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BNWL-2180 Vol. I
UC-12

A User's Manual for the BNW-I Optimization Code for Dry-Cooled Power Plants

Volume I

January 1977

Prepared for the Energy Research
and Development Administration
under Contract EY-76-C-06-1830

Battelle
Pacific Northwest Laboratories

BNWL-2180 Vol. I

PACIFIC NORTHWEST LABORATORY
operated by
BATTELLE
for the
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Under Contract EY-76-C-06-1830

Printed in the United States of America
Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22151
Price: Printed Copy $_____*; Microfiche $3.00

*Pages	NTIS Selling Price
001-025	$4.50
026-050	$5.00
051-075	$5.50
076-100	$6.00
101-125	$6.50
126-150	$7.00
151-175	$7.75
176-200	$8.50
201-225	$8.75
226-250	$9.00
251-275	$10.00
276-300	$10.25

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A USER'S MANUAL FOR THE BNW-I OPTIMIZATION CODE FOR DRY-COOLED POWER PLANTS

VOLUME I

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January 1977

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PREFACE

This document has been issued in four volumes to facilitate handling. Volume I is a narrative description of the code algorithms, as well as logic, input and output information. Volumes II through IV are appendices of Volume I, providing listings of the BNW-I optimization code for each of three dry-cooled heat rejection systems. These are:

Volume II - Metal Finned Tube Versions of the BNW-I Computer Code
Volume III - Plastic System Versions of the BNW-I Computer Code
Volume IV - Ammonia System Versions of the BNW-I Computer Code

Complete distribution has been given only to Volume I. Copies of one or more of the other three volumes may be obtained by writing the National Technical Information Service, U.S. Department of Commerce, 5281 Port Royal Road, Springfield, VA, 22151.

FOREWORD

The Dry Cooling Tower Development Program at Battelle, Pacific Northwest Laboratory was initiated with a program scope that included the following near-term and ultimate emphases.

Near-Term Objectives:

- Develop economic and performance models for cost optimization of total heat rejection systems using dry cooling.
- Analysis of, and dissemination of experience on, existing dry-cooled plant performance.
- Demonstrate certain features of existing technology equipment to provide confidence for specification by utilities.

Ultimate Objective:

- Promote water conservation through industry use of dry cooling by developing and demonstrating the reliability of lower-cost systems. The development of advanced dry/wet systems is also considered to be within this scope.

Over the past two years the following documents have been issued, reporting the results of the work toward these objectives.

Cost optimization of dry-cooled heat rejection systems:

A REVIEW AND ASSESSMENT OF ENGINEERING ECONOMIC STUDIES OF DRY COOLED ELECTRICAL GENERATING PLANTS. B. C. Fryer, BNWL-1976, March 1976.

HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF DRY TOWER EXTENDED SURFACES. PART I: HEAT TRANSFER AND PRESSURE DROP DATA. PFR Engineering Systems, Inc., PFR 7-100, March 1976.

HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF DRY TOWER EXTENDED SURFACES. PART II: DATA ANALYSIS AND CORRELATION. PFR Engineering Systems, Inc., PFR 7-102, June 1976.

Analysis of performance of existing dry-cooled plants:

> <u>DRY COOLING TOWER PROGRAM: RESULTS OF INDUSTRIAL CONTACTS THROUGH AUGUST 1974</u>. BNWL-1878, November 1, 1974.
>
> <u>A SURVEY OF MATERIALS AND CORROSION PERFORMANCE IN DRY COOLING APPLICATIONS</u>. A. B. Johnson, Jr., D. R. Pratt and G. E. Zima, BNWL-1958, March 1976.
>
> <u>EUROPEAN DRY COOLING TOWER OPERATING EXPERIENCE</u>. J. G. DeSteese and K. Simhan, BNWL-1995, March 1976.
>
> <u>MATHEMATICAL AND EXPERIMENTAL INVESTIGATIONS ON DISPERSION AND RECIRCULATION OF PLUMES FROM DRY COOLING TOWERS AT WYODAK POWER PLANT IN WYOMING</u>. Y. Onishi and D. S. Trent, BNWL-1982, February 1976.

Advanced dry (dry/wet) cooled systems:

> <u>PRELIMINARY EVALUATION OF WET/DRY COOLING CONCEPTS FOR POWER PLANTS</u>. W. V. Loscutoff, BNWL-1969.
>
> <u>COMPATIBILITY OF AMMONIA WITH CANDIDATE DRY COOLING SYSTEM MATERIALS</u>. D. R. Pratt, BNWL-1992, April 1976.
>
> <u>SCALE FORMATION IN DELUGED DRY COOLING SYSTEMS</u>. D. R. Pratt, BNWL-2060, March 1976.
>
> <u>AMMONIA AS AN INTERMEDIATE HEAT EXCHANGE FLUID FOR DRY COOLED TOWERS</u>. R. T. Allemann, B. M. Johnson and G. C. Smith, BNWL-SA-5997, September 1976.

A group of reports (including this report) has recently been issued. This group serves the dual purpose of developing cost optimization models for dry cooling systems based on available technology and comparing the results of analyzing the costs of these systems with the projected cost of several advanced dry and dry/wet systems. Included in this group are:

> <u>AN ENGINEERING AND COST COMPARISON OF THREE DIFFERENT ALL-DRY COOLING SYSTEMS</u>. B. C. Fryer, D. W. Faletti, Dan J. Braun, David J. Braun and L. E. Wiles, BNWL-2121, September 1976.
>
> <u>A STUDY OF THE COMPARATIVE COSTS OF FIVE WET/DRY COOLING TOWER CONCEPTS</u>. F. R. Zaloudek, R. T. Allemann, D. W. Faletti, B. M. Johnson, H. L. Parry, G. C. Smith, R. D. Tokarz, and R. A. Walter, BNWL-2122, September 1976.

DRY COOLING OF POWER GENERATING STATIONS: A SUMMARY OF THE ECONOMIC EVALUATION OF SEVERAL ADVANCED CONCEPTS VIA A DESIGN OPTIMIZATION STUDY AND A CONCEPTUAL DESIGN AND COST ESTIMATE. B. M. Johnson, R. T. Allemann, D. W. Faletti, B. C. Fryer and F. R. Zaloudek, BNWL-2120, September 1976.

COSTS AND COST ALGORITHMS FOR DRYING COOLING TOWER SYSTEMS. P. A. Ard, C. H. Henager, D. R. Pratt and L. E. Wiles, BNWL-2123, September 1976.

A USER'S MANUAL FOR THE BNW-I OPTIMIZATION CODE FOR DRY-COOLED POWER PLANTS. David J. Braun, Dan J. Braun, Warren V. De Mier, D. W. Faletti and L. E. Wiles, BNWL-2180, January 1977.

ACKNOWLEDGMENTS

This report contains an account of work sponsored by the Division of Nuclear Research and Applications of ERDA under the Dry Cooling Tower Program at Battelle, Pacific Northwest Laboratory (PNL). The project was completed under the direction of Mr. Ira Helms, Project Officer; Dr. W. F. Savage, Manager of the Advanced Concepts Evaluation Branch of ERDA; and B. M. Johnson, Manager of the Dry Cooling Tower Project at PNL.

The authors gratefully acknowledge Dr. B. C. Fryer's leadership and effort throughout the code development task.

The information provided by Dott. Ing. Carlo Rocco and Vincenzio Leonelli of Italimpianti, Genoa, Italy, and by Professor Carlo Roma of the University of Rome on the natural draft system currently under development by Italimpianti was of invaluable assistance in our evaluation of the induced draft PLASTIC system.

Acknowledgments are also due to the following cooling tower manufacturers and their representatives for their interest during this study: M. Larinoff and E. C. Smith of Hudson Products, and R. Landon and R. Cates of The Marley Company.

Many manufacturers of components for cooling tower systems were contacted during this study and provided essential cost information that was incorporated into the cost algorithms and estimates. This information is the basis of the comparisons made herein. We acknowledge this vital contribution to the study, although any mistakes in interpretation and formulization of this information are the responsibility of PNL.

SUMMARY

This User's Manual provides information on the use and operation of three versions of BNW-I, a computer code developed by Battelle, Pacific Northwest Laboratory (PNL) as a part of its activities under the ERDA Dry Cooling Tower Program. These three versions of BNW-I were used as reported elsewhere to obtain comparative incremental costs of electrical power production by two advanced concepts (one using plastic heat exchangers and one using ammonia as an intermediate heat transfer fluid) and a state-of-the-art system.

The computer program offers a comprehensive method of evaluating the cost savings potential of dry-cooled heat rejection systems and components for power plants. This method goes beyond simple "figure-of-merit" optimization of the cooling tower and includes such items as the cost of replacement capacity needed on an annual basis, and the optimum split between plant scale-up and replacement capacity, as well as the purchase and operating costs of all major heat rejection components. Hence, the BNW-I code is a useful tool for determining potential cost savings of new heat transfer surfaces, new piping or other components as part of an optimized system for a dry-cooled power plant.

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A USER'S MANUAL FOR THE BNW-I OPTIMIZATION CODE FOR DRY-COOLED POWER PLANTS

1.0 INTRODUCTION

This report provides information on BNW-I, a computer code developed by Battelle, Pacific Northwest Laboratory (PNL) as part of its activities under the ERDA Dry Cooling Tower Program. BNW-I was used in other studies[(1)] to obtain comparisons of the incremental cost (referred to as a zero-cost cooling system) of three all-dry cooling systems:

- the state-of-the-art indirect system (SOA);
- an advanced concept using plastic tube heat exchangers (PLASTIC);
- an advanced concept using ammonia as an intermediate heat transfer fluid (NH_3).

These studies showed that both the PLASTIC and the NH_3 systems had potential for significant cost savings.[(1)]

This User's Manual provides information on the use and operation of three versions of BNW-I used to study the all-dry systems listed above. The primary purpose of this report is to provide the reader with the information required to use the BNW-I code, i.e., to prepare the input required and to interpret the output.

In addition, this report covers the following topics:

- physical descriptions of the three systems
- the rationale used to optimize the cost of electricity
- calculational procedures
- basis for the performance and cost models.

2.0 PHYSICAL DESCRIPTION OF THE THREE COOLING SYSTEMS

General descriptions of the SOA, PLASTIC, and NH_3 cooling systems are provided in Sections 2.1 through 2.3. The descriptions are for each system operating at steady-state at full load. Startup, shutdown, load change, and system operational changes to accommodate meteorological changes for various plant operating modes are not considered.

2.1 SOA COOLING SYSTEM

The SOA cooling system modeled by the SOA code is illustrated in Figure 2-1. It is an indirect system with induced mechanical draft and a single-pressure surface condenser.

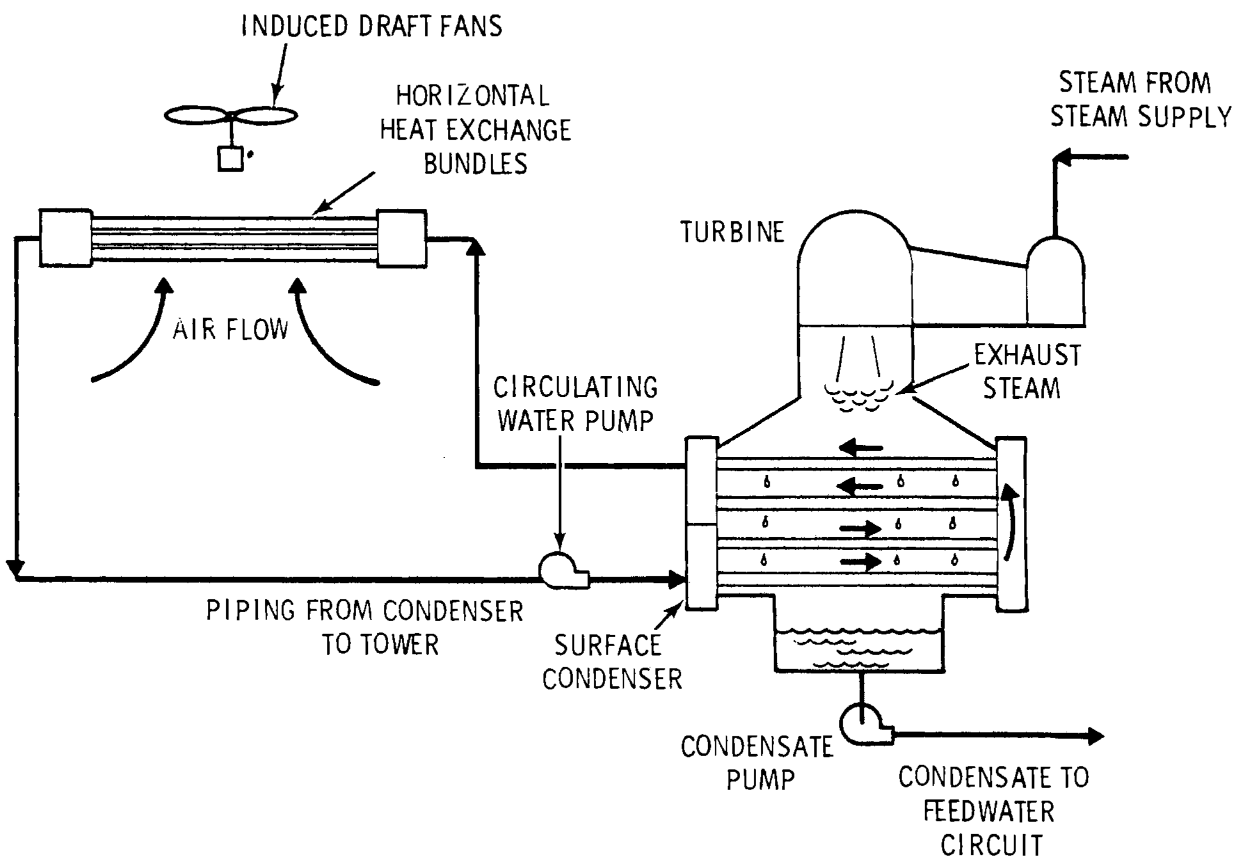

FIGURE 2-1. Indirect Dry Cooling System with Surface Condenser and Mechanical Draft Tower

The cooling process is best described with the aid of the temperature relationship diagram in Figure 2-2. Turbine exhaust steam is condensed at constant temperature on the outside of the condenser tubes. Water entering the condenser tubes is heated, then carried through the supply piping to the cooling tower.

The increase in water temperature is termed the range of water. The water temperature approaches the steam saturation temperature; the difference is termed the "terminal temperature difference," or TTD. At the cooling tower, the water cools while simultaneously heating the air as it passes over the finned tubes in the tower. This change in air temperature is termed the range of the air. The cooled water is then returned via piping to the condenser inlet. The heated air is discharged to the environment by the induced draft fans. A short description of the components of the SOA cooling system as contained in the computer code will now be presented.

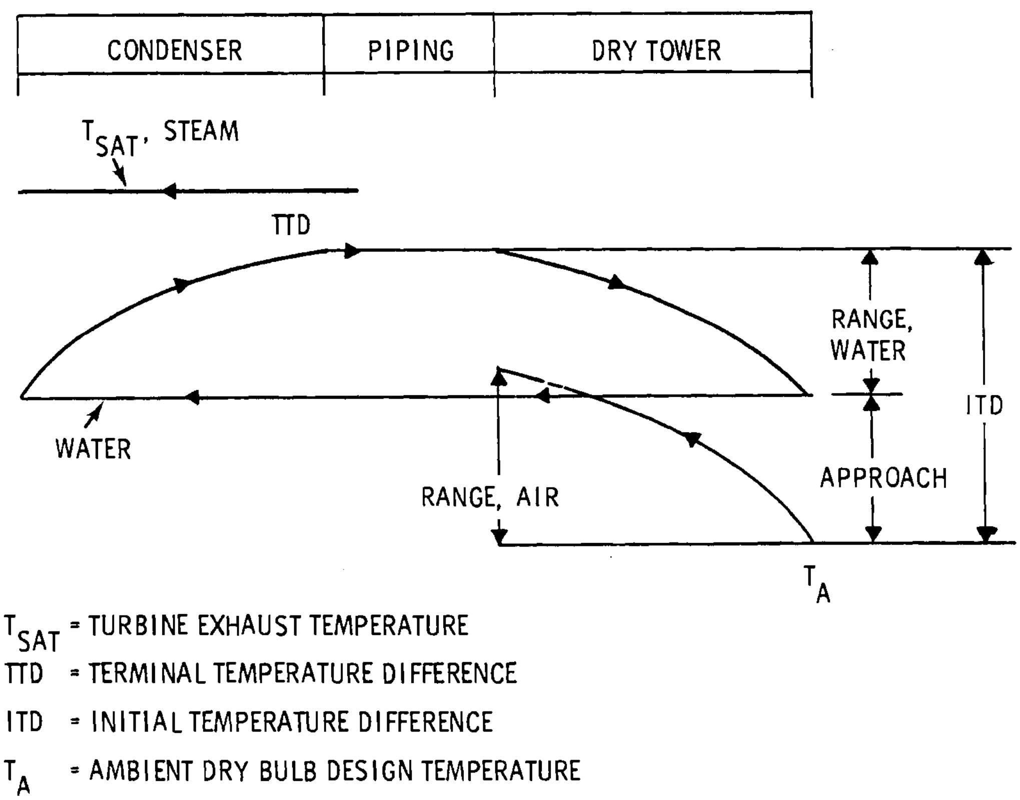

FIGURE 2-2. Temperature Relationships for SOA and Plastic Heat Exchanger Cooling Systems

2.1.1 Condenser

The SOA code considers a single-pressure surface condenser only. Investigation of multipressure surface condensers would require modification of the computer program.

2.1.2 Piping

The piping consists of main supply and return lines, tower distribution lines, headers, and pump station piping. The cost data used in the codes presented in this report are for carbon steel piping and fittings. The main supply and return piping and the tower distribution lines are considered to be below ground. The headers are considered to be above ground. The vertical inline pumps draw water from an open reservoir. A complete description of the piping layouts and algorithms is presented in Reference 2; Appendix B gives an abbreviated description.

2.1.3 Cooling Tower

The cooling towers are circular with vertical tube heat exchangers and induced draft fans. If desired, louvers can be provided for air flow control and hail protection. Similarly, fan diffuser stacks can be provided. The fan system consists of the fans, motors, gear boxes and electrical wiring. The fan blades are fiberglass. The gear boxes are single-speed bevel gear type. The fans have manually adjustable pitch blades.

Versions of the BNW-I code applicable to rectangular cooling towers with horizontal tubes were also developed and used.[1]

All three versions of the code can be used to model the tower configurations where the heat exchanger bundles are arranged in deltas (typically 60°) on the periphery of the towers.

2.2 PLASTIC HEAT EXCHANGER COOLING SYSTEM

The plastic heat exchanger cooling system is basically the same as the SOA system with the metal finned tubes replaced with plastic tubes. Details of plastic heat exchanger fabrication may be found in References 2 and 3.

Bare tube heat transfer models are used for the heat exchanger bundles of the PLASTIC system. Correlations for calculating the air-side heat transfer coefficients for both staggered and inline bare tube arrangements are included in Appendix A.

2.3 AMMONIA DRY COOLING SYSTEM

The ammonia dry cooling system, illustrated in Figure 2-3, differs from SOA systems in two ways. First, ammonia replaces water as the coolant. Second, a condenser/reboiler is used in place of the standard condenser.

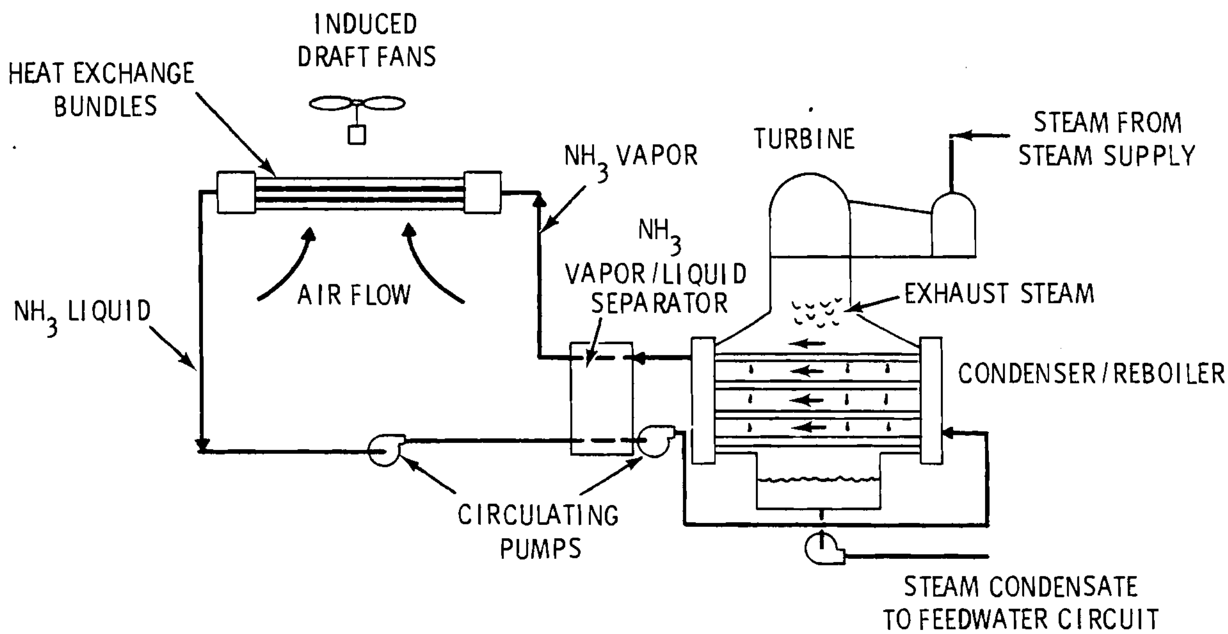

FIGURE 2-3. Ammonia Coolant Dry Cooling System

The cooling process can be explained by referring to Figures 2-3 and 2-4. Turbine exhaust steam is condensed at constant temperature on the outside of the condenser/reboiler tubes. Slightly subcooled liquid ammonia entering the condenser/reboiler tubes is first heated to saturation temperature, then partially boiled at constant temperature as it traverses the length of the tubes. A mixture of liquid and vapor ammonia (typically 50 to 80 percent quality, depending on the design) enters the separator, where the vapor is routed to the tower supply piping and the liquid is collected for recirculation to the condenser/reboiler. The vapor flows through the supply piping at near-saturation conditions. However, there is a pressure drop in the piping system from frictional effects. Because of the physical characteristics of ammonia vapor, this pressure drop is accompanied by a drop in temperature (TTD2). At the tower, the vapor is condensed inside the metal finned tubes at constant temperature. The condensed ammonia is then collected and returned to the condenser/reboiler. The basic terminology remains the same as for the SOA system except that the parameter, $RANGE_{Water}$, is eliminated and replaced by the vapor supply piping temperature drop, TTD2. Because of the economic trade-offs, the $RANGE_{Air}$, is larger for the ammonia system than for the SOA or PLASTIC systems.

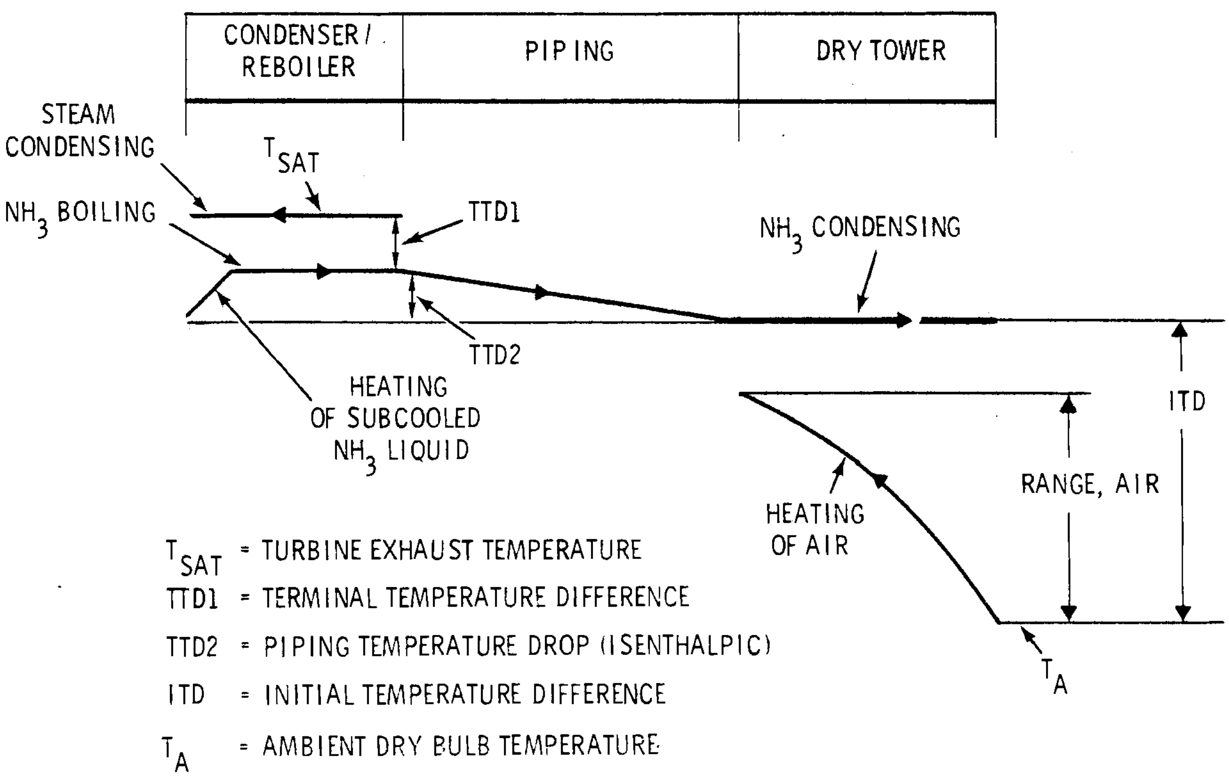

FIGURE 2-4. Temperature Relationships for Ammonia Coolant Dry Cooling System

2.3.1 Condenser/Reboiler

The condenser/reboiler is similar to the SOA condenser in general appearance. However, the water boxes are replaced with high pressure headers and the tube sheets are much thicker. Tubes of two types can be modeled: conventional smooth tubes of aluminum or stainless steel, or enhanced tubes which improve steam-side and ammonia-side heat transfer processes. With the enhanced tubes, the size of the condenser/reboiler is about the same as for a conventional condenser. With conventional tubes, the condenser/reboiler is significantly larger. Only horizontal tube arrangements with ammonia on the tube side are modeled. Other arrangements, including vertical tubes and/or ammonia on the shell side, are optional possibilities that could be evaluated by inserting appropriate cost and performance algorithms into the code.

2.3.2 Piping

The piping in the ammonia system is significantly different from that in the water systems. First, the diameter is typically smaller, particularly in the return line. Second, for a given diameter, the ammonia piping must be considerably thicker (neglecting corrosion allowances) to accommodate the higher pressure.

2.3.3 Cooling Tower

The ammonia cooling system considered consists of vertical tube, one-pass heat excangers. Ammonia vapor enters the top of the heat exchangers, and condenses as it flows down the inside of the finned tubes. This is in contrast to the two-pass SOA and PLASTIC systems in which water enters and exits the heat exchangers at the bottom of the vertical tube heat exchanger.

3.0 PLANT DESIGN OVERVIEW AND BASIS

The plant is designed to minimize the power production costs of an idealized utility system using a dry-cooled plant. The "cost" of dry cooling is defined as an incremental increase in power production costs compared to the cost of producing power with a plant having a zero-cost cooling system and a conventional turbine that operates at 3.5 in. Hg.

3.1 UTILITY MODEL

The idealized utility consists of one dry-cooled, base loaded power plant with a gas turbine complement[a] to provide power at higher ambient temperatures. (The reference plant requires no gas turbines complement because its turbine back pressure is constant at 3.5 in. Hg.) This idealized utility operates in an on-off mode. It supplies full power at a constant level when in the "on" mode and no power during the "off" mode. The fraction of time that the idealized utility is in the "on" mode is equal to the capacity factor. No real-life utility operates in this manner. However, the idealized utility used here can be viewed as a portion of a real utility, because the actual utility has a number of plants that supply the required power when one or more of the base loaded plants are down for either scheduled or unscheduled maintenance.

The base loaded dry-cooled plant is capable of supplying the entire power requirement of the utility plus the power requirements of the fans and pumps of its cooling system up to its "design temperature". Above the design temperature, the power output of the dry-cooled plant falls off because of rising turbine back pressure; the power deficit is made up by gas turbines. The size of the gas turbine contingent is determined by the power deficit of the dry-cooled plant (which is always operated "valve wide open") at the highest ambient temperature.

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[a] Other sources of replacement energy such as hydro or cycling plants can be simulated by appropriate changes in the capital and fuel costs used as inputs to the program.

The dry-cooled plant can produce more than the utility's power requirements below the design temperature. Because no market exists for this power, no credit is taken for this excess capacity. However, credit is given for reduced base plant fuel consumption resulting from the lower turbine back pressures during periods of lower-than-design ambient temperature.

3.2 STEAM SUPPLY AND PLANT SCALING

Both steam supply scaling and plant scaling are used in designing the plant. The steam supply system is scaled separately to meet the additional steam requirements of the dry-cooled plant's steam turbine at rated back pressure. This additional steam requirement arises from the difference in heat rate between

a) the reference base plant conventional turbine, and
b) the turbine selected for use in the power plant being designed (conventional, modified conventional, and high back pressure).

The difference in heat rate factors is shown graphically in Figure 3-1, where the difference being discussed is between Point 0 and Point 1 or 1'. Point 0 gives the heat rate factor and rated back pressure of the conventional turbine used in the reference plant. Points 1 and 1' are the heat rate factors at the rated back pressure for the modified conventional and high back pressure turbines, respectively. Steam supply scaling refers to increasing the size of the steam supply to compensate for the increase in heat rate between Points 0 and 1, or between Points 0 and 1'.

Plant scaling refers to increasing the size of the entire plant, including the steam supply system and the heat rejection system, so that the net power output of the plant (equal to the gross power output minus the requirements for cooling system fans and pumps) equals the utility's requirements at the design temperature.

Plant scaling is accomplished in two steps. The first step involves scaling only the plant size (including the steam supply system) up or down to account for the difference in turbine heat rates at plant design conditions (i.e., at the design temperature) and at rated back pressure. This can result in a decrease or increase in plant size depending on whether the design turbine back pressure is greater or less than the rated back pressure of the steam turbine. This process has nothing to do with the reference base plant conventional turbine, only the turbine used in the design plant. In Figure 3-1, the heat rate factors are shown as Points 1 and 2 (for a modified conventional turbine) or Points 1' and 2' (for the high back pressure turbine).

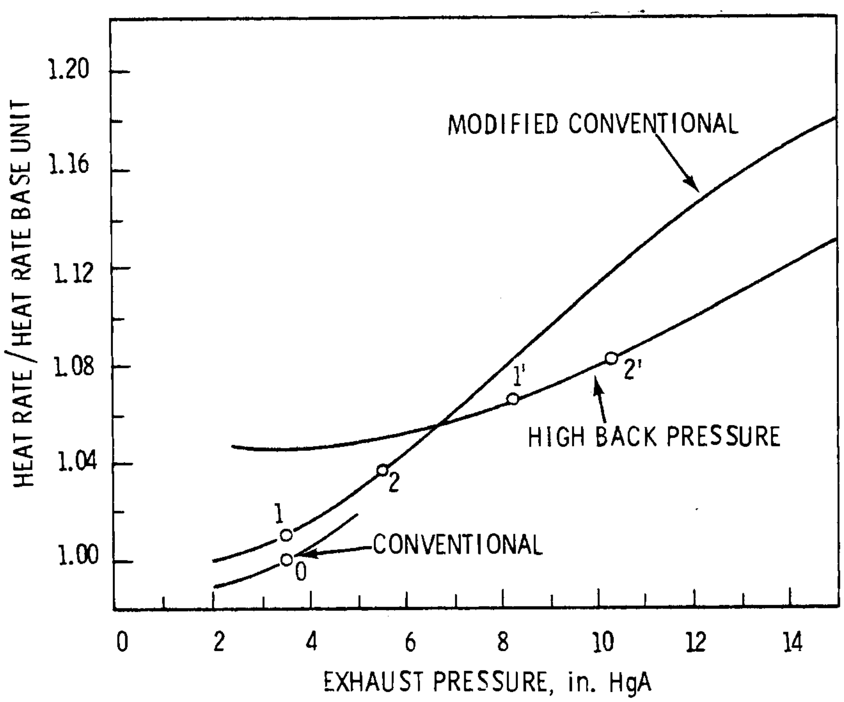

FIGURE 3-1. Heat Rate Versus Back Pressure--Fossil Turbines

The second step involves scaling up the plant and the dry cooling tower system to provide power for the cooling system fans and pumps. The relationships between the reference plant power output and the gross and net power output of the dry-cooled plant are shown on Figure 3-2. Point 2 on Figure 3-2 corresponds to Point 2 of Figure 3-1. When the plant is scaled up in size, none of the operating parameters is changed. Therefore, the power plants represented by Points 2 and 3 in Figure 3-2 have the same turbine design back pressure. The only difference between the two plants is that the scaled-up plant produces not only the power required by the utility system, but power for the fans and pumps of the dry cooling tower. Because the heat rejection load of the dry cooling towers has increased, the physical dimensions of the towers are scaled up to handle this increase in heat rejection.

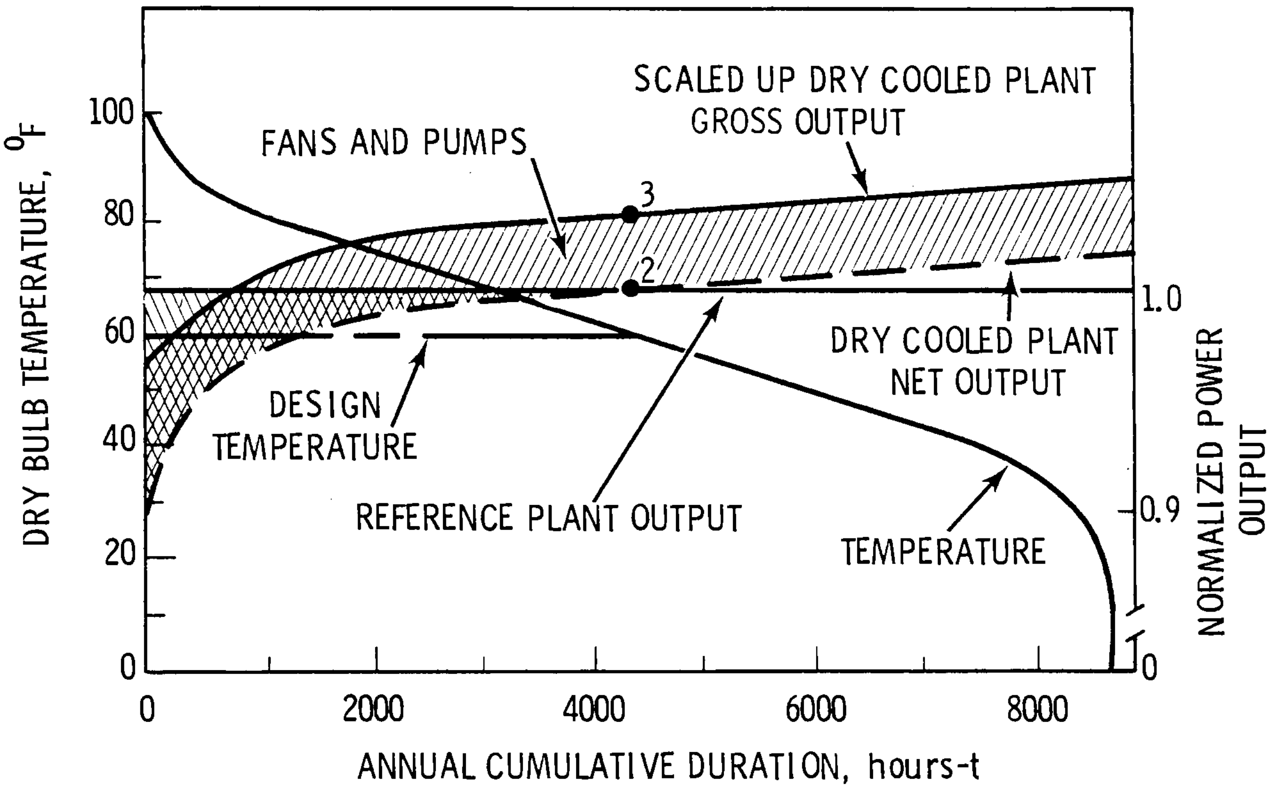

FIGURE 3-2. Typical Annual Performance of a Dry Cooled Plant

In both steps, the power plant is scaled up or down linearly. The dry cooling tower system in the second step is also scaled linearly. Therefore, second order effects were not taken into account by this particular scaling process; for example, the increased plant size required to produce power for the slightly larger cooling system is neglected.

A trade-off exists between plant scaling and the size of the gas turbine contingent. The optimization process takes this trade-off into account.

3.3 COSTING METHOD

The incremental cost of dry cooling includes the following capital items:

a) cooling system equipment and installation costs (including indirect costs such as profit, overhead and engineering)
b) increased steam supply system cost
c) change in plant cost due to difference between the turbine rated back pressure and the turbine design back pressure
d) increased base plant costs for supplying power to auxiliaries
e) gas turbine capital costs.

The incremental costs of dry cooling include the following fuel costs:

a) increased base plant fuel required for producing power for the pumps and fans
b) difference in the amount of base plant fuel required for the dry-cooled plant versus that of the reference plant operating at 3.5 in. Hg back pressure
c) gas turbine fuel.

Operation and maintenance costs are calculated as a percentage of the cooling system capital cost. No operation and maintenance costs are applied to the gas turbines.

Items and related costs not included within these codes are:

1) cover gas system
2) cooling system instrumentation costs
3) cooling loop water treatment costs.

The incremental cost is expressed in mills/kWh by

$$C = \frac{FCR\ (\text{capital costs}) + \text{Fuel Costs} + OM}{CF \times 24 \times 365 \times 1000 \times P}$$

where

FCR is the fixed charge rate

OM is the operations and maintenance cost

CF is the capacity factor (fraction of the year that the plant operates)

P is the power output, kilowatts

and the capital, fuel, and operation and maintenance costs (OM) are as given above.

All costs are in January 1976 dollars. Interest during construction and inflation prior to or during construction is not considered. Effects of the dry cooling system's construction schedules and their impact on the balance of plant construction schedule are also not considered. The uncertainty involved in defining these items appears to outweigh any additional understanding obtained by including them in a comparative analysis.

The heat transfer and the pressure drop correlations used by BNW-I are presented in Appendix A. Cost algorithms were developed for the components of the dry-cooled plant cooling system. These algorithms are described in detail in Reference 2. A brief description of the piping algorithms is presented in Appendix B.

3.4 OPTIMIZATION

Optimization of a dry cooling system involves:

a) selecting values for the set of independent parameters which fix the design, after which all other dependent parameters (including incremental power production costs) can then be calculated; and
b) repeating this process until the lowest incremental power production cost is obtained.

The computer optimization involves parameters of four types:

1. internally optimized independent parameters;
2. internally optimized dependent parameters;
3. externally optimized independent parameters; and,
4. external fixed parameters.

A list of the major parameters and options in each of the above groups is given in Table 3-1.

The external fixed parameters are those which are imposed on the analysis. They are generally fixed at a given site for a particular plant. However, the investigator may vary these "fixed" parameters when comparing the cost of dry cooling for nuclear plants with coal plants, or evaluating cost differences at different sites. The effect of changes in fuel costs, capacity factors, etc., can also be investigated by varying these parameters.

TABLE 3-1. Computer Analysis - Major Cooling System Parameters and Options

	INTERNALLY OPTIMIZED INDEPENDENT	INTERNALLY OPTIMIZED DEPENDENT	EXTERNALLY OPTIMIZED INDEPENDENT	EXTERNAL FIXED
I HEAT EXCHANGER	1. FRONTAL AREA 2. WIDTH/LENGTH RATIO	1. TUBE LENGTH* 2. NUMBER TUBES IN DEPTH* 3. TOTAL TUBES, (BUNDLE NUMBER), OR OVERALL SIZE	1. UNIT GEOMETRY (TUBE OD, TUBE PITCH, FIN PITCH, FIN HEIGHT, STAGGERED VS IN-LINE, ETC) 2. TUBE AND FIN MATERIAL 3. FIN TYPE 4. SPACER MATERIAL AND TYPE 5. HEADER TYPE	
II FAN SYSTEM		1. NUMBER 2. BLADE DIAMETER* 3. BLADE ANGLE* 4. NUMBER OF BLADES* 5. FAN POWER	1. VELOCITY RECOVERY VS NO VELOCITY RECOVERY	
III TOWER		1. NUMBER* (CIRCULAR) 2. DIAMETER (CIRCULAR)		
IV PIPING AND PUMP		1. PUMP POWER 2. PIPE LENGTHS 3. PIPE DIAMETER	1. DESIGN VELOCITY 2. PIPE WALL THICKNESS 3. PIPING TEMPERATURE DROP	
V CONDENSER (SOA AND PLASTIC)		1. TUBE LENGTH 2. TUBE NUMBER	1. TERMINAL TEMPERATURE DIFFERENCE 2. TUBE MATERIAL 3. TUBE DIAMETER 4. DESIGN VELOCITY	
V' CONDENSER/REBOILER (NH_3)	1. TERMINAL TEMPERATURE DIFFERENCE (NH_3)	1. TUBE NUMBER	1. TUBE LENGTH 2. TUBE MATERIAL 3. TUBE DIAMETER 4. TUBE TYPE 5. EXIT QUALITY	
VI TURBINE	1. DESIGN TURBINE EXHAUST TEMPERATURE*		1. TURBINE TYPE	
VII COMMON TO TWO OR MORE COMPONENTS	1. CAPACITY RATIO (WATER/AIR) 2. RANGE OF WATER* 3. RANGE OF AIR (NH_3)	1. AIR MASS FLOW RATE (RANGE OF AIR) 2. WATER MASS FLOW RATE 3. TOWER INITIAL TEMPERATURE DIFFERENCE (ITD)*	1. RATIO OF TOWER ROOF AREA TO FAN SWEPT AREA 2. DESIGN AMBIENT TEMPERATURE 3. DELTA VRS NON-DELTA HEAT EXCHANGER 4. CIRCULAR VRS RECTILINEAR TOWER ARRANGE	
VIII BASE PLANT				1. FUEL COST 2. PLANT COST 3. HEAT RATE 4. CAPACITY FACTOR 5. FIXED CHARGE RATE 6. SIZE
IX CAPACITY LEVELING				1. ENERGY PENALTY CHARGES 2. CAPACITY PENALTY CHARGES
X SITE				1. METEOROLOGY

*OPTIONAL TREATMENT AS EXTERNALLY FIXED PARAMETER

The externally optimized independent parameters or options are those optimized by changing one or more parameters or options at a time and observing the resultant changes in costs and cooling system design. This includes simple factors such as the effect of fin spacing, as well as more complex elements such as delta heat exchanger arrangements.

For a given set of external parameters, there are five internally optimized independent parameters. Once these five independent parameters are specified, all other system design and operating dependent parameters can be calculated. Several different combinations of five parameters could have been used in lieu of those in Table 3-1. However, combinations shown in Table 3-1 are those used in the MIT study[4] that served as one reference for the work described herein.

This particular set of parameters makes solving for the dependent parameters rather straightforward with a minimum of time-consuming iterative implicit calculations. However, this approach results in heat exchanger designs with a non-integer number of tube rows in the direction of air flow. Usually, this can be circumvented by applying constraints to the optimization analysis. However, difficulties can still result in attempting to arrive at optimized cooling systems which have an integer number of tube rows. For continuous matrix heat exchangers, the non-integer tube row problem does not exist. However, SOA, PLASTIC, and NH_3 systems investigated employ integer tube row heat exchangers. This is particularly important in the case of SOA systems utilizing two-pass heat exchangers where exchangers of interest are 2, 4, or 6 rows deep, resulting in relatively large discrete changes in the number of tube rows. In single-pass NH_3 systems, the problem is less severe. In PLASTIC systems, the problem is insignificant because exchangers with as many as 30 tube rows are typical.

The BNW-I computer program contains an optimization technique to determine which combination of internal dependent parameters results in the minimum incremental power production costs associated with dry cooling. This optimization technique, developed by Andeen and Glicksman of MIT,[4] is described in Appendix C.

The input to the program is illustrated in Figure 3-3. General output information is also indicated.

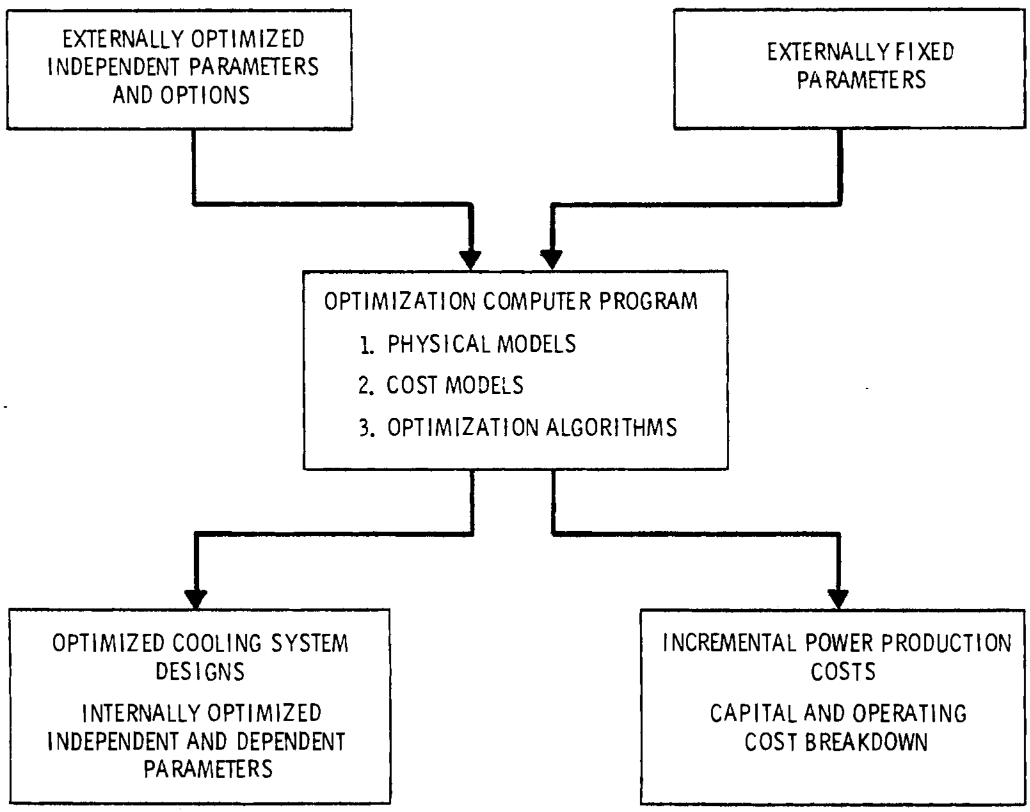

FIGURE 3-3. Dry Cooling System Computer Optimization

4.0 CALCULATIONAL PROCEDURE

The optimization of a dry cooling tower heat rejection system is accomplished by varying five independent variables to obtain the minimum incremental cost. The independent variables are varied according to the optimization routine developed by MIT[4] and discussed in Section 3 and Appendix C. Once these variables have been determined, the design and costing of the dry cooling tower system is carried from there.

The dry cooling tower code consists of two distinct parts:

- optimization
- design and costing of dry cooling towers.

The two parts of the code interact as shown in Figure 4-1. The optimization part of the code selects the values of the five independent parameters:

1. total frontal area of the heat exchanger - AFRON
2. width/length ratio of the heat exchanger - WLRAT
3. temperature range of water flowing through the heat exchanger - RANGE
4. steam exit temperature of the turbine - T1
5. heat capacity ratio of water to air flowing through the heat exchanger - CWARA

These variables are supplied to the design and costing portion of the code.

4.1 METAL FINNED TUBE (SOA) SYSTEM

The following is a discussion of the calculation procedure used in design and costing of the metal finned tube dry cooling towers. The discussion is subdivided into major categories describing specific calculations dealing with a particular aspect of the design or costing of the dry cooling towers. The subroutine in which a major portion of the computation is carried out is given in parentheses.

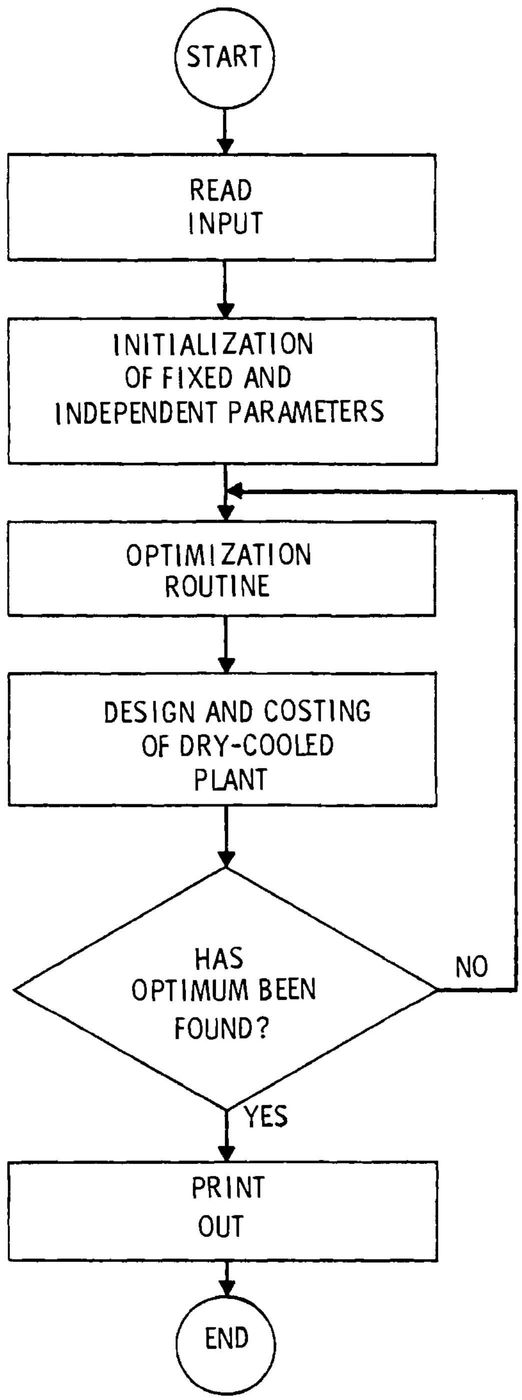

FIGURE 4-1. BNW-I Program Flow Diagram

SOA Dry Cooling Tower System Design (CALC)

Given a value for each of the five independent variables, the heat exchanger dimensions are determined by the stepwise calculational procedure shown below.

1. Given the exit temperature of the steam turbine, the turbine back pressure is calculated.
2. The heat rate factor and turbine efficiency is determined at the above back pressure.
3. Knowing the turbine efficiency and plant size, the heat input from the steam supply system and the heat rejected by the dry cooling towers is calculated.
4. The mass flow rate of air and water through the heat exchanger is determined.
5. Heat exchanger effectiveness and number of transfer units (NTU) are determined.
6. Heat exchanger dimensions per foot are calculated.
7. Air-side Reynolds number at the minimum cross-sectional area of the heat exchanger is determined.
8. Air-side friction factor and heat transfer coefficient are found (if Briggs, Robinson and Young's correlations[5,6] are chosen for use by setting input parameter FFHX equal to FRICHO).
9. Fin efficiency is determined.
10. Heat exchanger width is calculated by one of the following methods, depending on heat exchanger tube length options:
    a. Length fixed - Width a function of heat exchanger tube length and frontal area (AFRON);

b. Length allowed to vary - Width a function of the independent parameters of heat exchanger width length ratio (WLRAT) and heat exchanger frontal area (AFRON).

11. Number of tubes in width of the heat exchanger is determined.

START OF LOOP

Calculate Number of Tubes in Depth

12. Reynolds number of water flowing through the tubes is determined.
13. Water-side heat transfer coefficient is calculated.
14. Overall surface efficiency (fin effectiveness consideration) is found.
15. Overall heat transfer coefficient based on the air-side surface area is determined.
16. Number of heat exchanger tubes in depth is calculated.
17. Check if number of tubes in depth have converged.

END OF LOOP

18. Limit the number of tubes in depth to a specified range of values from input to the code (optional).
19. Velocity of the water through the tubes is found.
20. Length of heat exchanger tubes is determined.
21. Water-side friction factor is calculated.
22. Pressure drop through the heat exchanger tubes is determined.
23. Air-side hydraulic diameter is calculated.
24. Air-side friction factor using PFR correlations[1,8] (if PFR correlations are chosen for use by setting input parameter FFHX equal to PFR) is found.
25. Pressure drop through the heat exchanger is calculated.
26. Heat exchanger costing (Section 4.1.2) is performed.
27. Fan system design and costing (Section 4.1.3) is performed.

29. Piping and pumping system design and costing (Section 4.1.4) is performed.

30. The plant is scaled up to account for off-design conditions, fan and pump power (Section 4.1.5).

31. The following yearly performance parameters of the power plant and dry cooling tower (Section 4.1.6) are calculated:

    a. turbine outlet temperature

    b. turbine back pressure

    c. turbine heat rate factor

    d. temperature range of water passing through the heat exchanger

    e. terminal temperature difference of the condenser

    f. log mean temperature difference of the heat exchanger

    g. initial temperature difference of the heat exchanger

    h. power output from the plant to the electrical transmission network

    i. cost to produce power over the entire range of ambient air temperatures

    j. total yearly incremental power production cost.

32. Return to optimization procedure subroutine (SHOT, SERCH, or XTEND) which called subroutine CALC.

4.1.1 Condenser Design and Costing

The power plant condenser is a single-pressure surface condenser, designed according to Heat Exchanger Institute standards. The design is based on five parameters:

- temperature of steam exiting the turbine at plant design conditions;
- temperature range of cooling water passing through the condenser;
- temperature difference betweem water leaving the condenser and steam exiting the turbine;
- tube size; and
- nominal water velocity through the tubes.

The costing of the surface condenser is based on Westinghouse cost data for shells and tubing.[2]

4.1.1.1 Condenser Design (SPDES)

The surface condenser design is accomplished by calculating the following values in the sequence shown.

1. Initial temperature difference between steam and cooling water entering the condenser
2. Inlet, exit and average temperature of water in the condenser
3. Log mean temperature difference of the condenser

START OF LOOP
Size the Condenser

4. Guess the water velocity through the tubes
5. Overall heat transfer coefficient based on velocity of water through the tubes
6. Surface area per shell
7. Length of flow path per shell
8. Number of passes per shell
9. Tube length to nearest foot
10. Velocity of water through the tubes
11. Check if velocity is close to the previous value

END OF LOOP

12. Number of tubes per shell
13. Total condenser heat transfer area
14. Pressure drop across the condenser.

4.1.1.2 Condenser Costing (SURCON)

The surface condenser is costed by the following calculations performed in sequence:

1. Selection of tubing cost arrays to be used for costing (choice depends upon tubing materials specified)
2. Total length of tubing required
3. Cost discount rate on total amount of tubing based on quantity purchased
4. Tubing length penalty for the length of tube chosen
5. Total cost of the tubing material
6. Shell cost of the condenser
7. Field erection costs
8. Total cost of the condenser.

4.1.2 Heat Exchanger Design and Costing (COSTEX)

The heat exchanger bundle costing is based on the algorithms and cost models given in Reference 2. The cost of the heat exchanger bundle is broken down into several subcomponents:

- header
- tubing
- tubing sealant
- spacers
- bundle frame
- bundle assembly.

The total weight of the heat exchanger bundles consists of the weight of the heat exchanger materials and the water contained inside the tubing and the headers. The cost and weight of the heat exchanger bundles are calculated in the following order:

1. liner tubing costs
2. weight of fins

3. cost of bonding the fins to the tubing
4. protective coating costs
5. cost of the tubing and the fins
6. tubing spacer costs
7. end preparation costs
8. total cost of the cooling surface
9. header and nozzle costs, based on values calculated for
    a. number of bundles in entire cooling tower system
    b. number of tubes per bundle
    c. header depth and width
    d. header costs.
10. bundle assembly and frame costs
11. weight of heat exchanger bundles, consisting of
    a. material weight of heat exchanger bundle
    b. weight of water in heat exchanger bundle
12. total cost of the heat exchanger.

4.1.3 Fan System Selection (FAN)

The selected fan system is that which has the minimum annual capital and energy costs. The type of fan selected is specified in terms of:

- blade diameter
- hub diameter
- number of blades per fan
- blade angle.

Each fan system consists of a number of fans having the same diameter and number of blades. They all operate at the same blade angle. The cost for each fan system investigated is determined.

The number and diameter of circular towers are determined from the total fan swept area, the fan packing factor and the heat exchanger width. Knowing the total cost of the fan system and the circular tower structures, the annual cost of these two components can be determined. On this basis, the optimum fan system is determined.

The fan selection is performed as follows:

1. Determine the volumetric flow rate of the fan system using the mass flow rate of air through the heat exchanger, inlet temperature, and plant site elevation.
2. Choose fan to be analyzed (fan diameter, number of blades, blade angle).
3. Determine number of fans, volumetric flow rate through each fan and total pressure drop across each fan.
4. Determine horsepower required by each fan.
5. Determine number and diameter of towers.
6. Determine fan motor and electrical costs.
7. Calculate capital, stack, and shipping and assembly costs to determine fan system total capital cost.
8. Calculate fan system annual cost (the sum of the annual capital and energy costs).
9. Calculate annual costs for the foundation and structure needed for fans.
10. Total annual cost of the fan and fan system structure and foundation is calculated, then stored or dropped, depending on its value relative to the lowest cost of the previous fans. If the value is lower, the parameters of the fan system are saved.
11. Return to Step 2 and repeat the above process until all desired fan diameters have been investigated.

4.1.4 Piping Design and Cost Algorithm Outline

The piping design and cost algorithm performs two basic functions. First, it calculates the cost of all piping and pumps between the condenser and the heat exchanger bundles. Second, it determines the pressure drop in the piping system so that the total system pumping power requirements can be determined. Figure 4-2 represents a typical piping system layout. For additional details, see Appendix B. The algorithm is outlined on the following pages.

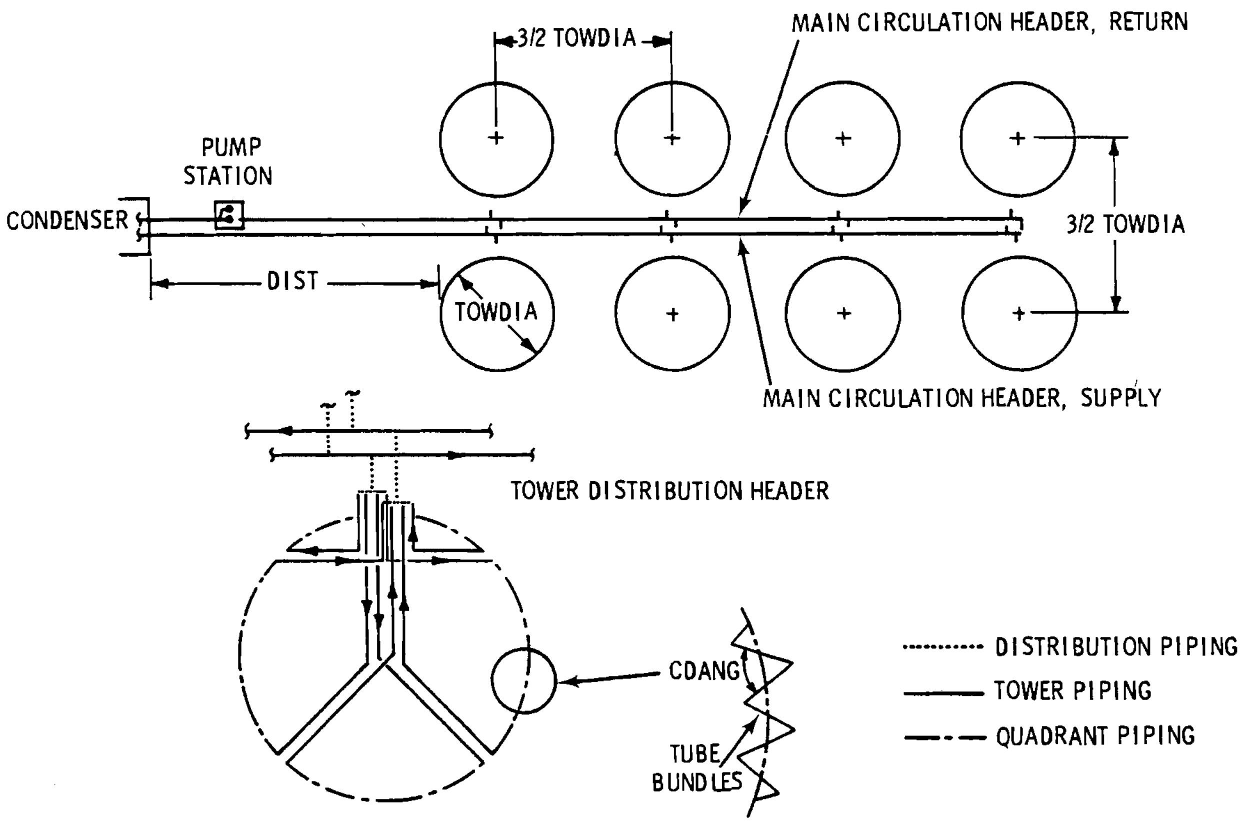

FIGURE 4-2. Typical Piping System Configuration

1. Input:

    a. total flow of coolant
    b. design velocity
    c. number and diameter of circular towers
    d. distance from condenser to towers
    e. width of tube bundle
    f. water-side pressure drop in heat exchanger and condenser.

2. Divide the total flow by the number of towers (bundles) to obtain the flow to each tower.

3. I = 1 (I is an index that applies to successive sections of the main supply header).

4. Determine the required diameter of header section 1, based on the coolant flow rate and the coolant design velocity.

5. It is assumed that pipe is available in size increments of six inches. Select DIAMETER (I) from the next largest incremental size.

6. If I = 1, go to 9.

7. It has been assumed that reducer prices apply whether the reduction is 6, 12, or 18 inches. This is based on a standard reducer length of two feet and an included angle of 22 1/2°. If DIAMETER (I) is not less than DIAMETER (I-1) by the size of the reduction (an input value which can be either 6, 12, or 18 inches) then set DIAMETER (I) equal to DIAMETER (I-1). In any other case, DIAMETER (I) = DIAMETER (I-1)-the reduction.

8. If a reduction has been made, then determine a reducer cost.

9. Determine the cost of pipe and a tee for header section I.

10. Determine the equivalent length of pipe and the true velocity of flow in header section I.

11. Determine the pressure drop in header section I.

12. Subtract out the flow to one tee branch. Determine the flow remaining in header Section I + 1.

13. If the flow remaining equals zero, go to 15.

14. I = I + 1, go to 4.

15. Determine miscellaneous costs of such items as valves, expansion joints, bundle fittings, etc.

16. If the quadrant header has been designed, then go to 19.

17. The quadrant header is designed similarly to the main circulation header. The total number of tube bundles and the flow to each quadrant and to each tube bundle can be determined from the input conditions.

18. Reset the index I = 1 and go to 4.

19. Size the tower distribution piping according to the design velocity and the flow to each tower and to each quadrant.

20. The total length of pipe for the four distribution lines between the tower distribution header and each quadrant is assumed to be 3.25 times the tower diameter. Determine the cost of this pipe and all other tower distribution pipe, fittings and valves.

21. Determine the equivalent length and the pressure drop for the tower distribution piping.

22. Determine the equivalent length and the pressure drop for the pump station.

23. Sum the pressure drops for the entire cooling system.

24. Pump station costs are determined from the total flow and the total system dynamic head loss.

25. Sum all costs for the supply and return piping.

4.1.5 Plant Scaling Due to Dry Cooling

4.1.5.1 Steam System Scaling (SSCALE)

The steam supply system of the power plant is scaled up so that the specified power output of the plant will be produced at the rated back pressure of the steam turbine. This accounts for the increased heat rate of the steam turbine being used over that of the reference plants conventional turbine rated at 3.5 in. Hg back pressure. This computation is carried out once for each set of input conditions.

The incremental cost due to scaling up the steam supply system is calculated in the following order:

1. Heat rate factor of the dry cooled plant turbine at rated back pressure
2. Plant cost as a result of steam supply system scale-up
3. Incremental costs of steam supply system due to a change in size of the steam supply system to meet increased steam requirements of the turbine.

4.1.5.2 Plant Scaling (SCALP)

The plant and dry cooling towers are scaled up or down linearly to account for two factors:

- the turbine back pressure at design conditions does not correspond to the rated back pressure of the dry cooled plant turbine, and
- part of the total power output of the dry cooled plant is used to supply the power requirements of the fan and pumping systems.

The following points on the heat rate factor curve are used in describing the scaling of the plant and dry cooling towers (See Figures 3-1 and 3-2). At each of these points, the heat rate factor, plant size and plant costs are determined.

Point 0 - Conventional turbine operating at rated back pressure of 3.5 in. Hg

Point 1 - Steam turbine operating at rated back pressure

Point 2 - Steam turbine operating at system design turbine back pressure with power output of turbine generator equal to the design power output

Point 3 - Steam turbine operating at system design turbine back pressure with the power output of plant equal to the design power output

The difference between Points 2 and 3 is that the plant has been scaled up to accommodate the power requirements of the cooling system fans and pumps. Between Points 1 and 2, the power plant has been scaled up or down to account for an increase or decrease in the heat rate factor.

The curves shown were taken from Reference 1. One is provided to the code as a third-order curve fit depending on which turbine has been selected for use.

Two scaling factors are calculated for use in scaling the plant and dry cooling towers. The purpose of each scaling factor is given separately below:

1. HRFA21 - Scale plant to account for difference in heat rate factor of the turbine at rated back pressure and design back pressure.

2. YFPP1 - Scale plant and dry cooling system to provide power for the dry cooling system fans and pumps at the turbine design back pressure.

The scaling factors are determined in the following sequence of calculations:

1. Scaling factor from Point 1 to Point 2 to account for change in heat rate factor
2. Power output of turbine at rated back pressure (Point 1) giving required power output at the design back pressure (Point 2).

3. Plant cost at design conditions, including the steam supply and plant back pressure scaling to Point 2.
4. Incremental cost of scaling up plant due to increased heat rate factor.
5. Scaling factor for the plant and dry cooling tower to meet pump and fan power requirements.
6. Gross power output of the fully scaled plant when operated at the rated back pressure (Point 1 of Figure 3-1) which meets the power requirements of the fans and pumps and the power output of the plant at the design back pressure (Point 2).
7. Total power output at Point 3.
8. Plant cost at Point 3, excluding the dry cooling tower cost.
9. Incremental cost of plant due to dry cooling if the plant operates at design conditions only.

Having calculated the scaling factor to go from Point 2 to Point 3 (YFPP1), the plant and dry cooling towers are scaled up to meet the increased heat rejection load. This is accomplished in the following sequence of calculations.

1. The following independent parameters are scaled up linearly:
   a. mass flow rate of water
   b. mass flow rate of air
   c. heat exchanger frontal area
   d. width of the heat exchanger.
2. Design and cost the condenser for increased water flow rate and heat load (Sections 4.1.1 and 4.3.1).
3. Scale the heat exchanger width and number of tubes in width.
4. Scale the fan and pumping system.
5. Scale the heat exchanger weight.

6. Calculate the incremental cost of fuel for the steam supply system due to dry cooling.
7. Scale the land and piping costs.
8. Scale the cost of the tower structure, foundation and plenum.
9. Calculate the hail and louver screen costs.
10. Sum the total capital cost of the dry cooling towers.
11. Calculate the total incremental cost of dry cooling, which is based upon the dry cooled plant operating at design conditions (unused in optimization procedure):
    a. yearly capital cost of the dry cooling towers[a]
    b. yearly maintenance cost of the dry cooling towers[a]
    c. incremental cost of plant due to dry cooling tower heat rejection
    d. yearly capital cost of the piping and pumping system[a]
    e. yearly capital cost of the surface condenser[a]
    f. incremental cost of fuel for the power plant with a dry cooling tower heat rejection system.

4.1.6 Dry-Cooled Power Plant Performance and Incremental Cost for a Yearly Temperature Profile (NOVART, VARIT)

Subroutine NOVART calculates the performance parameters of the plant and dry cooling towers over a yearly temperature profile. These are, in turn, used to calculate the incremental power production costs of the dry-cooled plant. NOVART is called for each set of independent parameters as NOVART but only for the optimum dry cooling tower design. It provides a printout of performance parameters and incremental costs of the dry-cooled plant.

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[a] Yearly cost for capital is equal to the fixed charge rate times the capital cost.

The performance of the power plant and the dry cooling tower is determined as a function of the yearly temperature profile which is an input to the code. Calculation of plant and dry cooling tower performance is based on no change in air-side velocity through the heat exchanger from the design conditions. Therefore, the air-side heat transfer coefficient is constant.

The calculations are performed in the following sequence:

1. Calculate thermal efficiency of the turbine at the design point of the plant.

2. Calculate thermal efficiency of the turbine at the rated back pressure.

START OF LOOP

Calculate Exit Temperature of Steam Turbine
as a Function of Ambient Air Temperature

3. Guess turbine exit temperature.

4. Calculate thermal efficiency of steam turbine for the specified exit temperature of the turbine.

5. Calculate temperature range of water flowing through the heat exchanger.

6. Calculate terminal temperature difference between the water and the steam entering the surface condenser.

7. Calculate exit temperature of the steam turbine based on the previous estimate.

8. Check if exit temperature has converged.

END OF LOOP

9. Calculate total power output of steam turbine for a particular ambient air temperature.

10. Calculate power output of plant to electrical power transmission network at a particutlar ambient air temperature.

11. Calculate cost of fuel for gas turbine makeup power at a particular ambient air temperature.

12. Calculate incremental cost of fuel for steam supply system due to steam turbine operating at off-design conditions.

13. Calculate capital cost of gas turbine used for makeup power when ambient air temperature is greater than design air temperature.

14. Calculate incremental cost of producing power at a particular ambient air temperature.

15. Calculate summation of incremental power production costs.

4.2 PLASTIC SYSTEM

The calculational procedure for the plastic tube advanced concept is very similar to that for the SOA system. However, the procedure differs in two areas:

1. calculation of the air-side heat transfer surface area and heat transfer coefficient, and

2. costing of the heat exchanger bundle.

The air-side heat transfer surface area per foot is less complicated to determine for the bare tubes of the PLASTIC concept than for the metal-finned tubes of the SOA system. The air-side heat transfer coefficient for the bare tubes of the PLASTIC system is determined by a set of correlations (Appendix A) taken from the literature. There is a set of correlations for each tube arrangement with each correlation covering a range of Reynolds numbers.

Because the PLASTIC code deals with heat exchangers having bare tubes, the costing algorithm for the heat exchanger differs from the SOA algorithm. The algorithm used for costing the plastic tube heat exchanger bundles is given in Reference 2.

4.3 AMMONIA (NH_3) SYSTEM

The calculational procedure used to design the ammonia cooling system is described here. The procedure begins after choosing five independent parameters by the pattern and gradient search routines developed by MIT(4) and detailed in Appendix C.

The five independent parameters determining the design of the ammonia cooling system are:

1. saturation temperature of steam leaving the turbine
2. temperature range of the air through the heat exchanger
3. temperature difference between the ammonia and saturated steam in the condenser/reboiler
4. total frontal area of the heat exchanger
5. width/length ratio of the heat exchanger.

NH_3 Dry Cooling Tower System Design (CALC)

All information required by this procedure not provided by the independent parameters is provided by the input (see Section 3.4). The steps of the calculation procedure are outlined below. Figure 4-3 illustrates the procedure in flow diagram form.

1. The heat rejected by the plant is determined by using saturation temperature of the steam leaving the turbine to find the turbine back pressure, which is then used to find the efficiency of the plant. The efficiency is then used along with the plant size to determine the heat rejected by the plant.
2. The properties of air and NH_3 are determined by using the temperature range of the air, the saturation temperature of the steam, and the temperature difference in the condenser/reboiler.
3. Using properties of the air and NH_3, the amount of heat rejected by the plant, and the range of the air, the mass flow rates of NH_3 and air are calculated.
4. The width and length of the heat exchanger are calculated from the frontal area and the width-to-length ratio of the heat exchanger.
5. The design of the condenser/reboiler is performed by using the heat rejected by the plant, the steam temperature, and the temperature difference between the steam and the NH_3 (see Section 4.3.1).

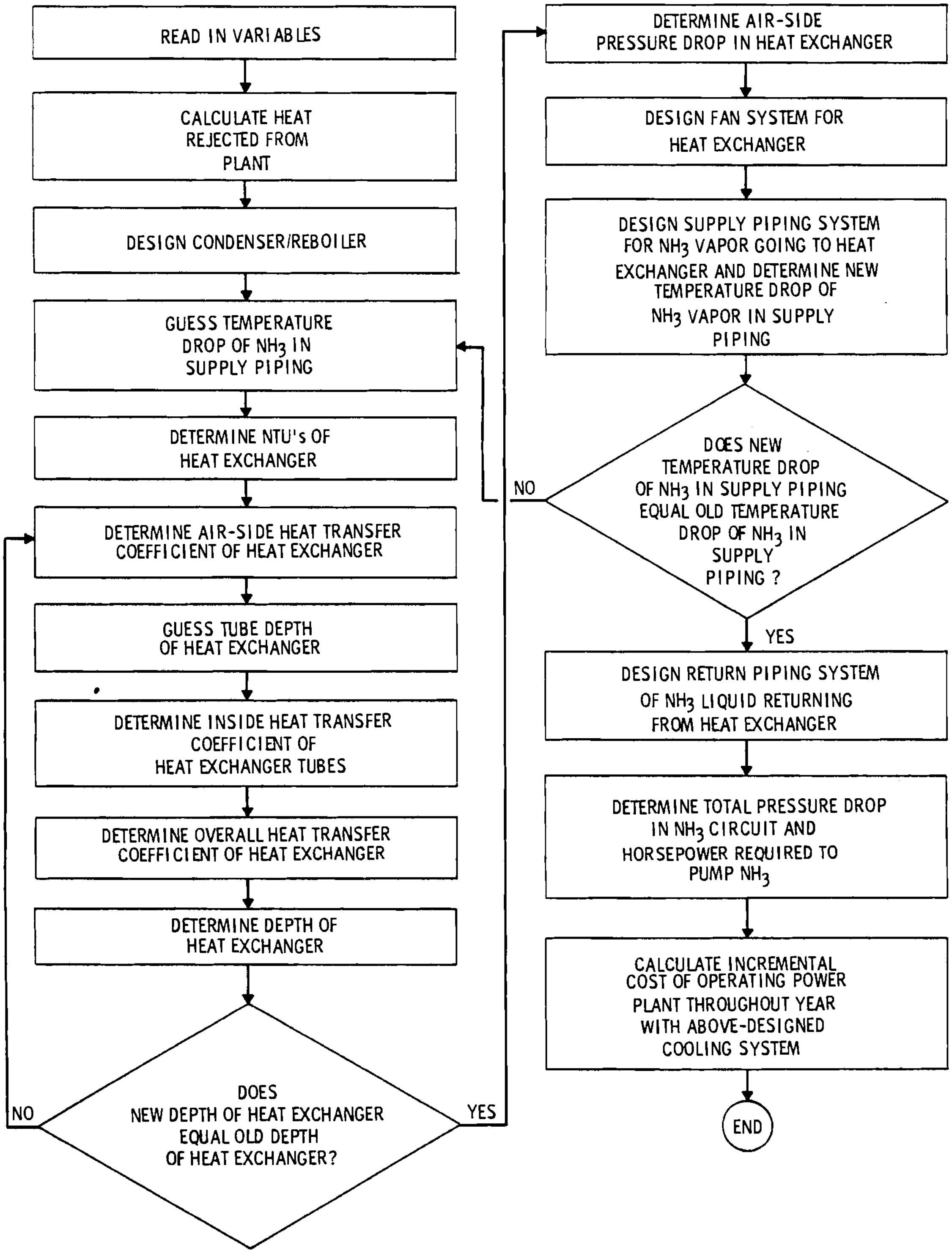


FIGURE 4-3. Subroutine CALC Flow Diagram (Ammmonia Version)

6. The tower system design, consisting of the number of towers, the tower diameter, and the tube depth of the heat exchanger is determined by a double interation loop. This calculation procedure consists of the following steps:

   (1) Determination of the number of transfer units (NTU) in the heat exchanger using the temperature range of the air, temperature of the ammonia vapor leaving the condenser/reboiler and an estimate of the temperature drop of the NH_3 ammonia in the piping going from the condenser/reboiler to the towers.

   (2) Calculation of the air-side heat transfer coefficient of the tubes in the heat exchanger from a knowledge of the air mass flow rate, the heat exchanger frontal area, and the tubing unit geometry.

   (3) The fin and surface efficiency of the heat exchanger tubes is calculated as a function of the air-side heat transfer coefficient.

   (4) The inside heat transfer coefficient is determined by an iteration loop that finds the depth of the heat exchanger. The following steps are contained in the loop:

      (i) An estimate of the heat exchanger depth is used to determine the liquid Reynolds number in the tubes, which is then used to find the inside heat transfer coefficient.

      (ii) The overall heat transfer coefficient is determined from the air-side and inside heat transfer coefficients and the surface efficiency of the outside surface.

      (iii) Using the number of transfer units (NTUs), the air mass flow rate, and the overall heat transfer coefficient the depth of the heat exchanger is determined. If the depth does not agree with the estimated tube depth in (i), a new depth is estimated and steps (i) through (iii) are performed again.

(5) Knowing the tube depth of the heat exchanger and the air-side Reynolds number, the pressure drop across the heat exchanger is calculated.

(6) The heat exchanger cost is determined from its size (see Section 4.1.2).

(7) Using the pressure drop and the frontal area of the heat exchanger, the least costly fan diameter, number of blades, and blade angle are determined. Once the type of fan is chosen, the number of fans and the number and diameter of the towers are calculated. (see Section 4.1.3).

(8) Using the number and diameter of the towers, the supply piping is designed and the temperature drop of the NH_3 vapor is determined. (Calculational procedure for the design of the supply piping is similar to the procedure in Section 4.1.4, except NH_3 vapor is the fluid designed for instead of water.)

(9) If the temperature drop in Step (8) differs significantly from the temperature drop used in Step (1), the temperature drop of Step (8) is used in Step (1) and the procedure is repeated.

7. After the tower system is known, the return piping is designed and the pressure drop of the ammonia liquid returning from the towers is calculated. (Calculational procedure for the return piping is similar to the procedure in Section 4.1.4, except NH_3 is the liquid being designed for instead of water.)

8. The pressure drop of the two-phase flow of ammonia in the heat exchanger tubes is calculated.

9. The pressure drop throughout the entire NH_3 loop is calculated along with the horsepower required for the pump power.

10. The plant is scaled up to obtain the designated output of the plant including the pump and fan power. Costs of the cooling system are also scaled up (see Section 4.1.5).

11. The incremental cost of operating the power plant with the cooling system over the entire year is calculated (see Section 4.3.2).

4.3.1 Condenser/Reboiler Design and Costing

4.3.1.1 Condenser/Reboiler Design (SPDES)

Subroutine SPDES determines the size of the condenser/reboiler on the basis of:

- the designated temperature difference between the saturated steam and ammonia
- the amount of heat to be transferred
- the temperature of the steam
- the length of the tubes, and
- exit quality of the ammonia leaving the condenser/reboiler tubes.

The design of the condenser/reboiler is done by a double iteration loop using correlations for the steam- and ammonia-side heat transfer coefficients to determine the temperature drops across both fluid boundaries and the average bulk velocity of the ammonia inside the condenser tubes. Once these values are found, the size of the condenser/reboiler is determined along with the system performance parameters.

The condenser/reboiler design is determined in the following sequence:

START OF LOOP I

Determine Average Velocity of NH_3 through Condenser/Reboiler Tubes

1. Calculate the forced convection coefficient by using a guess for the velocity of ammonia in the tubes.

START OF LOOP II

Determine Temperature Drop across Fluid Layers and Condenser Tube Wall

2.1 Calculate the heat flux through the outside tube wall by estimating the temperature drop on the inside fluid boundary and the forced convection coefficient.

2.2 Calculate the temperature drops across the steam condensate and the tube wall by the use of the heat flux through the wall.

2.3 Determine the total temperature drop caused by the estimation of the ammonia side temperature drop and then determine a new estimate for the ammonia side temperature drop and return to Step 2.1 until the total temperature drop converges to the desired value.

END OF LOOP II

3. Using the heat flux previously determined a new ammonia velocity is calculated. This new velocity is used in Step 1 and the same steps are repeated from there on until the velocity does not vary from its previous value by a certain percentage.

END OF LOOP I

4. Using the heat flux from Step 3 and the amount of heat transferred through the condenser/reboiler per unit time the outside heat transfer area and the overall, outside, and inside heat transfer coefficients are determined.

5. Using the velocity in the tubes, the pressure drop incurred by the ammonia flowing through the tubes of the condenser/reboiler is calculated.

4.3.1.2 Condenser/Reboiler Costing (SURCON)

Subroutine SURCON costs the components (shell, tubes, accessories) of the condenser/reboiler through a modification of the code for a water-side condenser so that boiling ammonia can be handled on the inside of the tube. The cost of field erection of the condenser/reboiler is also added into the total cost of the system. The modified code is set up to handle regular condenser tubing and enhanced tubing (enhanced heat transfer).

The calculations performed by subroutine SURCON proceed according to the following sequence:

1. Select tubing cost arrays to be used in costing.
2. Calculate total length of tubing required in the condenser/reboiler.
3. Determine the cost discount rate based on the total amount of tubing.
4. Determine the penalty for the length of tubing chosen.
5. Calculate the total cost of the tubing material.
6. Calculate the shell cost of the condenser/reboiler.
7. Calculate the field erection costs of the condenser/reboiler.
8. Determine the total cost of the condenser/reboiler.

4.3.2 Incremental Cost of the Dry Cooling System for the Power Plant Operating Over the Yearly Temperature Profile (NOVART and VARIT)

The incremental cost of operating a dry cooled plant over the entire year is determined by finding the operating conditions of the plant as a function of a yearly ambient temperature profile for the plant site. The performance of the plant is calculated on the basis of unchanged effectiveness of the tower heat exchanger and overall heat transfer coefficient in the condenser/reboiler.

The two routines proceed in the sequence shown below.

1. Determine the thermal effectiveness for the plant at both the design and rated back pressure of the low-pressure turbine.
2. Select the ambient air temperature at which the plant is operating from an ambient temperature array that approximates the yearly temperature profile at the plant site.
3. Estimate the saturation temperature of the steam in the condenser/reboiler using the ambient temperature.

4. Using the saturation temperature, calculate the turbine back pressure. The back pressure is then used to determine the thermal efficiency of the steam turbine and the amount of heat rejected from the plant.

5. The temperature difference in the condenser/reboiler, the pressure and temperature drops in the supply piping, and the range of the air in the heat exchanger are determined from the amount of heat rejected from the plant.

6. Using the temperature drop in the supply piping, the temperature difference in the condenser/reboiler, the range of the air, and the ambient temperature, a new steam saturation temperature is calculated.

7. If the steam saturation temperature of Step 5 varies from the saturation temperature used in Step 3 by a certain percentage, the new temperature is used as the new estimate for the saturation temperature and the procedure from Step 3 on is repeated.

8. The total power output of the steam turbine, the power output of the plant to the electrical power transmission network, the cost for auxiliary power, the increased fuel cost of the plant, and the capital cost of the auxiliary power are determined for the ambient temperature chosen, given the true saturation temperature.

9. Calculate the incremental cost of operating the power plant with the dry cooling system at the ambient temperature chosen. The cost is then multiplied by the percent of the year that the ambient temperature exists to obtain a portion of the yearly incremental power production cost.

10. Repeating Steps 2 through 9 for an ambient temperature array that approximates the yearly temperature profile of the plant site, a set of incremental power production costs is obtained.

11. Sum the set of incremental power production costs from Step 9 to obtain the incremental power production costs of operating the plant over the entire year.

5.0 COMPUTER PROGRAMS

This section describes the computer programs developed for the three dry cooling tower cost optimizations.

5.1 GENERAL INFORMATION

The dry cooling tower cost optimization models are coded in FORTRAN-IV EXTENDED. They were compiled and executed on a CDC-7600 computer under the SCOPE 2.1.3 System. Central memory requirements (octal words) are approximately 61000 for the PLASTIC tube model, 74000 for the NH_3 model, and 76000 for the SOA metal fin tube model. Execution time ranges from one to sixty or more seconds, depending upon the nearness of starting point to the final conditions and especially upon such factors as the number and severity of the constraints imposed (for example, heat exchanger geometry, turbine outlet temperature, range, number of different fans to be considered). Execution time is strongly affected by the number of different fans to be considered, as specified by the input parameters LFB and LFE (see card type S1 in Input Description). If the model is restricted to fans of only one diameter, execution time may be less than five seconds instead of the forty or more required if all fan diameters are considered.

Particular care should be used to avoid overconstraining the model. Experience has shown that the optimization scheme may fail to converge or may converge to a false minimum if the constraints are too numerous or too severe. Input variables which impose constraints and thus could interfere with optimization are:

- turbine outlet temperature constraints (TLIM, TFIX);
- water range[a] (RFIX);
- minimum air-side pressure drop (XDEPA);
- heat exchanger depth and length limits (DEEPL, FIXL);
- total number of towers (TOWMAX); and
- the minimum number of towers per group (TOWMIN).

Of these constraints, only XDEPA was present in the optimization routines (SHOT, CHNGE, SERCH, XTEND) in their original form. All other constraints

[a] does not exist in the ammonia version

have been incorporated in the development of the current code. The subroutines in which these variables are active, and the mechanisms through which they function to impose constraints, are summarized in Table 5-1.

TABLE 5-1. Subroutines Containing Constraint-Imposing Variables

Subroutine	Variable	Mechanism
CALC	DEEPL	Discard current trial point. Return to calling routine and begin new trial
	FIXL	Set length to FIXL instead of calculating it from the width/length ratio
SERCH	TFIX, RFIX	Trial values of turbine temperature or range are set to TFIX or RFIX
	TLIM	Trial values of turbine temperature exceeding TLIM are set equal to TLIM
	XDEPA	If air-side pressure drop is less than XDEPA, call subroutine CHNGE for two-variable search.
SHOT	TFIX, RFIX, TLIM	Same as in subroutine SERCH
	XDEPA	If pressure drop is less than XDEPA, reduce frontal area of heat exchanger
CHNGE	TFIX, RFIX, TLIM	Same as in subroutine SERCH
	XDEPA	Make two-point search by changing a variable different from that being changed by SERCH
XTEND	TFIX, RFIX, TLIM	Same as in subroutine SERCH
	XDEPA	Do not consider this point as a new starting point. Pick another one.
FAN	TOWMAX, TOWMIN	If total number of towers exceeds TOWMAX or if the number of towers per group is less than TOWMIN, reject this trial point by setting incremental cost to unreasonably high value (1×10^{38})

The complex interaction of variables in the system being optimized makes it extremely difficult to predict which constraints are likely to cause problems in any one case. In general, an unconstrained case should be run before any attempt is made to apply constraints.

Formal proof of correctness has not been attempted for the programs described here. However, comparison of computed costs of dry cooling system components with costs reported in the literature shows good agreement. Furthermore, the results of runs covering a range of system designs show no spurious results. In the course of code development, a number of potential problem areas were noted, most of them concerned with failure of the program to recognize physically impossible conditions. Corrective actions and warning messages have been incorporated in all problem areas which were found. As an aid to visualizing the interrelationships among subprograms, subroutine linkage charts in Appendixes E, F, and G indicate which subroutines may call or be called by others.

5.2 INPUT

All input to the programs is by punched cards. Sample input listings are shown in Figures 5-1 through 5-3 for the SOA, PLASTIC and NH_3 systems, respectively. The circled symbols on the figures are used as cross references to corresponding cards in later discussion of input/output.

```
A1   ***
     ***            EXAMPLE RUN - METAL FIN SOA VERSION
     ***
A2
B1   50.      50.      95.      1.       300000.  20.      80.
C1   100. 95. 85.  75. 65.  55. 45.  35. .0015.0100.0285.0822.0913.1096.1119.5650
C2
D1   1000.    .4       .65      .174     .01      1.       4410.    40.
E1   81.      24.      440.     0.       146.65   121000.  30.
E2   1.611    1.464    1.614    1.210    1.331    1.210    1.464
F1   3.5      1.            -.0049       .002675       -.0001125        MF1
G1   .85       125.
H1   1.       0.       153.     0.       2.       0.       0.       0.
H2   .90
I1   1.74     1.00     7.03     0.00     2.22     2.50     0.55     4.60
J1   TABLE             2.       12.      1.       5.
J2   12.      18.      12.      500.
K1   2.572    2.227    2.572    .082     2.572    .018
L1   1.00     .930     20.      STEEL WELDED      31.
M1   DOUBLE FOOTED     ALUMINUM 118.
N1   0.       0.       .55      .2       2.40
P1   FORM REMOVABLE    GALVANIZED.625
Q1   0.       2.       .5       20.      5.625    .48      5.
R1   2.       7.       1.       18.      .95      5.       ADMIRALTY
S1
T1   PFR
```

FIGURE 5-1. Sample Input for SOA Model

```
(A1) ***
     ***              EXAMPLE RUN - PLASTIC TUBE VERSION
     ***
(A2)
(B1)    50.0      50.0                  1.00         5.00E05  20.0      80.00
(C1) 100. 95.   85.   75.   65.   55.   45.   35.   .0015.0100.0285.0622.0913.1096.1119.5650
(C2)
(D1) 1000.      .4         .65        .174       .01        1.         4410.      40.
(E1) 81.        24.        440.       0.         146.65     121000.    30.
(E2) 1.611      1.464      1.614      1.21       1.331      1.21       1.464
(F1) 3.5        1.               -.0049          .002675          -.0001125
(G1)  0.85      125.
(H1)  1.0       0.0        153.0       0.0        2.00                             0.0
(H2) 0.9
(I1)   1.74      1.0       7.03       0.0        2.22       2.5        .55        4.6
(J1) TABLE                 2.         12.        1.         1.5
(J2)  12.00     18.        12.        500.0
(K1)  0.5000     0.43301    0.5000    STAGGERED 2            60.0
(L1)  0.2500     0.2000     20.00     POLYETHYLENE           0.2910
(M1) POLYCARBONATE         .25        .25        2.5        .5         .284       .04
(N1)   0.00       0.00      0.40
(P1) FORM REMOVABLE        GALVANIZED.625
(Q1)   0.00      0.00       0.050      20.0       5.625      0.000      5.0
(R1)   2.00      7.00       1.00       18.00      0.950      5.00       ADMIRALTY
(S1)    55   88
(U1)  $READIN NUCASE=0 $END
```

FIGURE 5-2. Sample Input for PLASTIC Model

```
(A1) ***
     ***              EXAMPLE RUN  -   AMMONIA VERSION
     ***
(A2)
(B1) 50.        50.        105.       3.         300000.    20.        80.0       3.7
(C1) 100. 95.   85.   75.   65.   55.   45.   35.   .0015.0100.0285.0622.0913.1096.1119.5650
(C2)
(D1) 1000.      .4         .65        .174       .01        1.         4410.      40.
(E1) 81.        24.        440.       0.         146.65     121000.    30.
(E2) 1.611      1.464      1.614      1.210      1.331      1.210      1.464
(F1) 3.5        1.               -.0049          .002675          -.0001125
(G1) .85
(H1) 1.0        0.         153.       6.         2.0                   6.
(H2) .90
(I1) 1.74       1.0        0.00       1.82       2.22       2.5        .55        4.6
(J1) EQUATION              1.         12.        1.         5.         90.
(J2) 15.        0.         0.         500.
(J3) 150.       0.         0.
(K1) 2.822      2.444      2.822      .082       2.572      .018
(L1) 1.         .93        20.        STEEL WELDED          31.
(M1) DOUBLE FOOTED         ALUMINUM   118.
(N1) 0.         0.         .55        .2         2.4
(P1) FORM REMOVABLE        GALVANIZED 1.125
(Q1) 0.         2.         .5         20.        10.15      .48        5.
(R1) 2.         1.         1.         12.        118.       5.         LINDE AL12
(R2) 1.         50.        .8
(R3)    1
(S1)    55   88
(T1) PFR
```

FIGURE 5-3. Sample Input for NH_3 Model

5.2.1 SOA Model Input Description

```
C
C                       I N P U T    D E S C R I P T I O N
C                      FOR BNW1 METAL FIN ROUND DCT
C
*-*-*-*-*-*-*             D E C E M B E R  1 0   1 9 7 6            *-*-*-*-*-*-*
C*** *** ***
C                       IN THE DESCRIPTIONS BELOW THE SYMBOL (DF) MEANS
C                       DECIMAL FRACTION
C
C ---- ------
C                                    T
C                                    Y
C CARD                     NAME   OF P
C TYPE COLUMNS  FORMAT   VARIABLE  E          DESCRIPTION AND (UNITS)
C ---- ------
C
C  A1     1-80  (8A10)   DESCR(I) R  COMMENTS OR CASE DESCRIPTION TO BE
C                                    PRINTED AT TOP OF FIRST OUTPUT PAGE.
C                                    ANY NUMBER OF TYPE A1 CARDS MAY BE
C                                    USED BUT A BLANK A1 CARD IS NOT
C                                    ALLOWED.
C
C
C.........
C
C  A2     1-80  BLANK IN ALL COLUMNS.  ONE TYPE A2 CARD ONLY
C
C.........
C.........
C
C  B1     1-10  (E10.0,  TSTAR     R  LOWEST DESIGN TEMPERATURE TO USE
C                                    (DEGREES F)
C
C        11-20   E10.0,  TEND      R  HIGHEST DESIGN TEMPERATURE TO USE
C                                    (DEGREES F)
C
C        21-30   E10.0,  VAS(1)    R  INITIAL VALUE FOR TURBINE OUTLET
C                                    TEMPERATURE (DEGREES F)
C
C        31-40   E10.0,  VAS(3)    R  INITIAL VALUE FOR CWARA, RATIO OF
C                                    MCP WATER / MCP AIR   (DF)
C
C        41-50   E10.0,  VAS(4)    R  INITIAL VALUE FOR AFRON, AIR SIDE
C                                    FRONTAL AREA (SQ FT)
C
C        51-60   E10.0,  VAS(5)    R  INITIAL VALUE FOR WLRAT, RATIO OF
C                                    HEAT EXCHANGER WIDTH/LENGTH   (DF)
C
C        61-70   E10.0,  FIXL      R  LENGTH OF HEAT EXCHANGER (FT).
C                                    IF THE ABSOLUTE VALUE OF FIXL=0.
C                                    THEN THE PROGRAM WILL CALCULATE
C                                    THE LENGTH.
C
C        71-75   F5.0,   DEEPL     R  LIMITING VALUE FOR NUMBER OF TUBES
C                                    IN DEPTH FOR HEAT EXCHANGER.
C                                   (NUMBER OF TUBES)
C
C                                    IF THE ABSOLUTE VALUE OF DEEPL = 0.0
C                                    THEN THE PROGRAM WILL COMPUTE THE
```

```
C                                               DEPTH WITH NO CONSTRAINTS
C
C
C                                               IF POMDPL IS BLANK OR ZERO THEN A
C                                               POSITIVE VALUE FOR DEEPL IMPLIES A
C                                               CONSTRAINT OF EQUAL TO OR GREATER
C                                               THAN DEEPL, WHILE A NEGATIVE VALUE
C                                               FOR DEEPL IMPLIES A CONSTRAINT OF
C                                               EQUAL TO OR LESS THAN DEEPL.
C
C        76-80    F5.0)    POMDPL    R  TOLERANCE ON DEEPL. NOTE THAT A ZERO
C                                               OR BLANK IS NOT INTERPRETED AS ZERO
C                                               TOLERANCE. (SEE DEFINITION OF DEEPL)
C
C.........
C.........
C
C  C1     1-40   (8F5.0,   TA(I)     R  TEMPERATURES REPRESENTATIVE OF
C                                               TPER(I) FRACTION OF YEAR (DEGREES F)
C                                               C A U T I O N  -  HIGHEST TA
C                                                                  MUST BE FIRST
C
C        41-80    8F5.0)   TPER(I)   R  FRACTION OF YEAR OVER WHICH TEMPERA-
C                                               TURE TA(I) IS TYPICAL. (DF)
C
C.........
C
C  C2     1-40   (8F5.0,   TA(I)     R  CONTINUATION OF TA(I) FOR I= 9 TO 16
C
C        41-80    8F5.0)   TPER(I)   R CONTINUATION OF TPER(I)
C
C        NOTE - A TYPE C2 CARD IS REQUIRED EVEN IF BLANK.
C
C.........
C.........
C  D1     1-10   (E10.0,   PSIZE     R  BASE PLANT SIZE (MEGAWATTS)
C
C        11-20    E10.0,   TEFF      R  BASE THERMAL EFFICIENCY (DF)
C
C        21-30    E10.0,   CAPF      R  CAPACITY FACTOR  (DF)
C
C        31-40    E10.0,   FCR       R  FIXED CHARGE RATE  (DF)
C
C        41-50    E10.0,   PER       R  RATIO MAINTENANCE COST TO CAPITAL
C                                               COST  (DF)
C
C        51-60    E10.0,   CCM       R  CONSTRUCTION COST MULTIPLIER (DF)
C
C        61-70    E10.0,   ELEV      R  SITE ELEVATION ABOVE SEA LEVEL (FT)
C
C        71-80    E10.0)   ROOFL     R  ROOF LOAD (LB/SQ FT)
C
C.........
C.........
C
C  E1     1-10   (E10.0,   FCOS      R  FUEL COST (CENTS/MMBTU)
C
C        11-20    E10.0,   PWCOS     R  REPLACEMENT POWER COST (MILLS/KWH)
C
C        21-30    E10.0,   PLANC     R  POWER PLANT CONSTRUCTION COST ($/KW)
C
```

```
C        31-40   E10.0,   COSTL      R  COST OF LAND  ($/SQ FT)
C
C        41-50   E10.0,   CSSPKW     R  COST OF STEAM SUPPLY ($/KW)
C
C        51-60   E10.0,   CAPCHG     R  CAPACITY CHARGE ($/MEGAWATT)
C
C        61-70   E10.0,   HPCST      R  COST OF ELECTRIC MOTORS ($/HP)
C
C        71-80   E10.0)   CTURB      R  EXTRA TURBINE COST BECAUSE OF
C                                       NUCLEAR POWER PLANT ($/KW)
C
C.........
C                                             INDIRECT COST FACTORS FOR -
C  E2     1-10  (E10.0,   POHBAF     R  BUNDLE ASSEMBLE AND FRAME
C
C        11-20   E10.0,   POHFAN     R  FANS
C
C        21-30   E10.0,   POHLEC     R  ELECTRICAL
C
C        31-40   E10.0,   POHCIR     R  CIRCULATION PIPING
C
C        41-50   E10.0,   POHCND     R  CONDENSER
C
C        51-60   E10.0,   POHSTC     R  STRUCTURE
C
C        61-70   E10.0)   POHSCL     R  SCREENS AND LOUVERS
C
C.........
C.........
C
C  F1     1-10  (F10.0,   RBP        R  TURBINE RATING BACK PRESSURE (INCHES
C                                       OF MERCURY)
C
C        11-70   4F15.8)  TPO(I)     R  COEFFICIENTS FOR THIRD ORDER POLY-
C                                       NOMIAL FOR HEAT RATE AS A FUNCTION
C                                       OF TURBINE BACK PRESSURE.
C
C.........
C.........
C
C  G1     1-10  (E10.0,   EFFP       R  EFFICIENCY OF PUMPS (DF)
C
C        11-20   E10.0)   DPRESS     R  PIPING DESIGN PRESSURE (PSI)
C
C                                       MAY BE
C                                               50.
C                                               75.
C                                              125.
C
C.........
C.........
C
C  H1     1-10  (E10.0,   VELREC     R  CONTROL VARIABLE.
C                                      =1 FOR VELOCITY RECOVERY
C                                      =0  FOR NO VELOCITY RECOVERY
C
C        11-20   E10.0,   XDEPA      R  MINIMUM AIR SIDE PRESSURE DROP THRU
C                                       HEAT EXCHANGER (LB FORCE/SQ FT)
C
C        21-30   E10.0,   TLIM       R  MAXIMUM STEAM TEMPERATURE FOR THE
C                                       TURBINE (DEGREES F).
```

```
C                                            ASSUMED 180 IF FIELD IS BLANK OR 0.
C
C       31-40   E10.0,  TOWMIN   R  THE MINIMUM ALLOWABLE NUMBER OF
C                                   TOWERS PER GROUP.
C                                   ASSUMED 1 IF NOT SPECIFIED
C
C       41-50   E10.0,  PFACT    R  PACKING FACTOR, RATIO OF CIRCULAR
C                                   TOWER ROOF AREA TO FAN SWEPT AREA
C                                   (DF)
C
C
C       51-60   E10.0,  TFIX     R  FIXED TURBINE OUTLET TEMPERATURE (F)
C
C                                   ENTRY IS IGNORED IF BLANK OR ZERO
C
C       61-70   E10.0,  RFIX     R  FIXED RANGE (F).  ENTRY IS IGNORED
C                                   IF BLANK OR ZERO
C
C       71-80   E10.0)  TOWMAX   R  MAXIMUM ALLOWABLE NUMBER OF TOWERS.
C                                   ASSUMED 9999 IF BLANK OR ZERO
C
C.........
C
C  H2    1-10   E10.0   GBEFF    R  FAN GEARBOX EFFICIENCY
C
C.........
C.........
C
C  I1    1-10  (E10.0,  CPM      R  PLENUM COST   ($/LB)
C
C       11-20   E10.0,  WPL      R  WEIGHT OF PLENUM MATERIAL (LB/SQ FT)
C
C       21-30   E10.0,  CLUVR    R  COST OF LOUVERS   ($/SQ FT)
C
C       31-40   E10.0,  CHAILS   R  COST OF HAIL SCREENS ($/SQ FT)
C
C       41-50   E10.0,  UCS      R  UNIT COST OF STACK $/LB
C
C       51-60   E10.0,  UWS      R  UNIT WT OF STACK  LB/CU FT
C
C       61-70   E10.0,  CVM      R  FAN RING MATERIAL UNIT COST $/LB
C
C       71-80   E10.0)  WFV      R  UNIT WT OF STRAIGHT CYLINDER LB/SQ FT
C
C.........
C.........
C
C  J1    1-20  (2A10,   NTUCAL   I  CONTROL VARIABLE FOR SPECIFYING THE
C                                   METHOD TO BE USED TO CALCULATE NTU.
C                                   NTUCAL MUST BEGIN IN COLUMN 1.
C                                                       CROSS FLOW
C                                                       COUNTER FLOW
C                                                       TABLE
C
C                                   CROSS FLOW  OR  COUNTER FLOW CAUSES
C                                   CALCULATION OF NTU BY EQUATIONS.
C                                   TABLE  CAUSES TABLE LOOKUP OF NTU.
C
C       21-30   F10.0,  HXNP     R  NUMBER OF PASSES THRU HEAT EXCHANGER
C
C       31-40   F10.0,  WB       R  HEADER LENGTH  (FEET)
```

```
C
C        41-50    F10.0,   NTUB       R   NUMBER OF TUBES THRU PLATE FINS
C
C        51-60    F10.0)   SS         R   SUPPORT SPACING (FT)
C
C.........
C
C   J2     1-10   (E10.0,  DESVEL     R   DESIGN WATER VELOCITY FOR PIPING
C                                         (FT/SEC)
C
C        11-20    E10.0,   REDUCE     R   MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                         FOR TOWER PIPING (INCHES)
C                                         MAY BE 6, 12, OR 18
C
C        21-30    E10.0,   QREDUCE    R   MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                         FOR QUADRANT HEADERING . (INCHES)
C                                         MAY BE 6, 12, OR 18
C
C        31-40    E10.0)   DIST       R   DISTANCE FROM TOWER TO CONDENSER
C                                         ROOM (FT)
C
C.........
C.........
C
C   K1     1-10   (E10.0,  XW         R   TUBE TRANSVERSE PITCH (NORMAL TO AIR
C                                         FLOW)  (INCHES)
C
C        11-20    E10.0,   XD         R   TUBE PITCH IN DIRECTION OF AIR FLOW
C                                         (INCHES)
C
C        21-30    E10.0,   XDG        R   TUBE DIAGONAL PITCH (INCHES)
C
C        31-40    E10.0,   SF         R   FIN SPACING (INCHES)
C
C        41-50    E10.0,   DFIN       R   FIN DIAMETER (INCH(S)
C
C        51-60    E10.0,   THFIN      R   FIN THICKNESS (INCHES)
C
C        61-70    E10.0)   DANGLE     R   BASE ANGLE FOR DELTA HEAT EXCHANGER
C                                         (DEGREES)
C
C
C.........
C.........
C
C   L1     1-10   (F10.0,  ODL        R   HX LINER OUTSIDE DIAMETER (INCHES)
C
C        11-20    F10.0,   DI         R   HX LINER INSIDE DIAMETER (INCHES)
C
C        21-30    F10.0,   GAGLIN     R   GAGE OF HX LINER TUBE  (MAY BE
C                                         22.  20.  19.  18.  17.  16.  15.
C                                         14.5 14.  13.  12.  11.  10.      )
C
C        31-50    2A10,    TUBMAT,    R   LINER MATERIAL. MUST BEGIN IN COL 31
C                          XTUBMA     R
C
C                                         MAY BE -
C                                                   ADMIRALTY
C                                                   COPPER
C                                                   CU-10 NI
C                                                   CU-30 NI
```

```
C                                                     ALUMINUM
C                                                     STEEL WELDED
C                                                     STEEL SEAMLESS
C                                                     WELDED SST
C
C       51-60    F10.0) CONL      R  THERMAL CONDUCTIVITY OF LINER MATER-
C                                    IAL  (BTU/(HR SQFT DEG F/ FT) )
C
C.........
C.........
C
C  M1    1-20   (2A10,   FINTYP   R  TYPE OF FIN.  MUST START IN COLUMN 1
C                        XFINTY   R
C
C                                    MAY BE -
C                                             STRAIGHT FIN
C                                             SINGLE FOOT
C                                             DOUBLE FOOT
C                                             EMBEDDED
C                                             EXTRUDED
C                                             PLATE
C
C
C       21-30   A10,     FINMAT   R  FIN MATERIAL.  MUST START IN COL. 21
C
C                                    MAY BE -
C                                             ALUMINUM
C                                             STEEL
C
C       31-40   F10.0)   CONF     R  THERMAL CONDUCTIVITY OF FIN MATERIAL
C                                    (BTU/(HR SQFT DEG F/FT))
C
C.........
C.........
C
C  N1    1-10   (E10.0,  CFB      R  FIN BONDING COST ($/SQ FT)
C
C       11-20   E10.0,   COATC    R  FIN PROTECTIVE COATING COST ($/SQFT)
C
C       21-30   E10.0,   ZINCC    R  ZINC COST FOR SPACERS ($/LB)
C
C       31-40   E10.0,   CASTC    R  FIXED CASTING COST FOR SPACERS
C                                    ($/SPACER)
C
C       41-50   E10.0)   EPREPC   R  END PREPARATION COST ($/TUBE)
C
C.........
C.........
C
C  P1    1-20   (2A10,   HEDTYP,  R  HEADER TYPE.  MUST START IN COLUMN 1.
C                        XHEDTY   R
C                                    MAY BE -
C                                             WELD REMOVABLE
C                                             WELD PLUG
C                                             FORM REMOVABLE
C                                             FORM PLUG
C
C
C       21-30   A10,     HEDMAT   R  HEADER MATERIAL. MUST START IN
C                                    COLUMN 21. MAY BE -
C                                                       ALUMINUM
```

```
C                                                         STEEL
C                                                         GALVANIZED
C
C       31-40    F10.0)    TW        R  THICKNESS OF HEADER MATERIAL, INCHES
C
C.........
C.........
C
C  G1    1-10   (E10.0,    CRJ       R  COST OF ROLLED JOINT, TUBE TO HEADER
C                                       ($/TUBE)
C
C       11-20    E10.0,    CWJ       R  COST OF WELDED JOINT, TUBE TO HEADER
C                                       ($/TUBE)
C
C       21-30    E10.0,    CHH       R  TUBE AND PLUG HOLE PREPARATION COST
C                                       ($/HOLE)
C
C       31-40    E10.0,    CN        R  NOZZEL AND ATTACHING COST ($/HOLE)
C
C       41-50    E10.0,    CMW       R  COST OF HEADER MACHINING AND WELDING
C                                       ($/FT)
C
C       51-60    E10.0,    CST       R  STRUCTURAL STEEL COST  ($/LB)
C
C       61-70    E10.0)    CBJ       R  COST OF BOLTED HEADER JOINT ($/FT)
C
C.........
C.........
C
C  R1    1-10   (E10.0,    XNS       R  NUMBER OF CONDENSER SHELLS
C
C       11-20    E10.0,    VELN      R  NOMINAL DESIGN VELOCITY THRU CONDEN-
C                                       SER TUBES (FT/SEC)
C
C       21-30    E10.0,    ODC       R  CONDENSER TUBE O.D. (INCHES)
C
C       31-40    E10.0,    GA        R  CONDENSER TUBE WALL GAGE.  MAY BE
C                                       12.  14.  16.  18.  20.  22.  24.
C
C       41-50    E10.0,    FC        R  CLEANLINESS FACTOR (DF)
C
C       51-60    E10.0,   TTD        R  TERMINAL TEMPERATURE DIFFERENCE
C                                       (DEGREES F)
C
C       61-80    2A10)    CONMAT,    R               CONDENSER TUBING MATER-
C                         CONMA2     R               IAL.  MUST START IN
C                                                    COLUMN 61.  MAY BE -
C
C                                                      ADMIRALTY
C                                                      CU-10 NI
C                                                    304 S/S WELDED
C
C.........
C.........
C
C  S1    1-5    (I5,       LFB       I  LFB AND LFE SPECIFY FIRST AND LAST
C        6-10    I5,       LFE       I  POSITION TO BE CONSIDERED IN THE
C                                       TABLE OF FANS.  DEFAULT VALUES ARE
C                                       LFB=1 AND LFE=153
C
C
```

```
C                                            DIAMETER       IFAN       LFAN
C
C                                               24.            1         27
C                                               26.           28         54
C                                               28.           55         88
C                                               30.           89        111
C                                               40.          112        135
C                                               60.          136        153
C
C          11-15     I5)       MXEXT       I  THE MAXIMUM ALLOWABLE NUMBER OF
C                                             CALLS TO SUBROUTINE XTEND FOR ANY
C                                             ONE DESIGN TEMPERATURE.
C
C                                             DEFAULT VALUE IS 10
C
C
C.........
C.........
C
C   T1     1-10     A10       FFHX       R  CONTROL VARIABLE FOR SPECIFYING
C                                           METHOD TO USE IN COMPUTING FRICTION
C                                           FACTOR.  MUST START IN COLUMN 1 .
C                                             MAY BE -
C
C                                                   PFR
C                                                   FRICHO
C
C.........
```

5.2.2 Plastic Model Input Description

```
C
C                        I N P U T   D E S C R I P T I O N
C
C                       FOR BNW1 PLASTIC CIRCULAR TOWER
*-*-*-*-*-*-*-*            D E C E M B E R  1 0   1 9 7 6            *-*-*-*-*-*-*-*
C*** *** ***
C                        IN THE DESCRIPTIONS BELOW THE SYMBOL (DF) MEANS
C                        DECIMAL FRACTION
C
C ---- ------
C                                      T
C                                      Y
C CARD                        NAME  OF P
C TYPE COLUMNS  FORMAT   VARIABLE E          DESCRIPTION AND (UNITS)
C ---- ------
C
C  A1     1-80  (8A10)    DESCR(I) R  COMMENTS OR CASE DESCRIPTION TO BE
C                                     PRINTED AT TOP OF FIRST OUTPUT PAGE.
C                                     ANY NUMBER OF TYPE A1 CARDS MAY BE
C                                     USED BUT A BLANK A1 CARD IS NOT
C                                     ALLOWED.
C
C
C.........
C.........
C
C  A2     1-80   BLANK IN ALL COLUMNS.  ONE TYPE A2 CARD ONLY
C
C.........
C.........
C
C  B1     1-10  (E10.0,   TSTAR      R  LOWEST DESIGN TEMPERATURE TO USE
C                                       (DEGREES F)
C
C        11-20   E10.0,   TEND       R  HIGHEST DESIGN TEMPERATURE TO USE
C                                       (DEGREES F)
C
C        21-30   E10.0,   VAS(1)     R  INITIAL VALUE FOR TURBINE OUTLET
C                                       TEMPERATURE (DEGREES F)
C
C        31-40   E10.0,   VAS(3)     R  INITIAL VALUE FOR CWARA, RATIO OF
C                                       MCP WATER / MCP AIR  (DF)
C
C        41-50   E10.0,   VAS(4)     R  INITIAL VALUE FOR AFRON, AIR SIDE
C                                       FRONTAL AREA (SQ FT)
C
C        51-60   E10.0,   VAS(5)     R  INITIAL VALUE FOR WLRAT, RATIO OF
C                                       HEAT EXCHANGER WIDTH/LENGTH  (DF)
C
C        61-70   E10.0,   FIXL       R  LENGTH OF HEAT EXCHANGER (FT).
C                                       IF THE ABSOLUTE VALUE OF FIXL=0.
C                                       THEN THE PROGRAM WILL CALCULATE
C                                       THE LENGTH.
C
C        71-80   E10.0)   DEEPL      R  LIMITING VALUE FOR NUMBER OF TUBES
C                                       (NUMBER OF TUBES)
C
C                                       IF THE ABSOLUTE VALUE OF DEEPL = 0.0
C                                       THEN THE PROGRAM WILL COMPUTE THE
C                                       DEPTH WITH NO CONSTRAINTS
```

```
C                                        +DEEPL IMPLIES CONSTRAINT OF GREATER
C                                          THAN OR EQUAL TO DEEPL
C
C                                        -DEEPL IMPLIES CONSTRAINT OF LESS
C                                         THAN OR EQUAL TO DEEPL.
C.........
C.........
C
C  C1    1-40  (8F5.0,  TA(I)      R  TEMPERATURES REPRESENTATIVE OF
C                                     TPER(I) FRACTION OF YEAR (DEGREES F)
C                                     C A U T I O N   -  HIGHEST TA
C                                                        MUST BE FIRST
C
C       41-80   8F5.0)  TPER(I)   R  FRACTION OF YEAR OVER WHICH TEMPERA-
C                                     TURE TA(I) IS TYPICAL. (DF)
C
C.........
C
C  C2    1-40  (8F5.0,  TA(I)      R  CONTINUATION OF TA(I) FOR I= 9 TO 16
C
C       41-80   8F5.0)  TPER(I)   R  CONTINUATION OF TPER(I)
C       NOTE - A TYPE C2 CARD IS REQUIRED EVEN IF BLANK.
C.........
C.........
C  D1    4-10  (E10.0,  PSIZE     R  BASE PLANT SIZE (MEGAWATTS)
C
C       11-20   E10.0,  TEFF      R  BASE THERMAL EFFICIENCY (DF)
C
C       21-30   E10.0,  CAPF      R  CAPACITY FACTOR  (DF)
C
C       31-40   E10.0,  FCR       R  FIXED CHARGE RATE  (DF)
C
C       41-50   E10.0,  PER       R  RATIO MAINTENANCE COST TO CAPITAL
C                                     COST  (DF)
C
C       51-60   E10.0,  CCM       R  CONSTRUCTION COST MULTIPLIER (DF)
C
C       61-70   E10.0,  ELEV      R  SITE ELEVATION ABOVE SEA LEVEL (FT)
C
C       71-80   E10.0)  ROOFL     R  ROOF LOAD (LB/SQ FT)
C
C.........
C.........
C
C  E1    1-10  (E10.0,  FCOS      R  FUEL COST (CENTS/MMBTU)
C
C       11-20   E10.0,  PWCOS     R  REPLACEMENT POWER COST (MILLS/KWH)
C
C       21-30   E10.0,  PLANC     R  POWER PLANT CONSTRUCTION COST ($/KW)
C
C       31-40   E10.0,  COSTL     R  COST OF LAND  ($/SQ FT)
C
C       41-50   E10.0,  CSSPKW    R  COST OF STEAM SUPPLY ($/KW)
C
C       51-60   E10.0,  CAPCHG    R  CAPACITY CHARGE ($/MEGAWATT)
C
C       61-70   E10.0)  HPCST     R  COST OF ELECTRIC MOTORS ($/HP)
C
C.........
C
C
```

```
C                                          INDIRECT COST FACTORS FOR -
C  E2    1-10  (E10.0,   POHBAF    R  BUNDLE ASSEMBLE AND FRAME
C
C       11-20   E10.0,   POHFAN    R  FANS
C
C       21-30   E10.0    POHLEC    R  ELECTRICAL
C
C       31-40   E10.0    POHCIR    R  CIRCULATION PIPING
C
C       41-50   E10.0    POHCND    R  CONDENSER
C
C       51-60   E10.0,   POHSTC    R  STRUCTURE
C
C       61-70  E10.0)   POHSCL     R  SCREENS AND LOUVERS
C
C.........
C.........
C  F1   1-10   (F10.0,   RBP       R  TURBINE RATING BACK PRESSURE (INCHES
C                                     OF MERCURY)
C
C       11-70   4F15.8) TPO(I)     R  COEFFICIENTS FOR THIRD ORDER POLY-
C                                     NOMIAL FOR HEAT RATE AS A FUNCTION
C                                     OF TURBINE BACK PRESSURE.
C
C.........
C.........
C
C  G1    1-10  (E10.0,   EFFP      R  EFFICIENCY OF PUMPS (DF)
C
C       11-20   E10.0)   DPRESS    R  PIPING DESIGN PRESSURE (PSI)
C                                      MAY BE
C                                              50.
C                                              70.
C                                             125.
C
C
C.........
C.........
C
C  H1    1-10  (E10.0,   VELREC    R  CONTROL VARIABLE.
C                                   =1 FOR VELOCITY RECOVERY
C                                   =0  FOR NO VELOCITY RECOVERY
C
C       11-20   E10.0,   XDEPA     R  MINIMUM AIR SIDE PRESSURE DROP THRU
C                                     HEAT EXCHANGER (LB FORCE/SQ FT)
C
C       21-30   E10.0,   TLIM      R  MAXIMUM STEAM TEMPERATURE FOR THE
C                                     TURBINE (DEGREES F).
C                                     ASSUMED 180 IF FIELD IS BLANK OR 0.
C
C       31-40   E10.0,   TOWMIN    R  THE MINIMUM ALLOWABLE NUMBER OF
C                                     TOWERS PER GROUP.
C                                     ASSUMED 1 IF NOT SPECIFIED
C
C       41-50   E10.0,   PFACT     R  PACKING FACTOR, RATIO OF CIRCULAR
C                                     TOWER ROOF AREA TO FAN SWEPT AREA
C                                     (DF)
C
C
C       51-60   E10.0,   TFIX      R  FIXED TURBINE OUTLET TEMPERATURE (F)
C
```

```
C                                            ENTRY IS IGNORED IF BLANK OR ZERO
C
C        61-70    E10.0,   RFIX        R   FIXED RANGE (F).  ENTRY IS IGNORED
C                                           IF BLANK OR ZERO
C
C        71-80    E10.0)   TOWMAX      R   MAXIMUM ALLOWABLE NUMBER OF TOWERS.
C                                           ASSUMED 9999 IF BLANK OR ZERO
C
C
C.........
C.........
C
C  H2     1-10    E10.0    GBEFF       R   FAN GEARBOX EFFICIENCY
C
C.........
C.........
C
C  I1     1-10   (E10.0,   CPM         R   PLENUM COST   ($/LB)
C
C        11-20    E10.0,   WPL         R   WEIGHT OF PLENUM MATERIAL (LB/SQ FT)
C
C        21-30    E10.0,   CLUVR       R   COST OF LOUVERS  ($/SQ FT)
C
C        31-40    E10.0,   CHAILS      R   COST OF HAIL SCREENS ($/SQ FT)
C
C        41-50    E10.0,   UCS         R   UNIT COST OF STACK $/LB
C
C        51-60    E10.0,   UWS         R   UNIT WT OF STACK  LB/CU FT
C
C        61-70    E10.0,   CVM         R   FAN RING MATERIAL UNIT COST $/LB
C
C        71-80    E10.0)   WFV         R   UNIT WT OF STRAIGHT CYLINDER LB/SQ FT
C
C
C.........
C.........
C
C  J1     1-20   (2A10,    NTUCAL      I   CONTROL VARIABLE FOR SPECIFYING THE
C                                           METHOD TO BE USED TO CALCULATE NTU.
C                                           NTUCAL MUST BEGIN IN COLUMN 1.
C                                                                CROSS FLOW
C                                                                COUNTER FLOW
C                                                                TABLE
C
C                                           CROSS FLOW  OR  COUNTER FLOW CAUSES
C                                           CALCULATION OF NTU BY EQUATIONS.
C                                           TABLE  CAUSES TABLE LOOKUP OF NTU.
C
C        21-30    F10.0,   HXNP        R   NUMBER OF PASSES THRU HEAT EXCHANGER
C
C        31-40    F10.0,   WB          R   HEADER LENGTH  (FEET)
C
C        41-50    F10.0,   NTUB        R   NUMBER OF TUBES THRU PLATE FINS
C
C        51-60    F10.0)   SS          R   SUPPORT SPACING (FT)
C
C.........
C.........
C
C  J2     1-10   (E10.0,   DESVEL      R   DESIGN WATER VELOCITY FOR PIPING
C                                           (FT/SEC)
```

```
C
C       11-20    E10.0,  REDUCE    R  MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR TOWER PIPING (INCHES)
C                                     MAY BE 6, 12, OR 18
C
C       21-30    E10.0,  QREDUCE   R  MIMIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR QUADRANT HEADERING. (INCHES)
C                                     MAY BE 6, 12, OR 18
C
C       31-40    E10.0)  DIST      R  DISTANCE FROM TOWER TO CONDENSER
C                                     ROOM (FT)
C
C.........
C.........
C
C  K1    1-10   (F10.0,  XW        R  TUBE TRANSVERSE PITCH (NORMAL TO AIR
C                                     FLOW)  (INCHES)
C
C       11-20    F10.0,  XD        R  TUBE PITCH IN DIRECTION OF AIR FLOW
C                                     (INCHES)
C
C       21-30    F10.0,  XDG       R  TUBE DIAGONAL PITCH (INCHES)
C
C       31-40    A10,    TUBARG    R TUBE ARRANGEMENT. MUST BEGIN IN
C                                     COLUMN 1.   MAY BE-
C                                                      STAGGERED
C                                                      INLINE
C
C                                     ASSUMED TO BE STAGGERED IF OMITTED
C
C       41-50    A10,    HCCALC    R  CONTROL VARIABLE TO SELECT METHOD
C                                     TO BE USED IN CALCULATING FRICTION
C                                     FACTOR AND HEAT TRANSFER COEFFICIENT
C                                     FOR AIR FLOWING ACROSS STAGGERED
C                                     TUBES.   MAY BE 1 OR 2
C                                     (ASSUMED 2 IF BLANK)
C
C                                     HCCALC=1 SPECIFIES MCADAMS RELATION-
C                                              SHIP FOR HEAT TRFR AND FOR
C                                              FRICTION FACTOR
C
C                                     HCCALC=2 SPECIFIES MCADAMS RELATION-
C                                              SHIP FOR FRICTION FACTOR
C                                              AND ZUKAUSKAS RELATIONSHIP
C                                              FOR HEAT TRANSFER COEFF.
C
C                                    HCCALC IS IGNORED IF TUBARG = INLINE
C
C       51-60    F10.0)  DANGLE    R  BASE ANGLE FOR DELTA HEAT EXCHANGER
C                                     (DEGREES)
C
C.........
C.........
C
C  L1    1-10   (F10.0, ODL        R  HX LINER OUTSIDE DIAMETER (INCHES)
C
C       11-20    F10.0, DI         R  HX LINER INSIDE DIAMETER (INCHES)
C
C       21-30    F10.0, GAGLIN     R  GAGE OF HX LINER TUBE  (MAY BE
C                                     22.  20.  19.  18.  17.  16.  15.
C                                     14.5 14.  13.  12.  11.  10.     )
```

```
C
C
C       31-50   2A10,    TUBMAT,  R  LINER MATERIAL. MUST BEGIN IN COL 31
C                        XTUBMA   R
C                                    MAY BE -
C                                                 POLYETHYLENE
C                                                 POLYPROPYLENE
C                                                 POLYVINYLCHLORIDE
C
C
C       51-60    F10.0) CONL      R  THERMAL CONDUCTIVITY OF LINER MATER-
C                                    IAL  (BTU/(HR SQFT DEG F/ FT) )
C.........
C.........
C
C  M1    1-20  (2A10,    SPAMAT   R  SPACER MATERIAL.  MUST BEGIN IN CARD
C                                   COLUMN 1.   MAY BE
C                                                     POLYVINYLCHLORIDE
C                                                     POLYCARBONATE
C
C       21-30   F10.0,   CSEALP   R  SEAL PLATE COST ($/LB)
C
C       31-40   F10.0,   TSEALP   R  SEAL PLATE THICKNESS (IN)
C
C       41-50   F10.0,   CELAST   R  ELASTOMER COST ($/LB)
C
C       51-60   F10.0,   TELAST   R  ELASTOMER LAYER THICKNESS (IN)
C
C       61-70   F10.0,   ROESEL   R  SEAL PLATE DENSITY (LB/CU IN)
C
C       71-80   F10.0,   ROEELA   R ELASTOMER DENSITY (LB/CU IN)
C
C.........
C.........
C
C  N1    1-10  (E10.0,    CASTC   R  FIXED CASTING COST FOR SPACERS
C                                    ($/SPACER)
C
C       11-20   E10.0,    EPREPC  R  END PREPARATION COST ($/TUBE)
C
C       21-30   E10.0)    CFS     R FRAME UNIT COST ($/LB)
C
C.........
C.........
C
C  P1    1-20  (2A10,    HEDTYP,  R  HEADER TYPE.  MUST START IN COLUMN 1.
C                        XHEDTY   R
C                                    MAY BE -
C                                                 WELD REMOVABLE
C                                                 WELD PLUG
C                                                 FORM REMOVABLE
C                                                 FORM PLUG
C
C       21-30   F10.0,    HEDMAT  R  HEADER MATERIAL. MUST START IN
C                                    COLUMN 21. MAY BE -
C                                                        ALUMINUM
C                                                        STEEL
C
C       31-40   F10.0)    TW      R  THICKNESS OF HEADER MATERIAL, INCHES
C
C.........
```

```
C.........
C
C  Q1     1-10  (E10.0,    CRJ       R  COST OF ROLLED JOINT, TUBE TO HEADER
C                                       ($/TUBE)
C
C        11-20   E10.0,    CWJ       R  COST OF WELDED JOINT, TUBE TO HEADER
C                                       ($/TUBE)
C
C        21-30   E10.0,    CHH       R  TUBE AND PLUG HOLE PREPARATION COST
C                                       ($/HOLE)
C
C        31-40   E10.0,    CN        R  NOZZEL AND ATTACHING COST ($/HOLE)
C
C        41-50   E10.0,    CMW       R  COST OF HEADER MACHINING AND WELDING
C                                       ($/FT)
C
C        51-60   E10.0,    CST       R  STRUCTURAL STEEL COST  ($/LB)
C
C        61-70   E10.0)    CBJ       R  COST OF BOLTED HEADER JOINT ($/FT)
C
C.........
C.........
C
C  R1     1-10  (E10.0,    XNS       R  NUMBER OF CONDENSER SHELLS
C
C        11-20   E10.0,    VELN      R  NOMINAL DESIGN VELOCITY THRU CONDEN-
C                                       SER TUBES (FT/SEC)
C
C        21-30   E10.0,    ODC       R  CONDENSER TUBE O.D. (INCHES)
C
C        31-40   E10.0,    GA        R  CONDENSER TUBE WALL GAGE.  MAY BE
C                                       12.  14.  16.  18.  20.  22.  24.
C
C        41-50   E10.0,    FC        R  CLEANLINESS FACTOR (DF)
C
C        51-60   E10.0,    TTD       R  TERMINAL TEMPERATURE DIFFERENCE
C                                       (DEGREES F)
C
C        61-80   2A10)    CONMAT,    R              CONDENSER TUBING MATER-
C                         CONMA2     R              IAL.  MUST START IN
C                                                   COLUMN 61.  MAY BE -
C
C                                                     ADMIRALTY
C                                                     CU-10 NI
C                                                   304 S/S WELDED
C
C
C.........
C.........
C
C  S1     1-5   (I5,       LFB       I  LFB AND LFE SPECIFY FIRST AND LAST
C         6-10   I5,       LFE       I  POSITION TO BE CONSIDERED IN THE
C                                       TABLE OF FANS.  DEFAULT VALUES ARE
C                                       LFB=1 AND LFE=153
C
C
C                                       DIAMETER       IFAN       LFAN
C
C                                           24.          1         27
C                                           26.         28         54
C                                           28.         55         88
```

```
C                                          30.        89        111
C                                          40.       112        135
C                                          60.       136        153
C
C       11-15   I5      MXEXT     I  THE MAXIMUM ALLOWABLE NUMBER OF
C                                    CALLS TO SUBROUTINE XTEND FOR ANY
C                                    ONE DESIGN TEMPERATURE.
C
C                                    DEFAULT VALUE IS 10
C
C.........
C.........
C
C  U1    2-72  NAMELIST INPUT CARD.      NAMELIST NAME IS  'READIN'
C              AT LEAST 1 TYPE U1 CARD IS REQUIRED.
C              IN ITS SIMPLEST FORM THE U1 CARD IS
C                                                 $READIN NUCASE=0 $END
C              BEGINNING IN CARD COLUMN 2.
C
C              VARIABLES IN NAMELIST 'READIN' ARE-
C
C              DANGLE,  DESVEL,  DIST,    DI,      FIXL,    LFB,
C              LFE,     NUCASE,  ODL,     PFACT,   QREDUCE, REDUCE,
C              RFIX,    TFIX,    TLIM,    TOWMAX,  TOWMIN,  VELREC,
C              XD,      XDG,     XW
C
C              AFTER THE INITIAL CASE IS DESCRIBED WITH CARD TYPES A1-U1
C              THEN ANY NUMBER OF TYPE U1 CARDS MAY BE USED TO REDEFINE
C              ANY OF THE VARIABLES IN THE NAMELIST FOR ADDITIONAL CASES
C
C              NUCASE   IS A CONTROL VARIABLE TO SPECIFY HOW THE DATA
C                       FOR THE NEXT CASE ARE TO BE READ.
C
C              NUCASE = 0        SPECIFIES THAT THE DATA FOR THE NEXT
C                                CASE ARE TO BE READ FROM A COMPLETE SET
C                                OF A1 THRU U1 CARDS
C
C              NUCASE = 1        SPECIFIES THAT DATA FOR THE NEXT CASE
C                                ARE TO BE READ FROM NAMELIST CARDS
C
```

5.2.3 NH_3 Model Input Description

```
C                        I N P U T   D E S C R I P T I O N
C                      FOR BNW1 METAL FIN ROUND DCT AMMONIA
C*-*-*-*-*-*-*-*            D E C E M B E R  1 0    1 9 7 6              *-*-*-*-*-*-*
C
C*** *** ***
C                        IN THE DESCRIPTIONS BELOW THE SYMBOL (DF) MEANS
C                        DECIMAL FRACTION
C
C ---- -------
C                                        T
C                                        Y
C CARD                         NAME   OF P
C TYPE COLUMNS  FORMAT   VARIABLE E          DESCRIPTION AND (UNITS)
C ---- -------
C
C  A1     1-80   (8A10     DESCR(I) R  COMMENTS OR CASE DESCRIPTION TO BE
C                                      PRINTED AT TOP OF FIRST OUTPUT PAGE.
C                                      ANY NUMBER OF TYPE A1 CARDS MAY BE
C                                      USED BUT A BLANK A1 CARD IS NOT
C                                      ALLOWED.
C
C
C.........
C
C  A2     1-80   BLANK IN ALL COLUMNS.  ONE TYPE A2 CARD ONLY
C
C.........
C.........
C
C  B1     1-10   (E10.0,   TSTAR      R  LOWEST DESIGN TEMPERATURE TO USE
C                                        (DEGREES F)
C
C        11-20   E10.0,    TEND       R  HIGHEST DESIGN TEMPERATURE TO USE
C                                        (DEGREES F)
C
C        21-30   E10.0,    VAS(1)     R  INITIAL VALUE FOR TURBINE OUTLET
C                                        TEMPERATURE (DEGREES F)
C
C        31-40   E10.0,    VAS(3)     R  INITIAL VALUE FOR TTD1, TERMINAL
C                                        TEMPERATURE DIFFERENCE (DEGREES F)
C
C        41-50   E10.0,    VAS(4)     R  INITIAL VALUE FOR AFRON, AIR SIDE
C                                        FRONTAL AREA (SQ FT)
C
C        51-60   E10.0,    VAS(5)     R  INITIAL VALUE FOR WLRAT, RATIO OF
C                                        HEAT EXCHANGER WIDTH/LENGTH  (DF)
C
C        61-70   E10.0,    FIXL       R  LENGTH OF HEAT EXCHANGER (FT).
C                                        IF THE ABSOLUTE VALUE OF FIXL=0.
C                                        THEN THE PROGRAM WILL CALCULATE
C                                        THE LENGTH.
C
C        71-75   F5.0,     DEEPL      R  LIMITING VALUE FOR NUMBER OF TUBES
C                                        IN DEPTH FOR HEAT EXCHANGER.
C                                       (NUMBER OF TUBES)
```

```
C
C                                             IF THE ABSOLUTE VALUE OF DEEPL = 0.0
C                                             THEN THE PROGRAM WILL COMPUTE THE
C                                             DEPTH WITH NO CONSTRAINTS
C
C
C                                             IF POMDPL IS BLANK OR ZERO THEN A
C                                             POSITIVE VALUE FOR DEEPL IMPLIES A
C                                             CONSTRAINT OF EQUAL TO OR GREATER
C                                             THAN DEEPL, WHILE A NEGATIVE VALUE
C                                             FOR DEEPL IMPLIES A CONSTRAINT OF
C                                             EQUAL TO OR LESS THAN DEEPL.
C
C        76-80   F5.0)    POMDPL     R   TOLERANCE ON DEEPL. NOTE THAT A ZERO
C                                             OR BLANK IS NOT INTERPRETED AS ZERO
C                                             TOLERANCE. (SEE DEFINITION OF DEEPL)
C
C.........
C.........
C
C   C1    1-40   (8F5.0,  TA(I)      R   TEMPERATURES REPRESENTATIVE OF
C                                             TPER(I) FRACTION OF YEAR (DEGREES F)
C
C        41-80   8F5.0)   TPER(I)    R   FRACTION OF YEAR OVER WHICH TEMPERA-
C                                             TURE TA(I) IS TYPICAL. (DF)
C                                             C A U T I O N   -  HIGHEST TA
C                                                                MUST BE FIRST
C
C.........
C
C   C2    1-40   (8F5.0,  TA(I)      R   CONTINUATION OF TA(I) FOR I= 9 TO 16
C
C        41-80   8F5.0)   TPER(I)    R CONTINUATION OF TPER(I)
C
C        NOTE - A TYPE C2 CARD IS REQUIRED EVEN IF BLANK.
C
C.........
C.........
C   D1    1-10   (E10.0,  PSIZE      R   BASE PLANT SIZE (MEGAWATTS)
C
C        11-20   E10.0,   TEFF       R   BASE THERMAL EFFICIENCY (DF)
C
C        21-30   E10.0,   CAPF       R   CAPACITY FACTOR  (DF)
C
C        31-40   E10.0,   FCR        R   FIXED CHARGE RATE  (DF)
C
C        41-50   E10.0,   PER        R   RATIO MAINTENANCE COST TO CAPITAL
C                                             COST  (DF)
C
C        51-60   E10.0,   CCM        R   CONSTRUCTION COST MULTIPLIER (DF)
C
C        61-70   E10.0,   ELEV       R   SI E ELEVATION ABOVE SEA LEVEL (FT)
C
C        71-80   E10.0)   ROOFL      R   ROOF LOAD (LB/SQ FT)
C
C.........
C.........
C
C   E1    1-10   (E10.0,  FCOS       R   FUEL COST (CENTS/MMBTU)
C
C        11-20   E10.0,   PWCOS      R   REPLACEMENT POWER COST (MILLS/KWH)
```

```
C       21-30   E10.0,  PLANC    R  POWER PLANT CONSTRUCTION COST ($/KW)
C
C       31-40   E10.0,  COSTL    R  COST OF LAND  ($/SQ FT)
C
C       41-50   E10.0,  CSSPKW   R  COST OF STEAM SUPPLY ($/KW)
C
C       51-60   E10.0,  CAPCHG   R  CAPACITY CHARGE ($/MEGAWATT)
C
C       61-70   E10.0,  HPCST    R  COST OF ELECTRIC MOTORS ($/HP)
C
C       71-80   E10.0)  CTURB    R  ADDITIONAL TURBINE COST BECAUSE OF
C                                   NUCLEAR POWER PLANT ($/KW)
C
C..........
C
C
C                                          INDIRECT COST FACTORS FOR -
C  E2    1-10  (E10.0,  POHBAF   R  BUNDLE ASSEMBLE AND FRAME
C
C       11-20   E10.0,  POHFAN   R  FANS
C
C       21-30   E10.0   POHLEC   R  ELECTRICAL
C
C       31-40   E10.0   POHCIR   R  CIRCULATION PIPING
C
C       41-50   E10.0   POHCND   R  CONDENSER
C
C       51-60   E10.0,  POHSTC   R  STRUCTURE
C
C       61-70  E10.0)  POHSCL    R  SCREENS AND LOUVERS
C
C..........
C
C  F1    1-10  (F10.0,  RBP      R  TURBINE RATING BACK PRESSURE (INCHES
C                                   OF MERCURY)
C
C       11-70   4F15.3) TPO(I)   R  COEFFICIENTS FOR THIRD ORDER POLY-
C                                   NOMIAL FOR HEAT RATE AS A FUNCTION
C                                   OF TURBINE BACK PRESSURE.
C
C..........
C..........
C
C  G1    1-10   E10.0   EFFP     R  EFFICIENCY OF PUMPS (DF)
C
C..........
C..........
C
C  H1    1-10  (E10.0,  VELREC   R  CONTROL VARIABLE.
C                                  =1 FOR VELOCITY RECOVERY
C                                  =0  FOR NO VELOCITY RECOVERY
C
C       11-20   E10.0,  XDEPA    R  MINIMUM AIR SIDE PRESSURE DROP THRU
C                                   HEAT EXCHANGER (LB FORCE/SQ FT)
C
C       21-30   E10.0,  TLIM     R  MAXIMUM STEAM TEMPERATURE FOR THE
C                                   TURBINE (DEGREES F).
C                                   ASSUMED 180 IF FIELD IS BLANK OR 0.
C
C       31-40   E10.0,  TOWMIN   R  THE MINIMUM ALLOWABLE NUMBER OF
```

```
C                                              TOWERS PER GROUP.
C                                              ASSUMED 1 IF NOT SPECIFIED
C
C        41-50    E10.0,   PFACT      R   PACKING FACTOR, RATIO OF CIRCULAR
C                                         TOWER ROOF AREA TO FAN SWEPT AREA
C                                         (DF)
C
C        51-60    E10.0,   TFIX       R   FIXED TURBINE OUTLET TEMPERATURE (F)
C                                         ENTRY IS IGNORED IF BLANK OR ZERO
C
C
C        61-70    F10.0)   TOWMAX     R   THE MAXIMUM ALLOWABLE NUMBER OF
C                                         TOWERS PER GROUP.  ASSUMED 9999 IF
C                                         NOT SPECIFIED.
C
C..........
C
C  H2     1-10    E10.0    GBEFF      R   FAN GEARBOX EFFICIENCY
C
C..........
C..........
C
C  I1     1-10   (E10.0,   CPM        R   PLENUM COST   ($/LB)
C
C        11-20    E10.0,   WPL        R   WEIGHT OF PLENUM MATERIAL (LB/SQ FT)
C
C        21-30    E10.0,   CLUVR      R   COST OF LOUVERS   ($/SQ FT)
C
C        31-40    E10.0,   CHAILS     R   COST OF HAIL SCREENS ($/SQ FT)
C
C        41-50    E10.0,   UCS        R   UNIT COST OF STACK $/LB
C
C        51-60    E10.0,   UWS        R   UNIT WT OF STACK  LB/CU FT
C
C        61-70    E10.0,   CVM        R   FAN RING MATERIAL UNIT COST $/LB
C
C        71-80    E10.0)   WFV        R   UNIT WT OF STRAIGHT CYLINDER LB/SQ FT
C
C..........
C..........
C
C  J1     1-20   (2A10,    NTUCAL     I   CONTROL VARIABLE FOR SPECIFYING THE
C                                         METHOD TO BE USED TO CALCULATE NTU.
C                                         NTUCAL MUST BEGIN IN COLUMN 1.
C                                                              CROSS FLOW
C                                                              COUNTER FLOW
C                                                              TABLE
C
C                                         CROSS FLOW  OR  COUNTER FLOW CAUSES
C                                         CALCULATION OF NTU BY EQUATIONS.
C                                         TABLE  CAUSES TABLE LOOKUP OF NTU.
C
C        21-30    F10.0,   HXNP       R   NUMBER OF PASSES THRU HEAT EXCHANGER
C
C        31-40    F10.0,   WB         R   HEADER LENGTH  (FEET)
C
C        41-50    F10.0,   NTUB       R   NUMBER OF TUBES THRU PLATE FINS
C
C        51-60    F10.0,   SS         R   SUPPORT SPACING (FT)
C
C        61-70    F10.0)   ANGLE      R   HEAT EXCHANGER ANGLE (DEGREES)
```

```
C
C.........
C
C  J2     1-10  (F10.0,  DESVEL    R  DESIGN VELOCITY (LIQUID) FOR PIPING
C                                     (FT/SEC)
C
C        11-20   F10.0,  REDUCE    R  MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR LIQUID PHASE TOWER PIPING (INCH-
C                                     ES)
C                                     MAY BE 6, 12, OR 18
C
C        21-30   F10.0,  GREDUCE   R  MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR LIQUID PHASE QUADRANT HEADERING
C                                     (INCHES)
C                                     MAY BE 6, 12, OR 18
C
C        31-40   F10.0)  DIST      R  DISTANCE FROM TOWER TO CONDENSER
C                                     ROOM (FT)
C
C.........
C
C  J3     1-10  (F10.0,  DESVELV   R  DESIGN VELOCITY (VAPOR) FOR PIPING.
C                                     (FT/SEC)
C
C        11-20   F10.0,  REDUCEV   R  MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR VAPOR PHASE TOWER PIPING
C                                     (INCHES)
C                                     MAY BE 6, 12, OR 18
C
C        21-30   F10.0)  GREDUCV   R  MINIMUM STEP CHANGE IN PIPE DIAMETER
C                                     FOR VAPOR PHASE QUADRANT HEADERING
C                                     (INCHES)
C                                     MAY BE 6, 12, OR 18
C
C
C.........
C.........
C
C  K1     1-10  (E10.0,  XW        R  TUBE TRANSVERSE PITCH (NORMAL TO AIR
C                                     FLOW)  (INCHES)
C
C        11-20   E10.0,  XD        R  TUBE PITCH IN DIRECTION OF AIR FLOW
C                                     (INCHES)
C
C        21-30   E10.0,  XDG       R  TUBE DIAGONAL PITCH (INCHES)
C
C        31-40   E10.0,  SF        R  FIN SPACING (INCHES)
C
C        41-50   E10.0,  DFIN      R  FIN DIAMETER (INCH(ES)
C
C        51-60   E10.0,  THFIN     R  FIN THICKNESS (INCHES)
C
C        61-70   E10.0)  DANGLE    R  BASE ANGLE FOR DELTA HEAT EXCHANGER
C                                     (DEGREES)
C.........
C.........
C
C  L1     1-10  (F10.0, ODL        R  HX LINER OUTSIDE DIAMETER (INCHES)
C        11-20   F10.0, DI         R  HX LINER INSIDE DIAMETER (INCHES)
C
C        21-30   F10.0, GAGLIN     R  GAGE OF HX LINER TUBE  (MAY BE
```

```
C                                         22.  20.  19.  18.  17.  16.  15.
C                                         14.5 14.  13.  12.  11.  10.     )
C
C        31-50   2A10,   TUBMAT,  R  LINER MATERIAL.  MUST BEGIN IN COL 3
C                        XTUBMA   R
C
C                                    MAY BE -
C                                             ADMIRALTY
C                                             COPPER
C                                             CU-10 NI
C                                             CU-30 NI
C                                             ALUMINUM
C                                             STEEL WELDED
C                                             STEEL SEAMLESS
C                                             WELDED SST
C
C        51-60   F10.0)  CONL     R  THERMAL CONDUCTIVITY OF LINER MATER-
C                                    IAL  (BTU/(HR SQFT DEG F/ FT) )
C
C.........
C.........
C
C   M1   1-20   (2A10    FINTYP   R  TYPE OF FIN. MUST START IN COLUMN 1.
C                        XFINTY
C
C                                    MAY BE -
C                                             STRAIGHT FIN
C                                             SINGLE FOOT
C                                             DOUBLE FOOT
C                                             EMBEDDED
C                                             EXTRUDED
C                                             PLATE
C                                             NO LINER
C
C                                    IF NO LINER THEN FIN MATERIAL
C                                    MUST BE ALUMINUM
C
C        21-30   A10,     FINMAT  R  FIN MATERIAL.  MUST START IN COL. 21
C
C                                    MAY BE -
C                                             ALUMINUM
C                                             STEEL
C
C        31-40   F10.0)   CONF    R  THERMAL CONDUCTIVITY OF FIN MATERIAL
C                                    (BTU/(HR SQFT DEG F/FT))
C
C.........
C.........
C
C   N1   1-10   (E10.0,   CFB     R  FIN BONDING COST ($/SQ FT)
C
C        11-20   E10.0,   COATC   R  FIN PROTECTIVE COATING COST ($/SQFT)
C
C        21-30   E10.0,   ZINCC   R  ZINC COST FOR SPACERS ($/LB)
C
C        31-40   E10.0,   CASTC   R  FIXED CASTING COST FOR SPACERS
C                                    ($/SPACER)
C
C        41-50   E10.0)   EPREPC  R  END PREPARATION COST ($/TUBE)
C
C.........
```

```
C.........
C
C  P1    1-20   (2A10,    HEDTYP,  R  HEADER TYPE.  MUST START IN COLUMN 1.
C                         XHEDTY   R
C                                     MAY BE -
C                                                WELD REMOVABLE
C                                                WELD PLUG
C                                                FORM REMOVABLE
C                                                FORM PLUG
C
C        21-30   A10,      HEDMAT  R  HEADER MATERIAL. MUST START IN
C                                     COLUMN 21. MAY BE -
C                                                      ALUMINUM
C                                                      STEEL
C
C        31-40   F10.0)    TW      R   THICKNESS OF HEADER MATERIAL
C                                     (INCHES)
C
C.........
C.........
C
C  Q1    1-10   (E10.0,   CRJ      R  COST OF ROLLED JOINT, TUBE TO HEADER
C                                     ($/TUBE)
C
C        11-20   E10.0,   CWJ      R  COST OF WELDED JOINT, TUBE TO HEADER
C                                     ($/TUBE)
C
C        21-30   E10.0,   CHH      R  TUBE AND PLUG HOLE PREPARATION COST
C                                     ($/HOLE)
C
C        31-40   E10.0,   CN       R  NOZZEL AND ATTACHING COST ($/HOLE)
C
C        41-50   E10.0,   CMW      R  COST OF HEADER MACHINING AND WELDING
C                                     ($/FT)
C
C        51-60   E10.0,   CST      R  STRUCTURAL STEEL COST  ($/LB)
C
C        61-70   E10.0)   CBJ      R  COST OF BOLTED HEADER JOINT ($/FT)
C
C.........
C.........
C
C  R1    1-10   (E10.0,   XNS      R  NUMBER OF CONDENSER SHELLS
C
C        11-20   E10.0,   VELN     R  NOMINAL DESIGN VELOCITY THRU CONDEN-
C                                     SER TUBES (FT/SEC)
C
C        21-30   E10.0,   ODC      R  CONDENSER TUBE O.D. (INCHES)
C
C        31-40   E10.0,   GA       R  CONDENSER TUBE WALL GAGE.  MAY BE
C                                     12.  14.  16.  18.  20.  22.  24.
C
C        41-50   E10.0,   TKCT     R  THERMAL CONDUCTIVITY OF CONDENSER
C                                     TUBING
C                                     (BTU/(HR SQFT DEG F/FT))
C
C        51-60   E10.0,   TTD2QE   R  TERMINAL TEMPERATURE DIFFERENCE
C                                     ESTIMATE
C                                     (DEGREES F)
C
C        61-80   2A10)   CONMAT,   R             CONDENSER TUBING MATER-
```

```
C                         CONMA2   R                   IAL.  MUST START IN
C                                                      COLUMN 61.  MAY BE -
C
C                                                        ADMIRALTY
C                                                        CU-10 NI
C                                                      304 S/S WELDED
C
C.........
C
C  R2     1-10   (E10.0,  XNP       R  NUMBER OF PASSES THRU CONDENSER
C
C         11-2    E10.0,  TLA       R  CONDENSER TUBING LENGTH (FEET)
C
C         21-3    E10.0)  XQUALY    R  AMMONIA VAPOR QUALITY OUT OF
C                                      CONDENSER.
C
C.........
C
C  R3     1-5    (I5)     LINOR     R  CONTROL VARIABLE
C                                      =1 FOR LINDE TUBING
C                                      =0 FOR BARE TUBES
C.........
C.........
C
C  S1     1-5    (I5,      LFB      I  LFB AND LFE SPECIFY FIRST AND LAST
C         6-10    I5,      LFE      I  POSITION TO BE CONSIDERED IN THE
C                                      TABLE OF FANS.  DEFAULT VALUES ARE
C                                      LFB=1 AND LFE=153
C
C
C                                      DIAMETER        IFAN       LFAN
C
C                                          24.            1         27
C                                          26.           28         54
C                                          28.           55         88
C                                          30.           89        111
C                                          40.          112        135
C                                          60.          136        153
C
C        11-15    I5)     MXEXT     I  THE MAXIMUM ALLOWABLE NUMBER OF
C                                      CALLS TO SUBROUTINE XTEND FOR ANY
C                                      ONE DESIGN TEMPERATURE.
C
C                                      DEFAULT VALUE IS 10
C
C
C.........
C.........
C
C  T1     1-10    A10      FFHX     R  CONTROL VARIABLE FOR SPECIFYING
C                                    METHOD TO USE IN COMPUTING FRICTION
C                                    FACTOR.  MUST START IN COLUMN 1 .
C                                     MAY BE -
C
C                                              PFR
C                                              FRICHO
C
```

5.3 OUTPUT

Complete output from typical runs of each model are presented and described in the following subsections.

5.3.1 SOA Model Output

Complete output from a typical run of the SOA model is shown in Figure 5-4. The circled numbers on the figure correspond to the numbers in parentheses in this section. The output report is divided into the following sections:

(100) through (300) Input Summary
(400) through (500) Plant scaling for increased steam, and for fans and pumps
(600) through (700) Heat Exchanger
(800) Piping
(900) Condenser
(1000) Fans
(1100) through (1200) Plant performance as a function of ambient temperature
(1300) Cost summary

(100) Output printed by subroutine SETUP.

This line begins a card-by-card summary of input data. Any cards preceding the first blank card are printed as title or description cards. After the first blank card the values are printed as they are read, and the variable name is printed immediately below the value. The line tags A1, B1, etc. correspond to the Card Type designations used in the Input Description (see Section 5.2.1). Variable names printed here are defined in Section 5.2.1. Every input variable read by the program appears in this summary.

(200) through (230) Output printed by subroutine INPSUM

Some of the more frequently changed input parameters are summarized here with more descriptive labels than are given in the preceding summary.

(205) The options summarized here are those selected by the variables FIXL, DEEPL AND POMDPL on Card Type B1; VELREC and TLIM on Card Type H1; and NTUCAL on Card Type J1.

(210) The starting point in the search for an optimum is specified by T1, CWARA, AFRON, and WLRAT. These correspond to the values of VAS(1), VAS(3), VAS(4) and VAS(5) on Card Type B1. (The initial value for the temperature range of the water is determined internally by the code.)

(220) Input data pertaining to the turbine are summarized here.

(230) Unit costs and costing factors from Card Types D1 and E1 are reported here.

(300) Output printed by Main Program.

TD is the design temperature (°F) used in calculations for this case. The design temperature of the plant is the air temperature at which the power plant and cooling towers are designed to give a net power of PSIZE.

(400) Output printed by subroutine OUT2.

This report is printed upon return from subroutine SSCALE, after the steam supply system is scaled to allow for increased heat rate at the rating back pressure. The term HRFAC1 is the heat rate factor at the rating back pressure, RBP.

(500) Output printed by subroutine OUT4.

Values reported here are intermediate results after the plant is scaled for increased steam, but before it is scaled for fan and pump power requirements. HRFAC2 is the heat rate factor at the indicated design backpressure. PLANC2 is the plant cost; PSIZ12 is the plant size needed at rating back pressure to provide the specified plant power output (PSIZE on Card Type D1) at design back pressure. PSIZ12 = PSIZE*(HRFAC2/HRFAC1). SPBP is the cost differential incurred in scaling the plant to allow for the difference between design conditions and base plant conditions (conventional turbine at rated back pressure).

(550) Output printed by subroutine OUT4.

The information reported here is for the plant after scaling to provide for additional power to drive fans and pumps. The scaling factor, S.F., is the ratio PSIZE/(PSIZE power required by fans and pumps). PSIZ13 corresponds to PSIZ12 above, but now the fan and pump power requirements are included. PSIZ13 = (PSIZ12) x (S.F.). The power PTOTAL is the size of the scaled plant. PTOTAL = (PSIZE) x (S.F.). PLANC3 is the plant cost per kilowatt, excluding fan and pump capital costs, at design conditions. SPC is the plant cost excluding fans and pumps. SPCD is the differential cost of the plant due to the use of a dry cooling tower as a heat rejection system. It is the difference in cost between the plant using a conventional turbine at rated back pressure and the plant using a nonconventional turbine at plant design conditions.

(600) Output printed by subroutine RPTHXD.

This report summarizes the input data pertaining to heat exchanger design, materials, and unit costs. Values are entered on Card Type K1, L1, M1, P1, and Q1.

(700) through (760) Output printed by subroutine RPTHXC.

The heat exchanger design, operating conditions, performance characteristics and costs are reported here.

(710) Fluid flow and heat transfer characteristics are summarized. The term CWARA is the ratio of MC_p of water to MC_p of air. CWARA corresponds to VAS(3) on Card Type B1.

(720) Heat exchanger geometry terms reported here are defined as

$$\text{AFTR} = \frac{\text{Fin Area}}{\text{Total Area for Heat Transfer}}$$

$$\text{ALPHA} = \frac{\text{Heat Transfer Area (ft}^2\text{)}}{\text{Unit Volume of Heat Exchanger (ft}^3\text{)}}$$

$$\text{SIGMA} = \frac{\text{Free Flow Area}}{\text{Frontal Area}}$$

$$\frac{\text{FITCO}}{\text{ATOT}} = \frac{\text{Finned Tube Cost (\$)}}{\text{Heat Transfer Area (ft}^2\text{)}}$$

(730) Weights of major dry cooling tower components are reported.

(740) The option selected for NTU calculation is shown here, based on NTUCAL input on Card Type J1.

(750) Fluid flow rates and heat exchanger geometry previously printed repeated here for convenience.

(760) An itemized list of costs associated with fabricating, shipping and installing the heat exchanger is printed.

(800) Output printed by subroutine RPRT1.

Piping costs are itemized separately for circulation piping, quadrant piping, and distribution piping. Pump and piping costs are reported in thousands of dollars ($ *10**3). For a complete description of the pump and piping, see Appendixes B and D.

(900) Output printed by subroutine RPRT2.

Condenser design and costing are reported in this section.

(1000) Output printed by subroutine RPTFAN.

Fan geometry, costs, and design conditions are reported here. Design condition values reported in the column labeled ADJUSTED are for air reduced to standard conditions. The space labeled DRIVE SYSTEM is reserved for future use. In its present form, the model assumes a spiral bevel drive system only.

(1100) Output printed by subroutine VARIT.

System performance and costs are reported for one year of operation at times and temperatures specified on Card Types C1 and C2. Abbreviations used in column headings are:

HX - heat exchanger
ITD - initial temperature difference
LMTD - log mean temperature difference
CAP - capacity
TEMP - temperature
PCT - percent
TTDP - terminal temperature difference (condenser).

(1200) Output printed by main program.

FINAL INCREMENTAL COST is the total of the entries in PORTION OF INCREMENTAL POWER COST column. Units are mills/kWh. Capacity charge of gas turbine is computed by the expression

CAPACITY CHARGE = (CAPCHG)(PSIZE-PGEN),

where

PGEN is the power generated at highest ambient temperature;

(PSIZE-PGEN) is the replacement capacity (kW);

CAPCHG is the cost of replacement capacity ($/kW);

PSIZE and CAPCHG are input values on Card Types D1 and E1.

(1300) Output printed by subroutine SUMCOS.

Major cost components are summarized here. Costs reported under the heading CAPITAL COST SUMMARY (1340) are in dollars; those reported under UNIT ENERGY COST SUMMARY are in mills/kWh. The ENERGY PENALTY (1350) is the summation of the products of two columns in the performance cost table (1100), that is, (COST TO REPLACE LOST CAP) x (PCT TIME AT AMBIENT TEMP). Similarly, the ADDITIONAL BASE PLANT FUEL (1360) is the summation of the products (CHANGE IN FUEL COST) x (PCT TIME AT AMBIENT TEMPERATURE). OPERATION AND MAINTENANCE unit cost (1370) is based on the cooling system capital cost and the input variable PER (Card Type D1), which is simply the ratio of maintenance cost to capital cost. The calculation is made in subroutine SCALP. The final TOTAL (1380) is the same as the FINAL INCREMENTAL COST reported in the preceding table (1200).

```
(100)  BNWL DRY COOLING TOWER COST MODEL --  METAL FIN TUBE, CIRCULAR TOWER VERSION. THIS RUN MADE  03/17/77     13.40.02
                   ***
               (A1)***            EXAMPLE RUN - METAL FIN SOA VERSION
                   ***

(B1)  50.000000     50.000000     95.000000     1.0000000     300000.00     20.000000     80.000000     -0.           -0.
       TSTAR          TEND          VAS(1)        VAS(3)        VAS(4)        VAS(5)        FIXL         DEEPL        PUMDPL

(C1)  100.0    95.0    85.0    75.0    65.0    55.0    45.0    35.0   .0015   .0100   .0285   .0822   .0913   .1096   .1119   .5650
      TA(1)   TA(2)   TA(3)   TA(4)   TA(5)   TA(6)   TA(7)   TA(8)  TPER(1) TPER(2) TPER(3) TPER(4) TPER(5)TPER(6) TPER(7) TPER(8)

(C2)  -0.0    -0.0    -0.0    -0.0    -0.0    -0.0    -0.0    -0.0  -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000
      TA(9)  TA(10)  TA(11)  TA(12)  TA(13)  TA(14)  TA(15)  TA(16) TPER(9) TPER(10)TPER(11)TPER(12TPER(13)TPER(14)TPER(15)TPER(16)

(D1)  1000.0000     .40000000     .65000000     .17400000     .10000000E-01 1.0000000     4410.0000     40.000000
       PSIZE          TEFF          CAPF          FCR           PER           CCM           ELEV          RJOFL

(E1)  91.000000     24.000000     440.00000     0.            146.65000     121000.00     30.000000     -0.
       FCJS          PWCOS         PLANC         COSTL         CSSPKW        CAPCHG        HPCST         CTUHB

(E2)  1.6110000     1.4640000     1.5140000     1.2100000     1.3310000     1.2100000     1.4640000
       POHRAF        POHFAN        POHLEC        POHCIR        POHCND        POHSTC        POHSCL

(F1)      3.50  1.0000000    -.49000000E-02  .26750000E-02 -.11250000E-03
       RRP        TPO(1)        TPO(2)         TPO(3)         TPO(4)

(G1)  .95000000     125.00000
       EFFP          OPRESS

(H1)  1.0000000     0.            153.00000     0.            2.0000000     0.            0.            0.
       VELREC        XOEPA         T_IM          TOWMIN        PFACT       TFIX          RFIX          IOWMAX

(H2)  .90000000
       GSEFF

(I1)  1.7400000     1.0000000     7.0500000     0.            2.2200000     2.5000000     .55000000     4.6000000
       CPM           WPL           C_UVR         CHAILS        JCS           JWS           CVM           WFV

(J1) TABLE                  2.0000000     12.000000     1.0000000     5.0000000
       NTUCAL                WKNP          W9            NTUS          SS

(J2)  12.000000     18.000000     12.000000     500.00000
       DESVEL        REDUCE        OREDUCE       DIST

(K1)  2.5720000     2.2270000     2.5720000     .82000000E-01 2.5720000     .18000000E-01 -0.
       XW            XD            XOG           SF            DFIN          THFIN         DANGLE

(L1)   1.00000    .93000  20.00000 STEEL WELDED       31.00000
       TOL          DI     GASLIN      TUBMA          CONL

(M1) DOUBLE FOOTED          ALUMINUM   118.00000
       FINTYP                FINMAT     CONF

(N1)  0.            0.            .55000000     .20000000     2.4000000
       CFS           COATC         ZINCC         CASTC         EPREPC

(P1) FORM REMOVABLE         GALVANIZED  .52500000
       MEDTYP                MEDMAT      TM
```

FIGURE 5-4. SOA Model Sample Output

```
      0.              2.0000000       .50000000       20.000000       5.6250000       .48000000       5.0000000
Q1      CAJ              CWJ             CAH             CV              CMW             CST             CAJ
R1    2.0000000       7.0000000       1.0000000       18.000000       .95000000       5.0000000       ADMIRALTY
        XNS             VELV            DDC             GA              FC              TTU              CONMAT
S1      =0       =0      =0
      LFB      LFE   MXEXT
T1 PFR
      FFAX
```

FIGURE 5-4. Continued

```
(200) I N P U T   S U M M A R Y   A N D   C A S E   D E S C R I P T I O N

(205) OPTIONS SELECTED FOR THIS RUN                        (210) STARTING CONDITIONS

   UPPER LIMIT ON TURBINE TEMP 153.0 F
  FRICTION FACTOR AND HT TRFR COEFF BY PFR
  EXCHANGER DEPTH UNCONSTRAINED                     TURBINE OUTLET TEMPERATURE, T1     95.000  F
  EXCHANGER LENGTH FIXED AT   80. FEET                       CAPACITY RATIO, CHARA     1.000
  RUN WITH VELOCITY RECOVERY                                  FRONTAL AREA, AFRON     300000.
  NTU CALCULATION IS BY TABLE                       WIDTH TO LENGTH RATIO, WLRAT     20.000

                                                    COEFFICIENTS FOR POLYNOMIAL CURVE FIT
                                                         A0           A1           A2           A3           A4
(220) HEAT RATE FACTOR AS FUNCTION OF BACKPRESSURE (INCHES OF MERCURY)   1.00000E+00 -4.90000E-03  2.67500E-03 -1.12500E-04
  TURBINE RATING BACKPRESSURE  3.50 INCHES OF MERCURY
  SITE ELEVATION  4410. FEET ABOVE SEA LEVEL

(230) UNIT COSTS AND COSTING FACTORS

FUEL    31.000 CENTS/MMBTU   POWER PLANT CONSTRUCTION    440.000 $/KW                         FIXED CHARGE RATE      .174
LAND     0.000 $/SQ FT       REPLACEMENT POWER            24.000 MILLS/KWH                     CAPACITY FACTOR      .650
MOTORS  30.000 $ / HP        STEAM SUPPLY                146.650 $/KW TURBINE OUTPUT   BASE THERMAL EFFICIENCY      .400
                             CAPACITY CHARGE          121000.000 $/MW           CONSTRUCTION COST MULTIPLIER      1.000
                                                                                MAINT COSTS /CAPITAL COSTS      .010
```

FIGURE 5-4. Continued

```
(300) TO USED IN THE FOLLOWING CALCULATIONS IS =  50.000

   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS          REJECTED

   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS          REJECTED

(400) STEAM SUPPLY SCALING TO COMPENSATE FOR INCREASED HEAT RATE AT RATING BACKPRESSURE.  RBP= 3.50 IN. HG          HRFAC1=   1.010795
                   STEAM SUPPLY COST DIFFERENTIAL $     1583133.                                TURBINE COST DIFFERENTIAL          0. $/KW
   PLANT COST INCLUDING STEAM SUPPLY AND TURBINE $  441583133.     PLANT UNIT COST INCLUDING STEAM SUPPLY AND TURBINE        442. $

   ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(500) CONDITIONS AFTER SCALING PLANT AND STEAM SUPPLY  (BEFORE SCALING FOR FANS AND PUMPS)  HRFAC2=    .9996632
                                                                                          PSIZ12=    989.2547      MW
                                                                                          PLANC2=    436.8382      $/KW
                                                                                            SPBP=-3161792.42       $
                                                                                   BACK PRESSURE=    1.995081      IN. HG
                                                                                 SAT TEMPERATURE=  100.000000      DEG F
              WATER FLOW RATE  202805420.      LB MASS / HR
              AIR   FLOW RATE  345014251.      LB MASS / HR
                   PUMP POWER  6199.50275      KILOWATTS
                    FAN POWER  16796.9954      KILOWATTS
              HX SURFACE AREA  31107612.9      SQ FT
                FRONTAL AREA   445600.000      SQ FT
          WIDTH/LENGTH RATIO   69.7812500
                        WIDTH      25045       TUBES
                        WIDTH  5592.50000      FEET
     CONDENSER HT TRFR AREA   550959.410       SQ FT TOTAL
       NO. CONDENSER TUBES  58348.0000
      CONDENSER TUBE LENGTH  36.0000000        FT.
              WATER VELOCITY  7.00000000       FT/SEC

   ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(550) SCALING FOR FAN AND PUMP POWER REQUIREMENTS                              SCALING FACTOR   1.02353779
                                                                                          PSIZE   1000.00000      MW
                                                                                         PSIZ13   1012.53961      MW
                                                                                         PTOTAL   1023.53779      MW
                                                                                         PLANC3   447.120412      $/KW
                                                                                            SPC   447120412.      $
                                                                                           SPCD   7120411.77      $

              WATER FLOW RATE  207575964.      LB MASS / HR
              AIR   FLOW RATE  364904015.      LB MASS / HR
                   PUMP POWER  5345.52770      KILOWATTS
                    FAN POWER  17192.2592      KILOWATTS
              HX SURFACE AREA  32310572.9      SQ FT
                FRONTAL AREA   457111.975      SQ FT
          WIDTH/LENGTH RATIO   71.4237461
                        WIDTH      25655       TUBES
                        WIDTH  5713.99969      FEET
     CONDENSER HT TRFR AREA   565927.779       SQ FT TOTAL
       NO. CONDENSER TUBES  59721.5527
      CONDENSER TUBE LENGTH  36.0000000        FT.
              WATER VELOCITY  7.00000000       FT/SEC
```

FIGURE 5-4. Continued

```
(600) N O N - D E L T A   H E A T   E X C H A N G E R   D E S C R I P T I O N

                SUPPORT SPACING   5.000  FEET          ROOT DIAMETER  1.0720 INCH
      TUBE PITCH NORMAL TO AIR FLOW   2.572  INCH          FIN THICKNESS   .0180 INCH
TUBE PITCH IN DIRECTION OF AIR FLOW   2.227  INCH          FIN DIAMETER    2.572INCH
           TUBE DIAGONAL PITCH   2.572  INCH          FIN SPACING     .0820 INCH
                    LINER GAGE  20.0                    FIN TYPE DOUBLE FOOTED
                LINER MATERIAL STEEL WELDED          FIN MATERIAL ALUMINUM
      LINER THERMAL CONDUCTIVITY 31.000            FIN THERMAL COND.   118.0  BTU/(HR SQ FT DEG F)/FT
   LINER TUBING OUTSIDE DIAMETER  1.0000  INCH          HEADER TYPE FORM REMOVABLE
   LINER TUBING INSIDE DIAMETER   .9300  INCH          HEADER LENGTH     12.000 FEET

                        *** U N I T   C O S T S ***

         FIN BONDING $ 0.       /SQ FT             HEADER MATERIAL (GALVANIZED)  $     .570/LB
  PROTECTIVE COATING $ 0.       /SQ FT             ROLLED JOINT, TUBE TO HEADER  $    0.000/TUBE
    ZINC FOR SPACERS $ .3500    /LB                WELDED JOINT, TUBE TO HEADER  $    2.000/TUBE
      SPACER CASTING $ .2000    /EA            TUBE AND PLUG HOLE PREPARATION    $     .500/HOLE
     END PREPARATION $ 2.400    /TUBE                     NOZZEL AND ATTACHING   $   20.000/HOLE
                                                          BOLTED HEADER JOINT    $    5.000/FT
        LINER TUBING    $ .32411    /FT              HEADER MACHINING AND WELDING $    5.625/FT
  PROTECTIVE COATING    $ 0.        /FT                       STRUCTURAL STEEL   $     .480/LB
             FINNING    $ .51691    /FT

                     FINNED TUBE  $ .94102    /FT
                    TUBE SPACERS  $ .50063    /FT

                  ****************************
             (700)H E A T   E X C H A N G E R                   37.5 FANS          2.0 PASS
                  ****************************

                                                 ***************************
                                             (710)  AIR            WATER
                                                 ***********  ***********
                               REYNOLDS NUMBER  7.68364E+03   4.75064E+04
                            VELOCITY (FT/SEC)   1.71345E+01   4.52651E+00
              PRESSURE DROP (LBS FORCE/SQ IN)   1.12822E-02   6.06090E+00
         HEAT TRFR COEFF (BTU/HR SQ FT DEG F)   7.23233E+00   1.08841E+03
                              FRICTION FACTOR   8.50160E-01   5.35782E-03
                        INLET TEMPERATURE (F)   5.00000E+01   9.56000E+01
                         EXIT TEMPERATURE (F)   7.52409E+01   7.03591E+01
                                    RANGE (F)   2.52409E+01   2.52409E+01
                                         CARA          1.00000E+00
                                          NTU          1.31194E+00
          OVERALL U (BTU/HR SQ FT DEG F)               5.20599E+00
                                EFFECTIVENESS              .5535
                               FIN EFFICIENCY              .8545

                                              (730)WT OF WATER      2450855. LB
(720) AFTR .94034152                                  WT OF TUBES      6522364. LB
      ALPHA 188.71115                                 WT OF FRAME      1799062. LB
      SIGMA .47822706                                 WT OF HEADER      933762. LB
                                                      WT OF STRUCT     6757737. LB
                                                         ROOF LOAD    18284479. LB     (    40.00 LB/SQ FT)
(740)FITCO/ATOT .12536394

TABLE        NTU EFFECTIVENESS CORRELATION USED
```

FIGURE 5-4. Continued

```
750  WATER FLOW RATE  207576953 LBM/HR
       AIR FLOW RATE  864904015 LBM/HR
      TUBES IN DEPTH       3.27 TUBES   (      .6 FT DEEP)
  TUBES ACROSS FRONT      26558 TUBES   ( 5713.9 FT WIDE)

              LENGTH        90. FEET

        FRONTAL AREA     457111 SQ FEET
        SURFACE AREA   52310572 SQ FT
```

```
                  COOLING SURFACE $ 11714357.
           760 HEADER AND NOZZEL $  1861126.
      BUNDLE ASSEMBLE AND FRAME $  3439761.
                          PLENUM $   206970.
                         LOUVERS $  4704556.
                    HAIL SCREENS $        0.
                       STRUCTURE $  5151423.
                      FOUNDATION $   483939.
      HEAT XCHR SHIP AND ERECT $  2041832.
```

FIGURE 5-4. Continued

****

(800) PIPING 125 PSI DESIGN PRESSURE

****

CIRCULAR TOWERS	12	TOWER GROUPS	1
TOWERS PER GROUP	12	TOWERS IN INCOMPLETE GROUP	0

CIRCULATION PIPING TO EACH TOWER GROUP

HEADER SECTION	DIAMETER (INCHES)	LENGTH (FEET)	PIPE	TEES	REDUCERS	EXPANSION JOINT(1)	FLANGES (3)
			COST IN 10**3 DOLLARS FOR				
1	120.	572.	920.	93.	0.	45.	99.
2	120.	0.	0.	93.	0.		
3	120.	222.	317.	93.	75.		
4	102.	0.	0.	61.	0.		
5	102.	222.	271.	61.	0.		
6	102.	0.	0.	61.	29.		
7	84.	222.	202.	40.	0.		
8	84.	0.	0.	40.	0.		
9	84.	222.	202.	40.	17.		
10	66.	0.	0.	23.	0.		
11	66.	222.	144.	23.	8.		
12	48.	0.	0.	9.	0.		

COST OF CIRCULATION PIPING FOR EACH TOWER GROUP $5597. *10**3
COST OF CIRCULATION PIPING FOR INCOMPLETE GROUP $ 0. *10**3

TOTAL COST OF CIRCULATION PIPING $5597. *10**3

TOWER PIPING

TOWER DIAMETER	148. FEET
DIAMETER OF PIPE FROM CIRCULATION HEADER TO TOWER DISTRIBUTION HEADER	36. INCHES
DIAMETER OF TOWER DISTRIBUTION PIPE	18. INCHES

COST OF DISTRIBUTION PIPING PER TOWER $ 297. *10**3

TOTAL COST OF TOWER DISTRIBUTION PIPING $3562. *10**3

FIGURE 5-4. Continued

QUADRANT PIPING

HEADER SECTION	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS
1	18.0	0.0	0.0	1.3	0.0
2	18.0	9.4	.8	1.3	0.0
3	18.0	9.4	.8	1.3	0.0
4	18.0	9.4	.8	1.3	0.0
5	18.0	9.4	.8	1.3	0.0
6	18.0	9.4	.8	1.3	0.0
7	18.0	9.4	.8	1.3	0.0
8	18.0	9.4	.8	1.3	0.0
9	18.0	9.4	.8	1.3	0.0
10	18.0	9.4	.8	1.2	0.0

TUBES PER BUNDLE 182
BUNDLES PER QUADRANT 10
COST OF HEADERS PER QUADRANT $ 62. *10**3

TOTAL COST OF QUADRANT HEADERS $2970. *10**3

PUMP STATION

DESIGN VELOCITY 12. FPS
FLOW PER BUNDLE 448. GPM
FLOW PER QUADRANT 5075. GPM
FLOW PER TOWER 33901. GPM
FLOW PER TOWER GROUP 406809. GPM
FLOW TO INCOMPLETE GROUP 0. GPM

FOR EACH TOWER GROUP		FOR INCOMPLETE TOWER GROUP	
DYNAMIC HEAD	59. FEET	DYNAMIC HEAD	0. FEET
PUMP COST	$ 961. *10**3	PUMP COST	$ 0. *10**3
PUMP FITTINGS COST	$ 718. *10**3	PUMP FITTINGS COST	$ 0. *10**3
PUMP STATION COST	$1679. *10**3	PUMP STATION COST	$ 0. *10**3

PUMP ELECTRICAL COSTS $ 155. *10**3

COST OF PUMP STATIONS $1679. *10**3

TOTAL PIPING SYSTEM COSTS $13966. *10**3

FIGURE 5-4. Continued

1000 **F A N S**

FANS EQUIPPED WITH VELOCITY RECOVERY DIFFUSERS

*** DRIVE SYSTEM ***

*** FAN GEOMETRY ***

HUB DIAMETER	13.68 FEET
BLADE DIAMETER	60.00 FEET
BLADE ANGLE	16.00 DEGREES
NUMBER OF BLADES/FAN	12.00
NUMBER OF FANS	37.

*** FAN COSTS ***

BLADE AND HUB COST/FAN $	54730.
MOTOR COST/FAN $	34770.
SPEED REDUCER DRIVE COST/FAN $	84726.
STACK COST/FAN $	62359.
ELECTRICAL COST/FAN $	59530.

*** ANNUAL OPERATIONAL COSTS $ 3935600.

*** DESIGN CONDITIONS ***

	(ACTUAL)	(ADJUSTED)
VOLUMETRIC FLOW RATE/FAN	6166745. CU FT/MIN	
TOTAL VOLUMETRIC FLOW RATE	224877512. CU FT/MIN	
TOTAL AIR PRESSURE DROP OF H.X.	.01765 PSI	.02112
PRESSURE DROP ACROSS H.X. BUNDLE	.01128 PSI	.01350
NON-RECOVERABLE VELOCITY HEAD	.00637 PSI	.00762
HORSEPOWER/FAN	619. HP	739.
FAN POWER REQUIREMENTS	22525. HP	

FIGURE 5-4. Continued

```
900 C O N D E N S E R

*** DESIGN CONDITIONS ***                         *** DESIGN OPERATING CONDITIONS ***

         STEAM TEMPERATURE 100.6 F        OVERALL HEAT TRANSFER COEFFICIENT   662.46 BTU/HR SQ FT DEG F
    INLET WATER TEMPERATURE  70.4 F       AVERAGE WATER VELOCITY THRU COND      7.0 FT/SEC
     EXIT WATER TEMPERATURE  95.6 F            PRESSURE DROP THRU CONDENSER  1107.1 LB/SQ FT
LOG MEAN TEMPERATURE DIFFERENCE  14.0 F              CONDENSER EFFECTIVENESS      .835

*** CONDENSER DESCRIPTION ***                     *** CONDENSER COSTS ***

         NUMBER OF SHELLS      2                        TUBING MATERIAL $   2104222.52  (ADMIRALTY
HEAT TRANSFER AREA PER SHELL .27549E+06 SQ FT                  SHELLS $   2405250.56
  NUMBER OF PASSES PER SHELL    2                     FIELD ERECTION $   1833317.45
                 TUBE LENGTH    56      FEET
  NUMBER OF TUBES PER SHELL 29174               TOTAL COST OF CONDENSER $     6342791.
```

FIGURE 5-4. Continued

(1100) AMBIENT TEMP	TURBINE OUTLET TEMP	BACK PRESSURE	HEAT RATE FACTOR	RANGE	TTDP	HX LMTD	HX ITD	POWER GENERATED	CHANGE IN FUEL COST	COST TO REPLACE LOST CAP	POWER COST AT AMBIENT TEMP	PCT TIME AT AMBIENT TEMP	PORTION OF INCREMENTAL POWER COST
DEG F	DEG F	IN. HG		DEG F	DEG F	DEG F	DEG F	MEGAWATTS	MILLS/KWH	MILLS/KWH	MILLS/KWH	PERCENT	MILLS/KWH
100.0	153.0	8.12	1.0753	26.436	5.237	21.33	48.00	927.337	.16221	1.7439	4.3944	.150	.65916E-02
95.0	147.5	7.09	1.0596	26.188	5.198	19.69	47.50	942.409	.16221	1.3822	4.0327	1.000	.40327E-01
85.0	135.7	5.38	1.0335	25.789	5.109	20.91	46.70	966.683	.15221	.79961	3.4501	2.850	.98328E-01
75.0	125.2	4.08	1.0169	25.521	5.055	20.64	46.16	992.967	.16221	.40880	3.0593	8.220	.25147
65.0	115.9	3.07	1.0059	25.358	5.023	20.48	45.83	992.886	.16221	.17075	2.8212	9.130	.25756
55.0	105.7	2.30	1.0015	25.267	5.005	20.39	45.65	999.381	.15221	.38852E-01	2.6894	10.960	.29475
45.0	95.6	1.71	.9989	25.223	4.996	20.34	45.56	1001.080	.15476	0.	2.6430	11.190	.29576
35.0	85.5	1.26	.9974	25.206	4.993	18.29	45.53	1002.141	.14744	0.	2.6357	56.500	1.4892

(1200) FINAL INCREMENTAL COST 2.7340

CAPACITY CHARGE $ 8792162.84

FIGURE 5-4. Continued

CAPITAL COST SUMMARY			
COOLING SYSTEM CAPITAL			61953951.
TOWER		41166631.	
HEAT EXCHANGER	19503659.		
PLENUM	205970.		
LOUVERS	3704556.		
HAIL SCREENS	0.		
FAN SYSTEM	11164524.		
STRUCTURE	5151423.		
FOUNDATION	145939.		
PIPING		14295233.	
CONDENSER		5492056.	
CAPACITY PENALTY			5792165.
PLANT SCALING			7120412.
STEAM SUPPLY		1602954.	
BASE PLANT		5517427.	
LAND			0.
TOTAL CAPITAL COST		$	77865525.

(1300) UNIT ENERGY COST SUMMARY			
COOLING SYSTEM CAPITAL			1.49322
TOWER		1.25794	
HEAT EXCHANGER	.596063		
PLENUM	.632400E-02		
LOUVERS	.145704		
HAIL SCREENS	0.		
FAN SYSTEM	.341109		
STRUCTURE	.157420		
FOUNDATION	.147854E-01		
PIPING		.436891	
CONDENSER		.167855	
CAPACITY PENALTY			.200675
PLANT SCALING			.217504
STEAM SUPPLY		.489945E-01	
BASE PLANT		.168604	
(1340) TOTAL CAPITAL COST			2.37948
(1350) ENERGY PENALTY			.926775E-01
(1360) ADDITIONAL BASE PLANT FUEL			.153054
(1370) OPERATION AND MAINTENANCE			.106606
(1380) TOTAL			2.73400

FIGURE 5-4. Continued

5.3.2 PLASTIC Tube Model Output

A complete output report from a typical run of the PLASTIC model is shown in Figure 5-5. The description of the metal fin tube output (Section 5.3.1) applies to the plastic tube model output with two exceptions:

1. The line tags A1, B1, etc., correspond to card type designations used in the Input Description for the PLASTIC model (Section 5.2.2).

2. The term AFTR is omitted from the geometry terms in report section (720) since it is meaningless for tubes without fins.

```
(100) 3N41 DRY COOLING TOWER COST MODEL -- PLASTIC TUBE, CIRCULAR TOWER VERSION.THIS RUN MADE  03/30/77      09.24.31
                   ***
              (A1) ***           EXAMPLE RUN - PLASTIC TUBE VERSION
                   ***

(B1) 50.000000    50.000000    -0.          1.0000000    500000.00    20.000000    80.000000    -0.
      TSTAR        TEND         VAS(1)       VAS(3)       VAS(4)       VAS(5)       FIXL         DEEPL

(C1) 100.0   95.0    85.0    75.0    65.0    55.0    45.0    35.0    .0015   .0100   .0285   .0622   .0915   .1095   .1119   .5650
     TA(1)   TA(2)   TA(3)   TA(4)   TA(5)   TA(6)   TA(7)   TA(8)   TPER(1) TPER(2) TPER(3) TPER(4) TPER(5)TPER(6) TPER(7) TPER(8)

(C2) -0.0    -0.0    -0.0    -0.0    -0.0    -0.0    -0.0    -0.0 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000
     TA(9)   TA(10)  TA(11)  TA(12)  TA(13)  TA(14)  TA(15)  TA(16) TPER(9) TPER(10)TPER(11)TPER(12TPER(13)TPER(14)TPER(15)TPER(16)

(D1) 1000.0000    .40000000    .65000000    .17400000    .10000000E-01 1.0000000   4410.0000    40.000000
      PSIZE        TEFF         CAPF         FCR          PER          CCM          ELEV         ROOFL

(E1) 81.000000    24.000000    440.00000    0.           146.65000    121000.00    50.000000
      FCUS         PACOS        PLANC        COST.  *     CSSPKA       CAPCHG       HPCST

(E2) 1.5110000    1.4640000    1.5140000    1.2100000    1.3310000    1.2100000    1.4640000
      POHBAF       POHFAN       POHLEC       POHCIR       POHCND       POHSTC       POHSCL

(F1)      3.50  1.0000000    -.49000000E-02  .26750000E-02 -.11250000E-03
      RRP        TPO(1)        TPO(2)          TPO(3)        TPO(4)

(G1) .35110000    125.00000
      EFFP         DPRESS

(H1) 1.0000000    0.           153.00000    0.           2.0000000    -0.          -0.          0.
      VELREC       XDEPA        TLIM         TOHMIN       PFACT        TFIX         RFIX         TOHMAX

(H2) .90000000
      GBEFF

(I1) 1.7400000    1.0000000    7.1500000    0.           2.2200000    2.5000000    .55000000    4.0000000
      CPM          APL          CLOVR        CHAILS       UCS          UHS          CVM          RFV

(J1) TABLE              2.0000000    12.000000    1.0000000    1.5000000
      VTICAL             HXNP         NB           NTUB         SS

(J2) 12.000000    18.000000    12.000000    500.00000
      DESVEL       HEDUCE       AREDUCE      DIST

(K1)  .50000     .43301     .50000 STAGGERED 2          60.00000
      XA         XD         XDG    TUBARS HCALC         DANGLE

   THEATA  2

(L1)  .25000     .20000  20.00000 POLYETHYLENE          .29100
      DOL          DI     GASLIN          TUBMA         CONL

(M1)POLYCARBONATE       .25000000    .25000000    2.5000000    .50000000    .28400000    .40000000E-01
      BPAMAT             CSEALP       TSEALP       CELAST       TELAST       RUESEL       MOEELA

(N1) 0.           0.           .40000000
      CASTC        EPREPC       CFS
```

FIGURE 5-5. PLASTIC Tube Model Sample Output

```
(P1) FORM REMOVABLE      GALVANIZED     .62500000
       MEDTYP              MEDMAT         TM

(Q1) 0.               0.              .50000000E-01  20.000000   5.6250000   0.          5.0000000
       CRJ              CWJ             CHM            CN          CMW         CST         LBJ

(R1) 2.0000000        7.0000000       1.0000000      18.000000   .95000000   5.0000000     ADMIRALTY
       XNS              VEN             DDC            GA          FC          TTD           CONMAT

(S1)   55     85    -0
     LFA    LFE  MXEXT
```

FIGURE 5-5. Continued

```
(200) I N P U T   S U M M A R Y   A N D   C A S E   D E S C R I P T I O N

(205) ******************************                           (210) ******************
      OPTIONS SELECTED FOR THIS RUN                                  STARTING CONDITIONS
      ******************************                                 ******************

   UPPER LIMIT ON TURBINE TEMP 133.0 F                    TURBINE OUTLET TEMPERATURE, T1    96.000  F
  EXCHANGER DEPTH UNCONSTRAINED                                 CAPACITY RATIO, CHARA        1.000
  EXCHANGER LENGTH FIXED AT    80. FEET                           FRONTAL AREA, AFRON     500000.
  RUN WITH VELOCITY RECOVERY                             WIDTH TO LENGTH RATIO, WLRAT       20.000
  NTU CALCULATION IS BY TABLE
      TUBE ARRANGEMENT STAGGERED

                                                                  COEFFICIENTS FOR POLYNOMIAL CURVE FIT
                                                           A0            A1            A2            A3            A4
(220) HEAT RATE FACTOR AS FUNCTION OF BACKPRESSURE (INCHES OF MERCURY)   1.00000E+00 -4.90000E-03  2.67500E-03 -1.12500E-04
  TURBINE RATING BACKPRESSURE  3.50 INCHES OF MERCURY
  SITE ELEVATION  4410. FEET ABOVE SEA LEVEL

                              (230) UNIT COSTS AND COSTING FACTORS
                                    *****************************

FUEL    51.000 CENTS/MMBTU   POWER PLANT CONSTRUCTION     440.000 $/KW                          FIXED CHARGE RATE     .174
LAND     0.000 $/SQ FT       REPLACEMENT POWER             24.000 MILLS/KWH                     CAPACITY FACTOR       .650
MOTORS  30.000 $ / HP        STEAM SUPPLY                 145.650 $/KW TURBINE OUTPUT   BASE THERMAL EFFICIENCY   .400
                             CAPACITY CHARGE           121000.000 $/MW            CONSTRUCTION COST MULTIPLIER  1.000
                                                                                    MAINT COSTS /CAPITAL COSTS   .010
```

FIGURE 5-5. Continued

```
(300) TO USED IN THE FOLLOWING CALCULATIONS IS =  50,000
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
   IN PIPING ROUTINE NUMBER OF TOWERS PER GROUP =0 WAS        REJECTED
(400) STEAM SUPPLY SCALING TO COMPENSATE FOR INCREASED HEAT RATE AT RATING BACKPRESSURE.  RBP= 3.50 IN. HG          HRFAC1=   1.010795

     STEAM SUPPLY COST DIFFERENTIAL $    1583133.               STEAM SUPPLY UNIT COST    1583133.  $/KW
   PLANT COST INCLUDING STEAM SUPPLY $  441583133.   PLANT UNIT COST INCLUDING STEAM SUPPLY     441.5833/KW

   ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(500) CONDITIONS AFTER SCALING PLANT AND STEAM SUPPLY  (BEFORE SCALING FOR FANS AND PUMPS)  HRFAC2=   .9998852
                                                                                           PSIZ12=   989.2547     MW
                                                                                           PLANC2=   436.8382     $/KW
                                                                                            SPBP=-3161792.42      $
                                                                                    BACK PRESSURE=   1.985081     IN. HG
                                                                                 SAT TEMPERATURE= 100.600000      DEG F
            WATER FLOW RATE  201209109.      LB MASS / HR
            AIR   FLOW RATE  676105977.      LB MASS / HR
                 PUMP POWER  5984.79062      KILOWATTS
                  FAN POWER  9201.45282      KILOWATTS
            HX SURFACE AREA  24985212.9      SQ FT
              FRONTAL AREA   591400.000      SQ FT
         WIDTH/LENGTH RATIO  92.4062500
                      WIDTH      177420      TUBES
                      WIDTH  7392.50000      FEET
    CONDENSER HT TRFR AREA   552072.919      SQ FT TOTAL
      NO. CONDENSER TUBES   58476.0000
     CONDENSER TUBE LENGTH   36.0000000      FT.
            WATER VELOCITY   6.92955599      FT/SEC

   ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(550) SCALING FOR FAN AND PUMP POWER REQUIREMENTS                                 SCALING FACTOR  1.01541970
                                                                                           PSIZE  1000.00000    MW
                                                                                          PSIZ13  1004.50875    MW
                                                                                          PTOTAL  1015.41970    MW
                                                                                          PLANC3  443.574121    $/KW
                                                                                             SPC  443574121.    $
                                                                                            SPCD  3574121.45    $

            WATER FLOW RATE  204311693.      LB MASS / HR
            AIR   FLOW RATE  695531227.      LB MASS / HR
                 PUMP POWER  6076.35350      KILOWATTS
                  FAN POWER  9343.33595      KILOWATTS
            HX SURFACE AREA  25370476.9      SQ FT
              FRONTAL AREA   600519.210      SQ FT
         WIDTH/LENGTH RATIO  93.8311266
                      WIDTH      180155      TUBES
                      WIDTH  7506.49013      FEET
    CONDENSER HT TRFR AREA   560593.616      SQ FT TOTAL
      NO. CONDENSER TUBES   59377.5923
     CONDENSER TUBE LENGTH   36.0000000      FT.
            WATER VELOCITY   6.92955599      FT/SEC
```

FIGURE 5-5. Continued

```
(600)   60.0 DEGREE   D E L T A   H E A T   E X C H A N G E R   D E S C R I P T I O N

               SUPPORT SPACING   1.500  FEET       SEAL PLATE THKNS       .250 INCH
     TUBE PITCH NORMAL TO AIR FLOW    .500  INCH     SEAL PLATE DENSITY      .284 LB/CU IN.
 TUBE PITCH IN DIRECTION OF AIR FLOW    .433  INCH       ELASTOMER THKNS       .500 INCH
             TUBE DIAGONAL PITCH    .500  INCH     ELASTOMER DENSITY      .040 LB/CU IN.
                      LINER GAGE  20.0
                  LINER MATERIAL POLYETHYLENE
      LINER THERMAL CONDUCTIVITY    .291
     LINER TUBING OUTSIDE DIAMETER   .2500  INCH          HEADER TYPE FORM REMOVABLE
      LINER TUBING INSIDE DIAMETER   .2000  INCH        HEADER LENGTH     12.000 FEET

                          *** U N I T   C O S T S ***

                                                 HEADER MATERIAL (GALVANIZED) $    .570/LB
                                                ROLLED JOINT, TUBE TO HEADER $   0.000/TUBE
                                                WELDED JOINT, TUBE TO HEADER $   0.000/TUBE
   SPACER CASTING $ 0.        /EA             TUBE AND PLUG HOLE PREPARATION  $    .050/HOLE
 END PREPARATION $ 0.         /TUBE                    NOZZEL AND ATTACHING  $  20.000/HOLE
      SEAL PLATE $ .2500     /LB                      BOLTED HEADER JOINT  $   5.000/FT
      FRAME COST $ .3000     /LB            HEADER MACHINING AND WELDING  $   5.625/FT
 LINER TUBING     $ .58782E-02/FT                        STRUCTURAL STEEL  $   0.000/LB
                  TUBE SPACERS  $ .51450E-02/FT


                 ******************************
            (700) H E A T   E X C H A N G E R                     176.9 FANS          2.0 PASS
                 ******************************


                                                 -------------------------
                                             (710)  AIR            WATER
                                                 -----------  -----------
                        REYNOLDS NUMBER  1.03553E+03  3.90812E+03
                     VELOCITY (FT/SEC)  9.96192E+00  1.73186E+00
        PRESSURE DROP (LBS FORCE/SQ IN)  8.67679E-03  7.70340E+00
   HEAT TRFR COEFF (BTU/HR SQ FT DEG F)  1.54015E+01  6.86147E+02
                       FRICTION FACTOR  1.20987E-01  1.00042E-02
                    INLET TEMPERATURE (F)  5.00000E+01  9.56000E+01
                     EXIT TEMPERATURE (F)  8.15467E+01  7.01591E+01
                              RANGE (F)  3.15467E+01  2.54409E+01
                                   CHARA        1.24000E+00
                                     NTU        2.06023E+00
        OVERALL U (BTU/HR SQ FT DEG F)        1.53801E+01
                           EFFECTIVENESS            .6918
                          FIN EFFICIENCY           0.0000
```

FIGURE 5-5. Continued

```
(720) ALPHA 45.531420
      SIGMA .50000000

 FITCO/ATOT .89812871E-01

(740) TABLE          NTU EFFECTIVENESS CORRELATION USED

(750) WATER FLOW RATE  204311593 LBM/HR
        AIR FLOW RATE  685331226 LBM/HR
      TUBES IN DEPTH       26.98 TUBES    (     1.0 FT DEEP)

 TUBES ACROSS FRONT       180155 TUBES    ( 7506.5 FT WIDE)

             LENGTH          90. FEET

       FRONTAL AREA       500819 SQ FEET
       SURFACE AREA     25370376 SQ FT
```

```
(730) WT OF WATER      6243628. LB
      WT OF TUBES     2675838. LB
      WT OF FRAME    12711698. LB
      WT OF HEADER    2235298. LB
      WT OF STRUCT    6997815. LB
         ROOF LOAD   24020768. LB      (    40.00 LB/SQ FT)
```

```
(760)    COOLING SURFACE $  5702405.
     HEADER AND NOZZEL $  4401470.
       BUNDLE ASSEMBLY $  2756714.
          BUNDLE FRAME $  3493400.
    SEALING ELASTOMER $   226438.
                PLENUM $   178528.
               LOUVERS $  3090250.
          HAIL SCREENS $        0.
             STRUCTURE $  5334435.
            FOUNDATION $   775737.
HEAT XCHR SHIP AND ERECT $  1962479.
```

FIGURE 5-5. Continued

```
****************
(800)   PIPING           125 PSI DESIGN PRESSURE
****************

CIRCULAR TOWERS     5          TOWER GROUPS                 1
TOWERS PER GROUP    5          TOWERS IN INCOMPLETE GROUP   0

          CIRCULATION PIPING TO EACH TOWER GROUP
```

HEADER SECTION	DIAMETER (INCHES)	LENGTH (FEET)	PIPE	TEES	REDUCERS	EXPANSION JOINT(1)	FLANGES (3)
			COST IN 1000$ DOLLARS FOR				
1	120.	515.	882.	55.	0.	45.	99.
2	120.	0.	0.	33.	75.		
3	102.	355.	431.	51.	29.		
4	84.	0.	0.	40.	17.		
5	65.	355.	229.	31.	0.		

```
COST OF CIRCULATION PIPING FOR EACH TOWER GROUP   $4145. *1000$
COST OF CIRCULATION PIPING FOR INCOMPLETE GROUP   $   0. *1000$

                    TOTAL COST OF CIRCULATION PIPING  $4145. *1000$


          TOWER PIPING

TOWER DIAMETER                                235. FEET
DIAMETER OF PIPE FROM CIRCULATION
HEADER TO TOWER DISTRIBUTION HEADER            54. INCHES
DIAMETER OF TOWER DISTRIBUTION PIPE            30. INCHES

COST OF DISTRIBUTION PIPING PER TOWER   $ 652. *1000$

              TOTAL COST OF TOWER DISTRIBUTION PIPING  $3159. *1000$
```

FIGURE 5-5. Continued

QUADRANT PIPING

HEADER SECTION	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS
1	30.0	0.0	0.0	2.5	0.0
2	30.0	1.5	.2	2.5	0.0
3	30.0	1.5	.2	2.5	0.0
4	30.0	1.5	.2	2.5	0.0
5	30.0	1.5	.2	2.5	0.0
6	30.0	1.5	.2	2.5	0.0
7	30.0	1.5	.2	2.5	0.0
8	30.0	1.5	.2	2.5	0.0
9	30.0	1.5	.2	2.5	0.0
10	30.0	1.5	.2	2.5	0.0
11	30.0	1.5	.2	2.5	0.0
12	30.0	1.5	.2	2.5	0.0
13	30.0	1.5	.2	2.5	0.0
14	30.0	1.5	.2	2.5	0.0
15	30.0	1.5	.2	2.5	0.0
16	30.0	1.5	.2	2.5	0.0
17	30.0	1.5	.2	2.5	.8
18	18.0	3.4	.3	1.3	0.0
19	18.0	3.4	.3	1.3	0.0
20	18.0	3.4	.3	1.3	0.0
21	18.0	3.4	.3	1.3	0.0
22	18.0	3.4	.3	1.3	0.0
23	18.0	3.4	.3	1.3	0.0
24	18.0	3.4	.3	1.3	0.0
25	18.0	3.4	.3	1.3	0.0
26	18.0	3.4	.3	1.3	0.0
27	18.0	3.4	.3	1.3	0.0
28	18.0	3.4	.3	1.3	0.0
29	18.0	3.4	.3	1.3	0.0
30	18.0	3.4	.3	1.3	0.0
31	18.0	3.4	.3	1.2	0.0

PUMP STATION

DESIGN VELOCITY	12. FPS
FLOW PER BUNDLE	651. GPM
FLOW PER QUADRANT	20180. GPM
FLOW PER TOWER	80721. GPM
FLOW PER TOWER GROUP	403605. GPM
FLOW TO INCOMPLETE GROUP	0. GPM

FOR EACH TOWER GROUP		FOR INCOMPLETE TOWER GROUP	
DYNAMIC HEAD	57. FEET	DYNAMIC HEAD	0. FEET
PUMP COST	$ 942. *10**3	PUMP COST	$ 0. *10**3
PUMP FITTINGS COST	$ 712. *10**3	PUMP FITTINGS COST	$ 0. *10**3
PUMP STATION COST	$1654. *10**3	PUMP STATION COST	$ 0. *10**3

PUMP ELECTRICAL COSTS $ 158. *10**3

COST OF PUMP STATIONS $1654. *10**3

TOTAL PIPING SYSTEM COSTS $13243. *10**3

FIGURE 5-5. Continued

```
                                   (900) ********************
                                         C O N D E N S E R
                                         ********************
      *** DESIGN CONDITIONS ***                                        *** DESIGN OPERATING CONDITIONS ***

              STEAM TEMPERATURE 100.6 F                   OVERALL HEAT TRANSFER COEFFICIENT   658.34 BTU/HR SQ FT DEG F
         INLET WATER TEMPERATURE  70.2 F                  AVERAGE WATER VELOCITY THRU COND      6.9 FT/SEC
          EXIT WATER TEMPERATURE  95.6 F                       PRESSURE DROP THRU CONDENSER   1067.1 LB/SQ FT
 LOG MEAN TEMPERATURE DIFFERENCE  19.1 F                           CONDENSER EFFECTIVENESS     .636

      *** CONDENSER DESCRIPTION ***                                        *** CONDENSER COSTS ***

                NUMBER OF SHELLS     2                             TUBING MATERIAL $    2108558.92  (ADMIRALTY
 HEAT TRANSFER AREA PER SHELL .27504E+06 SQ FT                             SHELLS $    2409641.79
     NUMBER OF PASSES PER SHELL      2                              FIELD ERECTION $    1837022.51
                    TUBE LENGTH     36       FEET
    NUMBER OF TUBES PER SHELL 22258                         TOTAL COST OF CONDENSER $      6355223.
```

FIGURE 5-5. Continued

(1000) F A N S

FANS EQUIPPED WITH VELOCITY RECOVERY DIFFUSERS

*** DRIVE SYSTEM ***

*** FAN GEOMETRY ***

HUB DIAMETER	6.31 FEET
BLADE DIAMETER	28.00 FEET
BLADE ANGLE	10.00 DEGREES
NUMBER OF BLADES/FAN	6.00
NUMBER OF FANS	179.

*** FAN COSTS ***

BLADE AND HUB COST/FAN $	5966.
MOTOR COST/FAN $	3139.
SPEED REDUCER DRIVE COST/FAN $	9156.
STACK COST/FAN $	16107.
ELECTRICAL COST/FAN $	15620.

*** ANNUAL OPERATIONAL COSTS $ 2622501.

*** DESIGN CONDITIONS ***

	(ACTUAL)	(ADJUSTED)
VOLUMETRIC FLOW RATE/FAN	1063393. CU FT/MIN	
TOTAL VOLUMETRIC FLOW RATE	192048225. CU FT/MIN	
TOTAL AIR PRESSURE DROP OF H.X.	.01256 PSI	.01525
PRESSURE DROP ACROSS H.X. BUNDLE	.00989 PSI	.01075
NON-RECOVERABLE VELOCITY HEAD	.00370 PSI	.00448
HORSEPOWER/FAN	70. HP	65.
FAN POWER REQUIREMENTS	12339. HP	

(1100)

AMBIENT TEMP	TURBINE OUTLET TEMP	BACK PRESSURE	HEAT RATE FACTOR	RANGE	ITDP	HX LMTD	HX ITD	POWER GENERATED	CHANGE IN FUEL COST	COST TO REPLACE LOST CAP	POWER COST AT AMBIENT TEMP	PCT TIME AT AMBIENT TEMP	PORTION OF INCREMENTAL POWER COST
DEG F	DEG F	IN. HG		DEG F	DEG F	DEG F	DEG F	MEGAWATTS	MILLS/KWH	MILLS/KWH	MILLS/KWH	PERCENT	MILLS/KWH
100.0	153.0	8.12	1.0763	26.645	5.237	17.96	49.00	927.914	.10611	1.7301	4.0608	.150	.60912E-02
95.0	147.5	7.09	1.0596	26.395	5.158	17.75	47.50	942.866	.10611	1.3712	3.7020	1.000	.37020E-01
85.0	136.7	5.38	1.0336	25.993	5.109	17.40	46.70	966.947	.10611	.79327	3.1240	2.850	.89035E-01
75.0	125.2	4.05	1.0169	25.723	5.055	17.17	45.16	983.102	.10611	.40556	2.7363	8.220	.22493
65.0	115.9	3.07	1.0069	25.559	5.023	17.02	45.83	992.942	.10611	.16939	2.5001	9.130	.22826
55.0	105.7	2.30	1.0015	25.469	5.005	16.93	45.65	998.394	.10611	.38543E-01	2.3693	10.960	.25968
45.0	95.6	1.71	.9989	25.423	4.996	16.91	45.56	1001.071	.09871	0.	2.3234	11.190	.25998
35.0	85.5	1.26	.9978	25.405	4.993	16.89	45.53	1002.124	.09146	0.	2.3161	56.500	1.3086

(1200) FINAL INCREMENTAL COST 2.4136

CAPACITY CHARGE $ 8722428.7

FIGURE 5-5. Continued

CAPITAL COST SUMMARY

COOLING SYSTEM CAPITAL			57214936.
TOWER		37314614.	
HEAT EXCHANGER	13656425.		
PLENUM	178528.		
LOUVERS	3090250.		
HAIL SCREENS	0.		
FAN SYSTEM	9121633.		
STRUCTURE	5334435.		
FOUNDATION	775737.		
PIPING		13446920.	
CONDENSER		6453503.	
CAPACITY PENALTY			8722429.
PLANT SCALING			3574121.
STEAM SUPPLY		1590271.	
BASE PLANT		1983851.	
LAND			0.
TOTAL CAPITAL COST		$	69511486.

(1300) UNIT ENERGY COST SUMMARY

COOLING SYSTEM CAPITAL			1.74840
TOWER		1.14026	
HEAT EXCHANGER	.576224		
PLENUM	.545553E-02		
LOUVERS	.944333E-01		
HAIL SCREENS	0.		
FAN SYSTEM	.276745		
STRUCTURE	.163012		
FOUNDATION	.237053E-01		
PIPING		.410914	
CONDENSER		.197209	
CAPACITY PENALTY			.266544
PLANT SCALING			.109220
STEAM SUPPLY		.485963E-01	
BASE PLANT		.606235E-01	
(1340) TOTAL CAPITAL COST			2.12417
(1350) ENERGY PENALTY			.919423E-01
(1360) ADDITIONAL BASE PLANT FUEL			.970036E-01
(1370) OPERATION AND MAINTENANCE			.100463
(1380) TOTAL			2.41359

FIGURE 5-5. Continued

5.3.3 NH_3 Model Output

Complete output from a typical run of the NH_3 model is shown in Figure 5-6. The circled numbers on the figure correspond to the numbers in parentheses in this section. The output report is divided into the following sections:

(100) through (300) Input Summary
(400) through (500) Plant scaling for increased steam, and for fans and pumps
(600) through (700) Heat Exchanger
(800) Piping
(900) Condenser
(1000) Fans
(1100) through (1200) Plant performance as a function of ambient temperature
(1300) Cost summary

(100) Output printed by subroutine SETUP.

This line begins a card-by-card summary of input data. Any cards preceding the first blank card are printed as title or description cards. After the first blank card the values are printed as they are read, and the variable name is printed immediately below the value. The line tags A1, B1, etc. correspond to the Card Type designations used in the Input Description (see Section 5.2.3). Variable names printed here are defined in the Input Description section. Every input variable read by the program appears in this summary.

(200) through (230) Output printed by subroutine INPSUM

Some of the more frequently changed input parameters are summarized here with more descriptive labels than are given in the preceding summary.

(205) The options summarized here are those selected by the variables FIXL, DEEPL AND POMDPL on Card Type B1; VELREC and TLIM on Card Type H1; and NTUCAL on Card Type J1.

(210) The starting point in the search for an optimum is specified by T1, TTD1, AFRON, and WLRAT. These correspond to the values of VAS(1), VAS(3), VAS(4) and VAS(5) on Card Type B1. (The initial value for the temperature range of the air is determined internally by the code.)

(220) Input data pertaining to the turbine are summarized here.

(230) Unit costs and costing factors from Card Types D1 and E1 are reported here.

(300) through (600)

These output report sections are as described in Section 5.3.1.

(700) through (760) Output printed by subroutine RPTHXC.

The heat exchanger design, operating conditions, performance characteristics and costs are reported here.

(710) Fluid flow and heat transfer characteristics are summarized. The term TTD1 is terminal temperature difference, and corresponds to VAS(3) on Card Type B1.

(720) Heat exchanger geometry terms reported here are defined as

$$\text{AFTR} = \frac{\text{Fin Area}}{\text{Total Area for Heat Transfer}}$$

$$\text{ALPHA} = \frac{\text{Heat Transfer Area (ft}^2\text{)}}{\text{Unit Volume of Heat Exchanger (ft}^3\text{)}}$$

$$\text{SIGMA} = \frac{\text{Free Flow Area}}{\text{Frontal Area}}$$

$$\frac{\text{FITCO}}{\text{ATOT}} = \frac{\text{Finned Tube Cost (\$)}}{\text{Heat Transfer Area (ft}^2\text{)}}$$

(730) Weights of major dry cooling tower components are reported.

(740) The option selected for NTU calculation is shown here, based on NTUCAL input on Card Type J1.

(750) Fluid flow rates and heat exchanger geometry previously printed are repeated here for convenience.

(760) An itemized list of costs associated with fabricating, shipping and installing the heat exchanger.

(800) Output printed by subroutine RPTSUP.

Supply piping costs are itemized separately for circulation piping, quadrant piping, and distribution piping. Pump and piping costs are reported in thousands of dollars ($ *10**3).

(850) Output printed by subroutine RPTRET. A cost report for the return piping system is provided here.

(900) through (1300)

These sections of the output are as described in Section 5.3.1.

```
(100)     BNWL DRY COOLING TOWER COST MODEL -- AMMONIA CIRCULAR TOWER VERSION   THIS RUN MADE  03/29/77      11.15.44
               ***
          (A1)           EXAMPLE RUN  -  AMMONIA VERSION
               ***

      50.000000     50.000000     105.00000     3.0000000     300000.00     20.000000     80.000000     5.7000000    -0.
(B1)   TSTAR          TEND          VAS(1)        VAS(3)        VAS(4)        VAS(5)        FIXL          DEEPL      PUMDPL

      100.0    95.0    85.0    75.0    65.0    55.0    45.0    35.0  .0015  .0100  .0285  .0822  .0913  .1096  .1119  .5650
(C1)  TA(1)   TA(2)   TA(3)   TA(4)   TA(5)   TA(6)   TA(7)   TA(8) TPER(1) TPER(2) TPER(3) TPER(4) TPER(5)TPER(6) TPER(7) TPER(8)

      -0.0     -0.0     -0.0     -0.0     -0.0     -0.0     -0.0     -0.0 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000 -0.0000
(C2)  TA(9)   TA(10)   TA(11)   TA(12)   TA(13)   TA(14)   TA(15)   TA(16) TPER(9) TPER(10)TPER(11)TPER(12TPER(13)TPER(14)TPER(15)TPER(16)

      1000.0000     .40000000     .65000000     .17400000     .10000000E-01 1.0000000     4410.0000     40.000000
(D1)  PSIZE          TEFF          CAPF          FCR           PER           CCM           ELEV          ROOFL

      91.000000     24.000000     440.00000     0.            146.65000     121000.00     30.000000     -0.
(E1)  PCOS           PACOS         PLANC         COSTL         CSSPKW        CAPCHG        HPCST         CTURB

      1.6110000     1.4640000     1.6140000     1.2100000     1.3310000     1.2100000     1.4640000
(E2)  POWBAF        POWFAN        POWLEC        POWCIR        POWCND        POWSTC        POWSCL

          5.50  1.0000000    -.49000000E-02  .26750000E-02 -.11250000E-03
(F1)  RBP        TPD(1)         TPD(2)         TPD(3)          TPD(4)

      .85000000
(G1)  EFFP

      1.0000000     0.            153.00000     6.0000000     2.0000000    -0.            6.0000000
(H1)  VELREC         XDEPA         TLIM          TOWMIN        PFACT         TFIX          TOWMAX

      .90000000
(H2)  GBEFF

      1.7490000     1.0000000     0.            1.8200000     2.2200000     2.5000000     .55000000     4.6000000
(I1)  CPM           WPL           CLUVR         CHAILS        JCS           JWS           CVM           WFV

     EQUATION           1.0000000     12.000000     1.0000000     5.0000000     90.000000
(J1)  NTUCAL             HXNP          W3            NTUS          SS            ANGLE

      15.000000     0.            0.            500.00000
(J2)  DESVEL        REDUCE        GREDUCE       DIST

      150.00000     0.            0.
(J3)  DESVELV       REDUCV        GREDUCV

      2.8220000     2.4440000     2.5220000     .82000000E-01 2.5720000     .18000000E-01 -0.
(K1)  XW            XD            XDG           SF            DFIN          THFIN         DANGLE

       1.00000    .93000  20.00000 STEEL WELDED         31.00000
(L1)  DOL         DI     GAGLTN      TUBMA             CONL

     DOUBLE FOOTED       ALUMINUM    119.00000
(M1)  FINTYP              FINMAT      CONP
```

FIGURE 5-6. NH_3 Model Sample Output

```
     0.              0.             .55000000      .20000000      2.4000000
(N1)  CFS             CDATC           ZINCC          CASTC          EPREPC

    FORM REMOVABLE       GALVANIZED  1.1250000
(P1) MEDTYP              MEDMAT      TW

     0.              2.0000000      .53000000      20.000000      10.130000      .48000000      5.0000000
(O1)  CRJ             CWJ            CMM            CN             CMW            CST            CBJ

     2.0000000       1.0000000      1.0000000      12.000000      118.00000      5.0000000       LINDE AL12
(R1)  KNS             VELV           DDC            GA            TKCT          TTDSWE            CONMAT

     1.0000000       50.000000      .80000000
(R2)  KNP             TLA          XQUALY

      1
(R3) LINDR

      55    85    -0
(S1) LPB   LPE MYEXT

    PFR
(T1)  PFMX
```

FIGURE 5-6. Continued

```
(200) I N P U T   S U M M A R Y   A N D   C A S E   D E S C R I P T I O N

(205)                                                       (210)
******************************                              ******************
OPTIONS SELECTED FOR THIS RUN                               STARTING CONDITIONS
******************************                              ******************

   UPPER LIMIT ON TURBINE TEMP 153.0 F
  FRICTION FACTOR AND HT TRFR COEFF BY PFR
  EXCHANGER DEPTH CONSTRAINED TO .GE.   3.7 TUBES                       TURBINE OUTLET TEMPERATURE, T1    105.000  F
  EXCHANGER LENGTH FIXED AT   80. FEET                          TERMINAL TEMPERATURE DIFFERENCE TTDI      5.000
  RUN WITH VELOCITY RECOVERY                                                   FRONTAL AREA, AFRON    30000.
  NTU CALCULATION IS BY EQUATION                                      WIDTH TO LENGTH RATIO, WLRAT     20.000

(220)                                                                  COEFFICIENTS FOR POLYNOMIAL CURVE FIT
                                                                  A0            A1             A2            A3            A4
HEAT RATE FACTOR AS FUNCTION OF BACKPRESSURE (INCHES OF MERCURY)   1.00000E+00 -4.90000E-03  2.67500E-03 -1.12500E-04
  TURBINE RATING BACKPRESSURE  3.50 INCHES OF MERCURY
  SITE ELEVATION  4410. FEET ABOVE SEA LEVEL

                                     (230) UNIT COSTS AND COSTING FACTORS
                                           ****************************

FUEL    81.000 CENTS/MMBTU   POWER PLANT CONSTRUCTION     440.000 $/KW                            FIXED CHARGE RATE      .174
LAND     0.000 $/SQ FT       REPLACEMENT POWER             24.000 MILLS/KWH                       CAPACITY FACTOR        .650
MOTORS 30.000 $ / HP         STEAM SUPPLY                 145.650 $/KW TURBINE OUTPUT      BASE THERMAL EFFICIENCY      .400
                             CAPACITY CHARGE          121000.000 $/MW               CONSTRUCTION COST MULTIPLIER     1.000
                                                                                      MAINT COSTS /CAPITAL COSTS      .010
```

FIGURE 5-6. Continued

```
(300) TO USED IN THE FOLLOWING CALCULATIONS IS =  50.000

(400) STEAM SUPPLY SCALING TO COMPENSATE FOR INCREASED HEAT RATE AT RATING BACKPRESSURE.  RBP= 3.50 IN. HG          HRFAC1=   1.010795
                 STEAM SUPPLY COST DIFFERENTIAL $     1583133.                        TURBINE COST DIFFERENTIAL         0. $/KW
  PLANT COST INCLUDING STEAM SUPPLY AND TURBINE $  441583133.     PLANT UNIT COST INCLUDING STEAM SUPPLY AND TURBINE       442. $

 ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(500) CONDITIONS AFTER SCALING PLANT AND STEAM SUPPLY  (BEFORE SCALING FOR FANS AND PUMPS)  HRFAC2=   1.000489
                                                                                            PSIZ12=   989.8629      MW
                                                                                            PLANC2=   437.1068      $/KW
                                                                                              SPSP=-2893239.72      $
                                                                                      BACK PRESSURE=   2.117190     IN. HG
                                                                                   SAT TEMPERATURE= 102.800000      DEG F
          AMMONIA FLOW RATE  10700122.2     LB MASS / HR
              HEAT REJECTED  .5124192456E+10 BTU/HR
            AIR   FLOW RATE  509948215.     LB MASS / HR
                 PUMP POWER  327.957958     KILOWATTS
                  FAN  POWER  14037.1252     KILOWATTS
           HX SURFACE AREA  37436708.2     SQ FT
              FRONTAL AREA  273000.000     SQ FT
       WIDTH/LENGTH RATIO  42.6552500
                      WIDTH       14510     TUBES
                      WIDTH  3412.50000     FEET
  CONDENSER HT TRFR AREA  532419.977     SQ FT TOTAL
    NO. CONDENSER TUBES  40674.3000
   CONDENSER TUBE LENGTH  50.0000000     FT.
          AMMONIA VELOCITY  .751707637     FT/SEC

 ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **  ** ** **

(550) SCALING FOR FAN AND PUMP POWER REQUIREMENTS                                   SCALING FACTOR  1.01457445
                                                                                             PSIZE  1000.00000      MW
                                                                                            PSIZ13  1004.28960      MW
                                                                                            PTOTAL  1014.57445      MW
                                                                                            PLANC3  443.477349      $/KW
                                                                                               SPC  443477349.      $
                                                                                              SPCD  3477349.25      $

          AMMONIA FLOW RATE  10856070.5     LB MASS / HR
              HEAT REJECTED  .5199554585E+10 BTU/HR
            AIR   FLOW RATE  515837572.     LB MASS / HR
                 PUMP POWER  332.737692     KILOWATTS
                  FAN  POWER  14241.7095     KILOWATTS
           HX SURFACE AREA  37982327.4     SQ FT
              FRONTAL AREA  275975.824     SQ FT
       WIDTH/LENGTH RATIO  43.2779412
                      WIDTH       14722     TUBES
                      WIDTH  3452.23530     FEET
  CONDENSER HT TRFR AREA  540173.703     SQ FT TOTAL
    NO. CONDENSER TUBES  41266.5010
   CONDENSER TUBE LENGTH  50.0000000     FT.
          AMMONIA VELOCITY  .751707537     FT/SEC
```

FIGURE 5-6. Continued

(600) NON-DELTA HEAT EXCHANGER DESCRIPTION

SUPPORT SPACING	5.000 FEET		ROOT DIAMETER	1.0720 INCH
TUBE PITCH NORMAL TO AIR FLOW	2.822 INCH		FIN THICKNESS	.0180 INCH
TUBE PITCH IN DIRECTION OF AIR FLOW	2.444 INCH		FIN DIAMETER	2.572INCH
TUBE DIAGONAL PITCH	2.822 INCH		FIN SPACING	.0820 INCH
LINER GAGE	20.0		FIN TYPE	DOUBLE FOOTED
LINER MATERIAL	STEEL WELDED		FIN MATERIAL	ALUMINUM
LINER THERMAL CONDUCTIVITY	31.000		FIN THERMAL COND.	119.0 BTU/(HR SQ FT DEG F)/FT
LINER TUBING OUTSIDE DIAMETER	1.0000 INCH		HEADER TYPE	FORM REMOVABLE
LINER TUBING INSIDE DIAMETER	.9300 INCH		HEADER LENGTH	12.000 FEET

*** UNIT COSTS ***

FIN BONDING	$ 0. /SQ FT		HEADER MATERIAL (GALVANIZED)	$ .570/LB
PROTECTIVE COATING	$ 0. /SQ FT		ROLLED JOINT, TUBE TO HEADER	$ 0.000/TUBE
ZINC FOR SPACERS	$ .5500 /LB		WELDED JOINT, TUBE TO HEADER	$ 2.000/TUBE
SPACER CASTING	$ .2000 /EA		TUBE AND PLUG HOLE PREPARATION	$ .500/HOLE
END PREPARATION	$ 2.400 /TUBE		NOZZEL AND ATTACHING	$ 20.000/HOLE
			BOLTED HEADER JOINT	$ 5.000/FT
			HEADER MACHINING AND WELDING	$ 10.130/FT
LINER TUBING	$ .32411 /FT		STRUCTURAL STEEL	$ .480/LB
PROTECTIVE COATING	$ 0. /FT			
FINNING	$ .51691 /FT			

FINNED TUBE	$ .94102 /FT
TUBE SPACERS	$ .57480 /FT

(700) HEAT EXCHANGER

127.2 FANS 1.0 PASS

(710)	AIR	AMMONIA
REYNOLDS NUMBER	8.67437E+03	9.21771E+03
VELOCITY (FT/SEC)	1.86005E+01	1.38713E+01
PRESSURE DROP (LBS FORCE/SQ IN)	1.46142E-02	3.65323E-01
HEAT TRFR COEFF (BTU/HR SQ FT DEG F)	7.80942E+00	1.28831E+03
FRICTION FACTOR	7.12148E-01	8.28820E-03
INLET TEMPERATURE (F)	5.00000E+01	9.58111E+01
EXIT TEMPERATURE (F)	8.49400E+01	9.58111E+01
RANGE (F)	3.49400E+01	
ITD1	3.40000E+00	
NTU	1.43942E+00	
OVERALL U (BTU/HR SQ FT DEG F)	5.63494E+00	
EFFECTIVENESS	.7627	
FIN EFFICIENCY	.8447	

FIGURE 5-6. Continued

```
(720) AFTR  .96834152
     ALPHA  156.72710
     SIGMA  .52445074

FITCO/ATOT .12536534

EQUATION (740) NTU EFFECTIVENESS CORRELATION USED

(750) AMMONIA FLOW RATE    10556070 LBM/HR
          AIR FLOW RATE   619557871 LBM/HR
         TUBES IN DEPTH        4.30 TUBES   (      .9 FT DEEP)
      TUBES ACROSS FRONT      14722 TUBES   ( 3462.2 FT WIDE)

                 LENGTH         80. FEET

           FRONTAL AREA      276978 SQ FEET
           SURFACE AREA    57982327 SQ FT
```

```
(730) WT OF AMMONIA          102579. LB
         WT OF TUBES        4735531. LB
         WT OF FRAME        1249901. LB
        WT OF HEADER        1104316. LB
        WT OF STRUCT        4962257. LB
           ROOF LOAD       11079153. LB     (   40.00 LB/SQ FT)

                 COOLING SURFACE $   8695739.
              HEADER AND NOZZEL $   1789607.
      BUNDLE ASSEMBLE AND FRAME $   2135848.
                          PLENUM $    125971.
(760)                    LOUVERS $         0.
                    HAIL SCREENS $    738004.
                       STRUCTURE $   3782728.
                      FOUNDATION $    523094.
      HEAT XCHR SHIP AND ERECT $   1514543.
```

FIGURE 5-6. Continued

****

(800) SUPPLY PIPING

****

CIRCULATION PIPING

HEADER	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS	EXPANSION JOINT(1)	FLANGES (3)
1	78.0	590.5	1240.1	92.9	84.1	79.5	101.7
2	72.0	0.0	0.0	80.2	68.2		
3	66.0	271.6	426.3	67.4	52.3		
4	54.0	0.0	0.0	44.3	26.7		
5	48.0	271.6	228.4	33.7	16.9		
6	36.0	0.0	0.0	16.1	0.0		

TOTAL COST OF CIRCULATION SUPPLY PIPING $2659. *10**3

QUADRANT SUPPLY PIPING

HEADER	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS
1	18.0	0.0	0.0	1.4	0.0
2	18.0	9.4	.9	1.4	0.0
3	18.0	9.4	.9	1.4	0.0
4	18.0	9.4	.9	1.4	0.0
5	18.0	9.4	.9	1.4	.8
6	12.0	10.3	.7	.7	0.0
7	12.0	10.3	.7	.7	0.0
8	12.0	10.3	.7	.7	0.0
9	12.0	10.3	.7	.7	0.0
10	12.0	10.3	.7	.7	0.0
11	12.0	10.3	.7	.7	.3
12	6.0	11.1	.4	.3	0.0

FITTINGS COST PER BUNDLE $ .340 *10**3

COST OF SUPPLY HEADERS PER QUADRANT $ 25.5 *10**3

TOTAL COST OF QUADRANT SUPPLY HEADERS $ 612. *10**3

FIGURE 5-6. Continued

```
                TOWER SUPPLY PIPING

NUMBER OF TOWERS                                    6
TOWER DIAMETER                                   184. FEET
DIAMETER OF PIPE FROM CIRCULATION
HEADER TO TOWER DISTRIBUTION HEADER        36. INCHES
DIAMETER OF TOWER DISTRIBUTION PIPE        18. INCHES

COST OF DISTRIBUTION PIPING PER TOWER  $ 284. *10**3

                              TOTAL COST OF TOWER DISTRIBUTION PIPING  $1705. *10**3


VAPOR DESIGN VELOCITY 150. FPS
MASS FLOW RATE 1.070E+07 LBM/HR
CONDENSER OUTLET TEMPERATURE  99.40 DEG. F
HEAT EXCHANGER INLET TEMPERATURE  95.81 DEG. F
SUPPLY PIPING PRESSURE DROP 1665.1 PSF
SUPPLY PIPING DYNAMIC HEAD 2535.3 FEET


                                 TOTAL SUPPLY PIPING SYSTEM COST   $ 6021. *10**3
```

FIGURE 5-6. Continued

....

(850) RETURN PIPING

....

CIRCULATION PIPING

HEADER	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS	EXPANSION JOINT(1)	FLANGES (3)
1	36.0	590.5	428.0	21.7	11.4	16.1	16.0
2	30.0	0.0	0.0	12.8	0.0		
3	30.0	271.6	134.4	12.8	6.8		
4	24.0	0.0	0.0	3.9	0.0		
5	24.0	271.6	72.0	3.9	2.2		
6	18.0	0.0	0.0	1.6	0.0		

TOTAL COST OF CIRCULATION RETURN PIPING $ 614. *10**3

QUADRANT RETURN PIPING

HEADER	DIAMETER (INCHES)	LENGTH (FEET)	COST IN 10**3 DOLLARS FOR PIPE	TEES	REDUCERS
1	12.0	0.0	0.0	.3	0.0
2	12.0	10.3	.9	.3	.4
3	5.0	11.1	.5	.4	0.0
4	5.0	11.1	.5	.4	0.0
5	5.0	11.1	.5	.4	0.0
6	5.0	11.1	.5	.4	0.0
7	5.0	11.1	.5	.4	0.0
8	5.0	11.1	.5	.4	0.0
9	5.0	11.1	.5	.4	0.0
10	5.0	11.1	.5	.4	0.0
11	5.0	11.1	.5	.4	0.0
12	5.0	11.1	.5	.3	0.0

FITTINGS COST PER BUNDLE $ .785 *10**3

COST OF RETURN HEADERS PER QUADRANT $ 21.2 *10**3

TOTAL COST OF QUADRANT RETURN HEADERS $ 508. *10**3

FIGURE 5-6. Continued

```
                TOWER RETURN PIPING

TOWER DIAMETER                               184. FEET
DIAMETER OF PIPE FROM CIRCULATION
HEADER TO TOWER DISTRIBUTION HEADER          18. INCHES
DIAMETER OF TOWER DISTRIBUTION PIPE          12. INCHES

COST OF DISTRIBUTION PIPING PER TOWER   $ 144. *10**3

                         TOTAL COST OF TOWER DISTRIBUTION PIPING  $ 865. *10**3


LIQUID DESIGN VELOCITY  15. FPS
MASS FLOW RATE 1.770E+07 LBM/HR
RETURN PIPING PRESSURE DROP  534.1 PSF
RETURN PIPING DYNAMIC HEAD   14.6 FEET


                              TOTAL RETURN PIPING SYSTEM COST   $ 2116. *10**3


                                 *********
                                 P U M P S
                                 *********


PUMP COST            $      250.*10**3
PUMP FITTINGS        $      399.*10**3
PUMP ELECTRICAL      $      189.*10**3

                                    PUMP STATION        $     837.*10**3


SEPARATOR COST       $       1211.*10**3
```

FIGURE 5-6. Continued

```
900 C O N D E N S E R

*** DESIGN CONDITIONS ***                               *** DESIGN OPERATING CONDITIONS ***

       STEAM TEMPERATURE 102.8 F                        OVERALL HEAT TRANSFER COEFFICIENT  2850.65 BTU/HR SQ FT DEG F
     AMMONIA TEMPERATURE  93.4F                           STEAM HEAT TRANSFER COEFFICIENT 5693.04BTU/HR SQ FT DEG F
    INLET AMMONIA VELOCITY      .75FT/SEC               AMMONIA HEAT TRANSFER COEFFICIENT13216.54BTU/HR SQ FT DEG F
   OUTLET AMMONIA VELOCITY    32.71FT/SEC                  WALL HEAT TRANSFER COEFFICIENT11516.65BTU/HR SQ FT DEG F
PRESSURE DROP THRU CONDENSER  227.0LB/SQ FT                FORCED CONVECTION COEFFICIENT  367.66BTU/HR SQ FT DEG F

*** CONDENSER DESCRIPTION ***                           *** CONDENSER COSTS ***

       NUMBER OF SHELLS      2                               TUBING MATERIAL $    4865842.01  (LINDE AL12
HEAT TRANSFER AREA PER SHELL .25621E+06 SQ FT                         SHELLS $    2179256.70
   NUMBER OF PASSES PER SHELL    1                           FIELD ERECTION $    1860205.55
              TUBE LENGTH    50      FEET
  NUMBER OF TUBES PER SHELL 20337                    TOTAL COST OF CONDENSER $      8905310.

1000 F A N S

FANS EQUIPPED WITH VELOCITY RECOVERY DIFFUSERS

*** DRIVE SYSTEM ***                                    *** FAN GEOMETRY ***

                                                                HUB DIAMETER    6.31 FEET
                                                              BLADE DIAMETER   25.00 FEET
                                                                 BLADE ANGLE   15.00 DEGREES
                                                         NUMBER OF BLADES/FAN    6.00
                                                              NUMBER OF FANS   127.

*** FAN COSTS ***                                       *** DESIGN CONDITIONS ***

                                                                                  (ACTUAL)               (ADJUSTED)
                                                      VOLUMETRIC FLOW RATE/FAN      1317879. CU FT/MIN
       BLADE AND HUB COST/FAN $     6966.            TOTAL VOLUMETRIC FLOW RATE   165264334. CU FT/MIN
               MOTOR COST/FAN $     6727.           TOTAL AIR PRESSURE DROP OF H.X.    .02059 PSI      .02506
SPEED REDUCER DRIVE COST/FAN $    15690.          PRESSURE DROP ACROSS H.X. BUNDLE    .01461 PSI      .01781
               STACK COST/FAN $    15107.           NON-RECOVERABLE VELOCITY HEAD    .00597 PSI      .00726
          ELECTRICAL COST/FAN $    27634.                     HORSEPOWER/FAN       150. HP         165.
                                                       FAN POWER REQUIREMENTS     18824. HP
*** ANNUAL OPERATIONAL COSTS $   3273437.

THE NUMBER OF TIME NO FAN WAS FOUND                6
THE NUMBER OF TIMES LOOP ESCAPE WAS REQUIRED =              1
```

FIGURE 5-6. Continued

(1100)

AMBIENT TEMP	TURBINE OUTLET TEMP	BACK PRESSURE	HEAT RATE FACTOR	RANGE	TTDP	HX LMTD	HX ITD	POWER GENERATED	CHANGE IN FUEL COST	COST TO REPLACE LOST CAP	POWER COST AT AMBIENT TEMP	PCT TIME AT AMBIENT TEMP	PORTION OF INCREMENTAL POWER COST
DEG F	DEG F	IN. HG		DEG F	DEG F	DEG F	DEG F	MEGAWATTS	MILLS/KWH	MILLS/KWH	MILLS/KWH	PERCENT	MILLS/KWH
100.0	152.9	9.10	1.0761	35.574	3.559	25.40	47.93	923.766	.10458	1.7096	3.7414	.150	.56121E-02
95.0	147.6	7.10	1.0599	36.242	3.527	25.21	47.53	943.230	.10458	1.3625	3.3943	1.000	.33943E-01
85.0	137.1	5.44	1.0344	35.702	3.474	24.84	46.82	966.790	.10458	.79703	2.8288	2.850	.80621E-01
75.0	127.0	4.16	1.0178	35.335	3.458	24.56	46.35	982.758	.10458	.41381	2.4456	8.220	.20103
65.0	117.1	3.18	1.0079	35.109	3.416	24.39	46.02	992.625	.10458	.17700	2.2086	9.130	.20166
55.0	107.6	2.43	1.0023	34.990	3.404	24.34	45.89	998.236	.10458	.42297E-01	2.0741	10.960	.22732
45.0	98.3	1.85	.9994	34.913	3.397	24.26	45.77	1001.176	.09546	0.	2.0237	11.190	.22645
35.0	89.2	1.41	.9981	34.885	3.394	24.22	45.71	1002.495	.08736	0.	2.0146	55.500	1.1382

(1200) FINAL INCREMENTAL COST 2.1149

CAPACITY CHARGE $ 8624165.25

FIGURE 5-6. Continued

CAPITAL COST SUMMARY			
COOLING SYSTEM CAPITAL			48199333.
TOWER		29945029.	
HEAT EXCHANGER	13341758.		
PLENUM	125971.		
LOUVERS	0.		
HAIL SCREENS	739034.		
FAN SYSTEM	9583930.		
STRUCTURE	5782725.		
FOUNDATION	325094.		
PIPING		10219210.	
CONDENSER		9035100.	
CAPACITY PENALTY			8624165.
PLANT SCALING			3477349.
STEAM SUPPLY		1599924.	
BASE PLANT		1887426.	
LAND			0.
TOTAL CAPITAL COST		$	60300853.

(1300)

UNIT ENERGY COST SUMMARY			
COOLING SYSTEM CAPITAL			1.47290
TOWER		.884516	
HEAT EXCHANGER	.438262		
PLENUM	.384947E-02		
LOUVERS	0.		
HAIL SCREENS	.225523E-01		
FAN SYSTEM	.295928		
STRUCTURE	.115594		
FOUNDATION	.987325E-02		
PIPING		.312284	
CONDENSER		.276099	
CAPACITY PENALTY			.263541
PLANT SCALING			.106263
STEAM SUPPLY		.485857E-01	
BASE PLANT		.576769E-01	
(1340) TOTAL CAPITAL COST			1.84270
(1350) ENERGY PENALTY			.937157E-01
(1360) ADDITIONAL BASE PLANT FUEL			.939423E-01
(1370) OPERATION AND MAINTENANCE			.846493E-01
(1380) TOTAL			2.11501

FIGURE 5-6. Continued

REFERENCES

1. B. C. Fryer et al., An Engineering and Cost Comparison of Three Different All-Dry Cooling Systems. BNWL-2121, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, September 1976.

2. P. A. Ard et al., Costs and Cost Algorithms for Dry Cooling Tower Systems. BNWL-2123, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, September 1976.

3. F. R. Zaloudek et al., A Study of the Comparative Costs of Five Wet/Dry Cooling Tower Concepts. BNWL-2122, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, September 1976.

4. B. R. Andeen and L. R. Glicksman, Dry Cooling Towers for Cooling Plants. Technical Report 73047-1, Engineering Project Laboratory, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA, February 1972.

5. K. K. Robinson and D. E. Briggs, "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes." Chemical Engineering Progress Symposium Series. 62(64): 1966.

6. D. E. Briggs and E. H. Young, "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes." Chemical Engineering Progress Symposium Series. 59(41): 1963.

7. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces, Part I: Heat Transfer and Pressure Drop Data. PFR Engineering Systems, Inc., Marina del Rey, CA, March 1, 1976.

8. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces, Part II: Data Analysis and Correlation. PFR Engineering Systems Inc., Marina del Rey, CA, June 15, 1976.

9. J. P. Rossie et al., Economics of the Use of Surface Condensers with Dry-Type Cooling Systems for Fossil-Fueled and Nuclear Generating Plants. R. W. Beck and Associates, Denver, CO, December 1973.

APPENDIX A

HEAT TRANSFER AND PRESSURE DROP CORRELATIONS AND DATA BASE

APPENDIX A

HEAT TRANSFER AND PRESSURE DROP CORRELATIONS AND DATA BASE

HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR FOR METAL FINNED TUBE HEAT EXCHANGERS

The performance of the metal finned tube heat exchanger is determined using the water-side and air-side heat transfer coefficients and friction factors. The performance is also influenced by the thermal resistance of the tube wall and fin, air-side and water-side fouling, and contact resistance at the liner-fin interface. For given values of the water and air flow rates and heat exchanger size, the prediction of the heat transfer coefficients and friction factors establishes the pumping power, fan power, and total heat load of the heat exchanger.

WATER-SIDE

The water-side performance parameters are calculated from established correlations discussed below.

Friction Factor

The friction factor is predicted from the Blasius equation[1] for flow in a circular conduit. The Blasius equation generally applies to the turbulent flow regime for Reynolds numbers up to 10^5. The range of Reynolds numbers observed in the parametric studies was 3×10^4 to 7×10^4, well within the limits of the Blasius equation. Other correlations are programmed for Reynolds numbers below 200 and above 10^5, but such conditions have not been shown to be economically attractive.

To evaluate the Reynolds numbers, the range of tube inside diameters was 0.680 to 1.908 inches. The optimized water velocity in the tubes ranged between 1.0 and 7.0 feet per second with the lower velocities characteristic of the larger diameter tubes.

Heat Transfer Coefficient

The water-side heat transfer coefficient is predicted with the Dittus-Boelter correlation[7] for turbulent flow in tubes. This correlation is applicable to Reynolds numbers greater than 10^4, a condition that was met for all observed cases in the parametric studies.

AIR-SIDE

The air-side performance parameters are not as well established as their water-side counterparts. The performance of a particular finned tube may vary significantly from the performance that would be predicted from any of the available correlations. This causes substantial concern because the performance of the heat exchanger, and thus, the accuracy of the economic analysis, is dominated by the air-side conditions.

It is the purpose of this section to present the two sets of correlations that were available for the optimization studies. The correlations will be evaluated by comparing them against each other and by comparing the performance predictions against actual finned tube experimental data. One set of correlations uses a friction factor developed by Briggs & Robinson[2] and a heat transfer coefficient developed by Briggs & Young.[3] The other set of correlations was developed by PFR Engineering Systems, Inc.,[4] at the request of PNL. The PFR correlations were used exclusively in the parametric studies of SOA and ammonia loop all-dry cooling systems using the metal finned tube heat exchanger. Both sets of air-side performance correlations apply only to staggered rows of finned tubes. Emphasis has been given to the analysis of the staggered tube arrangement because it has been shown to be economically more acceptable than the inline arrangement.[4,5]

In addition to developing the correlations, the PFR study was commissioned to collect performance data and to develop a figure of merit for the selection of finned tubes. The collection and review of finned tube air-side performance data is possibly the most thorough work of its kind available.[6] The PFR correlation was developed in consideration of all of this data.

The meaning of a "figure-of-merit" is that one could determine the economically optimum or near optimum finned tube unit geometry by considering only the performance of the finned tube, that is, without consideration of the economics of the entire cooling system. The attempt at PNL to develop a reliable and meaningful figure-of-merit has been unsuccessful. It is now evident that it is difficult, if not impossible, to successfully disassociate the economics of any one component from the remainder of the cooling system.

Friction Factor

Briggs and Robinson Correlation[2]

The friction factor correlation developed by Briggs and Robinson is

$$f = 18.93 \left(\frac{P_t}{D_r}\right)^{-.927} \left(\frac{P_t}{P_d}\right)^{.515} Re_r^{-.316} \qquad (1)$$

where

P_t = tube pitch normal to air flow
D_r = root diameter
P_d = tube pitch measured on diagonal between staggered tubes
Re_r = Reynolds number based on root diameter and the velocity at minimum cross section.

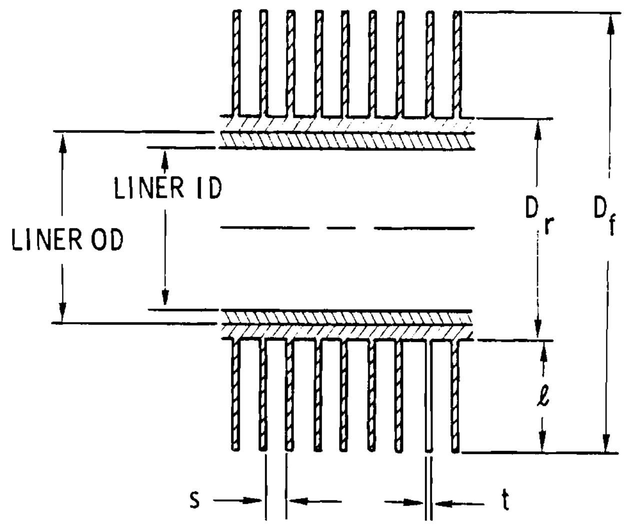
See Figure A-1 for a pictorial definition of these terms.

The friction factor was defined according to

$$f = \frac{\rho \; g_c \; \Delta P}{N \; G_m^2} \qquad (2)$$

where

ρ = air density
g_c = gravitational constant
ΔP = pressure drop across heat exchanger
N = tube rows in direction of air flow
G_m = air mass velocity at minimum cross section normal to direction of flow.



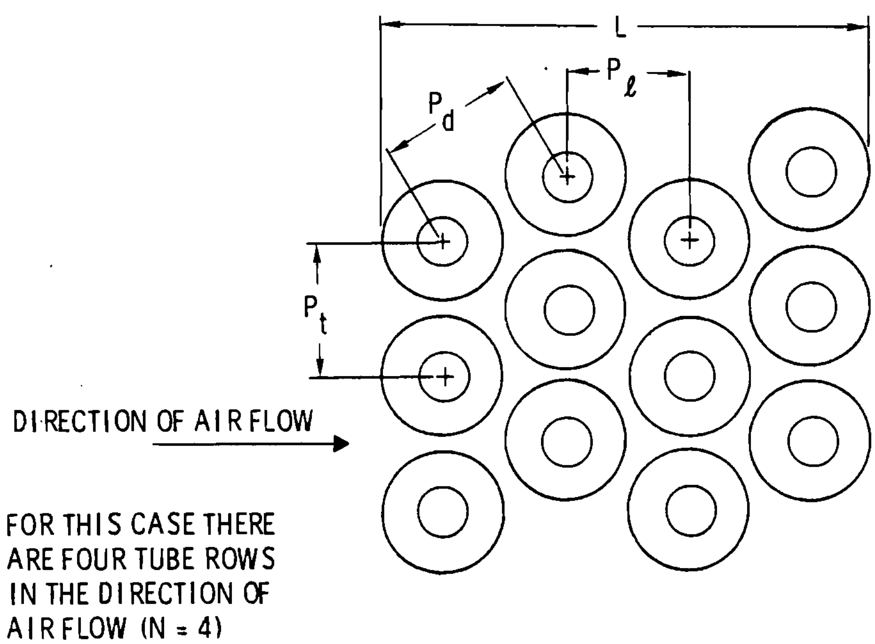


FIGURE A-1. Finned Tube Unit Geometry and Tube Pitch Parameters

The correlation, equation (1), applies to staggered tube arrangements. For equilaterally spaced tubes, the ratio of P_t/P_d is unity.

The range and variety of unit geometry variables for which the correlation was developed are relatively small. The range of the unit geometry variables was: root diameter (0.734 in. to 1.610 in.); fin diameter (1.561 in. to 2.750 in.); fin height (0.414 in. to 0.570 in.); fin thickness (0.0158 in. to 0.0235 in.); fin spacing (0.0729 in. to 0.1086 in.); and fins per inch (7.90 to 10.94). The tube pitch was varied between 1.687 inches and 4.500 inches with equilateral and isosceles arrangements being considered. Although the range of values is considered somewhat restricted, the near optimum unit geometry, as determined by the parametric studies, is bracketed by these values.

PFR Correlation

The friction factor correlations developed by PFR Engineering Systems, Inc.,[4] are:

$$f = \left(\frac{150}{Re_h} + \frac{1.8}{Re_h^{.2}}\right) \left(\frac{P_1}{D_h}\right)^{.35} \quad \text{for } \frac{P_1}{D_h} > 4, \tag{3}$$

and

$$f = 13.6\ Re_h^{-.3} \left(\frac{P_1}{D_h}\right)^{-.42} \quad \text{for } \frac{P_1}{D_h} \leq 4, \tag{4}$$

where

Re_h = Reynolds number based on hydraulic diameter and the velocity at minimum cross section

P_1 = tube pitch in direction of air flow

D_h = hydraulic diameter

PFR used the following equation to define friction factor:

$$f = \frac{2\ \rho\ g_c\ \Delta P}{N\ G_m^2} \tag{5}$$

The terms in the equation (5) were defined with equation (2). The correlations developed by PFR were multiplied by one-half so that they would be defined on a basis identical to that of the Briggs and Robinson correlation; that is $f = \rho g_c \Delta P / N G_m^2$. This allowed the comparison of the correlations.

The hydraulic diameter is defined as

$$D_h = \frac{4L}{N} \frac{A_c}{A_t} \tag{6}$$

where

L = depth of heat exchanger from leading edge of the first row of tubes to trailing edge of last row of tubes

A_c = minimum flow area between tubes normal to direction of air flow

A_t = heat transfer area per foot of finned tube

N = tube rows in direction of flow.

The PFR correlation was developed in order to include a large amount of data from many sources into a comprehensive, accurate correlation. An extensive quantity of data was reviewed to filter out the data that was considered to have been taken under questionable conditions. For example, friction factor data obtained from any module having less than five transverse tube rows was considered to have unacceptable edge effects. Some of the data was reanalyzed so that all the data could be reported on a common basis. All unit geometries were given equal weighting in the development of the correlation. The range of unit geometry variables was: root diameter (0.188 in. to 2.000 in.); fin diameter (0.375 in. to 3.250 in.); fin height (0.094 in. to 0.875 in.); fin thickness (0.008 in. to 0.080 in.); fin spacing (0.025 in. to 0.200 in.); and fins per inch (4.0 to 30.0).

Comparison of Friction Factor Correlations

The basic difference between the two friction factor correlations is the defining length for the Reynolds number. The PFR correlation uses hydraulic diameter. The hydraulic diameter is based upon the root diameter and the fin geometry parameters including fin thickness, fins per inch, and fin height. Briggs and Robinson use the root diameter which is the only unit geometry variable that appears explicitly in their correlation. While the Briggs and Robinson correlation does not explicitly include the effects of the fin geometry, changes in these parameters will cause a change in the friction factor predicted by the correlation. For example, if the fin thickness were increased and all other variables were unchanged, the air velocity through the minimum cross section would increase because of the increased restriction of the flow path. The velocity increase would result in a higher Reynolds number and, therefore, a slightly lower friction factor. The velocity increase would show up most dramatically as an increase in the air-side pressure drop.

While the use of the hydraulic diameter yields a correlation that is more explicitly sensitive to changes in some unit geometry variables, it is not certain that it yields a superior correlation. This may be especially true when the correlation is applied to near optimum unit geometries. Recall that the range of unit geometry variables which were used to develop the Briggs and Robinson correlation encompassed the near optimum values. The interest here is primarily in the near optimum unit geometries. In view of this emphasis, it is possible that the Briggs and Robinson correlation could be, in fact, the better correlation to use in this analysis. In other words, it could be postulated that in order to achieve general applicability, the PFR correlation was developed with a sacrifice in accuracy for specific unit geometries. Therefore, one should not draw conclusions regarding the applicability of either correlation based solely on the functional form.

To evaluate how the correlations compare, friction factors have been calculated for the near optimum unit geometry (NOUG) and for several other surfaces. These are compared in Table A-1 for a root diameter Reynolds number of 10^4, which is near optimum. (The PFR correlation can be converted to a root diameter Reynolds number form by replacing Re_h by $Re_r D_h / D_r$.)

These surfaces were selected from the PFR report on the basis of their approach to the near optimum unit geometry. The surfaces and the corresponding data are from several sources, including Briggs and Robinson (surfaces 75B

TABLE A-1. Comparison of Friction Factors Calculated from the Two Available Correlations Against Actual Finned Tube Data from Reference 6

Surface*	Briggs & Robinson	PFR	Actual Data
NOUG	.458	.771	--
75B	.363	.403	.331
78E	.512	.581	.430
80	.454	.558	.473
83	.513	.611	.704
84	.614	.803	.775
85	.509	.597	.743
86	.428	.620	.494
97	.400	.422	.395
108D	.542	.866	.613
109C	.404	.562	.394
120A	.518	.572	.840
143A	.456	.620	.467
144	.441	.855	.612
145A	.462	.689	.685
201	.513	.606	.756

*The surface numbers apply the the page numbers in the PFR report.(6)

and 78E). Surface 97 is the only surface that is not equilateral; the tube pitch in the direction of air flow (P_l) is 1.75 inches while the tube pitch normal to air flow (P_t) is 1.56 inches. Surfaces 144 and 145A are the closest available to an optimum unit geometry, while surface 143A is also very close.

The PFR friction factor exceeds the data for eleven of the fifteen observed cases. The Briggs and Robinson friction factor exceeds the data only four times. On this basis and by observing the result for the near optimum unit geometry one can conclude that the PFR correlation is significantly more conservative in terms of friction factor. If one were to develop a cooling system design using air flow characteristics predicted by a correlation, then the PFR correlation would be the likely choice. But, with the significant variation between the data and the correlations it can be concluded that the only way to obtain a reliable performance prediction, such as would be required in the actual design of a power plant cooling system, is to test the candidate finned tube under conditions similar to that of the proposed application.

Even though the friction factors are considerably different, the differences in the cooling system designs using either correlation are very subtle. Table A-2 presents optimized designs using either correlation with a near optimum set of input variables. The fan power is proportional to the product of the friction factor, the tube rows in depth, and the cube of the air velocity. Since the PFR friction factor is relatively high, the fan power must be kept down by reducing the tube rows in depth and the velocity. Reducing the velocity by only 15 percent reduces the cube of the velocity by nearly 40 percent. Both designs presented in Table A-2 require about the same heat transfer area. Therefore, the reduced heat exchanger depth for the PFR design results in a greater overall heat exchanger width which is reflected as an increase in the number of towers while the tower diameter remained nearly constant.

TABLE A-2. Comparison of Cooling System Designs Developed from the PFR Correlations and the Correlations of Briggs and Robinson and Briggs and Young

Tube Description: Liner OD = 1.00 in.
Fin OD = 2.57 in.
Fins/Inch = 10.0
Fin Thickness = 0.018 in.
Wrap-On Aluminum Fins
Carbon Steel Liner

	Air-Side Performance Correlation	
	PFR	Briggs & Robinson
DESIGN PARAMETERS		
Air Flow Rate, 10^8 lb/hr	8.65	7.54
Air Range, °F	25.2	28.9
Air Velocity, ft/sec	17.1	20.1
Friction Factor	0.850	0.473
$\Delta P_{air,HX}$, psi	0.0113	0.0126
$\Delta P_{air,discharge}$, psi	0.0064	0.0071
Fan Power, MWe	17.2	16.3
Fans Number/Dia, ft	37/60	31/60
Water Flow Rate, 10^8 lb/hr	2.08	2.10
Water Range, °F	25.2	24.9
Pump Power, MWe	6.35	6.15
LMTD, °F	20.4	18.6
ITD, °F	45.6	45.6
h, Btu/hr/ft^2/°F	7.23	7.62
U, Btu/hr/ft^2/°F	5.21	5.36
HX Heat Transfer Area, 10^6/ft^2	52.3	56.6
HX Width/Length, ft/ft	5710/80	4260/80
HX Depth Rows	3.27	4.74
Total HX Tubes	87200	94100
Towers Number/Dia, ft	12/148	8/166
COST SUMMARY		
Cooling System, mills/kWh	1.95	1.80
Capacity, mills/kWh	0.27	0.27
Scaling, mills/kWh	0.22	0.20
Energy, mills/kWh	0.09	0.09
Base Fuel, mills/kWh	0.15	0.15
O&M, mills/kWh	0.11	0.10
Total Incr. Cost, mills/kWh	2.80	2.61

The difference in air velocities and tower configuration between designs using the different correlations results in different fan system requirements. As a consequence of these basic differences in the trade-offs that determine the optimum designs, the incremental power production cost for a design developed with the PFR correlation is calculated to be about seven percent higher than when the Briggs and Robinson correlation is used. When comparing the metal finned tube systems (SOA and ammonia) to the plastic systems, this cost difference should be kept in mind. Use of the Briggs and Robinson correlation would widen the gap between the costs of the ammonia system and the plastic system.

Heat Transfer Coefficient

Briggs and Young Correlation

The correlation for the heat transfer coefficient developed by Briggs and Young is

$$h = 0.134 \frac{k}{D_r} Pr^{.333} \left(\frac{s}{l}\right)^{.200} \left(\frac{s}{t}\right)^{.1134} Re_r^{.681} \tag{7}$$

where

k = thermal conductivity of air
D_r = root diameter
Pr = Prandtl number of air
s = spacing between fins
l = fin height
t = fin thickness
Re_r = Reynolds number based on root diameter and air velocity at minimum cross section.

This correlation applies to both equilateral and isosceles staggered tube arrangements as it was concluded by Briggs and Young that tube pitch had a negligible effect upon heat transfer.

As with the Briggs and Robinson correlation for friction factor, this correlation was developed from an investigation of a relatively narrow range of unit geometry variables. However, the ranges include the near optimum unit geometry. The range of the unit geometry variables was: root diameter (.438 in. to 1.610 in.); fin diameter (.737 in. to 2.750 in.); fin height (0 in. to .656 in.); fin thickness (0.13 in. to .080 in.); fin spacing (.035 in. to .117 in.); and fins per inch (6.3 to 19.5). The tube pitch was varied between .964 inches and 4.370 inches.

PFR Correlation

The heat transfer correlation developed by PFR is

$$h = 0.290 \frac{k}{D_r} Pr^{.333} \left(\frac{A_o}{A_r}\right)^{-.17} Re_r^{.633} \tag{8}$$

where

k = thermal conductivity of air

D_r = root diameter

Pr = Prandtl number of air

A_o/A_r = ratio of total finned tube area to base tube area

Re_r = Reynolds number based on root diameter and air velocity at minimum cross section.

This correlation applies to both equilateral and isosceles staggered tube arrangements. Notice that the Reynolds number is defined here with the root diameter whereas the friction factor correlation of PFR uses a Reynolds number based on hydraulic diameter.

The data used in the development of the heat transfer correlation was screened in a fashion similar to that for the friction factor correlation. For example, all the tubes in the tube bank must have been heated in order to obtain usable heat transfer data; any other heating arrangement was considered to be unacceptable.

The range of unit geometry variables used to develop this correlation is about the same as is presented for the PFR friction factor, except in some cases where some finned tube data compiled by PFR was lacking in either heat transfer data or friction factor data. Regardless, the range of unit geometry is significantly broader here than the Briggs and Young heat transfer work. Similar arguments apply as to whether or not this is actually an improvement. That is, to achieve a correlation applicable to such a wide range of conditions, there may be a sacrifice of accuracy for specific unit geometries.

Comparison of Heat Transfer Correlations

An indication of the acceptability of either correlation is how they compare against each other and against actual data. Both correlations include the effects of fin thickness, fin pitch, and fin height, as well as the other pertinent variables. This was not the case with the friction factor correlations as the Briggs and Robinson correlation did not explicitly include the effect of these unit geometry variables.

Table A-3 compares heat transfer coefficients for a Reynolds number of 10^4 for the same surfaces as were compared for friction factor. In this case, surfaces have been excluded because heat transfer data were not available. These surfaces were chosen for comparison on the basis of their approach to the near optimum unit geometry (NOUG).

The agreement between the correlations and the data is obviously quite good. It appears that either correlation would be highly adequate for a generalized parametric study of this kind.

TABLE A-3. Comparison of Heat Transfer Coefficients Calculations Against Actual Finned Tube Data from Reference 6

Surface	B&Y	PFR	Actual Data
NOUG	8.54	8.16	--
80	10.01	10.66	9.43
83	9.80	9.70	9.13
84	6.30	6.40	6.75
85	8.85	8.82	7.92
86	9.32	9.94	8.88
97	16.23	16.41	21.11
143A	10.75	11.34	11.45
144	6.57	7.05	7.79
145A	9.27	9.37	9.98
201	8.85	8.82	9.19

The relatively high heat transfer coefficient of surface 97 in Table A-1 shows that this surface also has one of the lowest friction factors. This would seem to make surface 97 an ideal surface. However, surface 97 has the smallest root diameter (0.630 in.) and shortest fins (0.404 in.) of all the surfaces used in the comparison. This makes the heat transfer area per foot of finned tube relatively small. The cost of a unit of finned tube surface area is, therefore, relatively high.(7) The trade-offs that determine the optimum unit geometry go further than the simple trade-off between the unit cost and the performance of the surface. The trade-offs are dealt with more thoroughly in Section 6.2 of this report. As a result of considerations of unit cost and performance, as well as considerations of all other economic trade-offs, surface 97 is not necessarily superior to the others on an economic basis.

DISCUSSION

The parametric optimization studies indicate which value of a certain parameter results in the economically optimum cooling system design. By studying this parameter over a range of values on either side of the optimum it is possible to determine the sensitivity of costs and cooling system design to changes in this parameter. Determination of the optimum and the design and cost trends about the optimum of each independent parameter form the results of this study. It is generally true that similar trends and optimums are observed regardless of which set of air-side performance correlations has been used. This holds as long as each correlation is applied within the ranges of the variables for which it was developed.

For the purposes of development of an accurate code, it was difficult to assess which set of correlations would be more acceptable. If the attempt was to design a cooling system that was guaranteed to have a minimum heat rate capability, then one would certainly use actual data after scoping the problem using the more conservative PFR correlations. However, it is the purpose of the code to evaluate the relative economics of different designs. To do this most effectively, it would be necessary to have the same degree of conservatism as it affects the cost of power in all the cost models, physical models, assumptions, and correlations. This same conservatism would have to be applied to the descriptions of the advanced systems also. It is not realistic to expect that this could be done. The best available models were used with the hope that the conservatism in any one model was not so unbalanced that it would prevent an accurate identification of the true optimum. In this regard, either set of air-side performance correlations is probably acceptable.

HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR CORRELATIONS FOR PLASTIC TUBE HEAT EXCHANGERS

Heat transfer coefficient and friction factor correlations used in the optimization of a plastic tube dry cooling tower system for an electric generation plant are presented in this section. In the optimization procedure used for this system, the correlations presented here were the basis for determining the size of the cooling tower system and the mode in which it operated. The total cost of the system is highly dependent upon the values obtained from these correlations.

The heat transfer and pressure drop correlations presented here fall into two categories: air-side and water-side. The air-side is further subdivided into inline or staggered tube spacing (pitch) categories.

The nomenclature for these correlations is on page A-22.

WATER-SIDE

Friction Factor

The water-side pressure drop is determined through the use of a Fanning friction factor relationship for smooth tubes. The Fanning pressure drop equation is used in the following form:

$$\Delta p = 4f \frac{L}{D} (\rho V^2/2g_c) \tag{9}$$

The friction factor is provided by correlations covering three Reyolds number ranges:

- Laminar Range, ($0 < Re < 2000$):

$$f = 16/Re \tag{10a}$$

- Transitional ($2000 < Re < 1 \times 10^5$):

$$f = 0.0791/(Re)^{0.25} \tag{10b}$$

- Turbulent ($Re > 1 \times 10^5$):

$$f = 0.046/(Re)^{0.20} \tag{10c}$$

The water flow rate for the optimum plastic tubed system is in the transitional flow regime.

Heat Transfer Coefficient

The water-side heat transfer coefficient is estimated through the use of the Dittus-Boelter correlation[8] or the laminar flow relationship of W. Nusselt[9] depending on the flow regime of the cooling water.

Those relationships are expressed in the following form:

- Dittus-Boelter correlation (transitional-turbulent, Re > 2400):

$$h = 0.023(Re)^{0.80}(Pr)^{0.40}\ k/D \tag{11a}$$

- Nusselt Relationship (laminar, Re < 2400):

$$h = 3.66\ k/D \tag{11b}$$

For the range of parameters selected, the Reynolds number and velocities through the tubes vary from 3000 to 5000 and 1.00 to 4.00 ft/sec, respectively. This is in the region of transitional flow. The heat transfer coefficient is highly dependent upon the state of turbulence of the cooling water in the tube. The turbulent flow is preferred because it gives a higher heat transfer coefficient for only a slight increase in pressure drop. This is the case which is most likely to exist due to the high amount of turbulence which will exist in the headers. Only when the Reynolds number of the tube flow is below 2400 is Equation (11b) used.

AIR-SIDE

A literature search on the subject of air side heat transfer coefficient and friction factor through banks of bare tubes was provided to the Dry Cooling Tower Program at PNL by Dr. Knudson of Oregon State University. Reference 10 provided the most complete information. Prior to this literature search, correlations provided in Reference 11 had been used as the basis for the analysis.

Pressure Drop

The air-side pressure drop through a bank of tubes is determined through the use of a Fanning friction factor relationship,

$$\Delta p = 4fN(\rho V_{max}^2/2g_c) \tag{12}$$

The friction factor for both the inline and staggered tube arrangement were obtained from McAdams.[11]

Incorporation of information from Reference 9 into the optimization code would have required converting friction factor data from graphic to tabular form. Because this would have introduced an additional error into the computation scheme, the decision was made to use McAdams' information in the optimization code.

Staggered Tube Arrangement

The friction factor relationship provided by McAdams for a staggered tube arrangement is

$$f = \left(0.25 + \frac{0.1175}{(X_T - 1)^{1.08}}\right) (Re_{max})^{-0.16}. \tag{13a}$$

Inline Tube Arrangement

Inline tube arrangement friction factor is provided by

$$f = \left(0.044 + \frac{0.08 X_L}{(X_T - 1)^N}\right) (Re_{max})^{-0.15} \tag{13b}$$

in which $N = 0.43 + (1.13/X_L)$.

In the above relationships, X_L is the ratio of the longitudinal pitch to the tube outside diameter. X_T is the ratio of the transverse pitch to tube outside diameter. All pitches are measured from centerline to centerline.

Equations (13a) and (13b) are based on data which cover a Reynolds number range of 2000 to 4×10^4.

For the range of parameters selected in the optimization of the plastic tube advanced concept, the air-side Reynolds number varied from 500 to 5000. The air velocity at the minimum cross-sectional flow area ranged from 7.0 to 15.0 feet per second. The Reynolds number range lies within the laminar and transitional flow regimes.

The Reynolds number range straddles the lower limit of the friction factor relationships for both staggered and inline tube arrangements. To verify that these relationships give friction factors within a reasonable limit of actual data in the region less than 2000, they were compared against Zukauskas' data.[10] It was found that down to a Reynolds number of 500, friction factors calculated by equations (13a) and (13b) were within acceptable limits of actual data.

Heat Transfer Coefficient

The air-side heat transfer correlations for a bank of bare tubes are taken from McAdams[11] and Zukauskas.[10] The Zukauskas correlations appear to be based on more complete data than those of McAdams. Although both sets of correlations are provided in the optimization code, only Zukauskas' correlations were used in the optimization of the plastic tube advanced concept.

Staggered Tube Arrangement

McAdams recommends the relationship below for gases flowing normal to banks of staggered tubes for a Reynolds number from 2000 to 32000:

$$h = 0.33(Pr)^{0.33}(Re)^{0.60}\ k/D \tag{14}$$

Zukauska recommends the following set of relationships for determining the heat transfer rates in banks of tubes:

- Laminar, $10 < Re < 100$:

$$h = 0.90(Re)^{0.40}\ (Pr)^{0.36}\ k/D \tag{15a}$$

- Transitional, $1000 < Re < 2 \times 10^5$:

  For $a/b < 2$,

$$h = 0.35(a/b)^{0.2}(Re)^{0.60}(Pr)^{0.36}\ k/D \tag{15b}$$

  For $a/b > 2$,

$$h = 0.40(Re)^{0.60}(Pr)^{0.36}\ k/D \tag{15c}$$

- Turbulent, $Re > 2 \times 10^5$:

$$h = 0.022(Re)^{0.84}(Pr)^{0.36}\ k/D \tag{15d}$$

The Reynolds number region from 100 to 1000 is a transitional flow regime from laminar to turbulent, which is not covered by any correlation in Zukauskas' article. For completeness, correlations were derived for this region through use of an equation of the following form:

$$h = c(Re)^m(Pr)^{0.36}\ k/D$$

The constants c and m were determined by forcing this equation to fit values obtained at Re of 100 and 1000 by equations (15a-c). The resulting equations are:

- $100 < Re < 1000$

  For $a/b < 2$,

$$h = c(Re)^m(Pr)^{0.36}\ k/D. \tag{16a}$$

  where

$$c = 0.3755(100)^{(-0.02\ \log_{10}\ (a/b))}$$

$$m = 0.58983 + 0.2\ \log_{10}\ (a/b)$$

  For $a/b \geq 2$,

$$h = 0.28(Re)^{0.65}(Pr)^{0.36}\ k/D \tag{16b}$$

Inline Tube Arrangement

Only one set of correlations is used for the heat transfer in an inline tube bundle. These were taken from Zukauskas.[10] Again, the region from a Reynolds number of 100 to 1000 was not covered by a specific relationship. A relationship was derived from data obtained from adjacent correlations.

- Laminar, $10 < Re < 100$:

$$h = 0.80(Re)^{0.40}(Pr)^{0.36}\ k/D \tag{17a}$$

- Transitional:

For $100 < Re < 1000$,

$$h = 0.293(Re)^{0.618}(Pr)^{0.36}\ k/D \tag{17b}$$

For $1000 < Re < 2 \times 10^5$,

$$h = 0.27(Re)^{0.63}(Pr)^{0.36}\ k/D \tag{17c}$$

- Turbulent, $Re > 2 \times 10^5$,

$$h = 0.021(Re)^{0.84}(Pr)^{0.36}\ k/D. \tag{17d}$$

The range of Reynolds numbers found to be optimum for the plastic tube advanced concept is from 500 to 5000. This range includes the region in which heat transfer correlations were not available from Zukauskas. As described earlier, this region was accommodated by relationships derived from data obtained from adjacent correlations.

NOMENCLATURE

f - Fanning friction factor

h - heat transfer coefficient (Btu/hr-ft^2-°F)

k - thermal conductivity (Btu/hr-ft-°F)

μ - viscosity (lbm/ft-hr)

C_p - specific heat (Btu/lbm-°F)

ρ - density (lbm/ft^3)

V - velocity (ft/sec)

L - tube length

D - diameter

N - number of tubes in depth

g_c - gravitational constant

a - transverse pitch

b - longitudinal pitch

X_T - a/D

X_L - b/D

Re - $\rho VD/\mu$

Pr - $C_p\mu/k$

Nu - hD/k

HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR CORRELATIONS FOR THE AMMONIA COOLING LOOP

CONDENSER/REBOILER

The condenser/reboiler is similar in design to that of a conventional water shell and tube condenser. The differences are:

- ammonia instead of water passes through the tubes, and
- header wall thickness is increased to provide sufficient strength to withstand the pressure of the ammonia.

Condenser designs were considered with the smooth tubing used in commercial condensers built and used today and with enhanced tubing. The enhanced tubes have better heat transfer coefficients obtained by use of sintered metal on the inside surface and by use of a proprietary outside surface developed by LINDE Division of Union Carbide.

The nomenclature for correlations given in this section is found on page A-36.

Heat Transfer Correlations

Smooth Tubing

The design of the condenser/reboiler with smooth tubing is accomplished by determining the number and length of tubes in the condenser. This is done by finding the average overall coefficient for a specific temperature difference, heat load, and tube length. To find the overall coefficient the average inside and outside heat transfer coefficients of the horizontal tubes must be determined. The outside coefficient must take into account steam velocity, condensate film thickness, and condensate film buildup and removal due to splashing down from one tube to the next in a bank of tubes. The following correlation, used to handle these effects, was derived by Nusselt[12] and modified by Kern.

$$h_s = .728 \left(\frac{g\ \rho_L(\rho_L - \rho_v)k_L^3\ h_{fg}}{D_o\ \mu_L\ \Delta T_s}\right)^{1/4} \left(\frac{1}{N}\right)^{1/6} \qquad (18)$$

The original equation by Nusselt was derived with the number of tubes in depth (N) in the tube bank taken to the 1/4 power. Kern later modified the equation by changing it to the 1/6 power so that conditions in the condenser could be better predicted.

The steam-side coefficient obtained from equation (12) by the optimization code typically ranges from 1000 to 1500 Btu/hr-°F-ft^2 for the smooth tubing.

The inside heat transfer coefficient of the smooth tubing is composed of the sum of nucleate boiling and forced convection heat transfer:

$$h_i = h_{fc} + h_B \tag{19}$$

The forced convection coefficient in equation (19) is found by the correlation developed by Dittus-Boelter:[8]

$$h_{fc} = .023\, Re_L^{.8}\, Pr_L^{.4}\, k_L/D_i \tag{20}$$

The range of values obtained from this correlation by the optimization code is from 200 to 500 Btu/hr-°F-ft^2 with liquid inlet velocities into the tubes varying from 0.5 ft/sec to 1.5 ft/sec. The boiling coefficient is obtained from the correlation developed by Rohsenow:[13]

$$h_B = (Cp_L/h_{fg}(.015)Pr_L^{1.7})^3 \quad \frac{\mu_L\, h_{fg}\, \Delta T_a^2}{\sqrt{\dfrac{\sigma}{\rho_L - \rho_v}}} \tag{21}$$

The boiling coefficient ranges from 800 to 1200 Btu/hr-°F-ft^2 in the optimization code.

Enhanced Tubing

Design of the condenser/reboiler with enhanced tubing is done by the same method as for smooth tubing. Different heat transfer correlations are used to obtain the inside and outside coefficients of the horizontal tubes. The correlations used for the enhanced surfaces are derived from experimental data obtained from Union Carbide Corporation, LINDE Division. The correlations are derived from two data points of heat flux versus temperature difference.

The steam-side correlation is derived from experimental data[14] for steam condensing at one atmosphere on the outside of a single horizontal tube. The correlation is used for pressures below one atmosphere with the assumption that the effect of pressure does not have a large influence on the condensation coefficient. The influence of condensate film buildup due to the film of one tube falling on another tube in a tube bank is taken care of by adding to the experimental correlation the relationship for the tube bank depth (N) from equation (18). The following correlation is used to describe the condensation process.

$$h_s = 16700 \, \Delta T_s^{-.572} \left(\frac{1}{N}\right)^{1/6} \tag{22}$$

The steam-side coefficients obtained by the code from equation (22) range in values from 3000 to 6000 Btu/hr-°F-ft^2.

The heat transfer correlation for the ammonia-side of the enhanced tubing includes, as before, the effect of two processes; nucleate boiling and forced convection heat transfer. It is assumed that the forced convection correlation does not change from equation (21). The boiling correlation does change with the change in the tubing. The boiling correlation used was developed from experimental data[14] for a piece of the enhanced surface heated in a pool of ammonia so that boiling occurred at different temperatures of the surface. The correlation takes the form:

$$h_B = 18000 \, \Delta T_a^{1.72}. \tag{23}$$

The boiling coefficients obtained from equation (23) by the optimization code range in values from 14,000 to 20,000 Btu/hr-°F-ft^2.

Pressure Drop Correlations

Smooth Tubing

The correlation used to determine the pressure drop in the smooth tubes of the condenser/reboiler was developed by Lockhart and Martinelli[15] for two-phase flow:

$$\Delta P = \phi_L^2 \, f_L \, \frac{L}{D_i} \, \frac{G_L^2}{\rho_L} \tag{24}$$

The Martinelli function (ϕ_L) is provided by a curve fit of the Lockhart-Martinelli parameter (X):

$$X = f_L \; G_L^2 \; \rho_v / f_v \; G_v^2 \; \rho_L \tag{25}$$

The Fanning friction factor (f) is provided by four separate correlations covering three different regimes of flow. The regimes are differentiated by four ranges of Reynolds numbers.

- Laminar Regime ($0 < Re \leq 1600$)

$$f = 16/Re \tag{26}$$

- Transition Regime ($1600 < Re \leq 3950$)

$$f = .01 \tag{27}$$

- Turbulent Regime ($3950 < Re \leq 52{,}000$)

$$f = .0791/Re^{.25} \tag{28}$$

- Turbulent Regime ($52{,}000 \leq Re$)

$$f = .046/Re^{.20} \tag{29}$$

The flow regime usually found by the code is the turbulent flow regime. The pressure drops are within one-half to two pounds per square inch.

Enhanced Tubing

The method used to determine the pressure drop through the enhanced tubes of the condenser/reboiler is the same as for the smooth tubes.

TOWER

Arrangements

The arrangements of the tower depend on the type of flow inside the tube of the tower. The type of flow also affects the cost of the heat exchangers, the tower structure, and the piping to and in the towers. The two types of flow in the tubes are countercurrent and cocurrent.

Countercurrent Flow

The flow arrangement in the dry cooling tower in the inside of the tubes is important in terms of cost. If a counterflow arrangement as shown in Figure A-2 could be used, the cost associated with the structure and the piping could be greatly reduced as shown by a probable structural and piping arrangement in Figure A-3. One problem with counterflow situations is that entrainment of liquid (flooding) at the liquid-vapor interface can occur if certain conditions are present. For ammonia with its low surface tension and high vapor density, the condition of flooding does occur at low superficial vapor velocities (2-3 ft/sec) which requires that the length of a one-inch diameter tube be not above 17 feet in the tower if flooding is not to occur. Because the tubing in the tower would be longer than the minimum nonflooding length and because of the uncertainty of what happens to the inside heat transfer coefficient (enhancement or degregation) when flooding occurs, the countercurrent flow scheme was not pursued any further.

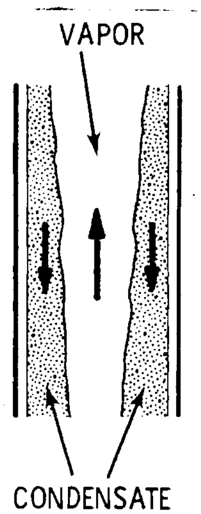


FIGURE A-2. Countercurrent Flow

Cocurrent Flow

The three cocurrent flow conditions possible in a condensation process are shown in Figures A-4 through A-6. The stratified flow condition in Figure A-4 results when low vapor velocities occur in the tubes which are either sloped or horizontal. Figure A-5 shows annular flow which occurs when high vapor velocities are prevalent in horizontal tubes and low and high vapor velocities in vertical tubes. Figure A-6 shows annular flow with mist in the vapor core which occurs at very high vapor velocities in both horizontal and vertical tubes. The last situation does not occur in an air-cooled condensation process because of the high heat transfer rate needed to obtain high vapor velocities.

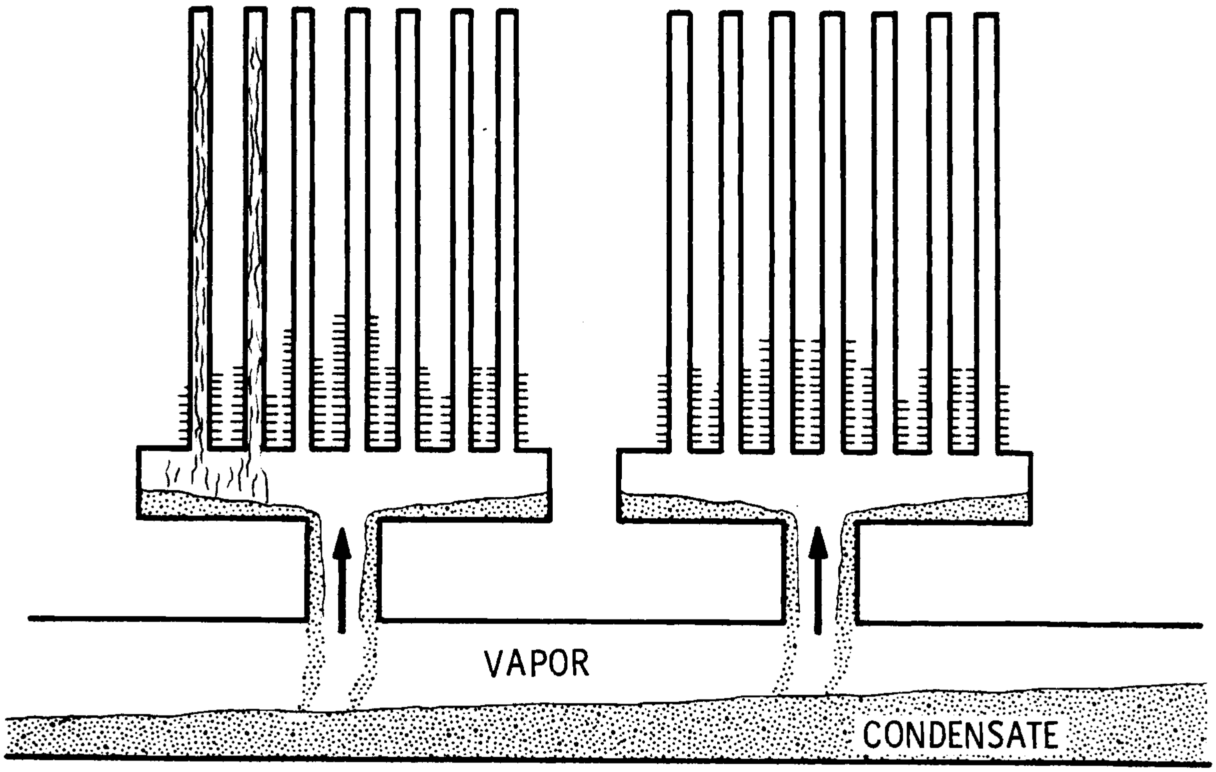


FIGURE A-3. Probable Structural and Piping Arrangement for Countercurrent Flow

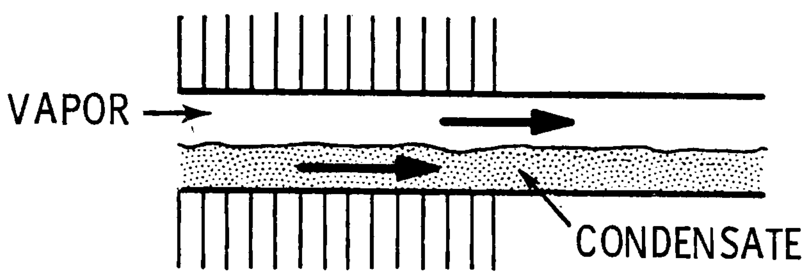


FIGURE A-4. Stratified Flow

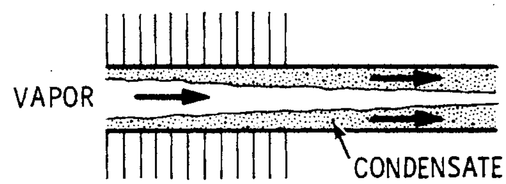


FIGURE A-5. Annular Flow

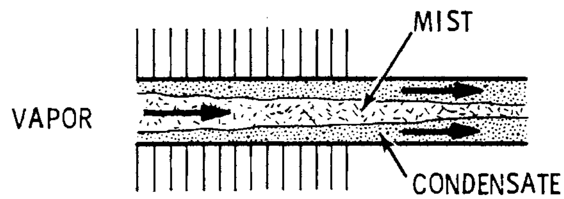


FIGURE A-6. Annular Flow with Mist

Tower Configuration

Two types of towers can be used when cocurrent flow is on the inside of the tubes. One is the rectilinear tower, shown in Figure A-7, which can handle both horizontal and sloped tubing. Because all or some of the weight of the heat exchanger plus the coolant must be supported by the structure of the tower, the structural cost for the rectilinear towers is greater than for the round tower. As shown in Figure A-8, the round tower supports the heat exchanger along the tower periphery, decreasing the structural cost.

When the rectiliner tower was investigated for the SOA system, it was found slightly more costly than the circular tower. So the analysis for the SOA system was mainly done with circular towers. When the difference between the circular and the rectilinear towers for the PLASTIC system was evaluated, a definite savings with the circular towers was realized.

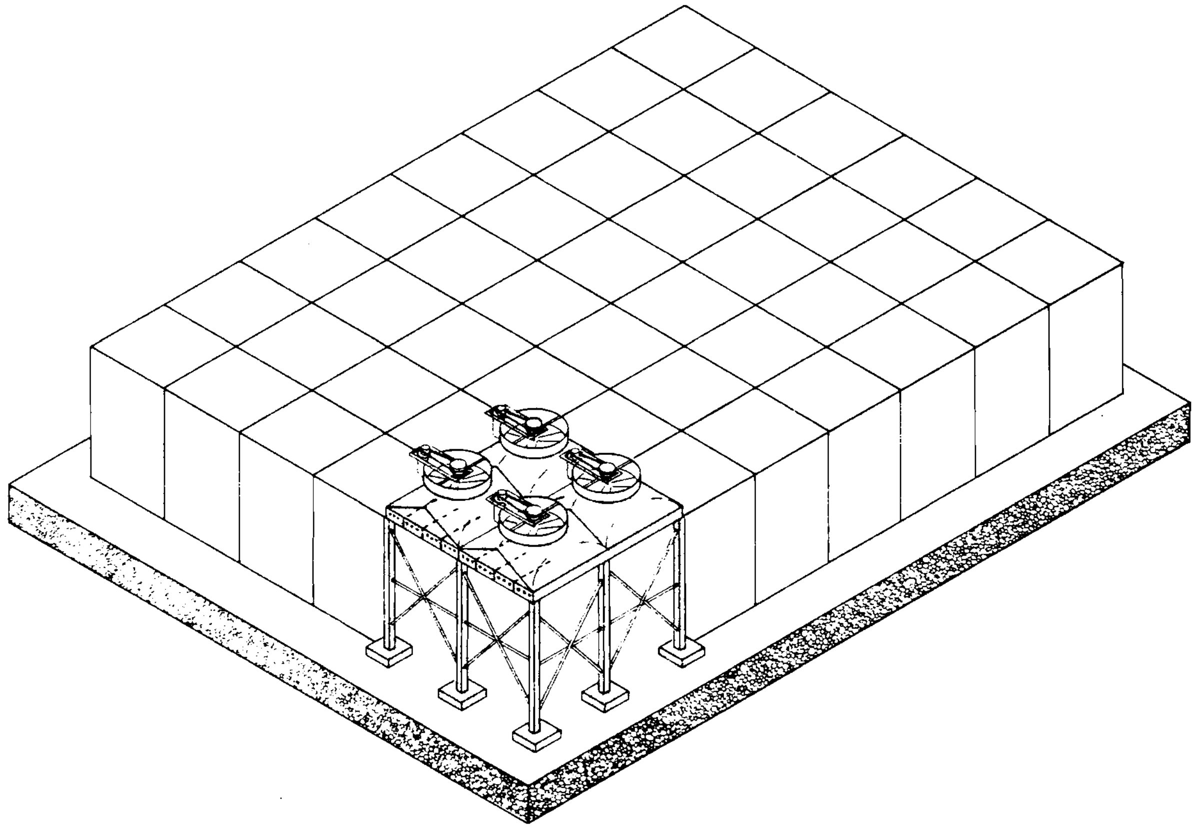


FIGURE A-7. Rectangular Tower

Because the major part of the analysis for the SOA and the PLASTIC systems was done with the circular tower, it was decided to analyze the NH_3 system with circular towers. The NH_3 system was not analyzed with rectilinear towers.

Heat Transfer Correlations - Tower Heat Exchanger

Air-Side

The air-side heat transfer correlations are the same as those described for the metal finned tube system.

Ammonia-Side

Because the condensation of ammonia only takes place in the vertical section of the tubes of the heat exchanger, three flow regimes must be modeled to find the correct heat transfer coefficient. All the regimes depend on the thickness of the average condensate layer and the velocity of the vapor entering the tube. This depends on whether gravity or vapor shear is the predominating force.

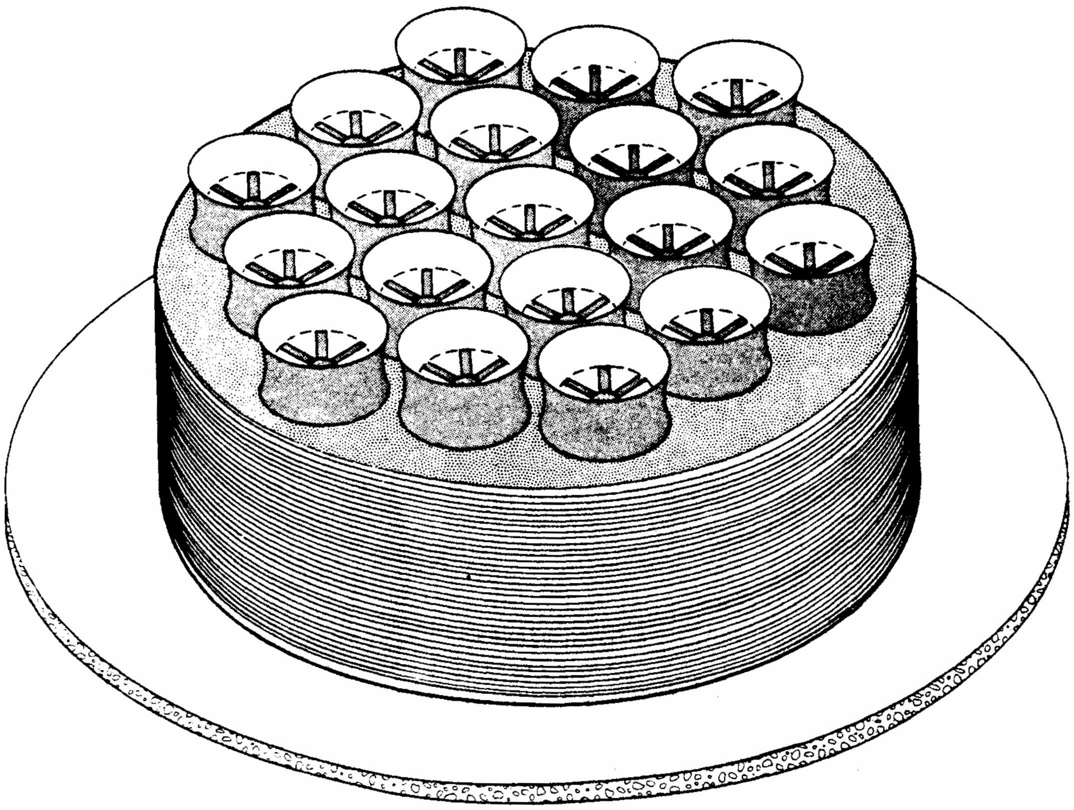


FIGURE A-8. Circular Tower

When the average condensate thickness is small and the velocity of the vapor is low (gravity force predominates over the vapor shear force) the resulting laminar film condensation can be predicted with the classical analysis of Nusselt as embodied in the following equation.

$$h_i = 1.47\ K_L\ (g \sin \Theta\ \rho_L\ (\rho_L - \rho_V)/\mu_L^2)^{1/3} / (Re_L(1 - X_o))^{1/3} \qquad (30)$$

If the condensate thickness is large or the velocity of the vapor is of medium velocity (gravity force still predominates over the vapor shear force) the resulting turbulent film condensation can be predicted by the correlation developed by Kirkbride:[16,17]

$$h_i = .0077\ K_L\ (g \sin \Theta\ \rho_L\ (\rho_L - \rho_V)/\mu_L^2)^{1/3}\ (Re_L\ (1 - X_o))^{.4}\ . \qquad (31)$$

When the annular ring of condensate is mainly controlled by the vapor shear force (predominating over the gravity force), the correlation by Boyko and Kruzhilin[18] was used for the turbulent liquid flim heat transfer coefficient:

$$h_i = .024\ K_L\ Re_L^{.8}\ Pr_L^{.43}/D_i\ \frac{\sqrt{1 + (\rho_L/\rho_V - 1)X_i} + \sqrt{1 + (\rho_L/\rho_V - 1)X_o}}{2} \qquad (32)$$

The range of values found by the optimization code for the inside coefficient is from 1000 to 1600 Btu/hr-°F-ft^2. The Reynolds number based on the liquid varies from 5000 to 15,000 with the entering vapor velocity ranging from 10 to 20 feet per second. The correlation most often used by the code to define the inside coefficient is equation (31).

Pressure Drop Correlation - Tower Heat Exchanger

The two-phase pressure drop in the tubes of the heat exchanger is found by the same set of equations as for the pressure drop in the condenser/reboiler. The only difference is that the Martinelli function (ϕ_L) is provided by a functional relationship of the Lockhart-Martinelli parameter (X) and not as a curve fit of the parameter.

PIPING

Pressure Drop Correlations

The pressure drop correlations in the optimization code are used to calculate the pressure drops for the supply and return piping. The supply piping transports vapor while the return line transports liquid.

The pressure drop in the vapor line is found on an incremental basis where each length of pipe of a particular diameter is divided into ten sections. The pressure drop across each section is calculated; then new vapor properties are found to be used to find the pressure drop in the next section.

The pressure drop in the liquid line is calculated for the whole length of pipe of one specific diameter.

Supply Piping

The pressure drop in the supply piping of the cooling loop is calculated by using the Fanning equation:

$$\Delta P = 4f_v \frac{L}{D_i} \frac{G_v^2}{2\rho_v} \tag{33}$$

The pressure drops found by the optimization code vary from approximately 10 to 20 pounds per square inch.

Return Piping

The pressure drop in the return piping is also calculated by using the Fanning equation.

$$\Delta P = 4f_L \frac{L}{D_i} \frac{G_L^2}{2\rho_L} \tag{34}$$

The pressure drops for the return piping vary from about 2 to 6 pounds per square inch.

Temperature Drop Correlation

Shown in Figure A-9 is a pressure-enthalpy diagram of the vapor dome for ammonia and a typical cycle ammonia passes through when it goes around the cooling loop. The diagram shows five important points of the cycle. Point A to Point B shows what happens to the ammonia in the condenser/reboiler. Point C to Point D shows the condensation of ammonia in the tower.

Point B to Point C is what occurs in the piping from the condenser/reboiler to the dry tower. The process is controlled by the pressure drop in the pipeline. Since the transfer of heat through the pipe walls is negligible and no pumping is done on the vapor the process through the pipe is isenthalpic. With the flow being isenthalpic and having a pressure drop the ammonia encounters a temperature drop from the beginning to the end of the supply piping.

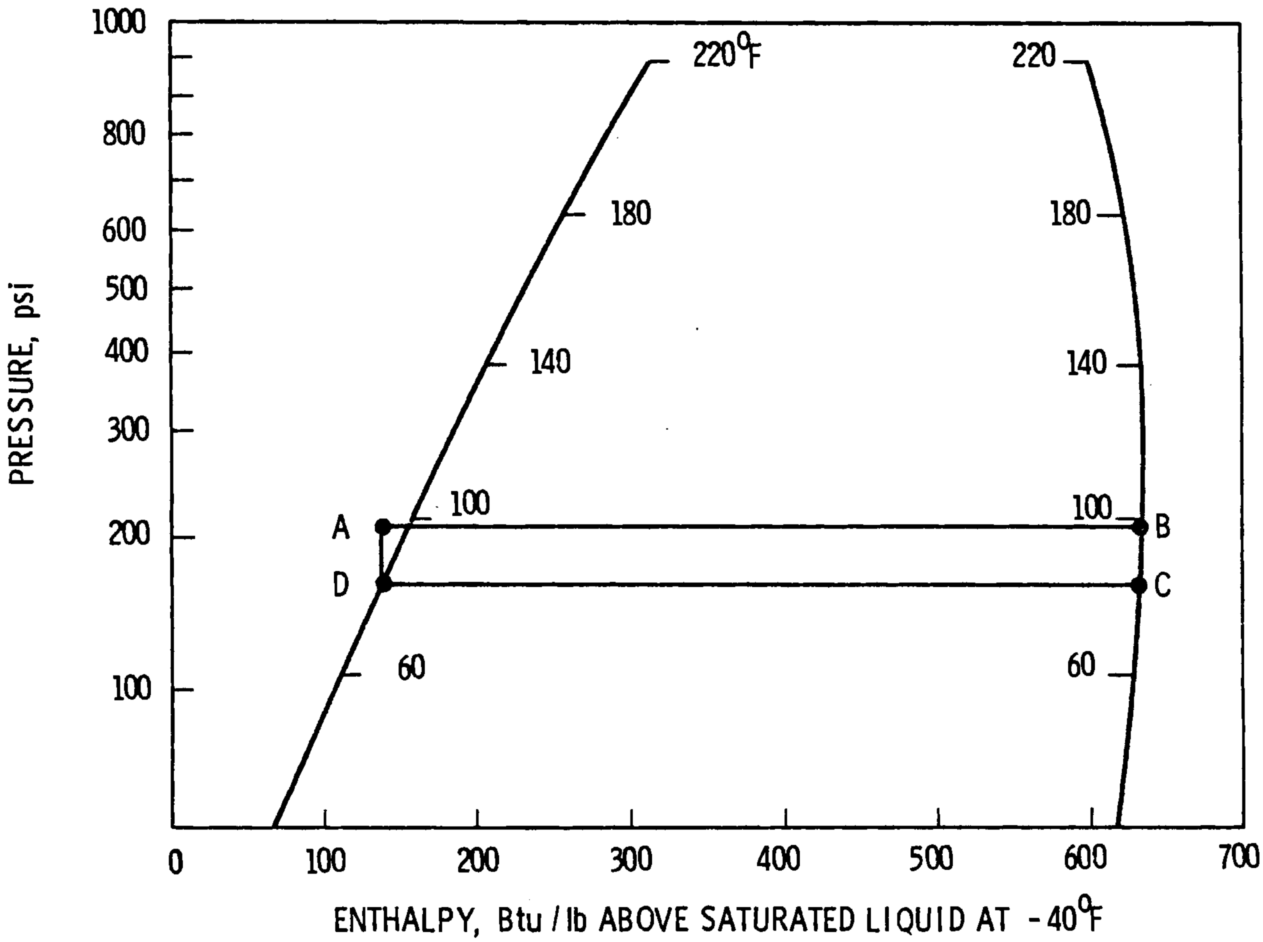


FIGURE A-9. Pressure-Enthalpy Diagram of Ammonia Vapor Dome

The pressure and temperature drops are calculated along the main piping and the quadrant supply runs by dividing the pipes into ten sections and calculating the pressure drop for each section. The temperature drop is calculated for each section of pipe by using the new pressure at the end of each section and a curve fit of the saturation line of pressure versus temperature.

$$P = a + bT + cT^2 \tag{35}$$

The optimization code typically calculates a temperature drop of the vapor from 3 to 5°F.

NOMENCLATURE

g - acceleration of gravity (ft/hr^2)

ρ_L - density of liquid (lbm/ft^3)

ρ_V - density of vapor (lbm/ft^3)

K_L - thermal conductivity of liquid (Btu/hr-°F-ft)

h_{fg} - heat of vaporization (Btu/lbm)

D_o - outside diameter of tube (in)

D_i - inside diameter of tube (in)

μ_L - viscosity of liquid (lbm/ft-hr)

N - number of tubes in depth

ΔT_s - temperature difference between saturated steam and tube surface (°F)

h_i - inside heat transfer coefficient of tube (Btu/hr-°F-ft)

h_{fc} - inside heat transfer coefficient of tube due to forced convection (Btu/hr-°F-ft^2)

h_B - inside heat transfer coefficient of tube due to boiling (Btu/hr-°F-ft^2)

Re_L - liquid Reynolds number

Pr_L - liquid Prandtl number

Cp_L - liquid specific heat (Btu/lbm-°F)

ΔT_a - temperature difference between saturated ammonia and inside tube surface, (°F)

σ - surface tension of liquid (lb/ft)

ΔP - pressure drop (lb/ft^2)

ϕ_L - Martinelli function

X - Lockhart-Martinelli parameter

f_L - liquid friction factor

f_V - vapor friction factor

L - length of pipe (ft)

G_L - mass flux of liquid (lbm/ft^2-hr)

G_V - mass flux of vapor (lbm/ft^2-hr)

θ - angle of tubes from the horizontal

X_o - quality of fluid leaving tubes

X_i - quality of fluid entering tubes

APPENDIX A REFERENCES

1. H. Schlichting, Boundary Layer Theory. McGraw-Hill Book Co., New York, NY, 1968.

2. K. K. Robinson and D. E. Briggs, "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes." Chemical Engineering Progress Symposium Series. 62(64): 1966.

3. D. E. Briggs and E. H. Young, "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes." Chemical Engineering Progress Symposium Series. 59(41): 1963.

4. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces, Part II: Data Analysis and Correlation. PFR Engineering Systems Inc., Marina del Rey, CA, June 15, 1976.

5. C. Weierman, J. Taborek and W. J. Marner, Comparison of the Performance of Inline and Staggered Banks of Tubes with Segmented Fins. Heat Transfer Research, Inc., Alhambra, CA, August 1975.

6. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces, Part I: Heat Transfer and Pressure Drop Data. PFR Engineering Systems, Inc., Marina del Rey, CA, March 1, 1976.

7. P. A. Ard, C. H. Henager, D. R. Pratt and L. E. Wiles, Cost Analysis and Algorithms for Dry Cooling Tower Systems. BNWL-2123, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, November 1976.

8. F. W. Dittus and L. M. K. Boelter, Univ. Calif., Pubs. Eng. 2: 443, 1930.

9. W. Z. Nusselt, "Die Abhangigkeit der Wärmeübergangszahl Von der Rohrlänge." Zeitschr. V. D. I. 54: 1910.

10. A. Zukauskas, "Heat Transfer from Tubes in Crossflow." Academy of Sciences of the Lithuanian SSR, republished in Advances in Heat Transfer, Volume 8. The Academic Press, 1972.

11. W. A. McAdams, Heat Transmission. McGraw-Hill Book Co., New York, NY, 1954.

12. W. Z. Nusselt, Ver. Dent. Ing. 60: 541, 569, 1916.

13. W. M. Rohsenow, "Heat Transfer with Boiling." Modern Developments in Heat Transfer. Pp. 85-158, 1963.

14. Heat Rejection by Dry Cooling in Steam Electric Power Stations. LINDE Division, Union Carbide, New York, NY, April 1976.

15. R. W. Lockhart and R. C. Martinelli, "Proposed Correlations for Isothermal Two-Phase, Two-Component Flow in Pipes." Chem. Eng. Prog. 45: 39, 1949.

16. C. G. Kirkbride, Trans. Am Inst. Chem. Engrs. 30: 170-186, 1933-1934.

17. C. G. Kirkbride, Ind. Eng. Chem. 26: 425-428, 1934.

18. L. D. Boyko and G. N. Kruzhilin, Int. Jour. Heat and Mass Transfer. 10: 361-373, 1967.

APPENDIX B

PIPING SYSTEM DESCRIPTION

APPENDIX B

PIPING SYSTEM DESCRIPTION

Two cooling tower arrangements were considered for dry type cooling systems: circular towers and rectilinear towers. Only circular towers are described here; a computer description of cost algorithms is presented elsewhere.(1) Circular towers are characterized by vertical tube bundles. For a 1000-MWe power station there would typically be several circular towers supplied from one main header. The rectilinear tower could have the tube bundles anywhere from horizontal to vertical, although most designs show the tube bundles to be within a few degrees of horizontal. A single rectilinear tower would be supplied by one main header.

A piping system design and cost model were developed for a cooling system using water, and for one using ammonia, as the primary coolant. A one-pass vertical heat exchanger was chosen for the ammonia system. With vapor entering at the top of the tube bundle it facilitates condensate collection. Thus, the ammonia lends itself to the circular tower arrangement. To compare the ammonia and water systems on an equivalent basis, emphasis was also placed on the analysis of circular towers for the water systems. In addition, a piping system model was developed for rectilinear towers using water.

Significant leakage of ammonia or water, or possible freezing of the water would necessitate draining the coolant from the heat exchanger and storing it until the problems were corrected. Such a drain and refill system has not been included in the piping model for the following reasons:

- Cost data on some of the required components was thought to be inadequate, particularly with regard to the storage tanks.
- The warmer meteorological sites such as Phoenix, Miami and San Francisco would not require freeze protection of the water system.
- Because there was little operating history on this subject, it was not clear what capacity would be required at any particular site.

A drain and refill system with capacity to store the entire volume of coolant could add as much as 20 percent to the cost of the piping system. If the attempt were to optimize the cooling system for a particular site that required a drain and refill system of known capacity, then the costs should be included because they would influence the design. However, such site-specific issues have not been included in these algorithms, and the drain and refill system was excluded from all the designs.

CIRCULAR TOWERS

In developing the piping system model for circular towers, four separate subsystems were identified. These were termed the pump station, quadrant piping, tower distribution piping and main circulation piping. They are identified in Figure B-1, drawn for a four-tower arrangement. The piping layout shown is representative of the configuration modeled in the piping system design and cost algorithm. This figure represents either the supply piping from the condenser to the heat exchangers or the return piping from the heat exchangers to the condenser. Only one is shown because, for our purposes, it was assumed that the layout of the supply and return piping could be essentially identical. Some adjustments are necessary, however, because the pump station is assumed to be included in the return piping. Also, for ammonia systems the supply coolant is a vapor and the return coolant is a liquid. The different densities and allowable design velocities result in different pipe sizes for the supply and return piping of the ammonia systems. A slightly different quadrant piping arrangement is required for the ammonia. This is discussed below.

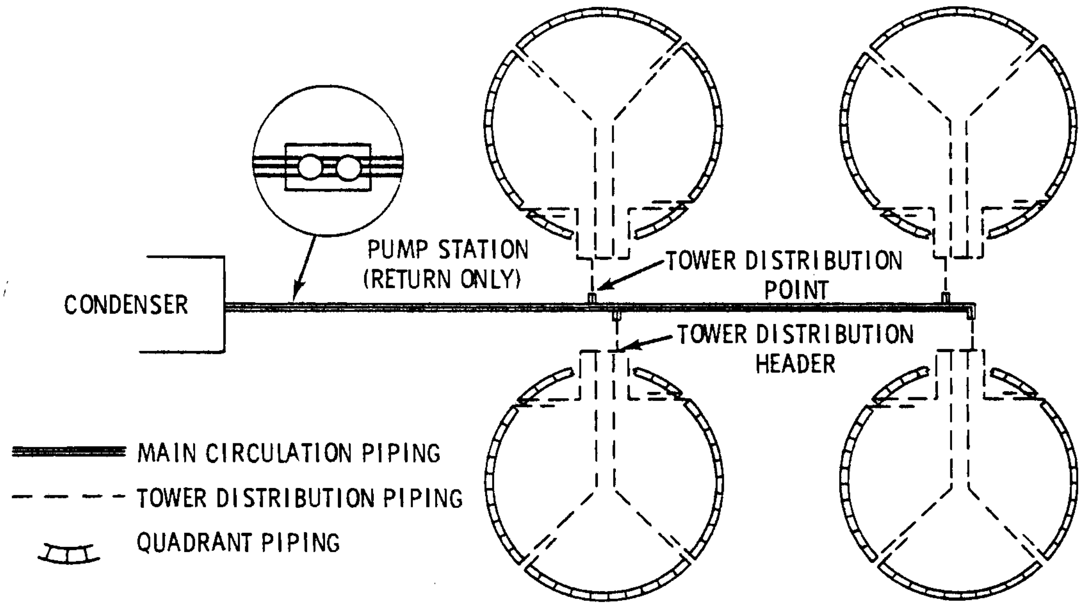


FIGURE B-1. Piping Subsystems for Circular Towers

PUMP STATION

The pump station for the water system is in the return line of the main circulation piping. It is assumed to be 50 feet in length from inlet to outlet. Two vertical inline pumps provide the capacity of the main circulation line. At the outlet of each pump is a motor-operated valve and an expansion joint. Other appropriate pipe and fittings are included in the pump station costs. No auxiliary pumping capacity is provided.

QUADRANT PIPING

The flow to each tower is distributed to four quadrants regardless of tower size. The supply and return quadrant interface with the tower distribution piping at opposite ends of the quadrant. Figure B-2 shows the quadrant headering for the water system, modeled with a two-pass heat exchanger. This enables the supply and return quadrant headers to be located at ground level. The ammonia system is modeled with a one-pass heat exchanger. The supply vapor is piped to the top of the quadrant where it is distributed to the heat exchanger bundles. The ammonia condensate is collected in a return header similar to the return header of a water system.

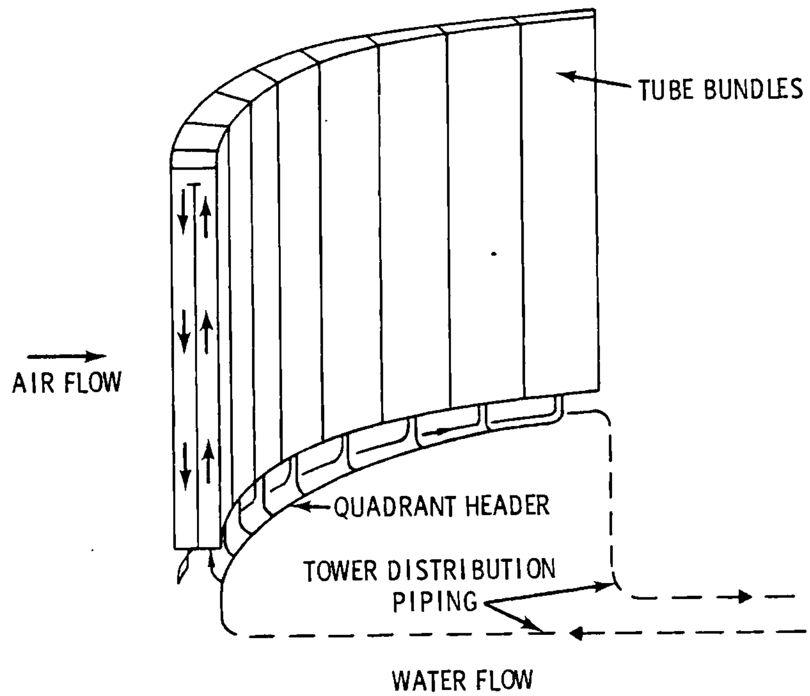


FIGURE B-2. Quadrant Headering for Water System

The quadrant header includes pipe, reducers and a tee at each tube bundle. The pipe and flanges connecting the header tees to the bundles are added to the quadrant headering cost.

TOWER DISTRIBUTION PIPING

The tower distribution piping is composed of the pipe and fittings from the tower distribution point to the quadrant headers. The flow is routed to each quadrant through the tower distribution header. Cooling system control is maintained by motor-operated valves located in the supply and return lines to each quadrant.

MAIN CIRCULATION PIPING

The main circulation piping includes the pipe and fittings from the condenser outlet flange to the distribution point for each circular tower. This piping is assumed to be entirely underground. The interface between the condenser and the piping system is a flanged expansion joint. The straight run of pipe from the condenser also includes the tees at each tower distribution point and appropriately sized reducers. The distribution point of the last tower is equipped with an elbow.

REFERENCE

1. P. A. Ard et al., Cost Analysis and Algorithms for Dry Cooling Tower Systems. BNWL-2123, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, September 1976.

APPENDIX C

OPTIMIZATION TECHNIQUE

APPENDIX C

OPTIMIZATION TECHNIQUE

As mentioned in Section 3.0, an optimization technique developed by Andeen and Glicksman[1] was used to determine the values of the five internally independent variables which give minimum incremental cost. Only minor changes were made to the subroutines incorporating the MIT optimization technique; these changes, described in Section 5.1, are related to system constraints, not to the optimization procedure. The basic optimization technique was not changed. However, major changes were made in the computational logic, cost algorithms, etc.

Approval was obtained from Dr. Glicksman to publish that portion of Reference 1 dealing with the optimization routine used in the BNW-I codes. The prose and flow charts of the subroutines are reproduced in this appendix.

VERBATIM DESCRIPTION OF MIT OPTIMIZATION TECHNIQUE

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

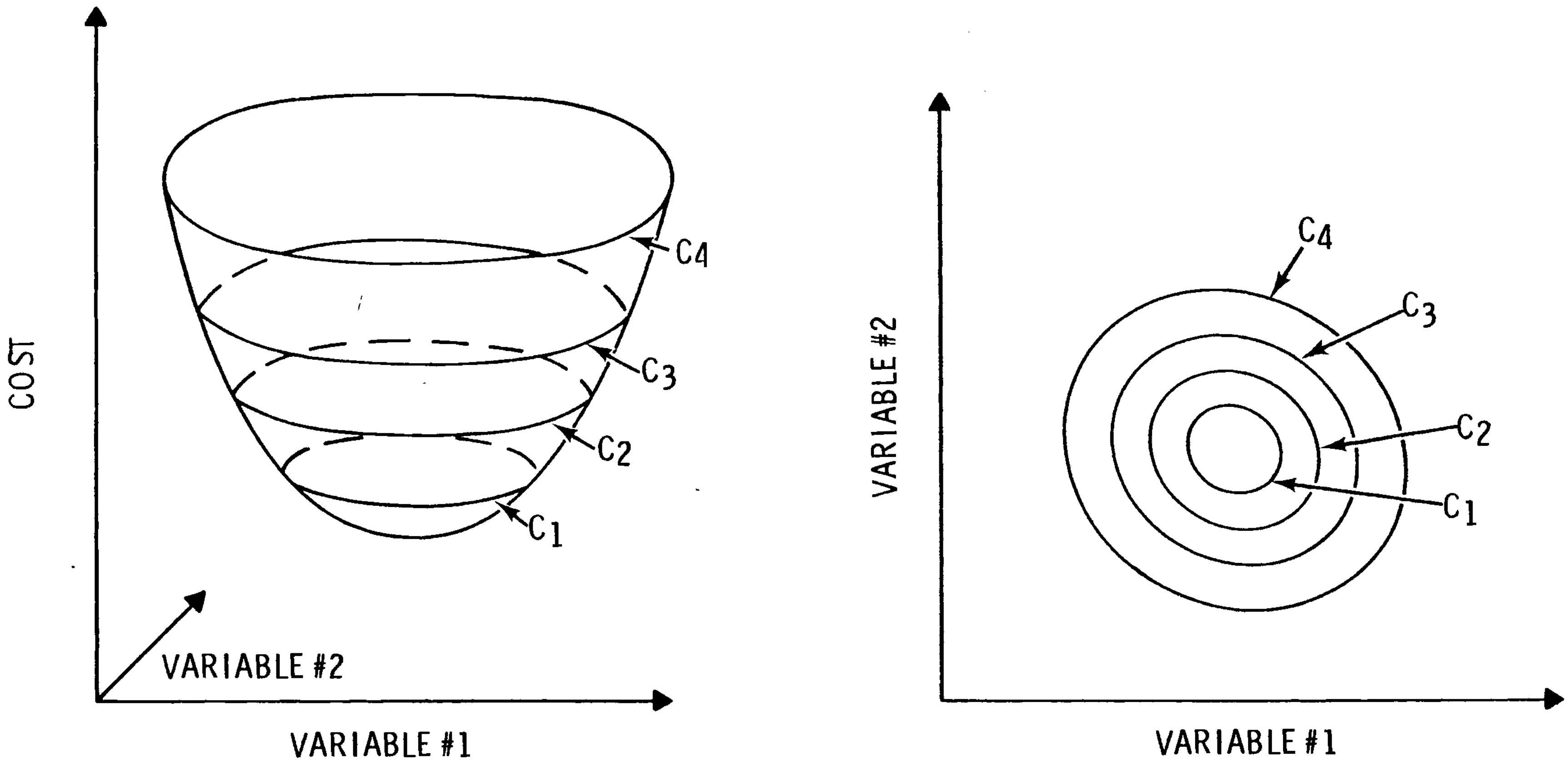
T_1	steam cycle output temperature (°F) at the low pressure turbine
RANGE	change in temperature (°F) of the water while passing through the cooler
CWARA	ratio of the heat capacity of water to that of air, where heat capacity of a fluid is the product of its specific heat and its mass flow rate
AFRON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

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

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

[a] PNL NOTE: TD is design temperature

C's - LINES OF CONSTANT COST - $C_1 < C_2 < C_3 < C_4$



Three-Dimensional Visualization
Figure IV-1A

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

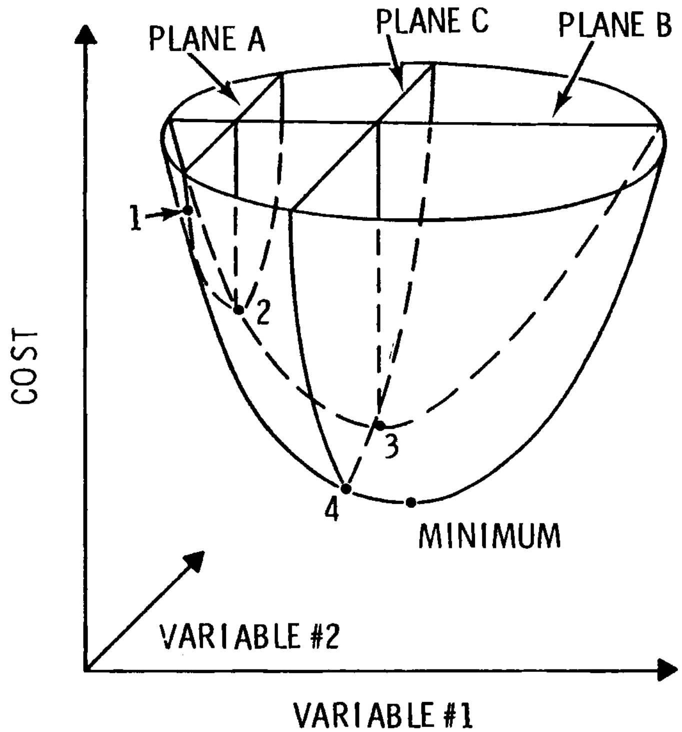
<u>FIGURE IV-1</u>. Cost as a Function of Two Variables

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

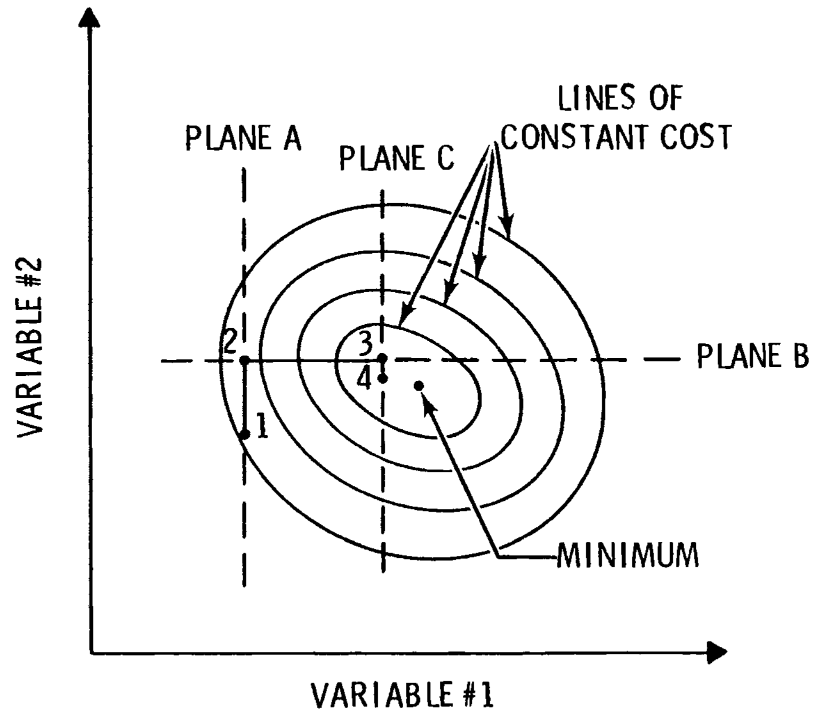
However, if the surface is not as well-behaved as that drawn in Figure IV-2, but has multiple minimal points as in Figure IV-3 the search procedure may end by "minimizing" in a relative minimum trough. To avoid such relative minima, two approaches have been taken: (1) the cost is evaluated at multiple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

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

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

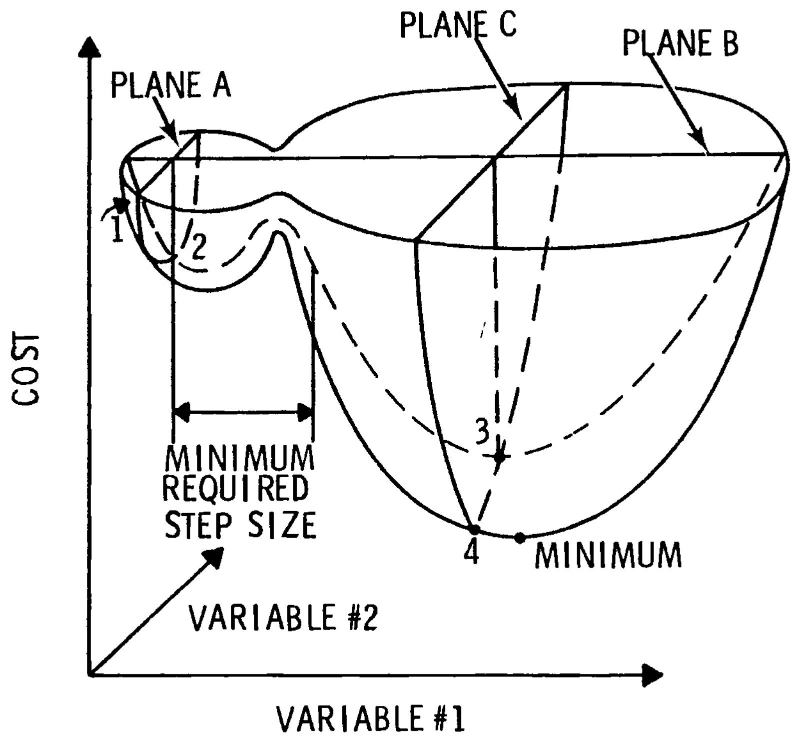


Three-Dimensional Visualization
Figure IV-2A



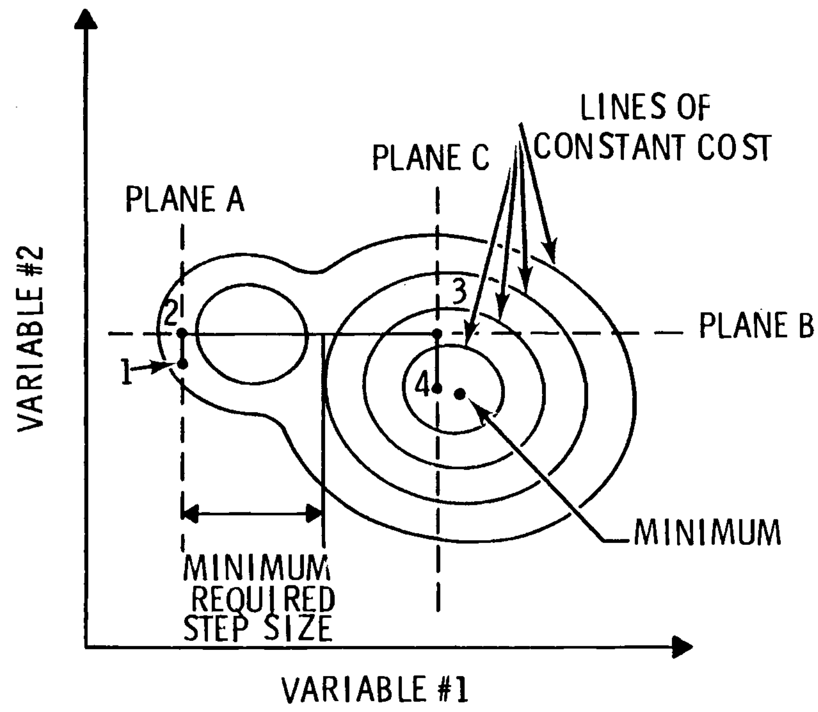
Two-Dimensional Visualization
Figure IV-2B

FIGURE IV-2. Minimization Technique on Simple Surface



Three-Dimensional Visualization

Figure IV-3A



Two-Dimensional Visualization

Figure IV-3B

FIGURE IV-3. Minimization Technique on Simple Surface

Initially the air velocities (VAIR) through the exchanger were maintained above a specified minimum. It was hypothesized that a high VAIR would minimize the effects of the ambient air velocity. However, specifying a minimum VAIR penalizes heat exchanger configurations having an inherently high air side friction. Specifying a minimum VAIR is the same as specifying a very high air side pressure drop for configurations with high friction factors. This in turn requires a much larger fan power to create this air side pressure drop. A more logical requirement is a minimum air side pressure drop, DELPA, through the heat exchanger. Since the driving force for the air flow is the pressure differential, as long as the maintained pressure differential is greater than an external differential caused by wind, flows through the exchanger will not be greatly affected. Note that this allows the "high friction" configurations to adopt lower values of VAIR, so long as DELPA is above a minimum, thereby saving on fan power.

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

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

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

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

So long as this function, or DELPA is maintained constant, we are at liberty to modify the variables which compose the function. Thus, if RANGE and AFRON are variables #1 and #2 respectively in Figure IV-4 an increase in range accompanied by a decrease in AFRON will move down line 2B towards B, while keeping DELPA fixed.

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

---

[a] PNL chose minimum pressure drop values well below the economically optimum values (in effect, no pressure drop constraint was applied).

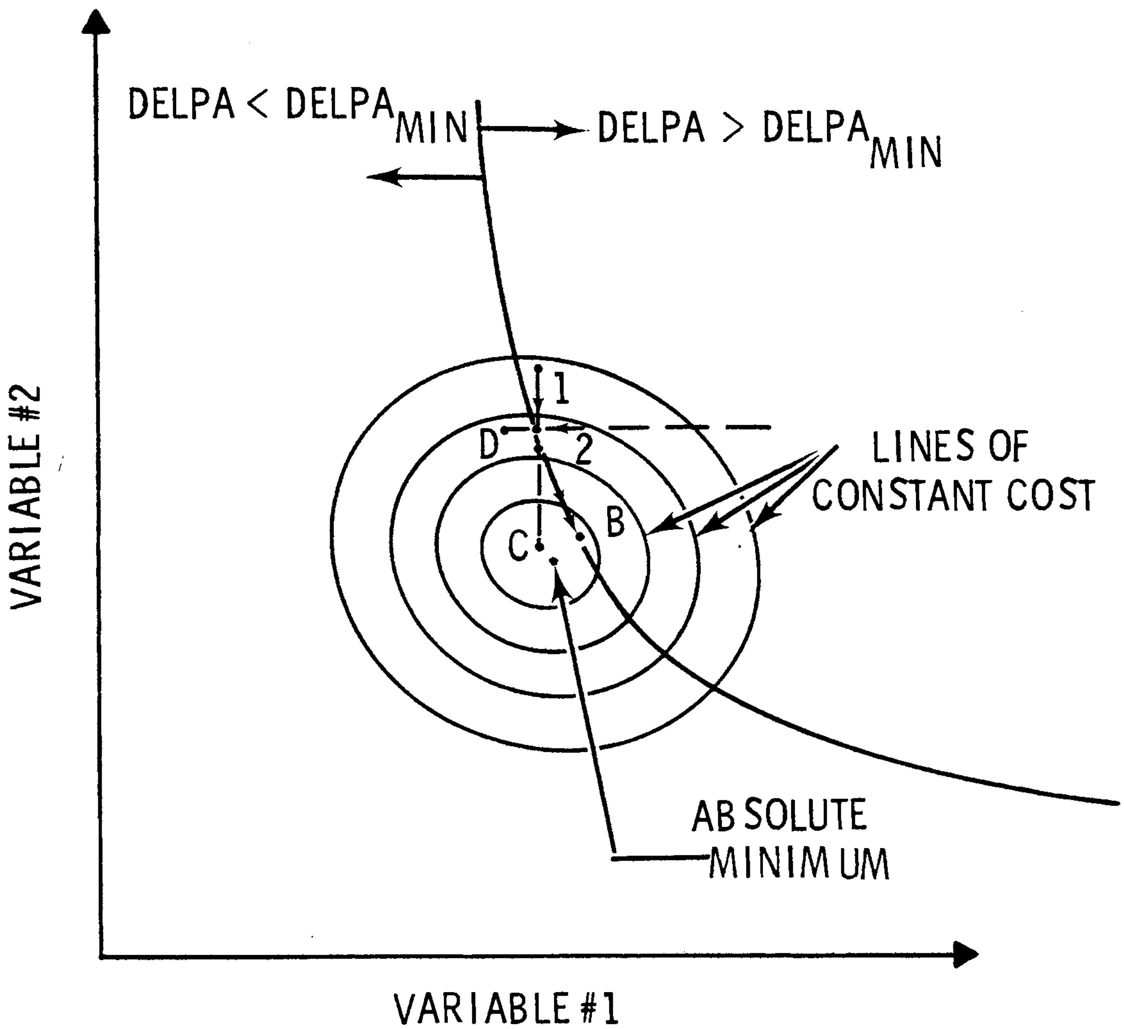


FIGURE IV-4. Minimization with Imposed Restrictions

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

This double shotgun method is pictured in Figure IV-5 for a two-variable case. Point 1 is the initial prescribed starting point. The first shotgun search investigates those points designated as circles, and yields point 2 as the minimum cost point. Extreme values, the diamonds, are then checked for lower cost points. Finding none in this case, a "finer" shotgun pattern is searched about point 2, yielding 3 as a lower cost point. Point 3 then becomes the starting point for the search procedure as described in Figure IV-2.

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

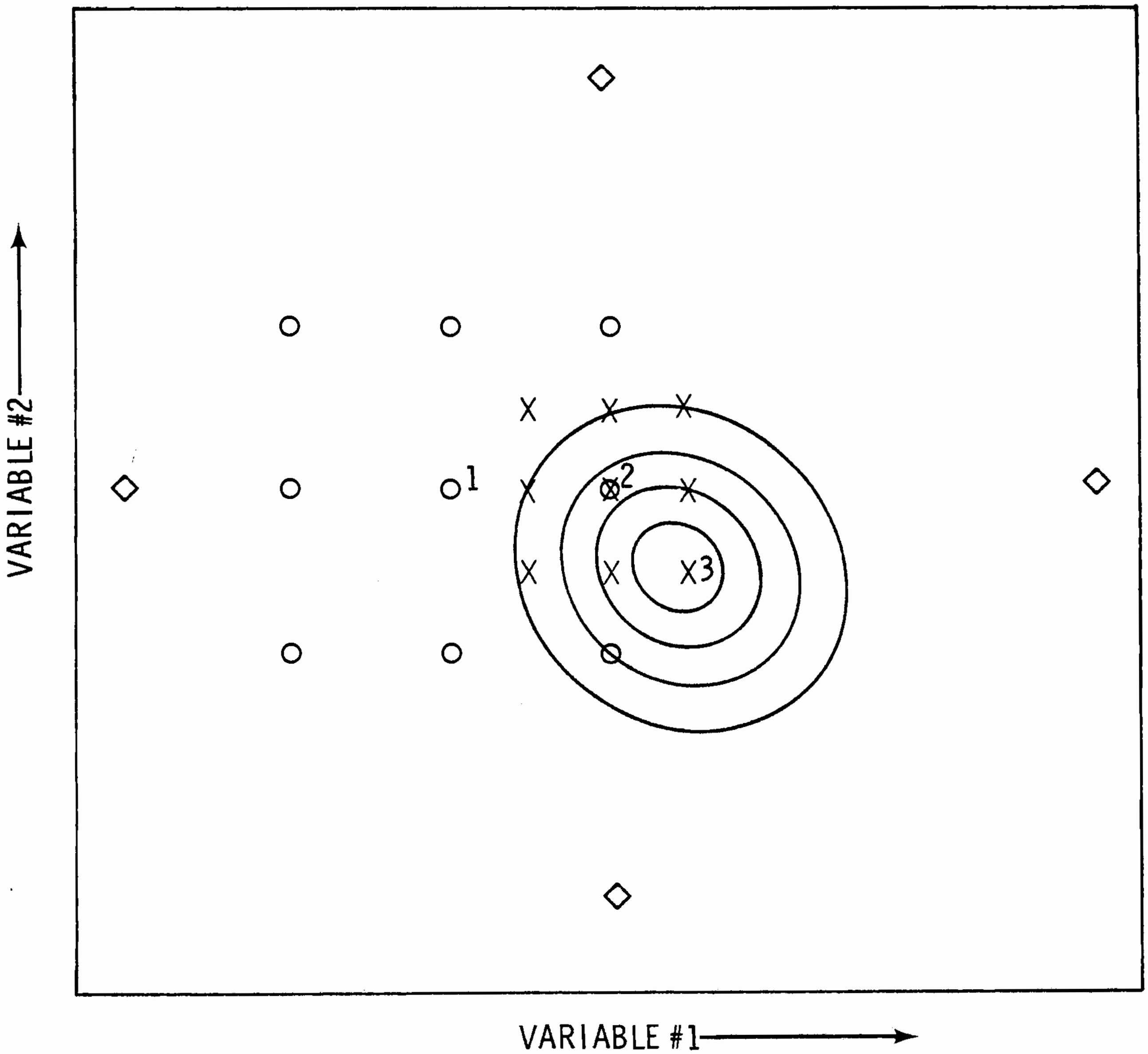


FIGURE IV-5. Double Shotgun Initializing Mehtod

MIT SUBROUTINES

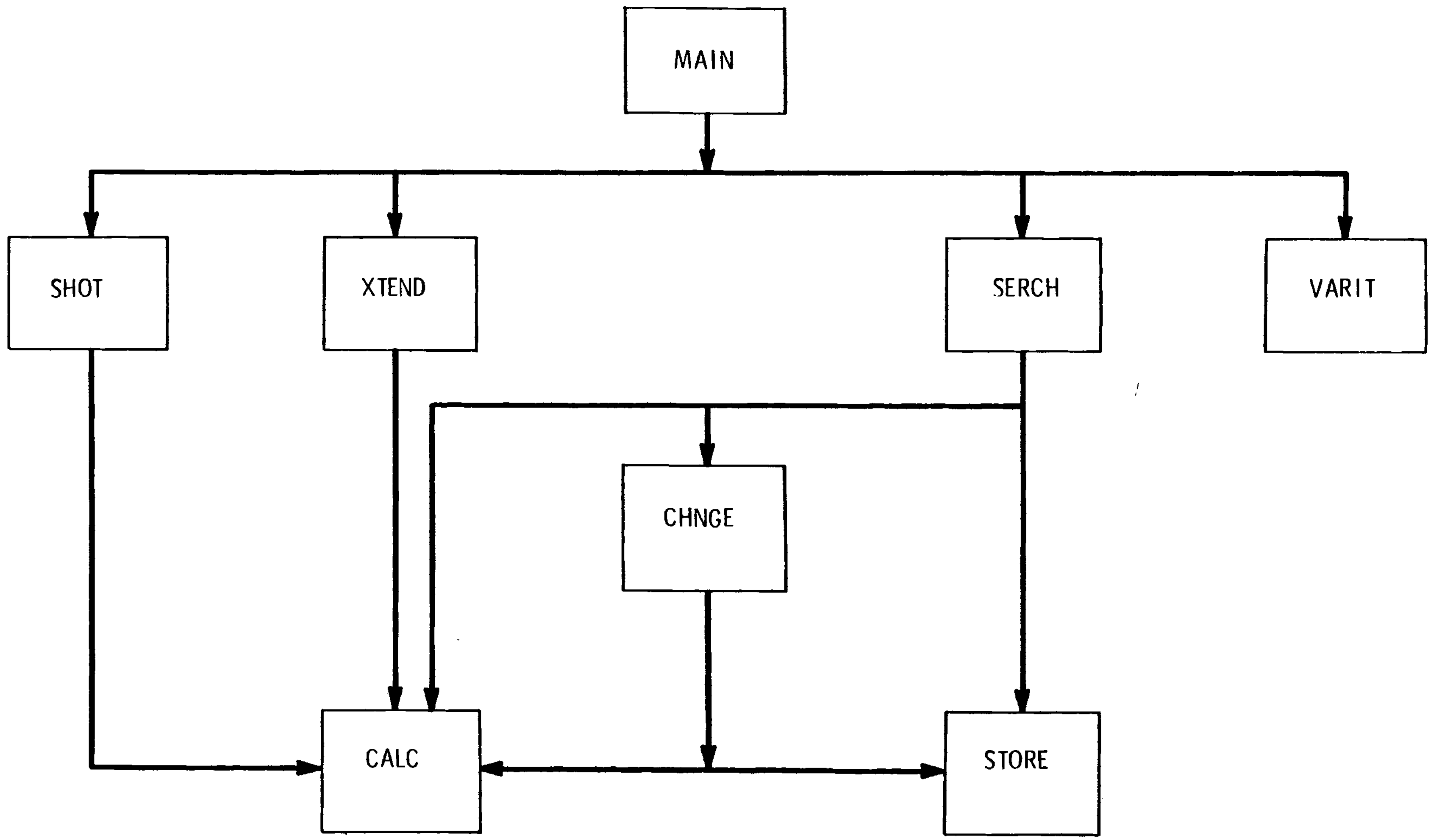
FUNCTION OF SUBROUTINES

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

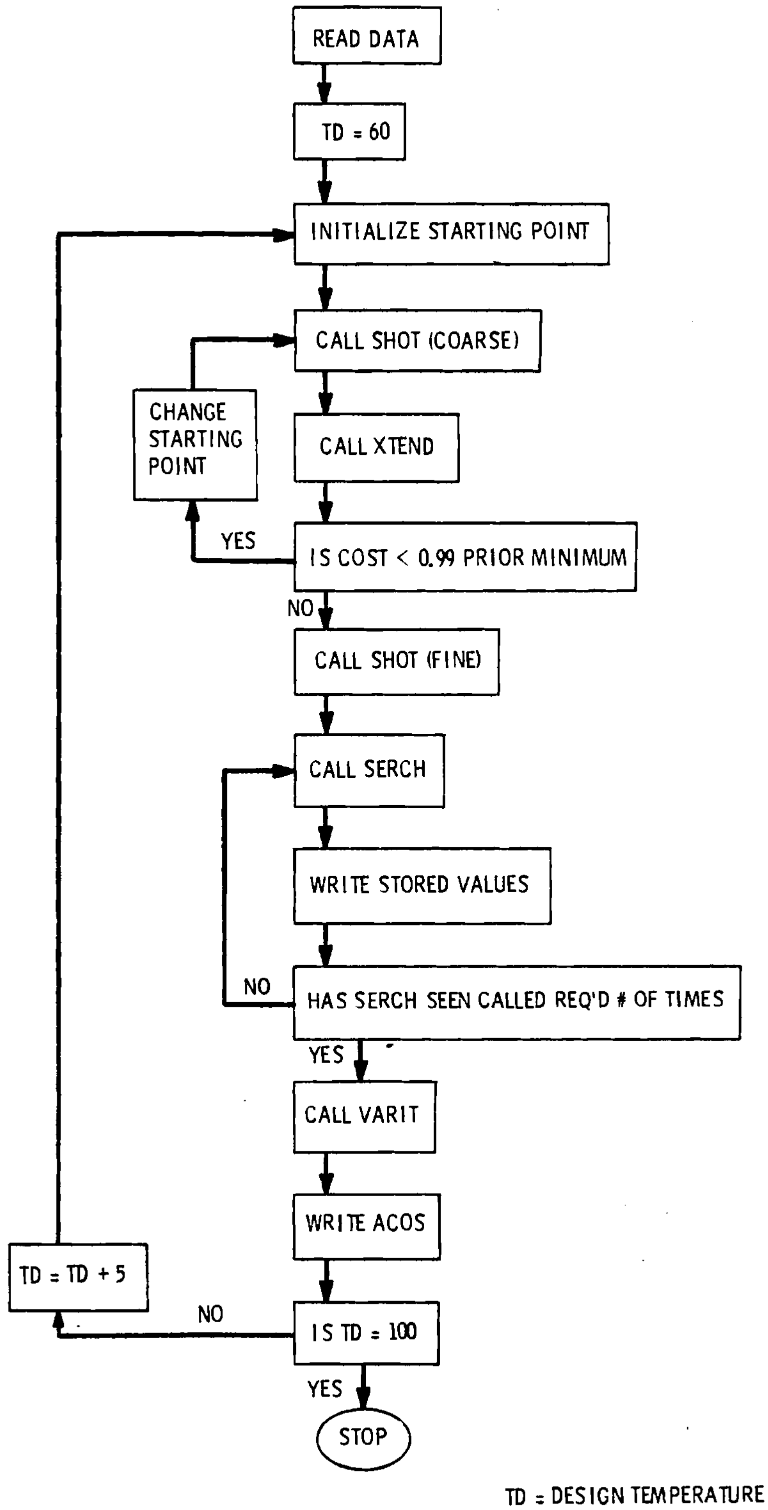
REFERENCE

1. B.R. Andeen and L. R. Glicksman, Dry Cooling Towers for Cooling Plants. Technical Report 73047-1, Engineering Project Laboratory, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA, February 1972.

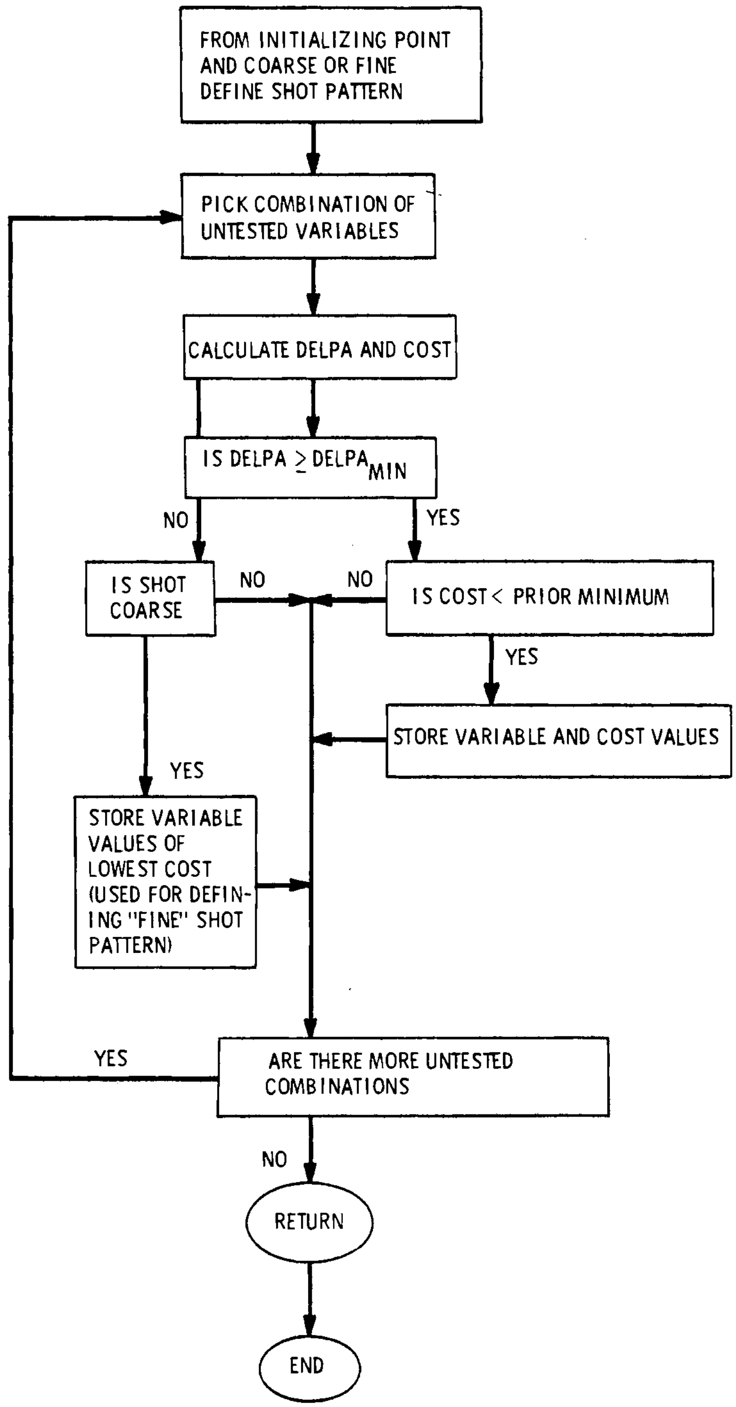
FLOW DIAGRAM OF SUBROUTINE CALLING



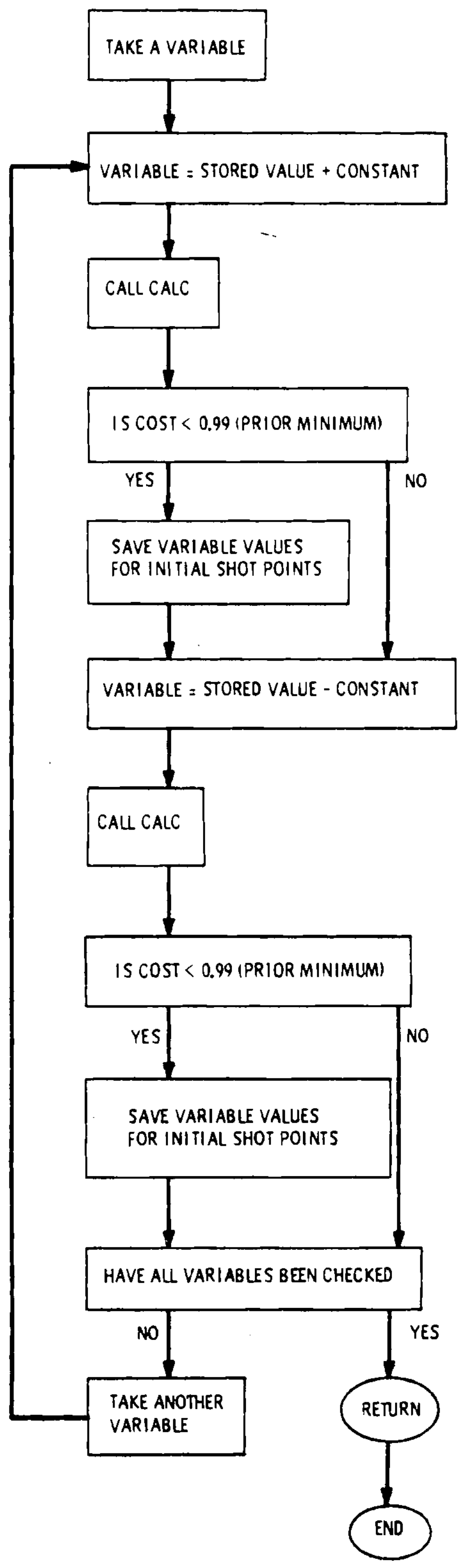
MAIN PROGRAM



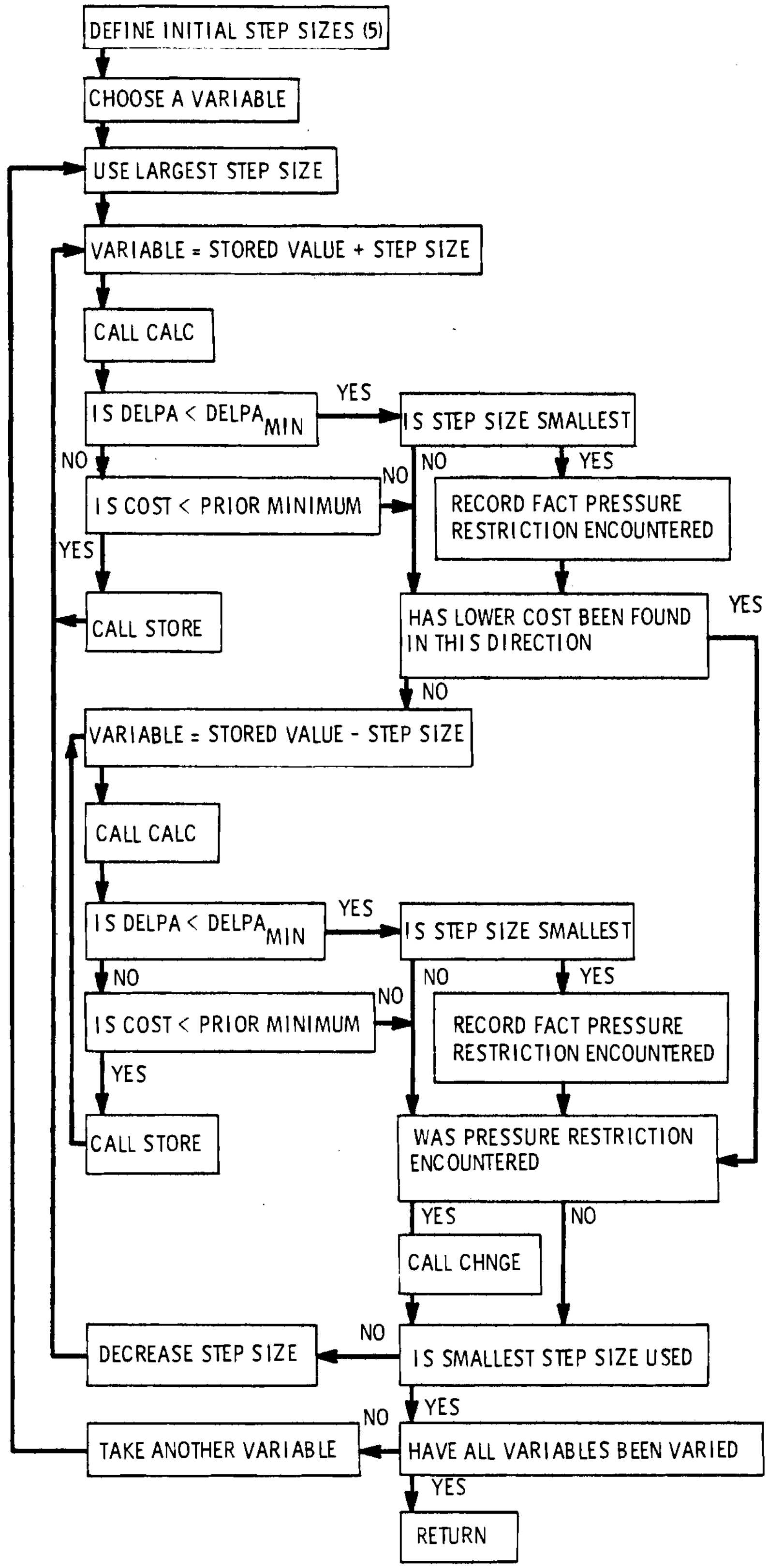
SHOT


FROM INITIALIZING POINT AND COARSE OR FINE DEFINE SHOT PATTERN
PICK COMBINATION OF UNTESTED VARIABLES
CALCULATE DELPA AND COST
IS DELPA ≥ DELPA$_{MIN}$
NO
YES
IS SHOT COARSE
NO
NO
IS COST < PRIOR MINIMUM
YES
STORE VARIABLE AND COST VALUES
YES
STORE VARIABLE VALUES OF LOWEST COST (USED FOR DEFINING "FINE" SHOT PATTERN)
YES
ARE THERE MORE UNTESTED COMBINATIONS
NO
RETURN
END

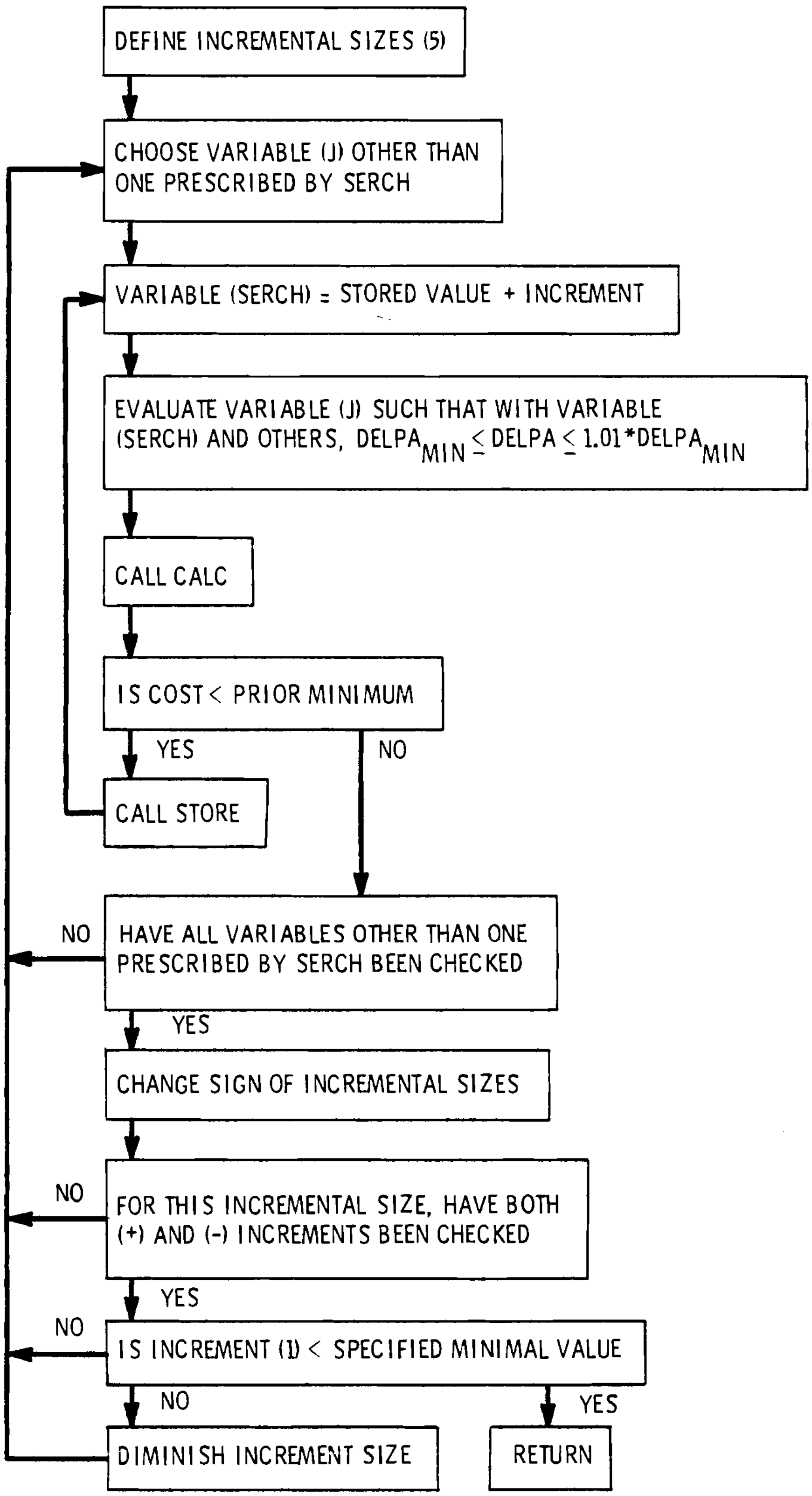
XTEND


TAKE A VARIABLE
VARIABLE = STORED VALUE + CONSTANT
CALL CALC
IS COST < 0.99 (PRIOR MINIMUM)
YES
NO
SAVE VARIABLE VALUES FOR INITIAL SHOT POINTS
VARIABLE = STORED VALUE - CONSTANT
CALL CALC
IS COST < 0.99 (PRIOR MINIMUM)
YES
NO
SAVE VARIABLE VALUES FOR INITIAL SHOT POINTS
HAVE ALL VARIABLES BEEN CHECKED
NO
YES
TAKE ANOTHER VARIABLE
RETURN
END

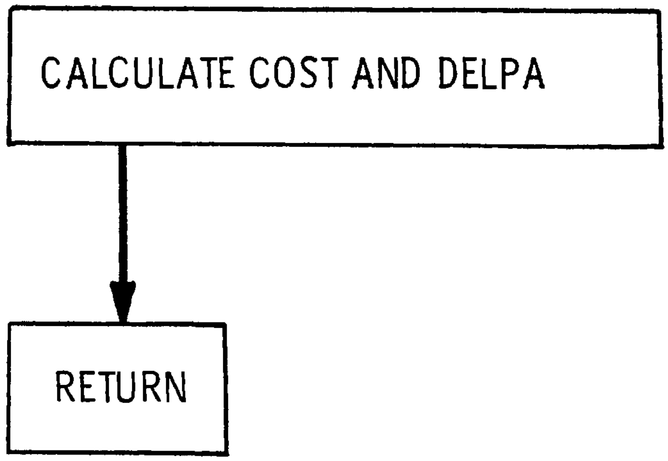
SERCH


DEFINE INITIAL STEP SIZES (5)
CHOOSE A VARIABLE
USE LARGEST STEP SIZE
VARIABLE = STORED VALUE + STEP SIZE
CALL CALC
IS DELPA < DELPA$_{MIN}$
YES
IS STEP SIZE SMALLEST
NO
NO
YES
IS COST < PRIOR MINIMUM
NO
RECORD FACT PRESSURE RESTRICTION ENCOUNTERED
YES
CALL STORE
HAS LOWER COST BEEN FOUND IN THIS DIRECTION
YES
NO
VARIABLE = STORED VALUE - STEP SIZE
CALL CALC
IS DELPA < DELPA$_{MIN}$
YES
IS STEP SIZE SMALLEST
NO
NO
YES
IS COST < PRIOR MINIMUM
NO
RECORD FACT PRESSURE RESTRICTION ENCOUNTERED
YES
CALL STORE
WAS PRESSURE RESTRICTION ENCOUNTERED
YES
NO
CALL CHNGE
DECREASE STEP SIZE
NO
IS SMALLEST STEP SIZE USED
YES
TAKE ANOTHER VARIABLE
NO
HAVE ALL VARIABLES BEEN VARIED
YES
RETURN

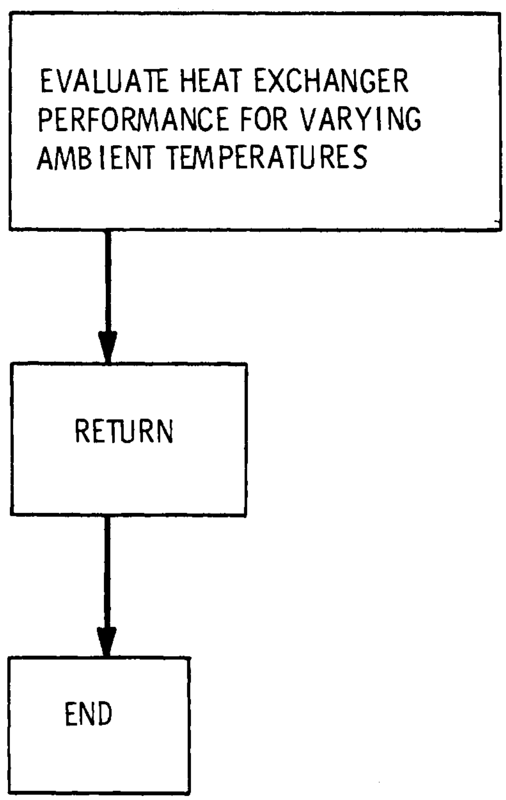
CHNGE


DEFINE INCREMENTAL SIZES (5)
CHOOSE VARIABLE (J) OTHER THAN ONE PRESCRIBED BY SERCH
VARIABLE (SERCH) = STORED VALUE + INCREMENT
EVALUATE VARIABLE (J) SUCH THAT WITH VARIABLE (SERCH) AND OTHERS, DELPA$_{MIN}$ ≤ DELPA ≤ 1.01*DELPA$_{MIN}$
CALL CALC
IS COST < PRIOR MINIMUM
YES
NO
CALL STORE
NO
HAVE ALL VARIABLES OTHER THAN ONE PRESCRIBED BY SERCH BEEN CHECKED
YES
CHANGE SIGN OF INCREMENTAL SIZES
NO
FOR THIS INCREMENTAL SIZE, HAVE BOTH (+) AND (-) INCREMENTS BEEN CHECKED
YES
NO
IS INCREMENT (1) < SPECIFIED MINIMAL VALUE
NO
YES
DIMINISH INCREMENT SIZE
RETURN

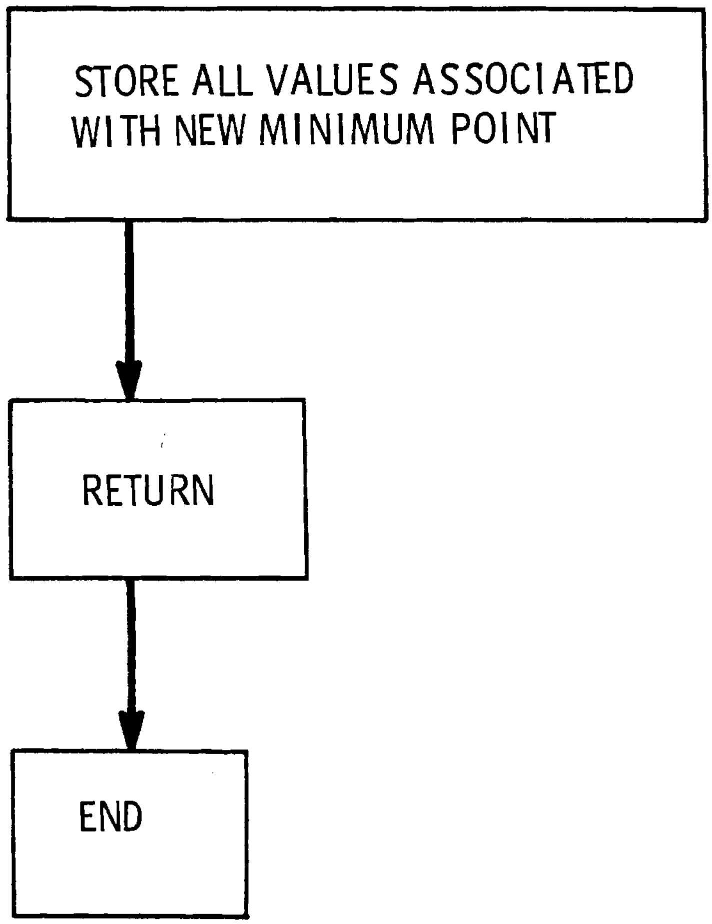
CALC


CALCULATE COST AND DELPA
RETURN

VARIT


EVALUATE HEAT EXCHANGER
PERFORMANCE FOR VARYING
AMBIENT TEMPERATURES
RETURN
END

STORE


STORE ALL VALUES ASSOCIATED
WITH NEW MINIMUM POINT
RETURN
END

APPENDIX D

SELECTION OF COMPLETE AND INCOMPLETE TOWER GROUPS

APPENDIX D

SELECTION OF COMPLETE AND INCOMPLETE TOWER GROUPS

A tower group is composed of a number of circular towers that are supplied by a main circulation header.

The number of circular towers per tower group is determined by consideration of the total coolant flow rate, the total number of circular towers, the largest pipe available for the main circulation header, and the design velocity for the coolant. The design velocity determines the maximum flow that can be transported through the largest pipe. This flow is compared against the flow to each circular tower to determine the largest integer number of towers that can be supplied through a common header. This is the number of towers per tower group.

Should the total number of towers be less than the number of towers per tower group, the tower group is considered to have the lower number of towers. If the total number of towers is an integer multiple of the calculated number of towers per tower group, there is that number of complete and identical tower groups. If the total number of towers otherwise exceeds the number of towers per tower group, a so-called "extra tower group" or "incomplete tower group" is created. For example, suppose there are seven towers with five towers forming a complete group, the two extra towers would form the incomplete group. Suppose there are thirteen towers with five towers in a complete group. There would be two complete groups with three towers in the incomplete group. The difference between a complete and incomplete tower group is in the main circulation header and pump station. The pump station of the incomplete group would have less total flow rate and its size and cost would be adjusted accordingly. The header pipe sizes for the incomplete group will be smaller although the header lengths will be identical. However, the header pipe diameters for the incomplete group will correspond to the diameters of the end headers of the complete group. For example, if there are five towers per group and two towers in an incomplete group, then the pipe diameter of the first and second header sections of the incomplete group will be the same as the pipe diameters of the fourth and fifth header sections of the complete group.

The distance between the condenser and the boundary of the tower group is the same for both the complete and incomplete group.

It has been assumed that the "incomplete group" problem does not apply to the ammonia system. It was believed that for any reasonable design the total flow could be transported through the largest available pipe. Thus, there would be no need to provide for the extra towers. However, for plant sizes larger than 1000 MWe or for below optimum design velocities of the coolant this may not be true. Problems could result that have not been provided for in the code logic.

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