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IMPROVEMENT OF THE
ENVIRONMENTAL AND ECONOMIC CHARACTERISTICS
OF COOLING TOWERS

PART II: PERIODIC COOLING TOWERS

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ABSTRACT

A periodic cooling tower was designed to operate at vastly lower evaporation rates than conventional wet towers and at a significant capital savings over dry cooling towers. In this new design discs of galvanized steel sheet metal rotate from the hot water to the cool air. A thin layer of oil floated on the water surface separates the water stream from the air stream and prevents any water carryover. Tests on a model of the periodic tower showed the amount of heat transfer due to evaporation to be less than four tenths of one per cent as compared to eighty per cent for conventional dry towers. Since one half of the total cost of a dry tower is heat transfer surface cost, the new design can significantly reduce the total cost of dry cooling towers.

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: Nomenclature

A	= surface area for heat transfer
B	= heat capacity flow rate times effectiveness for air
C	= heat capacity flow rate times effectiveness for water
C_{Max}	= heat capacity flow rate of water
C_{min}	= heat capacity flow rate of air
C_p	= heat capacity
D	= distance of spacing between plates
d'	= diffusivity
dh	= increment of flow channel height
e	= base of natural logarithms
F	= volume flow rate of air
f	= moody friction factor
g_o	= conversion factor
H	= maximum vertical height of flowchannel
h	= film heat transfer coefficient
h_o	= oil coating on plate thickness
k	= thermal conductivity
L	= channel flow length
M	= atomic mass number
P_o	= pressure in atmospheres
Pr	= Prandtl number
Q	= heat flux
Q/plate	= heat rejection per plate

Nomenclature continued

Q_{total}	= total heat rejection from plant
R	= thermal resistance
r	= plate radius
Re	= Reynolds number
t	= thickness
T_c	= local temperature of cold air
T_h	= local temperature of hot water
T_p	= temperature of plate
U	= overall conductance for heat transfer
V	= fluid velocity
v	= velocity of plate withdrawn from liquid
W	= work done in blowing air through exchanger
w	= mass evaporation rate

Subscripts

a	= air
w	= water
s	= steel
t	= total

Greek Symbols

τ	= atomic volume
ΔP	= pressure drop thru exchanger
ϵ	= effectiveness
μ	= viscosity
ρ	= density

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I. INTRODUCTION

The summary for the year's research on the improvement of the environmental and economic characteristics of cooling towers is encompassed in two separately bound volumes. This report constitutes Part II. Part I covers work on the optimized design program, fluidized beds, non-metallic surfaces, and packed beds.

The Need for a Periodic Cooling Tower

In the past power plants have used large bodies of water for once through cooling of their condensers. However the resultant increase in water temperature may adversely affect the ecology of the water body.

Evaporative cooling towers provide one alternative to once through cooling. Around eighty per cent of the heat transfer in a wet cooling tower is due to evaporation. When the ambient air is cold and has a high relative humidity, the moist air leaving the cooling tower may form a visible fog plume. This plume may obscure visibility on highways and airports; it may appear to merge and reinforce existing overcast conditions; or it may simply be aesthetically unpleasing. Wet cooling towers require a large amount of makeup water and may be prohibited in locations where water supplies are restricted. Dry cooling towers prevent all evaporation but may cost many times as much as a wet tower.

The periodic heat exchanger has been suggested as a compromise between the dry cooling tower and the wet tower. In a periodic exchanger the water to be cooled and the air to be heated are physically separated as in a dry tower. Instead of transferring heat through stationary walls as in a dry exchanger, a solid matrix rotates from the hot stream to the cold stream. The matrix is heated up by the water. Then the hot matrix is rotated into the cool air stream and cooled off. Finally the cooled matrix is rotated

back into the hot water to repeat the cycle.

There are three main advantages to the use of a periodic tower:

1. The rate of evaporation will be much less than in a wet tower if some means of preventing the water from being carried over into the air stream is found.
2. The cost per square foot of heat transfer area is less than that in a dry tower.
3. Because of the periodic flow reversals there are no permanent flow stagnation regions so that the surface tends to be self cleaning. This has been shown in the Ljungstrom - type air preheaters [1].

The major disadvantage of periodic heat exchangers is that when the periodic material rotates from the water side to the air side some water may wet the material and be carried to the air side. The percent of the total energy transfer from the rotating matrix to the air that is caused by evaporation of the carryover determines how effective a compromise the periodic cooling tower is.

In chapter two various means of reducing the amount of water carryover are investigated and a mechanism chosen. In chapter three an analysis is performed on the periodic tower design chosen. Chapter four and five explain the experimental technique and results of tests performed on a model to validate the evaporation suppression mechanism and the theoretical analysis. In chapter six an existing computer optimization program was redesigned and the analytical method for a periodic cooling tower was implemented to determine the optimum design. Chapter seven lists the conclusions and recommendations.

II Water Carryover on Periodic Surfaces

A Bare Aluminum and Teflon Coated Surfaces

One method to reduce the amount of water carried over is to coat the periodic material with a nonstick coating such as teflon. Since teflon coating is expensive the value of this method depends upon how much the evaporation is reduced by the coating and how much surface area would need to be coated. Tests were performed on both bare aluminum and teflon coated plates. Comparison of the results showed that the percent heat transfer due to evaporation was around three times higher for the plates without the nonstick coating. Table 1 shows the cost and percent evaporative heat transfer for conventional, aluminum plate, and teflon coated plate periodic cooling towers for a 1000 MWE plant. The values listed for the conventional towers are the total capital cost. The values for the periodic towers are just the cost for the heat transfer surface and don't include assembly, pump, and fan costs.

Table 1

Tower Type	% evaporative heat transfer	cost (\$ x 10 ⁶)
conventional wet tower	80	8.9
conventional dry tower	0	22.8
bare aluminum periodic plates	40	8.3
teflon coated steel plates	14	21.0

From Table 1 it can be seen that teflon coating is so expensive that it renders the coated design impractical. While the clean aluminum plates reduce the evaporation by a factor of two this gain may be lost with time if the surface of the plate became fouled. Also when fabrication, construction, fan and rotation motor costs are added to the high aluminum cost the economic advantage over dry towers might disappear.

B Hydrophobic Promotion of Periodic Surfaces

Since steel is the least expensive metal, methods of rust prevention and hydrophobic promotion were investigated. It was found that when a layer of oil was floated on the surface of water that the oil would preferentially wet metal surfaces which were dipped into the water. The oil coating kept the water from wetting the metal. When the metal was withdrawn from the water a thin film of oil adhered to the metal surface but no water would coat the surface. If the viscosity of the oil was too high then some small beads of water would become entrained in the oil film coating on the plates and get carried over into the air side. Since these small beads were coated with oil they couldn't evaporate when carried over to the air side. If the oil viscosity was low enough then the coating on the plate is so thin that no water gets carried over at all.

One cooling tower design using this mechanism consists of many thin circular sheet metal discs which rotate through a trough of hot water. The inlet and outlet for water to the trough are below the

surface level so that the oil is not entrained. Forced convection of air cools the top half of the discs before they rotate back into the hot water. A schematic drawing of the design is shown in figure 1. Figure 2 shows several of the rows of discs stacked up to conserve land area.

A small model of this type of tower was constructed to test the feasibility of the scheme. Tests were run on the model at different heat rejection rates, different rotational velocities, and different plate spacings. The purpose of the tests was to determine how effectively evaporation was suppressed, determine the heat transfer relationships for the model and to locate any unforeseen problems in the design.

Figure 1

Left Half of a Periodic Cooling Tower

Figure 2

C. SELECTION OF SURFACE GEOMETRY FOR PERIODIC HEAT EXCHANGERS

Along with the parallel plate geometry there are numerous other geometries which are candidates for the periodic heat exchanger. Before doing a computer optimization on any of the geometries it would be helpful if the most promising could be found by a simple study of the heat transfer and pressure drop characteristics. The goal of this simple study is to relate the heat transfer and pressure drop characteristics of a given geometry to the air side fan power and the total area necessary for heat transfer for a given power plant. To do this, the governing equation must be rewritten.

The pressure drop for air flow through the exchanger can be expressed as

$$\Delta p = \frac{G^2}{2g_c} f \frac{A}{A_c} v_m \quad (1)$$

where G is the air mass velocity (mass flow rate/ A_c). A_c is the minimum free flow air side area, A is the total heat transfer surface area wetted by the air, f is the friction factor, v_m is the average specific volume of the air and g_c is the proportionality factor in Newton's second law.

The fan power to force the air through the exchanger is,

$$p = \Delta p \dot{m} v_m / \eta_F = \Delta p G A_c v_m / \eta_F \quad (2)$$

where $\dot{m}$ is the air side mass flow rate and η_F is the fan efficiency.

The heat rejected from the plant must equal the energy rise of the air flowing through the exchanger,

$$q = (ITD) \epsilon G A_c (C_p)_{AIR} \quad (3)$$

q is the total heat transferred per unit time, C_p is the specific heat, ITD is the difference between the water inlet temperature and the air inlet temperature and ϵ is the heat exchanger effectiveness defined as,

$$\epsilon = \frac{\Delta T_{air}}{ITD} \quad (3a)$$

Combining equations 2 and 3 to remove the product GA_c

$$p = \frac{\Delta p \ v_m \ q}{\eta_F \ \epsilon (ITD)} \quad (4)$$

We can assume q , v_m and η_F will be approximately constant for any cooling tower designed for a given plant. Equation 4 demonstrates that the fan power can be reduced only by decreasing Δp and increasing ϵ or ITD . The minimum value of Δp is fixed by considerations of flow maldistribution under windy conditions; if Δp is allowed to get too small parts of the heat exchanger surface can be starved of flow under windy conditions. The temperature difference ITD must be set by a complete system optimization since raising this difference reduces the power plant efficiency. Such an optimization is described in last year's annual report. The effectiveness of ϵ can be increased by increasing the heat exchanger surface area but this in turn increases the capital cost of the exchanger. Equation 4 is important since it shows that there are only a very limited number of ways to decrease the total fan power. Furthermore, neither heat transfer nor the pressure drop characteristics of the surface explicitly enters equation 4.

The effectiveness of ϵ is a function of the heat transfer area, nondimensionalized as the number of transfer units, NTU. Neglecting the heat transfer resistance from the water to the rotor, NTU can be written as

$$NTU = hA/G \ A_c \ c_p \quad (5)$$

h is the heat transfer coefficient between the matrix of the rotor and the air. Experimental results for h for specific geometries can be expressed in terms of the Colburn j factor defined as

$$j = (h/G C_p) Pr^{2/3} \quad (6)$$

Pr is the Prandtl number of air, approximately equal to 0.7 at atmospheric conditions.

Combining equations 5 and 6 there results

$$NTU = \frac{j}{Pr^{2/3}} \frac{A}{A_c} \quad (7)$$

or

$$A = A_c (NTU) Pr^{2/3} / j \quad (8)$$

A_c can be removed from equation 8 by combining equations 1 and 3.

From equation 3,

$$A_c = q / (ITD) \epsilon G C_p \quad (9)$$

and G can be found by combining equations 1 and 7,

$$G = \frac{j \ 2g_c \ \Delta \ell}{\left[\frac{f}{NTU} Pr^{2/3} \right]^{1/2}} \quad (10)$$

Substituting equation 10 into 9 to eliminate G and then substituting equation 9 into 8 to eliminate A the resulting expression is,

$$A = \frac{1}{j} \sqrt{\frac{f}{j}} \frac{(NTU)^{3/2}}{[\Delta p \ 2g_c]^{1/2}} \frac{v_m^{1/2} q}{(ITD) \epsilon c_p} \quad (11)$$

For heat exchangers ϵ initially rises almost linearly with NTU and then ϵ approaches an asymptotic value as NTU gets large, ϵ is always equal to or less than unity. Figure 3 shows the ratio $(NTU)^{3/2} / \epsilon$ as a function of ϵ for a periodic heat exchanger. Note that knowledge of the specific surface geometry is not necessary to construct figure 3. As ϵ increases the ratio of NTU to ϵ in equation 11 always increases. Thus, the desire for a high ϵ to reduce the fan power as rotated by equation 4 is in

Figure 3

conflict with equation 11 which shows that the total surface area and the capital cost increases as ϵ is increased.

The term $1/j\sqrt{f/j}$ in equation 11 represents the influence of the particular surface geometry used. Both j and f have been experimentally measured for a great number of surface geometries. For a given geometry j and f vary with the Reynolds number, $4r_H G/\mu$, where $4r_H$ is the hydraulic diameter of the surface. Figure 4, shows how the ratio in equation 11 varies with the Reynolds for different geometries. The lower the value of the ratio, the lower the total heat transfer area necessary. The triangular flow passages, typical of the matrix surface Cercor used in gas turbine regenerators is poorer than screens or parallel plates. Surfaces with small spacings for air passage have low hydraulic diameter; this tends to give lower Reynolds numbers and smaller heat transfer areas. Surfaces with large friction factors give low values of G , see equation 10, for a given Δp . This will also lower the Reynolds number but at the expense of a larger frontal area.

For rotary regenerations the effectiveness, ϵ , is also a function of the speed of rotation. The nondimensional coefficient used is the ratio of the rotor heat capacity to the heat capacity of the air. The heat capacity is defined simply as the heat transferred to a body per unit time divided by the temperature change of the body due to the heat transfer.

$$\frac{C_{\text{ROTOR}}}{C_{\text{AIR}}} = \frac{(\text{RPM}) (\rho c)_{\text{ROTOR}} L A_{\text{FR}}}{G_T A_c (C_p)_{\text{AIR}}} \quad (12)$$

C is the heat capacity, c is the specific heat, A_{FR} is the frontal area, L is the depth of the rotor (in the direction of air flow).

Substituting for the product GA_c from equation 3 and using the geometric relationship,

$$L = r_H A/A_c \quad (13)$$

equation 12 can be rewritten as,

$$\frac{C_{\text{ROTOR}}}{C_{\text{AIR}}} = \frac{(A_{\text{FR}})}{A_c} \frac{(\text{RPM}) r_H A (\text{ITD}) (\rho c)_{\text{ROTOR}}}{q} \quad (14)$$

Figure 4

As this ratio increases the effectiveness increases for a given NTU. The ratio should be at least one for reasonable values of effectiveness. When the ratio reaches five further increases in its values have no influence on the effectiveness.

A numerical comparison of various surface geometries is shown in table II. Each geometry is used for a rotary heat exchanger for a 1000 MgW plant with a fixed air side fan power; other parameters fixed for all surfaces are listed in the table. Also, results for a conventional tube heat exchanger is given for comparative purposes. Note that the estimated surface costs for the rotary regeneration do not include costs of the structure, supports bearings, etc. for the rotor. If the surface costs are doubled to account for these factors, rotary heat exchangers still give much lower capital costs than conventional heat exchangers.

The parallel plate designs result in a low rotational speed; this is important to insure that most of the oil has a chance to drain from the surface. If the oil on the surface is thick, entrainment of the oil in the air stream may occur. Although stacked screens give a lower capital cost than parallel plates, the oil entrainment problem due to the high rotational speed and the geometry of the screens will discourage consideration of them for cooling towers.

If steel sheet is used for the discs, the maximum dimensions are limited to that normally available, approximately five feet.

TABLE II

<u>Surface</u>	<u>A_{Frontal}</u>	<u>RPM</u>	<u>A(total)</u>	<u>Estimated Cost/ft²</u>	<u>Estimated Surface Cost</u>	<u>L, Depth, ft.</u>
1. Parallel Plate Spacing = 1/2" Plate thickness=1/32"	$1.22 \times 10^5 \text{ ft}^2$	4.1	$1.4 \times 10^7 \text{ ft}^2$	20¢/ft ²	$\$2.8 \times 10^6$	2.45 ft.
2. Parallel Plate Spacing = 1/4"	$1.22 \times 10^5 \text{ ft}^2$	4.55	$1.26 \times 10^7 \text{ ft}^2$	20¢/ft ²	$\$2.52 \times 10^6$	1.23 ft.
3. Parallel Plate Spacing = 1"	$1.22 \times 10^5 \text{ ft}^2$	3.44	$1.67 \times 10^7 \text{ ft}^2$	20¢/ft ²	$\$3.34 \times 10^6$	5.7 ft.
4. Random Stacked Woven Screens 3 mesh, 21 gage	$3 \times 10^5 \text{ ft}^2$	57.5	$2.9 \times 10^6 \text{ ft}^2$	30¢/ft ²	$\$0.9 \times 10^6$	0.06 ft.
5. Random Screens NTU = 2 $\epsilon \approx 0.75$	$3.14 \times 10^5 \text{ ft}^2$	21.5	$7.7 \times 10^6 \text{ ft}^2$	30¢/ft ²	$\$2.31 \times 10^6$	0.15 ft.
6. Finned Circular Tubes Figure 10.82 B London & Kayes	3.05×10^5	-	9.2×10^6	\$1/ft ²	$\$9.2 \times 10^6$	0.52 ft.

For all surfaces

Plant Size = 1000 MgW

Rankine Cycle Thermodynamic Efficiency = 40%

$\Delta p = 4 \text{ lbm/ft}^2$

Fan Power = 20 MgW (17.4 MgW for surface #5)

ITD = 50°F

Ntu = 0.8

$\epsilon_r / C_{\text{Air}} = 2$

$\epsilon \approx 0.5$

III Analytical Approach

A Momentum Transfer

1. Velocity Distribution

To simplify the heat transfer analysis it is desirable to have the velocity of the air relative to the plates the same over all the airside plate surface. This allows the overall heat transfer coefficient to be a constant over the surface.

For turbulent flow the equation for the velocity in flow through channels neglecting entrance effects as given by Moody[2] can be reduced to

$$V = \sqrt{\frac{\Delta P D g_o}{f L \rho}} \quad 15$$

assuming P constant over the cross section and using the hydraulic diameter to extend the results to parallel plates. Where

ΔP = pressure drop across the channel, lbf/ft.²

D = distance between plates (one half the hydraulic diameter), ft.

g_o = conversion factor = 4.17×10^8 lbm ft/lbf hr²

f = Moody friction factor [2], dimensionless

L = channel flow length, ft.

ρ = density of air, lbm/ft.³

From equation 15 it can be seen that the velocity is inversely proportional to the square root of the flow path length. If the shroud directing the air flow were straight at the extreme radius as shown in figure 5 then the air velocity near the top of the

shroud would be much higher than that near the center of the discs. This would severely lower the effectiveness of the heat exchanger because more air would be going through the top where there is less area for heat transfer than would be going through the bottom. For optimum effectiveness it would be desirable to have more air pass through the lower area of the discs where there is more area for heat transfer. However, to make the heat transfer analysis more tractable the air is assumed to have a constant velocity over the cross-section. This can be achieved by either increasing the flow resistance in the upper regions by attaching the curved triangular sheets from which the discs were cut to the shroud, as shown in figure 6, or by designing the shroud so that the resistance of the flow through the shroud makes up for the shorter flow length near the top of the discs as shown in figure 7. In either case the velocity will be given by equation 5 with L equal to the diameter of the discs.

2 Oil Thickness

As the plates rotate through the oil layer into the air stream a thin coating of oil adheres to the metal. Because the oil preferentially wets the metal with respect to the water the oil coating continues to stick to the metal as the metal rotates through the water side of the exchanger. The oil coating serves to keep the plates from rusting when rotating through the water side and it keeps any water from adhering to the plates and being carried over to the air side to be evaporated.

However, as will be shown in the section of this chapter on heat

transfer the oil coating provides the major resistance to heat transfer on the water side and around twelve percent of the heat transfer resistance on the air side. For these reasons and to calculate the amount of oil necessary to operate the plant it is necessary to calculate the thickness of the oil layer adhering to the metal. The thickness is determined by balancing capillary, gravity, and shear forces. Levich[3] gives an equation for the thickness of a liquid coating, h_o , on a solid withdrawn from a liquid as

$$h_o = \left(\frac{\mu v}{\rho g}\right)^{1/2} f\left(\frac{\mu v}{\sigma}\right) \quad 16$$

$$\text{with } f\left(\frac{\mu v}{\sigma}\right) = .93\left(\frac{\mu v}{\sigma}\right)^{1/6} \quad \text{for } \frac{\mu v}{\sigma} < 1$$

$$\text{and } f\left(\frac{\mu v}{\sigma}\right) = 1 \quad \text{for } \frac{\mu v}{\sigma} \gg 1$$

where σ = surface tension, dynes

v = velocity of solid withdrawal rate from liquid, cm./sec.

ρ = density of coating, gm/cm³

h_o = coating thickness, cm.

μ = viscosity, centipoise

For a rotational velocity of ten RPM the maximum thickness which would occur at the outer edge of the five foot diameter discs would be .036 inches for Rubrex 100 oil, the oil used in the tests.

Figure 5

Figure 6

Figure 7

B Heat Transfer

1 Overall Heat Transfer Coefficient

The heat transfer rate equation can be stated as

$$\frac{dQ}{dA} = U(T_h - T_c) \quad 17$$

where A = surface area for heat transfer, ft^2

Q = heat flux, Btu/hr.

$\frac{dQ}{dA}$ = heat flux per unit transfer area at the location in the exchanger where the temperature difference is $(T_h - T_c)$, Btu/hr ft^2

T_h = local temperature of hot fluid, $^{\circ}\text{F}$

T_c = local temperature of cold fluid, $^{\circ}\text{F}$

U = overall conductance for heat transfer, Btu/hr $^{\circ}\text{F ft}^2$

The thermal resistance, $1/U$, is given by a sum of the series thermal resistances. For the oil coated plates under consideration these resistances consist of the following:

1) Conduction resistance from the interior of the plate to the surface. An upper bound of this resistance is given by

$$R_{\text{plate}} = \frac{t_{\text{plate}}}{2k_{\text{plate}}} \quad 18$$

2) Conduction resistance through the oil film coating the plate given by

$$R_{\text{oil}} = \frac{t_{\text{oil}}}{k_{\text{oil}}} \quad 19$$

3) The film resistance from the fluid to the oil surface on the plate given by

$$R_{\text{film}} = 1/h \quad 20$$

4) Because of the cleansing action of a periodic exchanger and the fact that the oil will be filtered there is no need for a scale heat transfer resistance.

where the above symbols mean:

$R_{\text{plate}}, R_{\text{oil}}, R_{\text{film}}$ = Thermal resistances of plate, oil, and film respectively, $\text{hr ft}^2 \text{ } ^\circ\text{F/Btu}$

$t_{\text{plate}}, t_{\text{oil}}$ = thickness of plate, and oil layer respectively in ft.

$k_{\text{plate}}, k_{\text{oil}}$ = Thermal conductivity of the plate, and oil, $\text{Btu/ft hr } ^\circ\text{F}$

For one thirty second of an inch thick steel plates the maximum plate resistance is

$$R_{\text{plate}} = .00005 \text{ hr ft}^2 \text{ } ^\circ\text{F/Btu} \quad 21$$

For the maximum oil thickness, which occurs near the outside radius of the disk as calculated in section A, the thermal resistance for oil with a conductivity of $.076 \text{ Btu/hr ft } ^\circ\text{F}$ is

$$R_{\text{oil}} = .039 \text{ hr ft}^2 \text{ } ^\circ\text{F/Btu} \quad 22$$

For flow in noncircular channels the McAdams correlation is used with the hydraulic diameter equal to twice the plate spacing.

Neglecting entrance effects, this gives:

$$h = .023 \frac{k_{\text{air}}}{2D} (\text{Re})^{.8} (\text{Pr})^{.4} \quad 23$$

where $\text{Re} = \text{Reynolds number} = \frac{V2D}{\mu}$, dimensionless

$\text{Pr} = \text{Prandtl number} = \frac{\mu c_p}{k_{\text{air}}}$, dimensionless

$\mu = \text{viscosity of air, lbm/hr ft}$

$c_p = \text{heat capacity of air at constant pressure, Btu/lbm } ^\circ\text{F}$

$k_{\text{air}} = \text{conductivity of air, Btu/hr ft } ^\circ\text{F}$

With an average airside velocity of 34.5 feet per second and a plate spacing of one third inch $h = 8$. Btu/hr ft² °F and therefore $R_{\text{film}} = .125 \text{ hr ft}^2 \text{ } ^\circ\text{F/Btu}$ for the air side film resistance. For the water side film resistance assuming the relative velocity of the plate to the water is only due to the speed of rotation of the plate equation 23 gives $h = 670$ so that $R_{\text{film}} = .00149$ for the water side.

Since the heat transfer resistance for conduction through the plate is much less than the film resistance the former can be neglected so that the overall heat transfer coefficient is given by

$$U = \frac{1}{R_{\text{oil}} + R_{\text{film}}} \quad 24$$

The value of U for the heat transfer between the air and the plate for an air velocity of 34.5 feet per second is $U_{\text{air}} = 7.88$ and that for the water side is $U_{\text{water}} = 33$. The water side is controlled by the oil thickness and will vary with speed of rotation of the plates and with oil viscosity. Since the water side heat

transfer coefficient is over four times that for the air side it can be seen that the air side will be the limiting side.

2 Optimum Fluid Level

Since the coefficient of heat transfer from the water to the plate is much greater than that from the plate to the air it is desirable to maximize the heated area of the plate which is exposed to the air. As can be seen in figure 8 the heated area of the plate exposed to the air is the area of the plates above the liquid level minus the area of the center circle which never gets heated by the water. An exact analytical derivation of the optimum water level is given in Appendix A. This level, which is a distance .303 times the disc radius down from the center gives an 11.9% increase in the air side heat transfer area.

3 Heat Exchanger Design

In order to design a heat exchanger an energy equation must be used along with the rate equation. The energy equation equates the enthalpy gain of the cold fluid with the enthalpy loss of the hot fluid. The two standard approaches for combining these equations to design a heat exchanger are the log mean rate equation method and the effectiveness NTU method. A comparison of both methods is given in reference 1. Mathematical difficulties arise when periodic exchangers are considered. Therefore computer solutions must be resorted to. London[1] compiles effectiveness versus NTU graphs

from the computer results of Lambertson [4] and Bahnke and Howard [5]. However, these results are not of use in the periodic cooling tower under consideration here. The above referenced solutions were for porous matrix surfaces in which the air was blown through the rotating material rather than parallel to its surface. As a result all the heated flow subchannels are the same length in the referenced solutions and the solution is more straight forward. Here, starting with the energy and rate equations a method similar to the NTU method will be derived and solved numerically on a computer.

The periodic flow cooling tower is actually two separate heat exchangers coupled together by the rotating material. Figure 9 shows the temperature distribution along the horizontal flow path of the periodic tower. If the metal plates are rotated fast enough the change in temperature of the plates will be small compared to the change in temperature of the air and water passing through the tower. Therefore the plate temperature will be nearly a constant as shown in the figure. The plates will be assumed to have a constant temperature in the analysis which follows.

Since the air side is controlling it will be considered first. The length of the heated surface the air passes varies with the vertical position as shown in figure 8. The shaded channels in the figure represent the heated portions of typical incremental flow subchannels of height dh . The width of the subchannels is equal to the spacing between the plates D . The heated length, L , and height of the subchannels can be found from geometry to be

Figure 8

$$L = 2 r \sin\theta \quad 180^\circ > \theta > 107.6^\circ \quad 25$$

$$L = 2 r (\sin\theta - \sqrt{.0918 - \cos^2\theta}) \quad 72.3^\circ < \theta < 107.6^\circ$$

$$dh = r \sin\theta \, d\theta$$

Where θ is measured from the bottom as shown in figure 8. When θ is greater than 107.6° the entire channel is above the unheated region.

Equating the heat flux in the energy equation to that in the rate equation for an increment of length dx gives

$$\rho \, c_p \, V \, dA_c \, dT_c = U \, dA \, (T_p - T_c) \quad 26$$

where ρ = air density, lbm/ft^3

c_p = air heat capacity, $\text{Btu/lbm}^\circ\text{F}$

V = air velocity, ft/hr

dA_c = crosssectional area for air flow = $D \, dh$, ft^2

dT_c = change in the air temperature while traveling the length dx , $^\circ\text{F}$

T_p = plate temperature, $^\circ\text{F}$

dA = area for heat transfer = $2 \, dh \, L$

integrating gives

$$\ln\left(\frac{T_p - T_{c1}}{T_p - T_{c2}}\right) = \frac{U \, dA}{\rho \, c_p \, V \, dA_c} \quad 27$$

taking the exponential and rearranging gives

$$T_{c2} - T_{c1} = (1 - e^{-\left(\frac{2 \, L \, U}{V \, c_p \, D}\right)}) (T_p - T_{c1}) \quad 28$$

The average value of the temperature increase of the air in passing through the tower for the assumed uniform velocity is given by

$$\begin{aligned}
 \overline{(T_{c2} - T_{c1})} &= \frac{\int_0^H (T_{c2} - T_{c1}) dh}{\int_0^H dh} \\
 &= \frac{(T_p - T_{c1})}{H} \int_0^H (1 - e^{-\left(\frac{2 L U}{\rho V c_p D}\right)}) dh
 \end{aligned} \tag{29}$$

It can be seen that the integral divided by the height of the shroud at $\theta=180^\circ$ above the liquid level, H , is the effectiveness for the air side.

$$\epsilon_a = \frac{\int_0^H (1 - e^{-\left(\frac{2 L U}{\rho V c_p D}\right)}) dh}{H} \tag{30}$$

Where ϵ_a = air side effectiveness = $\frac{T_{c2} - T_{c1}}{T_p - T_{c1}}$, dimensionless

so that $\overline{(T_{c2} - T_{c1})} = \epsilon_a (T_p - T_{c1})$ 31

The above integral can not be integrated analytically but the dependence of L and dh on θ were accounted for in a numerical integration for different values of the nondimensional parameter $NTU^* = \frac{4 r U}{\rho V c_p D}$.

The results of these integrations are plotted in figure 10 as effectiveness versus NTU^* . NTU^* is a modified value of NTU taking into account that the flow length is different for different subchannels. If all subchannels were of the maximum length $2 r$ then it can be seen that NTU would equal NTU^* .

$$NTU = \frac{A U}{c_{min}} = \frac{2 (2 r H) U}{\rho V c_p D H} = \frac{4 r U}{\rho V c_p D} = NTU^* \tag{32}$$

where $2 r$ = the diameter of the discs and the first 2 in the numerator comes from the fact that there are two sides transferring heat into every subchannel. A fifth order least squares curve fit was obtained

Figure 10 AIR SIDE EFFECTIVENESS VS. NTU*

for ϵ_a with values of NTU* between .1 and 4. The equation is:

$$\epsilon_a = .004367 + .684(NTU^*) - .1992(NTU^*)^2 + .01281(NTU^*)^3 + .006146(NTU^*)^4 - .001 (NTU^*)^5$$

The total heat transferred from one plate to the air is given by:

$$Q/\text{plate} = V c_p D H \epsilon_a (T_p - T_{c1}) \quad 33$$

Now for the water side the assumption that $T_h - T_p \gg \Delta T_p$ is not valid as was the corresponding relation for the air side. However if the plate is assumed to have its average $\bar{T}_p$ an estimate of the log mean temperature difference, if the plate were made up of equal flow length paths with fluid flowing in counterflow, show that the error in assuming the plate has its average temperature is only around 5%. From Figure 9 it can be seen that although $T_{h1} - T_{p2}$ on the right side is less than $T_{h1} - \bar{T}_p$, on the left side $T_{h2} - T_{p1}$ exceeds $T_{h2} - \bar{T}_p$ so that effects cancel out to within 5% for the temperatures of interest with a ΔT_p equal to as much as 5 °F. For higher rotational velocities and therefore lower ΔT_p the error is even less.

Now Proceeding in the same manner as for the air side we have the length and height of elementary subchannels given by

$$\begin{aligned} L &= 2 r \sin\theta & 0 < \theta < 72.3^\circ \\ dh &= r \sin\theta d\theta \end{aligned} \quad 34$$

The energy and rate equations are the same as for the airside with all

velocities, properties and temperatures referring to water instead of air. The derivation proceeds analogously to that for the air side and terminates with an equation similar to equation 33 written for the water side:

$$Q/\text{plate} = \rho V c_p D (T_{h1} - \bar{T}_p) \int_0^{H_w} (1 - e^{-\frac{2 L U}{(\rho V c_p D) h}}) dh \quad 35$$

The integral in equation 35 divided by $H_w = 2r - H$ has also been evaluated numerically as a function of $NTU^* = \frac{4 U r}{\rho V c_p D}$. It is shown

in figure 11. A fifth order least squares curve fit was found to be:

$$\epsilon_w = .00459 + .655(NTU^*) - .187(NTU^*)^2 + .00942(NTU^*)^3 + .00684(NTU^*)^4 - .00107(NTU^*)^5$$

For given air and water velocities and plate radius and spacing equations 33 and 35 may be rewritten as:

$$Q/\text{plate} = B(\bar{T}_p - T_{c1})$$

$$Q/\text{plate} = C(T_{h1} - \bar{T}_p)$$

where $B = \rho V c_p D H \epsilon_a$ for air

$C = \rho V c_p D H_w \epsilon_w$ for water

Q/plate = heat rejected per plate, Btu/hr plate

Equating the Q/plate and solving for $\bar{T}_p$ gives

$$\bar{T}_p = \frac{CT_{h1} + BT_{c1}}{B + C} \quad 37$$

Substituting this back into equation 36 gives the value of Q/plate in terms of the inlet temperatures, velocities and plate radius and spacing.

as

$$Q/\text{plate} = \frac{B C}{B + C} (T_{h1} - T_{c1}) \quad 38$$

Equation 38 may be rewritten as

$$Q/\text{plate} = C_{\text{air}} \epsilon_t (T_{h1} - T_{c1})$$

where $C_{\text{air}} = \rho V c_p$ H D = heat capacity flow rate of air and

$$\epsilon_t = \frac{\epsilon_a C}{B + C} = \text{Total effectiveness} = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} \quad \text{for } C_{\text{air}} = C_{\text{minimum}} \quad 39$$

substituting into equation 39 gives

$$\epsilon_t = \frac{\epsilon_a C_{\text{water}} \epsilon_w}{C_{\text{air}} \epsilon_a + C_{\text{water}} \epsilon_w} \quad 40$$

For the limiting case when $C_{\text{water}} \gg C_{\text{air}}$ and expression for ϵ_t can be derived as follows: the heat transfered per unit time to the air is

$$q = C_a (T_{c2} - T_{c1}) \quad 41$$

and since for this limiting case, $T_{h1} = T_{h2} = T_h$, the heat transfered per unit time from the water is

$$q = U_w A_w (T_h - T_p) \quad 42$$

equating the q's and rearranging yields

$$T_h - T_p = \frac{C_a (T_{c2} - T_{c1})}{U_w A_w} \quad 43$$

by definition,

$$T_h - T_{c1} = (T_p - T_{c1}) + (T_h - T_p)$$

and by substituting the equivalent of $(T_p - T_{c1})$ from equation 31 and using equation 43 for $(T_h - T_p)$, yields

$$T_h - T_{c1} = \frac{T_{c2} - T_{c1}}{\epsilon_a} + \frac{C_a}{U_w A_w} (T_{c2} - T_{c1}) \quad 44$$

substituting equation 44 into equation 39 results in

$$\epsilon_t = \frac{T_{c2} - T_{c1}}{(T_{c2} - T_{c1}) \left(\frac{1}{\epsilon_a} + \frac{C_a}{U_w A_w} \right)} = \frac{1}{\frac{1}{\epsilon_a} + \frac{C_a}{U_w A_w}} \quad 45$$

The number of plates, N, needed is found from the total heat rejection rate of the plant, Q_{total} to be

$$N = \frac{Q_{total}}{Q/plate} \quad 46$$

The total air flow, F, for the tower in cubic feet per hour is given by

$$F = N D H V \quad 47$$

The work done by the fans in driving this air is given by

$$W = \Delta P F \quad 48$$

where ΔP is found from equation 15. The change in temperature of the plates in passing from the hot water to the cool air is given by

$$\Delta T_p = \frac{Q/\text{plate}}{\rho_s r^2 t \pi c_{ps} \Omega 60}$$

49

Where Ω = rotational velocity, RPM

t = thickness of plate, ft

ρ_s = density of steel, lbm/ft³

c_{ps} = heat capacity of steel, Btu/lbm °F

The temperature increase of the air from an energy balance is

$$T_{c2} - T_{c1} = \frac{Q_{\text{total}}}{\rho c_p F}$$

50

4 Effect of Matrix Speed

Equation 49 shows the average plate temperature change as a function of rotational velocity. The important effect of this plate temperature change is to lower the effective ΔT for the subchannels. The higher the plate temperature change the lower is the ΔT . This effect can be seen in figure 9 where the dashed line is the extreme case for very slow velocities and the line marked $\overline{T_p}$ is the extreme case for very high velocities.

As the rotational velocity is increased from zero the effectiveness of the exchanger increases. However when the velocity becomes high enough so that the plate temperature change is small compared to the plate air temperature drop, then further increases in the rotational velocity fail to substantially change the effectiveness.

London [1] suggests using the following relationship between the effectiveness and rotational velocity after the preliminary exchanger

design is completed.

$$\epsilon \approx \epsilon_{\text{counterflow direct type}} \left(1 - \frac{1}{9(C_r/C_{\min})^{1.93}} \right) \quad 51$$

where $C_r = \pi r^2 t \rho_s c_p \Omega / 60$ for steel

and $C_{\min} = \rho V c_p D H$ for air

Although this relation is for a periodic exchanger with constant length flow channels the important effect which it describes is the change in temperature of the matrix relative to that of the air which is similar for all periodic heat exchangers so equation 51 should still give a good approximate relation for determining when the rotational velocity is affecting the efficiency.

From equation 51 it can be seen that operating at higher speeds allows the use of thinner plates. For sheet steel the cost is roughly proportional to the thickness of the plate so that higher velocities and thinner plates mean lower capital costs. However if the plate velocity becomes too high the oil may get churned up into the water and its coating thickness will increase and lower the heat transfer coefficient. Also there is a limit to how thin the plates can be and still support themselves without excessive buckeling and resulting channel blockage.

C Mass Transfer

The value of the periodic tower over the wet cooling tower stems from its evaporation suppression of water. However if large quantities of oil are evaporated into the air the environmental damage may well exceed that caused by wet towers. Therefore it is desirable to calculate the evaporation rate from the periodic tower.

For wetted-wall evaporation Rohsenow [6] gives the following equation

$$w/A = h_d(C_{ao} - C_{a\infty})$$

where C_{ao} = mass concentration of vapor at the wall, lbm/ft³

$C_{a\infty}$ = mass concentration far from the wall, lbm/ft³

w = mass evaporation rate, lbm/hr

h_d = mass transfer coefficient

For the case of oil the initial oil concentration in the atmosphere is zero so $C_{a\infty} = 0$. C_{ao} is given by the perfect gas law as

$$C_{ao} = \frac{P}{R T_r}$$

and h_d is given by the Gilliland relation as

$$h_d = .023 \frac{d' (V/2D)^{.83}}{2D} \left(\frac{\mu}{\rho d'} \right)^{.44}$$

where d' is the diffusivity given by

$$d' = .0069 \frac{T_r}{P_o (T_a^{.33} + T_b^{.33})^2} \sqrt{\frac{1}{M_a} + \frac{1}{M_b}}$$

Where T_r = temperature in °Rankine

P_o = pressure in atmospheres

T_a, T_b = atomic volumes

M_a, M_b = atomic mass numbers

The vapor pressure for Rubrex 100 the oil used in the tests is 10^{-4} , that for Rubrex 200 is 10^{-5} , and less than 10^{-8} lbf/in² for a silicone based oil. Assuming a chemical formula of $C_{22}H_{46}$ for Rubrex to obtain values of T_a and M_a allows us to calculate $d' = .16$ for Rubrex. Calculating C_{ao} from the vapor pressures gives $C_{ao} = 5 \times 10^{-6}$ lbm/ft³ for Rubrex 100 and C_{ao} less than 10^{-10} lbm/ft³ for the silicone oil. With an airside area of 13×10^6 ft² this gives an evaporation rate of 18,000 lbm/hr for Rubrex 100, 1,800 for Rubrex 200 and less than 1 lbm/hr for a silicone oil. If a fossil plant burns 1% sulfur fuel the amount of sulfur dioxide put into the air is 8,400 lbm/hr. Therefore although the unburned hydrocarbon is probably less harmful than sulfur dioxide it is definitely a problem to consider since the discharge rate can be as much as twice that of SO_2 for Rubrex 100.

The solution to the problem obviously lies in using an oil with a higher flash point or using a silicone oil for which the evaporation rate is negligible.

IV Experimental Technique

A model was built to determine the evaporation rate and the performance characteristics of a periodic cooling tower.

A System Configuration

The discs in the model were each one foot in diameter and made from one thirty second of an inch thick galvanized steel sheet. In the first set of tests the discs were spaced three eights of an inch appart and ten discs were used. To increase the air side effectiveness in the second set of tests the separation between discs was reduced to one quarter of an inch and thirteen discs were used. Because of bowing in the sheet steel, spacers were put at the five locations shown in figure 12 to keep the separation close to nominal. The spacers were made from three eights of an inch diameter cylinders. The height of the cylinders was the spacing desired for the set of tests. Both the spacers and the plates were drilled with a quarter inch hole and quarter inch threaded rod inserted through the plates and spacers as shown in figure 12. Nuts were tightened on the ends of the threaded rod which protruded through the plate-spacer-plate assembly to squeeze the plates and spacers tightly together as shown in figure 12. The maximum tolerance in spacing due to bowing of the plates between spacer locations was approximately 25% of the spacing distance. The clearance from the plates to the wall for the outside plates was half the plate spacing.

The water and oil were contained in a plexiglass trough 22 inches

Plate showing spacer location

Figure 12

long by 4 and one half inches wide by 7 inches deep. The bearings which supported the disc assembly rested in a groove on the top of the side walls of the trough as shown in figure 13. The threaded rod which ran through the center of the discs served as the axle for rotation. It extended through the bearings and was connected to a variable diameter pulley. The pulley was connected by a rubber belt to an electric motor. By changing the pulley diameter of the belt the speed of rotation could be changed.

The water was heated by four 500 watt Chromalox cartridge immersion heaters. The heaters were mounted through the holes in the walls of the trough as shown in figure 13. Each heater was cylindrical with a diameter of 2 and one sixteenth inches and a length of one and three quarter inches. The heaters were connected in parallel and the power controlled and measured by a variac with a built in voltmeter and ammeter. To prevent over heating in case of an unattended accident when left running over night a thermostat was installed. A temperature greater than 170 °F would cause the thermostat to open a relay and cut off all power to the heaters. This temperature limit was to prevent the plexiglass from softening and collapsing . Figure 14 shows the electrical circuitry schematic.

Air flow was provided by a centrifugal blower. The blower outlet was connected by a plexiglass channel to the plates as shown in figure 15. An even air flow distribution was obtained by placing layers of fine mesh wire screen between the blower and the inlet of the channel. The free flow length from the screen to the plates was one and a half feet.

The exhaust air from the discs was channeled through a drift

Figure 13

Figure 124

eliminator shown in figure 15 and 16. It turned out that the drift eliminators were unnecessary because no oil was entrained at the low air velocities which the blower produced.

B Flow and Temperature Monitoring

After passing through the drift eliminator the air was sent through a 3 inch pipe and a standard 2 inch orifice plate with flange pressure taps to measure the air flow rate. The pressure drop across the orifice plate was measured with a standard U tube manometer with water as the working fluid. The pressure drop across the plates was measured with an inclined manometer accurate to hundredths of an inch of water. The pressure taps for measuring the pressure drop across the plates were located on the top of the flow channel a distance three inches before and after the plates.

Copper constantine thermocouples were used to measure temperatures. The temperatures of the inlet air and water were each measured by a single thermocouple. The inlet air thermocouple was located after the blower and screen and before the plates. The water thermocouple was located in the trough directly below the plates in the center of the trough. This was the location of maximum water velocity. However since the exhaust air near the axle passes through a longer heated channel than the air passing across the top of the discs there is a temperature gradient in the exhaust air. Therefore the temperature was measured at seven different vertical heights each one half inch apart by the thermocouple rake shown in figure 17. The rake was moved to four

Figure 15

Drift Eliminator

Figure 16

Figure 17

different horizontal positions producing a cartesian grid of twenty eight temperature measurements. The above thermocouples were wired to a switch for readout singly at a potentiometer with ice bath junction. For the second set of tests three thermocouples were also mounted in the plates at radial locations of 1,3, and 5 inches. The thermocouples were connected through a switch and slip rings as shown in Figure 18 to a Kintel amplifier. The output of the amplifier was fed to a Sanborn 320 strip recorder so that the variation of temperature of the plates could be recorded as they rotated from the water to the air.

The dimensions and shape of the shroud are shown in figure 19. The shroud was sealed to the trough with plasticine to insure that all the air which left the blower passed through the plates and was measured at the orifice plate. The shape of the shroud did not produce a uniform vertical velocity distribution. The velocity distribution at the location of the thermocouple rake was measured with a pitot tube and the inclined manometer to determine the velocities to weight the temperature measurements with to determine an average temperature.

A small pump was used in one test to circulate the oil from the left side of the trough to the right side of the trough.

Figure 18

Figure 19

V Experimental Results

Tests were run on the model at different heat rejection rates, different rotational velocities, and different plate spacings. The purpose of the tests was to determine how effectively evaporation was suppressed, determine the heat transfer relationships for the model, and to demonstrate the feasibility and locate any unforeseen problems in such a design.

A Oil Characteristics

The oil used for the tests was a parafinic base nonadditive mineral oil. Detergent additives such as found in automobile motor oils are not desirable because they tend to cause emulsification of the oil with the water. Mobil's Rubrex 100 brand was used and can be purchased for around forty-one cents per gallon if bought in tank truck loads.

The drag force on the oil layer caused by the rotation of the circular discs tends to push the oil to the side where the discs leave the water as shown in figure 20. A certain height of oil on the right side of figure 20 is necessary to provide sufficient head to drive enough oil between the plates to keep the left side covered with oil. The height necessary depends upon the plate spacing, the viscosity of the oil, and the rotational velocity of the discs. For the model tested at a rotational velocity of 6.75 RPM around one half of an inch of oil was sufficient to insure coverage of the left side. Coverage of the water on the left side of the plates with oil is necessary only

Figure 20

to eliminate the evaporation from the water surface. Only coverage of the water surface to the right of the center axis of rotation is necessary to insure that no water is carried over with the plates.

In a large plant of this design the oil pushed to the right side would be pumped from the right side and sent through an oil filter and then sent back to the left side of the discs. The oil filter would remove any dust or soot deposited in the oil by the air. Since the oil was being pumped externally from the right side to the left side a thick layer of oil would not be needed on the right side. When an external pump was added to the model a one eighth inch thickness of oil was sufficient to insure complete coverage of the water surface.

The viscosity of oil varies with temperature. The lower the viscosity the thinner the film of oil which coats the plate and the smaller the heat transfer resistance of the oil film. If the viscosity of the oil is too high some small beads of water can get trapped in the oil film coating the plate and get carried over into the air stream. It was found that at operating temperatures around 130 °F where Rubrex 100 had a viscosity of around 10 centipoise, that the viscosity was sufficiently low so that no beads of water were carried over. With the water at a room temperature of 80 °F the oil had a viscosity of around 21 centipoise and some water beads were carried over. If lower operating temperatures are desired then a different oil with a lower viscosity should be used.

B Evaporation Rates

Tests were run both with and without the oil layer to observe the effect on the percent of heat transfer due to evaporation. Without any oil and operating at a heat rejection rate of 955 watts all the heat transfer was due to evaporation because the temperature of the water was less than the dry bulb temperature of the inlet air. In the test four pounds of water evaporated per hour. With the oil layer the water evaporation was less than .016 pounds per hour for the same heat rejection. This represents a reduction in water consumption by a factor of 250. Less than four tenths of one percent of that without the oil.

It is probable that the actual water evaporation rate is less than that measured in the extended duration test. In order to measure the .016 pounds per hour which is the most that could possibly have been evaporated the model was run over night for a sixteen hour duration test. Then in the morning the drop in the liquid level was just enough to be measurable and the weight of liquid loss was calculated from the known area of the tank. However in running the tests all night some oil which coated the plates flowed down the plate surface to the axle and then flowed along the axle through the bearings and down the outside of the tank forming a puddle on the table. No means was discovered to stop this very slow leak rate. It is probable that all the measured loss in fluid level was due to this oil seepage and not to evaporation of water. The evaporation of the oil for this small model for sixteen hours is infinitesimal. In all the heat transfer tests which were of shorter duration there was no noticeable fluid loss.

C Momentum Transfer Results

Two separate sets of tests were performed. In the first set ten discs were used spaced three eighths of an inch apart. In the second set of tests thirteen discs were used spaced one quarter of an inch apart.

In the first set of tests the mass flow rate as measured by the orifice plate and U tube manometer was 410 lbm/hr. With the three eighths of an inch spacing this resulted in a Reynolds number based on hydraulic diameter of 3,600. The pressure drop across the plates as measured by the inclined manometer was $.20 \pm .05 \text{ lbf/ft}^2$. This corresponds to an average velocity of ten feet per second through the plates.

In the second set of tests the mass flow rate was 390 lbm/hr which resulted in a Reynolds number of 2,290. The pressure drop across the plates was higher for the closer spacing and amounted to $.26 \pm .05 \text{ lbf/ft}^2$.

In the first set of tests the Reynolds number is above the transition Reynolds number so the flow should initially be laminar and then develop into turbulent flow. If the flow were completely laminar the pressure drop including entrance effects would have been calculated to be $.22 \text{ lbf/ft}^2$. Data for the entrance effects on turbulent flow friction factors in the low Reynolds number which occurred could not be found in the literature. If the ratio of the average turbulent friction factor including entrance effects to the fully developed numbers friction factor as given in the literature for Reynolds numbers greater than 10^4 is extrapolated down to $Re = 3600$ and multiplied by the fully

developed friction factor then the resulting pressure drop is calculated to be $.15 \text{ lbf/ft}^2$. Since the error bars are so large on the measured pressure drop both the laminar and turbulent calculated values are within the experimental error bars. The results are shown in table 3

In the second set of tests the Reynolds number is just below the transition Reynolds number so the flow should be laminar. Langhaar[7] gives a chart for the average friction factor for laminar flow in the entrance region of circular tubes. With the equivalent diameter of twice the plate spacing Langhaar's results give $f_{\text{avg}} \frac{L}{D} = .6$. Using this value in equation 1 gives for the calculated pressure drop $.256 \text{ lbf/ft}^2$. which is within the error limits of the measured value.

Table 3

	<u>experimental</u> <u>Pressure drop</u>	<u>analytical</u> <u>pressure drop</u>
set I	$.20 \pm .05 \text{ lbf/ft}^2$	$.22 - .35 \text{ lbf/ft}^2$
set II	$.26 \pm .05$	.256

D Heat Transfer Results

The results of seven of the the tests run are shown in table 4

Table 4

Test	T_{c2}	T_{c1}	mc_p	Q_{total}	T_{h1}	$\epsilon_{measured}$	RPM	D	$h_{measured}$
set I									
1	104	90	98.6	1376	134	32	6.75	.375"	4.4
2	109	92	98.6	1705	147	31	6.75	.375"	4.2
3	101	90	98.6	1095	125	32	6.75	.375"	4.4
4	108	88	98.6	1820	144	33	11.0	.375"	4.6
set II									
1	117	96	94.5	2060	146	43	6.75	.25"	5.0
2	117	98	94.5	1823	141	44	6.75	.25"	5.1
3	115	103	94.5	1080	132	41	6.75	.25"	4.7

where T_{c2} = Average temperature of outlet air °F

T_{c1} = temperature of inlet air °F

$\Delta T_c = T_{c2} - T_{c1}$, °F

mc_p = heat capacity flow rate of water Btu/hr °F

Q_{total} = Heat rejected to air Btu/hr

T_{h1} = water temperature °F

$\Delta T_{max} = T_{h1} - T_{c1}$

$\epsilon_a = \frac{\Delta T_c}{\Delta T_{max}} \times 100\%$

RPM = rotational velocity of discs, revolutions/minute

In test set one with the three eighths inch spacing the air is in turbulent flow. Kroll [8] finds that for turbulent flow of air in the

entrance region that

$$\frac{h_{\text{avg}}}{h_{\text{fully developed}}} = 1 + \left(\frac{2D}{L}\right)^{.7}$$

using this equation and the McAdams correlation for the fully developed heat transfer coefficient gives an analytical value of 4.0 Btu/hr ft² °F for h_{avg} . Using this analytical value for h_{avg} gives a value for NTU* equal to 4.75 which gives a calculated value of 29 for ϵ_a .

Test set two with a Reynolds number of 2,290 is in the laminar entrance region. Sparrow [9] gives results for average nusselt number for laminar flow in the entrance region between parallel plates. Using his results gives a heat transfer coefficient equal to 4.2 for h_{avg} . This analytical value of h_{avg} gives a value for NTU* equal to .675 which gives a calculated value of 39 for the effectiveness.

Starting with the measured values for effectiveness and working backwards to get a measured NTU* and from that deriving a measured h_{avg} gives the values listed in table 5.

Table 5

		Analytical	Experimental
set I	ϵ	29	32 $\pm$ 2
	h_{avg}	4.0 Btu/hr. ft ² °F	4.4 $\pm$.3 Btu/hr. ft ² °F
set II	ϵ	39	43 $\pm$ 2
	h_{avg}	4.2 Btu/hr. ft ² °F	5.0 $\pm$.3 Btu/hr. ft ² °F

From table 5 it can be seen that the analytical effectivenesses are within ten percent of the experimental ones. The reasons that the analytical values are more conservative than the experimental values are because of heat transfer to the air from the free oil surface which is not treated analytically and because the shroud is not designed for constant velocity airflow as was assumed in the analysis. The area of the free oil surface is 7.5% of the air side plate surface area for test set one and 5.7 % of the air side plate area for test set two. Since the shroud is not similar to either figure 1,2,or 3 it is difficult to tell whether the resulting flow distribution increases or decreases the effectiveness.

The fourth test in test set one was performed at a rotational velocity 1.6 times that for the other tests in set one. This fact lowered the average temperature drop of the plate in passing through the air stream from 4.4 °F to 2.7 °F. This had the effect of increasing the average plate temperature by 1 °F. If $\Delta T_{\max}$ is therefore increased by one degree to calculate the measured air side effectiveness it lowers the effectiveness to 32% which is equivalent to that for the other tests in set one. It is seen that for the heat rejection rate of test four in set one that the rotational velocity is already high enough at 6.75 RPM so that increasing the rotational velocity has a very small influence on the effectiveness.

The sanborn recorder produced graphs such as the one shown in figure 21 for the temperature variation of the plate with time. The average double amplitude of the oscillation was $2.6^{\circ}\text{F} \pm .5^{\circ}\text{F}$

Figure 21

which checks with the value of 2.5 which is the temperature change of the plate calculated from a heat balance for the test in which the measurement was taken.

VI Computer Design Program

A Objective

The larger the diameter of the discs are the more compact will be the heat exchanger. However, the largest width in which sheet steel is commercially available is 5 feet. Therefore the diameter of the discs will be five feet. With the diameter specified it was shown in chapter III that a periodic cooling tower could be designed for a given set of the five design variables:

1. D - the distance between plates, ft.
2. T_{h1} - the water temperature at the tower inlet, °F
3. T_D - the design ambient air temperature, this is the value of T_{c1} at which the power plant can deliver its fully rated power, °F
4. V - the air velocity through the tower, ft/hr
5. V_w - the water velocity through the tower, ft/hr

The objective of the computer program is to choose the best set of these five variables so as to minimize the total yearly incremental cost of operation of a power plant using the periodic cooling tower.

To minimize the entire plant's yearly incremental operation cost the effects of the cooling tower's performance on the rest of the power plant must be considered. An example of one of the economic trade-offs that must be considered is that of turbine outlet water temperature. The amount of heat rejected is proportional to the surface area of the cooling tower times the temperature

difference between the inlet air and the inlet water. Therefore with a given ambient air temperature the higher the water temperature the less surface area is needed. The smaller the surface area is the lower will be the capital cost of the cooling tower.

However, a higher water temperature causes a lower Rankine cycle efficiency, and a lower cycle efficiency requires a larger boiler capacity, greater fuel consumption, and more heat rejection.

The incremental cost of a periodic cooling tower will be determined to a large degree by the climate and geographic location of the power plant. So that the program may be used as a design tool in evaluating the cost of a periodic cooling tower at different plant locations, the local parameters such as, climate data, steam supply system cost, fuel cost, plant size, interest rates, base thermal efficiency, turbine heat rate vs. back pressure and land costs are left as input data.

The tower is first minimized for a given design ambient air temperature and then its performance is evaluated at the varying ambient air temperatures actually observed. The design temperatures are then varied to obtain the optimum design temperature. The optimization logic used in the program was the same as that of Andeen[10] and is explained in detail in that reference. Andeen's program optimized the cost of a conventional dry cooling tower which is a function of six design variables. To make his program optimize a periodic tower required extensive modifications causing more than half of the fortran statements to be rewritten.

The resulting periodic tower optimization program is listed in Appendix D.

B. Assumptions

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

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

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

4. The yearly cost of operation is the sum of

- the interest on capital costs (fixed charge rate of 15.8%)
- the cost of operating the required fans and pumps
- the cost of additional fuel required due to decreased thermal efficiency
- maintenance costs
- the cost of lost capacity at off design temperatures

5. Capital costs are broken down into

- cost of heat exchanger surface ($\$.4/\text{ft}^2$)
- cost of motors for fans and pumps ($\$30/\text{hp}$)
- cost of the fan blades ($\$7.5/\text{ft}^2$ area covered)
- cost of increased boiler capacity due to lower thermal efficiency ($\$45/\text{kW}$)
- land costs ($\$11,600/\text{acre}$)

6. Maintenance cost is expressed as a per cent 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 6. A listing of the order and format in which to input the data is given in Appendix B.

The land area used in determining land costs was the area directly under the plates assuming no stacking of plates. This cost is low in that it doesn't consider the land under the fans and other structural equipment but this may be balanced by the fact that the rows of disks may be stacked as in Figure 2.

Table 6

<u>Values for Input Data</u>		
<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
Maintenance Cost of cooling systems	Derived from Con Edison Data	1.09% of capital
Boiler Costs	Babcock & Wilcox	\$45/KW
Electric Motor Costs	Local Suppliers	\$30/HP
Fan Propeller Costs	Industrial Air Marley Company	\$7.5/ft ²
Fan Air Velocity	Industrial Air Marley Company	150 ft/sec
Surface cost	Assumed equal to twice material cost	\$.4/ft ²
Land Cost	Con Edison	\$11,600/Acre
Capacity Factor	Assumed	75 %
Thermal Efficiency	Assumed (for plant Operating with 3.5 inch Hg back pressure)	40%
Fan and Pump Efficiency	Assumed	85 %
Temperature Data	U.S. Weather Bureau 10 year, August for J.F.K. Airport	

C Results of Computer Program

Nonuniform air flow through the periodic tower may be caused by the local wind conditions. Since flow abnormalities reduce the effectiveness of the heat exchanger it is necessary to consider them in designing a cooling tower. Andeen[10] shows that if a cooling tower is modeled as a cylinder that the pressures caused by a 20 mph wind 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 the point 90° from the wind direction. Since flow rate is proportional to the square root of the pressure differential a heat exchanger designed for 4 lbf/ft^2 minimum pressure drop experiencing 20 mph wind would have approximately a 10 % increase in flow at the stagnation point and a 50 % reduction in flow at 90° from the wind direction.

Computer runs which were performed without any minimum pressure drop restriction optimized at a pressure drop of $.75 \text{ lbf/ft}^2$. This design would experience a reversal of flow at 90° from the wind direction and zero flow at some point between the stagnation point and the 90° point on a 20 mph wind velocity day.

Figure 22 shows the effect of different minimum pressure drop requirements on the cooling tower incremental cost. Figure 23 shows the effect on cooling tower capital costs of the minimum pressure drop requirement. From the figures it can be seen that, increasing Δp from zero causes the capital cost to rise along with the fan power costs. This is because although the velocity, heat

Design Temperature
COOLING TOWER INCREMENTAL COST VS. DESIGN TEMPERATURE
For Different Minimum Air Side Pressure Drops
Figure 22

COOLING TOWER CAPITAL COST VS. DESIGN TEMPERATURE
For Different Minimum Airside Pressure Drops
Figure 23

transfer coefficient, and effectiveness increase the plate spacing and therefore mass flow of air per plate decreases and so the heat rejection per plate decreases causing more plates to be needed.

The capital cost of the periodic surface is hard to determine. Although one thirty second of an inch thick galvanized steel sheet can be purchased for \$.20/ft² the cost of manufacturing and assembling the discs is unknown. If the cost of manufacture and assembly is expressed in a per square foot basis then the total cost of the surface, Surf, will equal the material cost plus the assembly cost. Figure 24 shows the effect of surface cost on the incremental cooling tower cost. Figure 25 shows the same effect on the capital cost. All the optimizations in figure 22 and 25 were performed at a minimum pressure drop requirement of 4 lbf/ft².

Therefore if the rule of thumb that material costs roughly equal manufacture costs and a minimum pressure drop of 4 lbf/ft² is used then the variation of the optimum incremental cost and capital cost with design temperature is shown in figure 26. For this optimum design the design variables and physical characteristics are listed in table 6. The optimum periodic exchanger costs are compared to those of optimum finned tubes in table 8.

A listing of the output symbols and their meaning is given in Appendix C. From the results of the computer optimization the prospects for a periodic cooling tower look optimistic.

Table 7

Optimum design for surf = .4 and $\Delta P = 4$

Plate spacing = .33 inches

Air velocity = 34.5 ft/sec

Water velocity = .976

Air design temperature = 75 °F

Water inlet temperature = 127 °F

Heat rejected per plate = 4720 Btu/hr

Number of plates = 1.12×10^6

Capital cost = $\$10.1 \times 10^6$

Additional boiler costs = $\$7.81 \times 10^5$

Maintenance cost = .0168 mills/kw-hr

Cost of increased plant size = $\$5 \times 10^6$

Airside effectiveness = 61.5 %

Airside heat transfer coefficient = 7.88 Btu/hr ft² °F

Land Cost = $\$4.06 \times 10^4$

Land Area = 1.52×10^5 ft²

Table 8

Comparison of Periodic Vs. Finned Tube Dry Tower Costs

Finned Tube Surface cost Including Assembly		Periodic Surface Cost Including Assembly		
		<u>\$.20/ft²</u>	<u>\$.40/ft²</u>	<u>\$.60/ft²</u>
\$1.00/ft ²	Ratio $\frac{(\text{Capital Cost})_{\text{periodic}}}{(\text{Capital Cost})_{\text{Finned}}}$	.57,	.86	1.10
	Ratio $\frac{(\text{Generating Cost})_{\text{periodic}}}{(\text{Generating Cost})_{\text{Finned}}}$	.63,	.87	1.08
\$2.00/ft ²	Ratio of Capital Costs	.30	.46	.60
	Ratio of Generation Costs	.51	.70	.87
\$3.00/ft ²	Ratio of Capital Costs	.21,	.31	.40
	Ratio of Generation Costs	.36,	.50	.62

COOLING TOWER INCREMENTAL COST VS. DESIGN TEMPERATURE
For Different Surface Costs
Figure 24

COOLING TOWER CAPITAL COST VS. DESIGN TEMPERATURE
For Different Surface Costs
Figure 25

COOLING TOWER INCREMENTAL COST AND CAPITAL COST VS. DESIGN TEMP.
For Optimum Design
Figure 26.

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VII Conclusions and Recommendations

The periodic cooling tower as described in this thesis has been shown to reduce evaporation to negligible levels. Its cost depends upon how large the fabrication costs are with respect to the material costs. At this time the periodic tower appears to hold an economic advantage over the conventional dry tower.

Because of the encouraging results obtained to date and because the design was just invented this year it is suggested that further study be made on the periodic cooling tower. Among the more important areas for further research are:

1. The selection of the optimum oil for use in the tower. A good selection must have low vapor pressure, low cost, low viscosity and maintain the ability to preferentially wet the metal surface.
2. Different rotational matrix surfaces should be investigated in which the heat transfer coefficients are higher than for flow between parallel plates. Wire mesh screens are a possibility with air flow through the screens rather than parallel to the surface if the oil entrainment in the air stream can be reduced.
3. A larger test model of around 100 to 200 kilowatts should be built and tested for long term performance.
4. Higher air velocities should be used to insure the heat transfer is well into the turbulent regime.
5. Some method for estimating the fabrication and construction costs should be discovered.

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Appendix A

Derivation of Optimum Water Level

The area available for air side heat transfer is

$$A = \pi R^2 \left(\frac{1}{2} + \frac{\arcsin \frac{r}{R}}{\pi} \right) + r \sqrt{R^2 - r^2} - \pi r^2$$

$$\frac{\partial A}{\partial r} = \frac{R^2 \frac{1}{R}}{\sqrt{1 - \left(\frac{r}{R}\right)^2}} - \frac{\frac{1}{2} r^2 \cdot 2}{\sqrt{R^2 - r^2}} + \sqrt{R^2 - r^2} - 2 \pi r = 0$$

$$2(R^2 - r^2) \sqrt{R^2 - r^2} - 2 \pi r (R^2 - r^2) = 0$$

therefore $r = R$ which is the minimum area or $r = \sqrt{\frac{R^2}{\pi^2 + 1}}$

or $r = \frac{R}{3.3} = .303 R$ and $\phi = 17.66^\circ$

With this value of r the area for heat transfer one side of a plate is $A = 1.878 R^2$ which is an 11.9 % increase in the available air side heat transfer area.

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4

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Appendix B

Computer Input

Card 1: Format 8f10.4

IREP : number of repetitions of cards 2-11

Card 2: Format 8F10.4

TSTAR: lowest design temperature to consider (°F)

TEND: highest design temperature to consider (°F)

VAS(1) : initial value for T_{h1} (°F)

VAS(3) : initial value for $C_{\text{water}}/C_{\text{air}}$

VAS(5) : initial value for 120 times D (ft)

Card 3: Format 8E10.3

VAS(4) : put any value here

Card 4: Format 8F5.1,8F5.4

TA(1) thru TA(8): Temperature representative of a small
span of yearly experienced temperatures (eg: TA =
72 for span of 70-75) (°F)

TPER(1) thru TPER(8): fraction of year plant will experience
temperature increment described by the corresponding
TA

Card 5: Format 8F10.4

PSIZE: plant size (MW)

FCOS: fuel cost (¢/MMBTU)

PWCOS: replacement power cost (mills/kw-hr)

BCOST: cost of steam supply system (\$/Kw)

HPCST: cost of electric motors (\$/hp)

PER: maintenance cost/capital cost

CAPF: capacity factor (%/100)

FCR: fixed charge rate (%/100)

Card 6: Format 8F10.4

TEFF: thermal efficiency (%/100)

EFFP: pump efficiency (%/100)

EFFF: fan efficiency (%/100)

PROPC: fan propeller cost/area covered (\$/ft²)

FANV: air velocity exiting fan (ft/sec)

XDEPA: minimum air side pressure drop through heat exchanger
(lbf/ft²)

RECOV: Fraction of air's energy of motion which is recovered
as with a diffuser

SURF: coefficient to multiply cost per area of heat exchanger
(\$/ft²)

Card 7; Format 8F10.4

PLANC: plant capital cost per kw (\$/kw)

Card 8: Format 8F10.4

TPO thru TPO(2): coefficients for 3rd order polynomial curve
fit for turbine heat rate performance:

Heat rate = F(back pressure)

Appendix C

Computer Output Definitions

ACOS: incremental cost of operation, including all off design costs (mills/kw-hr)

CAPIT: plant capital cost (CCOS + CBOIL + CPLNT) (\$)

COST OF LAND: land cost (\$)

AI: air side effectiveness

CCOS: capital cost of heat exchanger (\$)

CBOIL: capital cost of required increased steam supply system (\$)

COSM: maintenance cost (mills/kw-hr)

CPLNT: capital cost of required increased plant size (\$)

CSTCL: experienced cost of replacing lost capacity at a particular off design temperature (mills/kw-hr)

CWARA: heat capacity of water/heat capacity of air flow rate

D: PLATE SPACING (ft)

DELFC: change in fuel cost (mills/kw-hr)

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

DELPW: water side pressure drop across heat exchanger (lb/ft^2)

FPOW: required fan power (kw)

NPLAT: number of plates required for the tower

PCST: cost of off design temperature operation weighted by a fraction of year that temperature experienced (mills/kw-hr)

PGEN: power generation capability at off design operation (MW)

PPOW: required pump power (KW)

QPLAT: amount of heat rejection per plate (Btu/hr)
 REAIR: air side reynold's number
 REWAT: water side Reynolds number
 TA: ambient air temperature (off design °F)
 TCOS: incremental cost of operation at a particular temperature
 (on design temperature mills/kw-hr)
 T1: design turbine outlet temperature (°F)
 T11: off design turbine outlet temperature (°F)
 UA: heat transfer coefficient for air side (Btu/hr ft² °F)
 VAIR: air velocity thru heat exchanger (ft/sec)
 VA: air velocity thru heat exchanger (ft/hr)
 VW: water velocity thru heat exchanger (ft/hr)
 VWAT: water velocity thru heat exchanger (ft/sec)
 W3: mass flow water thru heat exchanger (lbm/hr)
 W4: mass flow air thru heat exchanger (lbm/hr)
 WI: water side effectiveness

Appendix D

Computer Program Listing

```

      DIMENSION PCST(8),TPER(8),TA(8)
      COMMON VAR(5),VAL(5),VAS(5),TPO(4),TCOST,CCOST,REAR1,N1,HRFAC,W31,
1W41,J1,DEPW1,DEPA1,PPOW1,FPOW1,CPLN1,COSM1,DEFC1,WIDT1,ELEN1,VAIR1
2,VWAT1,HI1,H01,T41,EFF1,QIN1,CBOIL,REWAR,NW1,TCOS,CCOS,REAR2,N,H
3PFAC,W3,W4,J,DELPW,DELPA,PPOW,FPOW,CPLN,COSM,DEFC,WIDTH,ELEN,VAI
4R,VWAT,HI,H0,T4,EFF,QIN,CBOIL,REWAR,NW1,TEFF,PSIZE,PWCOS,PER,FCO
5S,FCR,CAPF,PLANC
      COMMON BCOST,HPCST,EFFP,EFFF,PROPC,FANV,RECOV,C1,C2,C3,C4,C5,TJ,
CCAF,XDEPA,KCONV,SURF,JCONS
      COMMON R1,CC1,VAA1,VW1,PLTN1,TW1,AREA1,CLAN1,AI1,NI1,JA1,PLAT1,
C01,R,C,VA,VW,PLATN,TW1,AREA,BCOSTL,AI,NI,JA,PLAT,J,KKONS
      COMMON CLAND
      READ(5,73)IREP
      DO 6 IR=1,IREP
      READ(5,77)TSTAR,TEND,VAS(1),VAS(3),VAS(5)
      READ(5,75)(TA(I),I=1,8),(TPER(I),I=1,8)
      READ(5,77)PSIZE,FCOS,PWCOS,BCOST,HPCST,PER,CAPF,FCR
      READ(5,77)TEFF,EFFP,EFFF,PROPC,FANV,XDEPA,RECOV,SURF
      READ(5,77)PLANC,CLAND
      READ(5,77)(TPO(I),I=1,4)
      WRITE(6,79)
      WRITE(6,75)(TA(I),I=1,8),(TPER(I),I=1,8)
      WRITE(6,77)PSIZE,FCOS,PWCOS,BCOST,HPCST,PER,CAPF,FCR
      WRITE(6,77)TEFF,EFFP,EFFF,PROPC,FANV,XDEPA,RECOV,SURF
      WRITE(6,77)PLANC,CLAND
      WRITE(6,77)TPO(1),TPO(2),TPO(3),TPO(4)
75' FORMAT(8F5.1,8F5.4)
77' FORMAT(8F10.4)
73' FORMAT(8I10)
79' FORMAT('1')
81' FORMAT('0') USED IN THE FOLLOWING CALCULATIONS IS =18.3)
92' FORMAT(' CCOST= ',E10.3,' TCOST= ',E10.3,' REWAR= ',E10.3
C,' REAR= ',E10.3,' HRFAC= ',E10.3)
93' FORMAT(' W3= ',E10.3,' W4= ',E10.3,' DELPW= ',E10.3,' DELPA= ',E10
C.3,' PPOW= ',E10.3,' FPOW= ',E10.3)
94' FORMAT(' CBOIL= ',E10.3,' COSM= ',E10.3,' DEFC= ',E10.3,' CPLN= ',
C,E10.3,' VAIR= ',E10.3,' VWATER= ',E10.3)

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```

      87 FORMAT(' ACOS='E15.5)
      88 FORMAT(' CAPIT='E15.5)
101  FORMAT(1X,'D='E10.3,'VA='E10.3,'VW='E10.3,'T1 = 'E10.3)
102  FORMAT(1X,'TA = 'E10.3,' B = 'E10.3,' C = 'E10.3,'NPLAT = '
      C,E10.3,' QPLAT = 'E10.3)
103  FORMAT(1X,'JA = 'E10.3,' AI = 'E10.3,' AREA OF LAND = 'E10.3
      C,'CCOST OF LAND = 'E10.3,'WI = 'E10.3)
      VAS(4) = 2.E0
      DO 7 IS=1,5
7     VAR(IS)=0.0
      TD=TSTAR
1     WRITE(6,31)TD
      K=4
      KKK = 0
4     CALL SHOT(1)
5     L=0
      CALL XTEND(L)
      KKK = KKK + 1
      IF(KKK.GT.20) L = 0
      IF(L.EQ.1) GO TO 4
      CALL SHOT(2)
2     JCONS = 0
      CALL SERCH
      IF(JCONS.EQ.0) GO TO 10
      K=K-1
      IF(K.GT.0) GO TO 2
10    CAPIT = CCOS1 + CBOI1 + CP_LN1 + CLAN1
      WRITE(6,101) D1,VAA1,VW1,TW1
      WRITE(6,102) TD,B1,CC1,P_LN1,QPLT1
      WRITE(6,103) UA1,AI1,AREA1,CLAN1,WI1
      WRITE(6,33)CAPIT
      WRITE(6,32) CCOS1,TCOS1,REWA1,REA11,HRFA1
      WRITE(6,33) W31,W41,DEPW1,DEPA1,PPDW1,FPDW1
      WRITE(6,34) CBOI1,COSM1,DEFC1,CP_LN1,VAIR1,VWAT1
      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(6,37)ACOS
    TD=TD+5.
    IF(TD.LE.TEND) GO TO 1
5 CONTINUE
    STOP
    END

```

```

SUBROUTINE SHOT(L)
  DIMENSION T(3),CWAR(4),RANG(3),AFRO(3),W_LRA(4),VAM(5)
  COMMON VAR(5),VAL(5),VAS(5),TPD(4),TCOS1,CCOS1,REAI1,N1,HRA1,W31,
1 W41,J1,DEPW1,DEPA1,PPOW1,FPOW1,CPLN1,COSM1,DEFC1,WIDT1,ELEN1,VAIR1
2,VNAT1,HI1,HO1,T41,EFF1,QIN1,CBOIL,REWA1,NW1,TCOS,CCOS,REAR,N,H
3RFAC,W3,W4,J,DELPW,DELP,PPDW,FPOW,CPLN,COSM,DEFC,WIDTH,ELENG,VAI
4R,VNAT,HI,HO,T4,EFF,QIN,CBOIL,REWA,NW,TEFF,PSIZE,PWCOS,PER,FCO
5S,FCR,CAPF,PLANC
  COMMON BOOST,HPCST,EEFF,EEFFH,PROPC,FANV,RECOV,C1,C2,C3,C4,C5,TJ,
CCA,F,REFJ,XDEPA,KCONV,SURF,JCONS
  COMMON R1,CC1,VAA1,VW1,PLTN1,TNW1,AREA1,CLAN1,AI1,WI1,JA1,DPLT1,
C1,B,C,VA,VW,PLATN,TW1,AREA,KCOSTL,AI,WI,JA,DPLAT,D,KONS
  COMMON C_LAND
  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(3)+(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.
    DO 23 I=1,4
      RANG(I) = VAS(2) + (I - 2)*2.
23 IF(RANG(I).LE.0.) RANG(I) = .5
      AFRO(1) = VAS(4)
      IA = 1
      DT1=1.
      DRANG = 1.
      DCWAR=1.
      DW_LRA=1.
      LR = 4

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```

      _C=4
      _W=4
      GO TO 51
31 DT1=VAR(1)-VAM(1)
   DRNG = VAR(2) - VAM(2)
   DCWAR=VAR(3)-VAM(3)
   DWLRA=VAR(5)-VAM(5)
   IF(DT1.EQ.0.) GO TO 33
   DO 32 I=1,3
32 T(I)=VAR(1)-(I-1)*DT1/2.
   GO TO 28
33 DO 34 I=1,3
   T(I)=VAR(1)+(I-2)*2.
34 IF(T(I).GE.170.) T(I)=170.
28 IF(DRNG.EQ.0.) GO TO 26
24 DO 25 I =1,4
25 RANG(I) = VAR(2) - (I-1)*DRNG/3.
   GO TO 35
26 DO 27 I=1,3
27 RANG(I) = VAR(2) + (I - 2)
   _R = 3
35 IF(DCWAR.EQ.0.) GO TO 37
   DO 35 I=1,4
35 CWAR(I)=VAR(3)-(I-1)*DCWAR/3.
   GO TO 43
37 DO 38 I=1,3
38 CWAR(I)=VAR(3)+(I-2)
   _C=3
43 IF(DWLRA.EQ.0.) GO TO 45
   DO 44 I=1,4
44 WLRA(I)=VAR(5)-(I-1)*DWLRA/3.
   GO TO 51
45 DO 45 I=1,3
45 WLRA(I)=VAR(5)+(I-2)*5.
   _W=3
51 DO 10 IT=1,3

```

```

DO 13 IC=1,LC
DO 13 IR =1,LR
DO 13 IW=1, _W
KCALC=1
IF (IR.GT.1.OR.IC.GT.1) KCALC=2
IF (IW.GT.1) KCALC=3
CALL CALC(T(IT),RANG(IR),CWAR(IC),AFRO(IA), _RA(IW),KCALC)
IF (DELPA._T.XDEPA) GO TO 13
IF (TCOS1.EQ.0.) GO TO 14
IF (TCOS.GT.TCOS1) GO TO 16
14 TCOS2 = TCOS1
VAM(1) = VAR(1)
VAM(2) = VAR(2)
VAM(3) = VAR(3)
VAM(4) = VAR(4)
VAM(5) = VAR(5)
TCOS1 = TCOS
VAR(1) = T(IT)
VAR(2) = RANG(IR)
VAR(3) = CWAR(IC)
VAR(4) = AFRO(IA)
VAR(5) = _RA(IW)
IF (TCOS2.EQ.0.) GO TO 14
GO TO 15
18 IF (L.EQ.2) GO TO 15
IF (TCOS2.EQ.0.) GO TO 17
15 IF (TCOS1.EQ.TCOS2) GO TO 17
IF (TCOS.GT.TCOS2) GO TO 15
17 TCOS2 = TCOS
VAM(1)=T(IT)
VAM(2) = RANG(IR)
VAM(3)=CWAR(IC)
VAM(4)=AFRO(IA)
VAM(5)=W_RA(IW)
15 CONTINUE
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,  
C ' AFRO1='F10.3,' WLRA1 ='E10.3)  
RETURN  
END)
```



```

SUBROUTINE XTEND(L)
  DIMENSION TTRY(2),RTRY(2),CTRY(2),ATRY(2),WTRY(2)
  COMMON VAR(5),VAL(5),VAS(5),TPO(4),TCOS1,CCOS1,REAI1,V1,HREAI,W31,
  1W41,J1,DEPW1,DEPA1,PPOW1,FPOW1,CPLV1,COSM1,DEFC1,WID1,ELEV1,VAIR1
  2,VAA1,H1,H01,T41,EFF1,QIN1,CB011,REWA1,NW1,TCOS,CCOS,REAI2,N,H
  3RFAC,W3,W4,J,DELPW,DELPA,PPOW,FPOW,CPLV,COSM,DELFC,WIDT4,ELEV3,VAI
  4R,VAA1,H1,H0,T4,EFF,QIN,CB01L,REWA1,NW1,TEFF,PSIZE,PWCOS,PER,FCO
  5S,FCR,CAPE,PLANC
  COMMON BCOST,HPCST,EFFP,EFFM,PROPC,FANV,RECUV,C1,C2,C3,C4,C5,TD,
  CCAF,XREJ,XDEPA,KCONV,SURF,JCONS
  COMMON P1,CC1,VAA1,VW1,PLTN1,TW1,AREA1,CLAV1,A11,W11,I41,DP1T1,
  C01,B,C,VA,VW,PLATN,TW1,AREA1,COSTL,A1,W1,HA,DP1AT,D,KONS
  COMMON CLAVD
  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.
  RTRY(1) = VAR(2) - 3.
  IF(RTRY(1).LE.0.) RTRY(1) = .5
  RTRY(2) = VAR(2) + 2.
  CTRY(1)=VAR(3)-3.
  IF(CTRY(1).LE.0.) CTRY(1)=.5
  CTRY(2)=VAR(3)+2.
  WTRY(1) = VAR(5) - 3.
  IF(WTRY(1).LE.0.) WTRY(1) = .5
  WTRY(2) = VAR(5) + 2.
  DO 3 I=1,2
  CALL CALC(TTRY(I),VAR(2),VAR(3),VAR(4),VAR(5),1)
  IF(DELPA.LT.XDEPA) GO TO 3
  IF(TCOS.GT..99*TCOS1) GO TO 3
  TCOS1=TCOS
  VAS(1)=TTRY(1)
  VAS(2) = VAR(2)
  VAS(3)=VAR(3)
  VAS(4)=VAR(4)
  VAS(5)=VAR(5)

```

```

      _=1
3  CONTINUE
      DO 10 I=1,2
      CALL CALC(VAR(1),VAR(2),CTRY(I),VAR(4),VAR(5),I)
      IF(DELPA._T.XDEPA) GO TO 10
      IF(TCOS.BT..99*TCOS1) GO TO 10
      TCOS1=TCOS
      VAS(1)=VAR(1)
      VAS(2) = VAR(2)
      VAS(3)=CTRY(I)
      VAS(4)=VAR(4)
      VAS(5)=VAR(5)
      _=1
10 CONTINUE
      DO 14 I=1,2
      CALL CALC(VAR(1),RTRY(I),VAR(3),VAR(4),VAR(5),I)
      IF(DELPA._T.XDEPA) GO TO 14
      IF(TCOS.BT..99*TCOS1) GO TO 14
      TCOS1 = TCOS
      VAS(1) = VAR(1)
      VAS(2) = RTRY(I)
      VAS(3) = VAR(3)
      VAS(4) = VAR(4)
      VAS(5) = VAR(5)
      _ = 1
14 CONTINUE
      DO 12 I=1,2
      CALL CALC(VAR(1),VAR(2),VAR(3),VAR(4),WTRY(I),3+I)
      IF(DELPA._T.XDEPA) GO TO 12
      IF(TCOS.BT..99*TCOS1) GO TO 12
      TCOS1=TCOS
      VAS(1)=VAR(1)
      VAS(2) = VAR(2)
      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
```

```

SUBROUTINE SERCH
  DIMENSION CC(5)
  COMMON VAR(5),VAL(5),VAS(5),TPO(4),TCOS1,CCOS1,REAL1,N1,WRFA1,W31,
1W41,J1,DEPA1,DEPA1,PPOW1,FPDOW1,CPLV1,COSM1,DEFQ1,WID11,ELENI,VAIR1
2,VNAT1,H11,HJ1,T41,EFF1,DIV1,CBO11,REWA1,NW1,TCOS,CCOS,REAR1,N,H
3RFAC,V3,W3,J,DELPA,DELPA,PPOW,FPDOW,CPLV,COS4,DELFC,WIDTH,FLEN3,VAI
4R,VNAT,H1,HJ,T4,EFF,WIN,CBOIL,RENAT,NW,TEFF,PSIZE,PWCOS,PER,FCO
5S,FCR,CAPF,PLANC
  COMMON ROST,HPCST,EEFF,EEFF,PROPC,FANV,RFCOV,C1,C2,C3,C4,C5,TD,
CCAF,XREFJ,XDEPA,KCONV,SURF,JCONS
  COMMON R1,C1,VAA1,VW1,PLTV1,TW1,AREAL,CLAV1,AI1,WI1,JA1,DPAT1,
CJ1,R,C,VA,VW,PLATV,TW1,AREAL,COSTL,AT,WI,JA,DPAT,D,KKONS
  COMMON CLAV1
  CC(1) = 5.
  CC(2) = 1.
  CC(3) = 1.
  CC(5) = 1.
  DO 4 I=1,5
    IF(I.EQ.2) GO TO 8
    M=5-I
    K=3
    DO 9 IK=1,5
8    VA_(IK)=VAR(IK)
    _=1
    KC=0
1    J=1
2    VA_(M) = VAR(1) + CC(M)
    IF(M.GT.3) GO TO 10
    IF(VA_(1).GT.170.) GO TO 4
10 CALL CALC(VA_(1),VAL(2),VA_(3),VAL(4),VA_(5),_)
    _=M
    IF(DELPA._T.XDEPA) GO TO 3
    IF(TCOS.GT.TCOS1) GO TO 4
    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) - CC(M)
  IF(VAL(M).LE.0.0) GO TO 7
  CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),M)
  IF(DEIPA.LT.XUEPA) 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
  IF(K.EQ.7) GO TO 8
  CC(M) = CC(M)/5.
  GO TO 1
8 CONTINUE
  RETURN
  END

```

```

SUBROUTINE CHNGF(M)
  DIMENSION DP(5),DV(5),VAJ(5)
  COMMON VAR(5),VAL(5),VAS(5),TPO(4),TCOS1,CCOS1,REAL1,N1,HREF1,W31,
1W41,J1,DEPW1,DEPA1,PPOW1,FPDW1,CPLV1,COSM1,DEFC1,WIDT1,ELEN1,VAIR1
2,VNAT1,HI1,H01,T41,EFF1,QIN1,C30I1,REWA1,NW1,TCOS,CCOS,REATR,N,H
3RFAC,W3,W4,J,DELPW,DELPA,PPDW,FPDW,CPLV,COS1,DEFLC,WIDTH,F_LEN,VAI
4R,VNAT,HI,H0,T4,EFF,QIN,C30IL,RENAT,NW,T,FF,PSIZE,PWCOS,PER,FCO
5S,FCR,CAPF,PLANC
  COMMON HPCST,HPCST,EFFP,EFFP,PROPC,FANV,REFIV,C1,C2,C3,C4,C5,TD,
CCAF,REFJ,XDEPA,KCONV,SURF,JCONS
  COMMON R1,C1,VAA1,VW1,PLTN1,TNW1,AREAL,CLAN1,AI1,WI1,HA1,QPLT1,
C01,R,C,VA,VW,PLATN,TW1,AREAL,COSTL,AI,WI,JA,QPLAT,D,KONVS
  COMMON CLAND
  _=1
  <1=0
  DO 1 I=1,5
1  DV(I)=.05
  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.3) GO TO 10
    IF(VAL(1).GE.170..OR.VAL(1).LT.110.) GO TO 15
10  CALL CALD(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),1)
    IF(KCONV.EQ.1) GO TO 36
    DP(M)=DEPA-DEPA1
    DO 35 II=1,5
      J=5-II
      IF(J.EQ.4) GO TO 35
      IF(I.EQ.5.AND.DV(1).GT..005) GO TO 35
      IF(I.EQ.4) GO TO 35
11  VAL(I)=(1.+DV(I))*VAR(I)
      IF(VAL(I).LE.0.0) GO TO 34
      VAJ(I)=VAL(I)
      IF(I.GT.3) GO TO 17

```

```

      IF(VAJ(1).GE.170..OR.VAJ(1).LT.110.) GO TO 12
      GO TO 17
12  VAJ(I)=VAR(I)
      GO TO 34
17  J=I
      IF(4.LT.I) J=M
      CALL CALC(VAJ(1),VAJ(2),VAJ(3),VAJ(4),VAJ(5)+J)
      VAJ(I)=VAR(I)
      IF(KCONV.EQ.1) GO TO 34
      DP(I)=DELPA-DEPA1
      E=0.
      CC = 1.
      DD = 1.
18  IF(DP(I).EQ.0.) GO TO 34
      VAL(I) = (1. + DV(I)*(E - DP(M))/DP(I))*VAR(I)
19  IF(VAL(I).LE.0.0) GO TO 34
      IF(I.GT.3) GO TO 24
      IF(VAL(1).GE.170..OR.VAL(1).LT.110.) GO TO 34
24  CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5)+J)
      IF(KCONV.EQ.1) GO TO 34
      IF(DELPA.GE.XDEPA.AND.DELPA.LE.1.01*XDEPA) GO TO 25
      IF(DELPA.GE.1.01*XDEPA) GO TO 25
      CC = DD/10.
      IF(CC.LT..001) GO TO 34
      E = E + CC
      GO TO 18
25  DD = CC/10.
      IF(DD.LT..001) GO TO 34
      E = E - DD
      GO TO 18
26  IF(TCOS.3T.TCOS1) GO TO 34
      CALL STORE
      _=1
      GO 27 IJ=1,5
27  VAJ(IJ)=VAR(IJ)
      VAL(4)=(1.+DV(M))*VAR(M)

```

```

      IF (VAL(M).LE.0.0) GO TO 36
      IF (M.GT.3) GO TO 33
      IF (VAL(1).GE.170..OR.VAL(1).LT.110.) GO TO 35
33  CALL CALC(VAL(1),VAL(2),VAL(3),VAL(4),VAL(5),M)
      IF (CONV.EQ.1) GO TO 36
      DP(4)=DE_PA-DEPA1
      GO TO 11
34  VAL(1)=VAR(1)
35  CONTINUE
36  DO 37 I=1,5
37  DV(I)=-DV(I)
      KL=KL+1
      IF (KL.EQ.1) GO TO 4
      KL=0
      IF (DV(1).LE..005) GO TO 39
      DO 38 I=1,5
38  DV(I)=DV(I)/5.
      GO TO 4
39  IF (L.EQ.1) WRITE(6,103)
103  FORMAT('          AND CHANGES MADE')
      RETURN
      END

```



```

SUBROUTINE CALC(T1,RANGE,CWARA,AFRON,WLRAT,M)
COMMON VAR(5),VAL(5),VAS(5),TPD(4),TCOS1,CCOS1,REAL1,N1,HRFA1,W31,
1W41,J1,DEPN1,DEPA1,PPOW1,FPJW1,CPLN1,COSM1,DEFC1,WID11,ELEN1,VAIR1
2,VWAT1,HI1,HJ1,T41,EFF1,QIN1,C3O11,REWA1,VW1,TCOS,CCOS,REATR,N,H
3RFAC,W3,W4,J,DELPW,DELPA,PPJW,FPJW,CPLN,COSM,DEFLC,WID11,FLNG,VAT
4R,VWAT,HI,HJ,T4,EFF,QIN,C3O11,REWA1,VW,TEFF,PSIZE,P,COS,PF2,FCO
5S,FCR,CAPF,PLANC
COMMON BCOST,HPCST,EEFF,EEFF4,PRDPC,FANV,RECOV,C1,C2,C3,C4,C5,TJ,
CCAF,PREJ,XDEPA,KCONV,SURF,JCONS
COMMON R1,CC1,VAA1,VW1,P_TN1,TNW1,AREAL,CLAN1,AI1,WI1,JA1,DPLT1,
C01,R,C,VA,VW,PLATN,TW1,AREAL,COSTL,AI,WI,JA,DPLAT,D,KKONS
COMMON CLAND
IF(T1.LE.TD) GO TO 89
IF(RANGE.LE.0.) GO TO 89
IF(CWARA.LE.0.) GO TO 89
IF(AFRON._E.0.) GO TO 89
IF(WLRAT._E.0.) GO TO 89
KCONV=0
P=EXP(17.158-9240./(T1+450.))
HRFAC=TPD(1)+TPD(2)*P+TPD(3)*P**2+TPD(4)*P**3
TEFFM=TEFF/HRFAC
QIN=3413.*PSIZE*1000./TEFFM
PREJ=QIN*(1.-TEFFM)
T = WLRAT/120.
VW = 584./RANGE
VA = VW*1940./CWARA
JW = 33.333
JA = 1./(.03 + 2440.*((D**2)/((VA + 4710.)*.8)))
TA1 = TD
TW1 = T1
X = 536.*JA/(VA*D)
IF(X.LT..01) GO TO 5
IF(X.GT.4.) GO TO 1
AI = .004355972 +.5839515*X -.1991610*(X**2) +.0123113*(X**3.)
C+.005145541*(X**4.) -.00100051*(X**5.)
GO TO 2

```

```

      AI = X
      GO TO 2
1     AI = 1.
2     Y = .1612*JW/(VW*D)
      IF(Y.LT..01) GO TO 6
      IF(Y.GT.4.) GO TO 3
      WI = .004587799 + .6547554*Y -.1368092*(Y**2) +.009417724*(Y**3
C.) + .005835278*(Y**4.) +.001055019*(Y**5.)
      GO TO 4
5     WI = Y
      GO TO 4
3     WI = 1.
+     B = .056*VA*D*AI
      C = 108.*VW*D*WI
      QPLAT = 3*C*(TW1 - TA1)/(B + C)
      PLATN = QREJ/QPLAT
      W3 = PLATN*D*VW*108.
      W4 = PLATN*D*VA*.231
      VA1 = 53.35*(TD+ 450.)/(15.*144.)
      VAIR = VA/3500.
      VWAT = VW/3500.
      REWAT = (4710. + VW)*D*100.
      REAIR = (4710. + VA)*D*3.09
      IF(REWAT.GT.1.E5) GO TO 13
      FWAT = .0791/(REWAT**25)
      GO TO 14
13     FWAT = .045/(REWAT**2)
14     IF(REAIR.GT.1.E5) GO TO 15
      FAIR = .0791/((REAIR)**25)
      GO TO 16
15     FAIR = .045/(REAIR**2)
16     DELPW = (7.45 E-7)*FWAT*((4710. + VW)**2)/)
      DELPA = (9.5 E-10)*FAIR*((4710. + VA)**2)/)
      HPWAT = DELPW*W3/(62.*3600.*550.)
      HPAIR = W4*(DELP*VA1 + (1.-REC)V)*VAIR**2/54.4)/(3500.*550.)
      PPJW=HPWAT/(1.341*EFFP)

```

```

      FPOW=HPAIR/(1.341*EFFF)
      CCOS = (1.+ (FPOW+PPOW)/(PSIZE*1000.))*SJR*19.9*P_LATN
      C + (IPWAT + HPAIR)*HPCST + VAIR*7*P_LATN*3.25*PQPC/FAVV
      DELFC=FCOS*3413.*(HRFAC-1.)/TEFF*1.E-5
      COSM=CCOS*PER/(PSIZE*CAP*8760.)
      CROIL=(HRFAC-1.)*RCOST*1.E6
      CPLN =(FPOW+PPOW)*PLANC
      AREA_ = 0*P_LATN*5.
      COSTL = AREA_*CLAND
      TCOS = (CCOS + CROIL + CPLN + COSTL)*FCR/(PSIZE*CAP*8760.)
      C + COSM + DELFC
      RETURN
39  TCOS = 10.
      RETURN
      END

```

```

SUBROUTINE VARIT(TA,TPER,PCST)
COMMON VAR(5),VAL(5),VAS(5),TPO(4),TCOS1,CCOS1,REAL1,N1,HREF1,W31,
1W41,J1,DEPW1,DEPA1,PPOW1,FPDW1,CPLN1,COSM1,DEFC1,WIDT1,ELEV1,VAIR1
2,VWAT1,HI1,HJ1,T41,EFF1,QIN1,CBO11,REWA1,NW1,DUM1,CCOS,REATR,N,D
3JMY1,W3,N4,J,DELPW,DELPA,PPDW,FPDW,CPLN,COSM,DJMY2,WIDTH,ELENG,VAI
4R,VWAT,HI,HJ,T4,EFF,QIN,CBO11,REWA1,NW1,TEFF,PSIZE,PWCOS,PER,FCO
5S,FCR,CAFF,PLANC
COMMON ROST,HPCST,EFFP,EFFE,PRDPC,FANV,RECDV,C1,C2,C3,C4,C5,TD,
CCAF,QREJ,XDEPA,KCONV,SURF,JCONS
COMMON B1,CC1,VAA1,VW1,PLTN1,TNW1,AREA1,CLAN1,AI1,WI1,UA1,DPLT1,
CJ1,B,C,VA,VW,PLATN,TW1,AREA,COSTL,AI,WI,UA,DPLAT,D,KONS
COMMON CLAND
T1=140.
1 P=EXP(17.158-9240./(T1+450.))
HREFAC=TPO(1)+TPO(2)*P+TPO(3)*P**2+TPO(4)*P**3
TEFFM=TEFF/HREFAC
QREJ=QIN1*(1.-TEFFM)
T11 = TA + QREJ*(B1 + CC1)/(B1*CC1*PLTN1)
IF(ABS(T11-T1).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=FCOS*3413.*(HREFAC-1.)/TEFFM*1.E-5
TCOS = (CCOS1 + CBO11 + CPLN1 + CLAN1)*FCR/(PSIZE*CAFF*8760.)
C + COSM1 + DELFC + CSTCL
PCST=TPER*TCOS
WRITE(6,7)
7 FORMAT(' SUBROUTINE DATA')
WRITE(6,3)HREFAC,PGEN,CSTCL,DELFC
8 FORMAT(' HREFAC='F7.4,' PGEN='F3.3,' CSTCL='F15.5,' DELFC='E15.5)
WRITE(6,9)RANGE,TCOS,PCST,T11,TA
9 FORMAT(' RANGE='E10.3,'TCOS='E15.5,' PCST='E15.5,' T11='F7.3,

```

1. TA=*F6.2)
RETURN
END

```

SUBROUTINE STORE
COMMON VAR(5),VAL(5),VAS(5),TPO(4),S(27),W(27),TEFF,PSTZE,PWCRS,PE
IR,FCOS,FCR,CAPE,PLANC
COMMON BCOST,HPCST,EFFP,EFFI,PROPC,FANV,REDOV,C1,C2,C3,C4,C5,TD,
CCAF,JREJ,XDEPA,KCONV,SURF,JCONS
COMMON F(13),G(13),KKONS
COMMON C_LAND
DO 1 I=1,5
1 VAR(I)=VAL(I)
+ DO 2 I=1,27
+ DO 3 I=1,13
+ DO 3 I=1,13
+ F(I) = G(I)
JCONS=1
KKONS = 1
RETURN
END

```

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