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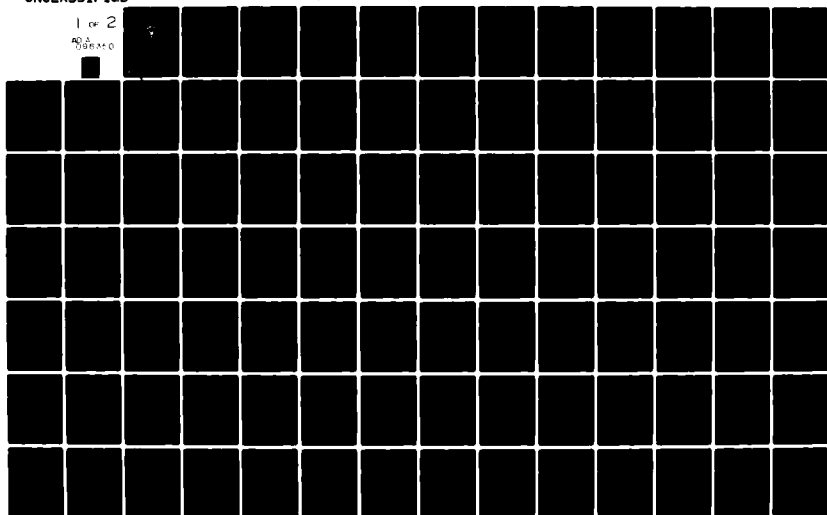
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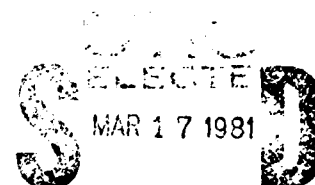
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NAVAL POSTGRADUATE SCHOOL
Monterey, California



THESIS

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HEAT EXCHANGER OPTIMIZATION

by

Conrad P. Hedderich

September 1980

Thesis Co-Advisors:

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The code is not limited to surfaces found in the literature, but will accommodate any triangular pitch bank of finned tubes in multiple-pass configurations.

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HEAT EXCHANGER OPTIMIZATION

by

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Lieutenant, United States Navy
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Submitted in partial fulfillment of the
requirements for the degree of

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ABSTRACT

A computer code was developed for the analysis of air-cooled heat exchangers and was coupled with a constrained function minimization code to produce an automated air-cooled heat exchanger design and optimization program with many new capabilities.

A general iteration free approximation method was used for the analysis which calculates the mean overall heat transfer coefficient and the overall pressure drop for many flow arrangements, taking into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of flow path.

The code is not limited to surfaces found in the literature, but will accommodate any triangular pitch bank of finned tubes in multiple-pass configurations.



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NOMENCLATURE

English Letter Symbols

- A - total heat transfer area, in^2
- c_p - specific heat, $\text{BTU/lbm-}^\circ\text{F}$
- $\dot{C}$ - heat capacity rate, $\text{BTU/hr-}^\circ\text{F} = \dot{m} c_p$
- D - diameter, in.
- f - friction factor
- F - LMTD correction factor
- g_c - acceleration of gravity, 32.2 ft/sec^2
- H - corrected heat transfer coefficient, $\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$
- h - bank height, in.
- J - Colburn factor
- k - thermal conductivity, $\text{BTU/hr-ft-}^\circ\text{F}$
- L - length, in.
- ℓ - fin height, in.
- $\dot{m}$ - mass flow rate, lbm/hr
- m - $\sqrt{2H_o/k_f t}$, ft^{-1}
- n - number of _____ (used with appropriate subscript)
- N - number of tubes
- p - pressure, psi
- P - pitch, in.
- Pr - Prandtl number
- $\dot{Q}$ - heat transfer rate, Btu/hr
- r - radius, in.
- R - heat transfer resistance, $\text{hr-ft}^2\text{-}^\circ\text{F/BTU}$

- Re - Reynolds number
- s - distance between adjacent fins, in.
- S - fin spacing center-to-center, in.
- t - fin thickness, in.
- T - true temperature, deg.
- U - overall heat transfer coefficient, BTU/hr-ft²-°F
- w - bank width, in.

Greek Letter Symbols

- ΔT - temperature difference, deg.
- η - surface efficiency
- ϕ - fin efficiency
- μ - viscosity, lbm/ft-hr
- ρ - density, lbm/ft³
- ψ - temperature correction

Subscripts

- a - air
- c - cold
- f - fin
- ff - free face
- h - hot
- i - inside
- j - reference number, I or II
- l - limiting
- L - longitudinal
- m - mean
- o - outside

p - pass
r - rows
t - transverse
T - total
w - water
x - cross-sectional (flow)
1 - entering
2 - leaving
 ∞ - ambient

Superscripts

' - uncorrected
* - estimated
i - initial

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I. INTRODUCTION

A. BACKGROUND

The cooling of fluids by passing ambient air over extended tube surfaces is a relatively recent development in heat exchangers. Its application has come about cautiously, due to the usual reluctance to change from well established and well documented methods, i.e. the shell-and-tube heat exchanger.

However, concern for the environment and economic pressures have necessitated the use of air as a coolant. Smith [1] has listed some typical advantages of direct cooling with air as compared to cooling with water in a shell-and-tube exchanger:

- a. Eliminates the problem of temperature rise in, and pollution of, water resources.
- b. Enables plant location to be independent of a water supply.
- c. Eliminates the necessity of much coolant piping.
- d. Reduces heat exchanger maintenance costs by eliminating the need of descaling water-side surfaces. The mechanical drives will operate in a noncorrosive atmosphere.
- e. Eliminates water treatment.
- f. Limiting coolant temperatures is unnecessary.
- g. Enables installation of exchangers at elevations above other operating equipment at no penalty, thus reducing ground area requirements.

With air colling becoming more and more competitive with water; even when water supplies are plentiful, an automated air-cooled heat exchanger design package (which could be used

for trade-off studies, first cut analysis, and conceptual design), would be of great use.

The design of an air-cooled cross flow heat exchanger is a complex task requiring the examination and optimization of a wide variety of heat transfer surfaces. Studies have shown that a poor choice of either the heat transfer surfaces or design parameters can more than double the costs chargeable to a heat exchanger [2].

For the optimized design of heat exchangers with the computer; reliable, but fast, calculation methods for the mean overall heat transfer coefficient and the overall pressure drops are needed for the following reasons:

- a. Conventional simple methods using mean values of temperatures as reference temperatures can lead to undesirable errors [3].
- b. Numerical stepwise integrations are prohibitively time consuming.

B. REVIEW

A number of heat exchanger design methods have been proposed to determine the optimum heat exchanger design. Bergles, et al. [4], performed an evaluation of different objective functions for compact heat exchangers with different heat transfer surfaces, but the same specifications. The method did not include any actual optimization techniques, but results did show that a great improvement in heat exchanger performance can be made by proper selection of design parameters.

The method of Fax and Mills [5], used Lagrange multipliers to optimize a heat exchanger design under specified constraints. This technique required that the objective function and constraints be expressed explicitly and be differentiable throughout the range of interest. The total number of constraints had to be less than the total number of variables, and all constraints had to be equality constraints. The method was obviously restricted to a very limited number of problems.

For a unit section of an air-cooled exchanger of standard length and width equipped with specified fans, both Schoonman [6], and Joyce [7], used factorial searches of fin spacing, number of rows and air rate to maximize the ratio of heat transfer to cost. Because the number of sections must be rounded to an integer, this method only appears suitable for large exchangers.

Nakayama [3], used a similar approach requiring the plotting of heat transfer coefficients and pressure drops which presumably could be programmed into a factorial search.

Kern [8] derived an analytic expression for the annual cost as a function of air rate and number of rows. (This was done by using a constant tubeside heat transfer coefficient and an arithmetic mean temperature difference.) The optimum was found by setting partial derivatives equal to zero. Alternatively, if the airside is assumed to control, the optimum allocation of the components of total cost might be

found through geometric programming, as illustrated by Auriel and Wilde [9]. Oshwald and Kochenberger [10], also presented a geometric programming method for heat exchanger optimization and used it to select heat exchanger fluids considering power requirements, cost, tube diameters, velocities, temperature, and other physical properties.

After discussing factorial, univariate, and random search methods for an optimum design of a shell-and-tube exchanger, Briggs and Evans [11, 12], discuss a "logical search method"; or what Peters and Nicole [13], call "heuristics". With this method, an engineer makes use of selecting design variables close to the optimum to obtain an optimum design. The heuristic method is less scientific, and is useful only when computer time and storage are at a premium. Due to the large number of discrete variables encountered in air-cooled heat exchanger design, Peters and Nicole [13], chose to base their cost-optimizing design programs on heuristic algorithms (starting close to optimum), specific to the equipment under consideration.

Mott, et al. [2], discuss a computerized procedure for designing a minimum cost heat exchanger. The method minimizes a cost index expressed as a function of fluid pumping power. The algorithm imposed no constraints.

To this point, no mention has been made of applying the concepts and techniques of nonlinear programming to optimizing the design of heat exchangers. However, Palen, et al. [14], in 1974 proposed using the Complex Method [15], for the heat

exchanger optimization problem. They found a minimum cost shell-and-tube exchanger by varying six geometrical parameters. The Complex Method requires several feasible starting designs before optimization can be performed.

Johnson, et al. [16], coupled an existing shell-and-tube condenser design code with a constrained function minimization code to produce an automated marine condenser design program of vastly different complexity.

The most complete work to date has been accomplished by Afimiwala [17]. He has applied various nonlinear programming methods of optimization to the heat exchanger design problem; including an experimental interactive graphical approach and exterior penalty function techniques. The gradient based search methods of Davidson - Fletcher - Powell and conjugate gradient were used for the resulting unconstrained minimizations. The exterior penalty method is extremely useful, since an initial solution satisfying the constraints is not required. The gradient based search methods are efficient when considering computer time.

Finally, Fontein and Wassink [18] utilized the complex method of Nelder and Mead [19], and a steepest descent method [20], for optimizing a shell-and-tube exchanger.

It can be seen that although there are many methods that have been presented for heat exchanger optimization, each of the methods has its own limitations; none is completely general. Of all the design procedures cited above (those of which are applicable to cross flow air-cooled heat exchangers), all are

limited to the 120 individual surfaces found in the open literature [21] for the calculation of the air-side heat transfer coefficient and friction factor. Therefore, the designer is faced with choosing an optimum surface from a number of individual optimal designs calculated from one of the above methods. In addition, the above methods treat the overall heat transfer coefficient as a constant, or they become involved with time-consuming numerical stepwise integrations in an attempt to account for the varying heat transfer coefficients.

This paper tries to bridge this gap by presenting an optimization routine that: selects an optimal surface, takes into account the varying heat transfer coefficients and friction factors across the exchanger, performs each analysis in an iterative-free manner, and may start with an infeasible design.

C. METHODOLOGY

With the Control Program for Engineering Synthesis and Constrained Function Minimization (COPES/CONMIN) optimizing scheme, a nonlinear optimization program is available that is capable of optimizing a wide class of engineering problems [22,23]. Therefore, for the heat exchanger design problem, it was necessary to develop a subroutine, which given a starting design, would analyze an air-cooled heat exchanger, and which would be suitable for coupling with the optimizer, COPES/CONMIN.

Figure 1 illustrates the procedure by which the heat exchanger was analyzed. Initial input consisted of a complete listing of all design parameters, whether known or estimated. Those that were estimates, i.e. unknowns, were later passed to the optimizer as design variables.

The analysis scheme and the optimizer will be discussed in much greater detail in the following chapters.

D. OBJECTIVES

The objectives of this thesis are two-fold.

The first objective was to develop a computer code, hereafter referred to as ANALIZ, which would analyze an air-cooled heat exchanger given any initial design. The analysis scheme was to: be iterative-free, take into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of flow path, and finally, be written in such a manner that it could be coupled with an existing optimizer COPES/CONMIN.

The second objective was to actually couple ANALIZ with a numerical optimization program. This would produce a detailed design program which would have the capability to determine an optimum surface, while optimizing the objective function such as size, weight, cost, etc.

II. NUMERICAL OPTIMIZATION

A. BACKGROUND

Almost all design problems require either the maximization or the minimization of some parameter or function. This parameter shall be called the design's objective function [24]. For example, the problem may call for a heat exchanger with a minimum volume. The expression for volume would be the design objective function. For the design to be acceptable, it must satisfy certain design constraints. For example, an air heater must be designed so that it will fit into a given space. Therefore, the engineer must set design constraints on the maximum size of the exchanger.

If the objective function could be easily formulated analytically, the maxima or minima could be found by using the methods of differential calculus. However, the limitations of this method are obvious.

Another numerical method that would be satisfactory for small scale problems would be an iterative solution technique. A computer program could be written containing a series of nested iteration loops that would vary the design parameters and solve the problem for a variety of values for each of the parameters. For other than small, easily formulated problems, the cost in central processor (CPU), time would be prohibitive.

Over the last twenty years, many numerical optimization techniques have been developed specifically for computer

utilization. These techniques usually do not require a specific algebraic equation, but rather any computer algorithm to which design variables can be input and from which the objective function and design constraint values can be determined is acceptable. For this reason, nonlinear programming methods were chosen for the air-cooled heat exchanger design. Some of these techniques were summarized by Shah et al. [25] in figure 2.

1. One-Dimensional Search Methods. Two of the most common of these search methods are the golden section [26] and quadratic interpolation [27]. The former isolates the minimum in regions of successively decreasing size, the latter performs a series of iterations approximating the objective function as a quadratic.
2. Multidimensional Unconstrained Search Methods. These unconstrained searches can be performed by a sequence of one-dimensional minimizations in the proper directions.
3. Multidimensional Constrained Search Methods. A common method here for enforcing the constraints in an optimization scheme are based on the sequential penalty function method. These techniques convert the constrained optimization problem into a sequence of unconstrained problems. This is accomplished by applying either an exterior or interior penalty to the objective function. The Complex Method [15], locates the optimum based on an intuitive approach in a n-dimensional space defined by the independent design variables. The method of feasible directions is used primarily for inequality constraints.

An optimization program based upon the Augmented Lagrangian Multiplier Method and the method of feasible directions was chosen for this research project.

B. CONSTRAINED FUNCTION MINIMIZATION (CONMIN)

Vanderplaats [22], developed an optimization program, CONMIN, based on the method of feasible directions which is

capable of optimizing a wide variety of engineering problems. CONMIN is a FORTRAN program, in subprogram form, that optimizes a function subject to a set of inequality constraints.

The following definitions will be useful in the following discussion:

1. Design Variables - those parameters which the optimization program can change in order to improve the design.
2. Design Constraints - those parameters which must not exceed given bounds for the design to be acceptable.
3. Objective Function - the parameter which is going to be minimized or maximized.

The general nonlinear inequality constrained optimization problem can be written mathematically as follows [28]:

$$\text{Minimize } F(\bar{X}) \quad (1)$$

Subject to:

$$g_j(\bar{X}) \leq 0 \quad j = 1, \text{NCON} \quad (2)$$

$$x_i^l \leq x_i \leq x_i^u \quad k = 1, \text{NDV} \quad (3)$$

where

$$\bar{X} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_{\text{NDV}} \end{bmatrix} \quad (4)$$

The vector $\bar{X}$ is the vector of design variables, with NDV equal to the number of design variables. The objective function,

$F(\bar{X})$, given by eq. (1), as well as the constraint functions given by eq. (2), may be linear or nonlinear functions of the design variables. They shall also be explicit or implicit functions of $\bar{X}$, but must have continuous first derivatives. NCON is the number of constraints. x_i^l and x_i^u are the lower and upper bounds or side constraints placed on the design variables. Side constraints could be included in eq. (2), but are treated separately for efficiency. Equality constraints are not dealt with by CONMIN, but will be treated separately by a multiplier method.

CONMIN requires that an initial vector of design variables, $\bar{X}$, which may or may not yield a feasible design, be specified. The design process continues iteratively as:

$$\bar{X}^{q+1} = \bar{X}^q + \alpha^* \bar{S}^q \quad (5)$$

where $\bar{S}^q$ is a vector search direction, α^* is a scalar quantity which determines the amount of change in $\bar{X}$ and q is the iteration number. At iteration q a direction $\bar{S}^q$ must be found which will reduce the objective (usable direction for minimization), without violating any constraints (feasible direction), see Figure 3 [30]. Once $\bar{S}^q$ is determined, eq. (5) becomes a one-dimensional search problem in which α^* must be found such that $F(\bar{X})$ is at a minimum (see Figure 4 [29]), a new constraint is encountered, or a currently active constraint ($g_j(\bar{X}) = 0$) is encountered again.

The design problem at iteration $q+1$ becomes one of finding a usable-feasible direction, $\bar{S}^q$, and a move parameter

a*. This process is illustrated geometrically by Johnson [30]. Consider a condenser problem with just two design variables, X_1 and X_2 , where

X_1 = condenser tube outside diameter,

X_2 = tube pitch to diameter ratio.

Let the objective function be condenser volume, $VOL(\bar{X})$.

Assume that the tube bundle diameter must be greater than a given value, BD_{min} , and that the cooling water pumping power must be less than a given value HP_{max} . Figure 5 illustrates the design problem geometrically.

It should be reiterated here, that while Johnson's example starts with a feasible initial design, (A), this is not a requirement and CONMIN is capable of optimizing given an infeasible initial design. This is obviously a great benefit.

The optimization process begins by calculating the gradient of the objective function by using finite difference. Each design variable is perturbed by .01 in a single forward step.

The gradient of the objective function, $\bar{\nabla}F$, shown in Figure 6 is simply the vector of the first partial derivatives with respect to the design variables; that is:

$$\bar{\nabla}F(\bar{X}) \equiv \bar{\nabla} VOL = \begin{bmatrix} \frac{\partial F}{\partial X_1} \\ \frac{\partial F}{\partial X_2} \end{bmatrix} \doteq \begin{bmatrix} \frac{\Delta VOL}{\Delta X_1} \\ \frac{\Delta VOL}{\Delta X_2} \end{bmatrix} \quad (6)$$

Therefore, because no constraints are active or violated at (A) . $\bar{\nabla}F$ defines the direction of steepest ascent. Because it is desired to minimize F , the greatest improvement can be made by moving in the negative gradient direction so that

$$\bar{S} = -\bar{\nabla}F = -\bar{\nabla}VOL . \quad (7)$$

With the value of $\bar{S}$ now determined, a search is performed from (A) until the minimum F is found at (B) on Figure 6. This is accomplished by taking several values of $\bar{X}^{q+1}$ in eq. (5) and interpolating for the α^* which will give the minimum value of F .

The second design iteration is begun at (B) by again perturbing $\bar{X}$ to find $\bar{\nabla}F$. Instead of moving in the direction of steepest descent, a conjugate direction, developed by Fletcher and Reeves [31], is chosen with this method, $\bar{S}$ is calculated as follows:

$$\bar{S}^q = -\bar{\nabla}F(\bar{X})^q + \frac{|\bar{\nabla}F(\bar{X})^q|^2}{|\bar{\nabla}F(\bar{X})^{q-1}|^2} \bar{S}^{q-1} , \quad (8)$$

see Figure 7.

The Fletcher-Reeves method is used in order to speed convergence. With the new conjugate direction, a search is performed in this direction until a constraint is encountered. This occurs at (C) of Figure 8 on the pumping power constraint.

At (C) , with the HP constraint active, not only is ∇F found, but the gradient of the active constraint is also computed, again using finite difference. The requirements on

the new search direction are now twofold; it must reduce the objective function and, at the same time, not violate the active constraint. This is solved by using the method of feasible directions developed by Zoutendijk [32], and implemented by Vanderplaats and Moses [33].

The problem of finding the new $\bar{S}$ can be stated as [34]:

Maximize β subject to the constraints

$$\bar{\nabla}F(\bar{X}) \cdot \bar{S} + \beta \leq 0 \quad (9)$$

$$\bar{\nabla}g_j(\bar{X}) \cdot \bar{S} + \theta_j \beta \leq 0 \quad j = 1, \text{NAC} \quad (10)$$

$$\bar{S} \cdot \bar{S} \leq 1 \quad (11)$$

where $\bar{\nabla}g_j(\bar{X}) = -\bar{\nabla}h_j$ and NAC is the number of active constraints (in this case NAC = 1).

If equation (9) is satisfied and β is positive, the search direction will reduce the objective function and is defined as a usable. If equation (10) is satisfied and β is positive, $\bar{S}$ is a feasible direction, because in this direction, no constraints will be violated if only a small move is taken. See Figure 3. θ_j is defined as the push-off factor for the active constraint and causes the design to move away from the constraint. θ_j must be greater than or equal to zero in order to maintain a feasible design. If the maximum value of β from equations (9) through (11) is zero, then there is no direction that will both reduce the objective function and also be feasible. Therefore, the current design is, at least, a local

minimum. In Johnson's example, a usable-feasible direction exists and a one-dimensional search leads to (D) in Figure 8 where the minimum bundle diameter, BD_{min} , constraint is met.

From (D) it should be noted here that CONMIN had information regarding the linearity of the BD_{min} constraint and, therefore, in (10), has set $\theta_j = 0$ to allow $\bar{S}$ to follow the constraint as shown in Figure 9. The one-dimensional search along this constraint is carried out until no further design improvement is realized. This occurs at (E).

This discussion of CONMIN would not be complete without citing the program's limitations. NDV directly affects the computational time required to reach the optimum. Since the calculation of gradients required for each design variable at the beginning of each design iteration is found by a finite difference step, which requires a complete pass through the analysis portion of the program, there is a subsequent increase in CPU time as NDV increases. Also, due to the interaction between design variables, as NDV gets larger, convergence slows during the optimization process. Vanderplaats [24], recommends that for most problems of general interest a practical limit of $NDV = 20$ be imposed. NCON does not present the same problem because gradient information is calculated simultaneously with VF and then only if the constraint is active or violated.

CONMIN offers no guarantees that a global minimum has been reached. Therefore, to lend some assurance, the design is

started with several different initial vectors until the same optimal design is reached.

Although CONMIN performs very well with inequality constraints, equality constraints such as:

$$h_K(\bar{X}) = 0$$

cannot be dealt with directly, but must be treated separately, using a different method, which will be discussed in the following sections.

C. CONTROL PROGRAM FOR ENGINEERING SYNTHESIS (COPES)

Recall that CONMIN was written in subroutine form, Vanderplaats [23], has developed a main program which greatly enhances the use of CONMIN.

For this main program, COPES, the user must supply an analysis subroutine titled ANALIZ. The subroutines CONMIN and ANALIZ are then used by COPES to optimize the objective function subject only to the inequality constraints.

ANALIZ must be organized into three segments: input, analysis and output. Based on the value of a counter, ICALC, ANALIZ performs the proper function in sequence.

The COPES program currently provides four specific capabilities:

1. Single Analysis - one cycle through the program, as if ANALIZ was executing alone.
2. Optimization - minimization or maximization of the objective function with constraints and side-constraints imposed.

3. Sensitivity Analysis - used to explore the effect of changing one or more design variables on one or more functions.
4. Two Variable Function Space - provides tables of data of all specified combination of two design variables.

However, a recent addition to COPES (still in the developmental stages), has put the Augmented Lagrangian Multiplier Method (ALMM) at the disposal of the programmer. Because of its good rate of convergence and its theoretical properties, the ALMM is preferred for equality constrained problems [35].

Therefore, COPES will call the various subroutines in order to optimize the objective function; subroutine ANALIZ for necessary analysis information, CONMIN for an optimum based on inequality constraints, and a subroutine (yet to be named), utilizing ALMM for an optimum satisfying the equality constraints.

For detailed explanation of the ALMM, its background and mathematical derivation, consult reference [35]; but for now consider the equality constrained problem:

$$\text{Min } f(\bar{X}) \quad (10a)$$

$$\text{Subject to } h_k(\bar{X}) = 0 \quad k = 1, \text{NECON} \quad (10b)$$

where NECON is the number of equality constraints.

Define the "modified" Lagrangian function as:

$$L(\bar{X}, \bar{\lambda}) = f(\bar{X}) + \sum_{i=1}^m \lambda_i h_k(\bar{X})$$

where λ_i is the Lagrangian multiplier. The problem can now be stated as:

$$\text{Min } L(\bar{X}, \bar{\lambda}) \quad (10c)$$

$$\text{Subject to } h_k(\bar{X}) = 0 \quad (10d)$$

Then, according to Lagrange, if a $\bar{\lambda}$ can be found for which $\bar{X}$ solves the problem stated above, then $\bar{X}$ is also the solution to the original problem, eq. (10a) and (10b).

The new problem is solved by the conventional exterior penalty function method, because this is believed to be one of the most efficient algorithms for the solution of such equality constrained problems [35].

II. HEAT EXCHANGER ANALYSIS

A. INTRODUCTION

To meet the objectives of this thesis, an analysis program for an air-cooled heat exchanger must be coupled with a numerical optimization scheme to produce a complete, detailed design package. COPES/CONMIN has greatly simplified this task.

This analysis program must be written in subroutine form, titled ANALIZ, and organized into three segments: input, execution, and output. The analysis subroutine must also:

1. Take into account the variation of the heat-transfer coefficients and differential pressure drop with temperature and/or length of flow path.
2. Be iterative free, if possible.
3. Be written in such a manner that the optimizer will play a role in surface selection.

With the number of design variables approaching the practical limit, the importance of an iterative free analysis subroutine cannot be over-emphasized. The reason being, that at the beginning of each design iteration in CONMIN, the calculation of all gradients (each design variable and active constraint) requires a complete pass through ANALIZ. Therefore, the computational time required by ANALIZ directly affects the time required to reach the optimum.

B. PROBLEM FORMULATION

The air-cooled heat exchanger is shown in Figure 10. A cross-flow arrangement with both fluids unmixed was chosen.

Cool air enters the heat exchanger at temperature T_{c_1} , pressure p_a , and constant specific heat c_{p_a} . The cool air makes one pass through the exchanger as it flows over an isosceles pitched bank of finned tubes. The air is heated by water, in single phase, at an entering temperature of T_{h_1} , and constant specific heat c_{p_w} .

The analysis of the air-cooled, cross-flow heat exchanger centers about the first law of thermodynamics and on the heat transfer equation. These equations as they apply to the exchanger of Figure 10 are as follows:

$$\dot{Q}_3 = \dot{m}_a c_{p_a} (T_{c_2} - T_{c_1}) \quad (12)$$

$$\dot{Q}_4 = \dot{m}_w c_{p_w} (T_{h_1} - T_{h_2}) \quad (13)$$

$$\dot{Q}_5 = U_m A \Delta T_m \quad (14)$$

where $\dot{m}_a$, T_{c_2} and $\dot{m}_w$, T_{h_2} are the fluid mass flow rates and exit temperatures of air and water respectively. U_m is the true mean overall heat transfer coefficient based on the outside root tube area, A is the total heat transfer surface area of the exchanger used to compute U_m , and ΔT_m is the mean temperature difference of the given exchanger.

The object of the analysis, therefore, is to determine $\dot{Q}_3$, $\dot{Q}_4$ and $\dot{Q}_5$ given an initial listing of values for the design parameters. The list includes the following:

Tubeside mass flow rate, lbm/hr

Entrance temperature of hot stream, °F

Exit temperature of hot stream, °F

Specific heat of hot fluid, BTU/lbm - °F

Air mass flow rate, lbm/hr

Entrance air temperature, °F

Exit air temperature, °F

Entrance air pressure, psi

Specific heat of air, BTU/lbm-°F

Tube inside diameter (ID), in.

Tube outside diameter (OD), in.

Fin height, in.

Fin thickness, in.

Fin spacing, in.

Transverse Pitch, in.

Longitudinal Pitch, in.

Bank height, in.

Bank width, in.

Cross-flow arrangement

Fin type

Number of rows

Number of passes

Given heat transfer rate, BTU/hr

Thermal Conductivity of tube material, BTU/hr.-ft.-°F

Thermal Conductivity of fin material, BTU/hr.-ft.-°F

Among the initial listing of design parameters above,
there are parameters that are known and will remain constant

throughout the design problem. Also, there will be those parameters that are unknown and can vary, i.e. design variables. On the way to determining the various heat transfer rates, other information will have been computed. This information includes the objective function, constraining functions, and other design data, such as the number of tubes per vertical row.

The optimizer will then manipulate the design variables in order to find an optimum, while at the same time, performing a heat balance, that is:

$$\dot{Q} = \dot{Q}_3 = \dot{Q}_4 = \dot{Q}_5 \quad (15)$$

where $\dot{Q}$ may be some given heat transfer rate.

C. PERFORMANCE CALCULATION PROCEDURE

With the temperatures, mass flow rates, and specific heats all specified in the listing of design parameters; whether they be constant or variable, the only unknown quantities on the right hand side of equations(12) through (14), are U_m , A , and ΔT_m . They will be determined as shown in Figure 1.

1. Mean Temperature Difference (MTD)

For many flow arrangements, various approaches for determining MTD, mainly using diagrams, are available [36], which have proven very useful in manual design efforts. For computerized design, however, an explicit, approximate equation is desirable in order to achieve a fast, sufficiently accurate

calculation of the mean temperature difference of a given flow arrangement.

Roetzel, et al. [37], presented such an approximate equation together with empirical coefficients for nine counter-current cross-flow arrangements as they apply to air-cooled heat exchangers.

Roetzel used the familiar equation for the MTD of the given flow arrangement ΔT_m :

$$\Delta T_m = F \cdot \Delta T_{\ell_m} \quad (16)$$

where ΔT_{ℓ_m} is the limiting case of pure countercurrent flow:

$$\Delta T_{\ell_m} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln \frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} \quad (17)$$

and F is a correction factor determined by a different set of coefficients for each flow arrangement. Roetzel reported the following function suitable for F :

$$F = 1 - \sum_{i=1}^m \sum_{k=1}^n a_{i,k} (1 - v_{1_m})^k \sin(2i \arctan R) \quad (18)$$

where v_{1_m} is the dimensionless LMTD.

$$v_{1_m} = \frac{\Delta T_{\ell_m}}{T_{h1} - T_{c1}} \quad (19)$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} \quad (20)$$

and the coefficients $a_{i,k}$ of the approximating equation (18), were calculated using a standard least squares estimation program [38], and are reported in reference [37]. The assumption that both streams were unmixed was used in their calculation.

When more than four tubeside passes are used, it is assumed that the heat exchanger has approached the limiting case of pure counterflow and F is set equal to one [2].

Having determined the MTD, the remainder of the analysis procedure follows Roetzel's [39] general approximation method for determining the mean overall heat transfer coefficient, U_m , for any flow arrangement while taking into account the variation of the heat transfer coefficients and the pressure drop with temperature and/or length of path.

Before continuing with specific analysis procedures, a brief summary of Roetzel's general approximation method is in order.

The local overall heat transfer coefficient based on the outside root tube area can be written as follows:

$$U = \frac{1}{\frac{A_o}{A_i} \frac{1}{H_i} + \frac{A_o \ln(r_o/r_i)}{2\pi kL} + \frac{1}{H_o \eta_f}} \quad (21)$$

where η_f is the efficiency of the extended surface.

In order to determine the individual convection heat transfer (film) coefficients, H_i and H_o , according to the

conventional methods, the coefficients would be considered constant, and the necessary fluid properties for their calculation would be evaluated at some mean bulk temperatures, T_{h_b} and T_{c_b} .

However, the film coefficients are not constant, but vary with temperature and/or length of flow path. Roetzel has taken these variations into account with the use of corrected reference temperatures. Two sets of corrected reference temperatures are determined: T_{h_I} , T_{c_I} and $T_{h_{II}}$, $T_{c_{II}}$. Therefore, for each set of corrected reference temperatures, the film coefficients are determined in the conventional manner using the reference temperatures in place of the bulk temperatures.

With the film coefficients, H_{i_I} , H_{o_I} , $H_{i_{II}}$ and $H_{o_{II}}$, two local overall heat transfer coefficients can be calculated from equation (21), U_I and U_{II} .

Finally, the true mean overall heat transfer coefficient is calculated as:

$$\frac{1}{U_m} = \frac{1}{2} \left[\frac{1}{U_I} + \frac{1}{U_{II}} \right] \quad (22)$$

2. Determination of Reference Temperatures

The reference temperatures for a pure counterflow heat exchanger must first be determined from [39]:

$$T'_{h_I} = T_{h_2} + (T_{h_1} - T_{h_2}) \left[\frac{\Delta T_I - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (23)$$

$$T'_{h_{II}} = T_{h_2} + (T_{h_1} - T_{h_2}) \left[\frac{\Delta T_{II} - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (24)$$

$$T'_{c_I} = T_{c_1} + (T_{c_2} - T_{c_1}) \left[\frac{\Delta T_I - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (25)$$

$$T'_{c_{II}} = T_{c_1} + (T_{c_2} - T_{c_1}) \left[\frac{\Delta T_{II} - (T_{h_2} - T_{c_1})}{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})} \right] \quad (26)$$

where

$$\Delta T_I = (T_{h_1} - T_{c_2}) \cdot .78868 \cdot (T_{h_2} - T_{c_1}) \cdot .21132 \quad (26a)$$

$$\Delta T_{II} = (T_{h_1} - T_{c_2}) \cdot .21132 \cdot (T_{h_2} - T_{c_1}) \cdot .78868 \quad (26b)$$

Equations (26a) and (26b) were derived by Roetzel in references [40] and [41]. For the special case where the fluid heat capacity rates, $\dot{C}$, are equal, that is,

$$\dot{m}_h C_{p_h} = \dot{m}_c C_{p_c}$$

the term

$$\frac{\Delta T_j - (T_{h_2} - T_{c_1})}{(T_{h_2} - T_{c_2}) - (T_{h_2} - T_{c_1})}$$

of equations (23) through (26), with $j = I$ and II , becomes .78868 or .21132 respectively.

3. Correction of Reference Temperatures for Given Arrangement

With inlet and outlet temperatures fixed, pure counter-flow yields the highest mean temperature difference. Therefore, for any other arrangement, the temperature difference would be smaller. Thus, the corrections are applied in the following manner [39]:

$$T_{h_I} = T'_{h_I} - \psi_{h_I}$$

$$T_{h_{II}} = T'_{h_{II}} - \psi_{h_{II}}$$

$$T_{c_I} = T'_{c_I} + \psi_{c_I}$$

$$T_{c_{II}} = T'_{c_{II}} + \psi_{c_{II}}$$

where ψ_{h_k} and ψ_{c_k} are the temperature corrections of the hot and cold streams, respectively. The corrections are calculated as follows:

$$\psi_{h_j} = \Delta T_j \left[\frac{1 - \Delta T_m / \Delta T_{i_m}}{1 + (\dot{C}_h / \dot{C}_c)^{2/3}} \right]$$

$$\psi_{c_j} = \Delta T_j \left[\frac{1 - \Delta T_m / \Delta T_{\ell_m}}{1 + (\dot{C}_c / \dot{C}_h)^{2/3}} \right]$$

The corrected reference temperatures are now used to determine the thermal conductivities and absolute viscosities of

fluids for later use in the calculation of the film coefficients. Thermal conductivity and viscosity data are usually presented in tabular form. However, for use on the computer, an explicit, simple approximate equation with temperature as the independent variable, was desirable. Water and air were chosen as two fluids that were likely to be involved in an air-cooled heat exchanger design. Figures 11, 12 and 13 indicate that the thermal conductivities of air and water and the viscosity of air can be approximated by a second order polynomial. The viscosity of the hot tubeside fluid, water, must be treated specially, due to the following considerations.

Calculations for the tubeside heat transfer coefficient in the laminar, transition and turbulent regions all involve the correction, $(\mu/\mu_{wall})^{.14}$ [42]. In the past, an iterative procedure was required to determine the inside tube wall temperature in order to evaluate μ_{wall} . Roetzel [43], has developed an iteration-free method for determining this correction. Roetzel's method assumes that the tubeside fluid's viscosity follows Andrade's Correlation, that is:

$$\mu = \alpha e^{\beta/T}$$

Using the viscosity data of saturated water from reference [44], the coefficients were determined through regression analysis with the resulting equation:

$$\mu_w = .01339 \exp\left(\frac{2715.7764}{T}\right) \quad (27)$$

where temperature is in degrees Rankine and viscosity is in lbm/ft-hr. Equation (27) was plotted next to the experimental data with results that indicate that water does indeed satisfy Andrade's equation, see Figure 14.

4. Uncorrected Tubeside Heat Transfer Coefficient

The tubeside heat transfer coefficient is calculated from one of three Nusselt-type empirical equations [42], as follows:

For laminar flows, Reynolds Number < 2,100

$$\frac{HD_i}{k} = 1.86 \text{ Re}^{1/3} \text{ Pr}^{1/3} \left(\frac{D_i}{L}\right)^{1/3} \left(\frac{\mu}{\mu_{\text{wall}}}\right)^{.14} \quad (28)$$

For transition regions, 2,100 < Re < 10,000

$$\frac{HD_i}{k} = [\text{Re}^{2/3} - 125] \left[1 + \left(\frac{D_i}{L}\right)^{2/3}\right] \text{ Pr}^{1/3} \left(\frac{\mu}{\mu_{\text{wall}}}\right)^{.14} \quad (29)$$

For turbulent flow, Re > 10,000

$$\frac{HD_i}{k} = .023 \text{ Re}^{.8} \text{ Pr}^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14} \quad (30)$$

Therefore, before any calculations can even begin, the Reynolds Number, Re, must be computed to determine the type of flow.

The Reynolds Number will be calculated as:

$$\text{Re}_j = \frac{D_i \dot{m}_h}{A_x \mu_{h_j}}$$

D_i , m_h is supplied in the initial list of parameters and μ_h is calculated from equation (27). This leaves only the cross-sectional flow area, A_x , to be determined, where:

$$A_x = N_p \frac{\pi D_i^2}{4}$$

The number of tubes per pass, N_p , is determined geometrically, having been given the tube bank height, number of rows, number of passes, transverse pitch, diameters, and fin height initially. Figure 15 details the procedure of finding N_p . Mathematically,

$$N_p = \left[\frac{n_{\text{rows}}}{n_{\text{pass}}} \right] \left[h - D_f - \frac{P_t}{2} \right] / P_t$$

and would be a rational number. This rational number is then truncated to an integer number of tubes.

With Re calculated, the Prandtl Number,

$$Pr_j = \frac{C_p \mu_{h_j}}{k_{h_j}}$$

is computed from given and previously determined thermo-physical properties.

The uncorrected tubeside heat transfer coefficients, H'_{i_j} , that is; H_{i_j} , without the factor $(\mu/\mu_w)^{.14}$, can be calculated from the proper choice of equations (28) through (29).

5. Calculation of Wall and Associated Resistances

Equation (21) can be written in a more general form as:

$$U = \frac{1}{R_i + R_{wall} + R_o} \quad (31)$$

where R_i , R_w and R_o are the inside wall and outside heat transfer resistances respectively. Additional resistances, such as contact or fouling, can also be added here.

In comparing equations (21) and (31), the resistances can be computed as:

$$R_{ij} = \frac{D_o}{D_i} \frac{1}{H_{ij}} \quad (31a)$$

$$R_{wall} = \frac{r_o \ln(D_o/D_i)}{k_{wall}} \quad (31b)$$

$$R_{oj} = \frac{1}{H_{oj} \eta_f} \quad (31c)$$

6. Calculation of Airside Heat Transfer Coefficients

In order to have the optimization program play a significant role in the selection of an optimized surface for a finned tube heat exchanger (which is one of the objectives of this project), an explicit equation for H_o involving tube and bank design parameters as independent variables is a necessity.

In the past, comparison methods, as described by Shah [45], were used to choose the best surface from a list of

surfaces for which experimental heat transfer and friction data existed. The data is presented in graphical form, where Friction and Colburn Factors are plotted as a function of Reynolds Number. With J , $\dot{m}_a$, c_p and Pr known, the film coefficient can be computed.

Therefore, in previous heat exchanger optimization programs, a given surface described by its pitch, outside fin diameter, fin thickness, fin spacing, and outside tube diameter, had to be chosen beforehand. After the surface configuration had been selected, expressions for f and J were obtained by fitting polynomials to the experimental data described earlier.

Briggs and Young [47] provide a means of getting past this obstacle with an improved convection heat transfer correlation for air flowing across triangular pitch banks of high finned tubes.

Briggs and Young expanded the work of Ward and Young [48], to cover a total of 18 differently configured finned tube banks in order to determine the effect of fin thickness and tube pitch on the airside heat transfer coefficient. The heat transfer data for the high-fin tube banks were correlated to give:

$$Nu = \frac{h_{o,j} D_o}{k_j} = .1378 Re^{.718} Pr^{1/3} \left(\frac{s}{l}\right)^{.296} \quad (32)$$

where s is the distance between adjacent fins and l is the fin height. Equation (32) is based on tubes having a wide range

of fin heights, fin thicknesses, fin spacing and outside tube diameter and can be used to predict H_{o_j} for a bank of tubes six rows deep. Figure 11 of reference [48] is used to correct H'_{o_j} for banks of other than six rows.

7. Calculation of Fin and Surface Efficiencies

In order to calculate the outside heat transfer resistance, which will be used to calculate the correction for the tubeside heat transfer coefficient and finally, the local overall heat transfer coefficients, the extended surface efficiency, η_f , must be computed.

The surface efficiency accounts for the temperature drop from the root to the tip of the fin, due to the thermal resistance of the fin material. Thus, even though the heat transfer has been increased by the additional area of the extended surface, the area is not as effective as if it were to be all at the root temperature.

The surface efficiency can be expressed as [49]:

$$\eta_f = 1 - \frac{A_f}{A} (1 - \phi)$$

where ϕ , the fin efficiency for a radial fin, can be found from [49]:

$$\phi = \frac{2 r_o}{m(r_f^2 - r_o^2)} \left[\frac{I_1(m r_f) k_1(m r_o) - k_1(m r_f) I_1(m r_o)}{I_0(m r_o) k_1(m r_f) + I_1(m r_f) k_0(m r_o)} \right]$$

The finned area, A_f , and the total heat transfer area, A , are computed as follows:

$$A_s = N_T w \eta_f \frac{\pi}{2} (D_f^2 - D_o^2)$$

$$A = N_T w \left[\frac{\eta_f \pi}{2} (D_f^2 - D_o^2) + (1 - \eta_f t) D_o \pi \right]$$

With R_o calculated from equation (31c), the correction to the tubeside film coefficient can be made.

8. Correction of the Tubeside Heat Transfer Coefficient

With the tubeside heat transfer coefficient dependent on the wall temperature, the dependence has either been neglected, or the wall temperature has been calculated with an iterative technique in the past. Roetzel [43], has proposed an improved iterative-free method for finding the "Seider-Tate" correction, $(\mu/\mu_{wall})^{.14}$, when the tubeside fluid obeys Andrade's viscosity correlation.

From Roetzel's formulation:

$$\left(\frac{\mu}{\mu_{wall j}} \right)^{.14} = - \frac{u_j}{2} + \left(\frac{u_j^2}{4} + v_j \right)^{1/2}$$

where

$$u_j = .007 \ln \left(\frac{\mu_{wall j}^*}{\mu_j} \right) \frac{1}{B_j} \left[\frac{1 - T_{c j} / T_{h j}}{T_{h j} / T_{wall}^* - 1} \right] + \frac{T_{c j}}{B_j T_{h j}} - 1$$

$$v_j = -.07 \ln \left(\frac{\mu_{wall j}^*}{\mu_j} \right) \frac{1}{B_j} \left[\frac{1 - T_{c j} / T_{h j}}{T_{h j} / T_{wall}^* - 1} \right] + \frac{T_{c j}}{B_j T_{h j}}$$

where

$$B_j = H'_{i_j} A (R_{wall} + R_{o_j})$$

(Note that all temperatures are in degrees absolute.)

All the parameters on the right-hand side of equation (21) are now available. Therefore, the two reference overall heat transfer coefficients, U_I and U_{II} , can be calculated. The mean overall heat transfer coefficient follows easily from equation (22).

With U_m , $\dot{Q}_5$ can be calculated, with the heat balance to be performed by the optimizer.

9. Calculation of Air and Tubeside Pressure Drops

The final calculations before computing objective and constraint functions for the optimization problem involve the pressure drops in the heat exchanger.

The basic equations that will be used for the calculation of the pressure losses are as follows:

a. Tubeside [42]

$$\Delta p_i = \frac{f_i \dot{m}_h^2 w n_p}{2 g_c A_{x_T}^2 \rho D_i \left(\frac{\mu}{\mu_w}\right)} z + \frac{(n_p - 1) \dot{m}_h^2}{2 g_c A_{x_T}^2 \rho} \quad (33)$$

where $z = .14$ below $Re = 2100$ and $z = .25$ for Re greater than 2100.

b. Airside [50]

$$\Delta p_o = \frac{f_o n_r \dot{m}_c^2}{A_{ff}^2 g_c \rho} \quad (34)$$

where A_{ff} is the minimum flow area.

The friction factor for the tubeside flow, f_i , is taken from Figure 9.5 of reference [42], assuming fully developed flow. For use on the computer, an explicit expression for f_i was obtained by fitting a line and an exponential to the experimental data of Figure 9.5. This follows:

$$\underline{\text{Re} \leq 1000} \quad f_i = .5/\text{Re}$$

$$\underline{\text{Re} > 1000} \quad f_i = .003215 \text{ Re}^{-.2694}$$

Just as in the case of the airside film coefficient, for surface optimization on the computer, an explicit equation for the airside friction factor, f_o , is a necessity. Robinson and Briggs [50], presented such an expression for f_o for air flowing across triangular pitch banks of finned tubes. Robinson and Briggs' work closely parallels that of Briggs and Young [47]. The Robinson-Briggs Correlation:

$$f_o = 18.93 \text{ Re}^{-.316} \left(\frac{P_t}{D_o}\right)^{-.927} \left(\frac{P_t}{P_L}\right)^{.515} \quad (34a)$$

covers the range of tube sizes and pitches used in air-cooled heat exchangers [50].

Therefore, with Re_I and Re_{II} , the four reference pressure drops, Δp_{ij} , and Δp_{oj} , may be computed from equations (33) and (34). Following Roetzel's general approximation method:

$$\Delta p_i = \left[\frac{\Delta p_{iI}}{U_I} + \frac{\Delta p_{iII}}{U_{II}} \right] / \left[\frac{1}{U_I} + \frac{1}{U_{II}} \right]$$

For a gas, an additional correction is needed because the density in equation (33) is strongly dependent on pressure, which is changing through the exchanger. Using the inlet pressure as reference:

$$\Delta p'_o = \left[\frac{\Delta p_{oI}}{U_I} + \frac{\Delta p_{oII}}{U_{II}} \right] / \left[\frac{1}{U_I} + \frac{1}{U_{II}} \right]$$

$$\Delta p_o = p_1 \left[1 - \left(1 - \frac{2\Delta p'_o}{p_1} \right)^{1/2} \right] \quad (35)$$

All the necessary information from an analysis viewpoint has now been calculated. Functions needed for the numerical optimization process shall follow.

10. Objective and Constraint Functions

The objective functions available for minimization are defined as follows:

a. Volume = $wh[D_f + (n_r - 1)P_L \cos \theta]$

where

$$\theta = \arcsin (P_t / 2P_L) \quad (36)$$

b. Heat Transfer Area = $N_T w \left[\frac{(D_f^2 - D_o^2) \pi}{2S} + \left(1 - \frac{t}{S} \right) D_o \pi \right]$

c. Air Horsepower = $\frac{\Delta p_a \cdot \dot{m}_a}{\rho}$

- d. Airside Pressure Drop
- e. Tubeside Pressure Drop

Limitations were imposed on the following inequality constraints in order to keep the design within practical physical bounds:

- a. The diameter ratio,

$$\text{DRATIO} = \frac{D_f}{D_o}$$

must be kept reasonable. This can also be accomplished to some extent by placing side constraints on the design variables, t and D_o . See Figure 16.

- b. The optimizer must be prevented from driving the tube thickness,

$$\text{TUBTH} = (D_o - D_i)/2$$

to zero.

- c. The tubes must be kept from touching in both the longitudinal and transverse directions,

$$\text{TOUCHN} = D_f - P_t$$

$$\text{TOUCHL} = D_f - P_L$$

TOUCHL and TOUCHN, therefore, must be kept below zero.

- d. Reasonable temperature profiles must be maintained at both ends of the exchanger. See Figure 10-7 of Ref. [44],

$$\text{PROFH} = T_{c2} - T_{h1}$$

$$\text{PROFC} = T_{c1} - T_{h2}$$

that is, PROFH and PROFC, must be negative.

- e. The number of tubes per vertical row, VROWR, shall not be allowed to go below 2.
- f. The free face area, that is, the minimum flow area for air, must obviously be greater than zero,

$$\text{DELSFF} = [\text{projected tube area}] - \text{hw}$$

where the projected tube area, STOTAL, is

$$\text{STOTAL} = N_r [D_o w + \frac{D_f t w}{s}]$$

and N_r is the number of tubes per vertical row. DELSFF must be less than zero.

- g. The airside and tubeside pressure drops must be kept within design constraints.
- h. From experience, the argument of the square root of equation (35),

$$\text{ARG5} = 1 - \frac{2\Delta p'}{P_1}$$

has been driven below zero. It must therefore be constrained.

- i. To maintain an isoscele pitch bank, the angle, θ , as shown in Figure 17 and as defined in eq. (36), must be constrained. One such constraining value is:

$$\theta_m = \arccos (D_f / 2P_L) \quad (37)$$

The arguments of the arcsin and arccos of eq. (36) and (37) must be constrained from going beyond 1.

- j. The heat balance described by eq. (15) is performed by three equality constraints:

$$\text{QRATIO} = \dot{Q}_5 / \dot{Q} \quad (38)$$

$$\text{QRATIO1} = \dot{Q}_3 / \dot{Q}_5 \quad (39)$$

$$QRATIO2 = \dot{Q}_4 / \dot{Q}_3 \quad (40)$$

All constraints are set equal to 1.

Besides the constraints described in paragraphs a. through j., above, lower and upper bounds are placed on the design variables to assure a reasonable design.

IV. RESULTS

A. BACKGROUND

Case studies were chosen as the best way to test the capabilities of the program for Heat Exchanger Design using Numerical Optimization (HEDSUP). The design problems posed were made as realistic as possible.

1. Capabilities

HEDSUP currently has the capability to design for nine different configurations of triangular pitch banks of finned tubes:

TYPE 1 - 1 ROW, 1 PASS

TYPE 2 - 2 ROW, 1 PASS

TYPE 3 - 3 ROW, 1 PASS

TYPE 4 - 4 ROW, 1 PASS

TYPE 5 - 2 ROW, 2 PASS

TYPE 6 - 3 ROW, 3 PASS

TYPE 7 - 4 ROW, 2 PASS

TYPE 8 - 4 ROW, 2 PASS

TYPE 10 - PURE COUNTERFLOW

TYPE 10 will include exchangers with a configuration of n rows, n passes, where n can go from five to 20.

The banks must be constructed of high-finned tubes ($\phi > .0625$ in. [42]) with the fins having a rectangular profile of constant thickness. Additional profiles can be inserted into HEDSUP quite simply, provided that its fin efficiency can

be expressed explicitly as:

$$\eta = f(\ell, t, H, k)$$

See Subroutine FINEFF of the program listing, Appendix D.

At present, HEDSUP can provide the design parameters for an air-cooled heat exchanger optimized for any one of the following design objectives:

- (a.) Minimum Volume
- (b.) Minimum Heat Transfer Surface Area
- (c.) Minimum Air Horsepower
- (d.) Minimum Airside Pressure Drop
- (e.) Minimum Tubeside Pressure Drop

Additional design objectives can be used, provided that they can be expressed explicitly as a function of the design variables and they are added to the common block. It should also be pointed out that any design variable may simultaneously be a design objective as long as it conforms to the restrictions of both. For example, an exchanger may be designed for minimum bank height.

The airside fluid is restricted to dry air. The tubeside fluid is presently limited to water in single phase. Other tubeside fluids can be used by HEDSUP, provided that their viscosities obey Andrade's Law and the fluid's thermal conductivities can be expressed explicitly as a function of temperature. The fluid's specific gravity would also have to be placed in the denominator of eq. (33).

B. CASE STUDIES

1. Case One

a. Problem Formulation

An air-cooled heat exchanger is to be designed for minimum volume with a heat transfer rate of 10,000,000 BTU's per hour. Water is to be cooled from 200°F to 125°F. Dry air will enter the exchanger at 95°F and leave at 130°F. Specifications call for a fan that can produce a pressure difference of two inches of water.

b. Design Variable Framework

From the list of design parameters in Section III.B, the design variables must be singled out, identified to COPES, and given side constraints. All parameters must be given an initial value. Only the values of the design variables will change.

Assuming constant specific heats,

$$c_{p_w} = 1.0 \quad \text{BTU/lbm-}^\circ\text{F}$$

$$c_{p_a} = .24 \quad \text{BTU/lbm-}^\circ\text{F}$$

the mass flow rates of both fluids can be determined from eqs. (12) and (13), because the heat transfer rate and temperature differences are given,

$$\dot{m}_w = \frac{\dot{Q}}{c_{p_h} \Delta T_n} = 133333. \quad \text{lbm/hr}$$

$$\dot{m}_a = \frac{\dot{Q}}{c_{p_c} \Delta T_c} = 190476.2 \quad \text{lbm/hr}$$

A cross-flow arrangement, fin profile, tube material, and fin material must be chosen.

The design variables for this example are, therefore:

$$\begin{array}{ll}
 .232 < D_i < 2.325 \text{ in.} & D_i^i = 2.0 \text{ inches} \\
 .24 < D_o < 2.5 \text{ in.} & D_o^i = 2.5 \text{ inches} \\
 .0625 \text{ in.} < \ell < \infty & \ell^i = .46 \text{ inches} \\
 .01 < t < .0235 \text{ in.} & t^i = .023 \text{ inches} \\
 .08 < S < .125 \text{ in.} & S^i = .111 \text{ inches} \\
 0.0 < P_L < 4.0 \text{ in.} & P_L^i = 2.125 \text{ inches} \\
 0.0 < P_t < 4.0 \text{ in.} & P_t^i = 4.0 \text{ inches} \\
 0.0 < w < 500 \text{ in.} & w^i = 490 \text{ inches} \\
 0.0 < h < 500 \text{ in.} & h^i = 350 \text{ inches}
 \end{array}$$

The side constraints on the design variables are of a practical nature with the exception of the lower bounds on fin height. Recall that the use of eq. (32) is restricted to high-fins. High-fins will also tend to keep the fluid unmixed, which was an assumption used when defining the coefficients $a_{i,k}$, used in eq. (18).

c. Constraint Framework

From the problem statement, the airside pressure drop must be less than two inches of water or .0722 psi,

$$0 < \Delta p_a < .0722 \text{ psi}$$

From a practical standpoint:

$$0.0 < \theta < 1.3$$

$$1.0 < \text{DRATIO} < 2.5$$

$$.018 < \text{TUBTH} < .18 \text{ inches}$$

$$-\infty < p_w < .14 \text{ psi}$$

The equality constraint

$$\frac{\dot{Q}_5}{\dot{Q}} = 1.0$$

where

$$\dot{Q}_5 = U A \Delta T_m$$

$$\dot{Q} = 10,000,000 \text{ Btu/hr}$$

will satisfy the heat balance.

The bounds on TOUCHN, TOUCHL ARG5, ARG7, ARG8, DELSFF and VROWR were discussed in Section III.C.10. Constraints on PROFH and PROFC were unnecessary because all temperatures were specified in the problem statement.

As with design variables, constraints must be identified to COPES by location in the common block. See the global catalog, Appendix A.

d. Methodology

Ideally, a three dimensional design matrix can now be constructed of optimum exchanger designs with minimum volumes. The matrix would be constructed by first holding the tube and fin materials constant and varying the configuration, i.e. Type 1, Type 2, Type 3, etc. Next the tube material would be varied with the fin material and exchanger configuration held constant and so forth. However, for case study one, the tube material will be chosen as copper, $k = 200 \text{ BTU/ft-hr-}^\circ\text{F}$, and the fin material will be aluminum, $k = 118 \text{ BTU/ft-hr-}^\circ\text{F}$.

Also, in order to simulate an actual trade off study, the constraint framework will be fixed throughout the individual case studies.

Problems arise in constructing the matrix when trying to determine the true minimum volume design for each configuration. Unfortunately, the choice of initial design parameters (starting point), coupled with the input parameters for ALMM, will sometimes lead to entirely different optimum designs with volumes differing by over 100%.

The ALMM parameters include the initial multiplier, CC, the multiplication factor, CMULT, and the maximum multiplier value, CCMAX. Experience has shown that setting

$$\begin{aligned}\text{CMULT} &= 2.0 \\ \text{CCMAX} &= 1000.\end{aligned}$$

will suffice for almost all problems. However, there is much "artwork" involved with the choice of CC. From experience,

an initial multiplier of 10 works well when starting far from the equality constraint, i.e.:

$$.8 < QRATIO < 1.2$$

However, when approaching the heat balance, i.e., the equality constraint, a $CC = 100$ is necessary for the heat balance to converge. Equation (15) is considered satisfied when the heat transfer ratios (eqs. (38) through (40)) are less than 0.1 percent.

Therefore, it is obvious why the chosen initial design is so critical. Together with the choice of CC it will determine how and to what design the optimizer will converge. As an example, see Table 1. Notice the calculated heat transfer for the initial design. This value is the product of U_m , A , and ΔT_m calculated by ANALIZ using the initial design parameters, some of which are mere estimates. Recall it is the job of the optimizer to vary the design variables in order to bring $\dot{Q}_5$ equal to $\dot{Q}_1$, $\dot{Q}_3$ and $\dot{Q}_4$ and at the same time minimize the objective function.

In order to remove some of the "artwork" and try to ensure a true optimum design, the following procedure is recommended to generate the design matrix:

- (1) Begin with a Type 2 configuration; input the initial design and constraint values enumerated in sections (a) through (d) above; let $CC = 10.$; execute.
- (2) If the heat balance of the resulting design has not converged, but is within 20%, use the design results as a new starting point and set $CC = 100$ (if the heat balance is not within 20%, let $CC = 10.$).

TYPE 7 EXCHANGER CONFIGURATION OPTIMUM DESIGN

DESIGN VARIABLES	RUN I		RUN II	
	INITIAL DESIGN	OPTIMUM DESIGN	INITIAL DESIGN	OPTIMUM DESIGN
D_i , inches	.6786	.5392	2.0	1.168
D_o , inches	.7201	.5767	2.5	1.278
l , inches	.1618	.128	.46	.294
t , inches	.0217	.0208	.023	.023
S , inches	.08	.08	.111	.08
P_t , inches	1.044	1.02	4.0	2.557
P_L , inches	1.044	.841	2.125	1.899
h , inches	342.1	238.4	350.	172.9
w , inches	286.6	186.2	490.	253.
Q , BTU/hr	21,988,624.	9,999,925.	20,650,352.	9,999,811.
Volume, ft ³	213.12	72.92	553.2	153.83

Table 1

- (3) Repeat step (2) until convergence. NOTE: CC may be adjusted up to 150 when approaching convergence. The design should converge following the use of CC = 100. If too much adjustment of CC is necessary, reaching the optimum from the starting point is unlikely.
- (4) Ensure a minimum design by beginning step (1) with a different initial design.
- (5) After finding an optimum design for a Type 2 configuration, use that design for the starting point for a Type 3 configuration. This assures a reasonable starting point, probably close to the optimum for Type 3.
- (6) Repeat steps (1) through (3) for a Type 3 configuration.
- (7) Repeat steps (5) and (6) for the remaining configuration.

e. Design Matrix

The design matrix is presented in Table 2. The optimum design is a Type Four configuration and is shown in Figure 18. Typically, for this case study, when starting far from the final design, COPES/CONMIN would require approximately 1700 calls to ANALIZ to reach an optimum. However when beginning from a reasonable starting point with CC = 100, only 600 calls were needed. Note that each call to ANALIZ requires approximately .06 seconds of CPU time on an IBM 360/67.

2. Case Study Two

a. Problem Formulation

An air heater is to be designed to fit into a space 8' X 24' X 4'. The heat exchanger is to heat 1,000,000 lbm/hr of dry air from 75°F to 130°F. 256,000 lbm/hr of water at 200°F is available. Design the heater so that the required air horsepower is at a minimum.

Table 2. CASE STUDY 1 RESULTS

		$\dot{m}_h = 133333 \text{ lbm/hr}$			$\dot{m}_a = 1190476 \text{ lbm/hr}$			Rectangular Fin Profile		
		$T_{h1} = 200 \text{ }^\circ\text{F}$			$T_{c1} = 95 \text{ }^\circ\text{F}$			Aluminum Fins		
		$T_{h2} = 125 \text{ }^\circ\text{F}$			$T_{c2} = 130 \text{ }^\circ\text{F}$			Copper Tubes		
								$\dot{Q} = 10,000,000 \text{ BTU/hr}$		
	INITIAL DESIGN TYPE 1	3 ROW 3 PASS	2 ROW 1 PASS	3 ROW 1 PASS	4 ROW 1 PASS	2 ROW 2 PASS	4 ROW 2 PASS	4 ROW 4 PASS	5 ROW 5 PASS	
D_i , inches	2.0	.48	1.096	.733	.444	.679	.539	.535	.573	
D_o , inches	2.5	.523	1.134	.772	.481	.720	.576	.575	.609	
ℓ , inches	.46	.0824	.393	.291	.276	.162	.128	.0856	.0733	
t , inches	.023	.0157	.0235	.0235	.0217	.0217	.0208	.0172	.0165	
S , inches	.111	.08	.08	.08	.08	.08	.08	.08	.08	
P_t , inches	4.0	.742	1.922	1.364	1.162	1.044	1.02	.922	1.044	
P_L , inches	2.125	.707	1.927	1.413	1.041	1.044	.84	.783	.8	
h , inches	350	499.5	224.2	174.0	155.4	342.1	238.4	500.	499.3	
w , inches	490.	120.8	373.8	229.1	195.7	285.6	186.2	106.3	99.2	
$\dot{Q}$, BTU/hr	20650350	9995582	9999134	10000534	9993329	9998108	9999925	9999968	9999278	
Volume, ft ³	553.2	66.1	174.1	88.4	63.8	110.1	72.9	81.3	91.2	

b. Design Variable Framework

Assuming constant specific heats,

$$c_{p_w} = 1.0 \text{ BTU/lbm-}^\circ\text{F}$$

$$c_{p_a} = .24 \text{ BTU/lbm-}^\circ\text{F}$$

eqs. (12) and (13) will yield the required heat transfer rate and the outlet water temperature.

$$\dot{Q} = \dot{m} c_{p_a} (T_{c_2} - T_{c_1}) = 13.2 \times 10^6 \text{ BTU/hr}$$

$$T_{h_2} = \frac{\dot{m}_h c_{p_w} T_{h_1} - \dot{Q}}{\dot{m}_h} = 148.44 \text{ }^\circ\text{F}$$

Therefore, the design variables are as follows:

$.232 < D_i < 2.325 \text{ in.}$	$D_i^i = 2.0 \text{ in.}$
$.25 < D_o < 2.5 \text{ in.}$	$D_o^i = 2.5 \text{ in.}$
$.0625 \text{ in.} < l < \infty$	$l^i = .46 \text{ in.}$
$.01 < t < .0235 \text{ in.}$	$t^i = .023 \text{ in.}$
$.08 < s < .125 \text{ in.}$	$s^i = .111 \text{ in.}$
$0.0 < P_L < \infty$	$P_L^i = 2.125 \text{ in.}$
$0.0 < P_t < \infty$	$P_t^i = 4.00 \text{ in.}$

c. Constraint Framework

When designing for minimum horsepower, the optimizer will naturally try to drive the design to a maximum volume in order to reduce airside pressure losses. Therefore, it is reasonable to constrain volume as follows:

$$0 < \text{Volume} < 768 \text{ ft}^3$$

The other constraints are:

$$0.0 < \theta < 1.3$$

$$1.0 < \text{DRATIO} < 2.5$$

$$.018 < \text{TUBTH} < .18 \text{ in.}$$

$$0 < \Delta p_w < .14 \text{ psi}$$

and for the heat balance:

$$\frac{\dot{Q}_5}{\dot{Q}} = 1$$

d. Design Matrix

With the tube and fin materials fixed, as in case one, the matrix is presented in Table 3. The optimum design is a Type 4 configuration, as shown in Figure 19. In case study two when starting far from the optimum, the optimizer called ANALIZ approximately 1900 times. When starting close to the final design, for example using the design for a Type 3 configuration as a starting point for the Type 4 design, COPES/CONMIN only required 596 calls.

Table 3. CASE II DESIGN MATRIX

$\dot{m}_h = 256,000 \text{ lbm/hr}$ $\dot{m}_c = 1,000,000 \text{ lbm/hr}$ Rectangular Fin Profile
 $T_{h1} = 200^\circ\text{F}$ $T_{C1} = 75^\circ\text{F}$ Aluminum Fins/Copper Tubes
 $T_{h2} = 148.44^\circ\text{F}$ $T_{C2} = 130^\circ\text{F}$ $\dot{Q} = 13,200,000 \text{ BTU/hr}$
 $P_\infty = 14. \text{ psi}$ $h = 96 \text{ in}$ $w = 288 \text{ in}$

DESIGN VARIABLES	INITIAL DESIGN TYPE 1	1 ROW 1 PASS	2 ROW 1 PASS	3 ROW 1 PASS	4 ROW 1 PASS	2 ROW 2 PASS	3 ROW 3 PASS	4 ROW 2 PASS	4 ROW 4 PASS
D_i , inches	2.0	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.14	1.81	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.31	1.93	2.32
D_o , inches	2.5	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.19	1.85	COULD NOT PROVIDE REQUIRED $\dot{Q}$	2.347	1.96	2.355
ℓ , inches	.46	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	1.21	.90	COULD NOT PROVIDE REQUIRED $\dot{Q}$	.385	.82	.12
t , inches	.023	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	.0235	.0235	COULD NOT PROVIDE REQUIRED $\dot{Q}$	.0189	.0234	.0176
S , inches	.111	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	.08	.08	COULD NOT PROVIDE REQUIRED $\dot{Q}$	.08	.085	.08
P_t , inches	4.0	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	4.90	4.93	COULD NOT PROVIDE REQUIRED $\dot{Q}$	3.12	4.93	2.68
P_L , inches	2.125	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	7.43	7.50	COULD NOT PROVIDE REQUIRED $\dot{Q}$	4.41	7.50	4.69
$\dot{Q}$, BTU/hr	4486222	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	13200986	13200819	COULD NOT PROVIDE REQUIRED $\dot{Q}$	13202784	13199363	13205473
FHP	51.96	COULD NOT PROVIDE REQUIRED $\dot{Q}$	COULD NOT PROVIDE REQUIRED $\dot{Q}$	25.96	22.06	COULD NOT PROVIDE REQUIRED $\dot{Q}$	128.6	23.52	395.8

V. CONCLUSIONS

The intent of this investigation was to couple an analysis program with a numerical optimization scheme, COPES/CONMIN, to produce a complete, detailed design program for an air-cooled heat exchanger, (HEDSUP). In addition, the analysis program was to be written such that:

1. the variation in the film coefficients with temperature/length of flow path would be taken into account
2. the surface would be optimized
3. it would be iterative free and thus minimize the CPU time required during an actual trade off study.

The results from test cases using ANALIZ coupled with COPES/CONMIN in its present form were unsatisfactory. Although COPES/CONMIN could optimize the objective function satisfying the inequality constraints, a reliable heat balance could not be obtained. The solution to this problem was the addition of the ALMM option to COPES. In this way, the method of feasible directions, which works best with inequality constraints, was used to satisfy the inequality constraints. The multiplier method, which works best with equality constraints, was used to perform the heat balance.

The results of the case studies show that HEDSUP will yield reliable designs for various design objectives and problems with only some trial and error application of the initial Lagrange multiplier. However, precautions must be taken to overcome the relative minima that plague this design

problem. Table 1 shows vividly the problem of relative minima with one "optimum" design having a volume over 100% greater than the "true" optimum.

The value of numerical optimization in a design problem of this size cannot be overemphasized. For example, in case study 1, the problem is taking place in a nine-dimensional design space and intuition on how an optimized design "should" turn out is quickly lost. Figure 20 helps to illustrate this point. When only varying two design variables, h and w , Figure 20 shows that, when beginning from the initial starting point with a Type 7 configuration, a design satisfying both the equality constraint and the air pressure drop constraint could never be found. However, Table 2 indicates how the numerical optimization routine has varied the other seven design variables in order to shift the constraints and yield an optimized design.

The results show that AHDOP did vary the surface design variables: D_i , D_o , λ , t , S , P_t and P_L , in order to produce an optimum heat exchanger. This capability of surface optimization is dependent upon the use of the Briggs-Young and Robinson-Briggs Correlations, eqs. (32) and (34a), respectively. The reliability of the correlations as compared to the "conventional" method is questionable. Actual experimental data for a particular tube and pitch will always be the most useful in predicting pressure drop and film coefficients of across banks of finned tubes. However, the correlations

mentioned above cover the ranges and pitches used in air-cooled heat exchangers, and should therefore be sufficiently accurate in predicting H_o and Δp_a .

VI. RECOMMENDATIONS

In addition to the insight that this investigation has given into the generation of an automated air-cooled heat exchanger design, it has also generated an awareness of this investigation's shortcomings. As mentioned in the review of previous work in this area, each of the optimization methods has its own limitations; none is completely general. Presented herein are recommendations for improving upon and furthering development of HEDSUP.

1. HEDSUP should be expanded to include the capability for two-phase tubeside fluids. Mott, et al. [2] discusses a method involving two-phase tubeside fluids that would be compatible with HEDSUP. The modular design of ANALIZ will aid this effort.

2. Research in the area of numerical optimization using discrete variables would benefit HEDSUP immensely. With discrete variables, the design of the exchanger could be accomplished with "off-the-shelf" materials. At present, the use of the optimizer is restricted to continuous variables with continuous first derivatives. The ability to work with discrete variables would also eliminate the need for the design matrix. The optimizer could optimize for type configuration, fin profile, fin material and tube material.

3. Additional research with ALMM is needed to remove the "artwork" involved with choosing an initial multiplier and

thus increase reliability and hence reduce CPU time. The research should be concentrated in two areas: 1) debugging the new optimizer with AFMM and 2) scaling of design variables.

4. The addition of cost as an objective function would increase the attractiveness of HEDSUP. Mott, et al. [2], and Fontein and Wassink, [18], have presented much useful information in this regard.

5. Mechanical constraints such as tube bursting stress and tube vibrations should be included in HEDSUP.

VII. FIGURES

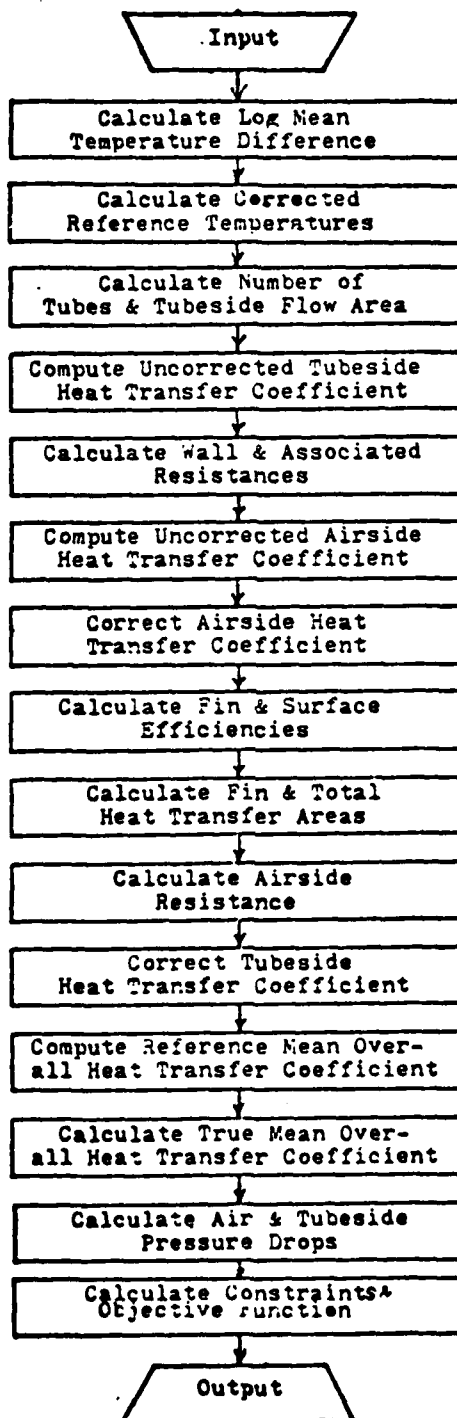


Figure 1

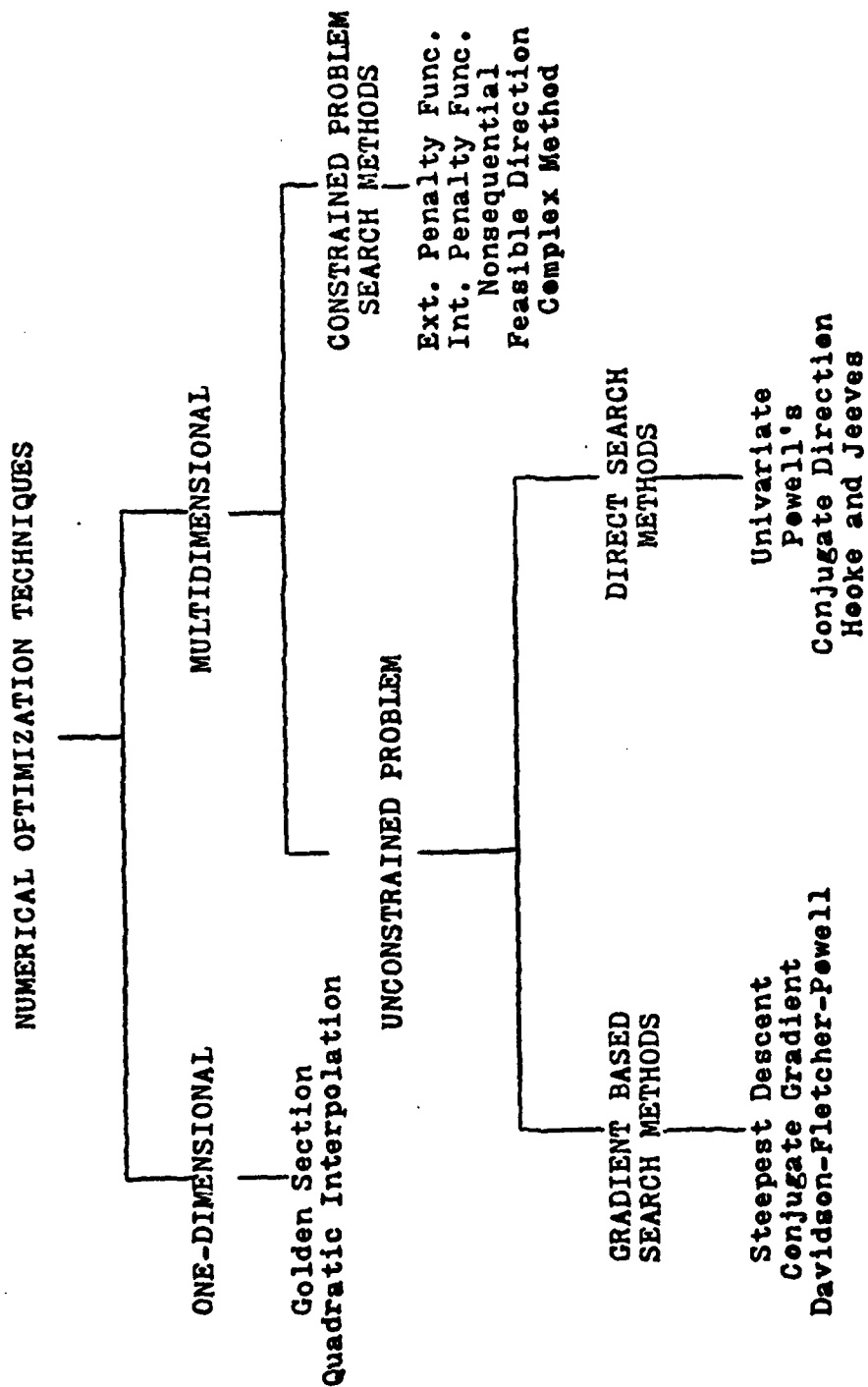
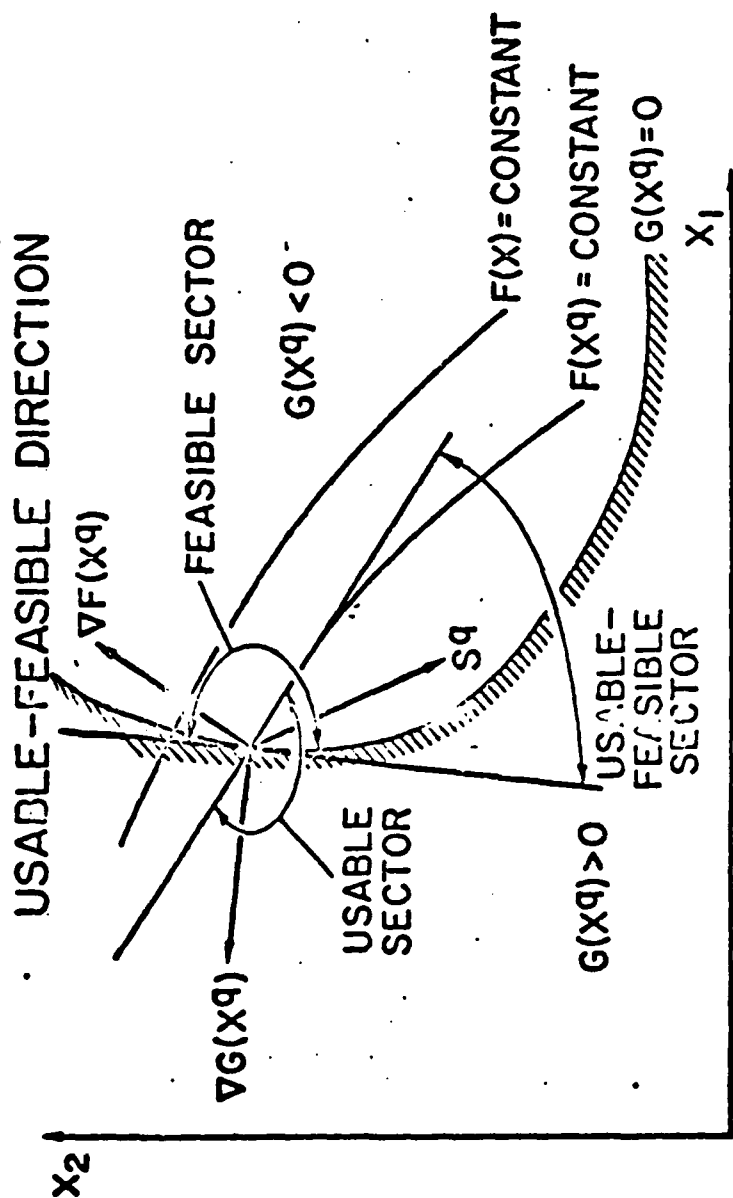


Figure 2



IF CURRENT DESIGN IS FEASIBLE (ALL $G_j(x_q) \leq 0$)

$$\nabla F(x_q) \cdot s_q \leq 0$$

USABLE

$$\nabla G_j(x_q) \cdot s_q \leq 0 \quad j=1, NAC$$

FEASIBLE

s_q BOUNDED

IF CURRENT DESIGN IS INFEASIBLE (SOME $G_j(x_q) > 0$)

ALL DIRECTIONS, s_q ARE USABLE

$$\nabla G_j(x_q) \cdot s_q \leq 0$$

FEASIBLE

Figure 3

ONE-DIMENSIONAL SEARCH IN DIRECTION S^q

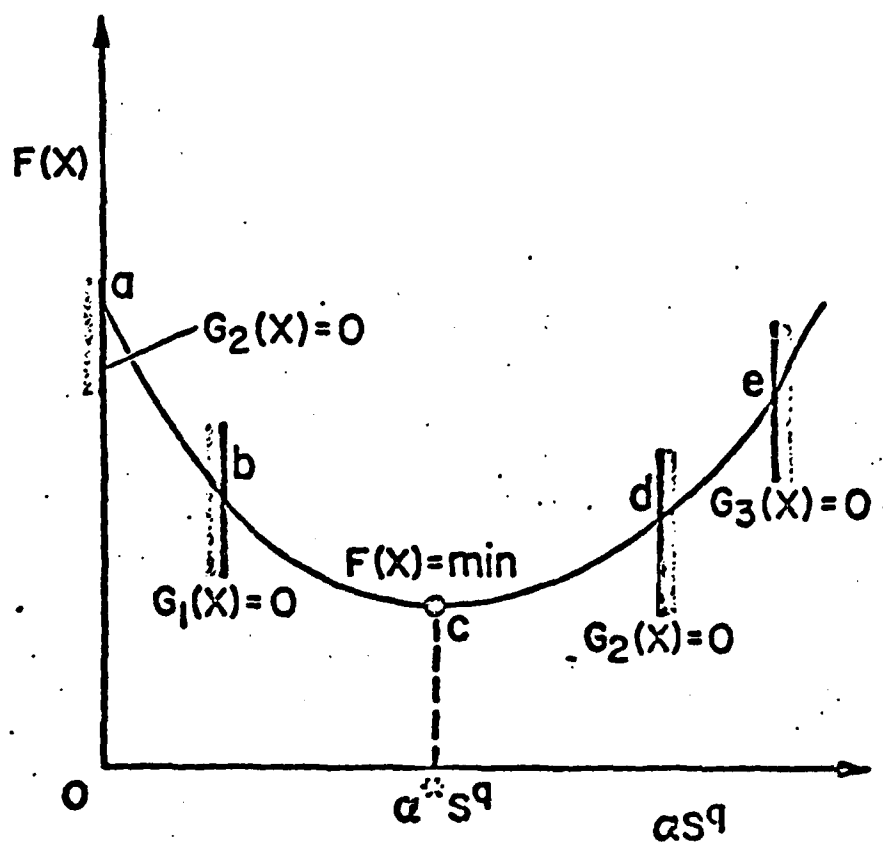


Figure 4

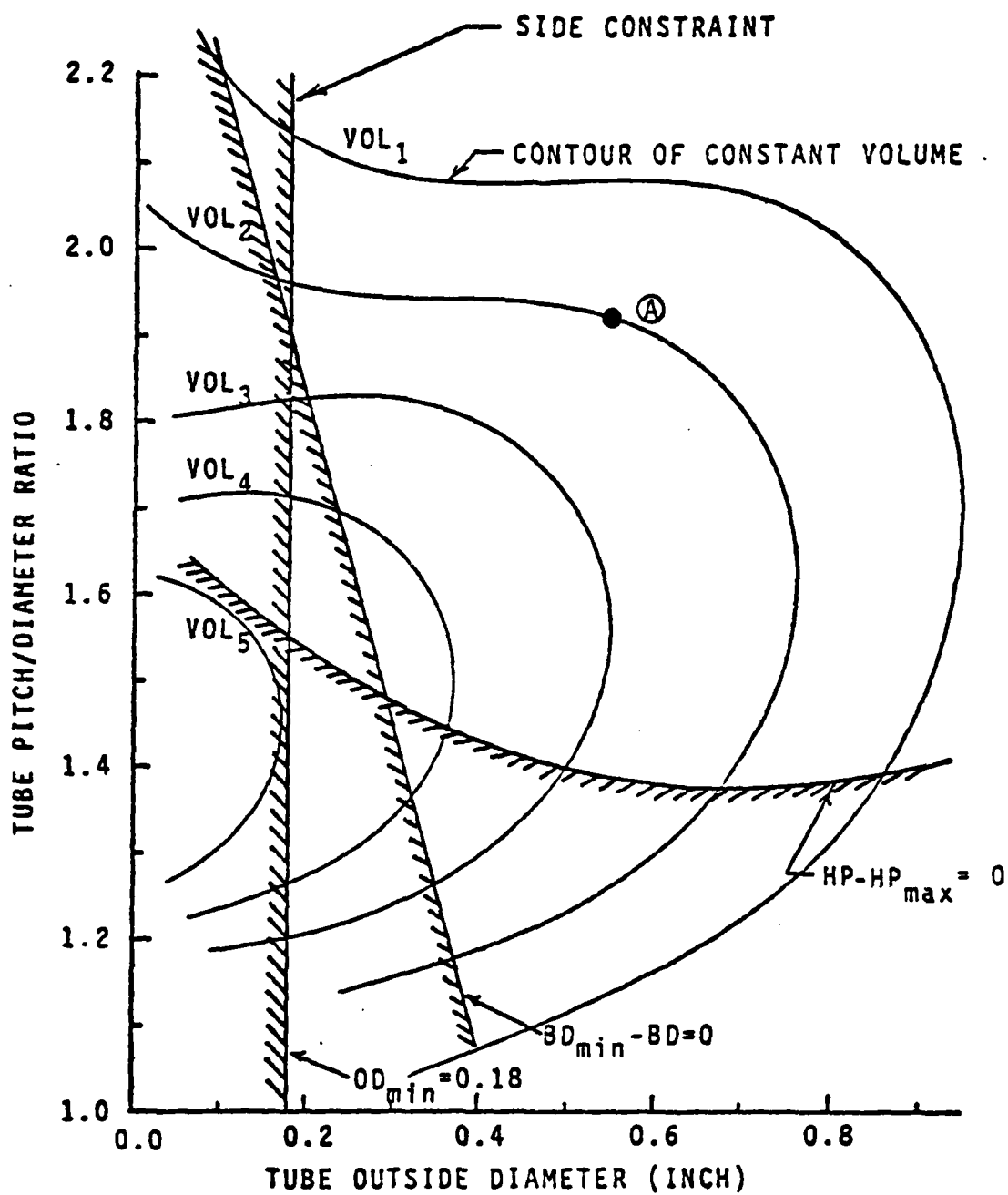


Figure 5

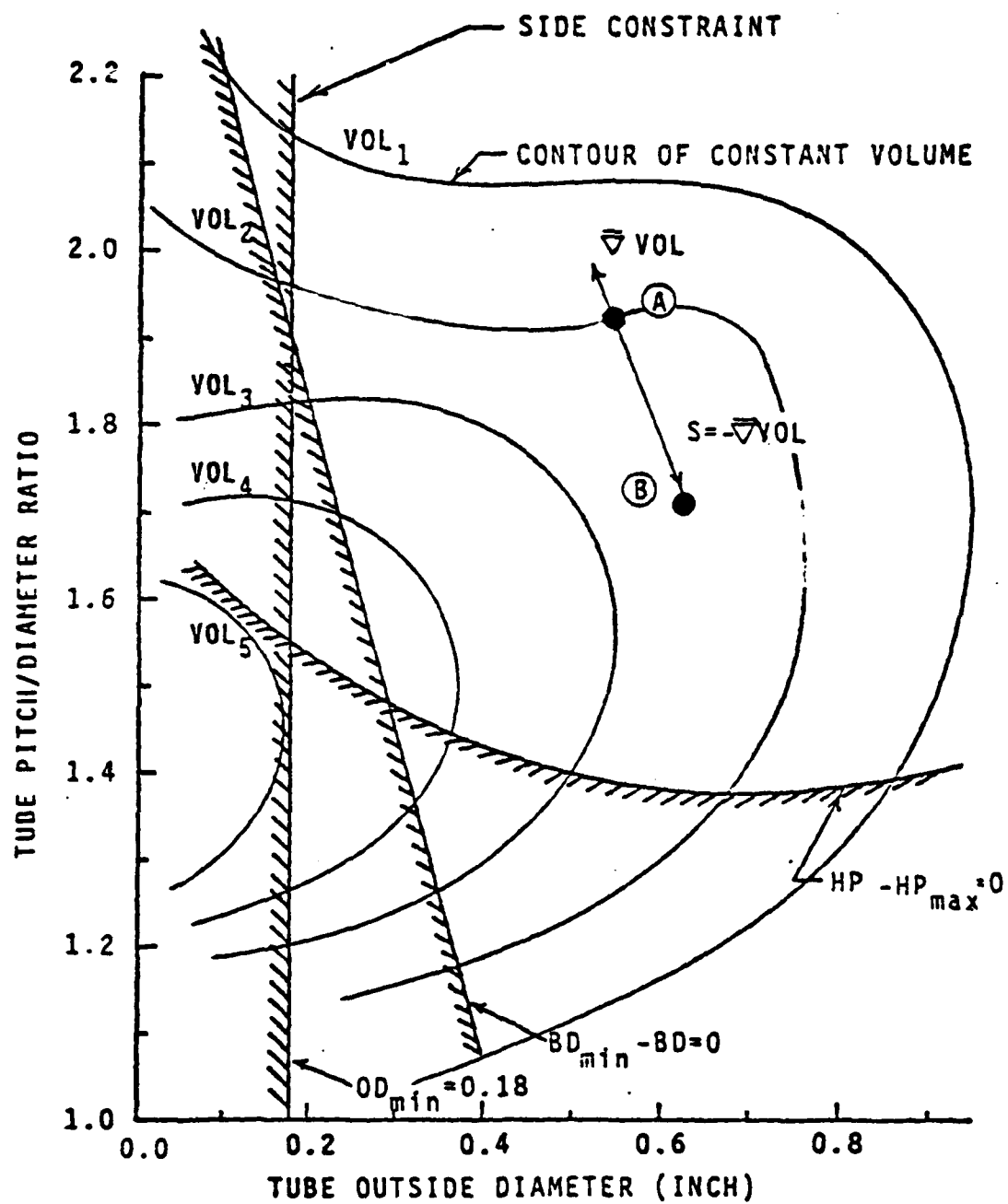


Figure 6

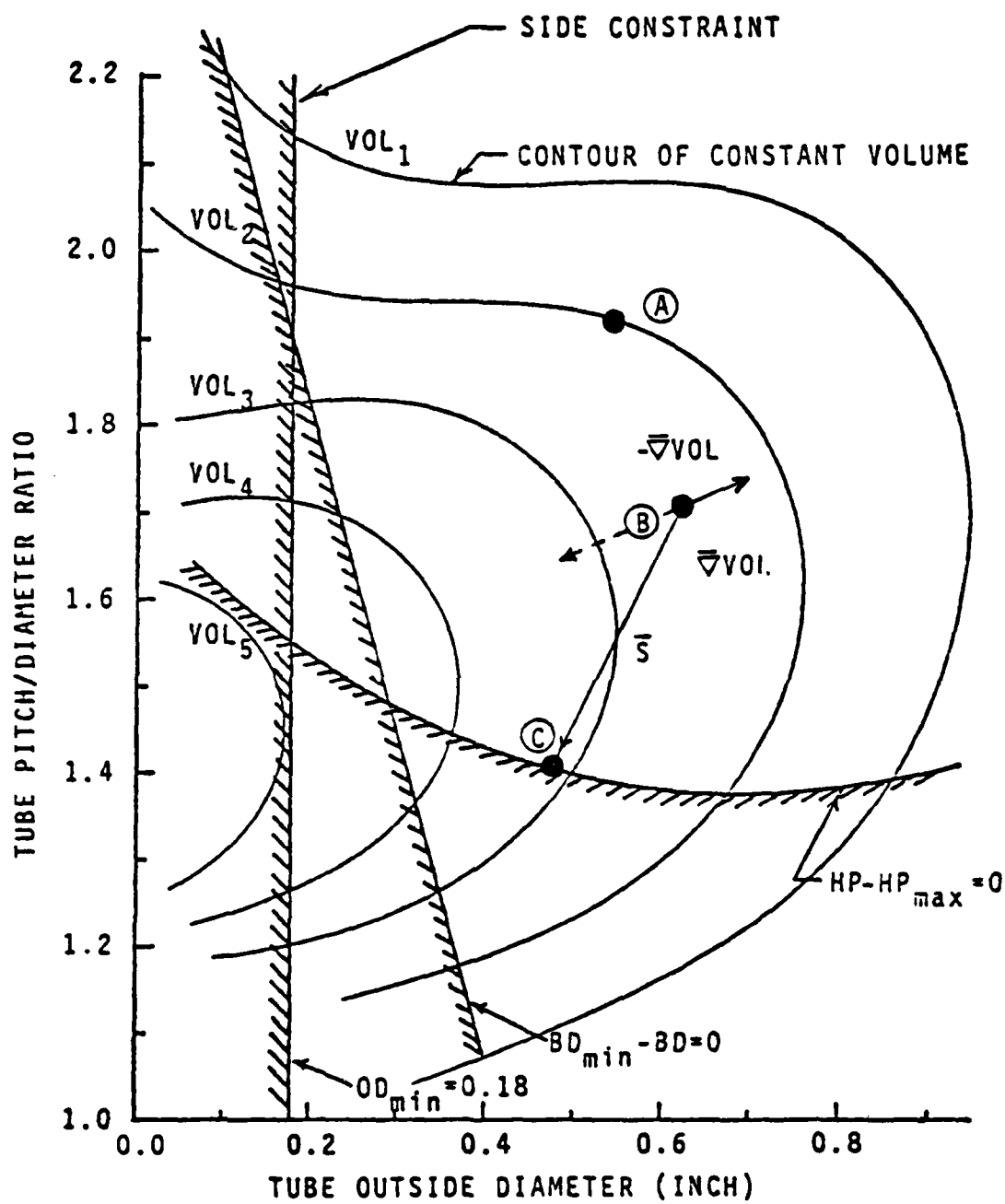


Figure 7.

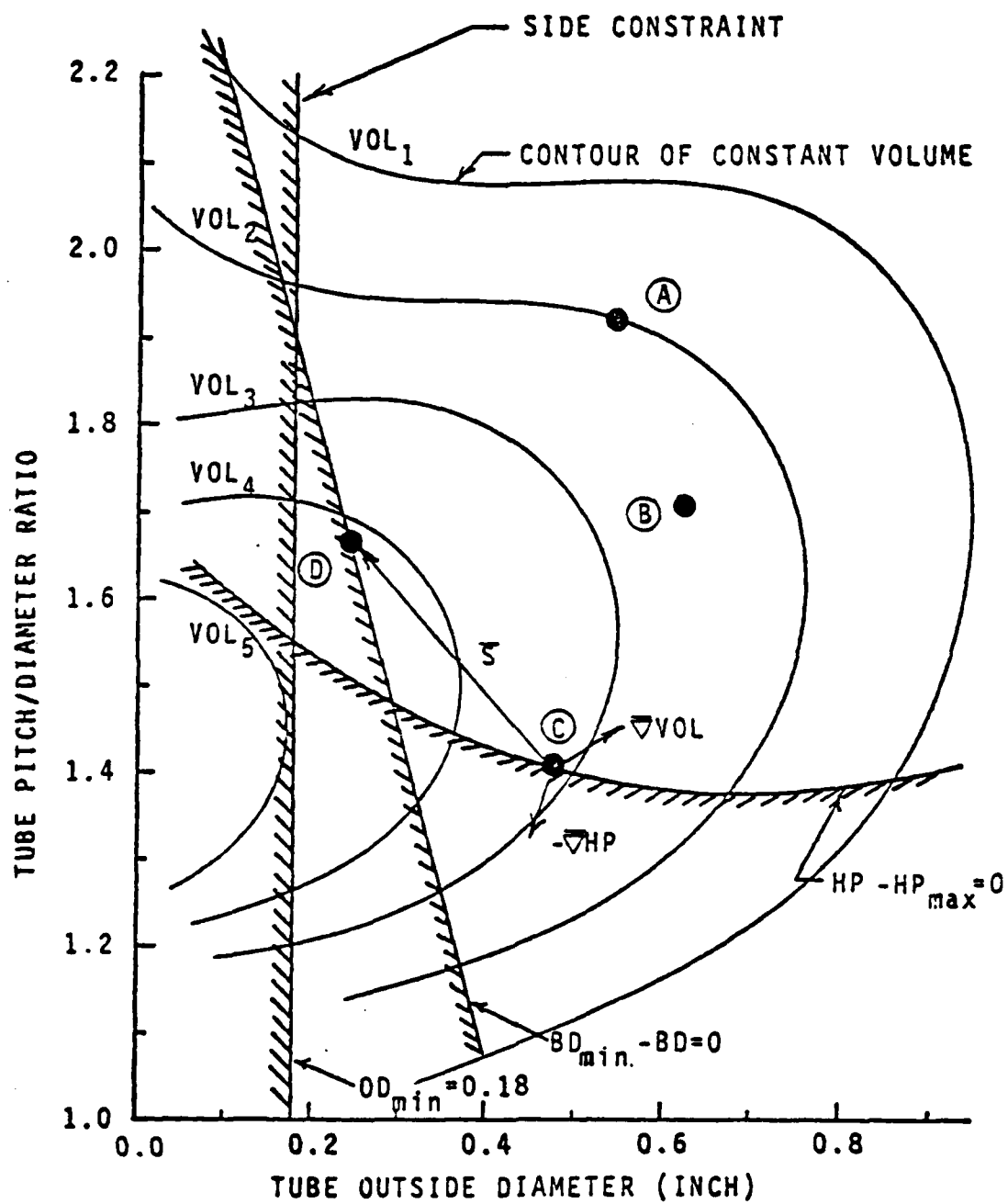


Figure 8

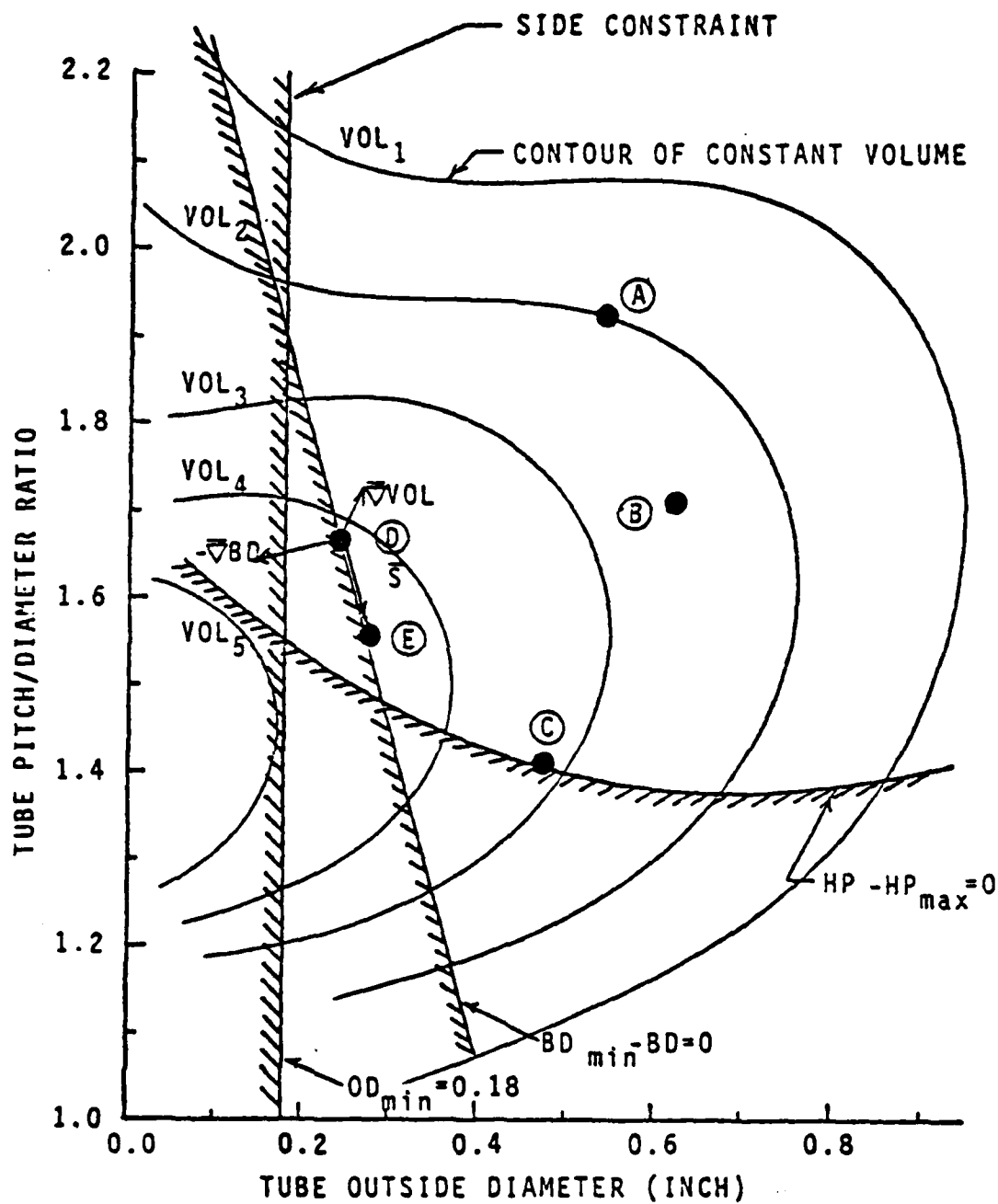


Figure 9

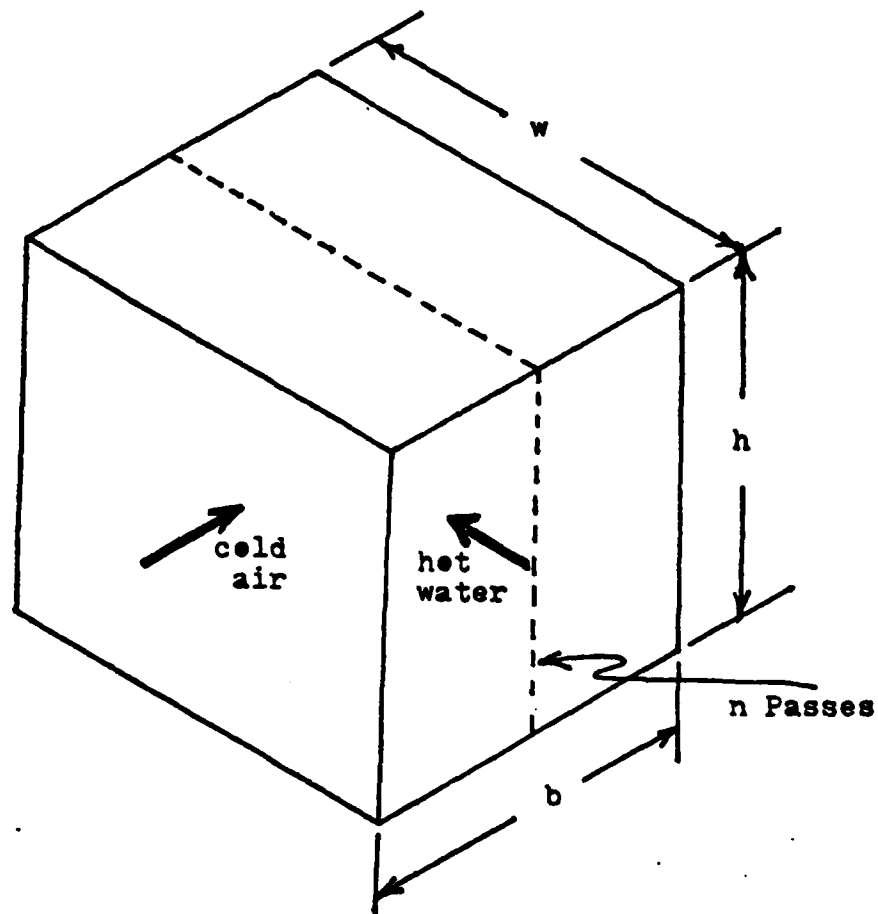


Figure 10. Configuration of Air-Cooled Heat Exchanger

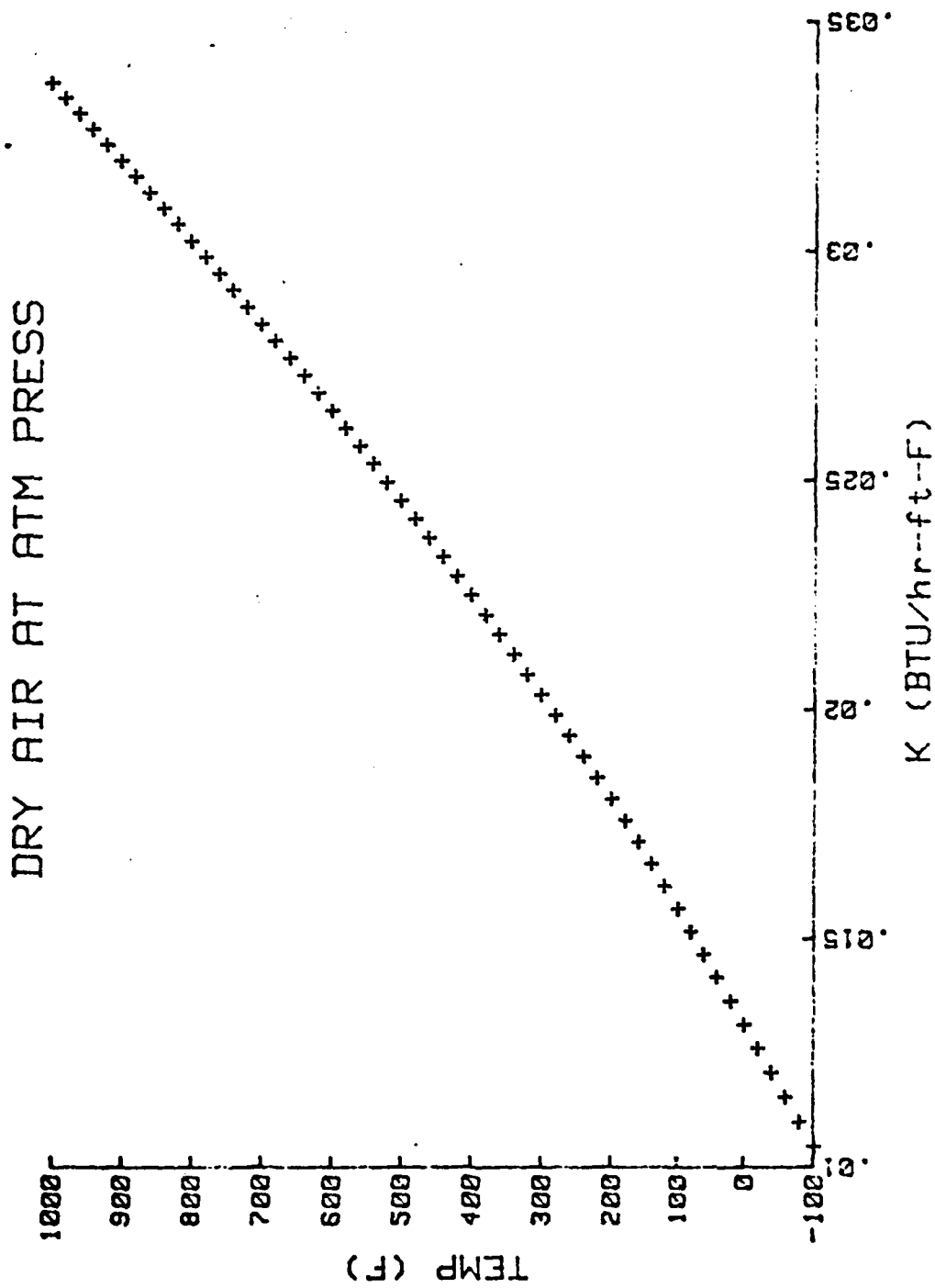


Figure 11

WATER (SATURATED LIQUID)

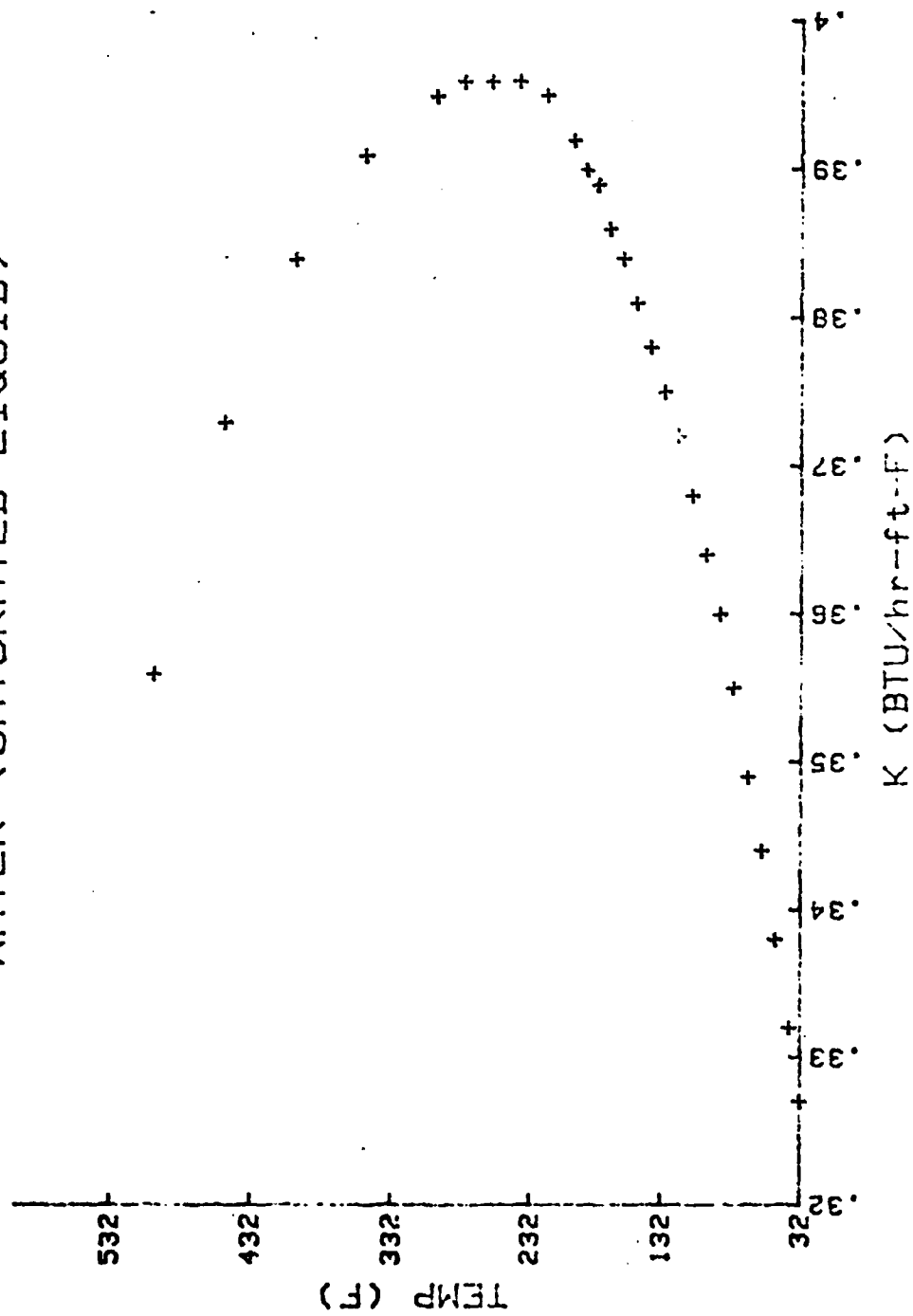


Figure 12

DRY AIR AT ATM PRESS

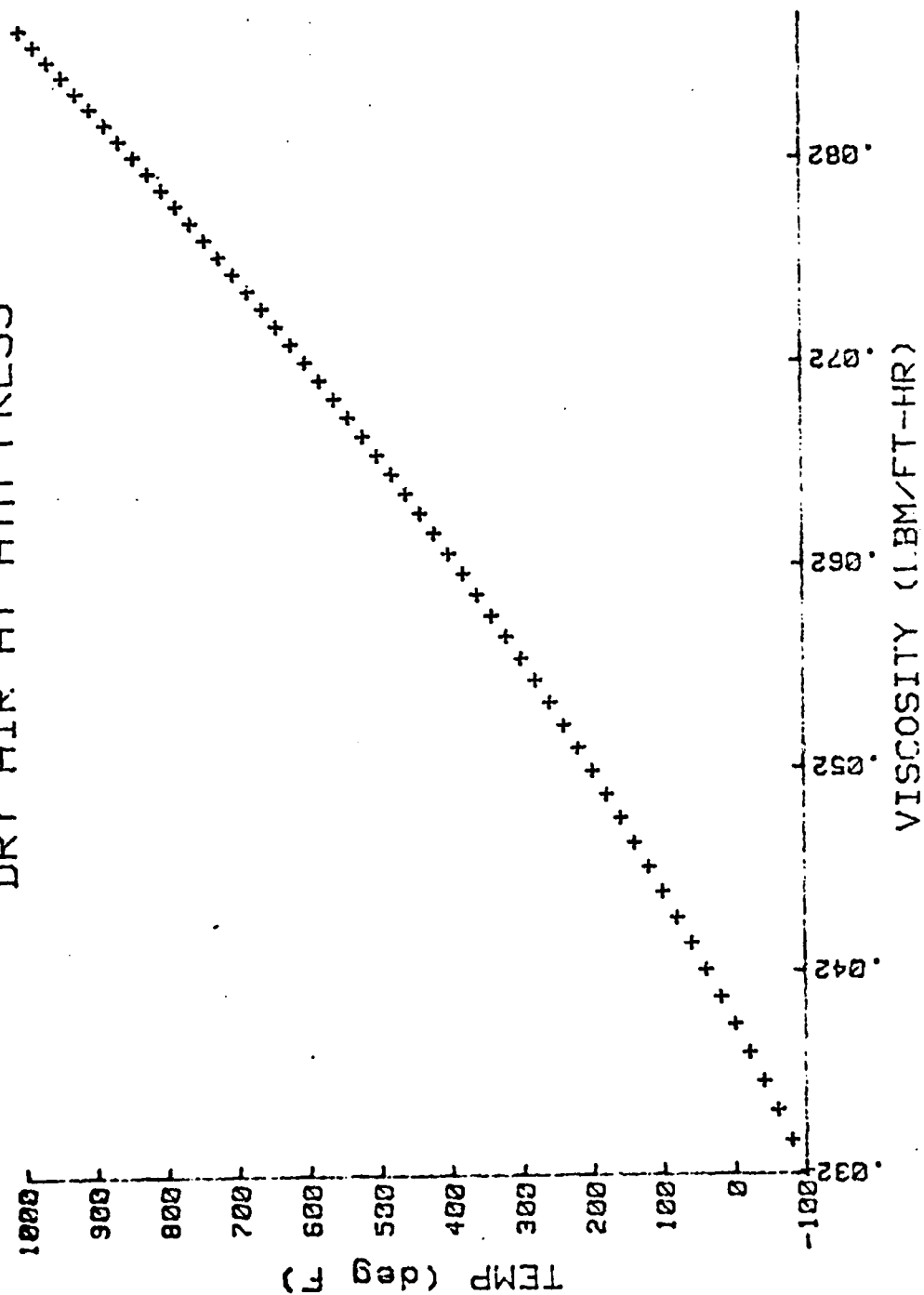


Figure 13

VISCOSITY OF WATER (Andrade's & Exp.)

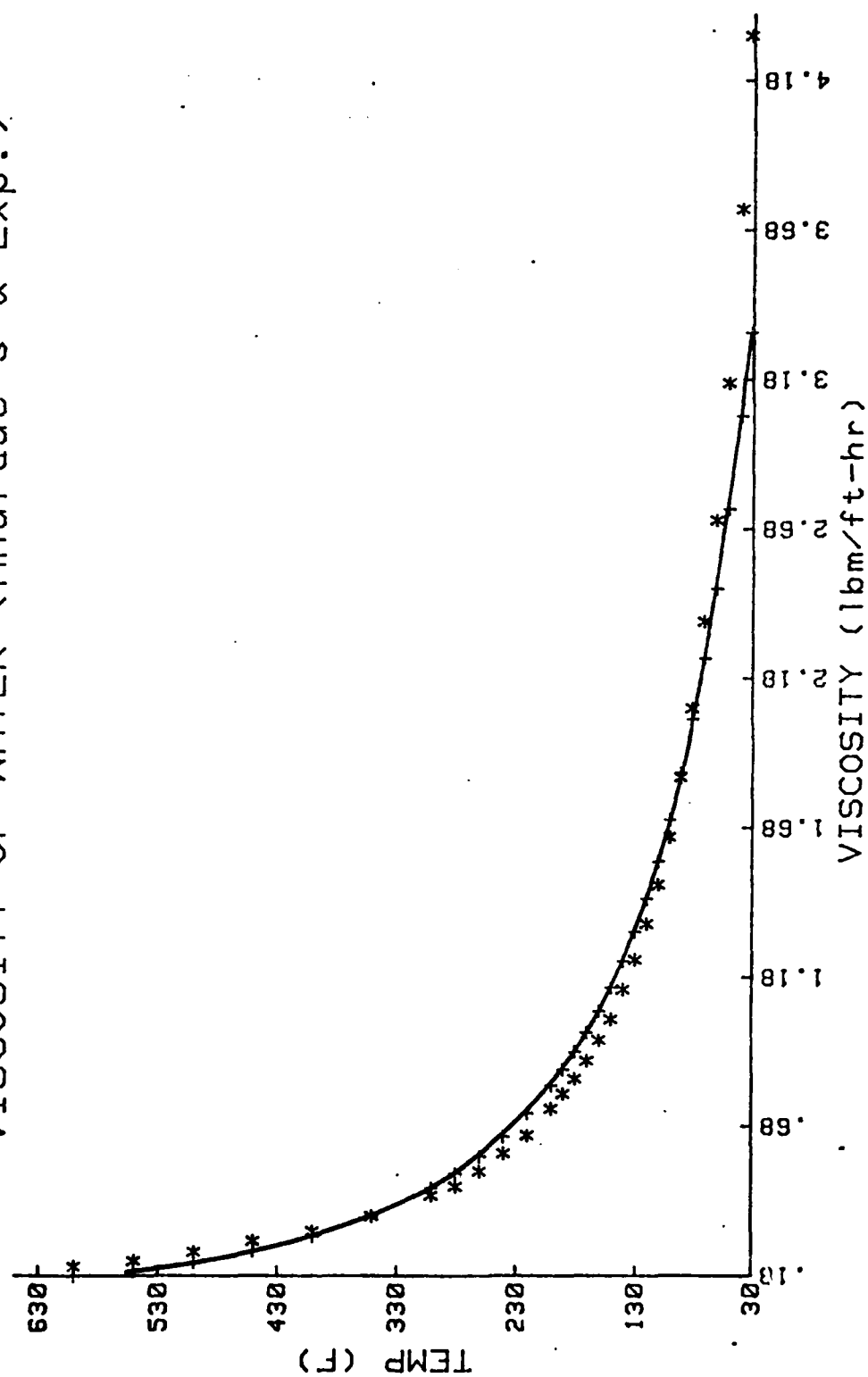


Figure 14

4 ROW, 2 PASS ARRANGEMENT

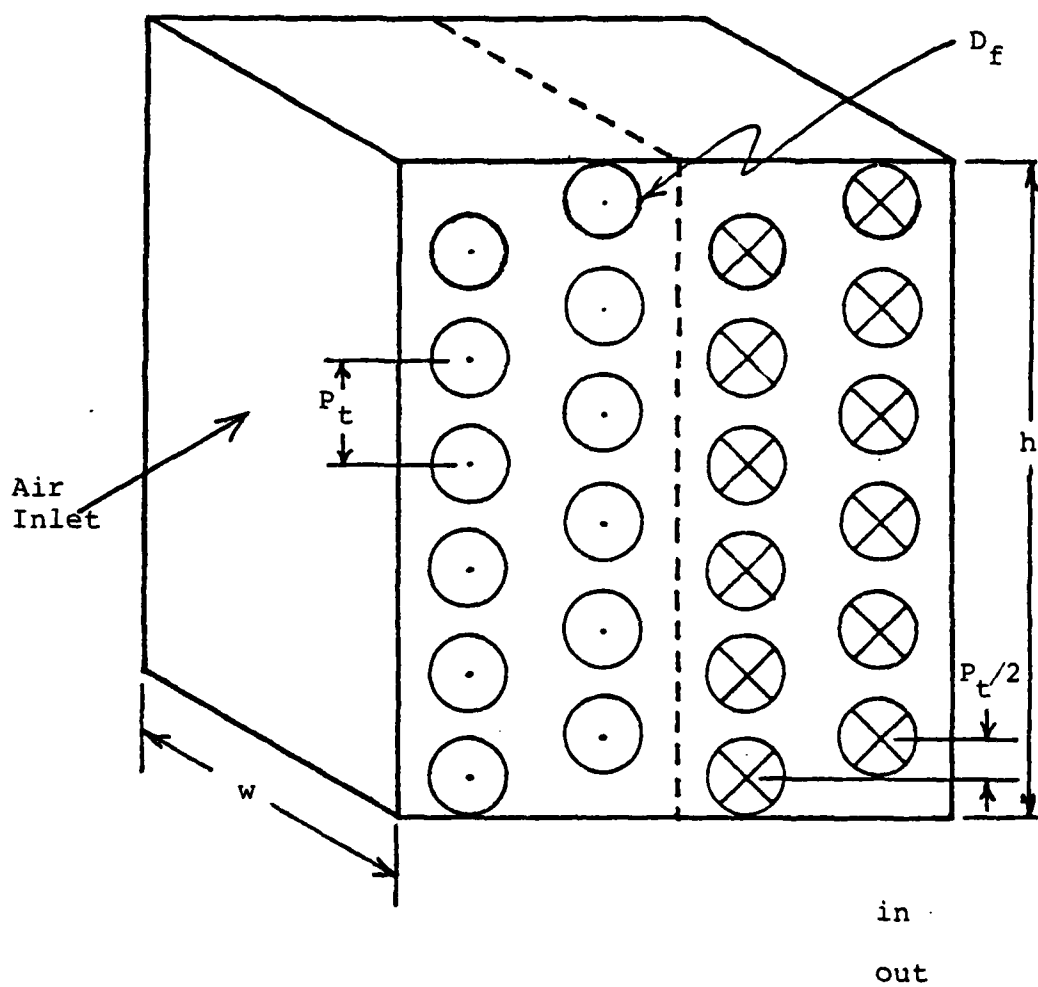


Figure 15

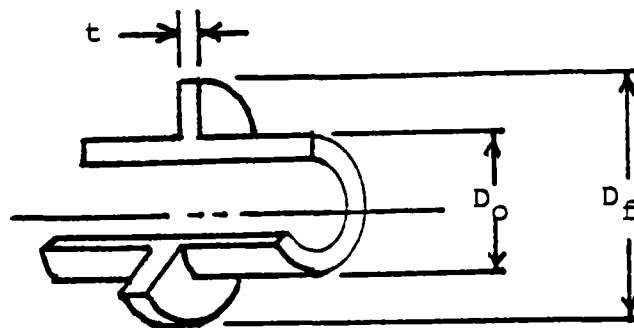


Figure 16

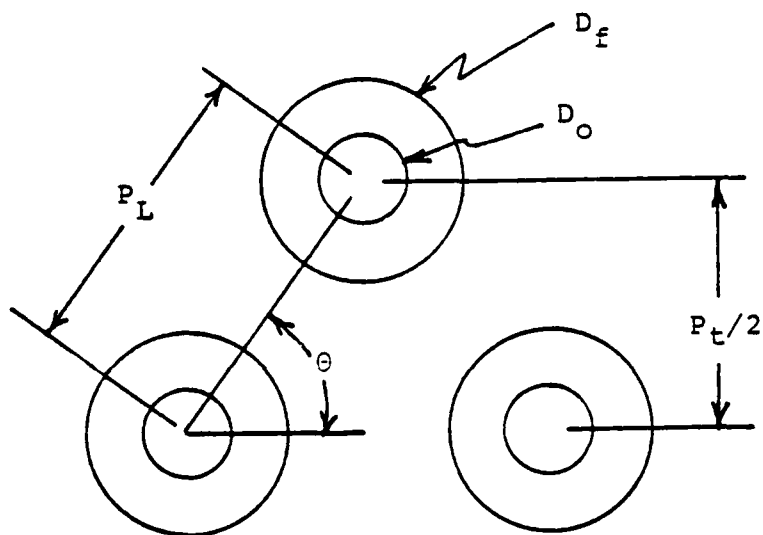


Figure 17

CASE STUDY ONE

DESIGN OPTIMUM

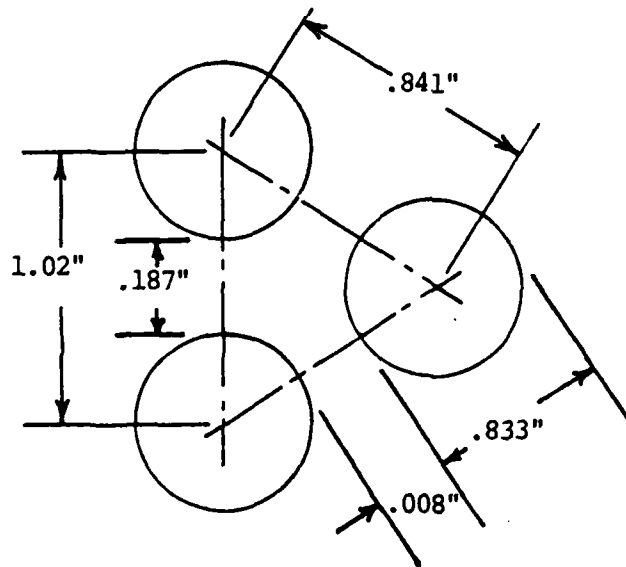
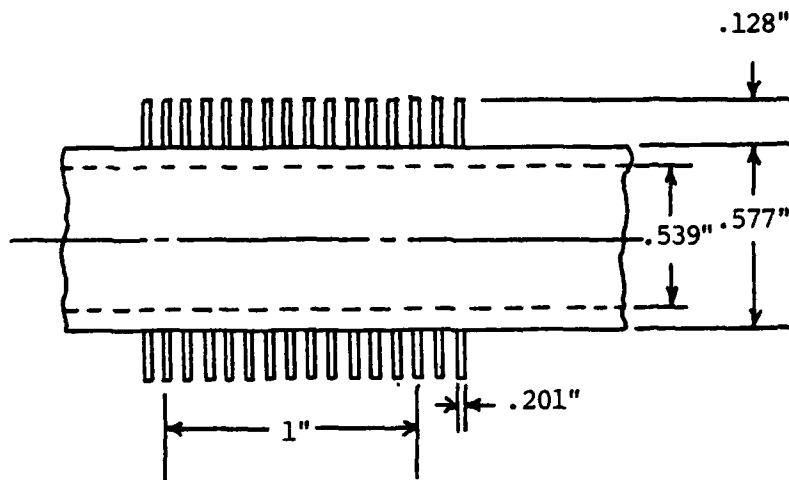


Figure 18

OPTIMUM DESIGN

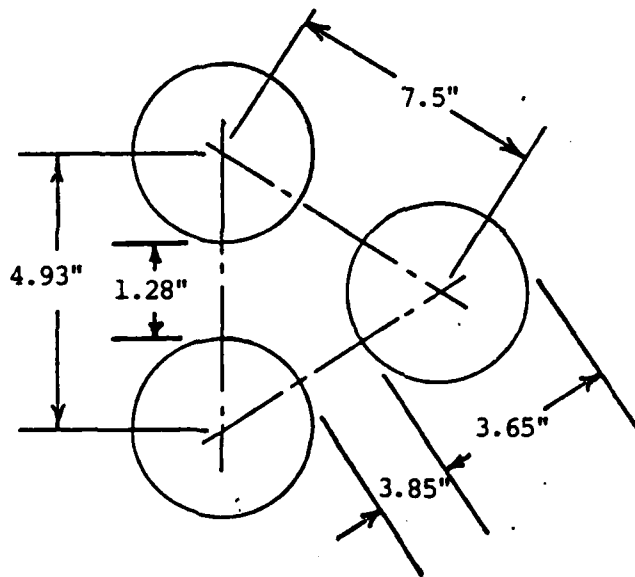
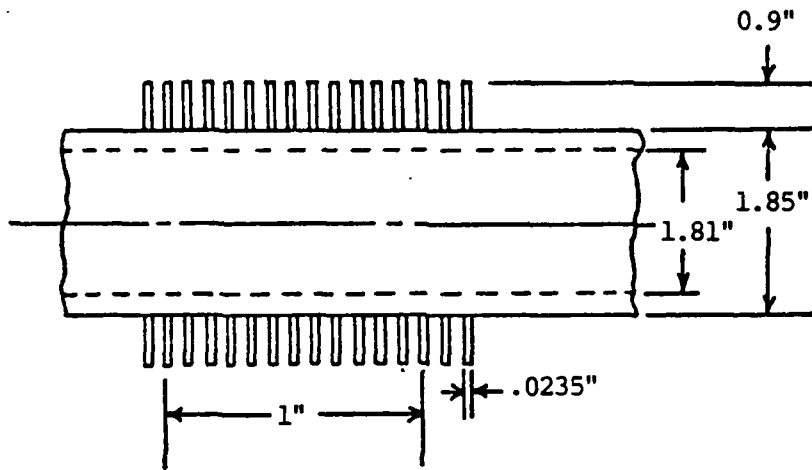


Figure 19

2 VARIABLE FUNCTION SPACE

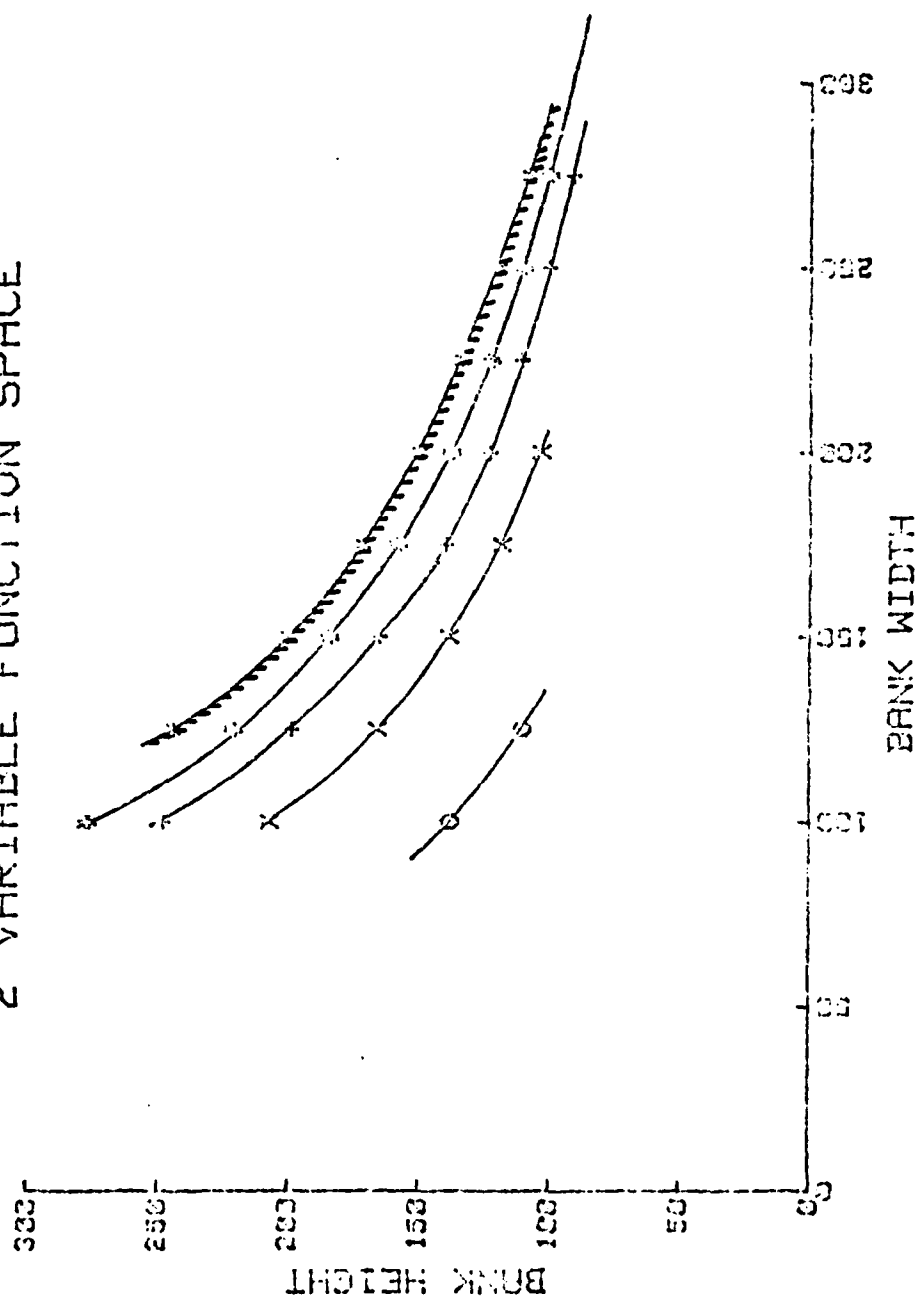


Figure 20

APPENDIX A

HEAT EXCHANGER DESIGN USING NUMERICAL OPTIMIZATION (HEDSUP)

USERS MANUAL

In order to execute HEDSUP, it is necessary to provide formatted data for COPES [25], followed by formatted data for the ANALIZ portion of the program. This section defines the data which must be supplied. The data is submitted in segmented blocks. All formats are alphanumeric for TITLE and END cards, F10 for real, and I10 for integer data when supplying COPES. For ANALIZ, formats are F14 for real and I10 for integer data.

While the COPES user's sheets define the data in formatted fields of ten, the COPES manual does provide means of simplifying this task through unformatted data input.

The included GLOBAL CATALOG defines objective functions, design variables and constraints along with their location for ease in compiling the necessary COPES data.

GLOBAL CATALOG

GLOBAL LOCATION	FORTTRAN NAME	DEFINITION
1	FMDOT	Fluid Mass Flow Rate, lbm/hr
2	FLTMP1	Inlet Fluid Temperature, °F
3	FLTMP2	Outlet Fluid Temperature, °F
4	AMDOT	Air Mass Flow Rate, lbm/hr
5	ARTMP1	Inlet Air Temperature, °F
6	ARTMP2	Outlet Air Temperature, °F
7	TUBEID	Tube Inside Diameter, in
8	TUBEOD	Tube Outside Diameter, in
9	FINHT	Fin Height, in
10	FINTH	Fin Thickness, in
11	FINSF	Fin Spacing, in
12	PITCHL	Longitudinal Pitch, in
13	PITCHN	Transverse Pitch, in
14	BANKW	Bank Width, in
15	BANKH	Bank Height, in
16	VOLUME	Volume, ft ³
17	AREA	Heat Transfer Area, ft ²
18	FHP	Air Horsepower, HP
19	PPA	Airside Pressure Drop, psi
20	PPW	Tubeside Pressure Drop, psi
21	DELQ1	$\dot{Q} - \dot{Q}_5$, BTU/hr
22	DELQ2	$\dot{Q}_5 - \dot{Q}_4$, BTU/hr
23	DELQ3	$\dot{Q}_4 - \dot{Q}_3$, BTU/hr
24	DRATIO	D_f/D_o
25	TUBTH	Tube Wall Thickness, in
26	TOUCHN	Tip-to-Tip Clearance, Transverse
27	TOUCHL	Tip-to-Tip Clearance, Longitudinal

GLOBAL CATALOG (Cont.)

[illegible]

DATA BLOCK A

DESCRIPTION: Title card.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7	8	FORMAT
TITLE								20A4

FIELD

CONTENTS

1-8 Any 80 character title may be given on this card.

DATA BLOCK B

DESCRIPTION: Program Control Parameters.

FORMAT AND EXAMPLE

		1	2							FORMAT
		NCALC	NDV							7110

CONTENTS

FIELD

- 1 NCALC: Calculation Control
 - 0 - Read input and stop. Data of blocks A, B and V is required. Remaining data is optional.
 - 1 - One cycle through program. The same as executing ANALIZ stand-alone. Data of blocks A, B and V is required. Remaining data is optional.
 - 2 - Optimization. Data of blocks A-I and V is required. Remaining data is optional.
 - 3 - Augmented Lagrangian Method.

DATA BLOCK C OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Integer optimization control parameters.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7	8	FORMAT
IPRINT	ITMAX	ICNDR	NSCAL	ITRM	LINOBJ	NACEX1	NFDG	8110

FIELD

CONTENTS

- 1 IPRINT: Print control used in the optimization program CONMIN.
 - 0 - No print during optimization.
 - 1 - Print initial and final optimization information.
 - 2 - Print above plus objective function value and design variable values at each iteration.
 - 3 - Print above plus constraint values, direction vector and move parameter at each iteration.
 - 4 - Print above plus gradient information.
 - 5 - Print above plus each proposed design vector, objective function and constraint values during the one-dimensional search.

FIELD

CONTENTS

- 2 ITMAX: Maximum number of optimization iterations allowed. DEFAULT = 20.
- 3 ICNDR: Conjugate direction restart parameter. DEFAULT = NDV + 1.
- 4 NSCAL: Scaling parameter. GT.0 - Scale design variables to order of magnitude one every NSCAL iterations. LT.0 - Scal design variables according to user-input scaling values.
- 5 ITRM: Number of consecutive iterations which must satisfy relative or absolute convergence criterion before optimization process is terminated. DEFAULT = 3.
- 6 LINOBJ: Linear objective function identifier. If the optimization objective is known to be a linear function of the design variables, set LINOBJ = 1.
DEFAULT = Non-linear.
- 7 MACMX1: One plus the maximum number of active constraints anticipated. DEFAULT = NDV + 2.
- 8 NFDG: Finite difference gradient identifier.
0 - All gradient information is computed by finite difference within CONMIN.
1 - All gradient information is computed analytically by the user-supplied code.
2 - Gradient of objective is computed analytically. Gradients of constraints are computed by finite difference within CONMIN.

REMARKS

- 1) Currently NFDG must be zero in COPES.

AD-A096 350

NAVAL POSTGRADUATE SCHOOL MONTEREY CA
HEAT EXCHANGER OPTIMIZATION.(U)
SEP 80 C P HEDDERICH

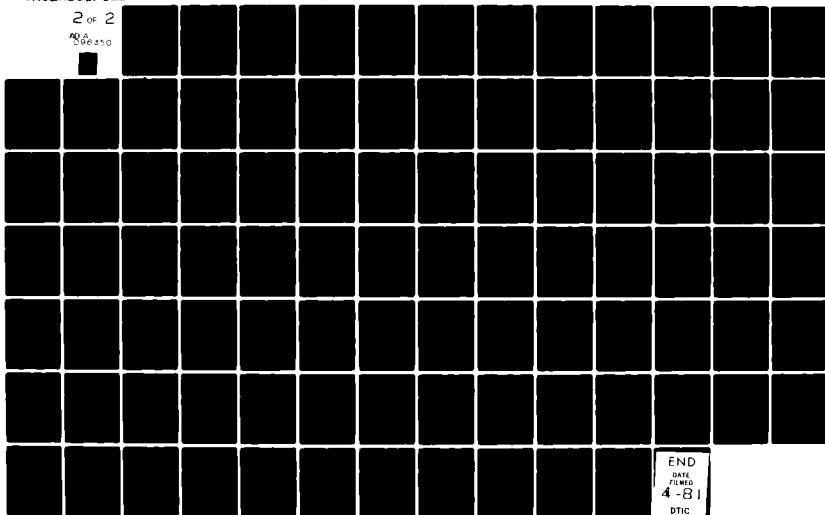
F/6 13/1

UNCLASSIFIED

NL

2 of 2

NOA
096450



END
DATE
FILMED
4-81
DTIC

DATA BLOCK D OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Floating point optimization program parameters.

FORMAT AND EXAMPLE

1	2	3	4	5	6	7	FORMAT
FDCH	FDCHM	CT	CTMIN	CTL	CTLMIN	THETA	7F10
DELFUN	DABFUN	ALPHAX	ABOBJ1				4F10

NOTE: TWO CARDS ARE READ HERE.

FIELD

CONTENTS

- 1 FDCH: Relative change in design variables in calculating finite difference gradients. DEFAULT = 0.01.
- 2 FDCHM: Minimum absolute step in finite difference gradient calculations. DEFAULT = 0.001.

FIELD

CONTENTS

- | | |
|---|--|
| 3 | CT: Constraint thickness parameter. DEFAULT = -0.05. |
| 4 | CTMIN: Minimum absolute value of CT considered in the optimization process.
DEFAULT = 0.004. |
| 5 | CTL: Constraint thickness parameter for linear constraints. DEFAULT = -0.01. |
| 6 | CTLMIN: Minimum absolute value of CTL considered in the optimization process.
DEFAULT = 0.001. |
| 7 | THETA: Mean value of push-off factor in the method of feasible directions.
DEFAULT = 1.0. |
| 1 | DELFUN: Minimum relative change in objective function to indicate convergence of
the optimization process. DEFAULT = 0.001. |
| 2 | DABFUN: Minimum absolute change in objective function to indicate convergence of
the optimization process. DEFAULT = 0.001 times the initial objective value. |
| 3 | ALPHAX: Maximum fractional change in any any design variable for first estimate of
the step in the one-dimensional search. DEFAULT = 0.1. |
| 4 | ABORJ1: Expected fractional change in the objective function for first estimate of
the step in the one-dimensional search. DEFAULT = 0.1. |

REMARKS

- 1) The DEFAULT values for these parameters usually work well.

DATA BLOCK E OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Total number of design variables, design objective identification and sign.

FORMAT AND EXAMPLE

1	2	3	FORMAT
NDVTOT	IOBJ	SGNOPT	2I10,F10

FIELD

CONTENTS

- 1 NDVTOT: Total number of variables linked to the design variables. This option allows two or more parameters to be assigned to a single design variable.
The value of each parameter is the value of the design variable times a multiplier, which may be different for each parameter. DEFAULT = NDV.
- 2 IOBJ: Global variable location associated with the objective function in optimization.
- 3 SGNOPT: Sign used to identify whether function is to be maximized or minimized.
 +1.0 indicates maximization. -1.0 indicates minimization. If SGNOPT is not unity in magnitude, it acts as a multiplier as well, to scale the magnitude of the objective.

DATA BLOCK F OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Design variable bounds, initial values and scaling factors.

FORMAT AND EXAMPLE

1	2	3	4	FORMAT
VLB	VUB	X	SCAL	4F10

NOTE: READ ONE CARD FOR EACH OF THE NDV INDEPENDENT DESIGN VARIABLES.

FIELD

CONTENTS

- 1 VLB: Lower bound on the design variable. If VLB.LT.-1.0E+15, no lower bound.
- 2 VUB: Upper bound on the design variable. If VUB.GT.10.E+15, no upper bound.
- 3 X: Initial value of the design variable. If X is non-zero, this will supercede the value initialized by the user-supplied subroutine ANALIZ.
- 4 SCAL: Design variable scale factor. Not used if NSCAL..GE.0 in BLOCK C.

DATA BLOCK H OMIT IF NDV = 0 IN BLOCK B

DESCRIPTION: Number of constrained parameters.

FORMAT AND EXAMPLE

1		FORMAT
NCONS		I10

FIELD

CONTENTS

1 NCONS: Number of constraint sets in the optimization problem.

REMARKS

1) If two or more adjacent parameters in the global common block have the same limits imposed, these are part of the same constraint set.

DATA BLOCK 1 OMIT IF NDV = 0 IN BLOCK B, OR NCONS = 0 IN BLOCK H

DESCRIPTION: Constraint Identification and constraint bounds.

FORMAT AND EXAMPLE

1	2	3	4	FORMAT
ICON	JCON	ICON		3110
BL	SCAL1	BU	SCAL2	

NOTE: READ TWO CARDS FOR EACH OF THE NCONS CONSTRAINT SETS.
EQUALITY CONSTRAINTS MUST FOLLOW ALL INEQUALITY CONSTRAINTS.

<u>FIELD</u>	<u>CONTENTS</u>
1	ICON: First global number corresponding to the constraint set.
2	JCON: Last global number corresponding to the constraint set. DEFAULT = ICON.
3	LCON: Linear constraint identifier for this constraint set. LCON = 1 indicates linear constraints, LCON = 2 indicates equality constraint.

FIELD

CONTENTS

- | | | |
|---|--------|--|
| 1 | BL: | Lower bound on the constrained variables. If BL.LT.-1.0E+15, no lower bound. |
| 2 | SCAL1: | Normalization factor on lower bound. DEFAULT = MAX of ABS(BL), 0.1. |
| 3 | BU: | Upper bound on the constrained variables. If BU.GT.1.0E+15, no upper bound. |
| 4 | SCAL2: | Normalization factor on upper bound. DEFAULT = MAX of ABS(BU), 0.1. |

REMARKS

- 1) The normalization factor should usually be defaulted.
- 2) The constraint functions sent to CONMIN are of the form;
(BL - VALUE)/SCAL1 .LE. 0.0 and (VALUE - BU)/SCAL2 .LE. 0.0.
- 3) Each constrained parameter is converted to two constraints in CONMIN unless ABS(BL) or ABS(BU) exceeds 1.0E+15, in which case no constraint is created for that bound.

✓

DESCRIPTION: COPE data 'END' card.

FORMAT AND EXAMPLE

1		FORMAT
END		3A1
END		

FIELD

CONTENTS

1 The word 'END' in columns 1-3.

REMARKS

- 1) This card MUST appear at the end of the COPES data.
- 2) This ends the COPES input data.
- 3) Data for the user-supplied routine, ANALIZ, follows this.

ANALIZ DATA

DESCRIPTION: Tubeside Parameters

FORMAT

	14	28	42	56
$\dot{m}_h$		T_{h1}	T_{h2}	C_{ph}
				4F14.4

FIELD

- $\dot{m}_h$ - fluid mass flow rate, lbm/hr
- T_{h1} - inlet fluid temperature, °F
- T_{h2} - outlet fluid temperature, °F
- C_p - specific heat, BTU/lbm-°F

REMARKS

- 1) This begins ANALIZ data input.
- 2) Input must be formatted.
- 3) Comment cards are no longer permitted.

DESCRIPTION: Airside Parameters

FORMAT

	14	28	42	56	70	
$\dot{m}_a$		T_{c1}	T_{c2}	C_{pa}	P_∞	5F14,4

FIELD

- $\dot{m}_a$ - air mass flow rate, lbm/hr
- T_{c1} - air inlet temperature, °F
- T_{c2} - outlet air temperature, °F
- C_{pa} - specific heat, BTU/lbm-°F
- P_∞ - ambient pressure, psi

DESCRIPTION: Tube Geometry

FORMAT

	14	28	42	56	70	
D_i		D_o	ℓ	t	S	5F14.4

FIELD

- D_i - inside tube diameter, inches
- D_o - outside root tube diameter, inches
- ℓ - fin height, inches
- t - fin thickness, inches
- S - fin spacing, center-to-center, inches

DESCRIPTION: Tube Arrangement

FORMAT

	14	28	42	56	
P_t					4F14.4

FIELD

- P_t - transverse pitch, inches
- P_L - longitudinal pitch, inches
- h - bank height, inches
- w - bank width, inches

FORMAT

	14	28	42	56	
ITYPE		JTYPE	NROWS	NPASS	
					4110

ITYPE ~ configuration

- 1 - 1 row, 1 pass
2 - 2 rows, 1 pass
3 - 3 rows, 1 pass
4 - 4 rows, 1 pass
5 - 2 rows, 2 passes
6 - 3 rows, 3 passes
7 - 4 rows, 2 passes

FIELD

- 8 - 4 rows, 4 passes
- 9 - pure crossflow
- 10 - pure counterflow

JTYPE - fin profile

- 1 - rectangular

NROWS - number of rows

NPASS - number of passes

DESCRIPTION: Miscellaneous Parameters

FORMAT

14	28	42	56
$\dot{Q}$	k_{TUBE}	k_{FIN}	R
			4F14.4

FIELD

- $\dot{Q}$ - given heat transfer rate, BTU/hr
- k_{TUBE} - thermal conductivity of tube material, BTU/ft-hr-°F
- k_{FIN} - thermal conductivity of fin material, BTU/ft-hr-°F
- R - gas constant of air; 53.34 ft-lbf/lbm-°R

DESCRIPTION: Penalty Parameters

FORMAT

	14	28	42	
R1	R2	R3		3F14.4

NOTE

Set $R1 = R2 = R3 = 0$ if $NCALC = 3$.

DESCRIPTION: Optimization Parameters

FORMAT

	14	28	42	56	70	
Y1		Y2	Y3	Y4	Y5	5F14.4

FIELD

- Y1 = 1 if minimizing volume, otherwise Y1 = 0
Y2 = 1 if minimizing area, otherwise Y1 = 0
Y3 = 1 if minimizing air horsepower, otherwise Y1 = 0
Y4 = 1 if minimizing air pressure drop, otherwise Y1 = 0
Y5 = 1 if minimizing tubside pressure drop, otherwise Y1 = 0

NOTE

If NCALC = 3 let Y1 = Y2 = Y3 = Y4 = Y5 = 0

DESCRIPTION: Augmented Lagrangian Function Method Data

FORMAT

	10	20	30	
CC		CMULT	CCMAX	3F10.0

FIELD

- CC - initial Lagrangian multiplier
- CMULT - Lagrangian multiplication factor
- CCMAX - maximum value of Lagrangian multiplier

APPENDIX B

SAMPLE USER'S INPUT

COPIES DATA

DATA BLOCK A

TITLE	FORMAT
AIR-COOLED HEAT EXCHANGER DESIGN - CASE I	20A4

DATA BLOCK B

NCALC	NDV	COMMENT
3	9	7110

DATA BLOCK C - OMIT IF NDV = 0

IPRINT	ITMAX	ICNDR	NSCAL	ITRM	LINOBJ	NACMX1	NFDG	FORMAT
0	0	0	10	0	0	0	0	8110

DATA BLOCK D - OMIT IF NDV = 0

							COMMENT	
FDCH	FDCRM	CT	CTMIN	CTL	CTLMIN	THETA	FORMAT	
0	0	0	0	0	0	0	7F10	
DEL.FUN	DABFUN	ALPIAX	ABOJJ1					FORMAT
0	0	0	0					4F10

DATA BLOCK E - OMIT IF NDV = 0

				COMMENT	
NDVTOF	IOBJ	SGNOBJ			FORMAT
9	16	-1.0			2I10,F10

DATA BLOCK F - OMIT IF NDV = 0

						COMMENT
VI.B	VUB	X	SCAL			FORMAT
.232	2.325	2.0	0.0			4F10
.25	2.5	2.5	0.0			

DATA BLOCK F - CONT.

.0625	1.0 + 20	.460	0.0		
.01	.0235	.023	0.0		
.08	.125	.111	0.0		
0.0	4.0	2.125	0.0		
0.0	4.0	4.0	0.0		
0.0	500.	490.	0.0		
0.0	500.	350.	0.0		

DATA BLOCK C - OMIT IF NDV = 0

NDSCN	IDSCN	AMULT	COMMENT
1	7	0.0	2110,F10
2	8	0.0	
3	9	0.0	
4	10	0.0	
5	11	0.0	
6	12	0.0	
7	13	0.0	
8	14	0.0	
9	15	0.0	

DATA BLOCK H - OMIT IF NDV = 0

\$			COMMENT
NCONS			FORMAT
11			110

DATA BLOCK I - OMIT IF NDV = 0 OR NCONS = 0

\$			COMMENT
ICON			FORMAT
24	24	0	3110
\$			COMMENT
EL			FORMAT
1.0	0.0	2.5	4F10
\$			COMMENT
ICON			FORMAT
25	25	0	

DATA BLOCK I - CONT.

\$						
BL	SCAL1	BU	SCAL2			
.018	0.0	.18	0.0			
\$						
ICON	JCON	LCON				
26	27	0				
\$						
BL	SCAL1	BU	SCAL2			
-1.0 + 20	0.0	0.0	0.0			
\$						
ICON	JCON	LCON				
19	19	0				
\$						
BL	SCAL1	BU	SCAL2			
-1.0 + 20	0.0	.0722	0.0			

DATA BLOCK I - CONT.

20	20	0			
\$					
EL	SCAL1	BU	SCAL2		
-1.0+20	0.0	.14	0.0		
\$					
ICON	JCON	LCON			
30	30	0			
\$					
EL	SCAL1	BU	SCAL2		
.7	0.0	1.0+20	0.0		
\$					
ICON	JCON	LCON			
32	32	0			
\$					
EL	SCAL1	BU	SCAL2		
-1.0+20	0.0	0.0	0.0		

DATA BLOCK I - CONT.

34	35	0			
\$					
BL	SCAL1	BU	SCAL2		
-1.0	0.0	1.0	0.0		
\$					
ICOH	JCON	LCON			
36	36	0			
\$					
BL	SCAL1	BU	SCAL2		
2.0	0.0	1.0 + 7	0.0		
\$					
ICOH	JCOH	LCON			
37	37	0			
\$					
EL	SCAL1	BU	SCAL2		
-1.0 + 20	0.0	0.0	0.0		

DATA BLOCK I - CONT.

33	33	2			
\$					
BL	SCAL1	BU	SCAL2		
1.0	0.1	1.0 + 20	0.0		
\$					
ICON	JCON	LCON			
\$					
BL	SCAL1	BU	SCAL2		
\$					
ICON	JCON	LCON			
\$					
BL	SCAL1	BU	SCAL2		
\$					
BL	SCAL1	BU	SCAL2		
\$					

DATA BLOCK V - COPIES END OF DATA CARD

END		FORMAT
END		3A1

ANALIZ DATA

$\dot{m}_w$	T_{h1}	T_{h2}	C_p						FORMAT
133333.33	200.	125.	1.						4F14.4
$\dot{m}_a$	T_{c1}	T_{c2}	C_p	P_∞					FORMAT
1190476.2	95.	130.	.24	14.0					5F14.4
D_i	D_o	ℓ	t	s					FORMAT
2.0	2.5	.46	.023	.111					5F14.4
P_t	P_L	h	w						FORMAT
4.0	2.125	350.	490.						4F14.4
ITYPE	JTYPE	NROWS	NPASS						FORMAT
7	1	4	2						4I10
$\dot{Q}$	k_{TUBE}	k_{FIN}	R						FORMAT
10000000.	220.	118.	53.34						4F14.4

ANALIZ DATA - CONT.

R1	R2	R3							FORMAT
0.0	0.0	0.0							3F14.4
Y1	Y2	Y3	Y4	Y5					FORMAT
0.0	0.0	0.0	0.0	0.0					5F14.4
CC	CMULT	CCMAX							FORMAT
10.0	2.0	1000.							3F10.0
									FORMAT

APPENDIX C
SAMPLE OUTPUT FROM COTES

AAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAAA

09/16/80 13.54.51
FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL
1

CCCCC	000000	PPPPPP	EEEEEE	SSSSSS
C	0	P	E	S
C	0	P	E	S
C	0	P	E	S
C	0	P	E	S
C	0	P	E	S
CCCCC	000000	P	EEEEEE	SSSSSS

CONTROL PROGRAM
FOR
ENGINEERING SYNTHESIS

TITLE

1 AIR HEATER DESIGN - OPTIMIZE VOLUME.
CARD IMAGES OF CONTROL DATA

CARD	IMAGE
0	
1)	AIR HEATER DESIGN - OPTIMIZE VOLUME.
2)	2.9
3)	\$ BLOCK C
4)	0.80, 0.10
5)	\$ BLOCK D
6)	0.
7)	0.
8)	\$ BLOCK E
9)	9.16, -1.
10)	\$ BLOCK F
11)	.232, 2.325, .6786
12)	.25, 2.5, .7201

FILE: SIV

DATA

T1

NAVAL POSTGRADUATE SCHOOL

131	.0625,1.0+20,.16		
131	.0625,1.0+20,.16		.16
141	.01,.0217,.0235		
141	.01,.0217,.0235	.0217	.0235
151	.08,.125,.08		
151	.08,.125,.08	.125	.08
161	0.,4.,1.04		
161	0.,4.,1.04	4.	1.04
171	0.,4.,1.04		
171	0.,4.,1.04	4.	1.04
181	0.,500.,285.		
181	0.,500.,285.	500.	285.
191	0.,500.,342.		
191	0.,500.,342.	500.	342.
201	\$ BLOCK G		
211	1,7		
211	1,7	1	7
221	2,8		
221	2,8	2	8
231	3,9		
231	3,9	3	9
241	4,10		
241	4,10	4	10
251	5,11		
251	5,11	5	11
261	6,12		
261	6,12	6	12
271	7,13		
271	7,13	7	13
281	8,14		
281	8,14	8	14
291	9,15		
291	9,15	9	15
301	\$ BLOCK H		
311	11		
311	11	11	
321	24		
321	24	24	
331	1.,0.,2.5		
331	1.,0.,2.5	0.	2.5
341	25		
341	25	25	
351	.018,0.,.18		
351	.018,0.,.18	0.	.18
361	26,27		
361	26,27	26	27
371	-1.0+20,0.,0.		
371	-1.0+20,0.,0.	0.	0.
381	19		
381	19	19	
391	-1.0+20,0.,.0722		
391	-1.0+20,0.,.0722	0.	.0722
401	20		
401	20	20	
411	-1.0+20,0.,.14		

FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL

```

41) -1.0+20 0. .14
42) 30 30
43) .7,0.,1.0+20
44) .7 0. 1.0+20
45) 38,38 38
46) 0.0,0.0,1.3,0. 0.0 1.3 0.
47) 3+,33 34 35
48) -1.0,0.,1. -1.0 0. 1.
49) 36 36
50) 2.,0.,1.0+7 2. 0. 1.0+7
51) 37 37
52) -1.0+20,0.,0. -1.0+20 0. 0.
53) 33,33,2 33 2
54) 1.,1,1.0+20 1. .1 1.0+20
55) END

```

1 TITLE: AIR HEATER DESIGN - OPTIMIZE VOLUME.

CONTROL PARAMETERS:
 CALCULATION CONTROL, NCALC = 2
 NUMBER OF GLOBAL DESIGN VARIABLES, NDV = 9
 INPUT INFORMATION PRINT CODE, IPNPUT = 0
 DEBUG PRINT CODE, IPDBG = 0

CALCULATION CONTROL, NCALC
 VALUE MEANING
 1 SINGLE ANALYSIS
 2 OPTIMIZATION
 3 SENSITIVITY
 4 TWO-VARIABLE FUNCTION SPACE
 5 OPTIMUM SENSITIVITY
 6 APPROXIMATE OPTIMIZATION

* * OPTIMIZATION INFORMATION

GLOBAL VARIABLE NUMBER OF OBJECTIVE = 16
 MULTIPLIER (NEGATIVE INDICATES MINIMIZATION) = -0.100CE 01

FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL

CONMIN PARAMETERS (IF ZERO, CONMIN DEFAULT WILL OVER-RIDE)

IPRINT	ITMAX	ICNDR	NSCAL	ITRM	LINDBJ	VACMAX	NFDG
0	80	0	10	0	0	11	3
FOCH		FOCHM		CT		CTMIN	
0.0		0.0		0.0		0.0	
CTL		CTLMIN		THETA		PHI	
0.0		0.0		0.0		0.0	
DELFLN		DABFUN		ALPHAX		ABOBJ1	
0.0		0.0		0.0		0.0	

DESIGN VARIABLE INFORMATION

NON-ZERO INITIAL VALUE WILL OVER-RIDE MODULE INPUT

D. V. NO.	LOWER BOUND	UPPER BOUND	INITIAL VALUE	SCALE
1	0.23200E 00	0.23250E 01	0.67660E 00	0.0
2	0.25000E 00	0.25000E 01	0.72010E 00	0.0
3	0.62500E -01	0.11000E 16	0.16000E 00	0.0
4	0.10000E -01	0.21700E -01	0.23500E -01	0.0
5	0.30000E -01	0.12500E 00	0.80000E -01	0.0
6	0.0	0.40000E 01	0.10400E 01	0.0
7	0.0	0.40000E 01	0.10400E 01	0.0
8	0.0	0.50000E 03	0.28500E 03	0.0
9	0.0	0.50000E 03	0.34200E 03	0.0

DESIGN VARIABLES

ID	D. V. NO.	GLOBAL VAR. NO.	MULTIPLYING FACTOR
1	1	7	0.10000E 01
2	2	8	0.10000E 01
3	3	9	0.10000E 01
4	4	10	0.10000E 01
5	5	11	0.10000E 01
6	6	12	0.10000E 01
7	7	13	0.10000E 01
8	8	14	0.10000E 01
9	9	15	0.10000E 01

CONSTRAINT INFORMATION

THERE ARE 11 CONSTRAINT SETS

ID	GLOBAL VAR. 1	GLOBAL VAR. 2	LINEAR	LOWER BOUND	NORMALIZATION FACTOR	UPPER BOUND
1	24	0	0	0.10000E 01	0.10000E 01	0.25000E 01
2	25	0	0	0.13000E -01	0.13000E -01	0.18000E 00
3	26	27	0	-0.11000E 16	0.11000E 16	0.0
4	19	0	0	-0.11000E 16	0.11000E 16	0.72200E -01
5	20	0	0	-0.11000E 16	0.11000E 16	0.14000E 00
6	30	0	0	0.70000E 00	0.70000E 00	0.11000E 16
7	38	38	0	0.0	0.10000E 00	0.13000E 01

FILE: SAV DATA T1 NAVAL POSTGRADUATE SCHOOL

12	34	35	0	-0.10000E 01	0.10000E 01	0.10000E 01
16	36	0	0	0.20000E 01	0.20000E 01	0.10000E 01
18	37	0	0	-0.11000E 16	0.11000E 16	0.0
19	33	33	2	0.10000E 01	0.10000E 00	0.11000E 16

TOTAL NUMBER OF CONSTRAINED PARAMETERS = 13

* * ESTIMATED DATA STORAGE REQUIREMENTS

	REAL	AVAILABLE	INTEGER	AVAILABLE
INPUT EXECUTION	106	5000	76	1000
1 AIR-COOLED HEAT EXCHANGER OPTIMIZATION				

INPUT DATA

TUBESIDE PARAMETERS

MASS FLOW RATE= 133333.3125 LBM/HR
 INLET TEMPERATURE= 200.0000 DEG F
 OUTLET TEMPERATURE= 125.0000 DEG F
 SPECIFIC HEAT= 1.0000 BTU/LB-F

AIRSIDE PARAMETERS

MASS FLOW RATE= 1190476.0000 LBM/HR
 INLET TEMPERATURE= 95.0000 DEG F
 OUTLET TEMPERATURE= 130.0000 DEG F
 SPECIFIC HEAT= 0.2400 BTU/LBM-F
 INLET PRESSURE= 14.0000 PSI

TUBE GEOMETRY

TUBE INSIDE DIA.= 0.8700 INCHES
 TUBE OUTSIDE DIA.= 1.3800 INCHES
 FIN HEIGHT= 0.4600 INCHES
 FIN THICKNESS= 0.0100 INCHES
 FIN SPACING, CENTER-TO-CENTER= 0.1110 INCHES

TUBE ARRANGEMENT

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NAVAL POSTGRADUATE SCHOOL

TRANSVERSE PITCH= 2.1250 INCHES
LONGITUDINAL PITCH= 2.1250 INCHES
BANK HEIGHT= 96.0000 INCHES
BANK WIDTH= 248.0000 INCHES

INTEGER PARAMETERS

TYPE OF CROSS FLOW ARRANGEMENT= 7
TYPE OF FIN PROFILE= 1
NUMBER OF ROWS= 4
NUMBER OF PASSES= 2

MISCELLANEOUS VARIABLES

GIVEN HEAT TRANSFER RATE= 1000000.0000 BTU/HR
THERMAL CONDUCTIVITY OF TUBE MATERIAL= 220.0000 BTU/HR-FT-F
THERMAL CONDUCTIVITY OF FIN MATERIAL= 118.0000 BTU/HR-FT-F
GAS CONSTANT= 53.3400 FT-LBF/LBM-R

PENALTY PARAMETERS

12.00000000 0.0 0.0

OPTIMIZATION PARAMETERS

0.0 0.0 0.0 0.0

* CC = 0.10000E 02 CMULT = 0.20000E 01 CCMAX = 0.10000E 04
KOUNT= 1

* * COMIN DETECTS INITIAL X(I).GT.VUB(I)
X(I) = 0.2350E-01 VUB(I) = 0.2170E-01
X(I) IS SET EQUAL TO VUB(I) FOR I = 4

KCUNT= 1 DEL= 0.77484E 01 OBJ= 0.75310E 02 JHJ1 = 0.72309E 02
-0.43510E 00-0.42590E 00-0.20534E-05-0.90000E 00-0.23558E 01
-0.84404E-01-0.55220E-01-0.69772E-01-0.41434E 00-0.67939E 01
-0.47701E 00-0.10287E 01-0.37127E 00-0.14952E 01-0.50473E 00
-0.10044E 03-0.97999E 00-0.16773E 06 0.77484E 00
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.77484E 01 0.0

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NAVAL POSTGRADUATE SCHOOL

KOUNT= 2

KOUNT= 2 DEL= 0.22126E 01 OBJ= 0.71452E 02 OBJ1 = 0.72922E 02
-0.45403E 00-0.41815E 00-0.13239E-01-0.39958E 00-0.18120E 01
-0.17881E-03-0.37723E-02-0.42521E-01-0.41393E 00-0.65290E 01
-0.49777E 00-0.15175E 01-0.39251E 00-0.15000E 01-0.50000E 00
-0.11359E 03-0.97493E 00-0.16405E 06-0.22125E 00
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.55358E 01 0.0

KOUNT= 3

KOUNT= 3 DEL= 0.70391E 00 OBJ= 0.71978E 02 OBJ1 = 0.72243E 02
-0.45477E 00-0.41815E 00-0.33662E-01-0.39651E 00-0.13523E 01
-0.26009E-01-0.15978E-01-0.55173E-02-0.41411E 00-0.65470E 01
-0.47617E 00-0.16041E 01-0.39087E 00-0.14485E 01-0.50155E 00
-0.11315E 03-0.97498E 00-0.16498E 06-0.70391E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.48320E 01 0.0

KOUNT= 4

KOUNT= 4 DEL= 0.10330E 01 OBJ= 0.71241E 02 OBJ1 = 0.71464E 02
-0.45445E 00-0.41820E 00-0.13034E-01-0.89814E 00-0.13535E 01
-0.87023E-04-0.32180E-02-0.34043E-02-0.41392E 00-0.65733E 01
-0.49436E 00-0.16110E 01-0.33399E 00-0.15000E 01-0.50001E 00
-0.11309E 03-0.97498E 00-0.16376E 06-0.51651E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.37990E 01 0.0

KOUNT= 5

KOUNT= 5 DEL= 0.10048E 01 OBJ= 0.71294E 02 OBJ1 = 0.71459E 02
-0.45451E 00-0.41820E 00-0.16434E-01-0.89835E 00-0.13538E 01
-0.56684E-01-0.33632E-02-0.34971E-02-0.41392E 00-0.65733E 01
-0.49436E 00-0.16110E 01-0.38399E 00-0.15000E 01-0.50003E 00
-0.11309E 03-0.97498E 00-0.16397E 06-0.50240E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.27942E 01 0.0

KOUNT= 6

KOUNT= 6 DEL= 0.94128E 00 OBJ= 0.71338E 02 OBJ1 = 0.71448E 02
-0.45455E 00-0.41813E 00-0.12723E-01-0.39873E 00-0.13599E 01
-0.17017E-02-0.37034E-02-0.35041E-02-0.41393E 00-0.65733E 01
-0.49435E 00-0.16110E 01-0.36900E 00-0.14999E 01-0.50010E 00
-0.11309E 03-0.97498E 00-0.15379E 06-0.47064E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.18529E 01 0.0

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NAVAL POSTGRADUATE SCHOOL

KOUNT= 7

KOUNT= 7 DEL= 0.18319E 01 OBJ= 0.71402E 02 OBJ1 = 0.71445E 02
-0.45456E 00-0.41318E 00-0.11355E-01-0.49386E 00-0.18603E 01
-0.21237E-02-0.33754E-02-0.39126E-02-0.41393E 00-0.55733E 01
-0.49436E 00-0.18110E 01-0.38700E 00-0.14999E 01-0.50015E 00
-0.11369E 03-0.99999E 00-0.16400E 06-0.45795E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 8 0.21066E-01

KOUNT= 8 DEL= 0.17517E 01 OBJ= 0.71483E 02 OBJ1 = 0.71445E 02
-0.45452E 00-0.41819E 00-0.11310E-01-0.49387E 00-0.18606E 01
-0.24354E-02-0.41425E-02-0.39128E-02-0.41394E 00-0.55733E 01
-0.49436E 00-0.18110E 01-0.38700E 00-0.14999E 01-0.50015E 00
-0.11310E 03-0.99999E 00-0.16400E 06-0.45795E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 9 -0.17306E 01

KOUNT= 9 DEL= 0.16983E 01 OBJ= 0.71555E 02 OBJ1 = 0.71446E 02
-0.45449E 00-0.41820E 00-0.11286E-01-0.49387E 00-0.18608E 01
-0.26459E-02-0.41444E-02-0.39128E-02-0.41394E 00-0.55733E 01
-0.49436E 00-0.18110E 01-0.38700E 00-0.14999E 01-0.50016E 00
-0.11310E 03-0.99999E 00-0.16400E 06-0.45795E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 10 -0.34209E 01

KOUNT= 10 DEL= 0.25459E 01 OBJ= 0.71530E 02 OBJ1 = 0.71599E 02
-0.45272E 00-0.41391E 00-0.13524E-01-0.49385E 00-0.18589E 01
-0.14077E-01-0.18301E-01-0.55764E-02-0.41411E 00-0.55702E 01
-0.49460E 00-0.18108E 01-0.38724E 00-0.14992E 01-0.50084E 00
-0.11344E 03-0.99999E 00-0.16522E 06-0.31824E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 11 -0.88302E 00

KOUNT= 11 DEL= 0.25754E 01 OBJ= 0.71609E 02 OBJ1 = 0.71596E 02
-0.45271E 00-0.41391E 00-0.13105E-01-0.49385E 00-0.18589E 01
-0.14143E-01-0.18274E-01-0.55797E-02-0.41411E 00-0.55701E 01
-0.49451E 00-0.18108E 01-0.38724E 00-0.14992E 01-0.50084E 00
-0.11344E 03-0.99999E 00-0.16522E 06-0.32192E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 12 0.16924E 01

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NAVAL POSTGRADUATE SCHOOL

KOUNT= 12

KOUNT= 12 DEL= 0.25936E 01 OBJ= 0.71691E 02 OBJ1 = 0.71594E 02
-0.45271E 00-0.41492E 00-0.12187E-01-0.39971E 00-0.13638E 01
-0.14184E-01-0.15257E-01-0.35943E-02-0.41411E 00-0.65700E 01
-0.49461E 00-0.15107E 01-0.33925E 00-0.14992E 01-0.50085E 00
-0.11345E 00-0.99993E 00-0.16522E 06 0.32420E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 13

KOUNT= 13 DEL= 0.17141E 01 OBJ= 0.71739E 02 OBJ1 = 0.71776E 02
-0.45178E 00-0.41929E 00-0.20004E-01-0.39774E 00-0.13638E 01
-0.17739E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
-0.49484E 00-0.15105E 01-0.33949E 00-0.14989E 01-0.50106E 00
-0.11372E 00-0.99998E 00-0.16497E 06-0.10738E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 14

KOUNT= 14 DEL= 0.16953E 01 OBJ= 0.71757E 02 OBJ1 = 0.71775E 02
-0.45178E 00-0.41929E 00-0.20004E-01-0.39774E 00-0.13638E 01
-0.17739E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
-0.49484E 00-0.15105E 01-0.33949E 00-0.14989E 01-0.50106E 00
-0.11372E 00-0.99998E 00-0.16497E 06-0.10595E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 15

KOUNT= 15 DEL= 0.16749E 01 OBJ= 0.71774E 02 OBJ1 = 0.71774E 02
-0.45178E 00-0.41929E 00-0.20017E-01-0.39774E 00-0.13638E 01
-0.17835E-01-0.13504E-01-0.11582E-01-0.41407E 00-0.65671E 01
-0.49484E 00-0.15105E 01-0.33949E 00-0.14989E 01-0.50107E 00
-0.11372E 00-0.99998E 00-0.16497E 06-0.10452E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 16

KOUNT= 16 DEL= 0.33295E 01 OBJ= 0.71799E 02 OBJ1 = 0.71773E 02
-0.45173E 00-0.41929E 00-0.20226E-01-0.39774E 00-0.13638E 01
-0.17863E-01-0.13493E-01-0.11582E-01-0.41407E 00-0.65671E 01
-0.49434E 00-0.15105E 01-0.33949E 00-0.14989E 01-0.50107E 00
-0.11371E 00-0.99998E 00-0.16497E 06-0.10405E-01
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
0.0 0.0 0.0 0.0 0.0
KOUNT= 17

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KOUNT= 17

KOUNT= 17 DEL= 0.21240E 01 OBJ= 0.71803E 02 OBJ1 = 0.71824E 02
 -0.45134E 00-0.41947E 00-0.22213E-01-0.89778E 00-0.13692E 01
 -0.19782E-01-0.16613E-01-0.11521E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.59993E 00-0.16529E 06 0.66376E-02
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 -0.20037E 01 0.0

KOUNT= 18

KOUNT= 18 DEL= 0.21358E 01 OBJ= 0.71817E 02 OBJ1 = 0.71823E 02
 -0.45134E 00-0.41947E 00-0.22213E-01-0.89778E 00-0.13692E 01
 -0.19801E-01-0.16613E-01-0.11513E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.59993E 00-0.16529E 06 0.66745E-02
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.12912E 00 0.0

KOUNT= 19

KOUNT= 19 DEL= 0.42976E 01 OBJ= 0.71837E 02 OBJ1 = 0.71822E 02
 -0.45134E 00-0.41946E 00-0.22079E-01-0.89778E 00-0.13692E 01
 -0.19810E-01-0.16602E-01-0.11513E-01-0.41412E 00-0.55560E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.59993E 00-0.16529E 06 0.67151E-02
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.44268E 01 0.0

KOUNT= 20

KOUNT= 20 DEL= 0.43011E 01 OBJ= 0.71856E 02 OBJ1 = 0.71821E 02
 -0.45134E 00-0.41946E 00-0.22038E-01-0.89778E 00-0.13692E 01
 -0.19803E-01-0.16579E-01-0.11513E-01-0.41412E 00-0.55561E 01
 -0.49492E 00-0.16104E 01-0.38957E 00-0.14988E 01-0.50118E 00
 -0.11381E 03-0.59993E 00-0.16529E 06 0.67204E-02
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.0 0.0
 0.0 0.0 0.0 0.87273E 01 0.0

1 KOUNT= 20 DEL= 0.43011E 01 OBJ= 0.71894E 02 OBJ1 =
 OPTIMIZATION RESULTS

OBJECTIVE FUNCTION
 GLOBAL LOCATION 16 FUNCTION VALUE 0.71821E 02

DESIGN VARIABLES

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DATA

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NAVAL POSTGRADUATE SCHOOL

ID	D. V. NO.	GLOBAL VAR. NO.	LOWER BOUND	VALUE	UPPER BOUND
1	1	7	0.2320E 00	0.5383E 00	0.2525E 01
2	2	8	0.2500E 00	0.5755E 00	0.2530E 01
3	3	9	0.8250E -01	0.1298E 00	0.1100E 16
4	4	10	0.9999E -02	0.2377E -01	0.2170E -01
5	5	11	0.7999E -01	0.7999E -01	0.1250E 00
6	6	12	0.0	0.8373E 00	0.3999E 01
7	7	13	0.0	0.1022E 01	0.4000E 01
8	8	14	0.0	0.1860E 03	0.4999E 03
9	9	15	0.0	0.2360E 03	0.4999E 03

DESIGN CONSTRAINTS

ID	GLOBAL VAR. NO.	LOWER BOUND	VALUE	UPPER BOUND
1	24	0.1000E 01	0.1451E 01	0.2570E 01
3	25	0.1800E -01	0.1839E -01	0.1830E 00
5	26	-0.1100E 16	-0.1867E 00	0.0
6	27	-0.1100E 16	-0.1980E -02	0.0
7	19	-0.1100E 16	0.7103E -01	0.7220E -01
8	20	-0.1100E 16	0.1333E 00	0.1400E 00
9	30	0.7000E 00	0.9898E 00	0.1100E 16
10	38	0.0	0.6566E 00	0.1307E 01
12	34	-0.1000E 01	0.6104E 00	0.1000E 01
14	35	-0.1000E 01	0.4358E 00	0.1000E 01
16	36	-0.2000E 01	0.2296E 03	0.1000E 03
18	37	-0.1100E 16	-0.1652E 05	0.0
19	33	0.1000E 01	0.9993E 00	0.1100E 16

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OUTPUT DATA

TUBESIDE PARAMETERS

MASS FLOW RATE= 13333.3125 LBM/HR
 INLET TEMPERATURE= 200.0000 DEG F
 OUTLET TEMPERATURE= 120.0000 DEG F
 SPECIFIC HEAT= 1.0000 BTU/LBM-F

AIRSIDE PARAMETERS

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DATA

TL

NAVAL POSTGRADUATE SCHOOL

MASS FLOW RATE= 1190476.0000 LB/HR
INLET TEMPERATURE= 95.0000 DEG F
OUTLET TEMPERATURE= 130.0000 DEG F
SPECIFIC HEAT= 0.2400 BTU/LB-F
INLET PRESSURE= 14.0000 PSI

TUBE GEOMETRY

TUBE INSIDE DIA.= 0.5388 INCHES
TUBE OUTSIDE DIA.= 0.5756 INCHES
FIN HEIGHT= 0.1299 INCHES
FIN THICKNESS= 0.0203 INCHES
FIN SPACING, CENTER-TO-CENTER= 0.0800 INCHES

TUBE ARRANGEMENT

TRANSVERSE PITCH= 1.0223 INCHES
LONGITUDINAL PITCH= 0.8374 INCHES
BANK HEIGHT= 236.0435 INCHES
BANK WIDTH= 186.0704 INCHES

THETA= 0.6556
DELPHI= -0.3920

INTEGER PARAMETERS

TYPE OF CROSS FLOW ARRANGEMENT= 7
TYPE OF FIN PROFILE= 1
NUMBER OF ROWS= 4
NUMBER OF PASSES= 2

VERTICAL ROWS CONTAIN 229 TUBES

MISCELLANEOUS VARIABLES

GIVEN HEAT TRANSFER RATE= 1000000.0000 BTU/HR
THERMAL CONDUCTIVITY OF TUBE MATERIAL= 220.0000 BTU/HR-FT-F
THERMAL CONDUCTIVITY OF FIN MATERIAL= 118.0000 BTU/HR-FT-F
GAS CONSTANT= 53.3490 FT-LB/7.34-R

PENALTY PARAMETERS

12.000000000 0.0 0.0

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DATA

T1

NAVAL POSTGRADUATE SCHOOL

OPTIMIZATION PARAMETERS

0.0

0.0

0.0

0.0

OTHER VALUES

HEAT TRANSFER RATES
QDOT1= 9953780.0000 BTU/HR
QDOT2= 999999.0000
QDOT3= 9959994.0000

HEAT TRANSFER DIFFERENCES
DELQ1= 6720.0000 BTU/HR
DELQ2= -6718.0000
DELQ3= 4.0000

PENALTY= 11.9919

CONSTRAINTS
DIAMETER RATIO= 1.4513
TUBE THICKNESS= 0.0184 INCHES
TOUCHN= -0.1869
TOUCHL= -0.0020
PROFH= -70.0000 F
PROFC= -30.0000 F

VALUES TO BE OPTIMIZED
VOLUME= 71.8214 CU.FT.
AREA= 10105.4219 SQ.FT.
AIR HP= 92.7296 HP
AIR PRESSURE DROP= 0.0710 PSI
TUBESIDE PRESSURE DROP= 0.1384 PSI

1 OBJECTIVE FUNCTION= 71.8214
PROGRAM CALLS TO ANALYZ

ICALL CALLS
1 1
2 1150
3 1

HEDSUP PROGRAM LISTING

SUBROUTINE ANALIZ (ICALC)

COMMON /GLOBBCM/ FMDOT,FLIMP1,FLIMP2,AMDOT,ARTMP1,ARTMP2,TUBEID,TUB
LEOD,FINHT,FINTH,FINSP,PITCHL,PITCHN,BANKW,BANKH,VOLUME,AREA,FHP,PP
2A,PPW,DELQ1,DELQ2,DELQ3,DRATIO,TUBTH,TUCHN,TOUCHL,PROFH,PROFC,ARG
35,QDOT1,DELPHI,QRATIO,ARG7,ARG8,VROWR,DELSFF,QRATUI,QRATIO2,THEA

IF (ICALC.GT.1) GO TO 10

---INPUT INITIAL DATA - WHETHER KNOWN OR ESTIMATED

TUBESIDE FLUID PROPERTIES:
READ (5,210) FMDOT,FLTMP1,FLTMP2,CP

EMDOT - FLUID MASS FLOW RATE, LBM/HR

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FLTMP - INLET FLUID TEMPERATURE, DEG F
FLTMP2 - OUTLET FLUID TEMPERATURE, DEG F
CP - SPECIFIC HEAT (ASSUMED CONSTANT), BTJ/LBM-F
AIR READ (5.220) ANDOT,ARTMP1,ARTMP2,ACP,PRES1

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AMDOT - AIR MASS FLOW RATE, LBM/HR

ARTMP1 - AIR INLET TEMPERATURE, DEG. F
ARTMP2 - AIR OUTLET TEMPERATURE, DEG. F
ACPP - SPECIFIC HEAT (ASSUMED CONSTANT), BTU/LBM-F
PRES1 - INLET AIR PRESSURE, PSI
TUBE GEOMETRY
TUBE READ (5,230) TJBEID,TUBEQD,FINHT,FINTH,FINSP

GEOMETRY
TUBEID, TUBEOD, FINHT, FINTH, FINSP
READ (5,230)

TUBE ID - TUBE INSIDE DIAMETER, INCHES

TUBEOD - TUBE OUTSIDE DIAMETER, INCHES

FINHT - FIN HEIGHT, INCHES

FINSP - DISTANCE BETWEEN ADJACENT KNEES. INCHES

PIPE ARRANGEMENT

READ (5,240) PITCHN,PITCHL,BANKH,BANKW

PITCHN - DISTANCE BETWEEN CENTERS OF ADJACENT TUBES NORMAL TO FLOW (TRANSVERSE PITCH), INCHES


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C0000      CALCULATE THE CORRECTION FACTOR FOR THE AIRSIDE FILM COEFFICIENT
C0010      BASED ON THE NUMBER OF ROWS AS COMPARED TO SIX.
C0020      CALL ROWCOR (VMAXS,NROWS,C)
C0030
C0040      THE CORRECT AIRSIDE FILM COEFFICIENTS ARE THE CALCULATED:
C0050
C0060      CHO(1)=C*HO(1)
C0070      CHO(2)=C*HO(2)
C0080
C0090      .....CALCULATE AIRSIDE RESISTANCE .....
C0100
C0110      TO CALCULATE THE AIRSIDE RESISTANCE, THE SURFACE EFFICIENCY MUST BE
C0120      COMPUTED. FIRST THE FIN EFFICIENCY FOR BOTH CORRECTED REFERENCE TEM-
C0130      PERATURES IS COMPUTED.
C0140      CALL FINEFF (FINHT,TUBEOD,AK,FINTH,HQ,JTYPE,EATA,AM,RO,RE,BETA,ARG
C0150      11,ARG2,B2,B1)
C0160
C0170      NEXT THE FINNED HEAT TRANSFER AND THE TOTAL HEAT TRANSFER AREA ON THE
C0180      AIRSIDE MUST BE CALCULATED.
C0190
C0200      CALL HTAREA (FINPIN,TUBEOD,FINTH,AS,AT,ASAT,TOTAL,BANKW)
C0210
C0220      THE SURFACE EFFICIENCY, SUREFF, IS NOW CALCULATED FOR EACH CORRECTED
C0230      REFERENCE AIRSIDE FILM COEFFICIENT.
C0240
C0250      SUREFF(1)=1.-ASAT*(1.-EATA(1))
C0260      SUKEFF(2)=1.-ASAT*(1.-EATA(2))
C0270
C0280      THE AIRSIDE RESISTANCE IS THEREFORE,
C0290
C0300      RESAIR(1)=1./(SUKEFF(1)*CHO(1))
C0310      RESAIR(2)=1./(SUREFF(2)*CHO(2))
C0320
C0330      NOW USING ROETZEL'S ITERATION FREE CALCULATION, COMPUTE THE TUBESIDE
C0340      FILM COEFFICIENT CORRECTION:

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C .....CORRECT TUBESIDE FILM COEFFICIENT .....
C CALL CAY (HI,W,TUBEID,RESAIR,TUBEOD,CT,CTP,CA)
C
C NOW CORRECT THE TUBESIDE FILM COEFFICIENT,
C
C   CHI(1)=CA(1)*HI(1)
C   CHI(2)=CA(2)*HI(2)
C
C WITH THE CORRECTED FILM COEFFICIENTS CALCULATED AT EACH CORRECTED
C REFERENCE TEMPERATURE, THE TWO REFERENCE OVERALL HEAT TRANSFER
C COEFFICIENTS BASED ON THE OUTSIDE ROOT TUBE AREA CAN BE COMPUTED.
C
C   UK(1)=1./(((TUBEOD/TUBEID)*(1./CHI(1)))+(TUBEOD/24.)*W+RESAIR(1))
C   UK(2)=1./(((TUBEOD/TUBEID)*(1./CHI(2)))+(TUBEOD/24.)*W+RESAIR(2))
C
C ..... TRUE MEAN OVERALL HEAT TRANSFER COEFFICIENT .....
C THEREFORE, IN ACCORDANCE WITH ROETZEL'S FORMULATION, THE TRUE MEAN
C OVERALL HEAT TRANSFER COEFFICIENT, UKM, IS, IN BTU/HR-SQ.FT.-F :
C   UKM=(2.*UK(1)*UK(2))/(UK(1)+UK(2))
C
C NOW TO CALCULATE PRESSURE DROP FOR THE AIRSIDE, TAKING INTO ACCOUNT
C THE DENSITY CHANGE.
C   CALL DELTAP (CTP,TUBEOD,AMDOT,SFF,PITCHM,PITCHL,PRES1,GASCON,NROWS
C   1,UK,DELPA,CDELPA,SVOL,ARG5)
C
C NOW TO CALCULATE THE TUBESIDE PRESSURE DROP, TAKING INTO ACCOUNT THE
C PROPERTY CHANGES ACROSS THE EXCHANGER.
C   CALL DELP (CT,TUBEID,FMDOT,FLAREA,CA,UK,DELPW,NPASS,BANKW)
C
C THE PARTICULARS HAVE NOW ALL BEEN CALCULATED. THE HEAT BALANCE MUST
C NOW BE PERFORMED ALONG WITH DEFINING OBJECTIVE AND CONSTRAINT
C FUNCTIONS.
C
C THE HEAT TRANSFER RATES:
C   QDOT1=UKM*AT*DELTM/144.
C   QDOT2=FLCAP*(FLTMP1-FLTMP2)

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A2380
 A2390
 A2400
 A2410
 A2420
 A2430
 A2440
 A2450
 A2460
 A2470
 A2480
 A2490
 A2500
 A2510
 A2520
 A2530
 A2540
 A2550
 A2560
 A2570
 A2580
 A2590
 A2600
 A2610
 A2620
 A2630
 A2640
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 A2660
 A2670
 A2680
 A2690
 A2700
 A2710
 A2720
 A2730
 A2740
 A2750
 A2760
 A2770
 A2780
 A2790
 A2800
 A2810
 A2820
 A2830
 A2840
 A2850

C QDOT3=ARCAP*(ARTMP2-ARTMP1)
 C
 C
 C DEFINING SOME HEAT TRANSFER DIFFERENCES FOR USE AS CONSTRAINING
 C FUNCTIONS AND AS PENALTIES.
 C
 C DELQ1=QDOT-QDOT1
 C
 C QRATIO=QDOT1/QDOT
 C
 C QRAT01=QDOT2/QDOT1
 C
 C QRAT02=QDOT3/QDOT2
 C
 C DELQ2=QDOT1-QDOT2
 C
 C DELQ3=QDOT2-QDOT3
 C
 C
 C DEFINE THE PENALTY FUNCTION AS:
 C
 C PENLTY=R1*QRATIO
 C
 C NOW DEFINE CONSTRAINTS WHICH WILL KEEP THE TUBE BANK WITHIN REASONABLE
 C LIMITS:
 C
 C THE RATIO OF FIN DIAMETER TO TUBE DIAMETER NEEDS TO BE KEPT WITHIN
 C LIMITS. THE LOWER LIMIT WILL BE HANDLED BY A SIDE CONSTRAINT ON FINHT
 C DRATIO=(2.*FINHT+TUBEOD)/TUBEOD
 C
 C THE TUBE THICKNESS MUST BE KEPT WITHIN REASONABLE LIMITS:
 C
 C TUBTH=(TUBEOD-TUBEID)/2.
 C
 C THE TUBES MUST BE KEPT FROM TOUCHING IN BOTH THE LONGITUDINAL AND
 C TRANSVERSE DIRECTIONS:
 C
 C TOUCHN=(TUBEOD+2.*FINHT)-PITCHN
 C
 C TOUCHL=(TUBEOD+2.*FINHT)-PITCHL
 C
 C THE TEMPERATURE PROFILES IN CROSS FLOW AT BOT ENDS OF THE EXCHANGER
 C MUST LOOK REASONABLE.
 C

A2860
 A2870
 A2880
 A2890
 A2900
 A2910
 A2920
 A2930
 A2940
 A2950
 A2960
 A2970
 A2980
 A2990
 A3000
 A3010
 A3020
 A3030
 A3040
 A3050
 A3060
 A3070
 A3080
 A3090
 A3100
 A3110
 A3120
 A3130
 A3140
 A3150
 A3160
 A3170
 A3180
 A3190
 A3200
 A3210
 A3220
 A3230
 A3240
 A3250
 A3260
 A3270
 A3280
 A3290
 A3300
 A3310
 A3320
 A3330

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C      PROFH=ARTMP2--LTMPI
C      PROFC=ARTMP1--LTMPI
C
C      TO PLACE CONSTRAINTS ON THE LONGITUDINAL AND TRANSVERSE PITCH IN ORDER
C      TO MAINTAIN AN ISOLES TRIANGULAR PITCH.
C      ARG7=(PITCHN/2.)/PITCHL
C      ARG8=((TUBEOD+2.*FINHT)/2.)/PITCHL
C      IF (ARG7.GT.1.) GO TO 20
C      IF (ARG8.GT.1.) GO TO 20
C      THETA=ARSIN((PITCHN/2.)/PITCHL)
C      THETAM=ARCOS(((TUBEOD+2.*FINHT)/2.)/PITCHL)
C      DELPHI=THETA-THETAM
C      GO TO 30
C      THETAM=0.0
C      THETA=3.14
C      DELPHI=3.14
C
C      DEFINE THE DIFFERENT OBJECTIVE FUNCTIONS THAT MAY BE USED. THE MUL-
C      TIPLIER IN FRONT OF THE PENALTY, Y, WILL EITHER BE SET TO 1 OR 0
C      DEPENDING ON WHETHER THE FUNCTION IS TO BE OPTIMIZED OR NOT.
C      EXCHANGER VOLUME IN CUBIC FEET:
C      VOLUME1=(BANKH*BANKW*((TUBEOD+2.*FINHT)+(NROWS-1))*((PITCHL*COS(THETA
30      1))))/1728.
C      VOLUME=VOLUME1+(Y1*PENLTY)
C      EXCHANGER AREA IN SQ. FEET:
C      AREA1=AT/144.
C      AREA=AT/144.+Y2*PENLTY
C      AIR HORSEPOWER:

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A3340
A3350
A3360
A3370
A3380
A3390
A3400
A3410
A3420
A3430
A3440
A3450
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A3470
A3480
A3490
A3500
A3510
A3520
A3530
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A3550
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A3570
A3580
A3590
A3600
A3610
A3620
A3630
A3640
A3650
A3660
A3670
A3680
A3690
A3700
A3710
A3720
A3730
A3740
A3750
A3760
A3770
A3780
A3790
A3800
A3810

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C      FHP1=CDELPA*AMDOT*SVOL*7.27273E-5
C      FHP=CDELPA*AMDOT*SVOL*7.27273E-5+Y3*PENLTY
C      PRESSURE DROP IN PSI:
C      PPA1=CDELPA
C      PPA=CDELPA+Y4*PENLTY
C      PPW1=DELPW
C      PPW=DELPW+Y5*PENLTY
C      RETURN
C      ----- PRINT RESULTS ----- PRINT RESULTS -----
C      WRITE (6,200)
C      WRITE (6,120) FMDOT,FLIMP1,FLIMP2,CP
C      WRITE (6,130) AMDOT,ARIMP1,ARIMP2,ACP,PRES1
C      WRITE (6,140) TUBEID,TUBEOD,FINHT,FINTH,FINSP
C      WRITE (6,150) PITCHN,PITCHL,BANKH,BANKW
C      WRITE (6,290) THETA,DELPHI
C      WRITE (6,160) ITYPE,JTYPE,VROWS,NPASS
C      WRITE (6,300) IVROW
C      WRITE (6,170) QDOT,TUBEK,AK,GASCON
C      WRITE (6,180) R1,R2,R3
C      WRITE (6,190) Y1,Y2,Y3,Y4,Y5
C      WRITE (6,320) QDOT1,QDOT2,QDOT3,DELQ1,DELQ2,DELQ3,PENLTY,DRATIO,TU
C      18TH,TOJCHN,TOUCHL,PROFH,PRJFC,VOLUMI,AREAI,FHP1,PPA1,PPW1
C      IF (Y1.EQ.1.) GO TO 50
C      IF (Y2.EQ.1.) GO TO 60

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A3820
A3830
A3840
A3850
A3860
A3870
A3880
A3890
A3900
A3910
A3920
A3930
A3940
A3950
A3960
A3970
A3980
A3990
A4000
A4010
A4020
A4030
A4040
A4050
A4060
A4070
A4080
A4090
A4100
A4110
A4120
A4130
A4140
A4150
A4160
A4170
A4180
A4190
A4200
A4210
A4220
A4230
A4240
A4250
A4260
A4270
A4280
A4290

C	IF (Y3.EQ.1.) GO TO 70	A4300
C	IF (Y4.EQ.1.) GO TO 80	A4310
C	IF (Y5.EQ.1.) GO TO 90	A4320
C	WRITE (6,310) VOLUME	A4330
50	GO TO 100	A4340
C	WRITE (6,310) AREA	A4350
60	GO TO 100	A4360
C	WRITE (6,310) FHP	A4370
70	GO TO 100	A4380
C	WRITE (6,310) PPA	A4390
80	GO TO 100	A4400
C	WRITE (6,310) PPW	A4410
90		A4420
C		A4430
C		A4440
C		A4450
C		A4460
C		A4470
C		A4480
C		A4490
C		A4500
C		A4510
C		A4520
C		A4540
C		A4550
C		A4560
C		A4570
C		A4580
C		A4590
C		A4600
100		A4610
110	RETURN	A4620
120	FORMAT (1H1,10X,38HAIR-COOLED HEAT EXCHANGER OPTIMIZATION,////,1	A4630
	15X,10HINPUT DATA)	A4640
	FORMAT (///,10X,19HTUBESIDE PARAMETERS,/,5X,16HMASS FLOW RATE=,F	A4650
	114.4,7H LBM/HR,/,5X,19HINLET TEMPERATURE=,F14.4,6H DEG F,/,5X,20H	A4660
	2OUTLET TEMPERATURE=,F14.4,6H DEG F,/,5X,15HSPECIFIC HEAT=,F14.4,	A4670
	310H BTU/LBM-F)	A4680
	FORMAT (///,10X,18HAIRSIDE PARAMETERS,/,5X,16HMASS FLOW RATE=,F1	A4690
	14.4,7H LBM/HR,/,5X,19HINLET TEMPERATURE=,F14.4,6H DEG F,/,5X,20H	A4700
	2OUTLET TEMPERATURE=,F14.4,6H DEG F,/,5X,15HSPECIFIC HEAT=,F14.4,1	A4710
	30H BTU/LBM-F,/,5X,16HINLET PRESSURE=,F14.4,4H PSI)	A4720
	FORMAT (///,10X,13HTUBE GEOMETRY,/,5X,18HTUBE INSIDE DIA.=,F14.4	A4730
	1,7H INCHES,/,5X,19HTUBE OUTSIDE DIA.=,F14.4,7H INCHES,/,5X,12HFIN	A4740
	2,7H INCHES,/,5X,15HFIN THICKNESS=,F14.4,7H INCHES,	A4750
	3/,5X,31HFIN SPACING, CENTER-TO-CENTER=,F14.4,7H INCHES,	A4760
	FORMAT (///,10X,16HTUBE ARRANGEMENT,/,5X,18HTRANSVERSE PITCH=,F1	A4770
	14.4,7H INCHES,/,5X,20HLONGITUDINAL PITCH=,F14.4,7H INCHES,/,5X,13	
	2HBANK HEIGHT=,F14.4,7H INCHES,/,5X,12HBANK WIDTH=,F14.4,7H INCHE	

149

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90-

B B B B B B B B B

FUNCTION VISCFL (T)
C CALCULATES THE VISCOSITY OF WATER IN LBM/FT-HR GIVEN TEMPERATURE IN
C DEGREES F ACCORDING TO ANDRADE'S LAW.
C
C VISCFL=.01339*EXP(2715.7764/(T+460.))
C
C RETURN
C END

```

C      FUNCTION TCFL (T)
C      CALCULATES THE THERMAL CONDUCTIVITY OF WATER IN BTU/HR-FT-F GIVEN
C      TEMPERATURE IN DEGREES F.
C      TCFL=.31128+5.84054E-4*T-9.931E-7*T*T
C      RETURN
C      END

```

```

C      20
C      30
C      40
C      50
C      60
C      70
C      80
C      90-

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D	10
D	20
D	30
D	50
D	60
D	70
D	80
D	90

```

FUNCTION VISCAR (T)
C CALCULATES THE VISCOSITY OF AIR IN LBM/FT-HR GIVEN TEMPERATURE IN
C DEGREES F.
C
C VISCAR=.03939*6.72E-5*T-2.1E-8*T*T
C
C RETURN
C END

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EEEEEEEEE

FUNCTION TCAR (T)
C CALCULATES THE THERMAL CONDUCTIVITY OF AIR IN BTU/HR-FT-F GIVEN
C TEMPERATURE IN DEGREES F.
C
C TCAR=.01312+2.62806E-5*T-7.0E-9*T*T
C
C RETURN
C END

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70
80
90
100
110
120
130
140-
F F F F F F F F F F F F F F

FUNCTION FF (REYN)
C CALCULATES THE TUBESIDE FRICTION FACTOR ACCORDING TO THE STANDARDS OF
C THE TUBULAR EXCHANGER MANUFACTURER'S ASSOCIATION, FIFTH ED., 1970.
C
C IF (REYN.LT.1000.) FF=.5/REYN
C IF (REYN.GE.1000.) FF=.0032149*(REYN**(-.2694))
C
C THE FRICTION FACTOR IS DIMENSIONAL, SQ.FT./SQ.IN.
C
C THE AREA BETWEEN REYN= 1000 TO 3000 IS UNSTABLE AND RESULTS ARE
C QUESTIONABLE IN THIS RANGE.
C RETURN
C END

```

C      FUNCTION FLDENS (T)
C      GIVEN A TEMPERATURE IN DEGREES F, THE FUNCTION WILL CALCULATE THE
C      DENSITY OF SATURATED WATER IN LBM/CU.FT.
C      FLDENS=62.67137-.0024345*T-5.089E-5*T*T
C      RETURN
C      END

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GGGGGGGG
10 20 30 40 50 60 70 80 90-

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C 20 P=DUMMY1/(FLTMP1-ARTMP1)
C Q=DUMMY2/(FLTMP1-ARTMP1)
C DUMMY3=(1.-Q)/(1.-P)
C IF (DUMMY3.LT.1.01) GO TO 30
C DIMENSIONLESS LMTD OF PURE COUNTERCURRENT.
C DIMM=(P-Q)/(ALOG(DUMMY3))
C GO TO 40
C 30 DIMM=ALMTD/(FLTMP1-ARTMP1)
C FOR PURE COUNTERFLOW
C FT=1.
C 40 IF (ITYPE.EQ.10) GO TO 160
C LISTING OF COEFFICIENTS FOR EACH TYPE ARRANGEMENT
C IF (ITYPE.GT.1) GO TO 50
C A(1,1)=-.462
C A(1,2)=5.087
C A(1,3)=-15.7
C A(1,4)=17.2
C A(2,1)=-.0313
C A(2,2)=-.529
C A(2,3)=-2.37
C A(2,4)=3.18
C A(3,1)=-.174
C A(3,2)=1.32
C A(3,3)=-2.93
C A(3,4)=1.99
C A(4,1)=-.042
C A(4,2)=.347
C A(4,3)=-.853
C A(4,4)=.649
C GO TO 130
C 50 IF (ITYPE.GT.2) GO TO 60
C A(1,1)=-.334

```

```

490
500
510
520
530
540
550
560
570
580
590
600
610
620
630
640
650
660
670
680
690
700
710
720
730
740
750
760
770
780
790
800
810
820
830
840
850
860
870
880
890
900
910
920
930
940
950
960

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970
 H 980
 H 990
 H 1000
 H 1010
 H 1020
 H 1030
 H 1040
 H 1050
 H 1060
 H 1070
 H 1080
 H 1090
 H 1100
 H 1110
 H 1120
 H 1130
 H 1140
 H 1150
 H 1160
 H 1170
 H 1180
 H 1190
 H 1200
 H 1210
 H 1220
 H 1230
 H 1240
 H 1250
 H 1260
 H 1270
 H 1280
 H 1290
 H 1300
 H 1310
 H 1320
 H 1330
 H 1340
 H 1350
 H 1360
 H 1370
 H 1380
 H 1390
 H 1400
 H 1410
 H 1420
 H 1430
 H 1440

A{1,2,3}=3.3
 A{1,2,3}=-8.7
 A{1,3,4}=8.7
 A{1,3,4}=-1.54
 A{2,1,3}=1.28
 A{2,1,3}=-3.35
 A{2,3,4}=2.83
 A{2,3,4}=-2.865
 A{3,1,2}=-.546
 A{3,1,2}=.929
 A{3,3,4}=-.471
 A{3,3,4}=.0553
 A{4,1,2}=-.405
 A{4,1,2}=-.953
 A{4,3,4}=-.717
 A{4,3,4}=-.0746

GO TO 130

IF (ITYPE.GT.3) GO TO 70

A{1,1,1}=-.0874
 A{1,2,3}=1.05
 A{1,2,3}=-2.45
 A{1,3,4}=3.21
 A{1,3,4}=-.0318
 A{2,1,3}=-.274
 A{2,1,3}=-.746
 A{2,3,4}=-.668
 A{2,3,4}=-.0183
 A{3,1,2}=-.123
 A{3,1,2}=-.156
 A{3,3,4}=-.0617
 A{3,3,4}=-.0071
 A{4,1,2}=-.0499
 A{4,1,2}=-.109
 A{4,3,4}=-.0746

GO TO 130

IF (ITYPE.GT.4) GO TO 80

A{1,1,1}=-.0414
 A{1,2,3}=-.615
 A{1,3,4}=-1.2
 A{1,3,4}=2.06
 A{2,1,3}=-.0139
 A{2,1,3}=-.123
 A{2,3,4}=-.345

C
 60
 C

C
 70
 C

H1450
 H1460
 H1470
 H1480
 H1490
 H1500
 H1510
 H1520
 H1530
 H1540
 H1550
 H1560
 H1570
 H1580
 H1590
 H1600
 H1610
 H1620
 H1630
 H1640
 H1650
 H1660
 H1670
 H1680
 H1690
 H1700
 H1710
 H1720
 H1730
 H1740
 H1750
 H1760
 H1770
 H1780
 H1790
 H1800
 H1810
 H1820
 H1830
 H1840
 H1850
 H1860
 H1870
 H1880
 H1890
 H1900
 H1910
 H1920

A(2,4)=.318
 A(3,1)=-.00723
 A(3,2)=.0566
 A(3,3)=-.0437
 A(3,4)=.0111
 A(4,1)=.0061
 A(4,2)=-.0463
 A(4,3)=.107
 A(4,4)=-.0757

GO TO 130

IF (ITYPE.GT.5) GO TO 90

A(1,1)=-.235
 A(1,2)=2.28
 A(1,3)=-6.44
 A(1,4)=6.24
 A(2,1)=-.0773
 A(2,2)=.632
 A(2,3)=-1.63
 A(2,4)=1.35
 A(3,1)=-.0598
 A(3,2)=.364
 A(3,3)=-.613
 A(3,4)=.276
 A(4,1)=.00525
 A(4,2)=-.0127
 A(4,3)=-.0114
 A(4,4)=.0272

GO TO 130

IF (ITYPE.GT.6) GO TO 100

A(1,1)=-.843
 A(1,2)=5.85
 A(1,3)=-12.8
 A(1,4)=9.14
 A(2,1)=.0302
 A(2,2)=-.00964
 A(2,3)=-.228
 A(2,4)=.266
 A(3,1)=.48
 A(3,2)=-3.28
 A(3,3)=7.11
 A(3,4)=-4.9
 A(4,1)=.0812

C 90 C

C 80 C

H1930
H1940
H1950
H1960
H1970
H1980
H1990
H2000
H2010
H2020
H2030
H2040
H2050
H2060
H2070
H2080
H2090
H2100
H2110
H2120
H2130
H2140
H2150
H2160
H2170
H2180
H2190
H2200
H2210
H2220
H2230
H2240
H2250
H2260
H2270
H2280
H2290
H2300
H2310
H2320
H2330
H2340
H2350
H2360
H2370
H2380
H2390
H2400

A(4,2)=-.834
A(4,3)=2.15
A(4,4)=-1.69
GO TO 130
IF (ITYPE.GT.7) GO TO 110

A(1,1)=-.605
A(1,2)=4.34
A(1,3)=-9.72
A(1,4)=7.54
A(2,1)=-.0231
A(2,2)=-.0059
A(2,3)=-.248
A(2,4)=-.287
A(3,1)=-.294
A(3,2)=-1.99
A(3,3)=4.32
A(3,4)=-3.
A(4,1)=-.0198
A(4,2)=-.305
A(4,3)=-.897
A(4,4)=-.731

GO TO 130

IF (ITYPE.GT.8) GO TO 120

A(1,1)=-.339
A(1,2)=2.38
A(1,3)=-5.26
A(1,4)=3.9
A(2,1)=-.0277
A(2,2)=-.0999
A(2,3)=-.0904
A(2,4)=-.000845
A(3,1)=-.179
A(3,2)=-1.21
A(3,3)=2.62
A(3,4)=-1.81
A(4,1)=-.0199
A(4,2)=-.04
A(4,3)=-.0494
A(4,4)=-.0981

GO TO 130

C
C
C
100

C
C
C
110

C
C

C TYPE 9 - PURE CROSSFLOW

120 A(1,1)=-.0669
 A(1,2)=-.278
 A(1,3)=1.11
 A(1,4)=.136
 A(2,1)=0.0
 A(2,2)=0.0
 A(2,3)=0.0
 A(2,4)=0.0
 A(3,1)=.0395
 A(3,2)=-.22
 A(3,3)=.454
 A(3,4)=-.258
 A(4,1)=0.0
 A(4,2)=0.0
 A(4,3)=0.0
 A(4,4)=0.0

C CALCULATION OF CORRECTION FACTOR, FT.

130 R=P/Q

SUMI=0.0

DO 150 I=1,4

DO 140 K=1,4

SUM=A(I,K)*((1.-DIMI)**K)*SIN(2.*I*ATAN(R))

SUMI=SUMI+SUM

CONTINUE

CONTINUE

FT=1.-SUMI

C THE GIVEN TEMP DIFFERENCE FOR THE ARRANGEMENT ...

DELTM=FT*ALMTD

RETURN

END

H2410
 H2420
 H2430
 H2440
 H2450
 H2460
 H2470
 H2480
 H2490
 H2500
 H2510
 H2520
 H2530
 H2540
 H2550
 H2560
 H2570
 H2580
 H2590
 H2600
 H2610
 H2620
 H2630
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 H2700
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 H2720
 H2730
 H2740
 H2750
 H2760
 H2770
 H2780
 H2790
 H2800
 H2810
 H2820
 H2830
 H2840-

[illegible]

I 490
I 500
I 510
I 520-
I 530-

T2P=ARTMP1+DUMMY5*DUMMY3
RETURN
END

C
C


```

C SUBROUTINE TUBES (BANKH,TUBEED,FINHT,PITCHN,NROWS,NPASS,TUBEID,TAR
C LEA,TOTAL,IVROW,FLAREA,VROWR)
C
C VERTICAL ROWS CONTAIN (EQUAL NO. PER VERT. ROW) .....
C
C VROWR=(BANKH-(TUBEED+2.*FINHT)-PITCHN/2.)/PITCHN
C
C IF (VROWR.LT.2.) VROWR=2.
C
C THIS IS A REAL NUMBER OF TUBES.
C
C TRUNCATING TO A WHOLE NUMBER OF TUBES.
C IVROW=VROWR
C
C TOTAL NUMBER OF TUBES IN THE BANK.
C
C TOTAL=IVROW*NROWS
C
C FLOW AREA PER TUBE IN SQ.IN.
C
C TAREA=3.14159*TUBEID*TUBEID/4.
C
C CALCULATE THE TOTAL FLOW AREA IN SQ.IN.
C
C IF (NPASS.EQ.1) GO TO 10
C
C IF (NPASS.EQ.NROWS) GO TO 20
C
C IF (NPASS.EQ.NROWS/2) FLAREA=TAREA*(2.*IVROW)
C
C GO TO 30
C
C FLAREA=TAREA*TOTAL
C
C GO TO 30
C
C FLAREA=TAREA*IVROW
C
C RETURN
C
C END

```

```

K 10
K 20
K 30
K 40
K 50
K 60
K 70
K 80
K 90
K 100
K 110
K 120
K 130
K 140
K 150
K 160
K 170
K 180
K 190
K 200
K 210
K 220
K 230
K 240
K 250
K 260
K 270
K 280
K 290
K 300
K 310
K 320
K 330
K 340
K 350
K 360
K 370
K 380
K 390
K 400-

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SUBROUTINE HTAREA (FINPIN,TUBEOD,FINHT,FINTH,AS,AT,ASAT,TOTAL,BANK
1W)
C THIS SUBROUTINE CALCULATES THE AIRSIDE FIN AREA, THE TOTAL HEAT
C TRANSFER AREA AND THEIR RATIO.
C FINNED AREA PER TUBE PER INCH IN SQ.IN.
C DUMMYO=FINPIN*((TUBEOD+2.*FINHT)**2-TUBEOD**2)*1.5708)
C BARE TUBE AREA PER INCH OF TUBE IN SQ.IN.
C DUMMY1=(1.-((FINPIN*FINHT))*TUBEOD*3.145927
C TOTAL INCHES OF TUBE IN BANK IN SQ.IN.
C DUMMY2=TOTAL*BANKW
C TOTAL FINNED AREA IN SQ. IN.
C AS=DUMMYO*DUMMY2
C TOTAL HEAT TRANSFER AREA IN SQ.IN.
C AT=(DUMMYO+DUMMY1)*DUMMY2
C RATIO OF FINNED AREA TO TOTAL
C ASAT=AS/AT
C RETURN
C END
SUBROUTINE FILMI (CT,TUBEID,FMDOT,FLAREA,L,CP,BANKW,HI)
C ***THIS SUBROUTINE IS FOR FORCED CONVECTION OF SINGLE PHASE,
C INCOMPRESSIBLE FLUID IN A SMOOTH TUBE.
C DIMENSION CT(2),HI(2)
C DO 10 I=1,2
C THE REYNOLDS NO. ....
C VISCOS - VISCOSITY OF THE FLUID
C VISCOS=VISCFL(CT(I))
C REYN=(TUBEID*FMDOT*12.)/((FLAREA*VISCOS)
C IS THE FLOW TURBULENT, LAMINAR OR IN TRANSITION?

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```

C
C
C TC - THERMAL CONDUCTIVITY OF THE FLUID.
C
C TC=TCFL(CT(1))
C
C THE PRANDTL NO. ....
C PR=(CP*VISCOS)/TC
C
C THE NUSSELT NO. ....
C
C ... SEIDER - TATE CORRELATION
C IF (L.EQ.1) FLNUS=1.86*((REYN*PR)**.333333)*((TUBEID/BANKW)**.3333
133)
C
C IF (L.EQ.0) FLNUS=.027*(REYN**.8)*(PR**.333333)
C
C ... HAUSEN CORRELATION ...
C
C IF (L.EQ.2) FLNUS=.116*((REYN**.666667)-125.)*(PR**.333333)*(1.+(
1TUBEID/BANKW)**.666667)
C
C THE FILM COEFFICIENT .....
C
C HI(1)=(FLNUS*TC*12.)/TUBEID
C
C CONTINUE
C
C RETURN
C
C END

```

```

M 180
M 190
M 200
M 210
M 220
M 230
M 240
M 250
M 260
M 270
M 280
M 290
M 300
M 310
M 320
M 330
M 340
M 350
M 360
M 370
M 380
M 390
M 400
M 410
M 420
M 430
M 440
M 450
M 460
M 470
M 480
M 490
M 500-

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[illegible]

C COEFFICIENT.
 RHO=.074
 C 10
 CONTINUE
 END

490
 500
 510
 520
 530
 540
 550-
 N N N N N N N

```

C C SUBROUTINE FILM3 (TUBEDD,AMDOT,SFF,FINSP,FINTH,S,FINHT,CTP,HO,ACP)
C C THE UNCORRECTED AIRSIDE FILM COEFFICIENT CAN BE CALCULATED DIRECTLY
C C USING THE BRIGGS-YOUNG CORRELATION. THE CORRELATION WILL YIELD GOOD
C C RESULTS FOR TRIANGULAR PITCH BANKS OF HIGH FINNED TUBES WITH SIX
C C ROWS. FOR OTHER THAN SIX ROWS A CORRECTION IS NECESSARY.
C C
C C DIMENSION CTP(2),HO(2)
C C DO 10 I=1,2
C C FIRST, THE REYNOLDS NO. ....
C C ... VISCOSITY ...
C C VISCA=VISCAR(CTP(I))
C C REYNA=(TUBEDD*AMDOT*12.)/(SFF*VISCA)
C C THE PRANDLT NO. ....
C C ... THERMAL CONDUCTIVITY ...
C C TCA=TCAR(CTP(I))
C C PRA=(ACP*VISCA)/TCA
C C CALCULATE THE DISTANCE BETWEEN FINS IN INCHES .....
C C S=FINSP-FINTH
C C CALCULATE THE NUSSELT NO. ....
C C ARNUS=.1378*(REYNA**.718)*(PRA**.33333333)*((S/FINTH)**.296)
C C THE FILM COEFFICIENT IN BTU/HR-SQ.FT-F .....
C C HO(I)=(ARNUS*TCA*12.)/TUBEDD
C C CONTINUE
C C RETURN
C C END
10
C

```

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00 110
00 120
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00 270
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00 290
00 300
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00 350
00 360
00 370
00 380
00 390
00 400
00 410-

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P 970
P 980
P 990
P1000
P1010
P1020
P1030
P1040
P1050
P1060
P1070-

```

C      IF (VMAXS-11.300.) C=((-.02175*ALOG(ROWS)+.9349)/((300.-VMAXS)*((.
102175*ALOG(ROWS)+.9349)-(.036*ALOG(ROWS)+.877)))/700.)))/(.975+((3
200.-VMAXS)*.025)/700.1))

C      IF (VMAXS-GT.1000.) C=((-.036*ALOG(ROWS)+.877)-((VMAXS-1000.)*((.0
12175*ALOG(ROWS)+.9349)-(.036*ALOG(ROWS)+.877)))/700.1)/(.95-((VMA
2XS-1000.)*.025)/700.1))

C50    RETURN
      END

```

174

```

C      FUNCTION B10 (X)
C      TAKEN FROM REFERENCE (46).
C      T=X/3.75
C      IF (ABS(X)-3.75) 10,10,20
C      B10=1.+3.5156229*T**2+3.0899424*T**4+1.2067492*T**6+0.2659732*T**8
C      1+0.0360768*T#10+0.0045813*T#12
C      RETURN
C      B10=(0.39A9422A+0.01328592/I+0.00225319/I**2-0.00157565/I**3+.0091
C      16281/I**4-.02057706/I**5+.02635537/I**6-.01647633/I**7+.00392377/I
C      2**8)*EXP(X)/SQRT(X)
C      RETURN
C      END

```


10	FUNCTION BK1 (X)	
20	TAKEN FROM REFERENCE (46).	
30	Y=X/2	
40	IF (X-2.) 10,10,20	
50	BK1=ALOG(T)*811(X)+(1.+15443144*T**2-.67278579*T**4-.18156897*T**	
60	16-.01919402*T**8-.00110404*T**10-.00004686*T**12)/X	
70	RETURN	
80	BK1=(1.25331414+.23498619/T-.0365562/T**2+.01504268/T**3-.00780353	
90	1/T**4+.00325614/T**5-.00068245/T**6)/SQR(T(X))/EXP(X)	
100	RETURN	
110	END	
120		
130		
140		
150		
160		
170		
180		

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400
410
420-

```

C      SUBROUTINE DELTAP (CTP,TUBEED,AMDOT,SFF,PITCHN,PITCHL,PRES1,GASCON
C      1,NROWS,UK,DELPA,CDELPA,SVOL,ARG5)
C      DIMENSION CTP(2),DELPAR(2),UK(2),DUMMY(2)
C      THIS SUBROUTINE CALCULATES THE AIRSIDE PRESSURE DROP, DELPA, USING
C      THE ROBINSON - BRIGGS CORRELATION FOR TRIANGULAR PITCH BANKS OF HIGH-
C      FINNED TUBES.
C      ROETZEL'S METHOD WHICH TAKES INTO ACCOUNT THE AIR'S DEPENDENCE ON
C      DENSITY AND ITS CHANGING PROPERTIES THROUGH THE TUBE BANK WILL BE
C      INCORPORATED TO CORRECT DELPA
C      THE REFERENCE PRESSURE DROPS WILL BE CALCULATED.
C      DO 10 I=1,2
C      CALCULATE THE VISCOSITY OF AIR AT THE REFERENCE TEMPERATURE.
C      VISCA=VISCAR(CTP(I))
C      THE REYNOLDS NO. ....
C      REYNA=(TUBEED*AMDOT*12.)/(SFF*VISCA)
C      CALCULATE THE FRICTION FACTOR ... FR. NOTE ITS LACK OF DEPENDENCE
C      ON THE NUMBER OF ROWS.
C      FR=18.93*(REYNA**(-.316))*((PITCHN/TUBEED)**(-.927))*((PITCHN/PITC
C      IHL)**.515)
C      BEFORE THE PRESSURE DROP CAN BE CALCULATED THE REFERENCE DENSITY MUST
C      BE CALCULATED FOR THE INLET PRESSURE IN LBM/CU.FT.
C      RHO=(PRES1*144.)/(GASCON*(CTP(I)+460.))
C      DUMMY(I)=RHO
C      THE PRESSURE DROP IS CALCULATED IN PSI ....
C      DELPAR(I)=(FR*NROWS*AMDOT*AMDOT*144.)/(4.18E8*SFF*SFF*RHO)
C      WHERE 4.18E 8 IS A CONSTANT IN FT/SQ.HR. : SFF, THE FREE FACE AREA IS
C      IN SQ. IN.
C      10 CONTINUE
C      FROM ROETZEL'S EQ. (47), THE UNCORRECTED PRESSURE DROP IS .....
C      DELPA=((DELPAR(1)/UK(1))+(DELPAR(2)/UK(2)))/((1./UK(1))+(1./UK(2)))

```

```

1)
C AS A CONTROL DURING OPTIMIZATION; TO ENSURE THAT THE SORT OF A
C NEGATIVE NJMBER IS NOT TAKEN DURING THE FINAL CORRECTION:
C
C ARG5=1.-(2.*DELPA/PRES1)
C
C CDELPA=DELPA
C
C IF (ARG5.GT.0.0) CDELPA=PRES1*(1.-SQRT(ARG5))
C
C WHERE CDELPA IS THE FINAL CORRECTED PRESSURE DROP.
C
C THE MEAN SPECIFIC VOLUME OF THE AIR FLOWING THROUGH THE EXCHANGER
C IN CU.FT./LBM.
C SVOL=2./((DUMMY(1))+DUMMY(2))
C
C RETURN
C END

```

```

490
W 500
W 510
W 520
W 530
W 540
W 550
W 560
W 570
W 580
W 590
W 600
W 610
W 620
W 630
W 640
W 650
W 660
W 670-

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C
SUBROUTINE DELP (CT,TUBEID,FMDOT,FLAREA,CA,UK,DELPW,NPASS,BANKW)
C
C   DIMENSION CT(2),CA(2),DP(2),UK(2)
C
C   THE TUBESIDE PRESSURE DROP IS CALCULATED IN ACCORDANCE WITH THE
C   STANDARDS OF THE TUBULAR EXCHANGER MANUFACTURER'S ASSOCIATION.
C
C   ROETZEL'S CORRECTION FOR CHANGING FLUID PROPERTIES IS APPLIED.
C   DO 10 I=1,2
C
C   THE VISCOSITY AT THE REFERENCE BULK TEMPERATURE.
C
C   VISCOS=VISCFL(CT(I))
C
C   THE REYNOLDS NO. ....
C
C   REYN=(TUBEID*FMDOT*12.)/(FLAREA*VISCOS)
C
C   THE TUBESIDE FRICTION FACTOR.
C
C   F=FF(KEYN)
C
C   ARECOMMENDED CORRECTION FACTOR :
C
C   IF (REYN.GT.2100.) PHI=CA(I)
C   IF (REYN.LE.2100.) PHI=CA(I)**1.7857
C
C   THE DENSITY AT THE REFERENCE TEMPERATURE IS CALCULATED IN LBM/CU.FT.
C
C   RHO=FLOENS(CT(I))
C
C   CALCULATE THE PRESSURE DROP DISREGARDING EXIT AND ENTRANCE LOSSES.
C
C   DEP=(F*FMDOT*FMDOT*BANKW*NPASS*20736.)/(2.**4.17E8*RHO*TUBEID*PHI*F
C   ILAREA*FLAREA)
C
C   TO THIS ADD THE ADDITIONAL LOSSES. ( EQ.(9-11) KERN & KRAUS)
C
C   ADD=((NPASS-1)*FMDOT*FMDOT*144.)/(2.*RHO*FLAREA*FLAREA*32.2*1.296E
C   17)
C
C   TO GET THE TOTAL PRESSURE DROP ....
C
C   DP(I)=DEP+ADD
C
C   CONTINUE
C
C   THE ACTUAL TUBESIDE PRESSURE DROP IN PSI .....

```

C
C
DEL PW= ((DP(1)/UK(1)))+(DP(2)/UK(2)))/((1./UK(1)))+(1./UK(2)))
RETURN
END

X 490
X 500
X 510
X 520
X 530-

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