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OPTIMUM DRY-COOLING SUB-SYSTEMS FOR A
SOLAR AIR CONDITIONER

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SUMMARY

Dry-cooling sub-systems for residential solar-powered Rankine-compression air conditioners have been economically optimized and compared with the cost of a wet cooling tower. Results in terms of yearly incremental busbar cost due to the use of dry-cooling have been presented for Philadelphia and Miami. With input data corresponding to local weather, energy rate and capital costs, condenser surface designs and performance, the computerized optimization program yields design specifications of the sub-system which has the lowest annual incremental cost.

INTRODUCTION

Air conditioners reject waste heat to the environment through evaporative (wet) cooling towers or air (dry) coolers. The loss of water from the former to the local atmosphere is quite extensive. A three-ton-solar-powered air conditioner equipped with flat plate collectors vaporizes approximately 41.3 kilograms (91 lb) of water per hour, based upon an overall coefficient of performance of 0.65 of the machine. During the summer months in an urban center, the addition of water vapor to the ambient air is often environmentally objectionable. It is also known that the increase of moisture in air causes low thermal efficiency of the collector. Furthermore, the circulating water is chemically treated for anti-corrosion and anti-freezing, and the discharge of such water for maintenance may overburden our existing water treatment plants in many metropolitan areas.

The dry-cooling system has four outstanding advantages: (a) no harmful or visible pollution discharge, (b) practically maintenance free, (c) no requirement for a water treatment plant, and (d) the elimination of make-up water supply.

Because of these considerations, solar-driven air conditioners designed for wide-spread residential utilization may be required to use dry-cooling. It is noted that the current design of the commercial lithium bromide-water absorption air conditioners is not suitable for using dry-cooling due to possible LiBr crystallization in the absorber at high ambient air temperatures. This problem must be solved before dry-cooling can be adapted to the air conditioner. Recent studies of solar absorption air-conditioning systems have been concerned with those equipped with wet-cooling towers (refs. 1 and 2). In this study the dry-cooling is proposed for a solar-powered Rankine-compression air conditioner.

As compared to wet-cooling, a possible disadvantage associated with dry-cooling is that the latter system requires higher capital and operating costs (although little maintenance cost is involved). The heat transfer by air is much less effective than by water. In addition, the overall coefficient of performance of an air conditioner equipped with nonconcentrating solar collectors is quite low and it rejects as much as twice the amount of heat as a conventional vapor com-
pression machine does. Thus, an unoptimized dry sub-system may be very costly.

The objective of this study is to economically optimize the dry-cooling sub-system for a solar-powered Rankine-compression air conditioner. The computerized program selects the best possible dry-cooling sub-system design by optimizing the heat transfer and thermodynamic relations with economic trade-offs.

ECONOMIC OPTIMIZATION

The economic analysis is based on the capital and operating costs of the solar-powered system which incorporates the Rankine-vapor compression cooling sub-system. Optimization is based on minimizing the annual total cost increment between dry-cooling and wet-cooling.

System Description

The schematic assembly of the solar-powered Rankine-compression air conditioner along with its thermodynamic pressure-enthalpy diagram for the working fluid is shown in figure 1. Only the cooling mode of the machine is considered in this study. Although heating can be accomplished with this system, it is not considered in this report. The working fluid chosen is Freon-12 (R-12). This does not rule out the use of other fluids. The effect of the thermodynamic properties of potential working fluids on the performance of a solar-driven heat pump has been analyzed in reference 3. The system is designed to provide three tons of air conditioning at temperatures of $10^0$ C ($50^0$ F) in the evaporator and $94.5^0$ C ($202^0$ F) in the boiler. The condensing temperature of the working fluid in the air-cooled condenser is one of the four system variables which will be discussed in a later section. A single fluid (R-12) is utilized for both Rankine and Compression cycles for simplified equipment design, although the use of different fluids (one for Rankine cycle and the other for the compression cycle) may yield better overall efficiencies.

The working fluid is vaporized by the heat supplied by the hot water from collector or storage tank (not shown in fig. 1), and then expands in the expander to provide the power for both compressor and pump. The exhaust vapor from the expander and the compressed vapor leaving the compressor are condensed in the air-cooled condenser. An appropriate amount of the condensate is pumped to the boiler for vaporization while the remaining liquid (the two mass flow rates are among the outputs of the design program) is adiabatically expanded through the expansion valve to the evaporator pressure.

The thermal efficiency of the Rankine cycle, $\eta_p$, is
\[ \eta_r = \frac{W_c - W_p}{Q_b} \]  (1)

where
\[ W_c = \text{power produced by the expander, kW (Btu/hr)} \]
\[ W_p = \text{power for the pump, kW (Btu/hr)} \]
\[ Q_b = \text{heat rate to the boiler supplied by solar collector-storage unit, kW (Btu/hr)} \]

The coefficient of performance, COP, of the compression cycle is,
\[ \text{COP} = \frac{Q_c}{W_c} \]  (2)

where
\[ Q_c = \text{cooling capacity of the air conditioner, kW (Btu/hr)} \]
\[ W_c = \text{power for the compressor, kW (Btu/hr)} \]

With \[ W_c = W_p + W_e \], the overall coefficient of performance, OCOP, is
\[ \text{OCOP} = \frac{Q_c}{Q_b} = \eta_r \cdot \text{COP} \]  (3)

Equations (1) to (3) for various condensing temperatures of R-12 are plotted in figure 2, where
\[ T_e = \text{evaporator temperature, } ^{\circ}\text{C (}^{\circ}\text{F)} \]
\[ T_b = \text{boiler temperature, } ^{\circ}\text{C (}^{\circ}\text{F)} \]
\[ \text{\( F_{\text{exp}} \)} = \text{expander efficiency, dimensionless} \]
\[ \text{\( F_p \)} = \text{pump efficiency, dimensionless} \]
\[ \text{\( F_{\text{comp}} \)} = \text{compressor efficiency, dimensionless} \]

Computerized Optimization Program

The program for the computer was written to carry out the minimization of the total annual cost increment. The same guidelines were applied for both dry-cooling and wet-cooling when determining system constraints, conditions, and the state of the art. The program assumptions, input data requirements, system variables, cost equations, and procedures are outlined below:

Assumptions. - The program specifications are the following:
1. The air conditioner performance varies with condensing temperature in accordance with the data given in figure 2.
2. The baseline solar collectors considered herein are selectively coated (such as black chrome) and have two glass covers which deliver 0.252 kW/m² (80 Btu/hr·ft²) of heat to the boiler.

3. The percentages of the cooling loads in various sites are based on the data reported in reference 4. Two cases considered herein are 77 percent of the total load for Philadelphia and 49 percent for Miami.

4. The air-cooled condenser is a single pass cross-flow type with both fluids (air side and Freon side) unmixed.

5. The air side pressure drop across the condenser is to be kept above a minimal value (0.76 kg/m², or 2 lb/ft²) in order to overcome a pressure differential caused by wind.

6. Condenser tubes are thin and their thermal resistance is negligible as compared to air film resistance.

7. The fixed charge rate is 10.2 percent on the basis of 8 percent interest on the capital cost for 20 years.

Input data. - Input data include air conditioner performance (fig. 2), solar heat collected and delivered to the boiler, percentage of cooling load supplied by solar, heat exchanger surface and its performance, minimum air side pressure drop of the condenser, rated cooling capacity of the air conditioner, boiler temperature, evaporator temperature, weather data (temperature) of the site, properties of the working fluid and at, design condensing temperature of wet-cooling system, thermal efficiencies of equipment components (expander, compressor, pump, fan, etc.), fixed charge rate, and costs of electricity, solar collector, heat exchanger surface, electric motor, and fan blades. These input parameters can be readily changed so as to allow the program to be quite general and adaptable to local conditions.

System variables. - With the input and parameters, the thermofluid, heat transfer and cost equations of the system can be written in terms of four variables which can be varied independently and which must be optimized to yield the least cost of the system.

The four variables are: (a) initial temperature difference (ITD), which is defined as the temperature difference between the working fluid entering the condenser and the ambient air, (b) condensing temperature of the working fluid, (c) number of tube banks in the direction of air flow, and (d) mass velocity ratio of the working fluid and the air.

Cost equations. - The increased cost due to the use of dry-cooling includes incremental capital cost, operating cost, and penalty cost due to decreased thermal efficiency and idled solar equipments whenever the ambient temperature is higher than the design condition.

The cost of the wet-cooling tower and the wet-condenser for the conventional
The system is estimated at $700. The maintenance cost of the wet-cooling system, which might be significant, is not included in the analysis due to lack of data in the field.

The incremental capital costs are broken down into:

1. **Cost of solar collector:** \( y_1 = C_1(A_1 - A_2) \), where \( C_1 \) is the cost per square foot of collector, \( A_1 \) the collector area required in dry-cooling and \( A_2 \) in wet-cooling.

2. **Cost of hot water storage tank:** \( y_2 = $0.75(A_1 - A_2) \); see reference 5.

3. **Cost of condenser surface and header:** \( y_3 = C_2 A_3 \), where \( C_2 \) is the cost per unit condenser surface area installed, and \( A_3 \) the condenser surface area.

4. **Cost of motors for fan and pump:** \( y_4 = C_3 (P_1 + P_2) \), where \( C_3 \) is the cost of motors for the fan and pump per horsepower; \( P_1 \) is the power required for the fan and \( P_2 \) the power for the pump.

5. **Cost of fan blades:** \( y_5 = C_4 A_4 \), where \( C_4 \) is the cost of fan blades per unit area covered, and \( A_4 \) the frontal area of the condenser.

Thus, the annual total incremental capital cost, \( y_c \), is

\[
y_c = (y_1 + y_2 + y_3 + y_4 + y_5 - 700)f_c
\]

where \( f_c \) is the fixed charge rate or capital-recovery factor.

The operating cost is for electricity to power the fan and is computed by

\[
y_o = C_s K H
\]

where \( C_s \) is the electricity cost per KWh, \( K \) the power for the fan, and \( H \) the total operating hours of the fan per year.

When the ambient air temperature is higher than design temperature, the thermal efficiency decreases. To meet the designed cooling load, conventional energy (electricity) is required to make up the deficit. The cost, \( y_e \), is the electrical energy penalty cost. Also, the decrease in capacity due to higher ambient air temperatures penalizes capital investment because of the partially or entirely idled solar collector-storage unit. The capacity penalty cost, \( y_f \), is computed on the basis of yearly lost capacity of the solar unit.

Thus, the yearly incremental cost of the solar-powered Rankine-compression air conditioner with dry-cooling is

\[
y = y_c + y_o + y_e + y_f
\]

**Procedures.** - The procedure is to minimize the annual total cost of the system for using dry-cooling for given values of condensing temperature of the working fluid and compare costs. The minimization method applied herein is Rosenbrock's constrained hill climbing procedure (ref. 6). Four constraints are
imposed on the system: (a) air velocity is to be kept below 45.7 m/sec (150 ft/sec),
(b) condensing temperature is in the range of 37.8°C (100°F) and 54.4°C (130°F),
(c) the minimum allowable air pressure drop across the condenser is 9.76 kg/m²
(2 lb/ft²), and (d) the lower bound and upper bound of the mass velocity ratio of
entering vapor and air are functions of number of tube banks, air velocity, and
air side pressure drop.

Once the dry-cooling system has been designed for a given ITD, its performance
is evaluated at ambient air temperatures representative of local yearly fluctuations.
For this design ITD, a fixed air-cooled condenser size, fan and pump power are
assumed. Thus, when the condensing temperature is higher than the design value
due to high ambient air temperature, the cost of both energy and capacity penalties
is then computed for this ambient temperature and multiplied by the percent of the
year this temperature is experienced at the site.

The output of the program is the annual total incremental cost (between dry
cooling and wet-cooling), the design ITD, number of tube banks, condenser size,
mass flow rates of working fluid and air, and other optimum design specifications.

The flow diagrams of the computer design program is displayed in appendix A.
The input data requirement, notation and listing of the program are presented in
appendix B. A sample design output for use in Philadelphia is given in appendix C.

RESULTS AND DISCUSSION

Figures 3(a) and (b) shows the annual incremental cost due to the use of dry-
cooling in Philadelphia and Miami for using two different condenser surfaces, which
are taken from reference 7. Surfaces A and B respectively refer to figures 10-83
and 10-84 in that reference.

The curves along with the values shown indicate the percentage of cooling load
supplied by solar energy. For a large condenser (small ITD), the capital and
operating incremental costs are high, but little or no cooling capacity is lost (thus
little or no energy and capital penalty cost is involved) at high ambient air tempera-
tures. To meet the same percentages of the yearly cooling load (49 percent for
Miami and 77 percent for Philadelphia when a wet-cooling system is used), the
incremental cost for adapting a dry-cooling system is quite high.

It is noted that each point on the curve is a minimal optimum point corre-
sponding to that design ITD, and hence the true optimum for the dry-cooling sub-
system is that design ITD which yields the least cost. Each point also identifies
the percentage of the cooling load provided by solar energy. Since there is a
unique value of solar percentage for each point along the curve, the curve can be
interpreted on the basis of solar percentage of the cooling load as well as the
ITD. The curves are fairly smooth and consistent except a few points which
fluctuate along the curves. This is believed to be the consequence of specific
design requirements such as the number of tube banks having to be an integer and
the imposed allowable minimum air pressure drop across the condenser.

The curves in figure 3(a) and (b) result in definite minimum values of the cost
increment. In all cases, the cost increment is quite sensitive to the solar per-
centage of the cooling load - particularly that portion of the curve with solar per-
centages higher than at the minimum cost point. For surface A under Miami con-
ditions, the cost increment is approximately 200 percent greater for a decrease in
solar percentage of 3 percent. Although the cost increment also rises as the
solar percentage drops below the optimum point, it increases only approximately
15 percent for solar percentage decrease of 8 percent. Results are similar for
the other 3 cases.

The significance of the condenser surface design is seen by a comparison at
the optimum points. Use of surface B would cost $50 per year more compared
to surface A, to meet 73 percent of the cooling load in Philadelphia.

Figure 4 illustrates the itemized annual incremental cost for using a dry-cooling
system in Philadelphia. The contributions of capital, operating, and penalty
costs to the total incremental cost are clearly shown. The shaded area indicates
the capital savings at those design ITD's corresponding to relatively small con-
densers. Figure 5 shows the effect of number of tube banks on the incremental
cost. The dashed lines are used to indicate only the general trend since the
number of tube banks must be an integer.

The highlight of the optimum design specifications for Philadelphia and
Miami is tabulated in Table I. The breakdown of costs is also given therein. A
sample output of the design program is given in appendix C.

SUMMARY OF RESULTS

An optimum design procedure has been developed to estimate the annual in-
cremental cost for a three-ton solar-powered Rankine-compression air con-
ditioner equipped with a dry-cooling condenser over that with a wet-cooling
tower. Conclusions may be made as follows:

1. The sensitivity of the cost increment between the two cooling systems
illustrates the need for an optimization program when considering a dry-cooling
solar system.

2. Within the conditions and guidelines stated within the report, it was con-
cluded that -

   a. The cost increment was significantly higher (a greater penalty) when
   values of solar percentage of the cooling load (an input parameter) was greater
   than at the optimum point compared to when the values were smaller than optimum.
b. Dry-cooling is economically competitive with conventional wet-cooling whenever the maintenance cost of the latter (which is ignored in this study) is in the range of $100 to $200 per year, or greater.

c. The optimization program has proved to be adaptable to various locales and design requirements.

3. The dry-cooling solar system, which eliminates water vapor in the atmosphere, offers a better environment with a slightly higher or no additional cost as compared to the wet-cooling tower system.

Some significant remaining problems or questions are as follows:

1. Absorption air conditioning has not yet been considered in a dry cooling system due to the crystallization of the working fluid at high ambient temperatures. If this problem can be addressed and solved, the benefits of dry cooling would be applicable for such units.

2. Maintenance cost for water treatment of the wet-cooling tower used for solar-driven residential air conditioners.

3. Cost of heat exchangers for rejecting waste heat to ambient air, as required by dry cooling.
APPENDIX A

FLOW DIAGRAMS

Flow Diagram of Subroutine calling

MAIN

START

OPTIM

F

CX

CG

CH

MAIN: Main program.

START Finds starting values of variables such that they satisfy both the explicit and implicit constraints and do not lie within any boundary zone.

OPTIM Seeks minimum by means of constrained Rosenbrock method.

F Objective function of the total yearly bus-bar costs.

CX Constraint functions.

CG Lower bounds of CX.

CH Upper bounds of CX.
Subroutine Start

1. PICK N(OB) and A(ITD) FROM MAIN PROGRAM
2. COMPUTE G(A) SATISFYING DELPA, DETERMINE BOUNDS OF GRAR
3. COMPUTE STARTING GRAR & TR = 105
4. RETURN
5. END
Main Program

- Baseline design of wet-cooling system; Percentage of cooling supplied by solar; Thermal efficiency of rankine cycle; COP; Overall COP; Weather; Costs; Heat exchanger surface; Minimum air pressure across heat exchanger; Fluid properties; Thermal efficiencies of components; etc.

Procedure:
1. READ DATA
2. I = 1, I1; KZ = 1
3. NOB = 1, 10
4. CALL START
5. CALL OPTIM
6. PRINT RESULT

Use the controller KZ to reduce the increment of LT D to one and compute results near prior minimum obtained from a particular I.
Pick base point $X_i$ and initial step sizes, $S_i$
\[ i = 1, 2, 3, \ldots, N \]
and evaluate objective function

1

increment $X_i$ from best point
A distance $S_i$ parallel to axis
and evaluate objective function

Point feasible?

Yes

In boundary zone?

No

Function improvement?

Modify function

Yes

\[ S_i^{\text{new}} = \alpha S_i^{\text{old}} \]
\[ 0 \leq \alpha \leq 1 \]

No

\[ S_i^{\text{new}} = -\beta S_i^{\text{old}} \]
\[ 0 < \beta < 1 \]

Yes

1 $\rightarrow$ N

No

Convergence obtained?

One success and one failure in each direction?

No

Rotate axes

Yes

Stop

Set step sizes
Function F

1. Pick data from optim
2. ThermoFluid calculation
3. Cost calculation
4. Calculate performances affected by local weather
5. Calculate yearly bus-bar cost F
6. Return
7. End
Function CX

CALCULATE CONSTRAINT FUNCTIONS, $CX_i$, $i = 1, 2, \ldots, L$

RETURN

END

Function CG

Find LOWER BOUNDS OF CX

RETURN

END

Function CH

Find UPPER BOUNDS OF CX

RETURN

END
APPENDIX B

PROGRAM INPUT DATA, NOTATION, AND LISTING

Input Data

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AITD</td>
<td>Initial temperature difference (temperature difference between the condensing R-12 and the ambient air), (°F)</td>
</tr>
<tr>
<td>AKA</td>
<td>Thermal conductivity of air (Btu/hr-ft-°F)</td>
</tr>
<tr>
<td>AKRL</td>
<td>Thermal conductivity of liquid R-12 (Btu/hr-ft-°F)</td>
</tr>
<tr>
<td>ALPHA</td>
<td>Ratio of total transfer area of air-side of heat exchanger to total exchanger volume</td>
</tr>
<tr>
<td>AMUA</td>
<td>Dynamic viscosity of air (lb/hr-ft)</td>
</tr>
<tr>
<td>AMURG</td>
<td>Dynamic viscosity of R-12 vapor (lb/hr-ft)</td>
</tr>
<tr>
<td>AMURL</td>
<td>Dynamic viscosity of liquid R-12 (lb/hr-ft)</td>
</tr>
<tr>
<td>C1 - C2</td>
<td>Coefficients of curve fit for Rankine cycle efficiency</td>
</tr>
<tr>
<td>C3 - C5</td>
<td>Coefficients of curve fit for compression cycle COP</td>
</tr>
<tr>
<td>C6 - C8</td>
<td>Coefficients of curve fit for enthalpy of R-12 vapor</td>
</tr>
<tr>
<td>C9 - C11</td>
<td>Coefficients of curve fit for enthalpy of R-12 liquid</td>
</tr>
<tr>
<td>C12 - C13</td>
<td>Coefficients of curve fit saturation pressure of R-12</td>
</tr>
<tr>
<td>C14 - C15</td>
<td>Curve fit constants for dimensionless heat transfer coefficients for heat exchanger surface</td>
</tr>
<tr>
<td>C16 - C17</td>
<td>Curve fit constants for friction factor for heat exchanger surface</td>
</tr>
<tr>
<td>C18 - C20</td>
<td>Coefficients of curve fit for specific volume of R-12 liquid</td>
</tr>
<tr>
<td>C21 - C23</td>
<td>Coefficients of curve fit for specific volume of R-12 vapor</td>
</tr>
<tr>
<td>COSEM</td>
<td>Cost of electric motor ($/Hh)</td>
</tr>
<tr>
<td>COSFB</td>
<td>Fan blade cost per area covered ($/ft²)</td>
</tr>
<tr>
<td>CPA</td>
<td>Specific heat of air (Btu/lb-°F)</td>
</tr>
<tr>
<td>CPRL</td>
<td>Specific heat of liquid R-12 (Btu/lb-°F)</td>
</tr>
<tr>
<td>CTPEA</td>
<td>Cost of cooling tower per unit total area of exchanger ($/ft²)</td>
</tr>
<tr>
<td>COSPW</td>
<td>Cost of electricity ($/kWh)</td>
</tr>
<tr>
<td>COSTC</td>
<td>Cost of Solar collector ($/ft²)</td>
</tr>
<tr>
<td>DEQ</td>
<td>Hydraulic diameter = 4 x flow passage hydraulic radius (ft)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>DITCH</td>
<td>Number of fins per foot (fins/ft)</td>
</tr>
<tr>
<td>EFFAN</td>
<td>Fan efficiency</td>
</tr>
<tr>
<td>EFP</td>
<td>Pump efficiency</td>
</tr>
<tr>
<td>FCR</td>
<td>Fixed-charge rate (percent/100)</td>
</tr>
<tr>
<td>FERC</td>
<td>Recoverable energy factor for moving air (percent/100)</td>
</tr>
<tr>
<td>FTHK</td>
<td>Fin thickness of exchanger (ft)</td>
</tr>
<tr>
<td>OD</td>
<td>Tube diameter of heat exchanger (ft)</td>
</tr>
<tr>
<td>PAI</td>
<td>Entering pressure of air ($\pi r^2$)</td>
</tr>
<tr>
<td>PERSS</td>
<td>Percentage of cooling load supplied by solar</td>
</tr>
<tr>
<td>PRA</td>
<td>Prandtl number of air</td>
</tr>
<tr>
<td>PRRL</td>
<td>Prandtl number of liquid R-12</td>
</tr>
<tr>
<td>QE</td>
<td>Cooling capacity of the unit (Btu/hr)</td>
</tr>
<tr>
<td>RFATA</td>
<td>Ratio of fin area to total area of the air side of exchanger</td>
</tr>
<tr>
<td>RHOA</td>
<td>Density of air (lb/ft³)</td>
</tr>
<tr>
<td>RHOR</td>
<td>Density of liquid R-12 (lb/ft³)</td>
</tr>
<tr>
<td>SIGMA</td>
<td>Ratio of free-flow to frontal area of air side of exchanger</td>
</tr>
<tr>
<td>SLF</td>
<td>Fin height of exchanger (ft)</td>
</tr>
<tr>
<td>TAI1 (F)</td>
<td>Ambient air temperatures used in evaluating dry-cooling tower performances (F)</td>
</tr>
<tr>
<td>TCB</td>
<td>Condensing temperature of R-12 of baseline design wet cooling system (F)</td>
</tr>
<tr>
<td>TEB</td>
<td>Fluid temperature in the evaporator (F)</td>
</tr>
<tr>
<td>TINXB</td>
<td>Temperature of fluid entering the expander (F)</td>
</tr>
<tr>
<td>TKF</td>
<td>Thermal conductivity of fin material of exchanger (Btu/hr-ft-F)</td>
</tr>
<tr>
<td>TPER1 (F)</td>
<td>Fraction of year TAI1 (F) experienced</td>
</tr>
<tr>
<td>VADF</td>
<td>Velocity of air delivered by fan (ft/sec)</td>
</tr>
<tr>
<td>XD</td>
<td>Longitudinal tube spacing (depth pitch) of exchanger (ft)</td>
</tr>
<tr>
<td>XW</td>
<td>Transverse tube spacing (width pitch) of exchanger (ft)</td>
</tr>
<tr>
<td>WETC</td>
<td>Cost of wet-cooling tower and &quot;wet-condensers&quot; ($)</td>
</tr>
</tbody>
</table>
Notation

A   Total heat transfer area of air-side of heat exchanger (ft²)
ACCL Annual penalty capital and operating costs due to ambient air temperatures above design temperature ($/yr)
ACOS Total annual bus-bar cost of the system ($/yr)
ACPC Annual penalty capital cost due to ambient air temperatures above design temperature ($/yr)
AEPC Annual penalty operating cost due to ambient air temperatures above design temperature ($/yr)
AFR Frontal area of heat exchanger (ft²)
ALMTD Log-mean temperature difference (F)
ALT Tube length of exchanger (ft)
AMC Mass flow rate of R-12 in the compression cycle (lb/hr)
AMP Mass flow rate of R-12 in the Rankine cycle (lb/hr)
ANTU Number of transfer unit of exchanger
ATUB Total area of tubes of exchanger (ft²)
COP Coefficient of performance of compression cycle
DA General storage vector
DELPA Air pressure drop across exchanger (lb/ft²)
DELPR R-12 pressure drop inside tube (lb/ft²)
DELTA Temperature increase of air across exchanger (F)
DELY Difference between current value and previous stage value of objective function ($/yr)
DEP Depth of exchanger (ft)
E Vector of initial step sizes
EFF Fin efficiency of exchanger (percent/100)
EFFP Thermal efficiency of the Rankine cycle (percent/100)
EFZ Temperature effectiveness of exchanger fin (percent/100)
EPS Heat exchanger effectiveness (percent/100)
FAIR Friction factor of air across exchanger
FR Friction factor of R-12 flowing inside of tube
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>GA</td>
<td>Mass velocity of air (lb/hr-ft²)</td>
</tr>
<tr>
<td>GM</td>
<td>Mean mass velocity of R-12 vapor (lb/hr-ft²)</td>
</tr>
<tr>
<td>GRAR</td>
<td>Mass velocity ratio of vapor R-12 to air</td>
</tr>
<tr>
<td>HA</td>
<td>Mean air-film conductance of a finned exchanger (Btu/hr-ft²-F)</td>
</tr>
<tr>
<td>HPA</td>
<td>Fan power for air (HP)</td>
</tr>
<tr>
<td>HPR</td>
<td>Pump power for R-12 (HP)</td>
</tr>
<tr>
<td>HR</td>
<td>Mean R-12 film conductance in a pipe (Btu/hr-ft²-F)</td>
</tr>
<tr>
<td>K</td>
<td>Point index</td>
</tr>
<tr>
<td>L</td>
<td>Number of variables plus number of constraints</td>
</tr>
<tr>
<td>LOCA</td>
<td>Number used to identify weather station</td>
</tr>
<tr>
<td>LOOPY</td>
<td>Maximum number of stages to be calculated</td>
</tr>
<tr>
<td>M</td>
<td>Optimization controller; +1 for maximization, -1 for minimization</td>
</tr>
<tr>
<td>N</td>
<td>Dimension limit in subfunctions F, CX, CG and CH</td>
</tr>
<tr>
<td>ND</td>
<td>Storage controller; 1 for storage in DA, 0 for no storage</td>
</tr>
<tr>
<td>NDATA</td>
<td>Number of data points to be stored in DA</td>
</tr>
<tr>
<td>NHXS</td>
<td>Number used to identify heat exchanger surface</td>
</tr>
<tr>
<td>NOB</td>
<td>Number of tube banks parallel to direction of air flow</td>
</tr>
<tr>
<td>NPAR</td>
<td>Dimension limit in subfunctions F, CX, CG, and CH</td>
</tr>
<tr>
<td>NSTEP</td>
<td>Control of step sizes for each rotation; 0 for original step size, 1 for step size from previous rotation of axes</td>
</tr>
<tr>
<td>OCOP</td>
<td>Overall coefficient of performance of the air conditioner (= EFP x COP)</td>
</tr>
<tr>
<td>OLHX</td>
<td>Overall length of the air-cooled condenser (ft)</td>
</tr>
<tr>
<td>OWHX</td>
<td>Overall width of the air-cooled condenser (ft)</td>
</tr>
<tr>
<td>P</td>
<td>Number of variables</td>
</tr>
<tr>
<td>PFKW</td>
<td>Fan power for air (KW)</td>
</tr>
<tr>
<td>PRKW</td>
<td>Pump power for R-12 (KW)</td>
</tr>
<tr>
<td>PR</td>
<td>Printing controller (number of stages between outputs)</td>
</tr>
<tr>
<td>QB</td>
<td>Solar energy collected by the collector (Btu/hr)</td>
</tr>
<tr>
<td>QBB</td>
<td>Solar energy collected by the collector for the baseline design unit (Btu/hr)</td>
</tr>
</tbody>
</table>
QRC  Heat rejection from the compression cycle (Btu/hr)
QRI  Heat rejection through the air-cooled condenser (Btu/hr)
QRP  Heat rejection from the Rankine cycle (Btu/hr)
REA  Reynolds number of air flow
RER  Reynolds number of R-12 flow
RTAIA Ratio of air-side area to R-12 side area of exchanger
SS   Fraction of cooling load supplied by solar energy
STPR Product of Stanton number and Prandtl number to the 2/3 power
SVRG Specific volume of R-12 vapor (ft³/lb)
SVRL Specific volume of liquid R-12 (ft³/lb)
TAI  Temperature of air entering the condenser (F)
TAO  Temperature of air leaving the condenser (F)
TR   Design condensing temperature of R-12 (F)
U    Overall heat transfer coefficient based on air-side area (Btu/hr-ft²-F)
UITA Overall heat transfer coefficient based on R-12 side area (Btu/hr-ft²-F)
V    Direction vector
VA   Air velocity (ft/sec)
VOL  Volume of heat exchanger (ft³)
VRI  Velocity of R-12 vapor entering the condenser (ft/sec)
WA   Mass flow rate of air (lb/hr)
WCOMP Power required by the compressor (Btu/hr)
WEXP Power generated by the expander (Btu/hr)
WPUMP Power required by the pump (HP)
WR   Total mass flow rate of R-12 (lb/hr)
X    Independent variables
XI   Area of solar collector (ft²)
YC   Total annual bus-bar capital cost ($) / yr
YCI  Capital cost of solar collector and storage tank ($)
YC2  Capital cost of air-cooled condenser ($)
YC3  Capital cost of electric motors ($)
<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>YC4</td>
<td>Capital cost of fan blades ($)</td>
</tr>
<tr>
<td>YCO</td>
<td>YC + YO</td>
</tr>
<tr>
<td>YO</td>
<td>Annual bus-bar operating cost ($/yr)</td>
</tr>
</tbody>
</table>
SAMPLE OF PROGRAM OUTPUT

**OPTIMUM DRY COOLING SYSTEMS FOR A SOLAR AIR CONDITIONER**

**MAIN PROGRAM**

```plaintext
DIMENSION DIT(15), DA(70), LE(3), XX(3), TC(2), Z(20), WW(50)
COMMON ACP, A10, AK, AKL, ALFA, AMU, AMRG, AMUL, C1, C2, C3, C4,
1C5, C6, C7, C8, C9, C10, C11, C12, C13, C14, C15, C16, C17, C18, C19, C20,
2C21, C22, C23, COP, COP, C0SP, COSF, COSF, COST, COST, CPA, CPRL, CTPA, DEO,
3DITCH, EFFAN, EFFPB, EFFC, FCR, FCR, FTHR, KOUNT, OD, PA1, PRA,
4PRRL, PZ, PRRA, KA1A, K1A, SIGA, SLF, ALA1, 151, RCS, TEB,
5STINX, TRF, TRIE, 151, VATD, WETC, X0, XW, XZ, H6, PERSS, Z1

INTEGER XZ

READ (5, 001) CIE, TRB, STIN, TEB
READ (5, 001) AMU, AK, CPA, PRA, RHOA, RHOR
READ (5, 001) (TA(I), I = 1, 5)
READ (5, 002) CPA, AMUL, AMRG, AKL, PRRL
READ (5, 002) COST, CIPE, COSM, COSF, FCR
READ (5, 002) VATD, PZ, EFFM, FERC, EFFAN
READ (5, 003) LOCA, (PERI(I), I = 1, 5)
READ (5, 004) NHXS, OD, PZ, DEQ, SIGA, ALFA, XM
READ (5, 004) NHXE, X0, FTHR, DITCH, TRF, SLF, HRY
READ (5, 005) C1, C2, C3, C4, C5
READ (5, 006) C6, C7, C8, C9, C10, C11, C12, C13
READ (5, 007) C14, C15, C16, C17
READ (5, 008) C18, C19, C20, C21, C22, C23
READ (5, 009) (IND(I), I = 1, 9) WETC, C0SP, PERSS

001 FORMAT (1X, 3F10.4)
002 FORMAT (5F10.4)
003 FORMAT (15, 5F5.2)
004 FORMAT (110, 6F10.5)
005 FORMAT (5F10.6)
006 FORMAT (6F10.4)
007 FORMAT (4F10.4)
008 FORMAT (1F10.6, 2F20.15, 2F10.6)
009 FORMAT (3F10.2)
010 FORMAT (6F10.4)

NDATA = 50
21 = PERI(11) + PERI(21) + PERI(13) + PERI(14) + PERI(15)
P2 = EP(C12 + C13 + DIF + 460.4)
H6 = C6 + C7 + TRB + CB + TR + EB
TR = TEB
EFFP = C1 + C2 + TR
COP = C1 + C4 + TR + CB + TR + EB
OCP = EFFP + COP
QB = QQ / COP
ACF = QQ / 80.
DA(51) = 0.2 * C171
DA(63) = 1 - FTHR * DITCH / (1 - RZK1A)
DA(64) = 0.0109755 * COP / (PRL + PR11 * (AMURG / OD) + OD)
DA(65) = 0.01F13.1416 * CD / CD)
```
DA(66)=2.0
DA(67)=16.1*3600.002*DEQ((AMUA/DLO)**C17/(C16+X0)
RTA10=1./C2.*C17)
DIM(13)
D1(11)=10.
DO 200 I=1,10
D1(11)*I=1,10
200 CONTINUE
C
*** CALCULATE MAX. AND MIN. MASS FLOW RATE WR ***
C
TC(11)=100.
TC(12)=130.
DO 102 K=1,2
TR=TC(K)
EFFP=C1*C2*TR
COP=C3*C4*TR+C5*TR+TR
Q=QL*QOP
QR=QF*QRF
HG=C6*C7*TR+C8*TR+TR
HF=C9*C10*C11*TR+TR
AMP=QR/PH
AMC=QF/PH
WR=AMP+AMC
DA(K)*601=WR
102 CONTINUE
WRITE(6,001) QE,TC,TR,INP6,TST
WRITE(6,001) AMUA,AKP,CPA,PA2,RHDA,RHOR
WRITE(6,002) TA11(11),I=1,5
WRITE(6,002) TPR,AMUR,AMUR,AKK,PRPL
WRITE(6,002) COSIC,CTPEC,COSEM,COFRC,FCR
WRITE(6,002) VADF,PA1,FF,FERC,FFAN
WRITE(6,003) LOCA,(P1ER1111),I=1,5
WRITE(6,004) NHXS,DD,FFATA,DEC,SIGMA,ALPHA,XW
WRITE(6,004) NHXS,XD,TH8,THC,TF8,SLF,MYR
WRITE(6,005) C1,C2,C3,C4,C5
WRITE(6,006) C6,C7,C8,C9,C10,C11,C12,C13
WRITE(6,007) C14,C15,C16,C17
WRITE(6,008) C18,C19,C20,C21,C22,C23
WRITE(6,009) WEL,COSPK,PLRSS
WRITE(6,3000) DA(11),DA(21),DA(31),DA(41),DA(51),DA(61),DA(67),
10A(79)
3000 FORMAT (HE12.5/1
R7=1
DO 100 I=1,11
A10=1111
118 DO 110 NOP=1,10
DA(21)=NOP
C
C FIND STARTING VALUES WITHIN CONSTRAINTS
C
CALL START (XX,DA)
GO TO 000,011,12
31 L1(11)=1.5
EC2=2.0
DO 101 J=1,NDATA
DA(J)=0.0
101 CONTINUE
   CALL OPTIMXX,LE,DA
   YCO=DA(35)+DA(35)
   ACOS=YCO+DA(45)
   WRITE (6,1005)
1005 FORMAT (' ')
   WRITE (6,1006)
1006 FORMAT (' ')
   WRITE (6,1007)
1007 FORMAT (*
   WRITE (6,1008) NHXS,A1TO,DA(21),DA(11),XX(1),XX(21),DA(521)
1008 FORMAT (*)
   WRITE (6,1009) M"POS" M"POS" M"POS" M"POS"
1009 FORMAT (*
1004 FORMAT (*)
   WRITE (6,1006) NHXS="12", A1TO="FS.1", M"POS"="1.4"," DRP"
   IF (7.3, 0.147, 1.1, 2.5, 0.2, 0.1) WRITE (6,1004) NHXS, A1TO, DA(21), DA(11), XX(11), XX(21), DA(521)
1004 FORMAT (*)
   WRITE (6,1008) DA(34), DA(351), YCO, DA(50), ACOS
1008 FORMAT (*)
   WRITE (6,1009) M"POS" M"POS" M"POS" M"POS"
1009 FORMAT (*)
   WRITE (6,1006) NHXS="11.4", YCO="11.4", ACOS="11.4"
1006 FORMAT (*)
   WRITE (6,1007)
1007 FORMAT (*)
1004 FORMAT (*)
   WRITE (6,1006) Z(1/10)=ACOS
   IF (NOB=10, 11) GO TO 110
   IF (Z(1/10)=-Z(1/10)) 110, 110, 111
111 GO TO (112, 113, 115, 199), K2
112 WW(1)=2(Z(1/10)-1)
   IF (WW(1)<LE, WW(1)-1)) GO TO 100
   IF (WW(1)-LE, WW(1)-1)) GO TO 100
   K2=2
   AL10=AL10-1.1, 0
   GO TO 114
113 AL10=AL10-1.0
   IF (AL10<AL10-1.21) GO TO 114
   K2=5
   AL10=AL10-1.1, 1.0
   GO TO 114
115 AL10=AL10+1.0
   IF (AL10<AL10-1.1) GO TO 114
   K2=0
   AL10=AL10+1.1
   GO TO 114
110 CONTINUE
100 CONTINUE
999 STOP
SUBROUTINE START (x,DA)
DIMENSION X(31),DA(70)
COMMON FCR,ATDC,AKA,AKRL,ALPHA,AMUI,AMURC,AMURC,C1,C2,C3,C4
CO,C6,C7,C8,C9,C10,C11,C12,C13,C14,C15,C16,C17,C18,C19,C20
2C21,C22,C23,C24,C25,C26,C27,C28,C29,C30,C31,C32,C33,C34,
30I8H,EFFAN,EFPR,ELF,FRE,FEK,FTHK,FHYR,KOUNT,OD,P2,P4,P5,
UPRL,CE,CFR,KFTA,KHOA,PHGR,RTAIA,SLF,TA111(T15),TCA,TEB,
STINX,TINF,TPER(T15),VADF,WEIT,XD,XW,XZ,H6,PERS,21
INTEGER XZ
DA(1)=DA(2)*X
VAMAX=120.
XMIN=DA(165)+DA(1)/3600.*RHOA*DA(2)*VAMAX
GAMIN=DA(67)+DA(66)/DA(2)*RHOA*DA(2)*VAMAX
XIMAX=DA(165)+DA(62)/DA(2)*GAMIN*DA(2)
DA(68)=XMIN
DA(69)=XMAX
VX=GAMIN/RHOA*3600.*
IF (VA-5.) 16,17,17
17 XX=MAX*DA(1)/DA(2)
XX=105.
XZ=2
WRITE (6,151) XX,XX+XX+XX
15 FORMAT (E15.7/)
GO TO 13
16 XZ=1
13 RETURN
END
IF (STEP.EQ.0) C(J) = INITIAL(J)
50 A(J) = 2.0
70 B(J) = 0.0
FREST = F1
80 I = 1
   IF (INIT.FEQ.0) GO TO 120
90 DO 110 K = 1, P
110 X(K) = X(K) + (1) * V(1, K)
   IF (X(K).LT.0.0) GO TO 501
   WRITE (6, 031) K, 1, (1) * V(1, K), X(K)
501 CONTINUE
   DO 50 K = 1, L
50 H(K) = 0.0
120 F1 = F(X, DA, N, NPAR)
   F1 = F1
   IF (ISW.EQ.0) FO = F1
   ISW = 1
   IF (APS.FEQ.0.0) DELY1 = 122, 122, 125
122 TERM = 1.0
   GO TO 450
125 CONTINUE
C
J = 1
C
130 XC = CX(X, DA, N, NPAR, J)
   LC = CG(X, DA, N, NPAR, J)
   UC = CH(X, DA, N, NPAR, J)
   IF (XC.LE.LC) GO TO 420
   IF (XC.GE.UC) GO TO 420
   IF (F1.LT.FO) GO TO 470
   IF (XC.LE.LC+AL(J)) GO TO 140
   IF (XC.GE.UC+AL(J)) GO TO 140
   H(K) = F0
   GO TO 210
140 CONTINUE
Bw = AL(J)
   IF (XC.LE.LC.OR.XC.LE.LC) GO TO 150
   IF (LC.LE.XC.AND.XC.LE.LC+LW) GO TO 160
   IF (LC.LE.XC.AND.XC.LE.LC) GO TO 170
   PH(J) = 1.0
   GO TO 210
150 PH(J) = 0.0
   GO TO 190
160 PW = (LC-PC-XC)/BW
   GO TO 180
170 PW = XC-UC+BW)/BW
180 PH(J) = 1.0-3.0*PW+4.0*PW-PW-2.0*PW+PW
190 F1 = H(J)+F1-H(J))*PH(J)
210 CONTINUE
   IF (J.EQ.L) GO TO 220
J = J + 1
   GO TO 130
220 INIT = 1
   IF (F1.LT.FO) GO TO 420
   D(J) = D(J) + 1.0

ORIGINAL PAGE IS OF POOR QUALITY
E(II)=3.0*E(I)
DO 230 JJ=1,P
IF (SA(JJ).GE.1.5) SA(JJ)=1.0
230 CONTINUE

C AXES ROTATION
C
DO 250 R=1,P
DO 250 C=1,P
250 VV(R,C)=0.0
DO 260 R=1,P
KR=R
DO 260 C=1,P
DO 260 K=KR,P
260 BMAG=0.0
DO 280 C=1,P
BMAG=BMAG+B(I,C)*B(I,C)
280 CONTINUE
BMAG=SQRT(BMAG)
BX(I)=BMAG
DO 310 C=1,P
310 VV(I,C)=BMAG
DO 390 R=1,P
IR=R-1
DO 390 C=1,P
SUMV=0.0
DO 420 KK=1,IR
SUMV=SUMV+V(K,C)
420 SUMV=SUMV+V(K,C)
DO 340 P=1,P
SUMAV=0.0
DO 350 K=1,P
350 BBMAG=SUMAV+B(R,K)*B(R,K)
SUMAV=SUMAV+B(R,K)
DO 340 C=1,P
340 VV(R,C)=BBMAG
LOOP=LOOP+1
LAP=LAP+1
IF (LAP.LE.NP) GO TO 450
GO TO 1000
420 IF (INIT.EQ.0) GO TO 450
DO 430 IX=1,P
430 XI=X(I)-X(1)*V(1,I)
E(I)=0.5*(X(I)
IF (SA(I).LT.1.5) SA(I)=0.0
GO TO 230
440 CONTINUE
IF (1.EQ.P) GO TO 60
I=1+1
GO TO 90
450 WRITE (6,003)
003 FORMAT (/7,2X,5HSTAGE,8X,8HJUNCTION,12X,8HPROGRESS,9X,)
116HORIZONTAL PROGRESS
WRITE (6,004) LOOP,LO,PMAG,HRMAG
004 FORMAT (1H,15,3E20.8)
WRITE (6,004) KJ,KJ
014 FORMAT (2X,33HNUMBER OF FUNCTION EVALUATIONS = ,I8)
WRITE (6,005)
005 FORMAT (2X,25HVALUES OF X AT THIS STAGE)
C
PRINT CURRENT VALUES OF X
WRITE (6,006) XM, XX(JM), XM=1,PM
006 FORMAT (3X,3E14.6,4X,11)
ELSE
IF LIMIT.EQ.0.0 GO TO 470
IF (TERM.EQ.1.0) GO TO 480
IF (LOOP.EQ.1.0) GO TO 480
GO TO 1000
470 WRITE (6,007)
007 FORMAT (7/4X,6H1N THE STARTING POINT MUST NOT VIOLATE THE CONSIDERED
1NTEGRALS. IT APPEARS TO HAVE DONE SO.)
480 CONTINUE
WRITE (6,019)
019 FORMAT (/5H**** OPTIMUM DESIGN SPECIFICATIONS *****/)
WRITE (6,015) DA(1),DA(4),DA(5),DA(6),DA(7)
WRITE (6,016) DA(8),DA(9),DA(10),DA(11),DA(12)
WRITE (6,017) DA(13),DA(14),DA(15),DA(16),DA(17)
WRITE (6,018) DA(18),DA(19),DA(20),DA(21),DA(22)
WRITE (6,020) DA(23),DA(24),DA(25),DA(26),DA(27)
WRITE (6,021) DA(28),DA(29),DA(30),DA(31),DA(32)
WRITE (6,022) DA(33),DA(34),DA(35),DA(36),DA(37)
WRITE (6,023) DA(38),DA(39),DA(40),DA(41),DA(42)
WRITE (6,024) DA(43),DA(44),DA(45),DA(46),DA(47)
WRITE (6,025) DA(48),DA(49),DA(50),DA(51),DA(52)
015 FORMAT (2X,YA-1,10.3, EPS-1,10.3, A-1,14.7, DEP-1,12.5,)
1'1. AR-1.14.7, 1)
016 FORMAT (4AI-1.12.5, AM-1.12.5, AIUB-1.14.7, U-1.12.5,)
1'1. PM-1.12.5, 1)
017 FORMAT (4AI-1.12.5, UM-1.12.5, VR-1.12.5, RIA-1.12.5,)
1'1. IR-1.12.5, 1)
018 FORMAT (4AI-1.14.7, WR-1.14.7, IA-1.12.5, DELT-1.12.5,)
1'1. AMLO-1.12.5, 1)
020 FORMAT (4AI-1.12.5, HPA-1.14.5, DEPN-1.12.5, HPR-1.12.5,)
1'1. ORJ-1.14.7, 1)
021 FORMAT (4AI-1.14.7, EFFP-1.12.5, OWX-1.12.5, OLMX-1.12.5,)
1'1. EFF-1.12.5, 1)
022 FORMAT (4AI-1.12.5, CPC-1.12.5, WCOMP-1.12.5,)
1'1. CPC-1.12.5, 1)
023 FORMAT (4AI-1.12.5, AMP-1.12.5, AMC-1.12.5, WPUH-1.12.5,)
1'1. AMP-1.12.5, 1)
024 FORMAT (4AI-1.12.5, YC1-1.12.5, YC2-1.12.5, YC3-1.12.5,)
1'1. YC4-1.12.5, 1)
025 FORMAT (4AI-1.12.5, ABCP-1.11.4, ACBC-1.11.4, D(5,6)=1.11.4/)
473 RETURN
END
FUNCTION F (X,DA,N=NPAR)
DIMENSION X (N), DA(NPAR)
COMMON ACR, ATO, ANA, APRL, ALPHA, AMUA, AMORG, AMULG, C1, C2, C4,
      C6, C7, C8, C9, C10, C11, C12, C13, C14, C15, C16, C17, C18, C19, C20,
      C21, C22, C23, COPR, COSEM, COSFB, COSPW, COSC, CPA, CPRL, CTPA, DEQ,
      ITIC, IIFAN, IFFP, IFF, ICR, IER, ITH, IHR, ION, IOD, IPZ, IFA, PRA,
      NPRL, O1, ORI, RATA, RHOA, RHOK, RT1A, SIGMA, SLF, TAI, T15, TCR, TCL,
      STNXB, TNP, TPER, VADF, WTC, XD, XW, XZ, H6, PEN55, Z1
GRAN=X(1)
IF (X(11) .LE. 0.1) GO TO 501
WRITE (6,001) X(1:1)
001 FORMAT (115.7)
501 CONTINUE
TR=X(1)
EFF=0.001**TR
COP=0.0001**TR
OCP=EFF*COP
QCR=0.001**COP
CR=0.01**CR
HE=0.001**TR
NE=0.01**TR
AF=0.001**AF
AM=0.01**AM
AT=0.001**AT
NM=0.01**NM
HO=0.001**HO
LR=0.01**LR
VR=0.001**VR
VRG=0.01**VR
G1=DA(65) 4 110 11.1
GM=0.58**GR1
GA=GR1 4 GRAN
VA=GA/10**20 *RH0A
KN=0.1**KN*AMUA
SN=0.1**SN*APRL
FA=0.1**FA**CPR 19 1.17
HA=CPA*GM**PR/PR/PR 19 1.67
AM=SQRT(1.*HA/11*THF*THK)
EFF=1 *AM 1.1131
THF=111.111111111
THK=111.111111111
RM=0.001**RM
MR=0.01**MR
UR=1.1111111111
UT=1.1111111111
AN=111.111111111
ANP=111.111111111111111
EPS=1.111111111
DELTA=111.1111111111
WAX=0.4111111111
AFA=0.4111111111
ALT=AFA/X
WCR=0.4111111111
AMORG
DA(301)=OWHX
DA(311)=GLHX
DA(321)=EFZ
DA(331)=A2
DA(341)=YC
DA(351)=YO
DA(361)=COP
DA(371)=COP
DA(381)=WCOMP
DA(391)=COP
DA(401)=COP
DA(411)=COP
DA(421)=AMC
DA(431)=WPUHP
DA(441)=WEXP
DA(451)=X1
DA(461)=YC1
DA(471)=YC2
DA(481)=YC3
DA(491)=YC4

C

***AMBENT AIR TEMPERATURE EFFECT***

AEPC=0.
ACPC=0.
TPSS=0.
COPR=COP
DO 100 K=1,5
TRR=ATD+TA11(K)
EFFP=C1+C2*TRR
COPR=C3+C4*TRR+C5*TRR+TRR
WCOMP=QE+COPR
QPP=QR-WCOMP-QE
GR=COPR/(1-EXP)
WM=WCOMP-EFFP+GR
IF (WM.LE.0.) GO TO 100
PERWM=WM/WCOMP
TPER=WPERWM*TPER1(K)/Z1
TPSS=TPSS+TPER
EPC=WM*2.298E-4*TPER1(K)*67.6*COSPW
CPC=WM/QL*COPB*YC1*FCR*TPER1(K)/Z1
AEPC=AEPC*EPC
ACPC=ACPC*EPC
ACCL=AEPC+ACPC

100 CONTINUE
DA(501)=ACCL
DA(521)=EFZ
DA(531)=A2
DA(551)=AEPC
DA(552)=ACPC
DA(553)=YC+YO+ACPC
F=YC+YO+ACCL
KOUNT=KOUNT+1
RETURN
END
FUNCTION CX (X, DA, N, NPAR, K)
DIMENSION X(N), DA(NPAR)
IF (K.GE.3) GO TO 1
CX=X(K)
GO TO 5
1 KK=K-2
GO TO (4,2,KK)
2 CX=DA(3)
GO TO 5
4 CX=DA(23)
5 RETURN
END

FUNCTION CG (X, DA, N, NPAR, K)
DIMENSION X(N), DA(NPAR)
GO TO (2,4,9,6,K)
2 CG=DA(8)
GO TO 9
3 CG=100.
GO TO 9
4 CG=DA(66)
GO TO 9
6 CG=4.0
9 RETURN
END

FUNCTION CH (X, DA, N, NPAR, K)
DIMENSION X(N), DA(NPAR)
GO TO (2,4,9,6,K)
2 CH=DA(6)
GO TO 9
3 CH=130.
GO TO 9
4 CH=DA(51)*DA(66)
GO TO 9
6 CH=130.
9 RETURN
END
### Subroutine Optim Data

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<tr>
<th>Stage</th>
<th>Function</th>
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<th>Lateral Progress</th>
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Number of function evaluations: 6

Values of \( x \) at this stage:
\[ x(1) = 5.40568*01 \quad x(2) = 1.04000*02 \]

### Optimum Design Specifications

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<th>DEP</th>
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<td>1.6779*06</td>
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REFERENCES


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<th>Miami</th>
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<td>Heat exchanger surface</td>
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<td>A</td>
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<td>11.1 (20)</td>
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<td>Number of tube banks</td>
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<td>7</td>
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<td>Depth, m (ft)</td>
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<tr>
<td>Overall width, m (ft)</td>
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<td>0.915 (3.0)</td>
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<tr>
<td>Overall length, M (ft)</td>
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<td>0.915 (3.0)</td>
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<td>Flow rate (Rankine), kg/hr (lb/hr)</td>
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<td>383 (825)</td>
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<td>321 (707)</td>
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<tr>
<td>Air velocity, m/sec (ft/sec)</td>
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<td>5.9 (19.5)</td>
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<td>Vapor velocity, m/sec (ft/sec)</td>
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<td>Inside pressure drop, kg/m² (lb/ft²)</td>
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<td>59 (635)</td>
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<td>Total</td>
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Figure 1. - Schematic solar Rankine compression air conditioner with P-h diagram for R-12.

Figure 2. - Rankine-compression system performance.
Figure 3a - Annual incremental cost of the dry-cooling system with heat exchanger surface A.
Figure 30. Annual incremental cost of the dry-cooling system with heat exchanger surface B.

Percentage of cooling load supplied by solar.
Figure 4 - Itemized annual incremental cost of the dry-cooling system in Philadelphia.

Figure 5 - Effect of number of heat exchanger tubes on cost in Philadelphia.