units (NTU) is a nondimensional expression of the heat transfer size of the exchanger. Kays and London use this equation for number or exchanger heat transfer units:[38]

$$N_{tu} = \frac{AU_{av}}{C_{min}} = \frac{1}{C_{min}} \int_0^A U \, dA$$

The number of exchanger heat transfer units is a function of three parameters: the effectiveness, the capacity rate ratio, and the heat

In Case 2:

$$TS2 = TS1 - \frac{Q \times 7.4805}{MS \times DENS \times CPS \times 60.0}$$

Of course, the fluid properties used in the previous two equations were only assumed average values. Once the initial value for the shell exit temperature is obtained, the mean fluid temperature is calculated by:

$$TSAVE = \frac{TS1 + TS2}{2}$$

Using this value for the mean shell-fluid temperature, the fluid property subroutines are called, the corrected values of density and specific heat are obtained. A new value for the shell-fluid exit temperature is calculated using the aforementioned equations, and the new value for the shell-fluid exit temperature is compared to the previously calculated value. If they differ by more than one degree of temperature the iterative process is continued. If the two values for the shell-fluid exit temperature do not differ by more than a single degree, the last calculated value for the exit temperature is recognized as the real shell-fluid exit temperature for use in any future calculations.

J. Calculation of Maximum and Minimum Capacity Rates

The mass flow rate x the specific heat of a fluid is defined as the capacity rate of that fluid.[36] Equations to calculate the capacity rates of the shell and tube fluid are:

$$CS = \frac{MS \times DENS \times CPS \times 60.0}{7.4805}$$

$$CT = \frac{MT \times DENT \times CPT \times 60.0}{7.4805}$$

I. Shell-Fluid Exit Temperature Determination

Since the exit temperature shell-fluid is an unknown, it can be found by doing an energy balance on the heat exchanger. Two possible conditions could exist. In Case 1, the tube fluid is being heated. Here the energy gained by the tube fluid must be lost by the shell-fluid. In Case 2, the tube-fluid is being cooled. Here the energy lost by the tube-fluid must be gained by the shell-fluid. The heat transfer rate of the tube-fluid is dependent upon these factors: the tube-fluid flow rate, the tube-fluid density, the tube-fluid specific heat, and the magnitude of the tube-fluid exit and entrance temperature difference.

In Case 1:

$$Q = \frac{MT \times DENT \times CPT \times 60.0}{7.4805} \times (TT2 - TT1)$$

In Case 2:

$$Q = \frac{MT \times DENT \times CPT \times 60.0}{7.4805} \times (TT1 - TT2)$$

Note that 60.0 is a conversion factor for minutes per hour and 7.4805 is a conversion factor for gallons per cubic feet.

Since an average shell-fluid temperature cannot be determined until the exit temperature of the shell-fluid is known, average values for the fluid properties are assumed depending upon the shell-fluid choice.

An iterative procedure will be followed in order to obtain the correct value of the shell exit temperature. The equations are:

In Case 1:

$$TS2 = TS1 - \frac{Q \times 7.4805}{MS \times DENS \times CPS \times 60.0}$$

H. Thermo-Hydraulic Properties of the Tube Fluid

Four basic properties of each fluid must be known to design a liquid-to-liquid heat exchanger. These are: density, viscosity, specific heat and thermal conductivity. The temperature at which the fluid properties are determined is the mean fluid temperature, which is simply the arithmetic average of the entrance and exit temperatures. The mean tube fluid temperature is:

$$TTAVE = \frac{(TT1 + TT2)}{2}$$

This computer program calculates these four fluid properties by way of calling separate subroutines. Each of the seven fluid choices has four subroutines which are associated with it, one for each of the fluid properties. For example, water has these subroutines:

Subroutine	Desired Fluid Property
DWATER	Density
CPWATR	Specific Heat
VWATER	Dynamic Viscosity
KWATER	Thermal Conductivity

Each subroutine contains temperature dependent data of its particular fluid property. An interpolation algorithm is employed to calculate the desires fluid temperature if the mean fluid temperature does not match the temperature data points filed for that particular fluid property. A complete listing of all of the fluid property subroutines for water is found in Appendix C. The other fluid subroutines are strikingly similar in form to these subroutines.

between the tubes, which depends on the tube pitch and tube size, that make up the crossflow area.

The tube clearance is simply:

$$TC = TPH - DTOT$$

The calculations of the shell-side flow rates can now be inscribed. The minimum shell-side flow rate for a heat exchanger using a K1 baffle spacing is:

$$QSK1L = 3.116875 \times (0.2 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

The maximum shell-side flow rate for the same exchanger is:

$$QSK1H = 3.116875 \times (0.2 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

The flow rate equations for exchangers which use the other two baffle spacings are:

For a K2 baffle spacing:

$$QSK2L = 3.116875 \times (0.45 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

$$QSK2H = 3.116875 \times (0.45 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

For a K3 baffle spacing:

$$QSK3L = 3.116875 \times DSI \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

$$QSK3H = 3.116875 \times DSI \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

G. Maximum and Minimum Shell-Side Flow Rates

In keeping with generally recommended practice the shell-side velocity should be at least 2 ft/sec but not more than 4.5 ft/sec.[33,34] The shell-side crossflow area is dependent on three factors: the tube pitch, the diameter of the shell, and the baffle spacing. The only parameter that has not been discussed previously is the baffle spacing. The baffle spacing is the distance between baffles within the exchanger. There is a generally recommended range of baffle spacings. The baffles should be no closer than a distance equal to one-fifth the shell diameter and should not be spaced farther apart than a distance equal to the shell diameter.[35] In keeping with this practice, three shell baffle spacings are considered in the calculations. They are the two extreme baffle spacing conditions and one midway between these two extremes. Numerically the baffle spacings are defined:

Baffle Spacing	Calculation
K1	DSI x 0.2
K2	DSI x 0.45
K3	DSI x 1.00

The estimation of the shell-side cross flow area follows a calculation procedure that is based upon the previously cited work of Kern.

If the shell inside diameter is divided by the tube pitch, a fictitious number of tubes which may be assumed to exist in the center of the shell is obtained. In actuality, there is more than likely no central row of tubes but rather a row of tubes on either side of the center with less tubes than is calculated by this procedure. This departure from reality is neglected. For each tube there is considered to be a clearance

$$QTH8 = \frac{3.116875 \times NT8 \times ATIN \times 8.0 \text{ ft/sec}}{8.0 \text{ passes}}$$

These two equations incorporate the assumption that there is an equal amount of tubes in each of the eight tube passes. This is consistent with actual shell and tube exchanger manufacture.[32]

The calculations for the remainder of the construction configurations are quite similar. For 6-pass construction:

$$QTL6 = \frac{3.116875 \times NT6 \times ATIN \times 2.0 \text{ ft/sec}}{6.0 \text{ passes}}$$

$$QTH6 = \frac{3.16875 \times NT6 \times ATIN \times 8.0 \text{ ft/sec}}{6.0 \text{ passes}}$$

For 4-pass construction:

$$QTL4 = \frac{3.116875 \times NT4 \times ATIN \times 2.0 \text{ ft/sec}}{4.0 \text{ passes}}$$

$$QTH4 = \frac{3.116875 \times NT4 \times ATIN \times 8.0 \text{ ft/sec}}{4.0 \text{ passes}}$$

For 2-pass construction:

$$QTL2 = \frac{3.116875 \times NT2 \times ATIN \times 2.0 \text{ ft/sec}}{2.0 \text{ passes}}$$

$$QTH2 = \frac{3.116875 \times NT2 \times ATIN \times 8.0 \text{ ft/sec}}{2.0 \text{ passes}}$$

$$NT6 = ARAV \times F1 \times F6$$

$$NT8 = ARAV \times F1 \times F8$$

E. Available Tube Surface Area

A useful preliminary area calculation is how much heat transfer area is provided per foot of length for a given tube bundle. The external surface area of a tube (ATSU) is fixed once the size is picked, so the area calculations for each of the construction configurations are:

$$AL2 = NT2 \times ATSU$$

$$AL4 = NT4 \times ATSU$$

$$AL6 = NT6 \times ATSU$$

$$AL8 = NT8 \times ATSU$$

F. Maximum and Minimum Tube-Side Flow Rates

In keeping with generally recommended practice the tube-side velocity should be at least 2 ft/sec but not more than 8 ft/sec.[30,31] With this in mind, the maximum and minimum flow rates for 2-, 4-, 6-, and 8-pass construction can be calculated. The tube inside area was fixed with the selection of the size of the tubing. To keep units correct, a conversion factor of 3.116875 is used. This allows for the flow rate to be in gal/min while the velocity is in ft/sec and the area is in square inches. The minimum flow rate for 8-pass construction is:

$$QTL8 = \frac{3.116875 \times NT8 \times ATIN \times 2.0 \text{ ft/sec}}{8.0 \text{ passes}}$$

The maximum flow rate for 8-pass construction is:

$$NT2 = F1 \times F2 \times ARAV$$

where F1 is a correction factor which accounts for the tube layout design. For triangular tube layouts, F1 = 0.9069. F2 is a correction factor which accounts for the loss of tubes due to pass partition dead space in 2-pass construction.

$$F2 = 1.0 - \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

The equations for the number of tubes in a shell with 4-, 6-, or 8-pass constructions are the same as for the 2-pass construction except that the F2 correction factor is changed due to the increased amount of dead space caused by the increase in the necessary flow partitions. The correction factors for these construction types are:

$$F4 = 1.0 - 2.125 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

$$F6 = 1.0 - 3.25 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

$$F8 = 1.0 - 4.0 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

and the corresponding equations for calculating the number of tubes for these construction types are:

$$NT4 = ARAV \times F1 \times F4$$

The number of tube-side passes affects the number of tubes in the shell. In Figure 2, partitions designs were shown. It is clear that no tubes may be placed in the plane of a partition. The thickness (TH) of this partition is dependent upon the size of the shell as follows:

Range of DSI(in)	TH(in)
6.0 to 18.0	0.25
20.0 to 36.0	0.50
38.0 to 60.0	0.75

The larger the number of tube-side passes, the smaller the number of tubes in the shell.

The size of the shell has an obvious part in the determining the number of tubes in the exchanger. The larger the shell diameter, the greater the number of tubes if all other parameters were kept constant. The entire shell diameter is not completely usable however. Structural necessity requires that at least a one-quarter inch free-space at the end of the tube sheet be maintained. Therefore the usable diameter (DSIU) calculation is as follows:

$$DSIU = DSI - 0.5$$

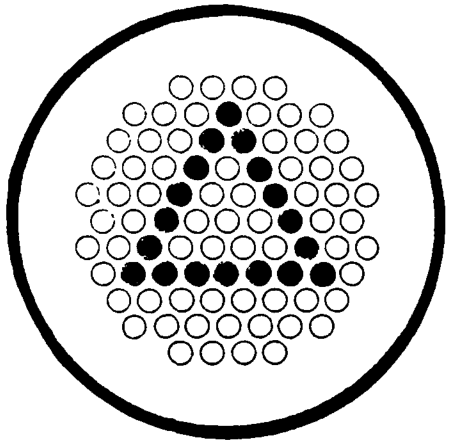
The maximum number of tubes that are possible in a given shell is calculated from:

$$ARAV = \left(\frac{DSIU - DTOT}{TPH} \right)^2$$

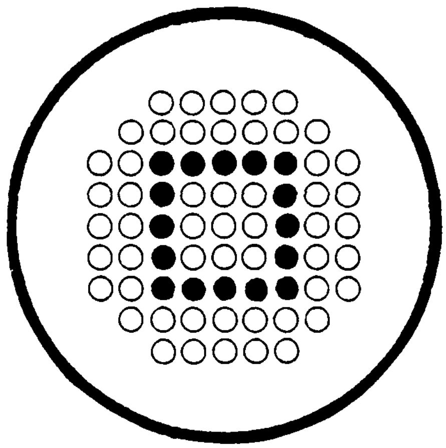
The actual number of tubes that will be found in a given shell with two tube-passes is given by:[29]

FIGURE 7

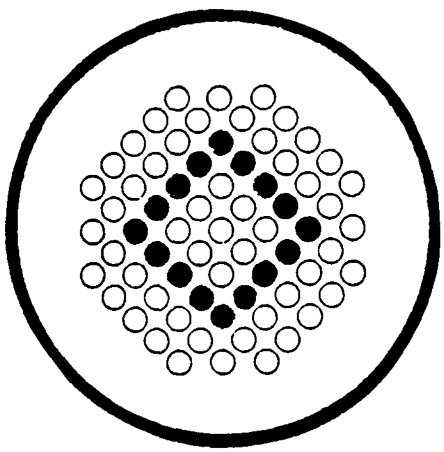
STANDARD TUBE LAYOUT DESIGNS



TRIANGULAR TUBE LAYOUT



SQUARE TUBE LAYOUT



ROTATED SQUARE TUBE LAYOUT

D. Number of Tubes Calculations

The number of tubes possible in a given shell diameter is dependent upon five factors: the tube size, the tube layout design, the tube pitch (TPH), the number of tube-side passes, and the diameter of the shell (DSI). Each of these parameters will be explained and the final equation for calculating the number of tubes will be presented.

The tube size fixes the outside diameter of the tubes. All other parameters equal, it is obvious that more 1.0 inch tubes would fit in a given shell than would 1.5 inch tubes.

The tube layout design is the pattern or tube construction. Three patterns have widespread usage today, they are: the triangular pattern, the square pattern and the oriented square pattern. Each of these patterns are shown in Figure 7. The spacing is clearly different for the triangular and square patterns. This computer design uses calculations for triangular tube layouts only.

The holes in the tubesheet cannot be drilled very closely together, since too thin a separation structurally weakens the tubesheet. The shortest distance between two adjacent tube holes is the clearance which has become standard in practice. The tube pitch is the shortest center-to-center distance between adjacent tubes. This dimension, which depends on the size of the tubing used, is also standard. For triangular tube layouts, the following tube pitches have been designated:

DTOT(in)	TPH(in)
0.75	15/16 or 0.9375
1.00	1 1/4 or 1.25
1.25	1 9/16 or 1.5625
1.50	1 7/8 or 1.875

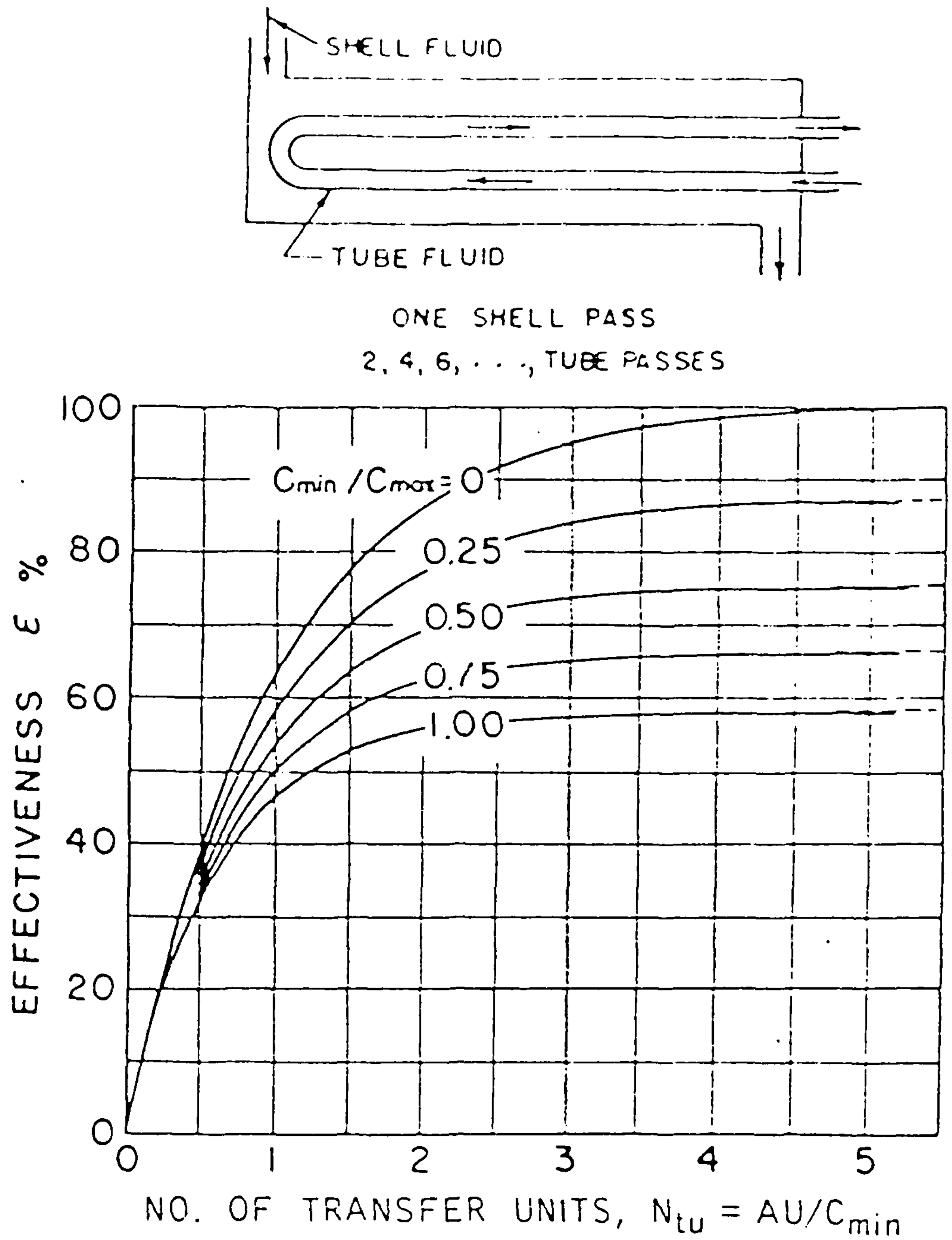
At this point, all the data that the computer program needs to provide a list of shell and tube heat exchanger configurations that will accomplish the required heat transfer is provided except for any restrictions that may need to be imposed.

The site that the heat exchanger will eventually occupy may be fixed, and only a certain space may be available. The length of the heat exchanger would certainly need to be restricted. Moreover, as the heat exchanger would most probably need some sort of inspection or maintenance done to it during its lifetime, it is common practice to require that the maximum length of the heat exchanger be less than one-half the available free-space at the site. This is because maintenance of the tubes requires the removal of the shell and obviously one would need at least twice the length of the heat exchanger to accomplish this task. Therefore, the user is asked to input the maximum allowable length of the heat exchanger to the program.

Another important limitation that normally is imposed on shell and tube heat exchanger designers is the amount of pressure drop that is allowed through the tubes and the shell. Normally, the size of the pumps and corresponding pumping power is fixed, and therefore a limit on the maximum head loss through a unit must be kept within known limits. Even if the pump sizes are not predetermined, reasonable pressure loss restrictions are imposed since the larger the pressure loss allowed, the larger the pumping power required. This capital cost outlay must be considered in the decision for optimum design. The user inputs these maximum allowable pressure losses as prompted by the program.

FIGURE 8

EFFECTIVENESS - NUMBER OF TRANSFER UNITS RELATIONSHIP FOR A 1-2 SHELL AND TUBE HEAT EXCHANGER



have a U-value of between 40-75 BTU/hr-ft^2-°F. These values are for unfouled tubes. Average (mid-range) values for all possible combinations of the seven fluid choices have been incorporated into the computer program as a design check of the actual U-value which is calculated later in the program. The source used for these estimated U-values is Heat Exchangers by Kakac and Bergles. These values are for plain (unfinned) tubes.

Tabulated values of fouling resistances are also common. If the user wants a fouling factor (RFOULT) added to the calculation of the overall U-value, the following equation is used.

$$U = \frac{1}{RFOULT + \frac{1}{UC}}$$

where UC is the U-value for clean (unfouled) tubes. Table 1 lists the overall U-values and fouling resistances used in the program.[41]

N. Determination of Possible Exchanger Configurations

Once the tube-fluid flow rate, the shell-fluid flow rate, the maximum tube length, and the maximum allowable tube-side and shell-side pressure losses have been specified, the computer does a systematic check of every possible heat exchanger configuration to see if it meets the problem specifications. It checks each available shell size (from six to sixty inches), with every possible number of tube-side passes (2, 4, 6, or 8) and every possible baffle spacing (0.2, 0.45, and 1.0), a total of 336 possible exchanger configurations, to see which, if any, meet the problem requirements.

TABLE 1

TYPICAL OVERALL DESIGN COEFFICIENTS FOR

SHELL AND TUBE HEAT EXCHANGERS

FLUID 1	FLUID 2	TOTAL FOULING RESISTANCE $\frac{HR\text{-}FT^2\text{-}°F}{BTU}$	OVERALL U-VALUE $\frac{BTU}{HR\text{-}FT^2 °F}$
Water	Water	0.0015	275.0
Water	Light Organic Liquids	0.0015	150.0
Water	Medium Organic Liquids	0.002	100.0
Water	Heavy Organic Liquids	0.0025	57.5
Light Organic Liquids	Light Organic Liquids	0.002	115.0
Light Organic Liquids	Medium Organic Liquids	0.0025	85.0
Light Organic Liquids	Heavy Organic Liquids	0.003	50.0
Medium Organic Liquids	Medium Organic Liquids	0.003	65.0
Medium Organic Liquids	Heavy Organic Liquids	0.0035	35.0
Heavy Organic Liquids	Heavy Organic Liqids	0.005	20.0

Note: Light Organics include: Methyl Alcohol and Gasoline

Medium Organics include: Dowtherm A and Kerosene

Heavy Organics include: SAE 10 Lubricating Oil and Ethylene Glycol

The first check that is made is to see whether the tube-fluid flow rate falls within the maximum and minimum allowable tube-fluid flow rates for that particular pass configuration. If the tube-fluid flow rate is acceptable, the shell-fluid flow rate is checked to see if it falls within the range of maximum and minimum allowable shell-fluid flow rates for any of the three baffle spacings. Both the tube-fluid and the shell-fluid must fall within their respective acceptance ranges for further checks to continue.

If the tube-fluid and shell-fluid flow rates are acceptable a pressure loss and length check subroutine PRESCK, is called. A listing of PRESCK is found in Appendix D.

This subroutine contains the bulk of the thermal-hydraulic design equations. Each of the major equations and their source will be explained in detail.

The velocities of the tube-fluid and the shell-fluid are calculated. The shell-fluid velocity (VS) is simply the shell-fluid flow rate (MS) divided by the crossflow area of the shell (ASC). The evaluation of the shell crossflow area was previously discussed in the determination of the minimum and maximum shell-side flow rates of the exchanger. The shell-fluid velocity reduces to:

$$VS = \frac{MS}{ASC} \times C$$

where C is a conversion factor (=.3208342) which allows velocity to be in ft/sec while mass flow rate is input in GPM and the area is in in^2.

The tube-side fluid velocity (VT) is simply the tube-fluid flow rate (MT) divided by the total tube-side area per pass. As noted earlier, the number of tubes that will be found in a given heat exchanger will decrease as the number of tube-passes increases. The tube-side velocity equation will therefore differ for each pass-construction type. This is because the tube-side area is equal to the number of tubes multiplied by the internal area of the tubes (ATIN). The tube-side velocity equation is for 2-pass construction is:

$$VT = \frac{C \times MT}{ATIN \times \dfrac{NT2}{2.0}}$$

where C is the same conversion factor that was used in the calculation of the shell-side velocity. The equations for 4-, 6-, or 8-pass construction are the same with the exception of the substitution of the appropriate number of tubes per pass for that particular pass construction.

The mass velocity of the shell-fluid (GS), and the mass velocity of the tube-fluid (GT) can now be found by multiplying the velocities by the density of the particular fluid.

$$GS = VS \times DENS$$

$$GT = VT \times DENT$$

The next step is to calculate the dimensionless Reynolds Number. The calculation of the tube-side Reynolds Number (RET) is straightforward. This is simply the tube-side mass velocity multiplied by the tube inside diameter (DTIN) divided by the tube-fluid dynamic viscosity (DVT).

$$RET = \frac{\frac{DTIN}{12.0} \times GT}{DVT}$$

The tube inside diameter of the tube, which is in inches, is divided by the conversion factor 12.0 to get it into proper units of feet.

The calculation of the shell-side Reynolds Number (RES) is not as direct. There is not a circular flow channel for the shell-fluid. An equivalent diameter must be calculated. The direction of the shell-fluid flow is partially along and partially at right angles to the long axes of the tubes in the tube bundle. The flow area at right angles to the long axes varies with each row of tubes. A hydraulic radius based upon the flow area across any one row could not distinguish between standard tube layout designs. In order to obtain a simple correlation combining both the tube size, the tube pitch (TPH), and their type of layout, excellent agreement is obtained if the hydraulic radius is calculated parallel to instead of perpendicular to the long axes of the tubes. The equivalent diameter for the shell is taken as four times the hydraulic radius obtained for that particular tube layout design. For the triangular tube layout design the shell equivalent diameter (DEQS) is:[42]

$$DEQS = \frac{4 \times \left(\frac{TPH}{2} \times 0.86\ TPH - \frac{\pi\ DTO^2}{8} \right)}{\frac{\pi\ DTO}{2}}$$

where DTO is the outside diameter of the tubes.

The evaluation of the shell-side Reynolds Number is now found by:

$$RES = \frac{\frac{DEQ}{12.0} \; GS}{DVS}$$

where DVS is the dynamic viscosity of the shell-fluid.

Evaluation of the shell-side heat transfer film coefficient (HO) can be done using this equation:[43]

$$\frac{hde}{k} = 0.36 \; (Re)^{0.55} \; (Pr)^{1/3} \; \left(\frac{\mu}{\mu f} \right)^{0.14}$$

using Appendix A nomenclature:

$$HO = \frac{KS}{\frac{DEQS}{12.0}} \; 0.36 \; RES^{0.55} \; \left(\frac{CPS \times DVS \times 3600.0}{KS} \right)^{1/3} \; \left(\frac{DVS}{DVSF} \right)^{0.14}$$

where KS is the thermal conductivity of the shell-fluid, CPS is the specific heat of the shell-fluid and DVSF is the dynamic viscosity of the shell-fluid evaluated at the fluid-tube wall interface temperature. This temperature has been estimated to be the average value of the shell-fluid and tube-fluid temperatures. The number 3600.0 in the Prandtl Number term is conversion factor for seconds per minute.

Evaluation of the tube-side heat transfer film coefficient (HI) can be made using the following equation:[44]

$$\frac{hid}{k} = 0.027 \; (Re)^{0.8} \; (Pr)^{1/3} \; \left(\frac{\mu}{\mu w} \right)^{0.14}$$

using Appendix A nomenclature:

$$HI = \frac{KT}{\frac{DTIN}{12.0}} \; 0.027 \; RET^{0.8} \; \left(\frac{CPT \times DVT \times 3600.0}{KT} \right)^{1/3} \; \left(\frac{DVT}{DVTF} \right)^{0.14}$$

where KT is the thermal conductivity of the tube-fluid, CPT is the specific heat of the tube-fluid and DVTF is the dynamic viscosity of the tube-fluid evaluated at the fluid-tube wall interface temperature.

The tube wall resistance (RW) is evaluated. Although this resistance can be neglected for many metals, it is evaluated for all since the more exotic metals such as zirconium and titanium have large resistances to heat transfer compared to copper and aluminum. TEMA standards handbook provides this equation for tube wall resistance calculations:[45]

$$RW = \frac{d}{24K} \times \left[Ln \left(\frac{d}{d-2t} \right) \right]$$

using Appendix A nomenclature:

$$RW = \frac{DTOT}{24\ KW} \quad Ln \left[\frac{DTOT}{DTOT - 2(DTOT - DTIN)} \right]$$

where KW is the thermal conductivity of the tube metal which was fixed once the choice of the tube metals was made in the program.

All the parameters needed for the determination of the overall heat transfer coefficient (UCALC) have now been determined. Using another equation from the TEMA standards handbook the U-value is calculated by:[46]

$$U = \frac{1}{\frac{1}{ho} + ro + rw + ri \left(\frac{Ao}{Ai} \right) + \frac{1}{hi} \left(\frac{Ao}{Ai} \right)}$$

where ro is the fouling resistance on the outside of the tubes, ri is the fouling resistance on the inside of the tubes and $\left(\frac{Ao}{Ai}\right)$ is the ratio of the outside to inside surface of the tubing. The calculation of the U-value differs depending upon the user's desire to add fouling factors. If the user wants fouling factors added the equation is:

$$UCALC = \frac{1}{\frac{1}{HO} + RO + RW + (RI \times TRATIO) + \left(\frac{1}{HI} \times TRATIO\right)}$$

where TRATIO is the area ratio described above. RO and RI are the fouling resistances on the outside and inside of the tube respectively. They have been computed depending upon the tube-fluid and shell-fluid choices. The total of the RI and RO is equal to the total fouling resistances found in Table 1.

If the user wants the U-value calculated for clean tubes the equation simplifies to:

$$UCALC = \frac{1}{\frac{1}{HO} + RW + \left(\frac{1}{HI} \times TRATIO\right)}$$

The required heat transfer area (ACORR) can now be calculated by:

$$ACORR = \frac{NTU \times CMIN}{UCALC}$$

Using this value of the heat transfer area, the required length of tubing (L) can be calculated depending upon the pass construction. For 2-pass construction:

$$L = \frac{ACORR}{AL2}$$

where AL2 is the tube bundle surface are per foot of tube length for 2-pass construction. The length calculation for 4-, 6-, or 8-pass construction is similar to this equation.

This calculated length must be less than the length restriction imposed by the user. This is the first of three checks that are done in the subroutine PRESCK. All three must be successfully achieved before the exchanger configuration is printed as a possible solution to the heat exchanger problem.

Checks 2 and 3 deal with the maximum allowable pressure losses in the exchanger.

The tube-side pressure loss is calculated using two equations from Kern. *The first equation accounts for the pressure drop in tubes.* This equation is:[47]

$$\Delta Pt = \frac{f \quad Gt^2 \quad L \quad m}{2g \; \rho D \; \left(\frac{\mu}{\mu w} \right)^{0.14}}$$

In Appendix A terminology:

$$PLTF = \frac{FRT \quad GT^2 \quad NP \quad L \quad .006944}{\left(\frac{DTIN}{12.0} \right) \; 2 \; AG \; \left(\frac{DVT}{DVTF} \right)^{0.14} \; DENT}$$

where NP is the number of the tube passes, .006944 is a conversation factor for pounds per square inch, AG is a gravitational acceleration constant and FRT is a tube-side friction factor. This friction factor is calculated by a subroutine called TFRICT. A listing of this subroutine can be found in Appendix E. This subroutine contains data on friction factors correlated by Sieder and Tate for fluids which are being heated or cooled in tubes. The subroutine uses a linear interpolation algorithm to calculate the friction factor depending on the Reynolds Number of the tube-side fluid. The Sieder and Tate correlated data was accepted and published by TEMA for use in pressure loss calculations.

Since the fluid must flow from one pass into the next pass by doing a 180° direction change in the tube channel area of the exchanger, an additional pressure drop is experienced. This loss is referred to as the return loss. Kern proposed this equation to account for this pressure drop:[48]

$$\Delta Pr = \frac{4.0\ n\ V^2}{S\ 2g}$$

in Appendix A nomenclature:

$$PLTR = \frac{4.0\ \ NP\ \ VT^2}{\left(\frac{DENT}{DENW}\right) 2\ \ AG}\ \ .4335$$

where .4335 is a conversion factor from feet of water to pounds per square inch and the ratio ($\frac{DENT}{DENW}$) is the specific gravity of the fluid. Manufacturers of heat exchangers use a modification of this return loss equation. The factor of 4.0 was determined to provide results that were much too conservative. Practice is to use a factor of 2.0 in calculations.[49] The equation used in PRESCK does in fact use this factor of 2.0.

The total tube-side pressure drop is therefore:

$$PLTOT = PLTF + PLTR$$

This total tube-side pressure loss is compared to the maximum allowable tube-side pressure loss imposed by the user. If the tube-side pressure loss is greater than the allowable loss, the exchanger configuration is not deemed acceptable.

The last check is associated with the shell-side pressure loss. The pressure drop in a shell and tube heat exchanger is proportional to the number of times the fluid crosses the tube bundle between tube sheets. It is also proportional to the distance the fluid flows across the tube bundle. The calculation of the pressure loss can be made from:[50]

$$\Delta Ps = \frac{f\ G^2\ Ds\ (N+1)}{2\ g\ \rho\ De\ \left(\frac{\mu}{\mu w}\right)^{0.14}}$$

where N is the number of baffles in the heat exchanger, De is the equivalent diameter of the shell cross flow channel, and f is the shell-side friction factor. This friction factor is calculated by a subroutine called SFRICT. A listing of this subroutine is in Appendix F. Subroutine SFRICT contains data on shell-side friction factors for bundles with 25% cut segmental baffles.[51] Like TFRICT, this subroutine uses a linear interpolation algorithm to calculate the fluid friction factor depending on the Reynolds Number of the shell-side fluid.

The shell-side pressure loss equation using nomenclature from Appendix A takes the form:

$$PLSTOT = \frac{FRS \ L \ GS^2 \ \left(\frac{L \times 12.0}{DSI \times PICKS} + 1 \right) \ .006944}{2 \ AG \ DENS \ DEQS \ \left(\frac{DVS}{DVFS} \right)^{0.14}}$$

where PICKS is the choice of baffle spacing, FRS is the shell-side fluid friction factor calculated in subroutine SFRICT, and .006944 is a conversion factor from pounds per square feet to pounds per square inch. This total shell-side pressure loss is compared to the maximum allowable shell-side pressure loss imposed by the user. If the calculated loss is less than the maximum allowable pressure loss the heat exchanger configuration is regarded as acceptable.

Finally, if the exchanger configuration successfully meets the length restriction, the tube-side pressure loss restriction and the shell-side pressure loss restriction it is considered as an acceptable exchanger configuration that will meet the limitations of the problem input by the user.

O. Output of Possible Exchanger Configurations

All heat exchanger configurations which meet the required heat duty of the problem and are within the length and pressure drop restrictions are output to the user. The output is performed by calling four subroutines, PRINT2, PRINT4, PRINT6, and PRINT8. Each of these subroutines correspond to their particular pass-construction type. A listing of these subroutines are in Appendix G.

If no exchanger configuration of a particular pass-construction meets the problem specifications, a verbal output to that effect is printed.

If no exchanger configuration of any pass-construction category meets the problem specifications, a message is printed which states that the length and pressure restriction cannot be met and the user must use split shell design or multiple shells in series. This computer program is not capable of handling those types of design.

P. User Selection of Possible Exchanger Configurations

Once the program has listed all the possible heat exchanger configurations by pass-construction, the user is asked to make a selection of a heat exchanger configuration. The user is reminded that only the listed configurations will meet the problem specifications.

The user responds to three input requests which pertain to his choice of exchanger configurations. The need only respond to the questions posed by the program. The first input is the shell diameter. The second input is the number of tube-side passes. The final input is the baffle spacing. Once the user specifies the choice of heat exchanger configurations, the program calculates the final output data.

Q. Calculation of Estimated Capital Cost

Many factors enter into how much a shell and tube heat exchanger may cost. The cost of engineering and design, raw materials, and labor are only three of the many parameters that must be explored. Each shell and tube heat exchanger manufacturer has its own method for doing cost calculations for a specific exchanger design.

One would assume that the length of a heat exchanger, as well as the shell diameter would affect the cost. Since material costs play an important role, the number of tubes would also seem to have an effect on

```
THE AVAILABLE HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:


THE AVAILABLE 2-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 14.00000  INCHES:
WITH A  1.000000  BAFFLE SPACING
SHELL DIAMETER 18.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 20.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING


THE AVAILABLE 4-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 18.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 20.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 22.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 24.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 26.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 28.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING


THE AVAILABLE 6-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 22.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 24.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 26.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 28.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 30.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 32.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 34.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING


THE AVAILABLE 8-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:
```

```
     9.0            CHROME MOLY STEEL (2.25%)
    10.0             CHROME MOLY STEEL (5%)
    11.0             CHROME MOLY STEEL (12%)
    12.0              STAINLESS STEEL (18%)
    13.0              STAINLESS STEEL (25%)
    14.0                 ADMIRALTY METAL
    15.0                     COPPER
    16.0                     NICKEL
    17.0                   ZIRCONIUM
    18.0                    TITANIUM

 PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF
 MATERIAL DESIRED.(ie 1.0  6.0  18.0 etc.)

6.0
 DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?

N

 DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?

N

 DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?

N

 DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR
 NOT A FOULING FACTOR IS USED ?

N

 DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?

N

 DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE
 PRESSURE LOSSES ?

N
```

```
NUMBER OF TUBE-SIDE PASSES:  8.0

BAFFLE SPACING:   0.20  OR

                   6.80  INCHES

EXCHANGER TUBE LENGTH:     1.461  FEET

TUBE-SIDE PRESSURE LOSS:    1.34  POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS:    0.18  POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST:  $ 418562.90

 U =  194.6903
 UCALC =  243.1180

 DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
 CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N

***** SINCE COPPER TUBING IS RELATIVELY EXPENSIVE, THE
***** TUBING MATERIAL WILL BE CHANGED TO REGULAR CARBON STEEL
***** TO SEE THE EFFECT ON THE FINAL RESULTS. ALL OTHER
***** PARAMETERS WILL BE LEFT UNCHANGED FROM THE ORIGINAL
***** PROBLEM STATEMENT.

 DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
 TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

Y

 DO YOU WANT TO CHANGE THE TUBING SIZE ?

N

 DO YOU WANT TO CHANGE THE TUBING MATERIAL ?

Y
 WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR
 THE TUBING ?

 CHOICE NUMBER            MATERIAL TYPE

      1.0              ALUMINUM TYPE 1100
      2.0              ALUMINUM TYPE 3003
      3.0              ALUMINUM TYPE 3004
      4.0              ALUMINUM TYPE 6061
      5.0              ALUMINUM TYPE 6063
      6.0                 CARBON STEEL
      7.0              CARBON MOLY STEEL
      8.0            CHROME MOLY STEEL (1%)
```

```
 2.0, 4.0, 6.0, OR 8.0

2.0

 WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

1.0

 THE FINAL CONFIGURATION IS :

SHELL DIAMETER:    14.0  INCHES

NUMBER OF TUBE-SIDE PASSES:  2.0

BAFFLE SPACING:   1.00  OR

                    14.00  INCHES

EXCHANGER TUBE LENGTH:     8.205  FEET

TUBE-SIDE PRESSURE LOSS:    1.59  POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS:    3.54  POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST:  $  46197.14

 U =  194.6903
 UCALC =  271.4765

 DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
 CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

Y
 WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 28.0 etc.

***** THE LARGEST CONFIGURATION WILL BE EXAMINED HERE.

34.0

 WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
 2.0, 4.0, 6.0, OR 8.0

8.0

 WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

0.2

 THE FINAL CONFIGURATION IS :

SHELL DIAMETER:    34.0  INCHES
```

```
SHELL DIAMETER 26.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 28.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING


THE AVAILABLE 6-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 22.00000  INCHES:
WITH A  0.4500000  BAFFLE SPACING
SHELL DIAMETER 24.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 26.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 28.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 30.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 32.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 34.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING


THE AVAILABLE 8-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 26.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 28.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 30.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 32.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING
SHELL DIAMETER 34.00000  INCHES:
WITH A  0.2000000  BAFFLE SPACING


THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
OF THE ABOVE LISTED HEAT EXCHANGER
CONFIGURATIONS. REMEMBER ONLY THESE LISTED
CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU
HAVE INPUT INTO THE PROGRAM

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
14.0, 26.0 etc.

***** THE SMALLEST CONFIGURATION WILL BE EXAMINED HERE.

14.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
```

```
 TRANSFER COEFFICIENT GIVEN FOR CLEAN TUBES ?
 ANSWER Y FOR YES AND N FOR NO.

Y

 WHAT IS THE MAXIMUM LENGTH THAT THE HEAT
 EXCHANGER TUBES CAN BE FOR THIS APPLICATION ?
 REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR
 CLEANING SO THE MAXIMUM LENGTH OF THE HEAT
 EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE
 LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL
 BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN
 FEET THAT THE HEAT EXCHANGER TUBES CAN BE.
 ( ie 5.0 , 7.5, 19.6 etc.)

10.0

 WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN
 THE TUBES IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)

5.0

 WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN
 THE SHELL IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)

5.0


 THE AVAILABLE HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:


 THE AVAILABLE 2-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

 SHELL DIAMETER 14.00000  INCHES:
 WITH A  1.000000  BAFFLE SPACING
 SHELL DIAMETER 18.00000  INCHES:
 WITH A  0.4500000  BAFFLE SPACING
 SHELL DIAMETER 20.00000  INCHES:
 WITH A  0.4500000  BAFFLE SPACING


 THE AVAILABLE 4-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

 SHELL DIAMETER 18.00000  INCHES:
 WITH A  0.4500000  BAFFLE SPACING
 SHELL DIAMETER 20.00000  INCHES:
 WITH A  0.4500000  BAFFLE SPACING
 SHELL DIAMETER 22.00000  INCHES:
 WITH A  0.4500000  BAFFLE SPACING
 SHELL DIAMETER 24.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING
```

```
    5.0                  KEROSENE
    6.0           SAE 10 LUBRICATING OIL
    7.0                ETHYLENE GLYCOL

 PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED
 TUBE FLUID. (ie 1.0 IF THE FLUID IS WATER)

1.0

 WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN
 THE SHELL OF THE EXCHANGER? THE FOLLOWING
 FLUID CHOICES ARE AVAILABLE:

 CHOICE NUMBER           FLUID NAME

    1.0                    WATER
    2.0                METHYL ALCOHOL
    3.0                  GASOLINE
    4.0                 DOWTHERM A
    5.0                  KEROSENE
    6.0           SAE 10 LUBRICATING OIL
    7.0                ETHYLENE GLYCOL

 PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED
 SHELL FLUID. (ie 1.0 IF THE FLUID IS WATER)

1.0

 WHAT IS THE ENTERING TEMPERATURE OF THE TUBE
 FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)

160.0

 WHAT IS THE EXIT TEMPERATURE OF THE TUBE FLUID
 IN DEGREES F ?

180.0

 WHAT IS THE FLOW RATE OF THE TUBE FLUID IN
 GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)

400.0

 WHAT IS THE ENTERING TEMPERATURE OF THE SHELL
 FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)

240.0

 WHAT IS THE FLOW RATE OF THE SHELL FLUID IN
 GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)

200.0

 DO YOU WANT TO ADD A FOULING FACTOR TO THE HEAT
```

29.0	1.25	16
30.0	1.25	17
31.0	1.25	18
32.0	1.25	20
33.0	1.5	10
34.0	1.5	12
35.0	1.5	14
36.0	1.5	16

PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF TUBING DESIRED.(ie 1.0 6.0 34.0 etc.)

9.0

WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR THE TUBING ?

CHOICE NUMBER	MATERIAL TYPE
1.0	ALUMINUM TYPE 1100
2.0	ALUMINUM TYPE 3003
3.0	ALUMINUM TYPE 3004
4.0	ALUMINUM TYPE 6061
5.0	ALUMINUM TYPE 6063
6.0	CARBON STEEL
7.0	CARBON MOLY STEEL
8.0	CHROME MOLY STEEL (1%)
9.0	CHROME MOLY STEEL (2.25%)
10.0	CHROME MOLY STEEL (5%)
11.0	CHROME MOLY STEEL (12%)
12.0	STAINLESS STEEL (18%)
13.0	STAINLESS STEEL (25%)
14.0	ADMIRALTY METAL
15.0	COPPER
16.0	NICKEL
17.0	ZIRCONIUM
18.0	TITANIUM

PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF MATERIAL DESIRED.(ie 1.0 6.0 18.0 etc.)

15.0

WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN THE TUBES OF THE EXCHANGER? THE FOLLOWING FLUID CHOICES ARE AVAILABLE:

CHOICE NUMBER	FLUID NAME
1.0	WATER
2.0	METHYL ALCOHOL
3.0	GASOLINE
4.0	DOWTHERM A

```
WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.
Y
    THE MAJOR SOURCES OF INFORMATION USED TO
FORMULATE THIS PROGRAM ARE:

BREESE,J. - SHELL AND TUBE HEAT EXCHANGERS
KAYS & LONDON - COMPACT HEAT EXCHANGERS
KERN,D. - PROCESS HEAT TRANSFER
HOLMAN,J.P. - HEAT TRANSFER
CHASE,T.A. - PERSONAL COMMUNICATION
FANARITIS,J.P. - HEAT EXCHANGER TECHNOLOGY
GAGGIOLI,R.A. - EFFICIENCY AND COSTING
PATTERSON-KELLEY CO. - HEAT EHCHANGERS
TEMA - STANDARDS OF TUBULAR EXCHANGERS

WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.
Y
    WHAT SIZE OF TUBING DO YOU WANT TO USE IN
YOUR HEAT EXCHANGER? (REMEMBER THE SMALLER THE
BWG NUMBER THE THICKER THE WALL OF THE TUBING)

CHOICE NUMBER  OUTSIDE DIAMETER(IN)    BWG #

     1.0              0.75              10
     2.0              0.75              11
     3.0              0.75              12
     4.0              0.75              13
     5.0              0.75              14
     6.0              0.75              15
     7.0              0.75              16
     8.0              0.75              17
     9.0              0.75              18
    10.0              0.75              19
    11.0              0.75              20

    12.0              1.0                8
    13.0              1.0               10
    14.0              1.0               11
    15.0              1.0               12
    16.0              1.0               13
    17.0              1.0               14
    18.0              1.0               15
    19.0              1.0               16
    20.0              1.0               18
    21.0              1.0               20

    22.0              1.25               7
    23.0              1.25               8
    24.0              1.25              10
    25.0              1.25              11
    26.0              1.25              12
    27.0              1.25              13
    28.0              1.25              14
```

IX. COMPUTER RESULTS

***** THIS IS A COPY OF THE ACTUAL RUN OF THE EXAMPLE
***** PROBLEM. THE ONLY ADDITIONS MADE ARE THE
***** COMMENTS INSERTED FOR CLARITY WHICH ARE PRECEEDED
***** WITH FIVE STARS.

```
$ SHELL
    THIS PROGRAM CALCULATES THE REQUIRED LENGTH,
 SIZE OF SHELL, BAFFLE SPACING, NUMBER OF TUBE
 PASSES, AND ESTIMATED CAPITAL COST OF A TEMA
 TYPE E SHELL AND TUBE HEAT EXCHANGER WITH 2, 4,
 6, OR 8 TUBE PASSES. YHIS PROGRAM ALSO PROVIDES
 CALCULATIONS OF THE SHELL-SIDE AND TUBE-SIDE
 PRESSURE DROPS. THE USER PICKS THE SIZE OF THE
 TUBING TO BE USED ALONG WITH THE TUBE MATERIAL.
 THE USER SETS THE MAXIMUM LENGTH ALONG WITH THE
 MAXIMUM ALLOWABLE PRESSURE DROPS ON THE SHELL
 AND TUBE SIDES.

 THE PROGRAM OUTPUTS THE EXCHANGER CONFIGURATIONS
 THAT WILL MEET THE PROBLEM LIMITATIONS. FINAL
 DESIGN DATA ON ANY CONFIGURATION IS PROVIDED AS
 REQUESTED. ALTERNATE EXCHANGER CONFIGURATIONS
 CAN BE COMPARED, AND ORIGINAL INPUT DATA CAN BE
 CHANGED TO DETERMINE THEIR EFFECTS ON THE FINAL
 DESIGN DATA.

    THIS PROGRAM IS LIMITED TO LIQUID-TO-LIQUID
 APPLICATIONS UNDER LOW PRESSURE CONDITIONS (<300 PSI).

     A LIMITED NUMBER OF COMMON FLUIDS
 ARE AVAILABLE TO BE CHOSEN AS FLUIDS IN THE
 TUBE AND SHELL. THIS PROGRAM CAN BE EASILY
 MODIFIED TO ACCEPT ADDITIONAL FLUIDS FOR WHICH
 DENSITY, VISCOSITY, THERMAL CONDUCTIVITY AND SPECIFIC
 HEAT DATA ARE AVAILABLE.

    THE USER MUST INPUT THE FOLLOWING DATA IN
 THE UNITS SPECIFIED:

       INPUT DATA                    UNITS

 TUBE FLUID ENTRANCE TEMP          DEGREES F
 TUBE FLUID EXIT TEMP              DEGREES F
 TUBE FLUID FLOW RATE          GALLONS PER MINUTE
 SHELL FLUID ENTRANCE TEMP         DEGREES F
 SHELL FLUID FLOW RATE         GALLONS PER MINUTE
 MAXIMUM TUBE LENGTH                 FEET
 MAX TUBE PRESSURE LOSS       POUNDS PER SQUARE INCH
 MAX SHELL PRESSURE LOSS      POUNDS PER SQUARE INCH
```

VIII. SAMPLE PROBLEM

In order to picture just what this program actually accomplishes, a sample problem will be examined. The example comes from an article listed as Reference 30.

Problem Conditions:

Tube fluid: water	Shell fluid: water
Tube entrance temperature: 160°F	Shell entrance temperature: 240°F
Tube exit temperature: 180°F	
Tube fluid flow rate: 400 GPM	Shell fluid flow rate: 300 GPM

Fouling factor will be added

Maximum allowable tube length: 10 feet

Tube outside diameter: 0.75 in

Tube pitch: 15/16 in

Tube layout design: equilateral triangle

Tube thickness: 18 BWG

Tube material: copper

No pressure loss limitations were mentioned in this example. In order to make this example meet the program's required inputs the following additional limitations will be imposed:

Maximum allowable tube-side pressure loss: 5.0 psi

Maximum allowable shell-side pressure loss: 5.0 psi

The actual inputs to the computer program and the results obtained are found in Chapter IX.

Baffle spacing

Exchanger tube length

Tube-side pressure loss

Shell-side pressure loss

Estimated capital cost

Each parameter is printed with its corresponding units. The user may request final design data on any or all of the listed exchanger configurations that meet the problem specifications.

S. User Directed Changes to Input Parameters

The user has the option of changing any or all of the input parameters. The program will query whether the user wants to change any of the initial inputs to the program. If the user responds affirmatively, the program responds with eleven questions representing all the inputs to the program. For example, the first question is, "Do you want to change the tubing size?" if the user responds "yes," the available tubing sizes will be reprinted on the screen and the user will be asked to make a choice of the size of tubing desired. The user may change all inputs or only selective inputs to the program. The parameters that are not changed will retain the value set by the initial problem input.

This aspect of the program is useful in that it enables the user to see how changes of tube size, fluid choices or temperature effect the list of possible heat exchanger configurations and the final design data.

Only the last of the preceeding conditions would impose a constraint on the flexibility of the program. There are eighteen choices of materials available to the user of the program, only one of which is 316 stainless steel. The solution to this problem was to assign cost factors to each of the available materials. Stainless steel (25%) was given a cost factor of 1.0 corresponding to 316 stainless steel. All other materials were given a cost factor higher or lower depending on cost of that particular material taken from a mechanical design text.[53]

Since this computer-aided design program only uses single shell pass construction with no additional shells in series or parallel, the cost equation, in Appendix A terminology reduces to:

For 2-pass construction:

$$Z = CF \quad 1150 \quad \left(\frac{DSI}{12.0}\right)^{1.05} L^{0.3} \left(\frac{TPH}{12.0}\right)^{0.75} NT2^{0.975}$$

where CF is the cost factor mentioned previously. The equations for 4-, 6-, or 8-pass constructions are similar.

It should be noted that this cost estimate is crude at best. It does however, give the user another quantitative factor from which his choice of optimum heat exchanger configurations can be made.

R. Final Design Output

The user is provided the following data for each heat exchanger configuration he chooses to meet the problem specifications:

Shell diameter

Number of the tube-side passes

the overall cost. One equation which gives an estimate of the capital cost of a shell and tube heat exchanger in terms of several geometric variables is:[52]

$$Z = 1150 \quad D^{1.05} \quad L^{0.3} \quad P^{0.75} \quad NNi^{0.975} \quad Ne^{0.1} \quad Ns \; Np$$

where

Z is cost in dollars

D is outside tube diameter (ft)

L is tube length (ft)

P is tube pitch (ft)

N is number of tubes per pass

Ni is number of tube passes

Ne is number of shell passes

Ns is number of shells in series

Np is number of shells in parallel

The use of this equation is limited to the following conditions:

- Length between three and thirty feet
- Tube diameters between 3/8 and 1.5 inches
- Tube pitch between 1.25 and 1.5 of the tube diameter
- Heat transfer area between 250 and 5000 square feet
- Fixed tube sheet construction
- Operating pressures less than 300 psi
- Material 316 stainless steel

```
 SHELL DIAMETER 26.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING
 SHELL DIAMETER 28.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING
 SHELL DIAMETER 30.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING
 SHELL DIAMETER 32.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING
 SHELL DIAMETER 34.00000  INCHES:
 WITH A  0.2000000  BAFFLE SPACING

***** NOTE THAT THE LIST OF AVAILABLE CONFIGURATIONS OF
***** HEAT EXCHANGERS IS NOT EFFECTED BY THE CHANGE IN
***** MATERIAL SELECTION. THE ESTIMATED COSTS WILL
***** BE SIGNIFICANTLY CHANGED HOWEVER.


 THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
 OF THE ABOVE LISTED HEAT EXCHANGER
 CONFIGURATIONS. REMEMBER ONLY THESE LISTED
 CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU
 HAVE INPUT INTO THE PROGRAM

 WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 28.0 etc.

***** THE SMALLEST CONFIGURATION WILL BE EXAMINED HERE.

14.0

 WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
 2.0, 4.0, 6.0, OR 8.0

2.0

 WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

1.0

 THE FINAL CONFIGURATION IS :

SHELL DIAMETER:    14.0  INCHES

NUMBER OF TUBE-SIDE PASSES:  2.0

BAFFLE SPACING:   1.00  OR

                   14.00  INCHES

EXCHANGER TUBE LENGTH:     8.814  FEET

TUBE-SIDE PRESSURE LOSS:    1.66  POUNDS PER SQUARE INCH
```

```
SHELL-SIDE PRESSURE LOSS:    3.80  POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST:  $   4285.43

 U =  194.6903
 UCALC =  252.7155

 DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
 CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

Y
 WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 28.0 etc.

***** THE LARGEST CONFIGURATION WILL BE EXAMINED HERE.

34.0

 WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
 2.0, 4.0, 6.0, OR 8.0

8.0

 WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

0.2

 THE FINAL CONFIGURATION IS :

SHELL DIAMETER:    34.0  INCHES

NUMBER OF TUBE-SIDE PASSES:  8.0

BAFFLE SPACING:   0.20  OR

                    6.80  INCHES

EXCHANGER TUBE LENGTH:     1.558  FEET

TUBE-SIDE PRESSURE LOSS:    1.36  POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS:    0.20  POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST:  $  38743.28

 U =  194.6903
 UCALC =  227.9624

 DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
 CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N
```

```
***** A DEMONSTRATION OF HOW A CHANGE IN THE LENGTH
***** RESTRICTION EFFECTS THE FINAL RESULTS WILL BE
***** MADE BY CHANGING THE MAXIMUM LENGTH TO 2.0 FEET.

 DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
 TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

Y

 DO YOU WANT TO CHANGE THE TUBING SIZE ?

N

 DO YOU WANT TO CHANGE THE TUBING MATERIAL ?

N

 DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?

N

 DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?

N

 DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?

N

 DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?

N

 DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR
 NOT A FOULING FACTOR IS USED ?

N

 DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?

Y

 DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE
 PRESSURE LOSSES ?

N
```

WHAT IS THE MAXIMUM LENGTH THAT THE HEAT
EXCHANGER TUBES CAN BE FOR THIS APPLICATION ?
REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR
CLEANING SO THE MAXIMUM LENGTH OF THE HEAT
EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE
LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL
BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN
FEET THAT THE HEAT EXCHANGER TUBES CAN BE.
(ie 5.0 , 7.5, 19.6 etc.)

2.0

NO 2-PASS HEAT EXCHANGER CONFIGURATIONS
WILL MEET THE PROBLEM SPECIFICATIONS.

NO 4-PASS HEAT EXCHANGER CONFIGURATIONS
WILL MEET THE PROBLEM SPECIFICATIONS.

THE AVAILABLE HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

THE AVAILABLE 6-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 30.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING
SHELL DIAMETER 32.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING
SHELL DIAMETER 34.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING

THE AVAILABLE 8-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 30.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING
SHELL DIAMETER 32.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING
SHELL DIAMETER 34.00000 INCHES:
WITH A 0.2000000 BAFFLE SPACING

***** NOTE THAT THE LIST OF AVAILABLE EXCHANGER
***** CONFIGURATIONS IS GREATLY REDUCED.

THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
OF THE ABOVE LISTED HEAT EXCHANGER
CONFIGURATIONS. REMEMBER ONLY THESE LISTED
CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU

```
 HAVE INPUT INTO THE PROGRAM

 WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 28.0 etc.

30.0

 WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
 2.0, 4.0, 6.0, OR 8.0

6.0

 WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

0.2

 THE FINAL CONFIGURATION IS :

SHELL DIAMETER:    30.0  INCHES

NUMBER OF TUBE-SIDE PASSES:  6.0

BAFFLE SPACING:   0.20  OR

                    6.00  INCHES

EXCHANGER TUBE LENGTH:     1.929  FEET

TUBE-SIDE PRESSURE LOSS:    1.01  POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS:    0.48  POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST:  $  28282.48

 U =  194.6903
 UCALC =  237.3057

 DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
 CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N

 DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
 TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

N

 DO YOU WANT TO DO ANY OTHER HEAT EXCHANGER
 PROBLEMS ? TYPE Y FOR YES AND N FOR NO.
N
Fortran STOP DONE
```

X. CONCLUSION

A. Discussion

The major intent of this thesis was to create a computer program which would provide the user with quick preliminary sizes of shell and tube heat exchangers which will accomplish the required heat duty while meeting the length and pressure drop limitations imposed. This program was developed with information and equations that are publicly available. No proprietary code was used. No attempt has been made to compete with the more sophisticated software of proprietary organizations such as HTRI and Heat Transfer and Fluid Flow Services (HTFS). It is the shell and tube exchanger manufacturer's job to squeeze every last BTU out of a given heat exchanger design and to use every last tenth of a PSI to shorten the length of an exchanger. Their reason is obvious; to give the buyer the cheapest, and most effective heat exchanger they can. While the program that has been developed will not provide the user the information needed to completely build a shell and tube heat exchanger, it does provide with reasonable accuracy the key information that one needs to know: the exchanger length, the exchanger shell diameter, the shell-side pressure loss, the tube-side pressure loss and the estimated capital cost. Armed with this information, one can seek out a manufacturer that will build the heat exchanger to specifications.

The program developed is very user-friendly. The user can continually change input parameters to see how the changes affect the final design options provided. This aspect gives the program some educational significance. The user can see what happens when a more viscous fluid is exchanged for a less viscous one. The addition or deletion of the fouling factor from the design input can also give the user some insight into just what a scaling deposit does to the heat transfer in an exchanger.

The program does indeed accomplish what it was expected to do. The time-intensive and laborious calculations needed to design a single heat exchanger have been organized into a compact program which provides nearly instantaneous results to the user.

B. Suggestions for Improvement

This program is suited to further expansion and revision. The program is limited to fixed tube sheet type "E" shells, with triangular tube pitch layout under single-phase liquid-to-liquid low pressure operations. Any reductions in these limiting features would greatly enhance the program capabilities. There is sufficient experimental data, equations, and commercial usage data available to widen the scope of the programs uses. The program could be expanded to handle TEMA type "F", "G", "H", "J", or "X" shells. Square and rotated square tube pitch layouts could be incorporated. Multiple-phase design can be added as well as gas-to-gas and gas-to-liquid operations. High pressure operations could be included.

The program can be expanded to also do more of the mechanical design. All of the integral parts of the shell and tube heat exchanger could be designed: the thickness of the tubesheets, the thickness of the shell, the size and type of gaskets, the size of the end covers and the number, type and size of bolts needed to name just a few.

Other improvements may be to increase the selection of fluids available and to allow for the evaluation of parallel and series heat exchanger configurations.

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APPENDIX A

VARIABLE LISTING

This is an alphabetical display of the variables used in the main computer program. Their explanation and units are provided for quick reference and numerical analysis checks.

Variable Name	Explanation	Units
ACALC	Answer designation to questions asked in the program.	-
ACORR	Required heat transfer area (calculated).	FT^2
AFFAC	Answer designation to change the fouling factor choice.	-
AFOUL	Answer designation to fouling factor question asked in the program.	-
AG	Gravitational acceleration constant = 32.1740.	FT/SEC^2
AGAIN	Answer designation to changing input parameters asked in the program.	-
AL2(I)	An array of tube bundle surface areas per foot of tube length for 2-pass construction.	FT^2/FT
AL4(I)	An array of tube bundle surface areas per foot of tube length for 4-pass construction.	FT^2/FT

AD-A153 779 COMPUTER-AIDED THERMOHYDRAULIC DESIGN OF TEMA TYPE E SHELL AND TUBE HEAT..(U) ARMY MILITARY PERSONNEL CENTER ALEXANDRIA VA N J KOLAR 01 APR 85 2/2

UNCLASSIFIED F/G 13/1 NL

END

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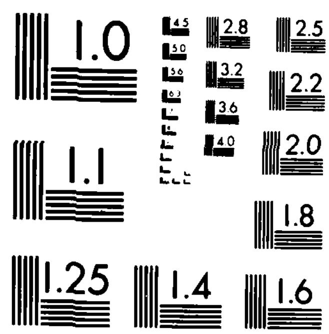

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963 A

Variable Name	Explanation	Units
AL6(I)	An array of tube bundle surface areas per foot of tube length for 6-pass construction.	FT^2/FT
AL8(I)	An array of tube bundle surface areas per foot of tube length for 8-pass construction.	FT^2/FT
ALENTH	Answer designation to length restriction question asked in the program.	-
ALREST	Answer designation to change the length restriction.	-
AMATE	Answer designation to change the tube material.	-
AMAXSP	Maximum allowable shell-side pressure loss.	LB/IN^2
AMAXTP	Maximum allowable tube-side pressure loss.	LB/IN^2
APRES	Answer designation of change the allowable pressure drops.	-
ARAV	Maximum number of tubes possible depending on available area.	-
ASC	Crossflow area of shell	IN^2

Variable Name	Explanation	Units
ASFLU	Answer designation to change the shell fluid.	-
ASGPM	Answer designation to change the shell-side flow rate.	-
ASTEM	Answer designation to change the shell-side temperature.	-
ATFLU	Answer designation to change the tube fluid.	-
ATGPM	Answer designation to change the tube-side flow rate.	-
ATI(I)	An array of available internal areas of tubes.	IN^2
ATIN	Choice of internal area of tube.	IN^2
ATS(I)	An array of available external surface areas of tubes per foot of tube length.	FT^2/FT
ATSU	Choice of external surface area of tube per foot of tube length.	FT^2/FT
ATTEM	Answer designation to change the tube-side temperatures.	-
ATUBE	Answer designation to change the tube size.	-

Variable Name	Explanation	Units
C	Constant = .3208342 (conversion factor).	-
CASE	Number representing whether the tube fluid is being heated or cooled.	-
CF	Cost factor of the tube material choice.	-
CFM(I)	An array of cost factors depending on material type.	-
CMAX	Capacity rate of the maximum fluid.	BTU/HR-°F
CMIN	Capacity rate of the minimum fluid.	BTU/HR-°F
CP	Specific heat at constant pressure value calculated in subroutine.	BTU/LBM-°F
CPS	Specific heat at constant pressure of the shell fluid.	BTU/LBM-°F
CPT	Specific heat at constant pressure of the tube fluid.	BTU/LBM-°F
CS	Capacity rate of shell fluid.	BTU/HR-°F
CT	Capacity rate of tube fluid.	BTU/HR-°F
DEN	Density value calculated in subroutine	LBM/FT^3
DENS	Density of shell fluid.	LBM/FT^3
DENT	Density of tube fluid.	LBM/FT^3

Variable Name	Explanation	Units
DENW	Density of water for specific gravity calculations.	LBM/FT^3
DEQS	Equivalent diameter of the fluid conduit crossing the tubes.	IN
DSI(I)	An array of available inside diameters of shells.	IN
DSIU(I)	An array of available usable diameters of shells.	IN
DTI(I)	An array of available inside diameters of tubes.	IN
DTIN	Choice of inside tube diameter.	IN
DTO(I)	An array of available outside diameters of tubes.	IN
DTOT	Choice of outside tube diameter.	IN
DV	Dynamic viscosity value calculated in subroutine.	LB/FT-SEC
DVS	Dynamic viscosity of shell fluid.	LB/FT-SEC
DVFS	Dynamic viscosity of shell fluid at at wall temperature.	LB/FT-SEC
DVT	Dynamic viscosity of tube fluid.	LB/FT-SEC

Variable Name	Explanation	Units
DVTF	Dynamic viscosity of tube fluid at wall temperature.	LB/FT-SEC
EFF	Heat transfer effectiveness.	-
F1	Constant = 0.9069, which accounts for triangular tube arrangement in calculating the number of tubes possible for a given shell diameter.	-
F2	Factor used in calculating the number of tubes possible for a given shell diameter for 2-pass construction.	-
F4	Factor used in calculating the number of tubes possible for a given shell diameter for 4-pass construction.	-
F6	Factor used in calculating the number of tubes possible for a given shell diameter for 6-pass construction.	-
F8	Factor used in calculating the number of tubes possible for a given shell diameter for 8-pass construction.	-
FRS	Shell-side friction factor.	-
FRT	Tube-side friction factor.	-

Variable Name	Explanation	Units
GS	Mass velocity of the shell fluid.	$LB/SEC\text{-}FT^2$
GT	Mass velocity of the tube fluid.	$LB/SEC\text{-}FT^2$
HI	Heat transfer film coefficient inside the tubes.	$BTU/HR\text{-}°F\text{-}FT^2$
HO	Heat transfer film coefficient outside the tubes.	$BTU/HR\text{-}°F\text{-}FT^2$
I	Counter	-
J	Counter	-
J2	Counter	-
J4	Counter	-
J6	Counter	-
J8	Counter	-
K	Thermal conductivity value calculated in subroutine.	$\frac{BTU}{HR\text{-}FT^2\text{-}°F}$
K1	Thermal conductivity interim value used in subroutines.	$\frac{BTU}{HR\text{-}FT^2\text{-}°F}$
K2	Counter	-
K4	Counter	-

Variable Name	Explanation	Units
K6	Counter	-
K8	Counter	-
KM(I)	An array of thermal conductivities of metals.	$\frac{\text{BTU-FT}}{\text{HR-FT}^2\text{-°F}}$
KS	Thermal conductivity of shell fluid.	-
KT	Thermal conductivity of tube fluid.	$\frac{\text{BTU}}{\text{HR-FT-°F}}$
KW	Choice of thermal conductivity of the tube metal.	$\frac{\text{BTU-FT}}{\text{HR-FT}^2\text{-°F}}$
L	Length of tubing of chosen heat exchanger configuration.	FT
LRESTR	Length restriction of the heat exchanger tubes.	FT
M1	Counter	-
M2	Counter	-
M3	Counter	-
MS	Shell-side fluid flow rate.	GPM
MT	Tube-side fluid flow rate.	GPM

Variable Name	Explanation	Units
N	Integer designation of shell diameter choice.	-
NP	Number designation of number of tube passes.	-
NT2(I)	An array of tube counts for 2-pass construction.	-
NT4(I)	An array of tube counts for 4-pass construction.	-
NT6(I)	An array of tube counts for 6-pass construction.	-
NT8(I)	An array of tube counts for 8-pass construction.	-
NTU	Number of heat transfer units.	-
OKFIN	Counter representing final OK to print exchanger configuration.	-
PI	Constant = 3.1415927	-
PICKD	Number designating the choice of shell diameter.	IN
PICKFS	Number designating the choice of fluid in the shell.	-

Variable Name	Explanation	Units
PICKFT	Number designating the choice of fluid in the tubes.	-
PICKM	Number designating the choice of metal of the tubing.	-
PICKP	Number designating the choice of number of tube passes.	-
PICKS	Number designating the choice of baffle spacing.	-
PICKT	Number designating the choice of tubing size.	-
PLSTOT	Total shell-side pressure loss.	LB/IN^2
PLTF	Tube-side pressure loss due to friction losses.	LB/IN^2
PLTR	Tube-side pressure loss due to return losses.	LB/IN^2
PLTTOT	Total tube-side pressure loss.	LB/IN^2
Q	Heat transfer rate	BTU/HR
QSK1H(I)	An array of maximum shell-side flow rates for .2 baffle spacing construction.	GPM

Variable Name	Explanation	Units
QSK1L(I)	An array of minimum shell-side flow rates for .2 baffle spacing construction.	GPM
QSK2H(I)	An array of maximum shell-side flow rates for .45 baffle spacing construction.	GPM
QSK2L(I)	An array of minimum shell-side flow rates for .45 baffle spacing construction.	GPM
QSK3H(I)	An array of maximum shell-side flow rates for 1.0 baffle spacing construction.	GPM
QSK3L(I)	An array of minimum shell-side flow rates for 1.0 baffle spacing construction.	GPM
QTH2(I)	An array of maximum tube-side flow rates for 2-pass construction.	GPM
QTH4(I)	An array of maximum tube-side flow rates for 4-pass construction.	GPM
QTH6(I)	An array of maximum tube-side flow rates for 6-pass construction.	GPM
QTH8(I)	An array of maximum tube-side flow rates for 8-pass construction.	GPM
QTL2(I)	An array of minimum tube-side flow rates for a 2-pass construction.	GPM

Variable Name	Explanation	Units
QTL4(I)	An array of minimum tube-side flow rates for a 4-pass construction.	GPM
QTL6(I)	An array of minimum tube-side flow rates for a 6-pass construction.	GPM
QTL8(I)	An array of minimum tube-side flow rates for 8-pass construction.	GPM
RATIO	Ratio of the minimum capacity fluid to the maximum capacity fluid.	-
RES	Reynolds number of the shell fluid.	-
RET	Reynolds number of the tube fluid.	-
RFOULT	Total fouling resistance in the heat exchanger.	$\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$
RI	Fouling resistance on inside of tubes.	$\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$
RO	Fouling resistance on outside of tubes.	$\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$
RW	Resistance of tube wall.	$\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$
SF(I)	An array of shell-side friction factors.	-

Variable Name	Explanation	Units
SRE(I)	An array of shell-side fluid Reynolds numbers.	-
T1MAX	Entrance temperature of the maximum fluid.	°F
T1MIN	Entrance temperature of the minimum fluid.	°F
T2MAX	Exit temperature of the maximum fluid.	°F
T2MIN	Exit temperature of the minimum fluid.	°F
TAVE	Average temperature value used in subroutine calculations.	°F
TC	Tube clearance.	IN
TF(I)	An array of tube-side friction factors	-
TH	Thickness of the partition wall for multiple pass construction.	IN
P(I)	An array of available tube pitches of tubes.	IN
TPH	Choice of tube pitch.	IN
TRATIO	Ratio of outside to inside surface of tubing.	-
TRE(I)	An array of tube-side fluid Reynolds numbers.	-

```
      DO 16, I=1,28

        QSK1H(I) = 3.116875*.20*DSI(I)*DSIU(I)*(TC/TPH)*4.5

16    CONTINUE

***   CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF .45*SHELL DIAMETER (K2) BASED ON A 2 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 17, I=1,28

        QSK2L(I) = 3.116875*.45*DSI(I)*DSIU(I)*(TC/TPH)*2.0

17    CONTINUE

***   CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF .45*SHELL DIAMETER (K2) BASED ON A 4.5 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 18, I=1,28

        QSK2H(I) = 3.116875*.45*DSI(I)*DSIU(I)*(TC/TPH)*4.5

18    CONTINUE

***   CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF 1.0*SHELL DIAMETER (K3) BASED ON A 2 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 19, I=1,28

        QSK3L(I) = 3.116875*1.0*DSI(I)*DSIU(I)*(TC/TPH)*2.0

19    CONTINUE

***   CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF 1.0*SHELL DIAMETER (K3) BASED ON A 4.5 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 20, I=1,28

        QSK3H(I) = 3.116875*1.0*DSI(I)*DSIU(I)*(TC/TPH)*4.5

20    CONTINUE

***   CALCULATE THE CAPACITIES OF THE SHELL AND TUBE FLUIDS


***   DEPENDING ON THE TUBE FLUID CHOICE CALCULATE THE TUBE
***   FLUID PROPERTIES OF DENSITY, SPECIFIC HEAT,  DYNAMIC
***   VISCOSITY AND THERMAL CONDUCTIVITY

      TAVE = TTAVE
```

```
      QTL4(I) = 3.116875 * NT4(I) * ATIN * 2.0 / 4.0

11    CONTINUE

***   CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 4-PASS
***   EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 12, I=1,28

        QTH4(I) = 3.116875 * NT4(I) * ATIN * 8.0 / 4.0

12    CONTINUE

***   CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 2-PASS
***   EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 13, I=1,28

        QTL2(I) = 3.116875 * NT2(I) * ATIN * 2.0 / 2.0

13    CONTINUE

***   CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 2-PASS
***   EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 14, I=1,28

        QTH2(I) = 3.116875 * NT2(I) * ATIN * 8.0 / 2.0

14    CONTINUE

***   CALCULATE THE TUBE CLEARANCE - THE SPACING BETWEEN THE
***   TUBES - FROM THE DEFINITION OF TUBE PITCH

      TC = TPH - DTOT

***   CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF .2*SHELL DIAMETER (K1) BASED ON A 2 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 15, I=1,28

        QSK1L(I) = 3.116875*.20*DSI(I)*DSIU(I)*(TC/TPH)*2.0

15    CONTINUE

***   CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***   SPACING OF .2*SHELL DIAMETER (K1) BASED ON A 4.5 FT/SEC
***   FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER
```

```
      DO 6, I=1,28

         AL2(I) = NT2(I) * ATSU
         AL4(I) = NT4(I) * ATSU
         AL6(I) = NT6(I) * ATSU
         AL8(I) = NT8(I) * ATSU

6     CONTINUE

***   CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 8-PASS
***   EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 7, I=1,28

         QTL8(I) = 3.116875 * NT8(I) * ATIN * 2.0 / 8.0

7     CONTINUE

***   CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 8-PASS
***   EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 8, I=1,28

         QTH8(I) = 3.116875 * NT8(I) * ATIN * 8.0 / 8.0

8     CONTINUE

***   CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 6-PASS
***   EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 9, I=1,28

         QTL6(I) = 3.116875 * NT6(I) * ATIN * 2.0 / 6.0

9     CONTINUE

***   CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 6-PASS
***   EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 10, I=1,28

         QTH6(I) = 3.116875 * NT6(I) * ATIN * 8.0 / 6.0

10    CONTINUE

***   CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 4-PASS
***   EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
***   AVAILABLE SHELL DIAMETER

      DO 11, I=1,28
```

```
      IF (AGAIN .EQ. 'Y') THEN
         GO TO 716
      END IF

***   THE NUMBER OF TUBES ALLOWABLE IS CALCULATED FOR COMMON
***   SHELL DIAMETERS STARTING WITH 6 INCHES AND INCREASING
***   IN INCREMENTS OF 2 INCHES TO A MAXIMUM OF 60 INCHES

3     DO 5, I=1,28

      IF(DSI(I) .GE. 6.0 .AND. DSI(I) .LE. 18.0) THEN
         TH = 0.25
      END IF

      IF(DSI(I) .GT. 18.0 .AND. DSI(I) .LE. 36.0) THEN
         TH = 0.50
      END IF

      IF(DSI(I) .GT. 36.0 .AND. DSI(I) .LE. 60.0) THEN
         TH = 0.75
      END IF

***   THE USABLE SHELL DIAMETER IS .5 INCHES LESS THAN THE
***   ACTUAL SHELL DIAMETER DUE TO TUBE SHEET CONSTRUCTION

      DSIU(I) = (DSI(I) - 0.5)

      F1 = 0.9069

      F2 = 1.0 - ((TH * DSIU(I))/((PI/4.0)*(DSIU(I)**2.0)))

      ARAV = ((DSIU(I) - DTOT)/TPH)**2.0

      NT2(I) = ARAV * F1 * F2

      F4 = 1.0-((2.125+TH * DSIU(I))/((PI/4.0)*(DSIU(I)**2.0)))

      NT4(I) = ARAV * F1 * F4

      F6 = 1.0-((3.25*TH * DSIU(I))/((PI/4.0)*(DSIU(I)**2.0)))

      NT6(I) = ARAV * F1 * F6

      F8 = 1.0-((4.0*TH * DSIU(I))/((PI/4.0)*(DSIU(I)**2.0)))

      NT8(I) = ARAV * F1 * F8

5     CONTINUE

***   CALCULATE THE AVAILABLE TUBE SURFACE AREA PER FOOT OF
***   LENGTH OF THE HEAT EXCHANGER FOR EACH AVAILABLE SHELL
***   DIAMETER
```

```
      PRINT*,' '
713   PRINT*,'WHAT IS THE ENTERING TEMPERATURE OF THE SHELL'
      PRINT*,'FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)'
      PRINT*,' '

      READ*,TS1

***   CASE 1 REPRESENTS THE TUBE FLUID BEING HEATED

      IF (TT1 .LT. TT2) THEN
         CASE = 1.0
      END IF

***   CASE 2 REPRESENTS THE TUBE FLUID BEING COOLED

      IF (TT1 .GT. TT2) THEN
         CASE = 2.0
      END IF

***   CHECH FOR IMPOSSIBLE TEMPERATURE CONDITIONS

      IF (CASE .EQ. 1.0 .AND. TS1 .LE. TT2) THEN
         PRINT*,' '
         PRINT*,'IT IS IMPOSSIBLE TO HAVE THE TUBE FLUID BEING'
         PRINT*,'HEATED AND THE SHELL FLUID ENTRANCE TEMPERATURE'
         PRINT*,'LESS THAN THE TUBE FLUID EXIT TEMPERATURE. TRY'
         PRINT*,'INPUTTING THE SHELL FLUID ENTRANCE TEMPERATURE'
         PRINT*,'AGAIN.'
         PRINT*,' '
         GO TO 713
      END IF

      IF (CASE .EQ. 2.0 .AND. TS1 .GE. TT2) THEN
         PRINT*,' '
         PRINT*,'IT IS IMPOSSIBLE TO HAVE THE TUBE FLUID BEING'
         PRINT*,'COOLED AND THE SHELL FLUID ENTRANCE TEMPERATURE'
         PRINT*,'GREATER THAN THE TUBE FLUID EXIT TEMPERATURE. TRY'
         PRINT*,'INPUTTING THE SHELL FLUID ENTRANCE TEMPERATURE'
         PRINT*,'AGAIN.'
         PRINT*,' '
         GO TO 713
      END IF

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 714
      END IF

      PRINT*,' '
715   PRINT*,'WHAT IS THE FLOW RATE OF THE SHELL FLUID IN'
      PRINT*,'GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)'
      PRINT*,' '

      READ*, MS
```

```
      READ*, PICKFS

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 708
      END IF

***   THE TEMPERATURES AND FLOW RATES ARE NOW INPUT BY THE
***   USER AS PROMPTED BY THE PROGRAM

      PRINT*,' '
709   PRINT*,'WHAT IS THE ENTERING TEMPERATURE OF THE TUBE'
      PRINT*,'FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)'
      PRINT*,' '

      READ*,TT1

      PRINT*,' '
      PRINT*,'WHAT IS THE EXIT TEMPERATURE OF THE TUBE FLUID'
      PRINT*,'IN DEGREES F ?'
      PRINT*,' '

      READ*,TT2

***   CHECK TO SEE IF THE HEAT EXCHANGER TEMPERATURE DATA IS
***   ACCEPTABLE

      IF (TT1 .EQ. TT2) THEN
         PRINT*,' '
         PRINT*,'THE ENTRANCE AND EXIT TEMPERATURES OF THE'
         PRINT*,'TUBE FLUID CANNOT BE EQUAL. PLEASE TRY AGAIN.'
         PRINT*,' '
         GO TO 709
      END IF

***   CALCULATES THE AVERAGE TUBE-SIDE TEMPERATURE TO BE USED
***   IN CALCULATING THE FLUID PROPERTIES

      TTAVE = (TT1 + TT2) / 2.0

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 710
      END IF

      PRINT*,' '
711   PRINT*,'WHAT IS THE FLOW RATE OF THE TUBE FLUID IN'
      PRINT*,'GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)'
      PRINT*,' '

      READ*, MT

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 712
      END IF
```

```
      GO TO 704
   END IF

***   THE USER IS ASKED TO PICK THE TYPE OF FLUIDS WANTED IN
***   THE SHELL AND IN THE TUBE OF THE EXCHANGER

      PRINT*,' '
705   PRINT*,'WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN'
      PRINT*,'THE TUBES OF THE EXCHANGER? THE FOLLOWING'
      PRINT*,'FLUID CHOICES ARE AVAILABLE:'
      PRINT*,' '

      PRINT*,'CHOICE NUMBER          FLUID NAME'
      PRINT*,' '
      PRINT*,'    1.0                     WATER'
      PRINT*,'    2.0                METHYL ALCOHOL'
      PRINT*,'    3.0                   GASOLINE'
      PRINT*,'    4.0                  DOWTHERM A'
      PRINT*,'    5.0                   KEROSENE'
      PRINT*,'    6.0            SAE 10 LUBRICATING OIL'
      PRINT*,'    7.0                ETHYLENE GLYCOL'
      PRINT*,' '

      PRINT*,'PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED '
      PRINT*,'TUBE FLUID. (ie 1.0 IF THE FLUID IS WATER)'
      PRINT*,' '

      READ*, PICKFT

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 706
      END IF

      PRINT*,' '
707   PRINT*,'WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN '
      PRINT*,'THE SHELL OF THE EXCHANGER? THE FOLLOWING'
      PRINT*,'FLUID CHOICES ARE AVAILABLE:'
      PRINT*,' '

      PRINT*,'CHOICE NUMBER          FLUID NAME'
      PRINT*,' '
      PRINT*,'    1.0                     WATER'
      PRINT*,'    2.0                METHYL ALCOHOL'
      PRINT*,'    3.0                   GASOLINE'
      PRINT*,'    4.0                  DOWTHERM A'
      PRINT*,'    5.0                   KEROSENE'
      PRINT*,'    6.0            SAE 10 LUBRICATING OIL'
      PRINT*,'    7.0                ETHYLENE GLYCOL'
      PRINT*,' '

      PRINT*,'PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED '
      PRINT*,'SHELL FLUID. (ie 1.0 IF THE FLUID IS WATER)'
      PRINT*,' '
```

```
      DTOT = DTO(I)
      DTIN = DTI(I)
      ATIN = ATI(I)
      ATSU = ATS(I)
      TPH  = TP(I)
      PI = 3.1415927

      IF (AGAIN .EQ. 'Y') THEN
         GO TO 702
      END IF

***   MATERIAL SPECIFICATIONS OF THE TUBING IS MADE

      PRINT*,' '
703   PRINT*,'WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR'
      PRINT*,'THE TUBING ?'
      PRINT*,' '
      PRINT*,'CHOICE NUMBER            MATERIAL TYPE '
      PRINT*,' '
      PRINT*,'     1.0                 ALUMINUM TYPE 1100'
      PRINT*,'     2.0                 ALUMINUM TYPE 3003'
      PRINT*,'     3.0                 ALUMINUM TYPE 3004'
      PRINT*,'     4.0                 ALUMINUM TYPE 6061'
      PRINT*,'     5.0                 ALUMINUM TYPE 6063'
      PRINT*,'     6.0                    CARBON STEEL'
      PRINT*,'     7.0                 CARBON MOLY STEEL'
      PRINT*,'     8.0               CHROME MOLY STEEL (1%)'
      PRINT*,'     9.0              CHROME MOLY STEEL (2.25%)'
      PRINT*,'    10.0               CHROME MOLY STEEL (5%)'
      PRINT*,'    11.0               CHROME MOLY STEEL (12%)'
      PRINT*,'    12.0                STAINLESS STEEL (18%)'
      PRINT*,'    13.0                STAINLESS STEEL (25%)'
      PRINT*,'    14.0                  ADMIRALTY METAL'
      PRINT*,'    15.0                      COPPER'
      PRINT*,'    16.0                      NICKEL'
      PRINT*,'    17.0                    ZIRCONIUM'
      PRINT*,'    18.0                     TITANIUM'
      PRINT*,' '

      PRINT*,'PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF '
      PRINT*,'MATERIAL DESIRED.(ie 1.0  6.0  18.0 etc.)'
      PRINT*,' '

      READ*,PICKM

      I = INT(PICKM)

***   MATERIAL PROPERTIES ARE SET ONCE A CHOICE IS MADE

      KW = KM(I)

      CF = CFM(I)

      IF (AGAIN .EQ. 'Y') THEN
```

```
      PRINT*,' '
      PRINT*,'CHOICE NUMBER  OUTSIDE DIAMETER(IN)    BWG # '
      PRINT*,' '
      PRINT*,'       1.0               0.75              10'
      PRINT*,'       2.0               0.75              11'
      PRINT*,'       3.0               0.75              12'
      PRINT*,'       4.0               0.75              13'
      PRINT*,'       5.0               0.75              14'
      PRINT*,'       6.0               0.75              15'
      PRINT*,'       7.0               0.75              16'
      PRINT*,'       8.0               0.75              17'
      PRINT*,'       9.0               0.75              18'
      PRINT*,'      10.0               0.75              19'
      PRINT*,'      11.0               0.75              20'
      PRINT*,' '
      PRINT*,'      12.0               1.0                8'
      PRINT*,'      13.0               1.0               10'
      PRINT*,'      14.0               1.0               11'
      PRINT*,'      15.0               1.0               12'
      PRINT*,'      16.0               1.0               13'
      PRINT*,'      17.0               1.0               14'
      PRINT*,'      18.0               1.0               15'
      PRINT*,'      19.0               1.0               16'
      PRINT*,'      20.0               1.0               18'
      PRINT*,'      21.0               1.0               20'
      PRINT*,' '
      PRINT*,'      22.0               1.25               7'
      PRINT*,'      23.0               1.25               8'
      PRINT*,'      24.0               1.25              10'
      PRINT*,'      25.0               1.25              11'
      PRINT*,'      26.0               1.25              12'
      PRINT*,'      27.0               1.25              13'
      PRINT*,'      28.0               1.25              14'
      PRINT*,'      29.0               1.25              16'
      PRINT*,'      30.0               1.25              17'
      PRINT*,'      31.0               1.25              18'
      PRINT*,'      32.0               1.25              20'
      PRINT*,' '
      PRINT*,'      33.0               1.5               10'
      PRINT*,'      34.0               1.5               12'
      PRINT*,'      35.0               1.5               14'
      PRINT*,'      36.0               1.5               16'
      PRINT*,' '

      PRINT*,'PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF '
      PRINT*,'TUBING DESIRED.(ie 1.0  6.0  34.0 etc.)'
      PRINT*,' '

      READ*,PICKT

      I = INT(PICKT)

***   TUBE DIMENSION VARIABLES ARE SET ONCE A CHOICE IS MADE
```

```
     $   0.606,0.620,0.634,0.652,0.666,0.680,0.670,0.732,
     $   0.760,0.782,0.810,0.834,0.856,0.870,0.902,0.930,
     $   0.890,0.920,0.982,1.010,1.032,1.060,1.084,1.120,
     $   1.134,1.152,1.180,1.232,1.282,1.334,1.370/

      DATA (TF(I), I=1,34)/1.152,1.008,0.864,0.720,0.360,
     $   0.2448,0.1728,0.144,0.12672,0.1008,0.0720,0.06048,
     $   0.05328,0.0504,0.04752,0.04464,0.0432,0.04176,
     $   0.04032,0.0396,0.03312,0.02952,0.02736,0.025776,
     $   0.02448,0.02304,0.02232,0.0216,0.021456,0.018,
     $   0.01656,0.015552,0.014688,0.012672/

      DATA (TRE(I), I=1,34)/60.0,70.0,80.0,100.0,200.0,300.0,
     $   400.0,500.0,600.0,700.0,1000.0,2000.0,3000.0,4000.0,
     $   5000.0,6000.0,7000.0,8000.0,9000.0,10000.0,20000.0,
     $   30000.0,40000.0,50000.0,60000.0,70000.0,80000.0,
     $   90000.0,100000.0,200000.0,300000.0,400000.0,
     $   500000.0,1000000.0/

      DATA (SF(I), I=1,34)/1.4112,0.8928,0.6912,0.6048,
     $   0.5472,0.5328,0.5184,0.504,0.4968,0.4896,0.4752,
     $   0.4176,0.3816,0.36,0.3456,0.3384,0.3312,0.32544,
     $   0.31392,0.3024,0.2664,0.2448,0.23184,0.2232,0.216,
     $   0.21024,0.20304,0.2016,0.19872,0.1728,0.1584,0.1512,
     $   0.144,0.1296/

      DATA (SRE(I), I=1,34)/50.0,100.0,200.0,300.0,400.0,
     $   500.0,600.0,700.0,800.0,900.0,1000.0,2000.0,3000.0,
     $   4000.0,5000.0,6000.0,7000.0,8000.0,9000.0,10000.0,
     $   20000.0,30000.0,40000.0,50000.0,60000.0,70000.0,
     $   80000.0,90000.0,100000.0,200000.0,300000.0,400000.0,
     $   500000.0,1000000.0/

      DATA (TP(I), I=1,11)/11*0.9375/
      DATA (TP(I), I=12,21)/10*1.25/
      DATA (TP(I), I=22,32)/11*1.5625/
      DATA (TP(I), I=33,36)/4*1.875/

      DATA (KM(I), I=1,18)/123.0,96.0,97.0,95.0,116.0,30.0,
     $   29.0,27.0,25.0,21.0,14.0,9.3,7.8,70.0,225.0,38.0,
     $   12.0,11.3/

      DATA (CFM(I), I=1,18)/0.7237,0.7389,0.7389,0.7542,0.7542,
     $   0.0778,0.0957,0.1221,0.2188,0.2977,0.5435,0.7542,
     $   1.00,0.8235,0.8569,2.2443,3.00,3.00/

***   THE USER PICKS THE SIZE OF TUBING HE WANTS FOR THE TUBES
***   OF HIS HEAT EXCHANGER. THESE COMMON SIZES OF TUBING AND
***   ASSOCIATED DIMENSIONS WERE TAKEN FROM THE PATTERSON-
***   KELLEY BOOK ON HEAT EXCHANGERS.

4     PRINT*,'     WHAT SIZE OF TUBING DO YOU WANT TO USE IN '
      PRINT*,'YOUR HEAT EXCHANGER? (REMEMBER THE SMALLER THE'
      PRINT*,'BWG NUMBER THE THICKER THE WALL OF THE TUBING)'
```

```
      PRINT*,'      INPUT DATA                        UNITS'
      PRINT*,' '
      PRINT*,'TUBE FLUID ENTRANCE TEMP         DEGREES F'
      PRINT*,'TUBE FLUID EXIT TEMP             DEGREES F'
      PRINT*,'TUBE FLUID FLOW RATE        GALLONS PER MINUTE'
      PRINT*,'SHELL FLUID ENTRANCE TEMP        DEGREES F'
      PRINT*,'SHELL FLUID FLOW RATE       GALLONS PER MINUTE'
      PRINT*,'MAXIMUM TUBE LENGTH                 FEET'
      PRINT*,'MAX TUBE PRESSURE LOSS     POUNDS PER SQUARE INCH'
      PRINT*,'MAX SHELL PRESSURE LOSS    POUNDS PER SQUARE INCH'
      PRINT*,' '

      PRINT*,'WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.'
      READ (5,101) ACALC

      PRINT*,'    THE MAJOR SOURCES OF INFORMATION USED TO '
      PRINT*,'FORMULATE THIS PROGRAM ARE:'
      PRINT*,'  '
      PRINT*,'BREESE,J. - SHELL AND TUBE HEAT EXCHANGERS'
      PRINT*,'KAYS & LONDON - COMPACT HEAT EXCHANGERS'
      PRINT*,'KERN,D. - PROCESS HEAT TRANSFER'
      PRINT*,'HOLMAN,J.P. - HEAT TRANSFER'
      PRINT*,'CHASE,T.A. - PERSONAL COMMUNICATION'
      PRINT*,'FANARITIS,J.P. - HEAT EXCHANGER TECHNOLOGY'
      PRINT*,'GAGGIOLI,R.A. - EFFICIENCY AND COSTING'
      PRINT*,'PATTERSON-KELLEY CO. - HEAT EHCHANGERS'
      PRINT*,'TEMA - STANDARDS OF TUBULAR EXCHANGERS'
      PRINT*,' '

      PRINT*,'WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.'
      READ (5,101) ACALC

      DATA (DSI(I), I=1,28)/6.0,8.0,10.0,12.0,14.0,16.0,18.0,
     $   20.0,22.0,24.0,26.0,28.0,30.0,32.0,34.0,36.0,38.0,
     $   40.0,42.0,44.0,46.0,48.0,50.0,52.0,54.0,56.0,58.0,
     $   60.0/

      DATA (DTO(I), I=1,11)/11*0.75/
      DATA (DTO(I), I=12,21)/10*1.0/
      DATA (DTO(I), I=22,32)/11*1.25/
      DATA (DTO(I), I=33,36)/4*1.50/

      DATA (ATI(I), I=1,36)/0.1822,0.2043,0.223,0.247,0.268,
     $   0.289,0.302,0.314,0.334,0.348,0.363,0.355,0.421,
     $   0.455,0.479,0.515,0.546,0.576,0.594,0.639,0.679,
     $   0.622,0.662,0.757,0.801,0.838,0.882,0.923,0.985,
     $   1.012,1.042,1.092,1.192,1.291,1.397,1.474/

      DATA (ATS(I), I=1,11)/11*0.1963/
      DATA (ATS(I), I=12,21)/10*0.2618/
      DATA (ATS(I), I=22,32)/11*0.3272/
      DATA (ATS(I), I=33,36)/4*0.3927/

      DATA (DTI(I), I=1,36)/0.482,0.510,0.532,0.560,0.584,
```

```
C     PROGRAM MAIN1    LAST UPDATED: 5 MARCH 85
C
C     NICHOLAS J. KOLAR
C     MECHANICAL ENGINEERING MASTER'S THESIS
C     DESIGN OF SHELL AND TUBE HEAT EXCHANGERS
C     JANUARY 1985

      DIMENSION DSI(28),QTL8(28),QTH8(28),QTL6(28),
     $   QTH6(28),QTL4(28),QTH4(28),QTL2(28),QTH2(28),
     $   QSK1L(28),QSK1H(28),QSK2L(28),QSK2H(28),QSK3L(28),
     $   QSK3H(28),TF(34),TRE(34),SF(34),SRE(34),TP(36),
     $   DTO(36),ATI(36),ATS(36),DTI(36),DSIU(28),NT2(28),
     $   NT4(28),NT6(28),NT8(28),AL2(28),AL4(28),AL6(28),
     $   AL8(28),KM(18),CFM(18)


      CHARACTER*1 ACALC,ALENTH,AFOUL,AGAIN,ATUBE,AMATE,ATFLU,
     $            ASFLU,ATTEM,ATGPM,ASTEM,ASGPM,AFFAC,ALREST,
     $            APRES
      REAL NTU ,MT,MS,LRESTR,L,NP,K,KS,KT,K1,KM,KW

      PRINT*,'    THIS PROGRAM CALCULATES THE REQUIRED LENGTH,'
      PRINT*,'SIZE OF SHELL, BAFFLE SPACING, NUMBER OF TUBE'
      PRINT*,'PASSES, AND ESTIMATED CAPITAL COST OF A TEMA'
      PRINT*,'TYPE E SHELL AND TUBE HEAT EXCHANGER WITH 2, 4,'
      PRINT*,'6, OR 8 TUBE PASSES. YHIS PROGRAM ALSO PROVIDES'
      PRINT*,'CALCULATIONS OF THE SHELL-SIDE AND TUBE-SIDE'
      PRINT*,'PRESSURE DROPS. THE USER PICKS THE SIZE OF THE'
      PRINT*,'TUBING TO BE USED ALONG WITH THE TUBE MATERIAL.'
      PRINT*,'THE USER SETS THE MAXIMUM LENGTH ALONG WITH THE'
      PRINT*,'MAXIMUM ALLOWABLE PRESSURE DROPS ON THE SHELL'
      PRINT*,'AND TUBE SIDES.'
      PRINT*,' '
      PRINT*,'THE PROGRAM OUTPUTS THE EXCHANGER CONFIGURATIONS'
      PRINT*,'THAT WILL MEET THE PROBLEM LIMITATIONS. FINAL'
      PRINT*,'DESIGN DATA ON ANY CONFIGURATION IS PROVIDED AS'
      PRINT*,'REQUESTED. ALTERNATE EXCHANGER CONFIGURATIONS'
      PRINT*,'CAN BE COMPARED, AND ORIGINAL INPUT DATA CAN BE'
      PRINT*,'CHANGED TO DETERMINE THEIR EFFECTS ON THE FINAL'
      PRINT*,'DESIGN DATA.'
      PRINT*,' '
      PRINT*,'    THIS PROGRAM IS LIMITED TO LIQUID-TO-LIQUID'
      PRINT*,'APPLICATIONS UNDER LOW PRESSURE CONDITIONS (<300 PSI).'
      PRINT*,' '
      PRINT*,'     A LIMITED NUMBER OF COMMON FLUIDS'
      PRINT*,'ARE AVAILABLE TO BE CHOSEN AS FLUIDS IN THE '
      PRINT*,'TUBE AND SHELL. THIS PROGRAM CAN BE EASILY '
      PRINT*,'MODIFIED TO ACCEPT ADDITIONAL FLUIDS FOR WHICH'
      PRINT*,'DENSITY, VISCOSITY, THERMAL CONDUCTIVITY AND SPECIFIC'
      PRINT*,'HEAT DATA ARE AVAILABLE.'
      PRINT*,' '
      PRINT*,'    THE USER MUST INPUT THE FOLLOWING DATA IN '
      PRINT*,'THE UNITS SPECIFIED:'
      PRINT*,' '
```

APPENDIX B

MAIN PROGRAM LISTING

Variable Name	Explanation	Units
TSAVE	Average temperature of the shell fluid.	°F
TS1	Entrance temperature of the shell fluid.	°F
TS2	Exit temperature of the shell fluid.	°F
TS2NEW	Adjusted exit temperature of the shell fluid.	°F
TTAVE	Average temperature of the tube fluid.	°F
TT1	Entrance temperature of the tube fluid.	°F
TT2	Exit temperature of the tube fluid.	°F
U	Average overall heat transfer coefficient of the exchanger (estimated).	$\frac{BTU}{HR\text{-}FT^2\text{-}°F}$
UC	Average overall heat transfer coefficient of exchanger with clean tubes.	$\frac{BTU}{HR\text{-}FT^2\text{-}°F}$
UCALC	Overall heat transfer coefficient of the exchanger (calculated).	$\frac{BTU}{HR\text{-}FT^2\text{-}°F}$
VS	Shell fluid velocity.	FT/SEC
VT	Tube fluid velocity.	FT/SEC
Z	Estimated capital cost of the exchanger.	$

```
      END IF

      IF(PICKFT .EQ. 6.0) THEN
         CALL DSAE(TAVE,DEN)
         DENT = DEN
         CALL CPSAE(TAVE,CP)
         CPT = CP
         CALL VSAE(TAVE,DV)
         DVT = DV
         CALL KSAE(TAVE,K)
         KT = K
      END IF

      IF(PICKFT .EQ. 7.0) THEN
         CALL DEGLY(TAVE,DEN)
         DENT = DEN
         CALL CPEGLY(TAVE,CP)
         CPT = CP
         CALL VEGLY(TAVE,DV)
         DVT = DV
         CALL KEGLY(TAVE,K)
         KT = K
      END IF

***   CALCULATE THE SHELL FLUID EXIT TEMPERATURE USING THE
***   HEAT BALANCE EQUATION. NOTE THAT 60.0 IS A CONVERSION
***   FOR MIN/HR AND 7.4805 IS A CONVERSION FOR GALLONS/
***   CUBIC FEET.

      IF (CASE .EQ. 1.0) THEN
         Q = ( MT * DENT * CPT * 60.0 / 7.4805 ) * ( TT2-TT1 )
      END IF

      IF (CASE .EQ. 2.0) THEN
         Q = ( MT * DENT * CPT * 60.0 / 7.4805 ) * ( TT1-TT2 )
      END IF

***   GET AN AVERAGE VALUE FOR DENSITY AND SPECIFIC HEAT FOR
***   INITIAL TRIAL AND ERROR GUESS

      IF(PICKFS .EQ. 1.0) THEN
         DENS = 62.4
         CPS = 1.0
      END IF

      IF(PICKFS .EQ. 2.0) THEN
         DENS = 48.1
         CPS = 0.62
      END IF

      IF(PICKFS .EQ. 3.0) THEN
         DENS = 42.7
         CPS = 0.565
      END IF
```

```
      IF(PICKFT .EQ. 1.0) THEN
         CALL DWATER(TAVE,DEN)
         DENT = DEN
         CALL CPWATR(TAVE,CP)
         CPT = CP
         CALL VWATER(TAVE,DV)
         DVT = DV
         CALL KWATER(TAVE,K)
         KT = K
      END IF

      IF(PICKFT .EQ. 2.0) THEN
         CALL DMETH(TAVE,DEN)
         DENT = DEN
         CALL CPMETH(TAVE,CP)
         CPT = CP
         CALL VMETH(TAVE,DV)
         DVT = DV
         CALL KMETH(TAVE,K)
         KT = K
      END IF

      IF(PICKFT .EQ. 3.0) THEN
         CALL DGAS(TAVE,DEN)
         DENT = DEN
         CALL CPGAS(TAVE,CP)
         CPT = CP
         CALL VGAS(TAVE,DV)
         DVT = DV
         CALL KGAS(TAVE,K)
         KT = K
      END IF

      IF(PICKFT .EQ. 4.0) THEN
         CALL DDOWA(TAVE,DEN)
         DENT = DEN
         CALL CPDOWA(TAVE,CP)
         CPT = CP
         CALL VDOWA(TAVE,DV)
         DVT = DV
         CALL KDOWA(TAVE,K)
         KT = K
      END IF

      IF(PICKFT .EQ. 5.0) THEN
         CALL DKERO(TAVE,DEN)
         DENT = DEN
         CALL CPKERO(TAVE,CP)
         CPT = CP
         CALL VKERO(TAVE,DV)
         DVT = DV
         CALL KKERO(TAVE,K)
         KT = K
```

```
      IF(PICKFS .EQ. 4.0) THEN
         DENS = 56.8
         CPS = 0.60
      END IF

      IF(PICKFS .EQ. 5.0) THEN
         DENS = 44.4
         CPS = 0.60
      END IF

      IF(PICKFS .EQ. 6.0) THEN
         DENS = 52.25
         CPS = 0.52
      END IF

      IF(PICKFS .EQ. 7.0) THEN
         DENS = 66.2
         CPS = 0.644
      END IF

***   SOLVE HEAT BALANCE EQUATION FOR INITIAL GUESS OF SHELL
***   FLUID EXIT TEMPERATURE

      IF (CASE .EQ. 1.0) THEN
         TS2 = TS1 - (Q/ ( MS * DENS * CPS * 60.0 / 7.4805))
      END IF

      IF (CASE .EQ. 2.0) THEN
         TS2 = TS1 + (Q/ ( MS * DENS * CPS * 60.0 / 7.4805))
      END IF

***   CALCULATE THE SHELL FLUID PROPERTIES BASED UPON THE
***   AVERAGE TEMPERATURE OF THE SHELL FLUID

48    TSAVE = (TS2+TS1) / 2.0
      TAVE = TSAVE

      IF(PICKFS .EQ. 1.0) THEN
         CALL DWATER(TAVE,DEN)
         DENS = DEN
         CALL CPWATR(TAVE,CP)
         CPS = CP
         CALL VWATER(TAVE,DV)
         DVS = DV
         CALL KWATER(TAVE,K)
         KS = K
      END IF

      IF(PICKFS .EQ. 2.0) THEN
         CALL DMETH(TAVE,DEN)
         DENS = DEN
         CALL CPMETH(TAVE,CP)
         CPS = CP
```

```
   CALL VMETH(TAVE,DV)
   DVS = DV
   CALL KMETH(TAVE,K)
   KS = K
END IF

IF(PICKFS .EQ. 3.0) THEN
   CALL DGAS(TAVE,DEN)
   DENS = DEN
   CALL CPGAS(TAVE,CP)
   CPS = CP
   CALL VGAS(TAVE,DV)
   DVS = DV
   CALL KGAS(TAVE,K)
   KS = K
END IF

IF(PICKFS .EQ. 4.0) THEN
   CALL DDOWA(TAVE,DEN)
   DENS = DEN
   CALL CPDOWA(TAVE,CP)
   CPS = CP
   CALL VDOWA(TAVE,DV)
   DVS = DV
   CALL KDOWA(TAVE,K)
   KS = K
END IF

IF(PICKFS .EQ. 5.0) THEN
   CALL DKERO(TAVE,DEN)
   DENS = DEN
   CALL CPKERO(TAVE,CP)
   CPS = CP
   CALL VKERO(TAVE,DV)
   DVS = DV
   CALL KKERO(TAVE,K)
   KS = K
END IF

IF(PICKFS .EQ. 6.0) THEN
   CALL DSAE(TAVE,DEN)
   DENS = DEN
   CALL CPSAE(TAVE,CP)
   CPS = CP
   CALL VSAE(TAVE,DV)
   DVS = DV
   CALL KSAE(TAVE,K)
   KS = K
END IF

IF(PICKFS .EQ. 7.0) THEN
   CALL DEGLY(TAVE,DEN)
   DENS = DEN
   CALL CPEGLY(TAVE,CP)
```

```
          CPS = CP
          CALL VEGLY(TAVE,DV)
          DVS = DV
          CALL KEGLY(TAVE,K)
          KS = K
       END IF

***    RECHECK SHELL EXIT TEMPERATURE CALCULATION USING NEW
***    VALUES FOR DENSITY AND SPECIFIC HEAT OF THE SHELL FLUID

       IF (CASE .EQ. 1.0) THEN
          TS2NEW = TS1 - (Q / ( MS * DENS * CPS * 60.0 / 7.4805))
       END IF

       IF (CASE .EQ. 2.0) THEN
          TS2NEW = TS1 + (Q / ( MS * DENS * CPS * 60.0 / 7.4805))
       END IF

       IF (ABS (TS2-TS2NEW) .GT. 1.0) THEN
          TS2 =TS2NEW
          GO TO 48
       END IF

***    DETERMINE WHICH FLUID HAS THE MAXIMUM CAPACITY RATE FOR
***    USE IN THE NTU METHOD

       CS = MS * DENS * CPS * 60.0 / 7.4805

       CT = MT * DENT * CPT * 60.0 / 7.4805

       IF (CS .GE. CT) THEN
          CMIN = CT
          T1MIN =TT1
          T2MIN = TT2
          CMAX = CS
          T1MAX = TS1
          T2MAX = TS2
       ELSE
          CMIN = CS
          T1MIN = TS1
          T2MIN = TS2
          CMAX = CT
          T1MAX = TT1
          T2MAX = TT2
       END IF

***    CALCULATE THE HEAT TRANSFER EFFECTIVENESS

       EFF = ABS(T2MIN -T1MIN) / ABS(T1MAX-T1MIN)

***    CHECK FOR IMPOSSIBLE CONDITIONS

       IF (EFF .GT. 1.0) THEN
          PRINT*,' '
```

```
            PRINT*,'THE TEMPERATURES PROVIDED AS INPUT ARE SUCH'
            PRINT*,'THAT THE HEAT EXCHANGER WOULD BE REQUIRED TO'
            PRINT*,'ACCOMPLISH AN IMPOSSIBLE FEAT. THE MAXIMUM'
            PRINT*,'POSSIBLE HEAT TRANSFER FOR AN EXCHANGER OF'
            PRINT*,'INFINITE LENGTH IS LESS THAN WHAT YOU REQUIRE'
            PRINT*,'THIS HEAT EXCHANGER TO ACCOMPLISH. TRY'
            PRINT*,'INPUTTING THE TEMPERATURES ONCE AGAIN.'
            PRINT*,' '
            GO TO 709
         END IF

         RATIO = CMIN / CMAX

***      CALCULATE THE NUMBER OF HEAT TRANSFER UNITS FOR THE
***      HEAT EXCHANGER USING EQUATION FROM HOLMAN'S - HEAT
***      TRANSFER ON PAGE 457

         NTU = ( -(1.0 + (RATIO **2.0))**(-0.5)) * LOG (((2.0/EFF) - 1.0
        $ - RATIO - ((1.0 + (RATIO **2.0))**0.5)) /
        $ (( 2.0/EFF) - 1.0 - RATIO + ((1.0 + (RATIO ** 2.0))**0.5)))

***      CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT USING
***      CLEAN TUBING. THE VALUES UTILIZED IN THIS PROGRAM WERE
***      TAKEN FROM HEAT EXCHANGERS THERMAL-HYDRAULIC
***      FUNDAMENTALS AND DESIGN BY KAKAC AND BERGLES.
***      VALUES FOR FOULING RESISTANCES ARE TAKEN FROM TEMA
***      STANDARDS HANDBOOK P 140. UNITS ARE HR-SQ FT-DEG F/BTU

         IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 1.0) THEN
               UC = 275.0
               RFOULT = .0015
               RO = .00075
               RI = .00075
         END IF

         IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 2.0) THEN
               UC = 150.0
               RFOULT = .0015
               RO = .00075
               RI = .00075
         END IF

         IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 3.0) THEN
               UC = 150.0
               RFOULT = .0015
               RO = .00075
               RI = .00075
         END IF

         IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 4.0) THEN
               UC = 100.0
               RFOULT = .002
               RO = .00125
               RI = .00075
```

```
      END IF

      IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 5.0) THEN
            UC = 100.0
            RFOULT = .002
            RO = .00125
            RI = .00075
      END IF

      IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 6.0) THEN
            UC = 57.5
            RFOULT = .0025
            RO = .00175
            RI = .00075
      END IF

      IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 7.0) THEN
            UC = 57.5
            RFOULT = .0025
            RO = .00175
            RI = .00075
      END IF

      IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 1.0) THEN
            UC = 150.0
            RFOULT = .0015
            RO = .00075
            RI = .00075
      END IF

      IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 2.0) THEN
            UC = 115.0
            RFOULT = .002
            RO = .001
            RI = .001
      END IF

      IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 3.0) THEN
            UC = 115.0
            RFOULT = .002
            RO = .001
            RI = .001
      END IF

      IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 4.0) THEN
            UC = 85.0
            RFOULT = .0025
            RO = .0015
            RI = .001
      END IF

      IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 5.0) THEN
            UC = 85.0
            RFOULT = .0025
```

```
          RO = .0015
          RI = .001
     END IF

     IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 6.0) THEN
          UC = 50.0
          RFOULT = .003
          RO = .002
          RI = .001
     END IF

     IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 7.0) THEN
          UC = 50.0
          RFOULT = .003
          RO = .002
          RI = .001
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 1.0) THEN
          UC = 150.0
          RFOULT = .0015
          RO = .00075
          RI = .00075
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 2.0) THEN
          UC = 115.0
          RFOULT = .002
          RO = .001
          RI = .001
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 3.0) THEN
          UC = 115.0
          RFOULT = .002
          RO = .001
          RI = .001
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 4.0) THEN
          UC = 85.0
          RFOULT = .0025
          RO = .0015
          RI = .001
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 5.0) THEN
          UC = 85.0
          RFOULT = .0025
          RO = .0015
          RI = .001
     END IF

     IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 6.0) THEN
```

```
      UC = 50.0
      RFOULT = .003
      RO = .002
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 7.0) THEN
      UC = 50.0
      RFOULT = .003
      RO = .002
      RI = .001
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 1.0) THEN
      UC = 100.0
      RFOULT = .002
      RO = .00075
      RI = .00125
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 2.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .00125
      RI = .00125
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 3.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .00125
      RI = .00125
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 4.0) THEN
      UC = 65.0
      RFOULT = .003
      RO = .00175
      RI = .00125
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 5.0) THEN
      UC = 65.0
      RFOULT = .003
      RO = .00175
      RI = .00125
END IF

IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 6.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .00225
      RI = .00125
END IF
```

```
IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 7.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .00225
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 1.0) THEN
      UC = 100.0
      RFOULT = .002
      RO = .00075
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 2.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .00125
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 3.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .00125
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 4.0) THEN
      UC = 65.0
      RFOULT = .003
      RO = .00175
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 5.0) THEN
      UC = 65.0
      RFOULT = .003
      RO = .00175
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 6.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .00225
      RI = .00125
END IF

IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 7.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .00225
```

```
      RI = .00125
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 1.0) THEN
      UC = 57.5
      RFOULT = .0025
      RO = .00175
      RI = .00075
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 2.0) THEN
      UC = 50.0
      RFOULT = .003
      RO = .001
      RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 3.0) THEN
      UC = 50.0
      RFOULT = .003
      RO = .001
      RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 4.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .0015
      RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 5.0) THEN
      UC = 35.0
      RFOULT = .0035
      RO = .0015
      RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 6.0) THEN
      UC = 20.0
      RFOULT = .005
      RO = .0025
      RI = .0025
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 7.0) THEN
      UC = 20.0
      RFOULT = .005
      RO = .0025
      RI = .0025
END IF


IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 1.0) THEN
```

```
            UC = 57.5
            RFOULT = .0025
            RO = .00175
            RI = .00075
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 2.0) THEN
            UC = 50.0
            RFOULT = .003
            RO = .001
            RI = .002
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 3.0) THEN
            UC = 50.0
            RFOULT = .003
            RO = .001
            RI = .002
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 4.0) THEN
            UC = 35.0
            RFOULT = .0035
            RO = .0015
            RI = .002
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 5.0) THEN
            UC = 35.0
            RFOULT = .0035
            RO = .0015
            RI = .002
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 6.0) THEN
            UC = 20.0
            RFOULT = .005
            RO = .0025
            RI = .0025
      END IF

      IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 7.0) THEN
            UC = 20.0
            RFOULT = .005
            RO = .0025
            RI = .0025
      END IF

***   THE USER IS ASKED IF FOULING RESISTANCE SHOULD BE
***   CONSIDERED IN THE EVALUATION OF THE OVERALL HEAT
***   TRANSFER COEFFICIENT.

      IF (AFFAC .EQ. 'N') THEN
         GO TO 50
```

```
      END IF

      PRINT*,' '
      PRINT*,'DO YOU WANT TO ADD A FOULING FACTOR TO THE HEAT'
      PRINT*,'TRANSFER COEFFICIENT GIVEN FOR CLEAN TUBES ?'
      PRINT*,'ANSWER Y FOR YES AND N FOR NO.'
      PRINT*,' '
      READ (5,101) AFOUL

50    IF (AFOUL .EQ. 'Y') THEN
         U = 1.0 / ( RFOULT + ( 1.0 / UC ) )
      ELSE
         U = UC
      END IF

***   DETERMINE WHAT LENGTH RESTRICTIONS ARE TO BE CONSIDERED
***   FOR THIS HEAT EXCHANGER APPLICATION

      IF (ALREST .EQ. 'N') THEN
         GO TO 51
      END IF

      PRINT*,' '
      PRINT*,'WHAT IS THE MAXIMUM LENGTH THAT THE HEAT '
      PRINT*,'EXCHANGER TUBES CAN BE FOR THIS APPLICATION ?'
      PRINT*,'REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR'
      PRINT*,'CLEANING SO THE MAXIMUM LENGTH OF THE HEAT'
      PRINT*,'EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE'
      PRINT*,'LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL'
      PRINT*,'BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN'
      PRINT*,'FEET THAT THE HEAT EXCHANGER TUBES CAN BE.'
      PRINT*,'( ie 5.0 , 7.5, 19.6 etc.)'
      PRINT*,' '

      READ*, LRESTR

51    IF (APRES .EQ. 'N') THEN
         GO TO 52
      END IF

***   DETERMINE WHAT PRESSURE LOSSES ARE ACCEPTABLE ON BOTH
***   THE SHELL AND THE TUBE SIDES OF THE EXCHANGER

      PRINT*,' '
      PRINT*,'WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN'
      PRINT*,'THE TUBES IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)'
      PRINT*,' '

      READ*, AMAXTP

      PRINT*,' '
      PRINT*,'WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN'
      PRINT*,'THE SHELL IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)'
      PRINT*,' '
```

```
      READ*, AMAXSP

***   NOW THAT THE TUBE FLOW RATE, THE SHELL FLOW RATE, THE
***   MAXIMUM TUBE LENGTH, AND THE MAXIMUM TUBE AND SHELL PRESSURE
***   LOSSES HAVE BEEN SPECIFIED, THE PROGRAM CHECKS EACH AVAILABLE
***   SHELL SIZE AND NUMBER OF TUBE PASSES TO SEE WHICH 1-2 HEAT
***   EXCHANGER CONFIGURATIONS MEET THE REQUIRED HEAT DUTY AND
***   LENGTH AND PRESSURE LIMITATIONS IMPOSED BY THE USER

***   THE FOLLOWING FIVE VARIABLES ARE USED AS COUNTERS IN THE
***   PRINTING ALGORITHM:

52    J = 0
      J2 = 0
      J4 = 0
      J6 = 0
      J8 = 0

      PRINT*,' '

***   ALL AVAILABLE 2-PASS EXCHANGER CONFIGURATIONS ARE CHECKED


      DO 60, I = 1,28

      K2 = 0
      M1 = 0
      M2 = 0
      M3 = 0

***   FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
***   RECOMMENDED MINIMUMS

      IF(MT .GE. QTL2(I) .AND. MT .LE. QTH2(I)) THEN
         K2 = 1
      END IF

      IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
         M1 = 1
      END IF

      IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
         M2 = 1
      END IF

      IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
         M3 = 1
      END IF

      IF(K2 .EQ. 0) THEN
         GO TO 60
      END IF
```

```
      WRITE(*,604) PICKS*DSI(N)
604   FORMAT ('                         ',F6.2, '  INCHES')
      PRINT*,' '
      WRITE(*,605) L
605   FORMAT ('EXCHANGER TUBE LENGTH:  ', F8.3, '  FEET')
      PRINT*,' '
      WRITE(*,606) PLTTOT
606   FORMAT ('TUBE-SIDE PRESSURE LOSS:  ',F6.2,
     $         '  POUNDS PER SQUARE INCH')
      PRINT*,' '
      WRITE(*,607) PLSTOT
607   FORMAT ('SHELL-SIDE PRESSURE LOSS:  ',F6.2,
     $         '  POUNDS PER SQUARE INCH')
      PRINT*,' '
      WRITE(*,608) Z
608   FORMAT ('ESTIMATED CAPITAL COST:  $ ',F9.2)
      PRINT*,' '

***   THESE VALUES ARE PRINTED TO SEE HOW CLOSE THE COMPUTED
***   VALUE OF THE OVERALL HEAT TRANSFER COEFFICIENT COMES
***   TO THE AVERAGE VALUE FOUND IN THE LITERATURE

      PRINT*,'U = ', U
      PRINT*,'UCALC = ', UCALC

      PRINT*,' '
      PRINT*,'DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER'
      PRINT*,'CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.'
      PRINT*,' '
      READ (5,101) ACALC
101   FORMAT (1A1)

      IF(ACALC .EQ. 'N') THEN
         GO TO 700
      END IF

      GO TO 71

700   PRINT*,' '
      PRINT*,'DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS'
      PRINT*,'TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO'
      PRINT*,' '

      READ (5,101) AGAIN

      IF (AGAIN .EQ. 'Y') THEN

      PRINT*,' '
      PRINT*,'DO YOU WANT TO CHANGE THE TUBING SIZE ?'
      PRINT*,' '

      READ (5,101) ATUBE

         IF (ATUBE .EQ. 'Y') THEN
```

```
***   DEPENDS ON THE REYNOLDS NUMBER USING SEIDER AND TATE
***   FRICTION FACTOR CORRELATIONS FROM PAGE 836 OF KERN

      CALL TFRICT(RET,TRE,TF,FRT)

***   CALCULATE THE PRESSURE LOSSES IN THE TUBE DUE TO BOTH
***   THE FRICTION LOSSES AND THE RETURN LOSSES USING
***   EQUATIONS FROM PAGE 148 AND 836 OF KERN. NOTE THAT
***   .006944 IS A CONVERSION FACTOR FOR PSF TO PSI AND
***   .4335 IS A CONVERSION FACTOR FOR FEET OF WATER TO PSI

      TAVE = TTAVE
      CALL DWATER(TAVE,DEN)
      DENW = DEN

      PLTF = (( FRT*NP*L* (GT**2.0) ) / ( (DTIN/12.0)* 2.0 * AG
     $   * DENT * ((DVT/DVTF)**0.14))) * 0.006944

      PLTR = (( 2.0 * NP * (VT**2.0)) / (2.0 * AG *
     $   (DENT/DENW) )) * 0.4335

***   CALCULATE THE TOTAL PRESSURE LOSS IN THE TUBES

      PLTTOT = PLTF + PLTR

***   CALCULATE THE SHELL-SIDE FLUID FRICTION FACTOR WHICH
***   DEPENDS ON THE REYNOLDS NUMBER TAKEN FROM A CORRELATION
***   OF FRICTION FACTORS FOR TUBE BUNDLES WITH 25% CUT
***   SEGMENTAL BAFFLES FROM KERN PAGE 838

      CALL SFRICT(RES,SRE,SF,FRS)

***   CALCULATE THE SHELL-SIDE PRESSURE LOSS VIA EQUATION
***   FROM KERN ON PAGE 147

      PLSTOT = (FRS * L * ( GS**2.0 ) *
     $   (INT( (L*12.0) / (DSI(N) * PICKS) ) + 1.0 ) * 0.006944)
     $   / ( 2.0 * AG * DENS *(DEQS/12.0)* ((DVS/DVSF) ** 0.14))

***   PRINT THE FINAL CONFIGURATION, LENGTH , AND PRESSURE
***   DROP FOR THE HEAT EXCHANGER

      PRINT*,' '
      PRINT*,'THE FINAL CONFIGURATION IS :'
      PRINT*,' '
      WRITE(*,601) DSI(N)
601   FORMAT ('SHELL DIAMETER:  ', F6.1,'  INCHES')
      PRINT*,' '
      WRITE(*,602) NP
602   FORMAT ('NUMBER OF TUBE-SIDE PASSES:  ', F3.1)
      PRINT*,' '
      WRITE(*,603) PICKS
603   FORMAT ('BAFFLE SPACING:  ', F5.2, '  OR')
      PRINT*,' '
```

```
***   USING THE CALCULATED VALUE OF THE OVERALL HEAT TRANSFER
***   COEFFICIENT, CALCULATE THE REQUIRED HEAT TRANSFER AREA.

      ACORR = (NTU * CMIN) / UCALC

***   USING THE CORRECT VALUE OF REQUIRED HEAT TRANSFER AREA,
***   CALCULATE THE LENGTH OF TUBING OF THE DESIRED HEAT
***   EXCHANGER CONFIGURATION.

      IF (PICKP .EQ. 2.0) THEN
         L = ACORR / AL2(N)
      END IF

      IF (PICKP .EQ. 4.0) THEN
         L = ACORR / AL4(N)
      END IF

      IF (PICKP .EQ. 6.0) THEN
         L = ACORR / AL6(N)
      END IF

      IF (PICKP .EQ. 8.0) THEN
         L = ACORR / AL8(N)
      END IF


      NP = PICKP
      AG = 32.1740

***   CALCULATE THE ESTIMATED COST OF THE HEAT EXCHANGER DESIGN
***   USING EQUATIONS FOUND IN THE BOOK - EFFICIENCY AND COSTING
***   BY GAGGIOLI ON PAGE 232

      IF (PICKP .EQ. 2.0) THEN
         Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
     $        ((TPH/12.0) ** 0.75) * (NT2(N) ** 0.975)
      END IF

      IF (PICKP .EQ. 4.0) THEN
         Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
     $        ((TPH/12.0) ** 0.75) * (NT4(N) ** 0.975)
      END IF

      IF (PICKP .EQ. 6.0) THEN
         Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
     $        ((TPH/12.0) ** 0.75) * (NT6(N) ** 0.975)
      END IF

      IF (PICKP .EQ. 8.0) THEN
         Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
     $        ((TPH/12.0) ** 0.75) * (NT8(N) ** 0.975)
      END IF

***   CALCULATE THE TUBE-SIDE FLUID FRICTION FACTOR WHICH
```

```
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VGAS (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 4.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VDOWA (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 5.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VKERO (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 6.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VSAE (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 7.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VEGLY (TAVE,DV)
            DVTF = DV
         END IF

         HI = (KT/(DTIN/12.0)) * 0.027 * (RET**0.8) *
     $      ((CPT * DVT * 3600.0/KT)**0.3333) * ((DVT/DVTF)**0.14)

***      CALCULATE THE TUBE WALL RESISTANCE FROM EQUATION FOUND IN
***      TEMA STANDARDS HANDBOOK P 137.

         RW = (DTOT/(24.0*KW)) * LOG (DTOT/(DTOT - 2.0*(DTOT-DTIN)))

***      CALCULATE THE AO/AI RATIO WHICH REDUCES TO DO/DI

         TRATIO = DTOT / DTIN

***      CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT

         IF (AFOUL .EQ. 'Y') THEN

            UCALC = 1.0/((1.0/HO) + RO + RW + (RI*TRATIO)
     $                 + ((1.0/HI)*TRATIO))

         ELSE IF (AFOUL .EQ. 'N') THEN

            UCALC = 1.0/((1.0/HO) + RW + ((1.0/HI)*TRATIO))

         END IF
```

```
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VMETH (TAVE,DV)
          DVSF = DV
       END IF

       IF(PICKFS .EQ. 3.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VGAS (TAVE,DV)
          DVSF = DV
       END IF

       IF(PICKFS .EQ. 4.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VDOWA (TAVE,DV)
          DVSF = DV
       END IF

       IF(PICKFS .EQ. 5.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VKERO (TAVE,DV)
          DVSF = DV
       END IF

       IF(PICKFS .EQ. 6.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VSAE (TAVE,DV)
          DVSF = DV
       END IF

       IF(PICKFS .EQ. 7.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VEGLY (TAVE,DV)
          DVSF = DV
       END IF

       HO = (KS/(DEQS/12.0)) * 0.36 * (RES**0.55) *
      $   ((CPS * DVS * 3600.0/KS)**0.3333) * ((DVS/DVSF)**0.14)

***    FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
***    CORPORATION P 136

       IF(PICKFS .EQ. 1.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VWATER (TAVE,DV)
          DVTF = DV
       END IF

       IF(PICKFS .EQ. 2.0) THEN
          TAVE = (TTAVE + TSAVE) / 2.0
          CALL VMETH (TAVE,DV)
          DVTF = DV
       END IF

       IF(PICKFS .EQ. 3.0) THEN
```

```
      IF (PICKP .EQ. 8.0) THEN
         VT = (C*MT) / (ATIN * (NT8(N)/8.0) )
      END IF

***   CALCULATE THE EQUIVALENT DIAMETER OF THE FLUID CONDUIT
***   CROSSING THE TUBES USING EQUATIONS FROM KERN IN HIS BOOK
***   PROCESS HEAT TRANSFER ON PAGE 139. THE EQUATION IS FOR A
***   TRIANGULAR PITCH ARRANGEMENT ONLY. UNITS ARE IN INCHES.

      DEQS=(4.0*((.5*TPH)*(.86*TPH)-((.5*PI*(DTOT**2.))/4.0)))
     $   /(.5*DTOT)

***   THE MASS VELOCITY OF THE SHELL FLUID IS CALCULATED. UNITS
***   ARE LB/SEC-SQUARE FEET

      GS = VS * DENS

***   THE REYNOLDS NUMBER OF THE SHELL FLUID IS CALCULATED

      RES = ( (DEQS / 12.0) * GS ) / DVS

***   THE MASS VELOCITY OF THE TUBE FLUID IS CALCULATED. UNITS
***   ARE LB/SEC-SQUARE FEET

      GT = VT * DENT

***   THE REYNOLDS NUMBER OF THE TUBE FLUID IS CALCULATED

      RET = ( (DTIN / 12.0) * GT) / DVT

***   NOW THAT THE CONFIGURATION HAS BEEN CHOSEN, CALCULATE THE
***   OVERALL U VALUE FOR THE HEAT EXCHANGER USING GENERALIZED
***   CORRELATIONS FOUND IN THE REFERENCES.

***   FROM TEMA PAGE 137, THE OVERALL HEAT TRANSFER COEFFICIENT
***   REQUIRES THE EVALUATION OF:
***   HO : THE FILM COEFFICIENT OF THE FLUID OUTSIDE THE TUBES
***   HI : THE FILM COEFFICIENT OF T8HE FLUID INSIDE THE TUBES
***   RO : THE FOULING RESISTANCE ON THE OUTSIDE OF THE TUBES
***   RI : THE FOULING RESISTANCE ON THE INSIDE OF THE TUBES
***   RW : THE RESISTANCE OF THE TUBE WALL REFERRED TO THE
***        OUTSIDE SURFACE OF THE TUBE WALL
***   AO/AI : THE RATIO OF OUTSIDE TO INSIDE SURFACE OF TUBING

***   FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
***   CORPORATION P 157

      IF(PICKFS .EQ. 1.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VWATER (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 2.0) THEN
```

```
***   IF THERE ARE AVAILABLE CONFIGURATIONS THAT MEET THE
***   DESIGN SPECIFICATIONS, THE USER IS ASKED FOR HIS
***   CHOICE OF CONFIGURATIONS THAT A FINAL OUTPUT IS DESIRED

70    PRINT*,' '
      PRINT*,' '
      PRINT*,'THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE'
      PRINT*,'OF THE ABOVE LISTED HEAT EXCHANGER '
      PRINT*,'CONFIGURATIONS. REMEMBER ONLY THESE LISTED'
      PRINT*,'CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU'
      PRINT*,'HAVE INPUT INTO THE PROGRAM'
      PRINT*,' '
71    PRINT*,'WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,'
      PRINT*,'14.0, 28.0 etc.'
      PRINT*,' '
      READ*,PICKD
      PRINT*,' '
      PRINT*,'WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE'
      PRINT*,'2.0, 4.0, 6.0, OR 8.0'
      PRINT*,' '
      READ*,PICKP
      PRINT*,' '
      PRINT*,'WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE'
      PRINT*,'0.2, 0.45, OR 1.0'
      PRINT*,' '
      READ*,PICKS

      N = INT( (PICKD/2.0) - 2.0 )

***   ONCE THE HEAT EXCHANGER CONFIGURATION HAS BEEN SELECTED, THE
***   PROGRAM CAN NOW CALCULATE THE ACTUAL SHELL AND TUBE FLUID
***   VELOCITIES. NOTE THAT C IS A CONVERSION FACTOR TO GET VELOCITY
***   IN FT/SEC BY INPUTTING A FLOW RATE OF GALLONS PER MINUTE AND
***   HAVING AREA IN INCHES.

      C = .3208342

      ASC = (DSIU(N)/TPH) * TC * (PICKS * DSI(N))

      VS = C * MS/ASC

      IF (PICKP .EQ. 2.0) THEN
         VT = (C*MT) / (ATIN * (NT2(N)/2.0) )
      END IF

      IF (PICKP .EQ. 4.0) THEN
         VT = (C*MT) / (ATIN * (NT4(N)/4.0) )
      END IF

      IF (PICKP .EQ. 6.0) THEN
         VT = (C*MT) / (ATIN * (NT6(N)/6.0) )
      END IF
```

```
      PICKP = 8.0
      N = I
      CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $         PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $         DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $         AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $         SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

      IF (OKFIN .EQ. 1) THEN
         CALL PRINT8 (J,J8,PICKS,PICKD)
      END IF

   ELSE IF (M3 .EQ. 1) THEN

      PICKD = (I + 2.0) * 2.0
      PICKS = 1.0
      PICKP = 8.0
      N = I
      CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $         PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $         DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $         AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $         SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

      IF (OKFIN .EQ. 1) THEN
         CALL PRINT8 (J,J8,PICKS,PICKD)
      END IF

   END IF

66    CONTINUE

      IF (J8 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'NO 8-PASS HEAT EXCHANGER CONFIGURATIONS'
         PRINT*,'WILL MEET THE PROBLEM SPECIFICATIONS.'
      END IF

***   IF NO SHELL CONFIGURATIONS MEET THE CRITERIA THE PROGRAM
***   MOVES TO THE END AND ASKS IF ANY MORE DESIGNS ARE TO
***   BE COMPUTED

      IF (J2 .EQ. 0 .AND. J4 .EQ. 0 .AND. J6 .EQ. 0 .AND.
     $    J8 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'NO AVAILABLE SHELL DIAMETERS WILL MEET THE '
         PRINT*,'LENGTH AND PRESSURE RESTRICTIONS GIVEN IN THIS'
         PRINT*,'PROBLEM. SPLIT SHELL DESIGN OR MULTIPLE SHELLS'
         PRINT*,'IN SERIES MUST BE USED. THIS COMPUTER PROGRAM'
         PRINT*,'IS NOT SET UP TO HANDLE THAT TYPE OF HEAT '
         PRINT*,'EXCHANGER DESIGN.'
         PRINT*,' '
         GO TO 700
      END IF
```

```
      M1 = 0
      M2 = 0
      M3 = 0

***   FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
***   RECOMMENDED MINIMUMS

      IF(MT .GE. QTL8(I) .AND. MT .LE. QTH8(I)) THEN
         K8 = 1
      END IF

      IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
         M1 = 1
      END IF

      IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
         M2 = 1
      END IF

      IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
         M3 = 1
      END IF

      IF(K8 .EQ. 0) THEN
         GO TO 66
      END IF

      IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
         GO TO 66
      END IF

***   A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
***   THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

      IF (M1 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.2
         PICKP = 8.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT8 (J,J8,PICKS,PICKD)
         END IF

      ELSE IF (M2 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.45
```

```
         IF (OKFIN .EQ. 1) THEN
            CALL PRINT6 (J,J6,PICKS,PICKD)
         END IF

      ELSE IF (M2 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.45
         PICKP = 6.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT6 (J,J6,PICKS,PICKD)
         END IF

      ELSE IF (M3 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 1.0
         PICKP = 6.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT6 (J,J6,PICKS,PICKD)
         END IF

      END IF

 64   CONTINUE

      IF (J6 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'NO 6-PASS HEAT EXCHANGER CONFIGURATIONS'
         PRINT*,'WILL MEET THE PROBLEM SPECIFICATIONS.'
      END IF

      PRINT*,' '

***   ALL AVAILABLE 8-PASS CONFIGURATIONS ARE CHECKED

 65   DO 66, I = 1,28

      K3 = 0
```

```
      END IF

      PRINT*,' '

***   ALL AVAILABLE 6-PASS CONFIGURATIONS ARE CHECKED

63    DO 64, I = 1,28

      K6 = 0
      M1 = 0
      M2 = 0
      M3 = 0

***   FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
***   RECOMMENDED MINIMUMS

      IF(MT .GE. QTL6(I) .AND. MT .LE. QTH6(I)) THEN
         K6 = 1
      END IF

      IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
         M1 = 1
      END IF

      IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
         M2 = 1
      END IF

      IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
         M3 = 1
      END IF

      IF(K6 .EQ. 0) THEN
         GO TO 64
      END IF

      IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
         GO TO 64
      END IF

***   A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
***   THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

      IF (M1 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.2
         PICKP = 6.0
         N = 1
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)
```

```
            PICKD = (I + 2.0) * 2.0
            PICKS = 0.2
            PICKP = 4.0
            N = I
            CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $           PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $           DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $           AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $           SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

            IF (OKFIN .EQ. 1) THEN
               CALL PRINT4 (J,J4,PICKS,PICKD)
            END IF

         ELSE IF (M2 .EQ. 1) THEN

            PICKD = (I + 2.0) * 2.0
            PICKS = 0.45
            PICKP = 4.0
            N = I
            CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $           PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $           DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $           AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $           SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

            IF (OKFIN .EQ. 1) THEN
               CALL PRINT4 (J,J4,PICKS,PICKD)
            END IF

         ELSE IF (M3 .EQ. 1) THEN

            PICKD = (I + 2.0) * 2.0
            PICKS = 1.0
            PICKP = 4.0
            N = I
            CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $           PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $           DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $           AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4.AL6,AL8,TRE,TF,
     $           SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

            IF (OKFIN .EQ. 1) THEN
               CALL PRINT4 (J,J4,PICKS,PICKD)
            END IF

         END IF

   62    CONTINUE

         IF (J4 .EQ. 0) THEN
            PRINT*,' '
            PRINT*,'NO 4-PASS HEAT EXCHANGER CONFIGURATIONS
            PRINT*,'WILL MEET THE PROBLEM SPECIFICATIONS.'
```

```
      END IF

60    CONTINUE

      IF (J2 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'NO 2-PASS HEAT EXCHANGER CONFIGURATIONS'
         PRINT*,'WILL MEET THE PROBLEM SPECIFICATIONS.'
      END IF

      PRINT*,' '

***   ALL AVAILABLE 4-PASS CONFIGURATIONS ARE CHECKED

61    DO 62, I = 1,28

      K4 = 0
      M1 = 0
      M2 = 0
      M3 = 0

***   FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
***   RECOMMENDED MINIMUMS

      IF(MT .GE. QTL4(I) .AND. MT .LE. QTH4(I)) THEN
         K4 = 1
      END IF

      IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
         M1 = 1
      END IF

      IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
         M2 = 1
      END IF

      IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
         M3 = 1
      END IF

      IF(K4 .EQ. 0) THEN
         GO TO 62
      END IF

      IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
         GO TO 62
      END IF

***   A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
***   THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

      IF (M1 .EQ. 1) THEN
```

```
      IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
         GO TO 60
      END IF

***   A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
***   THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

      IF (M1 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.2
         PICKP = 2.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT.DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT2 (J,J2,PICKS,PICKD)
         END IF

      ELSE IF (M2 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 0.45
         PICKP = 2.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP.AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT2 (J,J2,PICKS,PICKD)
         END IF

      ELSE IF (M3 .EQ. 1) THEN

         PICKD = (I + 2.0) * 2.0
         PICKS = 1.0
         PICKP = 2.0
         N = I
         CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI.DTOT,DENS.DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS.CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

         IF (OKFIN .EQ. 1) THEN
            CALL PRINT2 (J,J2,PICKS,PICKD)
         END IF
```

```
         GO TO 4
      END IF

   PRINT*,' '
702   PRINT*,'DO YOU WANT TO CHANGE THE TUBING MATERIAL ?'
   PRINT*,' '

   READ (5,101) AMATE

      IF (AMATE .EQ. 'Y') THEN
         GO TO 703
      END IF

   PRINT*,' '
704   PRINT*,'DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?'
   PRINT*,' '

   READ (5,101) ATFLU

      IF (ATFLU .EQ. 'Y') THEN
         GO TO 705
      END IF

   PRINT*,' '
706   PRINT*,'DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?'
   PRINT*,' '

   READ (5,101) ASFLU

      IF (ASFLU .EQ. 'Y') THEN
         GO TO 707
      END IF

   PRINT*,' '
708   PRINT*,'DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?'
   PRINT*,' '

   READ (5,101) ATTEM

      IF (ATTEM .EQ. 'Y') THEN
         GO TO 709
      END IF

   PRINT*,' '
710   PRINT*,'DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?'
   PRINT*,' '

   READ (5,101) ATGPM

      IF (ATGPM .EQ. 'Y') THEN
         GO TO 711
      END IF

   PRINT*,' '
```

```
712   PRINT*,'DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?'
      PRINT*,' '

      READ (5,101) ASTEM

         IF (ASTEM .EQ. 'Y') THEN
            GO TO 713
         END IF

      PRINT*,' '
714   PRINT*,'DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?'
      PRINT*,' '

      READ (5,101) ASGPM

         IF (ASGPM .EQ. 'Y') THEN
            GO TO 715
         END IF

      PRINT*,' '
716   PRINT*,'DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR'
      PRINT*,'NOT A FOULING FACTOR IS USED ?'
      PRINT*,' '

      READ (5,101) AFFAC

      PRINT*,' '
      PRINT*,'DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?'
      PRINT*,' '

      READ (5,101) ALREST

      PRINT*,' '
      PRINT*,'DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE'
      PRINT*,'PRESSURE LOSSES ?'
      PRINT*,' '

      READ (5,101) APRES

      GO TO 3

      END IF

      PRINT*,' '
800   PRINT*,'DO YOU WANT TO DO ANY OTHER HEAT EXCHANGER '
      PRINT*,'PROBLEMS ? TYPE Y FOR YES AND N FOR NO.'

      READ (5,101) ACALC

      IF (ACALC .EQ. 'N') THEN
         GO TO 999
      END IF

      IF (ACALC .EQ. 'Y') THEN
```

```
         AGAIN = 'N'
         AFFAC = 'Y'
         ALREST = 'Y'
         APRES = 'Y'
         GO TO 4
      END IF

999   STOP'DONE'

      END
```

APPENDIX C

FLUID PROPERTY SUBROUTINES (WATER ONLY)

```
      SUBROUTINE DWATER(TAVE,DEN)
***   THIS SUBROUTINE CALCULATES THE DENSITY OF WATER AT
***   THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
***   FROM 32.0 DEF  F TO 600 DEG F. UNITS ARE LBM/CUBIC FT


      IF (TAVE .LE. 32.0) THEN
         DEN = 62.4
         GO TO 100
      END IF

      IF (TAVE .LE. 50.0) THEN
         DEN = 62.4
         GO TO 100
      END IF

      IF (TAVE .EQ. 100.0) THEN
         DEN = 62.0
         GO TO 100
      END IF

      IF (TAVE .LT. 100.0) THEN
         T = 50.0
         T1 = 100.0
         D = 62.4
         D1 = 62.0
         FRAC = (100.0 - TAVE) / (T1 - T) * (D - D1)
         DEN = D1 + FRAC
         GO TO 100
      END IF

      IF (TAVE .EQ. 150.0) THEN
         DEN = 61.2
         GO TO 100
      END IF

      IF (TAVE .LT. 150.0) THEN
         T = 100.0
         T1 = 150.0
         D = 62.0
         D1 = 61.2
         FRAC = (150.0 - TAVE) / (T1 - T) * (D - D1)
         DEN = D1 + FRAC
         GO TO 100
      END IF

      IF (TAVE .EQ. 200.0) THEN
         DEN = 60.1
         GO TO 100
      END IF

      IF (TAVE .LT. 200.0) THEN
         T = 150.0
         T1 = 200.0
```

```
      D = 61.2
      D1 = 60.1
      FRAC = (200.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
   END IF

   IF (TAVE .EQ. 250.0) THEN
      DEN = 58.8
      GO TO 100
   END IF

   IF (TAVE .LT. 250.0) THEN
      T = 200.0
      T1 = 250.0
      D = 60.1
      D1 = 58.8
      FRAC = (250.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
   END IF

   IF (TAVE .EQ. 300.0) THEN
      DEN = 57.3
      GO TO 100
   END IF

   IF (TAVE .LT. 300.0) THEN
      T = 250.0
      T1 = 300.0
      D = 58.8
      D1 = 57.3
      FRAC = (300.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
   END IF

   IF (TAVE .EQ. 350.0) THEN
      DEN = 55.6
      GO TO 100
   END IF

   IF (TAVE .LT. 350.0) THEN
      T = 300.0
      T1 = 350.0
      D = 57.3
      D1 = 55.6
      FRAC = (350.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
   END IF

   IF (TAVE .EQ. 400.0) THEN
      DEN = 53.6
```

```
   GO TO 100
END IF

IF (TAVE .LT. 400.0) THEN
   T = 350.0
   T1 = 400.0
   D = 55.6
   D1 = 53.6
   FRAC = (400.0 - TAVE) / (T1 - T) * (D - D1)
   DEN = D1 + FRAC
   GO TO 100
END IF

IF (TAVE .EQ. 450.0) THEN
   DEN = 51.6
   GO TO 100
END IF

IF (TAVE .LT. 450.0) THEN
   T = 400.0
   T1 = 450.0
   D = 53.6
   D1 = 51.6
   FRAC = (450.0 - TAVE) / (T1 - T) * (D - D1)
   DEN = D1 + FRAC
   GO TO 100
END IF

IF (TAVE .EQ. 500.0) THEN
   DEN = 49.0
   GO TO 100
END IF

IF (TAVE .LT. 500.0) THEN
   T = 450.0
   T1 = 500.0
   D = 51.6
   D1 = 49.0
   FRAC = (500.0 - TAVE) / (T1 - T) * (D - D1)
   DEN = D1 + FRAC
   GO TO 100
END IF

IF (TAVE .EQ. 550.0) THEN
   DEN = 45.9
   GO TO 100
END IF

IF (TAVE .LT. 550.0) THEN
   T = 500.0
   T1 = 550.0
   D = 49.0
   D1 = 45.9
   FRAC = (550.0 - TAVE) / (T1 - T) * (D - D1)
```

```
         DEN = D1 + FRAC
         GO TO 100
      END IF

      IF (TAVE .EQ. 600.0) THEN
         DEN = 42.4
         GO TO 100
      END IF

      IF (TAVE .LT. 600.0) THEN
         T = 550.0
         T1 = 600.0
         D = 45.9
         D1 = 42.4
         FRAC = (600.0 - TAVE) / (T1 - T) + (D - D1)
         DEN = D1 + FRAC
         GO TO 100
      END IF

      IF (TAVE .GT. 600.0) THEN
         DEN = 42.4
      END IF

100   RETURN
      END
```

```
      SUBROUTINE CPWATR(TAVE,CP)
***   THIS SUBROUTINE CALCULATES THE SPECIFIC HEAT OF WATER
***   AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
***   FROM 32.0 DEG F TO 600 DEG F. UNITS ARE BTU/LBM-DEG F

      IF (TAVE .LE. 32.0) THEN
         CP = 1.01
         GO TO 101
      END IF

      IF (TAVE .EQ. 50.0) THEN
         CP = 1.00
         GO TO 101
      END IF

      IF (TAVE .LT. 50.0) THEN
         T = 32.0
         T1 = 50.0
         C = 1.01
         C1 = 1.00
         FRAC = (50.0 - TAVE) / (T1 - T) * (C - C1)
         CP = C1 + FRAC
         GO TO 101
      END IF

      IF (TAVE .EQ. 100.0) THEN
         CP = 0.998
         GO TO 101
      END IF

      IF (TAVE .LT. 100.0) THEN
         T = 50.0
         T1 = 100.0
         C = 1.00
         C1 = 0.998
         FRAC = (100.0 - TAVE) / (T1 - T) * (C - C1)
         CP = C1 + FRAC
         GO TO 101
      END IF

      IF (TAVE .EQ. 150.0) THEN
         CP = 1.00
         GO TO 101
      END IF

      IF (TAVE .LT. 150.0) THEN
         T = 100.0
         T1 = 150.0
         C = 0.998
         C1 = 1.00
         FRAC = (150.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
         GO TO 101
      END IF
```

```
IF (TAVE .LE. 200.0) THEN
   CP = 1.00
   GO TO 101
END IF

IF (TAVE .EQ. 250.0) THEN
   CP = 1.01
   GO TO 101
END IF

IF (TAVE .LT. 250.0) THEN
   T = 200.0
   T1 = 250.0
   C = 1.00
   C1 = 1.01
   FRAC = (250.0 - TAVE) / (T1 - T) * (C1 - C)
   CP = C1 - FRAC
   GO TO 101
END IF

IF (TAVE .EQ. 300.0) THEN
   CP = 1.03
   GO TO 101
END IF

IF (TAVE .LT. 300.0) THEN
   T = 250.0
   T1 = 300.0
   C = 1.01
   C1 = 1.03
   FRAC = (300.0 - TAVE) / (T1 - T) * (C1 - C)
   CP = C1 - FRAC
   GO TO 101
END IF

IF (TAVE .EQ. 350.0) THEN
   CP = 1.05
   GO TO 101
END IF

IF (TAVE .LT. 350.0) THEN
   T = 300.0
   T1 = 350.0
   C = 1.03
   C1 = 1.05
   FRAC = (350.0 - TAVE) / (T1 - T) * (C1 - C)
   CP = C1 - FRAC
   GO TO 101
END IF

IF (TAVE .EQ. 400.0) THEN
   CP = 1.08
   GO TO 101
```

```
      END IF

      IF (TAVE .LT. 400.0) THEN
         T = 350.0
         T1 = 400.0
         C = 1.05
         C1 = 1.08
         FRAC = (400.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
         GO TO 101
      END IF

      IF (TAVE .EQ. 450.0) THEN
         CP = 1.12
         GO TO 101
      END IF

      IF (TAVE .LT. 450.0) THEN
         T = 400.0
         T1 = 450.0
         C = 1.08
         C1 = 1.12
         FRAC = (450.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
         GO TO 101
      END IF

      IF (TAVE .EQ. 500.0) THEN
         CP = 1.19
         GO TO 101
      END IF

      IF (TAVE .LT. 500.0) THEN
         T = 450.0
         T1 = 500.0
         C = 1.12
         C1 = 1.19
         FRAC = (500.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
         GO TO 101
      END IF

      IF (TAVE .EQ. 550.0) THEN
         CP = 1.31
         GO TO 101
      END IF

      IF (TAVE .LT. 550.0) THEN
         T = 500.0
         T1 = 550.0
         C = 1.19
         C1 = 1.31
         FRAC = (550.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
```

```
         GO TO 101
      END IF

      IF (TAVE .EQ. 600.0) THEN
         CP = 1.51
         GO TO 101
      END IF

      IF (TAVE .LT. 600.0) THEN
         T = 550.0
         T1 = 600.0
         C = 1.31
         C1 = 1.51
         FRAC = (600.0 - TAVE) / (T1 - T) * (C1 - C)
         CP = C1 - FRAC
         GO TO 101
      END IF

      IF (TAVE .GT. 600.0) THEN
         CP = 1.51
      END IF

101   RETURN
      END
```

```
      SUBROUTINE KWATER(TAVE,K)
***   THIS SUBROUTINE CALCULATES THE THERMAL CONDUCTIVITY OF
***   WATER AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
***   FROM 32.0 DEG F TO 600 DEF F.UNITS ARE BTU/HR-FT-DEG F

      REAL K, K1

      IF (TAVE .LE. 32.0) THEN
         K = 0.319
         GO TO 107
      END IF

      IF (TAVE .EQ. 50.0) THEN
         K = 0.332
         GO TO 107
      END IF

      IF (TAVE .LT. 50.0) THEN
         T = 32.0
         T1 = 50.0
         K = 0.319
         K1 = 0.332
         FRAC = (50.0 - TAVE) / (T1 -T) * (K1 - K)
         K = K1 - FRAC
         GO TO 107
      END IF

      IF (TAVE .EQ. 100.0) THEN
         K = 0.364
         GO TO 107
      END IF

      IF (TAVE .LT. 100.0) THEN
         T = 50.0
         T1 = 100.0
         K = 0.332
         K1 = 0.364
         FRAC = (100.0 - TAVE) / (T1 -T) * (K1 - K)
         K = K1 - FRAC
         GO TO 107
      END IF

      IF (TAVE .EQ. 150.0) THEN
         K = 0.384
         GO TO 107
      END IF

      IF (TAVE .LT. 150.0) THEN
         T = 100.0
         T1 = 150.0
         K = 0.364
         K1 = 0.384
         FRAC = (150.0 - TAVE) / (T1 -T) * (K1 - K)
         K = K1 - FRAC
```

```
      GO TO 107
   END IF

   IF (TAVE .EQ. 200.0) THEN
      K = 0.394
      GO TO 107
   END IF

   IF (TAVE .LT. 200.0) THEN
      T = 150.0
      T1 = 200.0
      K = 0.384
      K1 = 0.394
      FRAC = (200.0 - TAVE) / (T1 -T) * (K1 - K)
      K = K1 - FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 250.0) THEN
      K = 0.396
      GO TO 107
   END IF

   IF (TAVE .LT. 250.0) THEN
      T = 200.0
      T1 = 250.0
      K = 0.394
      K1 = 0.396
      FRAC = (250.0 - TAVE) / (T1 -T) * (K1 - K)
      K = K1 - FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 300.0) THEN
      K = 0.395
      GO TO 107
   END IF

   IF (TAVE .LT. 300.0) THEN
      T = 250.0
      T1 = 300.0
      K = 0.396
      K1 = 0.395
      FRAC = (300.0 - TAVE) / (T1 -T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 350.0) THEN
      K = 0.391
      GO TO 107
   END IF

   IF (TAVE .LT. 350.0) THEN
```

APPENDIX E

TFRICT SUBROUTINE

```
      PLTTOT = PLTF + PLTR

***   CALCULATE THE SHELL-SIDE FLUID FRICTION FACTOR WHICH
***   DEPENDS ON THE REYNOLDS NUMBER TAKEN FROM A CORRELATION
***   OF FRICTION FACTORS FOR TUBE BUNDLES WITH 25% CUT
***   SEGMENTAL BAFFLES FROM KERN PAGE 838

      CALL SFRICT(RES,SRE,SF,FRS)

***   CALCULATE THE SHELL-SIDE PRESSURE LOSS VIA EQUATION
***   FROM KERN ON PAGE 147

      PLSTOT = (FRS * L * ( GS**2.0 ) *
     $   (INT( (L*12.0) / (DSI(N) * PICKS) ) + 1.0 ) * 0.006944)
     $   / ( 2.0 * AG * DENS *(DEQS/12.0)* ((DVS/DVSF) ** 0.14))

***   CHECK TO SEE IF THE PRESSURE LOSSES MEET THE RESTRICTIONS
***   IMPOSED BY THE USER

      IF (PLTTOT .LT. AMAXTP) THEN
         OKT = 1
      ELSE
         OKT = 0
      END IF

      IF (PLSTOT .LT. AMAXSP) THEN
         OKS = 1
      ELSE
         OKS = 0
      END IF

      IF (OKL .EQ. 1 .AND. OKT .EQ. 1 .AND. OKS .EQ. 1) THEN
         OKFIN = 1
      ELSE
         OKFIN = 0
      END IF

      RETURN
      END
```

```
***   CALCULATE THE LENGTH OF TUBING OF THE DESIRED HEAT
***   EXCHANGER CONFIGURATION.

      IF (PICKP .EQ. 2.0) THEN
         L = ACORR / AL2(N)
      END IF

      IF (PICKP .EQ. 4.0) THEN
         L = ACORR / AL4(N)
      END IF

      IF (PICKP .EQ. 6.0) THEN
         L = ACORR / AL6(N)
      END IF

      IF (PICKP .EQ. 8.0) THEN
         L = ACORR / AL8(N)
      END IF

***   CHECK TO SEE IF THE ACTUAL DESIGN LENGTH IS LESS THAN THE
***   LENGTH RESTRICTION IMPOSED BY THE USER.

      IF (L .LT. LRESTR) THEN
         OKL = 1
      ELSE
         OKL = 0
      END IF

      NP = PICKP
      AG = 32.1740

***   CALCULATE THE TUBE-SIDE FLUID FRICTION FACTOR WHICH
***   DEPENDS ON THE REYNOLDS NUMBER USING SEIDER AND TATE
***   FRICTION FACTOR CORRELATIONS FROM PAGE 836 OF KERN

      CALL TFRICT(RET,TRE,TF,FRT)

***   CALCULATE THE PRESSURE LOSSES IN THE TUBE DUE TO BOTH
***   THE FRICTION LOSSES AND THE RETURN LOSSES USING
***   EQUATIONS FROM PAGE 148 AND 836 OF KERN. NOTE THAT
***   .006944 IS A CONVERSION FACTOR FOR PSF TO PSI AND
***   .4335 IS A CONVERSION FACTOR FOR FEET OF WATER TO PSI

      TAVE = TTAVE
      CALL DWATER(TAVE,DEN)
      DENW = DEN

      PLTF = (( FRT*NP*L* (GT**2.0) ) / ( (DTIN/12.0)* 2.0 * AG
     $   * DENT * ((DVT/DVTF)**0.14))) * 0.006944

      PLTR = (( 2.0 * NP * (VT**2.0)) / (2.0 * AG *
     $   (DENT/DENW) )) * 0.4335

***   CALCULATE THE TOTAL PRESSURE LOSS IN THE TUBES
```

```
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VDOWA (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 5.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VKERO (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 6.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VSAE (TAVE,DV)
            DVTF = DV
         END IF

         IF(PICKFS .EQ. 7.0) THEN
            TAVE = (TTAVE + TSAVE) / 2.0
            CALL VEGLY (TAVE,DV)
            DVTF = DV
         END IF

         HI = (KT/(DTIN/12.0)) * 0.027 * (RET**0.8) *
     $      ((CPT * DVT * 3600.0/KT)**0.3333) * ((DVT/DVTF)**0.14)

***      CALCULATE THE TUBE WALL RESISTANCE FROM EQUATION FOUND IN
***      TEMA STANDARDS HANDBOOK P 137.

         RW = (DTOT/(24.0*KW)) * LOG (DTOT/(DTOT - 2.0*(DTOT-DTIN)))

***      CALCULATE THE AO/AI RATIO WHICH REDUCES TO DO/DI

         TRATIO = DTOT / DTIN

***      CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT

         IF (AFOUL .EQ. 'Y') THEN

            UCALC = 1.0/((1.0/HO) + RO + RW + (RI*TRATIO)
     $                 + ((1.0/HI)*TRATIO))

         ELSE IF (AFOUL .EQ. 'N') THEN

            UCALC = 1.0/((1.0/HO) + RW + ((1.0/HI)*TRATIO))

         END IF

***      USING THE CALCULATED VALUE OF THE OVERALL HEAT TRANSFER
***      COEFFICIENT, CALCULATE THE REQUIRED HEAT TRANSFER AREA.

         ACORR = (NTU * CMIN) / UCALC

***      USING THE CORRECT VALUE OF REQUIRED HEAT TRANSFER AREA,
```

```
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VGAS (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 4.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VDOWA (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 5.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VKERO (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 6.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VSAE (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 7.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VEGLY (TAVE,DV)
         DVSF = DV
      END IF

      HO = (KS/(DEQS/12.0)) * 0.36 * (RES**0.55) *
     $    ((CPS * DVS * 3600.0/KS)**0.3333) * ((DVS/DVSF)**0.14)

***   FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
***   CORPORATION P 136

      IF(PICKFS .EQ. 1.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VWATER (TAVE,DV)
         DVTF = DV
      END IF

      IF(PICKFS .EQ. 2.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VMETH (TAVE,DV)
         DVTF = DV
      END IF

      IF(PICKFS .EQ. 3.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VGAS (TAVE,DV)
         DVTF = DV
      END IF

      IF(PICKFS .EQ. 4.0) THEN
```

```
***   PROCESS HEAT TRANSFER ON PAGE 139. THE EQUATION IS FOR A
***   TRIANGULAR PITCH ARRANGEMENT ONLY. UNITS ARE IN INCHES.

      DEQS=(4.0*((.5*TPH)*(.86*TPH)-((.5*PI*(DTOT**2.))/4.0)))
     $   /(.5*DTOT)

***   THE VELOCITY OF THE SHELL FLUID IS CALCULATED. UNITS
***   ARE LB/SEC-SQUARE FEET

      GS = VS * DENS

***   THE REYNOLDS NUMBER OF THE SHELL FLUID IS CALCULATED

      RES = ( (DEQS / 12.0) * GS ) / DVS

***   THE MASS VELOCITY OF THE TUBE FLUID IS CALCULATED. UNITS
***   ARE LB/SEC-SQUARE FEET

      GT = VT * DENT

***   THE REYNOLDS NUMBER OF THE TUBE FLUID IS CALCULATED

      RET = ( (DTIN / 12.0) * GT) / DVT

***   NOW THAT THE CONFIGURATION HAS BEEN CHOSEN, CALCULATE THE
***   OVERALL U VALUE FOR THE HEAT EXCHANGER USING GENERALIZED
***   CORRELATIONS FOUND IN THE REFERENCES.

***   FROM TEMA PAGE 137, THE OVERALL HEAT TRANSFER COEFFICIENT
***   REQUIRES THE EVALUATION OF:
***   HO : THE FILM COEFFICIENT OF THE FLUID OUTSIDE THE TUBES
***   HI : THE FILM COEFFICIENT OF T8HE FLUID INSIDE THE TUBES
***   RO : THE FOULING RESISTANCE ON THE OUTSIDE OF THE TUBES
***   RI : THE FOULING RESISTANCE ON THE INSIDE OF THE TUBES
***   RW : THE RESISTANCE OF THE TUBE WALL REFERRED TO THE
***        OUTSIDE SURFACE OF THE TUBE WALL
***   AO/AI : THE RATIO OF OUTSIDE TO INSIDE SURFACE OF TUBING

***   FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
***   CORPORATION P 157

      IF(PICKFS .EQ. 1.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VWATER (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 2.0) THEN
         TAVE = (TTAVE + TSAVE) / 2.0
         CALL VMETH (TAVE,DV)
         DVSF = DV
      END IF

      IF(PICKFS .EQ. 3.0) THEN
```

```
      SUBROUTINE PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
     $        PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
     $        DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
     $        AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
     $        SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

***   THIS SUBROUTINE DOES A PRESSURE CHECK ON BOTH THE TUBE
***   AND THE SHELL SIDES, AND ALSO DOES A LENGTH RESTRICTION
***   CHECK. IF THE SUBROUTINE RETURNS THE VALUE OF OKFIN = 1
***   THE CONFIGURATION MET THE CRITERIA. A VALUE OF OKFIN = 0
***   MEANS THAT THE CONFIGURATION FAILED ONE OR MORE OF THE
***   DESIGN CRITERIA IMPOSED BY THE USER.

      DIMENSION DSIU(28),DSI(28),NT2(28),NT4(28),NT6(28),
     $  NT8(28),AL2(28),AL4(28),AL6(28),AL8(28),TRE(34),
     $  TF(34),SRE(34),SF(34)

      CHARACTER*1 AFOUL
      REAL NTU,MT,MS,LRESTR,NP,KS,KT,KW,L

      N = INT( (PICKD/2.0) - 2.0 )

***   FOR EACH HEAT EXCHANGER CONFIGURATION , THE PROGRAM CAN NOW
***   THE ACTUAL SHELL AND TUBE FLUID VELOCITIES, AND DO A LENGTH
***   RESTRICTION CHECK, A SHELL PRESSURE LOSS CHECK, AND A TUBE
***   PRESSURE LOSS CHECK.
***   NOTE THAT C IS A CONVERSION FACTOR TO GET VELOCITY UNITS IN
***   FT/SEC BY INPUTTING A FLOW RATE OF GALLONS PER MINUTE AND
***   HAVING AREA IN INCHES.

      C = .3208342

      ASC = (DSIU(N)/TPH) * TC * (PICKS * DSI(N))

      VS = C * MS/ASC

      IF (PICKP .EQ. 2.0) THEN
         VT = (C*MT) / (ATIN * (NT2(N)/2.0) )
      END IF

      IF (PICKP .EQ. 4.0) THEN
         VT = (C*MT) / (ATIN * (NT4(N)/4.0) )
      END IF

      IF (PICKP .EQ. 6.0) THEN
         VT = (C*MT) / (ATIN * (NT6(N)/6.0) )
      END IF

      IF (PICKP .EQ. 8.0) THEN
         VT = (C*MT) / (ATIN * (NT8(N)/8.0) )
      END IF

***   CALCULATE THE EQUIVALENT DIAMETER OF THE FLUID CONDUIT
***   CROSSING THE TUBES USING EQUATIONS FROM KERN IN HIS BOOK
```

APPENDIX D

PRESCK SUBROUTINE

```
         DV = 0.000064
         GO TO 102
      END IF

      IF (TAVE .LT. 550.0) THEN
         T = 500.0
         T1 = 550.0
         V = 0.000071
         V1 = 0.000064
         FRAC = (550.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 600.0) THEN
         DV = 0.000058
         GO TO 102
      END IF

      IF (TAVE .LT. 600.0) THEN
         T = 550.0
         T1 = 600.0
         V = 0.000064
         V1 = 0.000058
         FRAC = (600.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .GT. 600.0) THEN
         DV = 0.000058
      END IF

102   RETURN
      END
```

```
      T1 = 350.0
      V = 0.000126
      V1 = 0.000105
      FRAC = (350.0 - TAVE) / (T1 - T) * (V - V1)
      DV = V1 + FRAC
      GO TO 102
   END IF

   IF (TAVE .EQ. 400.0) THEN
      DV = 0.000091
      GO TO 102
   END IF

   IF (TAVE .LT. 400.0) THEN
      T = 350.0
      T1 = 400.0
      V = 0.000105
      V1 = 0.000091
      FRAC = (400.0 - TAVE) / (T1 - T) * (V - V1)
      DV = V1 + FRAC
      GO TO 102
   END IF

   IF (TAVE .EQ. 450.0) THEN
      DV = 0.000080
      GO TO 102
   END IF

   IF (TAVE .LT. 450.0) THEN
      T = 400.0
      T1 = 450.0
      V = 0.000091
      V1 = 0.000080
      FRAC = (450.0 - TAVE) / (T1 - T) * (V - V1)
      DV = V1 + FRAC
      GO TO 102
   END IF

   IF (TAVE .EQ. 500.0) THEN
      DV = 0.000071
      GO TO 102
   END IF

   IF (TAVE .LT. 500.0) THEN
      T = 450.0
      T1 = 500.0
      V = 0.000080
      V1 = 0.000071
      FRAC = (500.0 - TAVE) / (T1 - T) * (V - V1)
      DV = V1 + FRAC
      GO TO 102
   END IF

   IF (TAVE .EQ. 550.0) THEN
```

```
      END IF

      IF (TAVE .EQ. 200.0) THEN
         DV = 0.000205
         GO TO 102
      END IF

      IF (TAVE .LT. 200.0) THEN
         T = 150.0
         T1 = 200.0
         V = 0.000292
         V1 = 0.000205
         FRAC = (200.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 250.0) THEN
         DV = 0.000158
         GO TO 102
      END IF

      IF (TAVE .LT. 250.0) THEN
         T = 200.0
         T1 = 250.0
         V = 0.000205
         V1 = 0.000158
         FRAC = (250.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 300.0) THEN
         DV = 0.000126
         GO TO 102
      END IF

      IF (TAVE .LT. 300.0) THEN
         T = 250.0
         T1 = 300.0
         V = 0.000158
         V1 = 0.000126
         FRAC = (300.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 350.0) THEN
         DV = 0.000105
         GO TO 102
      END IF

      IF (TAVE .LT. 350.0) THEN
         T = 300.0
```

```
      SUBROUTINE VWATER(TAVE,DV)
***   THIS SUBROUTINE CALCULATES THE DYNAMIC VISCOSITY OF
***   WATER AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
***   FROM 32.0 DEG F TO 600 DEG F. UNITS ARE LB/FT-SEC


      IF (TAVE .LE. 32.0) THEN
         DV = 0.0012
         GO TO 102
      END IF

      IF (TAVE .EQ. 50.0) THEN
         DV = 0.00088
         GO TO 102
      END IF

      IF (TAVE .LT. 50.0) THEN
         T = 32.0
         T1 = 50.0
         V = 0.0012
         V1 = 0.00088
         FRAC = (50.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 100.0) THEN
         DV = 0.000458
         GO TO 102
      END IF

      IF (TAVE .LT. 100.0) THEN
         T = 50.0
         T1 = 100.0
         V = 0.00088
         V1 = 0.000458
         FRAC = (100.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
      END IF

      IF (TAVE .EQ. 150.0) THEN
         DV = 0.000292
         GO TO 102
      END IF

      IF (TAVE .LT. 150.0) THEN
         T = 100.0
         T1 = 150.0
         V = 0.000458
         V1 = 0.000292
         FRAC = (150.0 - TAVE) / (T1 - T) * (V - V1)
         DV = V1 + FRAC
         GO TO 102
```

```
      IF (TAVE .EQ. 550.0) THEN
         K = 0.325
         GO TO 107
      END IF

      IF (TAVE .LT. 550.0) THEN
         T = 500.0
         T1 = 550.0
         K = 0.349
         K1 = 0.325
         FRAC = (550.0 - TAVE) / (T1 -T) * (K - K1)
         K = K1 + FRAC
         GO TO 107
      END IF

      IF (TAVE .EQ. 600.0) THEN
         K = 0.292
         GO TO 107
      END IF

      IF (TAVE .LT. 600.0) THEN
         T = 550.0
         T1 = 600.0
         K = 0.325
         K1 = 0.292
         FRAC = (600.0 - TAVE) / (T1 -T) * (K - K1)
         K = K1 + FRAC
         GO TO 107
      END IF

      IF (TAVE .GT. 600.0) THEN
         K = 0.292
      END IF

107   RETURN
      END
```

```
      T = 300.0
      T1 = 350.0
      K = 0.395
      K1 = 0.391
      FRAC = (350.0 - TAVE) / (T1 -T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 400.0) THEN
      K = 0.381
      GO TO 107
   END IF

   IF (TAVE .LT. 400.0) THEN
      T = 350.0
      T1 = 400.0
      K = 0.391
      K1 = 0.381
      FRAC = (400.0 - TAVE) / (T1 -T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 450.0) THEN
      K = 0.367
      GO TO 107
   END IF

   IF (TAVE .LT. 450.0) THEN
      T = 400.0
      T1 = 450.0
      K = 0.381
      K1 = 0.367
      FRAC = (450.0 - TAVE) / (T1 -T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
   END IF

   IF (TAVE .EQ. 500.0) THEN
      K = 0.349
      GO TO 107
   END IF

   IF (TAVE .LT. 500.0) THEN
      T = 450.0
      T1 = 500.0
      K = 0.367
      K1 = 0.349
      FRAC = (500.0 - TAVE) / (T1 -T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
   END IF
```

```
      SUBROUTINE TFRICT(RET,TRE,TF,FRT)

***   THIS SUBROUTINE CALCULATES THE TUBE-SIDE FRICTION FACTOR
***   FOR USE IN THE DARCY-FANNING PRESSURE LOSS EQUATION FOR
***   THE TUBE-SIDE PRESSURE DROP DUE TO FRICTION. THE FACTOR
***   IS INTERPOLATED FROM VALUES TAKEN FROM A GRAPH OF TUBE-
***   SIDE FRICTION FACTORS PUBLISHED BY TEMA. THE FRICTION
***   FACTOR IS A DIMENSIONLESS PARAMETER.

      DIMENSION TRE(34),TF(34)

      I = 1
      IF(RET .LE. TRE(I)) THEN
         FRT = TF(I)
         GO TO 104
      END IF

      DO 103, I= 2,34
      IF(RET .LT. TRE(I)) THEN
         FRAC = (TRE(I)-RET)/(TRE(I)-TRE(I-1))*(TF(I-1)-TF(I))
         FRT = TF(I-1) - FRAC
         GO TO 104
      END IF

      IF(RET .EQ. TRE(I)) THEN
         FRT = TF(I)
         GO TO 104
      END IF

103   CONTINUE

      PRINT*,' THE TUBE-SIDE FLUID REYNOLDS NUMBER IS OUT OF'
      PRINT*,'RANGE OF THE PROGRAM. AN AVERAGE FRICTION'
      PRINT*,'FACTOR OF 0.035 WILL BE USED IN  CALCULATIONS.'

      FRT = 0.035

104   RETURN
      END
```

APPENDIX F

SFRICT SUBROUTINE

```
      SUBROUTINE SFRICT(RES,SRE,SF,FRS)

***   THIS SUBROUTINE CALCULATES THE SHELL-SIDE FRICTION FACTOR
***   FOR USE IN THE DARCY-FANNING PRESSURE LOSS EQUATION FOR
***   THE SHELL-SIDE PRESSURE DROP DUE TO FRICTION. THE FACTOR
***   IS INTERPOLATED FROM VALUES TAKEN FROM A GRAPH OF SHELL-
***   SIDE FRICTION FACTORS PUBLISHED BY KERN IN HIS BOOK PROCESS
***   HEAT TRANSFER. THE FRICTION FACTOR IS A DIMENSIONLESS
***   PARAMETER.

      DIMENSION SRE(34),SF(34)

      I = 1
      IF(RES .LE. SRE(I)) THEN
         FRS = SF(I)
         GO TO 106
      END IF

      DO 105, I= 2,34
      IF(RES .LT. SRE(I)) THEN
         FRAC = (SRE(I)-RES)/(SRE(I)-SRE(I-1))*(SF(I-1)-SF(I))
         FRS = SF(I-1) - FRAC
         GO TO 106
      END IF

      IF(RES .EQ. SRE(I)) THEN
         FRS = SF(I)
         GO TO 106
      END IF

105   CONTINUE

      PRINT*,' THE SHELL FLUID REYNOLDS NUMBER IS OUT OF'
      PRINT*,'RANGE OF THE PROGRAM. AN AVERAGE FRICTION'
      PRINT*,'FACTOR OF 0.30 WILL BE USED IN  CALCULATIONS.'

      FRS = 0.30

106   RETURN
      END
```

APPENDIX G

PRINT SUBROUTINES

```
      SUBROUTINE PRINT2(J,J2,PICKS,PICKD)

***   THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
***   CONFIGURATIONS FOR 2-PASS CONSTRUCTION



      IF (J .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J = 1
      END IF

      IF (J2 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE 2-PASS HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J2 = 1
      END IF

      PRINT*,'SHELL DIAMETER',PICKD,'  INCHES:'
      PRINT*,'WITH A ',PICKS,'  BAFFLE SPACING'

      RETURN
      END
```

```
      SUBROUTINE PRINT4(J,J4,PICKS,PICKD)

***   THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
***   CONFIGURATIONS FOR 4-PASS CONSTRUCTION



      IF (J .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J = 1
      END IF

      IF (J4 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE 4-PASS HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J4 = 1
      END IF

      PRINT*,'SHELL DIAMETER',PICKD,'  INCHES:'
      PRINT*,'WITH A ',PICKS,'  BAFFLE SPACING'

      RETURN
      END
```

```
      SUBROUTINE PRINT6(J,J6,PICKS,PICKD)


***   THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
***   CONFIGURATIONS FOR 6-PASS CONSTRUCTION


      IF (J .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J = 1
      END IF

      IF (J6 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE 6-PASS HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J6 = 1
      END IF

      PRINT*,'SHELL DIAMETER',PICKD,'  INCHES:'
      PRINT*,'WITH A ',PICKS,'  BAFFLE SPACING'

      RETURN
      END
```

```
      SUBROUTINE PRINT8(J,J8,PICKS,PICKD)

***   THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
***   CONFIGURATIONS FOR 8-PASS CONSTRUCTION


      IF (J .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J = 1
      END IF

      IF (J8 .EQ. 0) THEN
         PRINT*,' '
         PRINT*,'THE AVAILABLE 8-PASS HEAT EXCHANGER'
         PRINT*,'CONFIGURATIONS ARE AS FOLLOWS:'
         PRINT*,' '
         J8 = 1
      END IF

      PRINT*,'SHELL DIAMETER',PICKD,'  INCHES:'
      PRINT*,'WITH A ',PICKS,'  BAFFLE SPACING'

      RETURN
      END
```

END

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