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DRY COOLING TOWERS FOR COOLING PLANTS

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ABSTRACT

A literature survey of capital costs of dry cooling towers was performed. Heat exchanger surfaces were compared in terms of heat transfer coefficient and friction power per unit heat transfer area.

A computer program was developed to design the heat exchanger configuration resulting in the lowest total operating cost per kilowatt-hour for a dry cooling tower. Input data includes fuel costs, cost of lost capacity per kilowatt-hour, maintenance and interest rates, turbine heat rate versus back pressure, boiler costs, and friction and heat transfer coefficients for the heat transfer surface.

Studies of periodic exchangers showed that liquid or loose periodic material is impractical. However, rigid periodic surfaces are promising if liquid carryover and subsequent air side evaporation can be reduced, perhaps by using a non-wetting surface. Carryover and evaporation tests of a rigid Teflon-coated periodic surface are underway. If a method to remove large drops from the periodic surface can be perfected, its evaporation rate will be an order of magnitude less than the wet tower's, while the costs of the two should be competitive.

A packed bed heat exchanger consists of small balls attached to the outside of tubes carrying hot liquid. The balls act as fins; air blown through the balls cools the liquid. Several methods of joining balls were tested to find the method which minimizes heat transfer resistance from ball to ball. Zinc coated steel balls and epoxy coated aluminum balls have shown promising performance in simple configurations. The first assemblies made with these techniques exhibited considerable blockage of the air passages between the balls.

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I. INTRODUCTION

This report summarizes the first year's research on dry cooling towers.

Traditionally, power plants have rejected heat from their condensers to large bodies of water. The heat transferred to the water raises the water's temperature, increases evaporation, and may adversely affect the ecology of the water system. To avoid raising the temperature of the water bodies, new power plants use cooling ponds or evaporative cooling towers for condenser heat rejection. However, water losses from these systems are extensive, and an objectionable fog plume may be formed. If saline water is used for evaporative cooling, salt particles may be entrained in the air and deposited in neighboring areas. Because of these problems, many power plants constructed in the future may not be allowed to use evaporative cooling systems. They will be required to use non-evaporative air cooling towers, so called dry towers. Currently the cost of dry towers is significantly higher than that of evaporative cooling systems.

The aim of this program is to develop heat exchangers which will significantly reduce the cost of dry cooling towers for power plants where no other cooling method can be used. To achieve substantial improvements, the emphasis of the research is on unconventional concepts rather than on modest improvements of current designs.

Chapters II and III of the report present the results of a literature survey on dry cooling tower costs and a preliminary comparison of promising heat transfer surfaces. Because of the lack of a systemized

way to compare various heat transfer surfaces or to relate surface characteristics to tower costs, a design program was developed which gives the optimum design, i.e., lowest overall operating cost for a cooling tower made with various heat transfer surfaces. The surfaces can now be compared on the basis of overall system generating costs. Chapter IV describes the details of the computer design program.

The feasibility of air-water periodic heat exchangers is examined in Chapter V, and preliminary tests of a promising concept for a periodic exchanger are reported in Chapter VI. Chapter VII contains test results for a heat exchanger with the air side area augmented by a bed of metal balls.

II. LITERATURE SURVEY ON DRY COOLING TOWER COSTS

A literature survey was made to determine the range of optimized designs by other investigators. Comparisons are difficult to make, as assumptions on costs, design, and climatic conditions vary considerably. Table II-1 contains a breakdown of capital investment costs and some design optimums for given references. Most references rate the cooling tower in accordance with the additional capital required to install the cooling system. Comparisons are confused because designs based on a minimum capital cost must have parameters such as the design temperature arbitrarily chosen.

A more meaningful rating is the increased bus-bar cost due to the addition of a dry cooling system. The bus-bar cost increase must include operating cost, maintenance cost, and interest and amortization of capital cost. The parameters such as design temperature are varied until the minimum bus-bar cost increase is achieved. This rating is used in the computer program described in Chapter IV. Results of the program are included in the table.

TABLE II-1

BREAK-DOWN OF ESTIMATED DRY-COOLING TOWER COSTS

Ref	capital Investment \$/KW	T ₁ , °F	ITD °F	Range °F	Approach °F	TD	Δ Cost ¹ °F mills/kWH	Tower cost incl.	Temp. variation incl.
Indirect System - Forced Draft									
1	12.-19.	155-181				100	.21-4.51 ²	X	
2	17.	150	60			90			
3	26.5	140	45	12	33	95	.82 ²		
4	19.2						.68	X	
5	31.3		55	17	38				
6	22.4	120	25	10	15	95			
7	13.	120		10-20			.51-.64	X	
10	17.	159	60			99	1.25 ²	X	
10-82-A ³	15.8	136.1	61.1	5.6	54.5	75	.65 ²		X
10-81-A ³	16.4	137.5	62.5	6.1	56.4	75	.66 ²		X
10-81-C ³	14.8	134.8	59.8	6.2	53.6	75	.59 ²		X
Indirect System - Natural Draft									
2	20.	146	56			90			
3	24.5	150	55	17	38	95	.77 ²		
4	39.						.98	X	
5	31.3		44	26	18				
7	20.	120		10-20			.51-.64	X	
10	19.	154-155	55-56			99	1.17 ²	X	
Direct System - Forced Draft									
3	19.9	140	45	20	25	95	.59 ²		
7	15.	120		10-20			.416	X	
8	36.8	158	78			80			

1. Additional buss bar cost as compared to a once-through river condenser-
2. Original data was compared to base other than once-through river system (as increased cost over a forced draft wet cooling tower). Numbers have been modified to river condenser using numerical values of ref. [4].
3. Computer programs of figures. Kays, W. M., and A. L. London, Compact Heat Exchangers, McGraw-Hill, New York, 1964.

Legend -

Ref	See reference list
T_1	Steam temperature out of the turbine
ITD	Initial temperature difference, of the maximum ΔT between air and water
Range	Number of degrees water is cooled in the tower
Approach	The minimum Δt between the air and the water
TD	Design ambient air temperature

Note: $\text{Range} + \text{Approach} = \text{ITD}$

$$\text{Range} + \text{Approach} + \text{TD} = T_1$$

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III. PROMISING HEAT TRANSFER SURFACES

At the outset of the program, several novel heat exchanger configurations appeared promising for air cooling towers. In order to evaluate these in a preliminary fashion, existing data was used.

Figure III-1 shows the heat transfer coefficient versus air flow rate per unit frontal area for flow through packed beds, fluidized beds, standard finned tubes, figures 10-83 and 10-79, Kays and London,* and bare tubes. In each case the air flowed on the outside of the tubes. Figure III-2 illustrates the heat exchanger geometries presented in fig. III-1.

A packed bed consists of a bare tube surrounded by metal particles usually spherical in size, the particles act like an extended surface or fin. Results are shown for steel particles with diameters of one quarter of an inch and one half inch. The balls are packed three deep about the tube. The tubes were one inch in diameter. The fin efficiency of the particles was estimated assuming the contact resistance between particles is negligible. This assumption is a key one; experiments are being conducted in this program to determine fabrication techniques which will eliminate the contact resistance.

A fluidized bed is made up of fine loose particles through which air flows vertically. The air velocity is adjusted so that the drag force on the particle balances the gravity force. The particles became highly mobile and are reported to have high heat transfer coefficients. The line labeled G_{MIN} refers to the minimum air flow rate at which the bed of particles becomes fluidized. The line labeled G_{MAX} is the air flow rate above which

* Kays, W. M., and A. C. London, Compact Heat Exchangers, McGraw-Hill, New York, 1964.

FIGURE III-1

Fig. III-2

FLUIDIZED BED

BARE TUBES

PACKED BED

STANDARD SURFACE 'B'

STANDARD SURFACE 'A'

HEAT EXCHANGER CONFIGURATIONS

FIGURE III-3

particles are carried out of the bed by the air stream. The fluidized bed data is shown as a shaded region indicating the approximate uncertainty in the data.

The finned tube data is plotted two ways, based on total heat transfer surface area and based on tube area. The latter method of presentation is used since packed bed and fluidized bed data is also based on tube area. It is expected that the cost of the finned tube is proportional to the total area while the cost of fluidized beds is mainly due to the tube costs. The cost of packed beds is between these extremes because of the fabrication costs to reduce contact resistance between the balls.

Figure III-3 shows the data replotted as heat transfer coefficient versus friction power per square foot of heat transfer area on the air side. To determine friction power for a given value of 'h' one first determines the flow rate from figure III-1. From the data on the particular surface, see Kays and London, the friction factor is found. The friction power is

$$\text{Friction Power} = \frac{(\Delta P)}{\rho} \frac{\dot{m}}{(A)_{HT}} \quad (\text{III-1})$$

From the definition of the friction factor, this can be rewritten as,

$$\text{Friction Power} = \frac{\mu^3}{2\rho^2 g_c} \left(\frac{1}{4r_H}\right)^3 (f) (N_R)^3 \quad (\text{III-2})$$

where N_R is defined as

$$N_R = 4 r_H G / \mu \quad (\text{III-3})$$

Δp is the pressure drop, ρ is the fluid density, $\dot{m}$ is the total fluid flow rate, $(A)_{HT}$ is the total area for heat transfer, N_R is the Reynolds number,

μ and ρ are the air viscosity and density, f is the friction factor, G is mass velocity or flow rate per unit frontal area and r_H is the hydraulic radius which is determined by the geometry of the surface.

On the basis of figure III-3 the packed bed and fluidized bed look promising. It must be kept in mind that figures 1 and 3 do not tell the complete story. No allowance for water side pumping power is made, this varies with tube diameter and length, nor can the system cost be readily ascertained, e.g. what is the trade off between fan power and heat exchanger capital cost. To compare the system costs with various heat exchanger configurations, it is necessary to have a computer program which determines the optimum design of the heat exchanger for each heat exchanger surface. One can not compare various surfaces for the same heat exchanger design because heat exchangers with different surfaces will have different optimum design conditions. For example, if all heat exchangers were compared at a design condition which has a very high frontal air velocity, packed beds would give poor results because of the high fan power requirements while bare tubes would give better performance.

For each heat exchanger surface the optimum heat exchanger design must be determined which gives the smallest incremental increase in the total cost of generated power. The surfaces can then be compared by comparing the performance of their respective optimum designs.

IV. COMPUTER DESIGN PROGRAM

The objective of the computer program is to design the best possible heat exchanger by optimizing heat transfer and thermodynamic relations with economic trade-offs. Previous investigators have made optimization studies, but invariably have optimized their designs using predesigned heat exchanger modules. These prior studies hence become less of an optimization program and more of a study on the best use of available subsystems. To be truly optimizing, a program must not only design a heat exchanger, but also design a heat exchanger as an integral part of the entire power system. Thus in raising the outlet temperature of the turbine, the benefits of heat transfer with a higher temperature differential must be compared with the resulting decrease in Rankine cycle thermal efficiency.

The computer program minimizes the total cost of yearly operation for an indirect cooling tower. This requires varying six variables: design temperature, turbine outlet temperature, range, the ratio of the heat capacity of water to that of air, the cooler's frontal area, and its width-to-length ratio.

Rather than work from predesigned cooling tower modules, the program works from the ground up -- designing the configuration, requirements, costs of the heat exchanger, and the costs of the ensuing changes in the entire thermodynamic system which result from the use of a dry cooling tower. Thus the entire-system minimum is found, not just the minimum between compatible subsystems.

Assumptions built into the program include the use of a single pass, cross flow heat exchanger, and turbine performance which varies with outlet temperature in accordance with a General Electric graph entitled "Estimated Turbine Generation Heat-rate Variation with Elevated Pressures."* The system was also assumed to be induced-draft.

Other parameters, including fan blade costs, boiler costs, fuel costs, replacement power costs, interest rates, heat exchanger surface performance, and base thermal efficiency are required as input data. This allows the program to be general and adaptable to nuclear or fossil fuel plants, different heat exchanger surfaces, and costs merely by changing the input data. Land costs were not considered in the program optimization.

Object

The object of the computer program is to design a dry cooling tower so as to minimize the cost of yearly operation. With input data corresponding to local costs, heat exchanger surface performance, and efficiencies, the main output is the net cost of operation, the design temperature (TD), and the design configuration of the heat exchanger surface. Design temperature is defined as the maximum ambient air temperature at which the plant can generate 100% of its rated capacity.

Description

The procedure is to optimize the design of the heat exchanger for TD's between 60° and 100° F, and compare costs.

* Rossie, J. P., and E. A. Cecil, "Research on Dry-Type Cooling Towers for Thermal Electric Generation: Part I," for the Water Quality Office, Environmental Protection Agency, November 1970, p. 79.

It may be beneficial to indicate briefly an example of the trade-offs which exist in the problem considered. Since q is proportional to $A \Delta T$ for a given quantity of heat (q) to be rejected, the larger the temperature difference (ΔT) between the two fluids, the smaller the required surface area (A) of the heat exchanger. Since heat exchanger costs are proportional to surface area, a higher temperature difference will be reflected as a lower heat exchanger cost. However, for a fixed ambient temperature a higher temperature difference corresponds to a higher turbine exhaust temperature, resulting in a lower cycle efficiency. For a given power generation, a lower efficiency requires a larger boiler capacity, greater fuel consumption, and hence greater costs on the boiler side of the cycle. Note that the temperature difference (ΔT) depends strongly on the ambient air temperature. Hence daily and seasonal fluctuations in air temperatures must be accounted for in the trade-off described above.

With a given heat exchanger surface, a given design temperature, cost data, and power generation, the heat transfer and cost equations may be reduced to five variables which can be varied independently and which must be optimized:

T_1	steam cycle output temperature ($^{\circ}\text{F}$) at the low pressure turbine
RANGE	change in temperature($^{\circ}\text{F}$) of the water while passing through the cooler
CWARA	ratio of the heat capacity of water to that of air, where heat capacity of a fluid is the product of its specific heat and its mass flow rate
AFRON	the air side frontal area (ft^2) of the heat exchanger
WLRAT	ratio of width to length of AFRON, length being the tube length

Thus the cost of adding the dry cooling tower to the power plant is a function of these five variables. The problem, or the optimization process, is to minimize this cost for given values of TD and, by comparison, to choose the TD with the least cost.

In order to explain the minimizing procedure used, let us consider a simplified case. Assume cost is a function of only two independent variables. Hence, cost may be pictured as a surface in three-dimensional space. Figure IV-1A shows this surface in a three-dimensional sketch. Figure IV-1B shows the same figure, with lines of constant cost projected on the plane defined by variables 1 and 2, the horizontal plane of Fig. IV-1A.

For any given starting point, the minimum may be approached by:

- (1) holding variable 1 fixed (e.g. staying in plane A of Figure IV-2A and finding the minimum costs, point 2 on plane A);
- (2) holding variable 2 fixed, staying in plane B, and finding a second minimum, point 3, plane B;
- and (3) continuing the process indefinitely, thus "spiraling in" on the absolute minimum. Figure IV-2B illustrates this same process in a two-dimensional projection of Figure IV-2A.

However, if the surface is not as well-behaved as that drawn in Figure IV-2, but has multiple minimal points as in Figure IV-3 the search procedure may end by "minimizing" in a relative minimum trough. To avoid such relative minima, two approaches have been taken: (1) the cost is evaluated at multiple points and the lowest cost point is used as the starting point for the minimizing procedure. This is a sort of "shotgun" approach, sampling many points chosen to represent a cross-section of reasonable values. A minimum of 687 initial points are checked. (2) the cost is evaluated at discrete, incremental step changes in the variable's

Three-Dimensional Visualization
Figure IV-1A

Two-Dimensional Visualization
with Constant Cost Loci
Figure IV-1B

Figure IV-1 Cost as a Function of Two Variables

Three-Dimensional Visualization
Figure IV-2A

Two-Dimensional Visualization
Figure Iy-2B

Figure IV-2 Minimization Technique on Simple Surface

Three-Dimensional Visualization
Figure IV-3A

Two-Dimensional Visualization
Figure IV-3B

Figure IV-3 Minimization on a "Multi-Minimal" Surface

magnitude. Both positive and negative step changes are searched for points of minimum cost. If lower cost is not determined by the initial step size, the step size is diminished and the process is checked again. Hence the step size decreases as the minimum is approached.

The use of the shot-gun approach affords a starting point somewhere in the proximity of the minimum, while the second approach allows the search procedure to step over, or out of, non-minimum troughs.

Imposed restrictions must also be considered. For example, flow abnormalities through the heat exchanger may be experienced as a result of external conditions such as wind. An air stream flowing over a surface of the heat exchanger creates locations of both high and low external pressures. A location of increased external pressure will experience an enhancement of air flow through the exchanger, while a point of decreased external pressure will be "starved" of flow. Hence, depending on the shape of the heat exchanger, improperly directed wind could nullify or greatly hinder local air flows through the exchanger. As flow abnormalities tend to reduce the effectiveness of the heat exchanger, it is desirable to reduce flow abnormalities.

Initially the air velocities (VAIR) through the exchanger were maintained above a specified minimum. It was hypothesized that a high VAIR would minimize the effects of the ambient air velocity. However, specifying a minimum VAIR penalizes heat exchanger configurations having an inherently high air side friction. Specifying a minimum VAIR is the same as specifying a very high air side pressure drop for configurations with high friction factors. This in turn requires a much larger fan power to create this air side pressure drop. A more logical requirement is a

minimum air side pressure drop, ΔP_{air} , through the heat exchanger. Since the driving force for the air flow is the pressure differential, as long as the maintained pressure differential is greater than an external differential caused by wind, flows through the exchanger will not be greatly affected. Note that this allows the "high friction" configurations to adopt lower values of ΔP_{air} , so long as ΔP_{air} is above a minimum, thereby saving on fan power.

To approximate the pressure differential caused by wind, consider the case of potential non-viscous flow around a cylinder. This would be analogous to wind blowing around a chimney or a heat exchanger of cylindrical configuration. Pressures caused by a wind of 20 mph would range from a pressure increase of 1.02 lbf/ft^2 at the stagnation point to a decrease of 3.06 lbf/ft^2 at a point 90° from the wind direction. In the present optimization runs, the minimum air side pressure drop ($\Delta P_{\text{air, min}}$) was specified as 4.0 lbf/ft^2 . Since flow rate is directly proportional to the square root of the pressure differential, a heat exchanger designed for a 4 psf minimum pressure drop and being subjected to 20 mph winds would experience approximately a 10% increase of flow at the point facing the wind (stagnation point) and approximately a 50% reduction in flow at a point 90° from the direction of the wind. Computer runs made without pressure drop restrictions optimize at air pressure drops of about 1 psf. Such a design in a 20 mph wind would experience a 40% increase in flow at the stagnation point, and a reversal of flow at a point 90° to the wind direction. There would also be a point of no flow somewhere between these two extremes. This definitely indicates a need for the pressure drop

restriction. Flow irregularities may be further minimized by specifying a higher minimum pressure drop, an input variable to the optimization program.

Pictorially, restrictions on the air velocity or tube length pass a plane or surface through the three-dimensional cost surface, eliminating from consideration certain areas of the cost surface. As illustrated in Figure IV-4, the eliminated points may well include the absolute minimum, making some other point (point B) the desired minimum.

Simple restrictions on the value of the variables may lead to incorrect answers if only the minimization procedure of Figure IV-2 is used. For example, in Figure IV-4, suppose the search started at point 1 with variable #1 being fixed. Attempting to reach planar minimum C, variable #2 is stopped at point 2 by the restriction on DELPA. Now holding variable #2 constant, variable #1, attempting to reach planar minimum D, cannot pass point 2 either. The search procedure could not proceed further, and would evaluate point 2 as the minimum. To arrive at B, both variables are allowed to change simultaneously, thus proceeding down line 2B towards B. In the five-variable case,

$$\text{DELP} = f(T_1, \text{RANGE}, \text{CWARA}, \text{AFRON}, \text{WLRAT})$$

So long as this function, or DELPA is maintained constant, we are at liberty to modify the variables which compose the function. Thus, if RANGE and AFRON are variables #1 and #2 respectively in Figure an increase in range accompanied by a decrease in AFRON will move down line 2B towards B, while keeping DELPA fixed.

Figure IV-4 Minimization with Imposed Restrictions

Since one can also expect to encounter sub-minimal troughs while moving along line 2B, a step size procedure similar to that in Figure IV-2 is used. Again, the step size diminishes as the minimum is approached.

In order both to widen the initial search range and to more accurately determine a starting point, the shot-gun method has been changed in the following manner: The original one-step shot-gun has been replaced with a multi-step shot-gun. Initially, a starting point is prescribed by specifying values for the five variables, and a sequence of 432 points in the vicinity of the starting point are checked. The point of lowest cost with a DELPA greater than $DELPA_{min}$ is taken, and becomes the initializing point for the next procedure. Holding four of the five variables constant, the cost is evaluated, taking extreme values of the fifth variable both above and below the values searched by the initial shot-gun. This is sequentially done for each variable. If the minimum of one of these "extended" valued costs is at least 1% less than the minimum cost as calculated in the initial shot-gun, this point becomes the prescribed point, and a new initial shot-gun is performed about this point. If no new minimum is found by checking the extremes, the minimum cost point determined by the initial shot-gun becomes the initial point for a second, "finer", shot-gun check. This second shot-gun search is "finer" in that the points checked are closer to the "initializing point" than in the "coarse" shot-gun. This is the so-called "double shot-gun" method.

This double shot-gun method is pictured in Figure IV-5 for a two-variable case. Point 1 is the initial prescribed starting point. The first shot-gun search investigates those points designated as circles, and yields point 2 as the minimum cost point. Extreme values, the diamonds, are then checked for lower cost points. Finding none in this case, a "finer"

FIGURE IV-5
DOUBLE SHOT-GUN INITIALIZING METHOD

shot-gun pattern is searched about point 2, yielding 3 as a lower cost point. Point 3 then becomes the starting point for the search procedure as described in Figure IV-2.

Expanding the process from two variables to five is like working in six-dimensional space rather than three-dimensional: pictorially clumsy, but not conceptually difficult. It merely entails sequentially holding all variables but one constant, while finding the planar minima, and repeating the process a given number of times. If an imposed restriction is encountered, the variable is allowed to change simultaneously with, in turn, each of the other variables.

Once the tower has been designed for a given ambient air temperature, a subroutine is called which evaluates the tower's performance at temperatures representative of the yearly fluctuation. For this, a fixed cooler configuration, boiler capacity, and pump and fan power are assumed. This fixes the values of CWARA, AFRON, and WLRAT. The remaining values, T1 and RANGE, are a function of the ambient air temperature, both increasing with ambient air temperature. T1 and RANGE are determined by iterative procedures for a given turbine performance map and the fixed heat exchanger geometry. The cost of operation is then calculated for this ambient temperature and multiplied by the percent of the year the temperature is experienced. The sum of these partial costs is the total yearly operating cost.

Program Assumptions

1. Turbine performance varies with outlet temperature in accordance with General Electric data incorporated in "Estimated Turbine Generation Heat Rate Variation with Elevated Pressures."

2. Heat exchanger is a single pass cross-flow type with the air side unmixed.

3. Tube walls are thin, i.e. they contribute negligible thermal resistance in comparison to the fluid film coefficients.

4. Air side pressure drop is to be kept above some minimal value (4 lbf/ft^2).

5. A percentage of the fan velocity energy is recovered (50%).

6. The yearly cost of operation is composed of the sum of:

- the interest on the capital costs, using a fixed charge rate (15.8%)
- the cost of operating the required fans and pumps
- the cost of additional fuel required due to a decreased thermal efficiency
- maintenance costs
- the cost of replacing lost capacity (15 mills/kWh)

7. Capital costs are broken down into:

- cost of the heat exchanger surface and headers (taken from graph in Heat Exchanger Design, by A.P. Fraas and M. Necati Ozisik) *
- cost of motors for the fans and pumps (\$30/hp)
- cost of the fan blades ($\$7.5/\text{ft}^2$ area covered)
- cost of increased boiler capacity required over that of base capacity due to a lower thermal efficiency (\$45/kW)

8. Maintenance cost is expressed as a certain percent of the capital cost (1.09%)

All cost data and the values of the imposed restriction are required as input parameters and can be readily changed. A breakdown of the input data values and sources is listed in Table IV-1.

*Fraas, A.P., and Ozisik, M. N., Heat Exchanger Design, John Wiley and Sons, Inc., New York, 1965.

Table IY-1Values for Input Data

<u>INPUT</u>	<u>SOURCE</u>	<u>VALUE</u>
Plant Size	Assumed	1,000 MW
Fuel Cost	Con Edison	\$.50/Million BTU's
Replacement Power Costs	Con Edison	15 Mills/KW-Hr
Fixed Charge Rate	Con Edison	15.8% of capital investment
Maintenance Cost of cooling systems	Derived from Con Edison Data	1.09% of capital investment
Boiler Costs	Babcock & Wilcox	\$45/KW
Electric Motor Costs	Local Suppliers	\$30/HP
Fan Propeller Costs	Industrial Air Marley Company	\$7.50/ft ²
Fan Air Velocity	Industrial Air Marley Company	150 ft/sec
Tube Costs	Frass & Ozisik Heat Exchanger Design approx. values: \$.90/ft ² (based on total H.T. area)	Curve fit
Capacity Factor	Assumed	75%
Thermal Efficiency	Assumed (for plant Operating with 3.5" Hg back press)	40%
Fan and Pump Efficiency	Assumed	85%
Temperature Data	U.S. Weather Bureau 10 year, August for J.F.K. Airport	

CCOS = 0.13245E 08 TCOS = 0.76855E 00 RANGE = 4.938 EFF = 0.6965 AFRON = 0.13264E 06 WLRAT = 294.677
REWAT = 40597. REAIR = 5951. N = 10 NW = 24366 HRFAC = 1.0327 W3 = 0.10933E 10 W4 = 0.52708E 09
1 U = 12.270 DELPW = 65.993 DELPA = 4.000 PPOW = 515.634 FPOW = 16354.852 CBOIL = 0.14723E 07
OCOS = 0.25306E 00 COSM = 0.21974E-01 DELFC = 0.13959E 00 T4 = 117.678 CWARA = 8.6429 BEDIA = 0.0
VAIR = 23.515 VWAT = 3.517 HAIR = 14.030 HWAT = 871.699 WIDTH = 5196.152 LENGTH = 21.216 T1 = 136.272

SUBROUTINE DATA

HRFAC= 1.1021 PGEN= 937.028 CSTCL= 0.94459E 00 DELFC= 0.48017E 00
RANGE= 5.134 TCOS= 0.20537E 01 PCST= 0.12322E-02 T11=160.711 TA= 97.00

SUBROUTINE DATA

HRFAC= 1.0827 PGEN= 953.834 CSTCL= 0.69248E 00 DELFC= 0.38201E 00
RANGE= 5.082 TCOS= 0.17034E 01 PCST= 0.54510E-02 T11=155.060 TA= 92.00

SUBROUTINE DATA

HRFAC= 1.0626 PGEN= 971.847 CSTCL= 0.42229E 00 DELFC= 0.28395E 00
RANGE= 5.026 TCOS= 0.13352E 01 PCST= 0.14687E-01 T11=149.362 TA= 87.00

SUBROUTINE DATA

HRFAC= 1.0490 PGEN= 984.507 CSTCL= 0.23240E 00 DELFC= 0.21915E 00
RANGE= 4.986 TCOS= 0.10805E 01 PCST= 0.32415E-01 T11=143.872 TA= 82.00

SUBROUTINE DATA

HRFAC= 1.0374 PGEN= 995.525 CSTCL= 0.67119E-01 DELFC= 0.16534E 00
RANGE= 4.952 TCOS= 0.86142E 00 PCST= 0.59438E-01 T11=138.445 TA= 77.00

SUBROUTINE DATA

HRFAC= 1.0277 PGEN=1004.838 CSTCL= 0.0 DELFC= 0.12166E 00
RANGE= 4.923 TCOS= 0.75062E 00 PCST= 0.79565E-01 T11=133.084 TA= 72.00

SUBROUTINE DATA

HRFAC= 1.0200 PGEN=1012.503 CSTCL= 0.0 DELFC= 0.86877E-01
RANGE= 4.899 TCOS= 0.71584E 00 PCST= 0.71584E-01 T11=127.788 TA= 67.00

SUBROUTINE DATA

HRFAC= 1.0127 PGEN=1019.738 CSTCL= 0.0 DELFC= 0.54995E-01
RANGE= 4.876 TCOS= 0.68396E 00 PCST= 0.46523E 00 T11=122.507 TA= 62.00

ACOS= 0.72960E 00

Computer Output

On page 35 is a sample of computer output. Labeled are (1) output from the main design procedure, where the tower is minimized for a specified ambient design temperature, and (2) the output of the subroutine where the performance of the tower is evaluated at varying ambient temperatures.

Terms are defined as follows:

AFRON: the air side frontal area of the heat exchanger (ft^2)

BEDIA: if a packed bed heat exchanger is used, the diameter of the balls comprising the bed (in)

CBOIL: cost of increased boiler capacity (\$/kW)

CCOS: capital cost (\$)

COSM: maintenance cost (\$/kWh)

CSTCL: cost of replacing lost capacity (mills/kWh)

CWARA: ratio of the heat capacity of water to that of air

DELFC: additional fuel cost due to change in thermal efficiency (mills/kWh)

DELPA: air side pressure drop through the heat exchanger (lb/ft^2)

EFF: heat exchanger effectiveness

FPOW: required fan power (kW)

HAIR: air side heat transfer coefficient ($\text{Btu}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$)

HRFAC: ratio of the heat rate of the designed tower to the assumed base heat rate

HWAT: water side heat transfer coefficient ($\text{Btu}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$)

LENGTH: heat exchanger tube length (ft)

N: number of tubes parallel to direction of air flow
 NW: number of tubes perpendicular to direction of air flow
 OCOS: operating cost (mills/kWh)
 PCST: cost of yearly operation at the specific ambient air temperature, assuming an average yearly occurrence of that temperature (mills/kWh)
 PGEN: heat rejection capabilities at specific ambient air temperature in terms of maximum generatable power (MW)
 PPOW: required pump power (kW)
 RANGE: change in temperature of the water while passing through the heat exchanger (°F)
 REAIR: air side Reynolds number
 REWAT: water side Reynolds number
 TA: ambient air temperature (°F)
 TCOS: in designing program: total cost of yearly operation if operated year-round at the design temperature (mills/kWh)
 in subroutine: total cost of yearly operation of designed heat exchanger if operated year-round at the specific ambient air temperature (mills/kWh)
 T1: turbine outlet temperature (°F)
 T11: turbine outlet temperature (°F) in subroutine
 T4: temperature of air exhausted from exchanger (°F)
 U: overall heat transfer coefficient ($\text{Btu/hr-ft}^2\text{-°F}$)
 VAIR: air velocity (ft/sec)
 VWAT: water velocity (ft/sec)
 WIDTH: AFRON/LENGTH
 WLRAT: WIDTH/LENGTH
 W3: flow rate of water through the exchanger (lb/hr)
 W4: flow rate of air through the exchanger (lb/hr)

Output

Figures IV-6 through IV-15 illustrate output for different heat exchanger configurations. Numbers of the configurations refer to the Kays and London reference number. Capital cost and incremental costs are plotted as functions of design temperature. It should be noted that each point is a minimal optimum point for that design temperature, and hence the true optimum is that design temperature which results in the lowest cost. Here the incremental cost is the important factor, for it reflects the total additional cost of operation (incremental bus bar cost) due to the use of the dry cooling tower. Capital cost merely reflects the additional cost of adding the exchanger to the plant, and is included as one factor in determining the incremental cost.

The curves are fairly smooth and internally consistent. Some of the fluctuation in the curves seems to be a result of discontinuities of the cost surface, caused by design requirements (as, number of tubes must be an integer). What the output curves really represent are envelopes of performance curves. By specific program requirements (as the imposed restriction on the air side pressure drop) possibly part of the envelope is excluded. Computer runs made with an increased minimum pressure drop tend to result in smoother output curves.

Table IV-2 is a listing comparing the optimized values for four different heat exchanger surfaces. The important thing to notice is the wide variation in optimized values with surface geometry.

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Packed Beds

Figure IV-6

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Standard Finned Tubes

Figure IV-7

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Standard Finned Tubes

Figure IV-8

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Standard Finned Tubes

Figure IV-9

Fig. 10-82A

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Standard Finned Tubes

Figure IV-10

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
For Dry Cooling Tower Using Standard Finned Tubes

Figure IV-11

Fig. 10-83

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
 For Dry Cooling Tower Using Plate Finned Tubes
 Figure IV-12

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMPERATURE
 For Dry Cooling Tower Using Plate-Finned Tubes
 Figure IV-13

COOLING TOWER CAPITAL COST VS. DESIGN TEMPERATURE
For Different Minimum Air Side Pressure Drops

Figure IV-14

COOLING TOWER INCREMENTAL COST VS. DESIGN TEMPERATURE
For Different Minimum Air Side Pressure Drops
Figure IV-15

TABLE IV-2

	IV-6 PACKED BED	IV-7 FINNED TUBES	IV-11 FINNED TUBES	IV-13 PLATE FINNED
Power Generated at $T_{AMB} = TD$ (MW)	1000	1000	1000	1000
TD ($^{\circ}F$)	75	75	75	75
ACOS $\left(\frac{\text{mills}}{\text{KW-HR}}\right)$	.68391	.80504	.72960	.56958
TCOS $\left(\frac{\text{mills}}{\text{KW-HR}}\right)$	.72675	.84276	.76855	.54147
CCOS ($\$ \times 10^{-6}$)	13.713	16.359	14.717	8.9812
T1 ($^{\circ}F$)	139.3	137.469	136.3	131.635
AFRON ($\text{ft}^2 \times 10^{-6}$)	.34394	.16560	.13264	.15434
RANGE ($^{\circ}F$)	5.433	6.044	4.938	4.511
CWARA	9.9991	7.0103	8.6429	10.2743
WLRAT	685.1	477.2	294.7	1050.7
LENGTH (ft)	22.406	18.628	21.216	12.120
N (tubes)	3	5	10	6
NW (tubes)	73679	68514	24366	101876
W3 $\left(\frac{\text{lbm}}{\text{hr}} \times 10^{-9}\right)$	1.0203	.89661	1.0933	1.1814
W4 $\left(\frac{\text{lbm}}{\text{hr}} \times 10^{-9}\right)$	.42515	.53292	.52708	.47912

Comments on Computer Output

In interpreting the output of the computer program, several items should be noted.

In general, the minimizing cost curve (ACOS) is relatively flat between the TD values of 70 and 80°F. Hence, in this region, cost is not a strong function of design temperature, and any design temperature in this range may be used without significant cost increases.

Heat exchanger cost data was derived from Heat Exchanger Design by Arthur Frass, and is based on a cost per water side area. This data is expressly for assembled finned tubes and is somewhat dated, as it was published in 1965, but due to a lack of any better information this data was used for all the heat exchanger configurations. Therefore, the output needs further interpretation, depending on the specific heat exchanger configuration. Listed below are the configuration, with the probable cost deviations from that calculated by the computer:

Finned Tube:	Fairly accurate
Plate Fin:	Evaluated costs most likely underestimated. Plate fins would probably require higher assembly costs.
Packed Bed:	The actual cost per water side area is probably lower than the cost used, but because of increased weight, the cost of the supporting structure will probably be higher than for the other configurations.

What is needed is better cost data for specific heat exchangers. Some concrete cost data has recently been acquired, and will be used in future runs.

For comparison purposes, other program assumptions need to be altered. These include the incorporation of a multiple pass heat exchanger and the evaluation of a natural draft system.

Figures IV-14 and IV-15 are examples of how the costs vary with a change in the minimum required air side pressure drop through the exchanger. Note with an increase in required pressure drop, the capital costs decrease and the incremental costs increase. Capital costs decrease because of a savings in the structural compactness of the exchanger; the frontal area decreases and the depth increases. Incremental costs increase because of increased operating costs, namely the greater fan power required to create the pressure differential.

Flow Diagrams

The flow diagrams of the program and its subroutines are shown in Appendix A. Dotted lines indicate either (1) inputs required by the program or subroutine; or (2) assumptions on performance which are built into the program or subroutine. The required input may be information read in as data, or information generated by another subroutine. Brackets indicate the source of generated information. Built-in assumptions, such as turbine performance vs. back pressure, may be readily changed by modifying the appropriate card in the program or subroutine.

Program Listing

A listing of the computer program appears as Appendix C. It is in Fortran IV and is compatible with the IBM 370/155 system.

V. PERIODIC HEAT EXCHANGERS

In Chapters III and IV several novel heat exchanger configurations were examined. These configurations have heat transfer surfaces which remain at a constant temperature once steady state conditions are attained. In periodic heat exchangers solid material moves continually from the hot stream to the cold stream. The heat is exchanged from the hot stream to the solid, and from the solid to the cold stream. The hot and cold streams are physically separated, and the solid material transports the energy from the hot to the cold stream. The solid may be in loose form or in a rigid core or matrix. The feasibility of periodic heat exchangers for dry cooling towers will be examined in this chapter.

A. Liquid Coupled Regenerative Cycle

To be completely general, the circulating material in a periodic exchanger may be a fluid rather than a solid. However, this possibility was eliminated because there is no way to completely eliminate vaporization and entrainment of small drops of the fluid on the air side with the consequent make-up costs and adverse environmental effects. For example, in a 1000 MW plant, a 0.1% loss of fluid per cycle results in a loss of more than one million pounds per day. For fluids other than water, this loss rate is prohibitive. If water were used, the system would be essentially a wet cooling tower. Vaporization or entrainment can be reduced if the liquid particles are large. However, for good heat transfer from the particles to the air, precisely the opposite is desirable: small fluid elements with a large surface-to-volume ratio. The detrimental effects of large particle size will be illustrated in a later section.

B. Rigid Core Periodic Exchangers

In a rigid core periodic exchanger, the solid material is rigidly attached to a frame which rotates so that the core passes alternately through the hot and cold streams. Alternatively, the core is stationary, and the hot and the cold fluid periodically pass through the core, with the flow regulated by valves in the flow passages. The heat exchanger analysis for the rotating and the valved types is identical; for dry towers the rotary appears more feasible because of the high flow rates involved.

Design Calculations: Indirect System

Design calculations were made for an indirect condensing system where the vapor from the turbine is condensed in a direct contact condenser and hot water from the condenser is air-cooled in a rotating periodic exchanger and returned to the condenser. A typical matrix core geometry with triangular flow passages was used for the exchanger.*

The values assumed for the design parameters were:

$$\text{Ratio of Thermal Capacity} = \frac{(\dot{m} C_p)_{\text{air}}}{(\dot{m} C_p)_{\text{water}}} = 0.50 \quad (\text{V-1})$$

$$\text{Effectiveness} = \frac{(\Delta T)_{\text{air}}}{T_{\text{water,in}} - T_{\text{air,in}}} = 0.50 \quad (\text{V-2})$$

$$\text{Generated Power} = 1000 \text{ MW}$$

$$\text{Power Plant Thermal Efficiency} = 36\%$$

$$T_{\text{water,in}} = T_1 = 140^\circ\text{F}$$

* Figure 7-11, Kays and London, Ibid. The data is almost identical to that supplied by Corning Glass Works for their ceramic matrix material, Cercor.

$$T_{\text{air,in}} = T_D = 80^\circ\text{F}$$

$$\Delta T_{\text{water}} = \text{RANGE} = 15^\circ\text{F}$$

$$\begin{aligned} \text{Matrix Capacity Rate} &= C_r \\ &= \frac{\text{Cycles}}{\text{Hour}} (\text{matrix mass}) (\text{specific heat of solid}) \end{aligned} \quad (\text{V-3})$$

For all designs it was assumed that

$$C_r = (\dot{m} C_p)_{\text{air}}$$

(The cycles per hour are the number of hot-cold-hot cycles each element of matrix material experiences per hour.)

These design values were chosen as typical; they do not necessarily represent the optimum performance point of the periodic heat exchanger.

Table V-1 presents a comparison of several periodic heat exchangers with a conventional finned tube heat exchanger. The finned tube data was taken from the results of Chapter IV. The range and initial temperature difference of the finned tube and the periodic exchangers are the same, and the results are for a 1000 MW plant.

The second periodic exchanger listed in Table V-1 has a 300% higher air velocity than the first exchanger and thus a reduced air side frontal area. The large fan power for the final periodic design could be reduced by increasing the frontal area.

The quantity A/Vol is the ratio of heat transfer area to volume for the matrix core. Values near $1000 \text{ ft}^2/\text{ft}^3$ are typical for regenerators. Lower values of A/Vol reduce the water carryover as discussed below.

The cost of the regenerator core is based on estimates given to the authors.* The total regenerator costs are much higher than the core cost

* A. L. London, Stanford University, personal communication.
C. Shay, Corning Glass Works, personal communication.

because they include the costs of the supporting structure for the core, a mechanism to rotate the core, and the water distribution system. The costs are less than a conventional heat exchanger since small tubes to carry the water, tube attachments to headers, tube supports, etc. are all unnecessary.

Possible Regenerator Geometry

Figure V-1 shows a sketch of a possible configuration for a regenerative heat exchanger. The sketch shows a direct system where the steam is directly condensed in the exchanger. For an indirect system the steam inlets would be replaced by hot water inlets. The core is cold when hot water is forced through it. The core increases in temperature while the water is cooled. After the water drains from the core, the core is exposed to the air stream which re-cools it. A number of identical water and air inlets are provided to increase the number of cycles per hour of the core for a given rotational speed. The number of cycles per hour must be kept within practical limits; the period of one cycle should be of the same order of magnitude as the thermal time constant of the core material. For the regenerative exchanger listed first in Table V-1, with an annular cross section, an O.D. of 300 ft., and steam inlets 8 ft. apart, the angular velocity of the core is twelve revolutions per hour. If the core inner diameter is 100 ft., twelve such cores are needed for a 1000 MW plant. Using the second regenerator listed in Table V-1, only four cores are needed for the same size plant. If the cores are supported on steel wheels which ride a steel track, the power to turn each core is less than 10 KW.

MODULAR UNIT

LARGER UNIT

PERIODIC HEAT EXCHANGER
WITH NON-WETTING SURFACE

Water and Air Carryover

Some of the advantages of a regenerative heat exchanger over a conventional heat exchanger are compactness and lower cost. However, regenerators also have disadvantages. The water may pick up impurities; moving elements in the exchanger increase the mechanical complexity; air will be trapped in the core and transferred to the water, necessitating the installation of a de-aerator before the boiler. Another serious problem is the carryover of water to the air side. If the amount of water carried over is appreciable, a significant portion will be entrained or evaporated by the air, causing a high rate of water consumption, and the unit may become simply an expensive evaporative cooling system.

On Table V-1, the maximum percentage of the water flowing through the exchanger which would be carried over to the air side is given. This value was calculated assuming that the water wets the core material and that the film thickness is equivalent to the thickness of a water film on a flat plate which is withdrawn from a stationary pool of water. * As the area-to-volume ratio of the core decreases, the weight of the core is increased for the same heat transfer area. For higher core weights and a constant matrix capacity rate, C_r , the cycles per hour of the core decreases (see eq. V-3). With a constant wetted area and a decrease in the number of cycles per hour, the carryover decreases.

Mass transfer calculations were performed for the design regenerators to find the fraction of liquid carried over to the air side which would evaporate. If the entire heat transfer area was covered with a water film and dry air entered the exchanger, the rate of evaporation of the water

* Levich, V. G., Physicochemical Hydrodynamics, Prentice-Hall, 1962, pp. 674-683.

film would be approximately equal to the boiler flow rate. Evaporative cooling alone would account for almost all of the heat rejection from the power plants. Samples of Corning Glass Works' ceramic matrix materials will be tested to substantiate the calculations of liquid carryover.

Clearly the carryover must be drastically reduced for this design to be feasible for a dry cooling tower. Carryover per unit time can be decreased by using a core with a smaller heat transfer area to volume ratio or by increasing the density of the core material. These steps will reduce carryover somewhat but will increase the cost and size of the exchanger, nullifying the advantages of periodic exchangers over conventional exchangers. The carryover may also be reduced by using a core with a non-wetting surface. The amount of liquid in the form of drops on a non-wetting surface may be larger than the amount of liquid in a film, but the drops can be blown off the surface before it contacts the air, and the surface area of the drops is much smaller than the total heat transfer area of the core.

Rigid Core Regenerators - Direct System

In the direct system, steam from the turbine exhaust is condensed directly in the heat exchanger. The cost differences of direct versus indirect systems, exclusive of the heat exchanger, have already been adequately examined * and will not be considered here. Rather a comparison of the merits of a direct versus an indirect periodic regenerator will be considered.

The steam condenses directly on the cold surface of the core, raising the temperature of the core material. Space for condensate run-off is pro-

* R. W. Beck and Associates, "Engineering Studies of Thermal-Electric Generating Plants Equipped with Dry-Cooling Towers," prepared for Consolidated Edison Company of New York, Inc., March 1971.

vided, and the warm core is air-cooled. Here, as in the indirect system, carryover of air and water should be the major problem. The air which is carried over into the steam side acts as a non-condensable gas, reducing the heat transfer rate during condensation. However, the resistance to heat transfer on the air side should still be more significant than on the steam side. In addition, since the steam is at sub-atmospheric pressure, seals must be provided to prevent a large leakage of air into the steam passages.

Since the air flow rate is so much larger than the steam flow rate, the frontal area for steam flow is small, and the equivalent power for pumping steam through the core is small compared to the air side values. For a steam velocity of 1000 ft/sec, the pressure drop would be equivalent to a pumping power of 2540 KW for the last design listed on Table V-1. High steam velocities would reduce the effects of the non-condensable air on the steam side and remove excess liquid from the surface before the surface was exposed to the air. Since it is impossible to maintain high steam velocities throughout the length of the flow passage if all of the steam is condensed, high velocity air jets can also be used to blow off the large drops.

If the core surface were non-wetting or if a non-wetting promoter were introduced into the steam, the condensate would not form a continuous film; rather, discrete drops would form. For dropwise condensation, most of the surface area is covered by very small drops, while most of the liquid is contained in a comparatively few large drops.* Calculations for

* Hunt, Andrew, M. S. Thesis, Dept. of Mechanical Engineering, MIT, June 1971.

the designs given in Table V-1 show that in the time the surface is exposed to the cooling air, the small drops are completely evaporated, but only a small fraction of the volume of the large drops is evaporated. Since most of the condensate is contained in the large drops, only about 15% of the total condensate is evaporated. The evaporation can be further reduced if high velocity air is used to blow the large drops off the surface. If drops of 0.02 inch diameter and larger can be blown off, the evaporation rate on the air side can be cut to approximately ten percent of the condensate flow rate. In contrast, a wet tower has an evaporation rate close to 100% of the condensate flow rate.

C. Periodic Heat Exchangers with Heat Transfer Material in Loose Form

In the previous paragraphs, periodic exchangers with cores made of sections or rigid core materials were considered. Loose particles can also be used to exchange heat between hot and cold streams. The particles are cooled by air in one tower and transported to a second tower where they cool water or condense steam. Disadvantages of this method as compared to a rigid matrix are the lack of precise control of the geometry of the particles as they fall, and the difficulty of transporting the particles from tower to tower. If the particles are denser than the fluid in the tower, they drop downward through the fluid while the fluid flows upward.

Table V-2 summarizes the calculations of a drop tower with aluminum particles falling through air. For each particle size the terminal velocity was found by equating the air drag forces on the particle to the gravity forces:

Table V - 1

Heat Exchangers for Dry Cooling Tower, 1000 Mw Plant

Heat Exchanger	Pressure Drop		Pumping Power		Frontal Area Air Side	Matrix* Core Weight	Capital** Cost of Exchanger	Maximum Water*** Carryover, Percent of Water Flow Rate	
	ΔP_{air}	ΔP_{water}	P_{air}	P_{water}					
	lb_f/ft^2	lb_f/ft^2	KW	KW	ft^2	lb_m	Dollars	%	
Computer Design Calculation Finned tube, Fig. IV-10	4.00	7.86	18,749	544	1.5×10^5	--	15.8×10^6	---	61
Periodic $A/(Vol) =$ $840 \text{ ft}^2/\text{ft}^3$	1.36	60	6,600	525	7.38×10^5	2.5×10^6	8.1×10^6	25	
Periodic $A/(Vol) =$ $840 \text{ ft}^2/\text{ft}^3$	9.4	60	45,600	525	2.46×10^5	19×10^5	8.6×10^6	19	
Periodic $A/(Vol) =$ $84 \text{ ft}^2/\text{ft}^3$	6.33	60	30,800	525	2.46×10^5	35.2×10^6	12.2×10^6	2.5	

* Assuming water-side flow plus Drain Volume = Air Side Volume

** Assume Total Regenerator Cost = $\$0.50 \text{ ft}^2$ + Fan Costs + Interest Cost (Exclude Piping Costs)

*** See Remarks in Text.

Table V-2

LOOSE PARTICLES - DROP TIMES AND DISTANCES IN AIR

Particle Diameter (inches)	Air Carryover Velocity (ft/sec)	Frontal Area for Air Flow at 2/3 Carryover Velocity (ft ²)	Particle Temperature Drop in Tower (°F)	Avg Particle Separation Distance (Inches)	Tower Height (ft)	Power to Raise Particles to Tower Height (MgW)
10^{-2}	6.1	9.25×10^5	30	0.10	38	12.5
10^{-1}	40	1.38×10^5	15	0.80	124	94.0

$$F_{\text{drag}} = C_d \cdot \rho_{\text{air}} \cdot v^2 \cdot \frac{\pi D^2}{4} = F_{\text{gravity}} = g \cdot \rho_{\text{particle}} \cdot \frac{\pi D^3}{8} \quad (\text{V-4})$$

The drag forces are proportional to the cross-sectional area of the particles, and the gravity forces are proportional to the volume of the particles. Therefore, the terminal velocity increases as the particle diameter increases.

$$v = \sqrt{\frac{\rho_{\text{particle}} g D}{2 \rho_{\text{air}} C_d}} \quad (\text{V-5})$$

To preclude carrying the particles out of the tower with the air stream, the upward air velocity must be less than the terminal velocity. The upward air velocity was assumed to be two-thirds the terminal velocity of a particle. Therefore, small diameter particles necessitate low air velocities and a large cross-sectional area of the tower for a given air flow rate.

The time of exposure of the particle to the air is the ratio of the tower height to the velocity of the particle relative to the air. The exposure time must be long enough to allow the particle to be cooled. Large particles have a smaller surface-area-to-volume ratio and require a long cooling time. Since large particles also have high velocities, the cooling tower must be high, resulting in a prohibitively large power to lift the particles to the top of the tower.

There must be provision for a particle separator in the air outlet since some of the particles will break in the drop tower, and smaller particles will be entrained in the air stream and carried out. The loose particle regenerator also suffers from the disadvantages of the rigid core, namely air and water carryover.

Parallel Flow Particle - Air Tower

To eliminate the power necessary to lift particles to the top of the tower, a parallel flow particle-air tower was suggested. Hot particles are injected into cold air at the base of a tower. The air is heated by the particles and rises by natural convection, entraining the particles, which continue to heat the rising air column. The cool particles are separated from the air at the top of the tower and are transferred to another tower containing vapor. The cool particles fall through the vapor and condense it.

The mass of particles which can be carried per mass of air can be estimated by a simple analysis. Assume the air in the tower is at a constant temperature equal to its maximum value, $T_{\max}$. This assumption will give an upper limit on the mass of particles which can be carried upward. Neglecting frictional losses and assuming that the tower has a constant cross section and that steady flow exists, the momentum equation can be written for the tower shown in Fig. V-2.

For steady flow and constant cross sectional area, the momentum of the entering and leaving streams are the same, resulting in a simple force balance between the weight of the gas and particles in the tower and the force due to the pressure difference between the inlet and the outlet.

$$(M_{\text{part}} + M_{\text{air}}) g = (p_1 - p_2) A \quad (\text{V-4})$$

The pressure difference is set by the hydrostatic head of the ambient air.

$$p_1 - p_2 = \rho_{\text{AMB}} g L \quad (\text{V-5})$$

ρ_{AMB} represents the average density of the ambient air.

FORCE BALANCE FOR TOWER

FIGURE V-2

$$(p_1 - p_2)A = \rho_{AMB} g L A \quad (V-6)$$

For the air in the tower, assuming the particles occupy a small percentage of the volume,

$$M_{air} \approx \rho(T_{MAX}) L A \quad (V-7)$$

$$(p_1 - p_2)A = \frac{\rho_{AMB}}{\rho(T_{MAX})} M_{air} g \quad (V-8)$$

Substituting eq. (8) into eq. (4),

$$(M_{part} + M_{air})g = \frac{\rho_{AMB}}{\rho(T_{MAX})} M_{air} g \quad (V-9)$$

$$\frac{M_{part}}{M_{air}} = \frac{\rho_{AMB}}{\rho(T_{MAX})} - 1 \quad (V-10)$$

The upper limit on the density ratio can be found by using the perfect gas relationships assuming the pressure is the same inside and outside the tower at corresponding elevations.

$$\frac{M_{part}}{M_{air}} = \frac{T_{MAX}}{T_{AMB}} - 1 \quad (V-11)$$

The temperatures must be given in an absolute scale. Assuming that the air changes at most by 50°F and that the ambient temperature is 60°F,

$$\frac{M_{part}}{M_{air}} \leq \frac{110 + 460}{60 + 460} - 1 = 0.10 \quad (V-12)$$

Since in the limit the particles can move only as fast as the air, the mass ratio must be less than or equal to the ratio of mass flow rates of particles to air. From a simple energy balance:

$$\dot{m}_{\text{part}} (C_p)_{\text{part}} \Delta T_{\text{part}} = \dot{m}_{\text{air}} (C_p)_{\text{air}} \Delta T_{\text{air}} \quad (\text{V-13})$$

where $\dot{m}$ is the mass flow rate and C_p is the specific heat.

$$\frac{\dot{m}_{\text{part}}}{\dot{m}_{\text{air}}} = \frac{(C_p)_{\text{air}}}{(C_p)_{\text{part}}} \frac{\Delta T_{\text{air}}}{\Delta T_{\text{part}}} \quad (\text{V-14})$$

For a 100°F change in the particle temperature, requiring at least a 100°F difference between the turbine exhaust temperature and the ambient air temperature, the air temperature increase in the tower is:

$$\Delta T_{\text{air}} = (100) \frac{\dot{m}_{\text{part}}}{\dot{m}_{\text{air}}} \frac{(C_p)_{\text{part}}}{(C_p)_{\text{air}}} \quad (\text{V-15})$$

With eq. (V-12) and the assumption that the specific heat ratio is unity,

$$\Delta T_{\text{air}} < (100) (0.1) = 10^\circ\text{F} \quad (\text{V-16})$$

Since the product $(\dot{m} C_p \Delta T)_{\text{air}}$ must remain constant for a given heat rejection from the power plant, a very low air ΔT results in an impractically high air flow rate and tower cross-sectional area.

Use of Loose Particles to Condense Steam Directly

The particles cooled in the air tower may be used to directly condense the steam by dropping them in another tower with steam flowing counter-currently. Although the steam flow rate is considerably less than the air flow rate, a large cross-sectional area is needed for the tower to accommodate all the particles. Notice in Table V-2 the small average distance between particles in the air cooling tower.

An alternative arrangement is to transport the particles to a location close to the turbine outlet to condense the steam. This scheme reduces the distance the steam must be transported, but the particle

transportation becomes unwieldy. Using five foot wide conveyor belts moving at the highest recommended speed*, 100 belts would be needed to transport the particles necessary for a 1000 MW plant. The power to move the particles 400 feet (round trip) on the belts is 22 MW.

Periodic Exchangers - Conclusions

Periodic exchangers with loose particles or liquid serving as the heat exchange medium appear impractical. Use of a rigid core matrix may be feasible if the air and water carryover problem can be solved. The periodic exchanger has the potential of being much less expensive than conventional finned tube exchangers. Preliminary experimental results of water carryover and evaporation on a non-wetting surface are given in Chapter V.

* Marks' Standard Handbook for Mechanical Engineers, Seventh Edition, 1967.

VI. Periodic Heat Exchangers -- Experimental

Experimental work testing the feasibility of the use of a periodic surface was initiated. Preliminary work was done on a test apparatus schematically illustrated in Figure VI-1 . The test plate is coated with a non-wetting surface (teflon) to promote dropwise condensation, and is equipped with an aluminum foil "rain gutter" to collect condensate which runs off the surface. Hence, weighing the plate will also weigh all the water which was ever condensed on the surface.

The procedure is to condense steam on the test plate, then flush the plate with air. By measuring the weight-time and temperature-time history of the plate, the percent of heat transferred by evaporation can be determined.

Initial tests using a thin test plate yielded results typified by Figure VI-2 . This information can also be plotted as weight loss vs. temperature change, as shown in Figure VI-3 . The two runs plotted in Figure VI-3 illustrate the band of uncertainty for the runs.

From Figure VI-3 the amount of cooling done by evaporation may be calculated. Since all the heat was initially transferred to the test plate by condensation, and all the condensate is still on the plate or in the rain gutter, the percent of condensate evaporated directly reflects the percent of cooling by evaporation. Hence in Figure VI-3 for a 50% temperature recovery, one run has had about 25% evaporative cooling, while the other had about 12% evaporative cooling. The amount of water evaporated in these runs (30-50% of boiler flow rate) was indeed less than that which would be evaporated in a conventional wet tower (about 100% of boiler flow rate) but greater than had been expected (5-25% of boiler flow rate).

It was postulated that the use of a thicker plate (i.e., a greater heat capacity) would result in the formation of larger condensate drops. Larger drops are less susceptible to evaporation in a two-fold manner: (1) they are more likely to run off a non-wetting surface and therefore not be subject to evaporation; and (2) they have a smaller surface-area-to-volume ratio.

A thicker plate was tried, but due to the thicker plate's increased time constant (time necessary to cool the plate) any advantage of larger drops was nullified. No measurable decrease in evaporation was evident. In an attempt to overcome this, a short burst of high-velocity steam was used, trying to blow condensate drops off the surface before cooling the plate. However, the lab steam source was supplying slightly wet steam, resulting in entrained water being deposited on the surface. The combination of a steam separator placed in the high pressure line and an isenthalpic expansion is now producing slightly superheated steam.

Measurements made with a burst of high velocity steam are delicate. The vertical component of the drag forces on the plate due to the steam or air jets are of the same order of magnitude as the measured condensate weight. This vertical force is measured in the weighing process, and is recorded as condensed water weight. To counteract this, the exhaust system must be carefully balanced with the jet system. Further, if all steam is not drawn out of the exhaust, steam escapes through the top, and condensation occurs on the members supporting the test plate. Again, this condensate is improperly recorded as test plate condensate.

Further refinements are to be made to the test apparatus. This includes further studies with bursts of high velocity fluids (steam or air) to strip the test surface of condensate, and tests using hot water as the high temperature fluid.

TEST APPARATUS SCHEMATIC
Figure VI-1

Fig. VI-2

PLATE TEMPERATURE AND CONDENSATE WEIGHT VS. TIME

For Non-Wetting Periodic Surface

Fig. VI-3

VII. PACKED BED HEAT EXCHANGERS

The results of Chapters III and IV indicate that one promising heat transfer surface is the packed bed. A packed bed heat exchanger has relatively large pipes carrying the fluid to be cooled, surrounded by a packed bed of metal balls. Heat is transferred from the hot fluid through the pipe wall to the balls and from the balls to air which is blown through the bed. The balls act as fins to increase the heat transfer on the air side. For a tube surrounded by five layers of balls, the heat transfer area is fifteen times greater than that for a bare tube. In addition, the hydraulic diameter for the air passages is small, resulting in good heat transfer - friction power characteristics.

The balls should be less expensive to fabricate than fins since no machining of the balls is necessary.

A. Joining Methods

The most critical problem in fabricating packed beds is joining the balls to each other and to the tube wall. The structure must be self-supporting, and the thermal resistance between balls must be small. If the thermal resistance between the balls is large, very little heat will be conducted to the balls one or two rows away from the tube. Finally, the fabrication costs must be low if the overall cost of the packed bed heat exchanger is to be much lower than that of the finned tube heat exchanger.

Four joining techniques have been studied: soldering, welding, coating with zinc, and coating with epoxy. For simplicity, four or five steel balls were joined together in a straight line, and the thermal resistance was

evaluated. The details of the experiment are given in Appendix I of the Interim Report of June 30, 1971. The row of balls was heated at one end, the surface of the balls was air-cooled, and the temperature of each ball was measured.

The average thermal resistance from the center of one ball to the center of the next was calculated from the temperature distribution. The ball-to-ball resistance includes the conduction resistance through one ball plus the resistance at the joint. If the former is negligible, the ball-to-ball resistance represents the joint resistance. Table VII-1 gives the average value of the ball-to-ball thermal resistance calculated from the temperature distribution; the welded configuration serves as a standard. Also shown in Table VII-1 are calculated approximate values of the conduction resistance through a right circular cylinder with a length equal to one ball diameter, and two values of the cross-sectional area of the cylinder, corresponding to a cylinder radius of three-quarters and one-half the ball radius, respectively. The values for the cylinders represent a rough approximation of the ball-to-ball conduction resistance with negligible joint resistance.

In general, the resistance to heat transfer is defined by the expression:

$$\text{Heat Transfer Rate (q)} \equiv \frac{\text{Temperature Difference } (\Delta T)}{\text{Resistance (R)}} \quad (\text{VII-1})$$

$$R = \frac{\Delta T}{q} \quad (\text{VII-2})$$

From the results shown in Table VII-1, the soldered joints and the zinc coatings give approximately the same thermal resistance as the welded joints. The values of the overall ball-to-ball resistance for all three joining methods are approximately the same as the conduction resistance through the right circular cylinder. Therefore the thermal resistance of the joint must be small. The overall ball-to-ball resistance is the sum of the conduction resistance in the ball with a perfectly conducting joint plus the conduction resistance of the joint. Since the former is influenced by the size of the web between balls, it cannot be calculated exactly, and quantitative values of the joint resistance cannot be found from the measured overall resistance.

The high resistance of the epoxy joint is due in part to poor fabrication technique. The epoxy was applied and then cured in an oven. The oven temperature was too high, causing a large amount of run-off and a very narrow web. More accurate results for epoxy joints are given later in this chapter.

The material cost of the solder (50% tin - 50% lead) was estimated to be 9.8 cents per square foot of heat transfer area. For zinc the material cost was estimated to be 1.6 cents per square foot. The difference in cost plus the fact that solder is likely to creep rupture at the temperature level of the heat exchanger makes zinc a logical choice for further investigation.

B. Epoxy Joints

A further study of the use of epoxy joints for packed beds was also made. Two types of epoxy were used: a common epoxy cement (Hysol Epoxi-Patch); and epoxy mixed with graphite filler to increase the thermal

TABLE VII-1

AVERAGE THERMAL RESISTANCE BETWEEN BALLS

Ball Diameter = 1/2 inch

Joining Method	Thermal Resistance (+ Standard Deviation) Between Two Balls (°F-hr/BTU)
Soldered	5.24 $\pm$ 0.64
Zinc Coated	4.33 $\pm$ 2.06
Welded	4.86 $\pm$ 0.52
Epoxy Coated	11.08 $\pm$ 1.55
Calculated:	
Right Circular Cylinder L = 1/2" , R = 3/8"	2.7
Right Circular Cylinder L = 1/2" , R = 1/4"	6.05

conductivity (Stycast 2850KT). The ball-to-ball resistance was found by transient test of a row of balls. The balls were heated in an oven to a uniform temperature. They were removed from the oven and one end was quenched in cold water. The temperature-time history of the balls was recorded, and the resistance was calculated from the measured time constant. For comparison, balls were attached with silver solder, which should give a very small joint resistance. The results are presented in Table VII-2. The results cannot be compared directly with Table VII-1 because the balls used for Table VII-2 were one-quarter inch in diameter, while the balls used for Table VII-1 were one-half inch in diameter. The conduction resistance in the ball should vary inversely with the diameter to the first power.

For these tests with epoxy, the web between the balls was thick, giving better results for the epoxy than those listed in Table VII-1. The high conductivity epoxy appears promising. However, the material cost and production cost of heat transfer surfaces with epoxy must be investigated.

The details of the tests summarized above are given in Appendix 2 of the June 30, 1971 Interim Report.

C. Packed Bed Assembly

Two assemblies of balls were fabricated: one of aluminum balls joined with graphite filled epoxy, and one of steel balls joined by a zinc coating. The balls were assembled around a one-inch tube, a geometry similar to that used in cooling towers. The aluminum balls were used to reduce the overall weight of the packed bed assembly.

TABLE VII-2

AVERAGE THERMAL RESISTANCE BETWEEN BALLS

Ball Diameter = 1/4 inch

<u>Joining Method</u>	<u>Thermal Resistance Between Two Balls (°F-hr/BTU)</u>
Hysol Epoxi-Patch	39.6 $\pm$ 1.9
Stycast 2850KT (high thermal conductivity graphite filled epoxy)	13 $\pm$ 2
Silver Solder	9.6 $\pm$ 0.8
Calculated:	
Right Circular Cylinder L = 1/4" , R = 3/16"	5.4
Right Circular Cylinder L = 1/4" , R = 1/8"	12.1

The assemblies were tested for air side heat transfer and friction drop characteristics. A blower supplied a flow of air through the balls. The upstream and downstream static pressure and temperature of the air was measured along with the air flow rate. A mixture of steam and water flowed through the one inch tube. The flow rate was adjusted so that the calculated heat transfer coefficient from the water to the tube wall was an order of magnitude larger than the measured overall heat transfer coefficient from the water to the air. The overall coefficient is therefore approximately equal to the air side heat transfer coefficient.

The air side heat transfer coefficient and friction power per unit area were calculated from the test data using the surface area of the one inch tube as the reference area. The results are plotted on Figure VII-1 along with the predicted values as explained in Chapter III.

The experimental results are far below the theoretical curve. This is primarily due to a thick coating of zinc or epoxy which tended to plug the clearance spaces between the balls. The air flow was not uniform, and the resistance to air flow was considerably increased. Although the structure appeared mechanically sound, a few joints were observed which had poor ball-to-ball contact, increasing the resistance to heat transfer.

Additional work on epoxy coated aluminum balls is underway, specifically aimed at reducing the plugging problem by modifying the coating procedure.

Figure VII-1

VIII. CONCLUSIONS AND RECOMMENDATIONS

Computer Design Program

The computer design program has proved to be viable and adaptable to many systems, locales, costs, and heat exchanger surfaces. With accurate representation of costs and heat exchanger surface characteristics, the program accurately determines an optimized design.

In future work, more accurate cost data for heat exchanger surfaces should be obtained and used, and the program should be modified to accommodate a natural draft system. The program should be used to evaluate the effects of multipass heat exchangers and different turbine performances. The sensitivity of the system to changes in specific parameters should be studied to identify the key parameters -- those yielding the greatest return for a small change. Finally, an extensive study of different heat exchanger surfaces (finned tube, packed bed, plastics, etc.) should be made with the computer program to find the most desirable surface.

Periodic Heat Exchangers

Liquid or loose-particle periodic material is impractical. However, rigid periodic surfaces are promising if water carry-over can be reduced. The use of non-wetting surfaces has been tested, but additional means of stripping water drops, such as high velocity jets of steam or air or surface alterations (such as grooves) to augment run-off, should be investigated.

Periodic exchangers should be tested in an indirect mode. That is, rather than condensing turbine steam directly on the periodic surface, hot water cooled on the surface will be used to condense the turbine steam.

Packed Bed Heat Exchangers

Initial tests of small packed bed assemblies showed good heat transfer performance. However, in larger fabricated test sections the distribution of bond material varied from a thin coat (poorly bonded balls) to a thick coat (plugged air passages). This variation resulted in non-uniform flows, decreased heat transfer coefficients, and increased pressure drops.

Consequently, a more uniform coating of bond material is needed. This may require the use of less viscous epoxy, different epoxy fillers, or in the zinc-bond case, longer times at higher temperatures for curing. Tests should be conducted with larger test sections or smaller balls to minimize flow abnormalities caused by random ball packing. Small balls, however, may increase the amount of clogging caused by the bond material.

Heat Exchanger Surfaces

Preliminary order-of-magnitude studies show that fluidized beds give high heat transfer rates with modest pressure drops, and suggest that fluidized beds should be examined further.

Plastic surfaces should also be examined to determine if an inexpensive plastic surface can be found which is rigid yet thin enough to reduce conduction resistance across the plastic.

APPENDIX A
FLOW DIAGRAMS

FUNCTION OF SUBROUTINES

SHOT	Tests multiple points. Performs "shot-gun" search.
XTEND	Tests extreme values of minimal cost point, as determined from "coarse" shot-gun
SERCH	Seeks minimum by means of one variable search method.
CHNGE	Seeks minimum when pressure drop restriction encountered by two variable search method.
VARIT	Evaluates system performance and cost, considering average annual temperature fluctuations.
CALC	Calculates cost for a given set of five variables.
STORE	Stores variable and calculated values of minimum cost.

FLOW DIAGRAM OF SUBROUTINE CALLING

MAIN PROGRAM

TD = design temperature

SHOT

XTEND

SERCH

CHNGE

CALC

VARIT

STORE

APPENDIX B
INPUT DATA

DATA COMMON TO ALL PROGRAMS

Plant Size

Fuel Cost

Replacement Power Cost

Additional Boiler Capacity Cost

Pump and Fan Costs

Maintenance Cost

Capacity Factor

Fixed Charge Rate

Base Thermal Efficiency

Pump and Fan Efficiency

Propeller Cost

Air Velocity, Fan

Percent Energy Recovery of Fan Velocity

Minimum Air Side Pressure Drop

Constants for Heat Exchanger Costs

Performance and Physical Dimensions of Heat Exchanger Surface

Constants for Turbine Performance

INPUT DATA

AFTR	airside fin area/total air side area
ALPHA	total air side area/total volume (ft^2/ft^3)
BCOST	boiler cost (\$/KW)
BEDIA	packed bed ball diameter (inches)
C1 - C5	polynomial coefficients for cost data curve fit
CAPF	capacity factor: $\frac{\text{actual yearly power generation}}{\text{maximum possible power generation}}$
CONF	conductivity of fin material (Btu/hr-ft°F)
DEQ	equivalent diameter = 4 rh (ft)
DFIN	fin diameter (inches)
EFFF	fan efficiency
EFFP	pump efficiency
FAC	a constant for a given surface, from friction vs. Reynolds Number graph. For determining air side friction.
FAS	scope of friction vs. Reynolds number graph
FANV	air velocity delivered by fan (ft/sec)
FCOS	fuel cost (¢/MMBTU)
FCR	fixed charge rate (%/100)
HPCST	cost of electric motors (\$/HP)
OD	tube diameter (inches)
PER	yearly maintenance cost/capital investment
PROPC	fan blade cost per area covered ($\$/\text{ft}^2$)
PSIZE	plant size (MW)
PWCOS	replacement power cost (mills/KW-HR)
RECOV	for energy of moving air; $\frac{\text{recoverable energy}}{\text{energy of initiating motion}}$

SAC	a constant for a given surface from Colburn J. Factor vs. Reynolds Number graph. For determining air side h
SAS	slope of Colburn J Factor vs. Reynolds Number graph
SIGMA	free flow area/frontal area for air side
TA(1) - TA(8)	air temperatures - used in evaluating yearly exchanger performance (°F)
TEFF	base thermal efficiency (%/100)
THFIN	fin thickness (inches)
TPER(1)-TPER(8)	fraction of year TA(1) - TA(8) experienced
TPO(1)-TPO(3)	Polynomial coefficients for fit of turbine performance vs. back pressure curve
XD	depth (separation) pitch (inches)
XDEPA	minimum air side pressure drop (psf)
XW	transverse (width) pitch (inches)

APPENDIX C
PROGRAM LISTING

```

    DIMENSION PCST(8),TPER(8),TA(8)
    COMMON VAR(5),VAL(5),VAS(5),TPC(3),TCOS1,TCOS,CCCS1,CCOS,REAI1,
1 REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2 DELPA,PPCW1,PPCW,FPCW1,FPOW,CCCS1,CCCS,CCSM1,CCSM,DEFC1,DEFC,
3 WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,H11,H1,H01,H0,T41,T4
4 EFF1,EFF,QIN1,QIN,CECI1,CBOIL,TEFF,PSIZE,PWCCS,PER,FCOS,FCR,CAPF
    CCMCN BCCST,HPCST,EFFP,EFFF,PRCFC,FANV,      RECCV,C1,C2,C3
1,C4,C5,XW,XD,DEQ,DD,CCNF,AFTR,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2 THFIN,BEDIA,CAF,REWA1,REWAT,TD,QREJ,XDEPA,KCONV
    READ(5,75)(TA(I),I=1,8),(TPER(I),I=1,8)
    READ(5,77)PSIZE,FCOS,PWCCS,BCOST,HPCST,PER,CAPF,FCR
    READ(5,77)TEFF,EFFP,EFFF,PROPC,FANV,      XDEPA,RECCV
    READ(5,77)(TPC(I),I=1,3)
    READ(5,77)C1,C2,C3,C4,C5
    READ(5,77)XW,XD,DEQ,CC,CONF,AFTR,ALPHA,SIGMA
    READ(5,77)FAC,FAS,SAC,SAS,DFIN,THFIN,BEDIA
    WRITE(6,77)PSIZE,FCOS,PWCCS,BCOST,HPCST,PER,CAPF,FCR
    WRITE(6,77)TEFF,EFFP,EFFF,PRCPC,FANV,      XDEPA,RECCV
    WRITE(6,76)C1,C2,C3,C4,C5
    WRITE(6,77)XW,XD,DEQ,CC,CONF,AFTR,ALPHA,SIGMA
    WRITE(6,77)FAC,FAS,SAC,SAS,DFIN,THFIN,BEDIA
75 FORMAT(16F5.4)
76 FORMAT(8E10.3)
77 FORMAT(8F10.4)
    DO 7 IS=1,5
7 VAR(IS)=C.0
    TD=6C.
1 WRITE(6,81)TD
81 FORMAT(' TD USED IN THE FOLLOWING CALCULATIONS IS ='F8.3)
    K=4
    VAS(1)=135.
    VAS(3)=7.

```

```

MAIN0001
MAIN0002
MAIN0003
MAIN0004
MAIN0005
MAIN0006
MAIN0007
MAIN0008
MAIN0009
MAIN0010
MAIN0011
MAIN0012
MAIN0013
MAIN0014
MAIN0015
MAIN0016
MAIN0019
MAIN0020
MAIN0021
MAIN0022
MAIN0023
MAIN0026
MAIN0027
MAIN0028
MAIN0029
MAIN0030
MAIN0031
MAIN0032
MAIN0033
MAIN0034
MAIN0035
MAIN0036

```

```

      VAS(4)=2.E5
      VAS(5)=5C.
4  CALL SHOT(1)
      IF(TCOS1.GT.C.0) GO TC 5
      VAS(4)=VAS(4)/4.
      GO TC 4
5  L=0
      CALL XTEND(L)
      IF(L.EQ.1) GO TC 4
      CALL SHCT(2)
2  CALL SERCH
      WRITE(6,82)CCOS1,TCCS1,VAR(2),EFF1,VAR(4),VAR(5)
82  FORMAT(' CCOS='E15.5,' TCCS='E15.5,' RANGE='F6.3,' EFF='F7.4,
1' AFRON='E15.5,' WLRAT='F8.3)
      WRITE(6,83)REWA1,REA11,N1,NW1,HRFA1,W31,W41
83  FORMAT(' REWA='F9.0,' REAIR='F9.0,' N='I5,' NW='I7,' HRFAC='
1' F7.4,' W3='F15.5,' W4='E15.5)
      WRITE(6,84)U1,DEPW1,DEFA1,PPCW1,FPOW1,CBCI1
84  FORMAT(' U='F10.3,' DELPW='F10.3,' DELPA='F10.3,' PPCW='F10.3,
1' FPCW='F10.3,' CBCIL='E12.5)
      WRITE(6,85)CCCS1,CCSM1,DEFC1,T41,VAR(3),EEDIA
85  FORMAT(' CCOS='E15.5,' CCSM='E15.5,' DELFC='E15.5,' T4='F8.3,
1' CWARA='F7.4,' BEDIA='F5.3)
      WRITE(6,86)VAIR1,VWA11,HCI,H11,WIDT1,ELEN1,VAR(1)
86  FORMAT(' VAIR='F10.3,' VWAT='F10.3,' HAIR='F10.3,' HWAT='F10.3,
1' WIDTH='F10.3,' LENGTH='F10.3,' T1='F7.3)
      K=K-1
      IF(K.GT.0) GO TC 2
      DO 3 I=1,8
3  CALL VARIT(TA(I),TPER(I),PCST(I))
      ACOS=PCST(1)+PCST(2)+PCST(3)+PCST(4)+PCST(5)+PCST(6)+PCST(7)+PCST(
18)
      WRITE(6,87)ACOS
87  FORMAT(' ACOS='E15.5)
      TD=TD+5.
      IF(TC.LE.100.) GO TC 1

```

```

MAIN0037
MAIN0038
MAIN0039
MAIN0040
MAIN0041
MAIN0042
MAIN0043
MAIN0044
MAIN0045
MAIN0046
MAIN0047
MAIN0048
MAIN0049
MAIN0050
MAIN0051
MAIN0052
MAIN0053
MAIN0054
MAIN0055
MAIN0056
MAIN0057
MAIN0058
MAIN0059
MAIN0060
MAIN0061
MAIN0062
MAIN0063
MAIN0064
MAIN0065
MAIN0066
MAIN0067
MAIN0068
MAIN0069
MAIN0070
MAIN0071
MAIN0072

```


6 CONTINUE
STOP
END

MAIN0073
MAIN0074
MAIN0075

```

SUBROUTINE SHOT(L)
  DIMENSION T(3),CWAR(4),RANG(3),AFRO(3),WLRA(4),VAM(5)
  COMMON VAR(5),VAL(5),VAS(5),TPO(3),TCCS1,TCOS,CCCS1,CCOS,REAI1,
1  REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2  DELPA,PPCW1,PPCW,FPCW1,FPOW,OCOS1,OCOS,CCSM1,CUSM,DEFC1,DELC,
3  WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,H11,F1,F01,F0,T41,T4
4  EFF1,EFF,QIN1,QIN,CECI1,CBJIL,TEFF,PSIZE,PWCCS,PER,FCUS,FCR,CAPF
  COMMON BCCST,HPCST,EFFP,EFFF,PRCFC,FANV,      RECOV,C1,C2,C3
1  C4,C5,XW,XD,DEQ,UD,CCNF,AFTF,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2  THFIN,BEDIA,CAF,REWA1,REWAT,TC,QREJ,XDEPA,KCONV
  IF(L.EQ.2) GO TO 31
  TCOS1=0.
  TCOS2=0.
  DO 20 I=1,3
    T(I)=VAS(1)+(I-2)*5.
    IF(T(I).GE.170.) T(I)=170.
20  IF(T(I).LE.100.) T(I)=110.
    DO 21 I=1,4
      CWAR(I)=VAS(2)+(I-2)*2.
21  IF(CWAR(I).LE.0.) CWAR(I)=.5
    DO 22 I=1,4
      WLRA(I)=VAS(5)+(I-2)*10.
22  IF(WLRA(I).LE.0.) WLRA(I)=5.
    AFRO(1)=VAS(4)/2.
    AFRO(2)=VAS(4)
    AFRC(3)=VAS(4)*2.
    CT1=1.
    ECWAR=1.
    CAFRC=1.
    CWLRA=1.
    DO TC 51
31  CT1=VAR(1)-VAM(1)
    ECWAR=VAR(3)-VAM(3)
    CAFRC=VAR(4)-VAM(4)
    CWLRA=VAR(5)-VAM(5)
    IF(CT1.EQ.0.) GO TO 32

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SHOT0001
SHOT0002
SHOT0003
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SHOT0027
SHOT0028
SHOT0029
SHOT0030
SHOT0031
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SHOT0033
SHOT0034
SHOT0035
SHOT0036

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```

      CC 32 I=1,3
32  T(I)=VAR(1)-(I-1)*DT1/2.
      GO TC 35
33  DO 34 I=1,3
      T(I)=VAR(1)+(I-2)*2.
34  IF(T(I).GE.170.) T(I)=170.
35  IF(DCWAR.EQ.C.) GC TC 37
      DO 36 I=1,4
36  CWAR(I)=VAR(3)-(I-1)*[CWAR/3.
      GO TC 39
37  DO 38 I=1,3
38  CWAR(I)=VAR(3)+(I-2)
39  IF(DAFRC.EQ.C.) GO TC 41
      DO 40 I=1,3
40  AFRC(I)=VAR(4)-(I-1)*[AFRC/2.
      GO TO 43
41  CC 42 I=1,3
42  AFRC(I)=VAR(4)+(I-2)*.5E5
43  IF(DWLRA.EQ.C.) GO TC 45
      DO 44 I=1,4
44  WLRA(I)=VAR(5)-(I-1)*DWLRA/3.
      GC TC 51
45  DO 46 I=1,3
46  WLRA(I)=VAR(5)+(I-2)*5.
51  CC 15 IT=1,3
      T1=T(IT)
      F=EXP(17.168-9240./(T1+460.))
      HRFAC=1.+TPC(1)*P+TPC(2)*P**2+TPC(3)*P**3
      TEFFM=TEFF/HRFAC
      QIN=3413.*PSIZE*1000./TEFFM
      GREJ=QIN*(1.-TEFFM)
      LC=4
      IF(DCWAR.EQ.C.) LC=3
      DO 15 IC=1,LC
      CWARA=CWAR(IC)
      IF(CWARA.GE.1.) GO TC 1

```

```

SHOT0037
SHOT0038
SHCT0039
SHCT0040
SHOT0041
SHOT0042
SHOT0043
SHOT0044
SHOT0045
SHOT0046
SHCT0047
SHCT0048
SHOT0049
SHCT0050
SHOT0051
SHOT0052
SHOT0053
SHOT0054
SHOT0055
SHCT0056
SHOT0057
SHCT0058
SHOT0059
SHOT0060
SHOT0061
SHOT0062
SHOT0063
SHOT0064
SHOT0065
SHCT0066
SHOT0067
SHOT0068
SHOT0069
SHOT0070
SHCT0071
SHCT0072

```

```

RANGL=(1.-EXP(-CWARA))*(T1-TD)/CWARA
GO TC 2
1 RANGL=(1.-EXP(-1./CWARA))*(T1-TD)
2 RANG(1)=.9*RANGL
  RANG(2)=.7*RANGL
  RANG(3)=.5*RANGL
DO 15 IR=1,3
  RANGE=RANG(IR)
  W3=QRFJ/RANGE
  W4=W3/(CWARA*.24)
  IF(CWARA.CE.1.) GO TC 3
  EFF=RANGE/(T1-TD)
  CRAT=CWARA
  CMIN=W3
  GO TC 4
3 EFF=CWARA*RANGE/(T1-TD)
  CRAT=1./CWARA
  CMIN=.24*W4
4 ANTU=-ALOG((1./CRAT)*LOG(1.-CRAT*EFF)+1.)
DO 15 IA=1,3
  AFRN=AFRC(IA)
  GAIR=W4/(AFRCN*SIGMA)
  VAIR=GAIR/(.C73*3600.)
  REAIR=DEC*GAIR/.046
  STPR=SAC/(REAIR**SAS)
  FAIR=FAC/(REAIR**FAS)
  HC=STPR*GAIR*.24/.793
  IF(BEDIA.NE.C.0) GO TC 5
  BFIN=(DFIN/OD+4./3.)/7.
  EFFIN=1./(1.+(BFIN*HC*(DFIN-OD)**2)/(24.*CONF*THFIN))
  GO TO 6
5 BFIN=SQRT(144.*HD/(CONF*BEDIA))
  EFFIN=TANH(BFIN*BEDIA/4.)/(BFIN*BEDIA/4.)
6 EFS=1.-AFTR*(1.-EFFIN)
  LW=4
  IF(DWLRA.EQ.C.0) LW=3

```

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SHOT0073
SHOT0074
SHOT0075
SHOT0076
SHOT0077
SHOT0078
SHOT0079
SHOT0080
SHOT0081
SHOT0082
SHOT0083
SHOT0084
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SHOT0086
SHOT0087
SHOT0088
SHOT0089
SHOT0090
SHOT0091
SHOT0092
SHOT0093
SHOT0094
SHOT0095
SHOT0096
SHOT0097
SHOT0098
SHOT0099
SHOT0100
SHOT0101
SHOT0102
SHOT0103
SHOT0104
SHOT0105
SHOT0106
SHOT0107
SHOT0108

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```

CO 15 IW=1,LW
WLRAT=WLRAT(IW)
WIDTH=SQRT(WLRAT*AFRCN)
NW=WIDTH/XW*12
KN=C
N=2
7 REWAT=48.*W3/(3.14*1.65*CD*N*NW)
HI=.1836*(REWAT**.8)/CD
U=1./(1./(EFS*HD)+1./((1.-AFTR)*HI))
AAIR=ANTL*CMIN/U
VOL=AAIR/ALPHA
DEPTH=VOL/AFRDN
ND=DEPTH/XD*12
KN=KN+1
IF(KN.GT.4) GO TO 9
NDEL=N-ND
IF(NDEL*NDEL.LT.1) GO TO 9
IF(NC.LT.1) GO TO 8
N=ND
GO TO 7
8 N=1
GO TO 7
9 GWAT=576.*W3/(3.14*CD*CD*N*NW)
ELENG=WIDTH/WLRAT
IF(REWAT.GT.2000.) GO TO 10
FWAT=16./REWAT
GO TO 12
10 IF(REWAT.GT.1.E5) GO TO 11
FWAT=.0791/(REWAT**.25)
GO TO 12
11 FWAT=.046/(REWAT**.2)
12 DELPW=48.*FWAT*ELENG*(GWAT/3600.)**2/(CD*62.*64.4)
T4=CWARA*RANGE+TD
AV4AR=(T4+460.)/(TD+460.)
AVMAR=((T4+TD)/2.+460.)/(TD+460.)
VA=53.35*(TD+460.)/(15.*144.)

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SHOT0109
SHOT0110
SHOT0111
SHOT0112
SHOT0113
SHOT0114
SHOT0115
SHOT0116
SHOT0117
SHOT0118
SHOT0119
SHOT0120
SHOT0121
SHOT0122
SHOT0123
SHOT0124
SHOT0125
SHOT0126
SHOT0127
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SHOT0130
SHOT0131
SHOT0132
SHOT0133
SHOT0134
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SHOT0136
SHOT0137
SHOT0138
SHOT0139
SHOT0140
SHOT0141
SHOT0142
SHOT0143
SHOT0144

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AFACT=(1.+SIGMA**2)*(AV4AR-1.)+FAIR*DEPTH*4.*AVMAR/DEQ	SHOT0145
DELP=(GAIR/3600.)*2*VA*AFACT/64.4	SHOT0146
HPWAT=DELPW*W3/(62.*3600.*550.)	SHOT0147
FPCW=HPWAT/(1.341*EFFF)	SHOT0148
HPAIR=W4*(DELP*VA+VAIR**2/(64.4*(1.-RECCV)))/(3600.*550.)	SHOT0149
FPCW=HPAIR/(1.341*EFFF)	SHOT0150
ATUB=(1.-AFTR)*AAIR	SHOT0151
XN=N	SHOT0152
CPERA=C1+C2*XN+C3*XN**2+C4*XN**3+C5*XN**4	SHOT0153
CBOIL=(HRFAC-1.)*BCOS1*1.E6	SHOT0154
CCOS=ATUB*CPERA+(HPWAT+HPAIR)*HPCST+VAIR*AFRON*PROPC/FANV	SHOT0155
CCCS=(PPCW+FFCW)/(PSIZE*1000.)*PWCOS	SHOT0156
DELFC=FCCS*3413.*(HRFAC-1.)/TEFF*1.E-5	SHOT0157
COSM=CCCS*PER/(PSIZE*(APF*8760.))	SHOT0158
TCOS=(CCCS+CBOIL)*FCR/(PSIZE*1000.*CAPF*8.76)+CCCS+COSM+DELFC	SHOT0159
IF(DELP.LT.XDEPA) GO TO 16	SHOT0160
IF(TCOS1.EQ.C.) GO TO 14	SHOT0161
IF(TCOS.GT.TCOS1) GO TO 15	SHOT0162
GO TO 14	SHOT0163
16 IF(L.EQ.2) GO TO 15	SHOT0164
IF(TCOS2.EQ.C.) GO TO 17	SHOT0165
IF(TCOS.GT.TCOS2) GO TO 15	SHOT0166
17 TCOS2=TCOS	SHOT0167
VAM(1)=T1	SHOT0168
VAM(3)=CWARA	SHOT0169
VAM(4)=AFRON	SHOT0170
VAM(5)=WL RAT	SHOT0171
GO TO 15	SHOT0172
14 TCOS1=TCOS	SHOT0173
VAR(1)=T1	SHOT0174
VAR(2)=RANGE	SHOT0175
VAR(3)=CWARA	SHOT0176
VAR(4)=AFRON	SHOT0177
VAR(5)=WL RAT	SHOT0178
CAF=AFRO(1)*.1	SHOT0179
15 CONTINUE	SHOT0180

```
WRITE(6,553)TCOS1,VAR(1),VAR(2),VAR(3),VAR(4),VAR(5)
553 FORMAT(' TCOS1='E12.5,' T11='F4.0,' RANG1='F7.4,' CWAR1='F7.3,
1' AFRO1='E10.3,' WLRA1='F4.0)
RETURN
END
```

```
SHOT0181
SHOT0182
SHOT0183
SHOT0184
SHOT0185
```

```

SUBROUTINE XTEND(L)
  DIMENSION TTRY(2),RTRY(2),CTRY(2),ATRY(2),WTRY(2)
  COMMON VAR(5),VAL(5),VAS(5),TPO(3),TCOS1,TCOS,CCOS1,CCOS,REAL1,
1  REALR,N1,N,NW1,NW,HREAL,HREAL,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2  DELPA,PPCW1,PPCW,FPCW1,FPCW,OCOS1,OCOS,CCSM1,CCSM,DEFC1,DEFC,
3  WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,F11,F1,HU1,HU,T41,T4
4  EFF1,EFF,QIN1,QIN,CBCI1,CBOIL,TEFF,PSIZE,PWCCS,PER,FCOS,FCR,CAPF
  COMMON BCOST,HPCST,EFFP,EFFP,PROPC,FANV, RECDV,C1,C2,C3
1  C4,C5,XW,XD,DEQ,JD,CCNF,AETR,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2  THFIN,BEDIA,CAF,REWA1,REWA,TD,QREJ,XDEPA,KCCNV
  TTRY(1)=VAR(1)-20.
  IF(TTRY(1).LE.100.) TTRY(1)=110.
  TTRY(2)=VAR(1)+20.
  IF(TTRY(2).GE.170.) TTRY(2)=170.
  CTRY(1)=VAR(3)-3.
  IF(CTRY(1).LE.0.) CTRY(1)=.5
  CTRY(2)=VAR(3)+2.
  ATRY(1)=VAR(4)*4.
  ATRY(2)=VAR(4)/4.
  WTRY(1)=VAR(5)+40.
  WTRY(2)=VAR(5)/3.
  DO 3 I=1,2
    IF(VAR(3).GE.1.) GO TC 1
    RANGL=(1.-EXP(-VAR(3)))*(TTRY(I)-TD)/VAR(3)
    GO TC 2
1  RANGL=(1.-EXP(-1./VAR(3)))*(TTRY(I)-TD)
2  RTRY(1)=.9*RANGL
  RTRY(2)=.7*RANGL
  DO 3 IR=1,2
    CALL CALC(TTRY(IR),RTRY(IR),VAR(3),VAR(4),VAR(5),1)
    IF(DELPW.LT.XDEPA) GO TO 3
    IF(TCOS.GT..99*TCOS1) GO TO 3
  TCOS1=TCOS
  VAS(1)=TTRY(1)
  VAS(3)=VAR(3)
  VAS(4)=VAR(4)

```

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XTENCC01
XTEN0002
XTEN0003
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XTEN0035
XTEN0036

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      VAS(5)=VAR(5)
      L=1
3  CONTINUE
      DO 10 I=1,2
      IF(CTRY(I).GE.1.) GC TC 8
      RANGL=(1.-EXP(-CTRY(I)))*(VAR(1)-TD)/CTRY(I)
      GO TC 9
8  RANGL=(1.-EXP(-1./CTRY(I)))*(VAR(1)-TD)
9  RTRY(1)=.9*RANGL
   RTRY(2)=.7*RANGL
      DO 10 IR=1,2
      CALL CALC(VAR(1),RTRY(IR),CTRY(I),VAR(4),VAR(5),I)
      IF(DELPA.LT.XDEPA) GC TC 10
      IF(TCOS.GT..99*TCOS1) GO TO 10
      TCCS1=TCCS
      VAS(1)=VAR(1)
      VAS(3)=CTRY(I)
      VAS(4)=VAR(4)
      VAS(5)=VAR(5)
      L=1
10 CONTINUE
      DO 11 I=1,2
      CALL CALC(VAR(1),VAR(2),VAR(3),ATRY(I),VAR(5),2*I)
      IF(DELPA.LT.XDEPA) GC TO 11
      IF(TCOS.GT..99*TCOS1) GO TO 11
      TCCS1=TCCS
      VAS(1)=VAR(1)
      VAS(3)=VAR(3)
      VAS(4)=ATRY(I)
      VAS(5)=VAR(5)
      L=1
11 CONTINUE
      DO 12 I=1,2
      CALL CALC(VAR(1),VAR(2),VAR(3),VAR(4),WTRY(I),3+I)
      IF(DELPA.LT.XDEPA) GC TO 12
      IF(TCOS.GT..99*TCOS1) GO TO 12

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XTEN0037
XTEN0038
XTEN0039
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XTEN0064
XTEN0065
XTEN0066
XTEN0067
XTEN0068
XTEN0069
XTEN0070
XTEN0071
XTEN0072

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```
TCCS1=TCCS
VAS(1)=VAR(1)
VAS(3)=VAR(3)
VAS(4)=VAR(4)
VAS(5)=WTRY(I)
L=1
12 CONTINUE
  IF(L.EQ.1) WRITE(6,13)
13 FORMAT(' XTEND CALLED, AND CHANGE MADE')
  RETURN
  END
```

```
XTEN0073
XTEN0074
XTEN0075
XTEN0076
XTEN0077
XTEN0078
XTEN0079
XTEN0080
XTEN0081
XTEN0082
XTEN0083
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```

SUBROUTINE SERCH
DIMENSION C(5)
COMMON VAR(5),VAL(5),VAS(5),TPO(3),TCOS1,TCOS,CCCS1,CCOS,REAI1,
1 REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2 DELPA,PPCW1,PPCW,FPCW1,FPOW,CCCS1,CCCS,CCSM1,CUSM,DEFC1,DELC,
3 WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,H11,H1,H01,H0,T41,T4
4 EFF1,EFF,QIN1,CIN,CBC11,CBUIL,TEFF,PSIZE,PWCOS,PER,FCOS,FCR,CAPF
COMMON BCOST,HPCST,EFFP,EFFF,PROFC,FANV,RECCV,C1,C2,C3
1,C4,C5,XW,XD,DEQ,OD,CCNF,AFT,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2 THFIN,BEDIA,CAF,REWA1,REWA,TD,QREJ,XDEPA,KCONV
C(1)=5.
C(2)=1.
C(3)=1.
C(4)=CAF
C(5)=5.
DO 8 I=1,5
M=6-I
K=3
DO 9 IK=1,5
9 VAL(IK)=VAR(IK)
L=1
KC=C
1 J=0
2 VAL(M)=VAR(M)+C(M)
IF(M.NE.2) GO TO 1C
IF(VAL(3).GE.1.) GO TO 60
RANGL=(1.-EXP(-VAL(3)))*(VAL(1)-TD)/VAL(3)
GO TO 61
60 RANGL=(1.-EXP(-1./VAL(3)))*(VAL(1)-TD)
61 IF(VAL(2).GE.RANGL) GO TO 4
10 IF(VAL(1).GT.170.) GO TO 4
IF(VAL(3).GE.-1./ALCG(1.-VAL(2)/(VAL(1)-TD))) GO TO 4
CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),L)
L=M
IF(DELPW.LT.XDEPA) GO TO 3
IF(TCOS.GT.TCOS1) GO TO 4

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SERC0001
SERC0002
SERC0003
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SERC0035
SERC0036

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```

J=1
CALL STORE
GO TO 2
3 IF(K.EQ.1) KC=1
4 IF(J.EQ.1) GO TO 7
5 VAL(M)=VAR(M)-C(M)
  IF(VAL(M).LE.0.0) GO TO 7
  IF(VAL(1).LE.TD+VAL(2)/(1.-EXP(-1./VAL(3))).CR.VAL(1).LT.110.) GO
1 TO 7
  CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),M)
  IF(DELPA.LT.XDEPA) GO TO 6
  IF(TCOS.GT.TCOS1) GO TO 7
  CALL STORE
  KC=0
  GO TO 5
6 IF(K.EQ.1) KC=1
7 IF(KC.EQ.1) CALL CHNGE(M)
  K=K-1
  WRITE(6,18)K,M,VAR(M),M,VAL(M),TCOS,TCOS1,DELPA
18 FORMAT(' K='I2,' VAR('I1,')='E15.5,' VAL('I1,')='E15.5,' TCOS='
1E15.5,' TCOS1='E15.5,' DELPA='F10.6)
  IF(K.EQ.0) GO TO 8
  C(M)=C(M)/5.
  GO TO 1
8 CONTINUE
RETURN
END

```

```

SERC0037
SERC0038
SERC0039
SERC0040
SERC0041
SERC0042
SERC0043
SERC0044
SERC0045
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SERC0047
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SERC0049
SERC0050
SERC0051
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SERC0058
SERC0059
SERC0060
SERC0061
SERC0062
SERC0063

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```

SUBROUTINE CHNGE(M)
  DIMENSION DP(5),DV(5),VAJ(5)
  COMMON VAR(5),VAL(5),VAS(5),TPU(3),TCOS1,TCOS,CCCS1,CCOS,REAI1,
1 REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2 DELPA,PPCW1,PPOW,FPCW1,FPCW,CCCS1,CCCS,CCSM1,COSM,DEFC1,DELCF,
3 WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,H11,HI,HO1,HO,T41,T4
4 EFF1,EFF,QIN1,QIN,CECI1,CHOIL,TEFF,PSIZE,PWCOS,PER,FCOS,FCR,CAPF
  COMMON BCOST,HPCST,EFFP,EFFF,PRCFC,FANV, RECCV,C1,C2,C3
1 C4,C5,XW,XD,DEQ,CD,CCNF,AETR,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2 THFIN,BEDIA,CAF,REWA1,REWAT,TD,QREJ,XDEPA,KCCNV
  WRITE(6,100)
100 FORMAT(' SUBROUTINE CHNGE CALLED')
  L=C
  K1=0
  DO 1 I=1,3
1 CV(I)=.05
  DO 2 I=1,2
2 DV(I+3)=.2
  DO 3 I=1,5
  VAJ(I)=VAR(I)
3 VAL(I)=VAR(I)
4 VAL(M)=(1.+DV(M))*VAR(M)
  IF(VAL(M).LE.0.0) GO TO 36
  IF(M.GT.4) GO TO 10
  IF(M-2)5,6,9
5 IF(VAL(1).GE.170.) GO TO 36
  IF(VAL(1).LE.TD+VAL(2)/(1.-EXP(-1./VAL(3))).CR.VAL(1).LT.110.) GO
1 TO 36
  GO TO 10
6 IF(VAL(3).GE.1.) GO TO 7
  RANGL=(1.-EXP(-VAL(3)))*(VAL(1)-TD)/VAL(3)
  GO TO 8
7 RANGL=(1.-EXP(-1./VAL(3)))*(VAL(1)-TD)
8 IF(VAL(2).GE.RANGL) GO TO 36
  GO TO 10
9 IF(VAL(3).GE.-1./ALCG(1.-VAL(2)/(VAL(1)-TD))) GO TO 36

```

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CHNG0001
CHNG0002
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CHNG0035
CHNG0036

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```

10 CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),1)
   IF(KCCNV.EQ.1) GO TO 36
   CP(M)=DELP-DEPA1
   CC 35 II=1,5
   I=6-II
   IF(I.EQ.5.AND.DV(1).GT..005) GO TO 35
   IF(I.EQ.M) GC TO 35
11 VAL(I)=(1.+DV(I))*VAR(I)
   IF(VAL(I).LE.0.0) GC TO 34
   IF(I.GT.3) GO TO 17
   IF(I-2)12,13,16
12 IF(VAL(1).GE.170.) GC TO 34
   IF(VAL(1).LE.TD+VAR(2)/(1.-EXP(-1./VAR(3))).CR.VAL(1).LT.110.) GC
   1 TC 34
   GO TC 17
13 IF(VAR(3).GE.1.) GO TC 14
   RANGL=(1.-EXP(-VAR(3)))*(VAR(1)-TD)/VAR(3)
   GO TC 15
14 RANGL=(1.-EXP(-1./VAR(3)))*(VAR(1)-TD)
15 IF(VAL(2).GE.RANGL) GC TO 34
   GO TO 17
16 IF(VAL(3).GE.-1./ALCG(1.-VAR(2)/(VAR(1)-TD))) GO TO 34
17 VAJ(I)=VAL(I)
   J=I
   IF(M.LT.I) J=M
   CALL CALC(VAJ(1),VAJ(2),VAJ(3),VAJ(4),VAJ(5),J)
   VAJ(I)=VAR(I)
   IF(KCCNV.EQ.1) GO TO 34
   CP(I)=DELP-DEPA1
   E=0.
   C=1.
   L=1.
18 VAL(I)=(1.+DV(I)*(E-CP(M))/DP(I))*VAR(I)
   IF(VAL(I).LE.0.0) GC TO 34
   IF(I.GT.3) GC TO 24
   IF(I-2)19,20,23

```

```

CHNGCC37
CHNG0038
CHNG0039
CHNG0040
CHNG0041
CHNG0042
CHNG0043
CHNG0044
CHNG0045
CHNG0046
CHNG0047
CHNG0048
CHNG0049
CHNG0050
CHNG0051
CHNG0052
CHNG0053
CHNG0054
CHNG0055
CHNG0056
CHNG0057
CHNG0058
CHNG0059
CHNG0060
CHNG0061
CHNG0062
CHNG0063
CHNG0064
CHNG0065
CHNG0066
CHNG0067
CHNG0068
CHNG0069
CHNG0070
CHNG0071
CHNG0072

```

```

19 IF(VAL(1).GE.170.) GC TO 34
   IF(VAL(1).LE.TD+VAL(2)/(1.-EXP(-1./VAL(3))).OR.VAL(1).LT.110.) GC
   1 TC 34
   GO TO 24
20 IF(VAL(3).GE.1.) GO TC 21
   RANGL=(1.-EXP(-VAL(3)))*(VAL(1)-TD)/VAL(3)
   GO TC 22
21 RANGL=(1.-EXP(-1./VAL(3)))*(VAL(1)-TC)
22 IF(VAL(2).GE.RANGL) GC TO 34
   GO TC 24
23 IF(VAL(3).GE.-1./ALCG(1.-VAL(2)/(VAL(1)-TD))) GO TC 34
24 CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),J)
   IF(KCONV.EQ.1) GC TO 34
   IF(DELPA.GE.XDEPA.AND.DELPA.LE.1.01*XDEPA) GC TO 26
   IF(DELPA.GE.1.01*XDEPA) GO TO 25
   C=D/10.
   IF(C.LT..001) GO TO 34
   E=E+C
   GO TC 18
25 E=C/10.
   IF(C.LT..001) GO TO 34
   E=E-D
   GO TC 18
26 IF(TCOS.GT.TCOS1) GC TO 34
   CALL STORE
   L=1
   DO 27 IJ=1,5
27 VAJ(IJ)=VAR(IJ)
   VAL(M)=(1.+DV(M))*VAR(M)
   IF(VAL(M).LE.0.C) GC TO 36
   IF(M.GT.3) GC TO 33
   IF(M-2)28,29,32
28 IF(VAL(1).GE.170.) GC TO 36
   IF(VAL(1).LE.TD+VAL(2)/(1.-EXP(-1./VAL(3))).OR.VAL(1).LT.110.) GC
   1 TC 36
   GO TO 33

```

```

CHNG0073
CHNG0074
CHNG0075
CHNG0076
CHNG0077
CHNG0078
CHNG0079
CHNG0080
CHNG0081
CHNG0082
CHNG0083
CHNG0084
CHNG0085
CHNG0086
CHNG0087
CHNG0088
CHNG0089
CHNG0090
CHNG0091
CHNG0092
CHNG0093
CHNG0094
CHNG0095
CHNG0096
CHNG0097
CHNG0098
CHNG0099
CHNG0100
CHNG0101
CHNG0102
CHNG0103
CHNG0104
CHNG0105
CHNG0106
CHNG0107
CHNG0108

```

```

29 IF(VAL(3).GE.1.) GO TC 30
   RANGL=(1.-EXP(-VAL(3)))*(VAL(1)-TD)/VAL(3)
   GO TC 31
30 RANGL=(1.-EXP(-1./VAL(3)))*(VAL(1)-TC)
31 IF(VAL(2).GE.RANGL) GC TO 36
   GO TC 33
32 IF(VAL(3).GE.-1./ALCG(1.-VAL(2)/(VAL(1)-TC))) GO TC 36
33 CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),M)
   IF(KCONV.EQ.1) GO TO 36
   CP(M)=DELPA-CEPA1
   GO TC 11
34 VAL(I)=VAR(I)
35 CONTINUE
36 DO 37 I=1,5
37 DV(I)=-DV(I)
   K1=K1+1
   IF(K1.EQ.1) GO TO 4
   K1=0
   IF(DV(1).LE..005) GO TO 39
   DO 38 I=1,5
38 DV(I)=DV(I)/5.
   GO TC 4
39 IF(L.EQ.1) WRITE(6,103)
103 FORMAT('          AND CHANGES MADE')
   RETURN
   END

```

```

CHNG0109
CHNG0110
CHNG0111
CHNG0112
CHNG0113
CHNG0114
CHNG0115
CHNG0116
CHNG0117
CHNG0118
CHNG0119
CHNG0120
CHNG0121
CHNG0122
CHNG0123
CHNG0124
CHNG0125
CHNG0126
CHNG0127
CHNG0128
CHNG0129
CHNG0130
CHNG0131
CHNG0132
CHNG0133
CHNG0134

```



```

SUBROUTINE CALC(T1,RANGE,CWARA,AFR0N,WLR,T,M)
  CCMCN VAR(5),VAL(5),VAS(5),TPU(3),TCOS1,TCOS,CCOS1,CCOS,REAI1,
1 REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
2,DELPA,PPCW1,PPOW,FPCW1,FPOW,UCOS1,UCCS,CGSM1,COSM,DEFC1,DELFC,
3WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,H11,H1,H01,H0,T41,T4
4,EFF1,EFF,QIN1,QIN,CBCI1,CBOIL,TEFF,PSIZE,PWCCS,PER,FCOS,FCR,CAPF
  CCMCN BCOST,HPCST,EFFP,EFFF,PROPC,FANV, RECOV,C1,C2,C3
1,C4,C5,XW,XD,DEQ,DD,CCNF,AFTF,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2THFIN,BEDIA,CAF,REWA1,REWAT,TD,QREJ,XDEPA,KCCNV
  KCCNV=0
  IF(M.EQ.5) GO TO 23
  IF(M.EQ.4) GO TO 22
  IF(M.EQ.3) GO TO 21
  IF(M.EQ.2) GO TO 20
  F=EXP(17.168-9240./(T1+460.))
  HRFAC=1.+TPC(1)*P+TPC(2)*P**2+TPC(3)*P**3
  TEFFM=TEFF/HRFAC
  CIN=3413.*PSIZE*1000./TEFFM
  QREJ=QIN*(1.-TEFFM)
20 W3=QREJ/RANGE
21 W4=W3/(CWARA*.24)
  IF(CWARA.GE.1.) GO TO 1
  EFF=RANGE/(T1-TD)
  CRAT=CWARA
  CMIN=W3
  GO TO 2
1 EFF=CWARA*RANGE/(T1-TD)
  CRAT=1./CWARA
  CMIN=.24*W4
2 ANTU=-ALOG(1./CRAT)*ALOG(1.-CRAT*EFF)+1.)
22 GAIR=W4/(AFRCN*SIGMA)
  VAIR=GAIR/(.073*3600.)
  REAIR=DEQ*GAIR/.046
  STPR=SAC/(REAIR**SAS)
  FAIR=FAC/(REAIR**FAS)
  FG=STPR*GAIR*.24/.793

```

```

CALC0001
CALC0002
CALC0003
CALC0004
CALC0005
CALC0006
CALC0007
CALC0008
CALC0009
CALC0010
CALC0011
CALC0012
CALC0013
CALC0014
CALC0015
CALC0016
CALC0017
CALC0018
CALC0019
CALC0020
CALC0021
CALC0022
CALC0023
CALC0024
CALC0025
CALC0026
CALC0027
CALC0028
CALC0029
CALC0030
CALC0031
CALC0032
CALC0033
CALC0034
CALC0035
CALC0036

```

IF(BEDIA.NE.C.0) GO TC 3	CALC0037
PFIN=(DFIN/OD+4./3.)/7.	CALC0038
EFFIN=1./(1.+(BFIN*HC*(DFIN-JD)**2)/(24.*CONF*THFIN))	CALC0039
GO TC 4	CALC0040
3 BFIN=SQRT(144.*HC/(CCNF*BEDIA))	CALC0041
EFFIN=TANH(BFIN*BEDIA/4.)/(BFIN*BEDIA/4.)	CALC0042
4 EFS=1.-AFTR*(1.-EFFIN)	CALC0043
23 WIDTH=SQRT(WLRAT*AFRCN)	CALC0044
NW=WIDTH/XW*12	CALC0045
KN=C	CALC0046
N=2	CALC0047
5 REWAT=48.*W3/(3.14*1.65*CC*N*NW)	CALC0048
FI=.1836*(REWAT**8)/CD	CALC0049
U=1./(1./(EFS*HC)+1./((1.-AFTR)*FI))	CALC0050
AAIR=ANTL*CMIN/U	CALC0051
VOL=AAIR/ALPHA	CALC0052
DEPTH=VOL/AFRCN	CALC0053
ND=DEPTH/XD*12	CALC0054
NDEL=N-ND	CALC0055
IF(NDEL*NDEL.LT.1) GO TC 11	CALC0056
KN=KN+1	CALC0057
IF(KN.GT.4) GO TO 10	CALC0058
IF(ND.LT.1) GO TC 9	CALC0059
N=ND	CALC0060
GO TC 5	CALC0061
9 N=1	CALC0062
GO TO 5	CALC0063
10 WRITE(6,80)N,ND	CALC0064
80 FORMAT(' TUBE BANKS NOT CONVERGENT, WITH N=',I6,' AND ND=',I6)	CALC0065
KCCNV=1	CALC0066
11 GWAT=576.*W3/(3.14*CC*CD*N*NW)	CALC0067
VWAT=GWAT/(62.*3600.)	CALC0068
ELENG=WIDTH/WLRAT	CALC0069
IF(REWAT.GT.2000.) GO TO 12	CALC0070
FWAT=16./REWAT	CALC0071
GO TC 14	CALC0072

```

12 IF(REWAT.GT.1.E5) GC TC 13
   FWAT=.0791/(REWAT**.25)
   GC TC 14
13 FWAT=.046/(REWAT**.2)
14 CELPW=48.*FWAT*ELENG*(GWAT/ 3600.)**2/(OC*62.*64.4)
   T4=CWARA*RANGE+TD
   AV4AR=(T4+46C.)/(TD+460.)
   AVMAR=((T4+TC)/2.+46C.)/(TD+460.)
   VA=53.35*(TD+46C.)/(15.*144.)
   AFACT=(1.+SIGMA**2)*(AV4AR-1.)+FAIR*DEPTH*4.*AVMAR/DEQ
   DELPA=(GAIR/3600.)**2*VA*AFACT/64.4
   HPWAT=DELPW*W3/(62.*3600.*550.)
   FPOW=HPWAT/(1.341*EFFF)
   HPAIR=W4*(DELPW*VA+VAIR**2/(64.4*(1.-RECCV)))/(3600.*550.)
   FPOW=HPAIR/(1.341*EFFF)
   ATUB=(1.-AFTF)*AAIR
   XN=N
   CPERA=C1+C2*XN+C3*XN**2+C4*XN**3+C5*XN**4
   CBOIL=(HRFAC-1.)*BCOST*1.E6
   CCOS=ATUB*CPERA+(HPWAT+HPAIR)*HPCST+VAIR*AFRCN*PRCFC/FANV
   CCGS=(PPCW+FFOW)/(PSIZE*1000.)*PWCOS
   DELFC=FCCS*3413.*(HRFAC-1.)/TEFF*1.E-5
   COSM=CCOS*PER/(PSIZE*CAPF*8760.)
   TCOS=(CCCS+CBOIL)*FCR/(PSIZE*1000.*CAPF*8.76)+OCOS+COSM+DELF
RETURN
END

```

```

CALCC073
CALC0074
CALC0075
CALC0076
CALC0077
CALC0078
CALC0079
CALC0080
CALC0081
CALC0082
CALC0083
CALC0084
CALC0085
CALC0086
CALC0087
CALC0088
CALC0089
CALC0090
CALC0091
CALC0092
CALC0093
CALC0094
CALC0095
CALC0096
CALC0097
CALC0098

```

```

SUBROUTINE VARIT(TA,TFER,PCST)
  COMMON VAR(5),VAL(5),VAS(5),TPO(3),TCOS1,CUM1,CCCS1,CCOS,REAI1,
1 REAIR,N1,N,NW1,NW,HRFA1,DUMY1,W31,W3,W41,W4,U,DEPW1,DELPW,DEPA1
2 DELFA,PPCW1,PPCW,FPCW1,FPOW,CCCS1,CCCS,CCSM1,CCSM,DEFC1,DUMY2,
3 WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,F11,F1,F01,F0,T41,T4
4 EFF1,EFF,QIN1,QIN,CBCI1,CBOIL,TEFF,PSIZE,PWCOS,PER,FCUS,FCR,CAPF
  COMMON BCCST,HPCST,EFFP,EFFF,PROFC,FANV,      RECOV,C1,C2,C3
1 C4,C5,XW,XD,DEQ,JD,CCNF,AFT,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
2 THFIN,BEDIA,CAF,REWA1,REWA,TD,DUMY3,XDEPA,KCCNV
  T1=140.
1 P=EXP(17.168-9240./(T1+460.))
  HRFAC=1.-.0049*P+.002675*P**2-.0001125*P**3
  TEFFM=TEFF/HRFAC
  GREJ=QIN1*(1.-TEFFM)
  RANGE=GREJ/W31
  IF(VAR(3).GT.1.) GO TO 2
  T11=RANGE/EFF1+TA
  GO TO 3
2 T11=RANGE*VAR(3)/EFF1+TA
3 IF((T11-T1)**2.LE.1.) GO TO 4
  T1=T11
  GO TO 1
4 PGEN=QIN1*TEFFM/(3413.*1000.)
  IF(PGEN.GE.PSIZE) GO TO 5
  CSTCL=(PSIZE-PGEN)/PSIZE*PWCOS
  GO TO 6
5 CSTCL=0
6 DELFC=FCCS*3413.*(HRFAC-1.)/TEFFM*1.E-5
  TCOS=(CCOS1+CBOI1)*FCR/(PSIZE*CAFF*8760.)+QCCS1+CCSM1+DELFC+CSTCL
  PCST=TPER*TCOS
  WRITE(6,7)
7 FORMAT(' SUBROUTINE DATA')
  WRITE(6,8)HRFAC,PGEN,CSTCL,DELFC
8 FORMAT(' HRFAC='F7.4,' PGEN='F8.3,' CSTCL='E15.5,' DELFC='E15.5)
  WRITE(6,9)RANGE,TCOS,PCST,T11,TA
9 FORMAT(' RANGE='F6.3,' TCCS='E15.5,' PCST='E15.5,' T11='F7.3,

```

```

VAR10001
VAR10002
VAR10003
VAR10004
VAR10005
VAR10006
VAR10007
VAR10008
VAR10009
VAR10010
VAR10011
VAR10012
VAR10013
VAR10014
VAR10015
VAR10016
VAR10017
VAR10018
VAR10019
VAR10020
VAR10021
VAR10022
VAR10023
VAR10024
VAR10025
VAR10026
VAR10027
VAR10028
VAR10029
VAR10030
VAR10031
VAR10032
VAR10033
VAR10034
VAR10035
VAR10036

```

1* TA='F6.2)
RETURN
END

VARI0037
VARI0038
VARI0039

```

SUBROUTINE STORE
  CCMCN VAR(5),VAL(5),VAS(5),TPC(3),TCOS1,TCJS,CCOS1,CCJS,REAI1,
  1REAIR,N1,N,NW1,NW,HRFA1,HRFAC,W31,W3,W41,W4,U1,U,DEPW1,DELPW,DEPA1
  2,DELPA,PPCW1,PPCW,FPCW1,FPCW,OCOS1,OCOS,CCSM1,CJSM,DEFC1,DELFC,
  3WIDT1,WIDTH,ELEN1,ELENG,VAIR1,VAIR,VWAT1,VWAT,F11,F1,HU1,HU,T41,T4
  4,EFF1,EFF,QIN1,QIN,CBCI1,CBUIL,TEFF,PSIZE,PWCCS,PER,FCUS,FCR,CAPF
  CCMCN BCUST,HPCST,EFFP,EFFF,PROPC,FANV,      RECDV,C1,C2,C3
  1,C4,C5,XW,XD,DEQ,OD,CCNF,AFTF,ALPHA,SIGMA,FAC,FAS,SAC,SAS,DFIN,
  2THFIN,BEDIA,CAF,REWA1,REWAT,TD,QREJ,XDEPA,KCCNV
  VAR(1)=VAL(1)
  VAR(2)=VAL(2)
  VAR(3)=VAL(3)
  VAR(4)=VAL(4)
  VAR(5)=VAL(5)
  TCCS1=TCCS
  CCOS1=CCCS
  REWA1=REWAT
  REAI1=REAIR
  N1=N
  NW1=NW
  HRFA1=HRFAC
  W31=W3
  W41=W4
  U1=U
  DEPW1=DELPW
  DEPA1=DELPA
  PPCW1=PPCW
  FPCW1=FPCW
  CCCS1=CCCS
  CCSM1=CJSM
  DEFC1=DELFC
  WIDT1=WIDTH
  ELEN1=ELENG
  VAIR1=VAIR
  VWAT1=VWAT
  F11=F1

```

```

STCR0001
STOR0002
STOR0003
STCR0004
STOR0005
STOR0006
STOR0007
STOR0008
STOR0009
STCR0010
STOR0011
STCR0012
STOR0013
STOR0014
STOR0015
STOR0016
STCR0017
STOR0018
STOR0019
STCR0020
STOR0021
STOR0022
STOR0023
STOR0024
STCR0025
STCR0026
STOR0027
STCR0028
STOR0029
STOR0030
STCR0031
STOR0032
STCR0033
STOR0034
STOR0035
STOR0036

```

FC1=FC
T41=T4
EFF1=EFF
GIN1=QIN
CBCI1=CBCIL
RETURN
END

STOR0037
STOR0038
STOR0039
STOR0040
STOR0041
STOR0042
STOR0043